THE BOILER MAKER

VOLUME XXXII

JANUARY TO DECEMBER, 1932

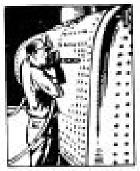
PUBLISHED BY
SIMMONS-BOARDMAN PUBLISHING COMPANY
30 Church Street, New York, N. Y., U. S. A.

INDEX

Note—Illustrated articles are marked with an (*) asterisk; Book reviews (B. R.); Communications (C); Editorials (E); Paragraphs (P); Questions and Answers (Q).

The Boiler Maker

Reg. U. S. Pat. Off.



Boiler Failures

From the Bureau of Locomotive Inspection chief inspector's report it is evident that, in spite of the fact there were about 10 percent fewer accidents in 1931 than for the previous year, caused by the failure of locomotive boilers, the number of persons killed and injured from this cause increased. Boiler explosions accounted for 93.7 percent of the fatalities from all locomotive accidents, as compared with 84.6 for the preceding year. As in former years also, crown sheet failures from low water continued to be the source of most of the boiler accidents.

While the increase of fatalities in the current record is to be deplored, it is accounted for largely by the increased size of locomotive boilers and the higher pressures carried. It is only necessary to compare the present statistics of failures with those of 1912, when the present law went into effect, and that of intervening years to note the tremendous improvement in this respect. In 1912 there were 94 low-water failures as compared with 14 in 1931, a reduction of 85.1 percent. From the standpoint of fatalities, the same condition is true, 54 persons were killed in 1912 as compared with 15 in 1931; and 168 persons were injured as against 41 for the present fiscal year.

The potential danger from crown sheet failures cannot be minimized, however, because of improvements in the record. Suggestions of the chief inspector, that the safest practicable firebox construction must be adopted, and reliable boiler feeding and water level devices utilized, should be observed. Then if the engine crews will maintain the proper water level on such boilers when in service, an entirely clear record in this direction is possible.

Help for the Railroads

The constructive action now being taken in favor of the railroads by the Government and by the executives of the industry should within a comparatively short time be reflected in increased activity in the shops. It is certain that much of the equipment and rolling stock of the railroads of the country require repair and replacement. In fact from 1929 on, maintenance needs have not had the attention necessary in normal times. While this condition might exist for some time when the transportation demands of industry are at a low level, the point has undoubtedly been reached beyond which it is hardly safe to continue if the railroads are ever economically to rehabilitate their facilities.

This state of physical equipment has been a factor in the consideration of relief measures, and it is certain to receive early attention when funds become available from new rates now in effect and credits which are expected to be established shortly.

Since the railroads constitute a fundamental basis of industrial life and activity in the United States, there can be no question but that the present adverse conditions, threatening their very existence, will be corrected. With improvement in this direction, industry in general will commence its recovery, as it has done many times in past periods of depression.

New Markets for Manufacturers

Boiler manufacturers who, at this time, will take inventory of their equipment and the ability of their men, will undoubtedly find that they could manufacture other marketable products besides boilers. With the development of stainless and other alloy steels as well as the many new non-ferrous alloys which, because of their non-corrosive and acid-resisting qualities, are desirable in the chemical and allied fields, boiler manufacturers in many cases are well equipped to fabricate products for various industries. The fact that a number of the more progressive boiler manufacturers have reorganized their plants to build pressure vessels for oil refineries, as well as boilers, indicates that new markets for the products of the boiler shop are available.

In the oil industry high-pressure equipment such as cracking coils, stills and condensers is in demand; while in the chemical field, special alloy pressure vessels are required for various processes. Again, the manufacture of food products in many cases requires large double-jacketed steam kettles of stainless steel or special alloy construction. Pasteurizing plants found in the dairy industry, as well as equipment used in the making of candy are among the applications of pressure vessels, tanks, dished plate and welded plate products. The fabrication of such products would require the use of flanging and dishing presses, spinning machines, plate rolls and welding equipment, facilities to be found in every modern boiler manufacturing plant.

Present business conditions will not continue forever, but there is no certainty that, when general business revives, future markets will be the same as those of the past. In preparing for the future, the wise executive will either find new markets for his products or new products for his plant. He must analyze his facilities and determine the direction in which his activities will give him the greatest measure of security in the industrial age of the future.

Fig. 1.—Result of crown sheet failure caused by low water

Annual Report of Bureau of Locomotive Inspection

A. G. Pack, chief inspector of the Bureau of Locomotive Inspection, in the twentieth annual report to the Interstate Commerce Commission, states that during the fiscal year ended June 30, 1931, there were fewer locomotive accidents than at any time during the past 20 years and that this reduction has been consistently made in about the same ratio as the condition of locomotives has been improved. As in former years, boiler explosions continue to be the most prolific source of accidents. During the fiscal year ended June 30, 1931, boiler explosions caused 15 fatalities and 41 injuries. Only 16 fatal accidents resulted from failures of steam locomotives and tenders and their appurtenances, the one death, not accounted for from boiler explosions, being caused by a defective coupler.

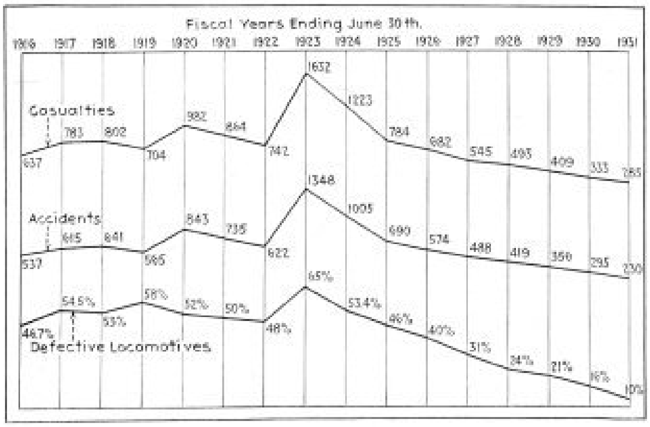


Fig. 2.—Relation of defective locomotives to accidents and casualties resulting from locomotive failures

The fact that the railroads were able to continue to improve their record with respect to defective locomotives, handicapped as they have been by curtailed expenditures for maintenance, speaks well for the effective work being done by the mechanical departments in meeting the emergencies created by the depression and also for the standards of maintenance established in past years. Following is an abstract of Mr. Pack's report:

During the year 10 percent of the steam locomotives inspected by our inspectors were found with defects or errors in inspection that should have been corrected before being put into use as compared with 16 percent for the previous year. This reflects the best condition of locomotives in service ever recorded. A summary of all accidents and casualties to persons occurring in connection with steam locomotives compared with the previous year shows a decrease of 22 percent in the number of accidents, an increase of 23 percent in the number of persons killed, and a decrease of 15.9 percent in the number of persons injured. The increase in the number of persons killed was due to one particularly violent boiler explosion in which six persons were killed. This explosion was due to lapse of ordinary caution on the

Table 1—Condition of Locomotives, Found by Inspection, in Relation to Accidents and Casualties

Fiscal year ended June 30	Percent of locomotives inspected found defective	Number of locomotives ordered out of service	Number of accidents	Number of persons killed	Number of persons injured
1923	65	7,075	1,348	72	1,560
1924	53	5,764	1,005	66	1,157
1925	46	3,637	690	20	764
1926	40	3,281	574	22	660
1927	31	2,539	488	28	517
1928	24	1,725	419	30	463
1929	21	1,490	356	19	390
1930	16	1,200	295	13	320
1931	10	688	230	16	269

Table 2—Reports and Inspections—Steam Locomotives

	Year ended June 30—					
	1931	1930	1929	1928	1927	1926
Number of locomotives for which reports were filed	60,841	61,947	63,562	65,940	67,835	69,173
Number inspected	101,224	100,794	96,465	100,415	97,227	90,475
Number found defective	10,277	16,300	20,185	24,051	29,995	36,354
Percentage inspected found defective	10	16	21	24	31	40
Number ordered out of service	688	1,200	1,490	1,725	2,539	3,281
Total number of defects found	36,968	60,292	77,268	85,530	112,008	136,973

Table 3—Accidents and Casualties Caused by Failure of Some Part of the Steam Locomotive, Including Boiler, or Tender

	Year ended June 30—					
	1931	1930	1929	1928	1927	1926
Number of accidents	230	295	356	419	488	574
Percent increase or decrease from previous year	22	17.1	15	14.1	14.9	16.8
Number of persons killed	16	13	19	30	28	22
Percent increase or decrease from previous year	[1]23	31.6	36.6	[1]7.1	[1]27.3	[1]10
Number of persons injured	269	320	390	463	517	660
Percent increase or decrease from previous year	15.9	17.9	15.8	10.4	21.6	13.6

[1] Increase.

Table 4—Accidents and Casualties Caused by Failure of Some Part or Appurtenance of the Steam Locomotive Boiler[1]

	Year ended June 30—							
	1931	1930	1929	1928	1927	1926	1915	1912
Number of accidents	91	105	119	150	185	247	424	856
Number of persons killed	15	12	14	26	20	18	13	91
Number of persons injured	122	113	133	174	205	287	467	1,005

[1] The original act applied only to the locomotive boiler.

part of the enginehouse force rather than to any structural defect in the boiler.

The decrease in accidents and casualties brought about by the decrease in defective locomotives, and the converse, are graphically shown by Fig. 2. The percentage of locomotives inspected found defective; number ordered out of service; number of accidents resulting from the failure of some part or appurtenance of the locomotive or tender, including the boiler; number of persons killed; number of persons injured, are shown in Table 1, for the fiscal years ended June 30, 1923 to 1931.

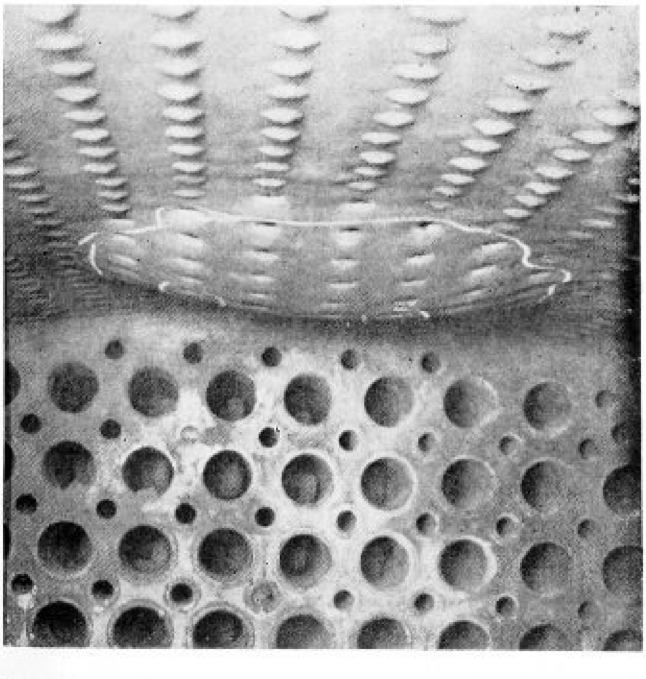
It will be noted from the table that from 1923 to 1931 both inclusive, the percentage of locomotives inspected found defective consistently decreased from 65 to 10; the number of locomotives ordered out of service decreased in the same manner from 7,075 to 688, or 90.3 percent; the number of accidents decreased from 1348 to 230, or 82.9 percent; the number of persons killed decreased from 72 to 16, or 77.8 percent; the number of persons injured decreased from 1560 to 269, or 82.8 percent.

Fig. 3.—Two were killed in this low-water disaster

It may be noted from Fig. 2, that there were fewer locomotive accidents during the year than at any time during the past 20 years, and that this reduction has been consistently made in about the same ratio as has the condition of locomotives been improved. If proper inspections and repairs are made in accordance with the requirements of the law and rules, many accidents will be avoided. Some of the carriers are maintaining their locomotives in such condition as to fully meet all the requirements of the law and the rules, regulations, and instructions made or given thereunder while other carriers were found to be delinquent in various degrees.

As in former years, boiler explosions caused by crown-sheet failures continue to be the source of most of the fatal accidents; 93.7 percent of the fatalities during the year occurred from this cause as compared with 84.6 percent for the previous year. It has been

Fig. 4.—Pocketed crown sheet caused by overheating



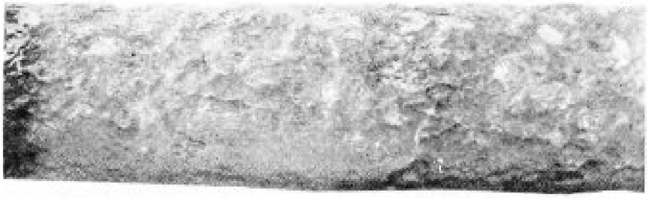


Fig. 5.—Pitted superheater flue

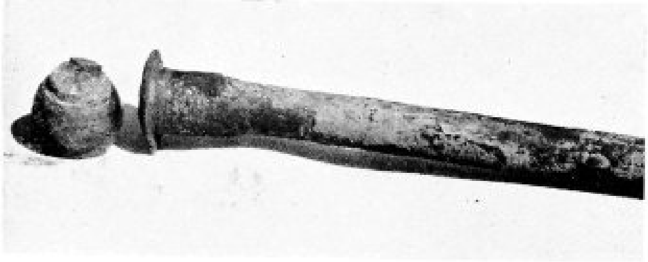


Fig. 6.—Broken flexible radial stay

pointed out in former reports that the increasing size of locomotive boilers and the higher pressures carried therein tend to increase the violence of explosions and cause increase in the average number of casualties per accident. Notwithstanding, there has been a marked decrease in the number of boiler explosions and firebox failures as the result of low water and other causes. For instance, during 1912 there were 94 accidents as compared with 14 during 1931, or a reduction of 85.1 percent; 54 persons killed as compared with 15, or a reduction of 72.2 percent; and 168 persons injured as compared with 41, or a reduction of 75.6 percent. This class of accidents can be minimized only by the use of the safest practicable firebox construction, reliable boiler feeding and water level indicating devices, and maintenance of proper water level in the boiler.

Four hundred and fifty-two applications were filed for extensions of time for removal of flues, as provided in Rule 10. Our investigations disclosed that in 34 of these cases the condition of the locomotives was such that extensions could not properly be granted. Seventy-three were in such condition that the full extensions requested could not be authorized, but extensions for shorter periods of time were allowed. Forty-one extensions were granted after defects disclosed by our investigations had been repaired. Seventeen applications were cancelled for various reasons. Two hundred and eighty-seven applications were granted for the full periods requested.

Under Rule 54 of the Rules and Instructions for Inspection and Testing of Steam Locomotives, 845 specification cards and 7138 alteration reports were filed, checked, and analyzed. These reports are necessary in order to determine whether or not the boilers represented were so constructed or repaired as to render safe and proper service and whether the stresses were within the allowed limits. Corrective measures were taken with respect to numerous discrepancies found.

Under Rules 328 and 329 of the Rules and Instructions for Inspection and Testing of Locomotives Other Than Steam, 114 specifications and 6 alteration re-

Table 5.—Number of Steam Locomotives Reported, Inspected, Found Defective, and Ordered From Service

Parts defective, inoperative or missing, or in violation of rules	Year ended June 30					
	1931	1930	1929	1928	1927	1926
1. Air compressors	481	873	1,202	1,282	1,679	2,151
2. Arch tubes	60	87	104	103	127	204
3. Ash pans and mechanism	81	76	132	133	192	211
4. Axles	10	12	20	7	13	8
5. Blow-off cocks	191	325	442	469	650	780
6. Boiler checks	263	521	761	914	1,043	1,200
7. Boiler shell	430	579	841	954	1,422	1,888
8. Brake equipment	1,923	2,706	3,894	5,214	6,572	7,062
9. Cabs, cab windows, and curtains	1,484	3,066	2,140	1,670	2,055	2,666
10. Cab aprons and decks	415	710	1,005	852	1,086	1,307
11. Cab cards	211	226	305	378	575	696
12. Coupling and uncoupling devices	98	122	154	179	289	394
13. Crossheads, guides, pistons, and piston rods	856	1,421	1,887	2,088	2,602	3,018
14. Crown bolts	96	95	129	164	235	334
15. Cylinders, saddles, and steam chests	1,265	2,311	3,210	3,264	4,526	5,080
16. Cylinder cocks and rigging	411	848	967	1,007	1,634	1,904
17. Domes and dome caps	83	154	227	281	388	463
18. Draft gear	568	950	1,310	1,453	2,037	2,634
19. Draw gear	640	1,003	1,367	1,650	2,210	3,140
20. Driving boxes, shoes, wedges, pedestals, and braces	925	1,359	1,993	1,990	2,710	3,342
21. Fire-box sheets	341	471	657	730	796	1,129
22. Flues	187	254	334	464	465	556
23. Frames, tailpieces, and braces, locomotive	740	1,271	1,377	1,354	1,682	1,973
24. Frames, tender	105	177	297	256	264	373
25. Gauges and gauge fittings, air	192	290	309	461	721	886
26. Gauges and gauge fittings, steam	324	553	678	969	1,425	2,038
27. Gauge cocks	415	783	1,114	1,413	2,024	3,068
28. Grate shakers and fire doors	410	767	295	377	613	720
29. Handholds	562	865	1,125	1,373	2,285	3,100
30. Injectors, inoperative	55	103	86	93	84	78
31. Injectors and connections	1,815	3,275	4,484	5,563	7,188	8,303
32. Inspections and tests not made as required	4,862	7,456	9,246	6,623	8,889	10,646
33. Lateral motion	289	372	618	699	673	758
34. Lights, cab and classification	77	119	121	118	107	106
35. Lights, headlights	180	373	488	571	835	946
36. Lubricators and shields	176	312	423	500	746	883
37. Mud rings	318	445	636	822	1,073	1,458
38. Packing nuts	523	828	991	1,263	1,851	2,772
39. Packing, piston rod and valve stem	706	1,429	1,708	1,904	2,214	2,489
40. Pilots and pilot beams	160	272	371	386	507	638
41. Plugs and studs	182	348	482	619	740	1,087
42. Reversing gear	299	579	788	967	1,247	1,539
43. Rods, main and side, crank pins, and collars	1,520	2,488	3,465	4,152	5,137	5,683
44. Safety valves	61	116	170	172	212	270
45. Sanders	314	804	1,008	1,031	1,268	1,769
46. Springs and spring rigging	2,161	3,311	4,557	4,939	5,956	6,826
47. Squirt hose	184	313	387	478	644	975
48. Staybolts	293	395	542	590	631	905
49. Stay bolts, broken	938	1,098	1,197	1,867	2,373	3,582
50. Steam pipes	512	730	925	1,020	1,308	1,587
51. Steam valves	226	399	471	708	774	962
52. Steps	676	1,021	1,394	1,817	2,440	3,227
53. Tanks and tank valves	732	1,426	1,717	1,941	2,747	3,430
54. Telltale holes	151	183	174	241	377	487
55. Throttle and throttle rigging	574	1,175	1,554	1,889	2,233	2,618
56. Trucks, engine and trailing	714	1,141	1,605	1,914	2,363	2,860
57. Trucks, tender	1,059	1,531	2,144	2,610	4,114	4,929
58. Valve motion	497	827	1,067	1,262	1,568	1,576
59. Washout plugs	815	1,283	1,871	2,211	2,786	3,649
60. Train control equipment	9	48	60	112	.....	.....
61. Water glasses, fittings, and shields	955	1,501	1,816	2,115	2,973	3,621
62. Wheels	750	1,025	1,325	1,609	2,119	2,243
63. Miscellaneous—Signal appliances, badge plates, brakes (hand)	418	691	1,101	1,273	1,511	1,746
Total defects	36,968	60,292	77,268	85,530	112,008	136,973
Locomotives reported	60,841	61,947	63,562	65,940	67,835	69,173
Locomotives inspected	101,224	100,794	96,465	100,415	97,227	90,475
Locomotives defective	10,277	16,300	20,185	24,051	29,995	36,354
Percentage of inspected found defective	10	16	21	24	31	40
Locomotives ordered out of service	688	1,200	1,490	1,725	2,539	3,281

Table 6.—Accidents and Casualities Resulting from Failures of Steam Locomotives and Tenders and Their Appurtenances

Part or appurtenance which caused accident	Year ended June 30— 1931 Accidents			1930			1929			1928			1927		
	Accidents	Killed	Injured	Accidents	Killed	Injured	Accidents	Killed	Injured	Accidents	Killed	Injured	Accidents	Killed	Injured
Air reservoirs	..	..	..	..	..	..	..	..	..	..	..	..	3	..	3
Aprons	3	..	3	..	..	..	2	..	2	5	..	5	6	..	6
Arch tubes	2	..	3	..	..	..	1	..	2	1	..	1	2	..	5
Ash-pan blowers	..	..	..	3	1	2	..	..	..	1	..	1	2	..	2
Axles	6	..	8	7	..	9	7	..	8	5	..	8	6	..	7
Blow-off cocks	5	..	3	4	..	4	7	..	10	7	..	7	10	1	9
Boiler checks	1	..	1	5	..	5	1	..	1	3	..	4	2	..	2
Boiler explosions:															
A. Shell explosions	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..
B. Crown sheet; low water; no contributory causes found	10	7	32	6	7	5	11	11	12	15	16	25	14	14	14
C. Crown sheet; low water; contributory causes or defects found	3	8	8	5	4	8	6	2	8	7	4	12	5	3	12
D. Fire box; defective staybolts, crown stays, or sheets	1	..	4	1	..	1	1	..	3	..	..	..	..	..	..
Brakes and brake rigging	8	..	9	21	..	23	16	..	17	14	..	14	25	1	26
Couplers	9	1	10	9	1	13	5	..	6	13	1	14	15	..	16
Crank pins, collars, etc.	7	..	8	3	..	5	2	..	2	8	..	8	3	..	4
Crossheads and guides	4	..	4	4	..	5	3	..	10	3	..	3	7	..	7
Cylinder cocks and rigging	3	..	3	1	..	1	1	..	1	6	..	6	3	..	3
Cylinder heads and steam chests	..	..	..	2	..	2	4	..	4	1	..	1	4	..	4
Dome caps	..	..	..	..	..	..	..	..	..	1	..	1	..	..	..
Draft appliances	..	..	..	1	..	1	3	..	3	1	..	2	2	..	2
Draw gear	..	..	..	1	..	1	5	..	6	2	..	2	5	..	6
Fire doors, levers, etc.	2	..	2	8	..	8	4	..	4	8	..	8	6	..	6
Flues	13	..	13	10	..	14	7	1	7	17	..	21	23	1	26
Flue pockets	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..
Footboards	4	..	4	7	..	7	7	..	7	11	..	11	10	..	10
Gauge cocks	..	..	..	..	..	..	1	..	1	..	..	..	..	..	..
Grease cups	1	..	1	3	..	3	5	..	6	1	..	1	1	..	1
Grate shakers	8	..	8	18	..	18	16	..	16	25	..	25	29	..	29
Handholds	6	..	6	5	..	5	10	1	9	12	..	12	12	1	11
Headlights and brackets	1	..	1	2	..	2	2	1	1	3	1	2	6	1	5
Injectors and connections (not including injector steam pipes)	5	..	5	4	..	4	6	..	6	7	..	7	12	..	12
Injector steam pipes	1	..	1	2	..	2	2	..	2	3	..	3	4	..	5
Lubricators and connections	5	..	5	1	..	1	5	..	5	8	..	8	7	..	8
Lubricator glasses	1	..	1	..	..	..	2	..	2	1	..	1	..	..	..
Patch bolts	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..
Pistons and piston rods	5	..	5	..	..	..	4	..	4	2	..	2	4	1	3
Plugs, arch tube and washout	..	..	..	2	..	3	2	..	2	1	2	1	6	1	8
Plugs in firebox sheets	..	..	..	..	..	..	1	..	1	..	..	..	1	..	2
Reversing gear	12	..	12	14	..	14	23	..	23	35	..	35	30	..	30
Rivets	1	..	1	..	..	..	3	..	3	1	..	1	2	..	2
Rods, main and side	4	..	4	11	..	15	14	..	17	11	1	13	16	1	18
Safety valves	..	..	..	..	..	..	..	..	..	1	..	1	..	..	..
Sanders	3	..	3	2	..	2	3	..	3	2	..	2	5	..	5
Side bearings	..	..	..	..	..	..	1	..	1	..	..	..	..	..	..
Springs and spring rigging	4	..	4	4	..	4	10	..	10	10	1	11	14	..	18
Squirt hose	7	..	9	20	..	20	23	..	23	32	..	33	33	..	33
Staybolts	4	..	4	1	..	1	4	..	4	5	2	4	8	..	8
Steam piping and blowers	3	..	3	5	..	5	4	..	6	7	1	10	11	..	11
Steam valves	4	..	4	6	..	6	..	..	..	2	..	2	6	..	6
Studs	1	..	1	..	..	..	2	..	5	1	..	1	3	..	3
Superheater tubes	4	..	5	5	..	7	1	..	1	1	..	2	5	..	7
Throttle glands	1	..	1	..	..	..	..	..	..	1	..	1	2	..	2
Throttle leaking	..	..	..	..	..	..	..	..	..	1	..	1	..	..	..
Throttle rigging	1	..	1	3	..	3	2	..	2	3	..	3	6	1	6
Trucks, leading, trailing or tender	1	..	2	2	..	2	4	..	4	3	..	4	4	1	4
Valve gear, eccentrics and rods	6	..	7	5	..	5	14	..	16	8	..	9	22	..	23
Water glasses	8	..	8	15	..	15	18	..	18	13	..	13	10	..	11
Water glass fittings	2	..	2	1	..	1	1	..	1	1	..	1	2	..	2
Wheels	1	..	1	3	..	4	8	1	16	5	..	13	5	..	6
Miscellaneous	49	..	50	63	..	64	71	2	69	84	1	87	69	1	68
Total	230	16	269	295	13	320	356	19	390	419	30	463	488	28	517

ports were filed for locomotive units and 6 specifications and 13 alteration reports were filed for boilers mounted on locomotives other than steam. These were checked and analyzed and corrective measures taken with respect to discrepancies found.

Three suits for penalties, involving 44 counts for alleged violations of the locomotive inspection law and rules, were pending in the district courts at the beginning of the year. Judgments in favor of the Government were obtained in 2 cases, involving 14 counts; five counts were dismissed by stipulation or agreement, and penalties imposed on nine counts in the sum of $900. One case, involving 30 counts, was pending at the end of the year.

No formal appeal by any carrier was taken from the decisions of any inspector during the year.

The following details of several of the most disastrous explosions and boiler failures which occurred during the year are given, with illustrations showing the results that accompany such disasters.

Fig. 1, shows the result of a crown sheet failure caused by overheating due to low water; this accident caused the death of two employees and the serious injury of one employee.

The force of the explosion tore the boiler from the frame and hurled it forward 429 feet. The boiler alighted on the track and then slid forward for some distance where the locomotive running gear and train collided with it resulting in derailment of the running gear, tender, and 14 freight train cars, 8 of which caught fire and were destroyed.

Fig. 3, shows the result of a crown sheet failure caused by overheating due to low water; the explosion caused the death of two employees.

The locomotive was moving backward at the time of the accident. The force of the explosion tore the boiler from the frame and hurled it forward or in the opposite direction from which the locomotive was moving. The boiler first alighted on the back head, then bounded and alighted on the front end, after which it rolled over and came to rest 155 feet east of and 35 feet north of the point of explosion.

The top water-glass cock was found in closed position and apparently the enginemen were misled as to the height of water in the boiler because of false indication of the water glass.

Fig. 4, shows a pocket in the crown sheet caused by overheating. The locomotive was ascending a 2.2 percent grade at time of accident and the water level in the boiler was not being maintained at a sufficient height to protect the front end of the crown sheet. Two employees were seriously injured.

Fig. 5, shows the pitted condition of a superheater flue that had not been removed from the boiler at time of resetting fire tubes. The removal of superheater flues has not been insisted upon when fire tubes are removed provided the superheater flues are in good condition and the boiler can be thoroughly cleaned and inspected without their removal. This exception is subject to withdrawal if carriers permit defective or unserviceable superheater flues to remain in service.

An example is given in the report of a boiler tube that failed in two places, resulting in the serious injury of one employee. One break occurred in a safe end weld 15 inches from back flue sheet and the other break occurred approximately 4 inches to the rear of the failed weld and 2 inches forward of another safe end weld; material had been overheated in welding and failed weld had been improperly made.

Fig. 6, shows a broken flexible radial stay that blew out of crown sheet while boiler maker was attempting to stop leakage with the boiler under pressure. The stay broke at a 75 percent old fracture at the top end near the ball nut; the head had been excessively flattened by repeated working and the threads in crown sheet and threads on end of stay that engaged crown sheet had been practically destroyed by leakage and by hammering of the bolt in attempt to stop leakage. This accident resulted in serious injury of one employee while in the firebox attempting to stop leakage while the boiler was under steam pressure.

Incinerator Made from Scrap Boiler

The illustration shows the use of a scrap locomotive-type boiler for the economical disposal of refuse, garbage and miscellaneous scrap material at one of the largest passenger terminal yards in the country.

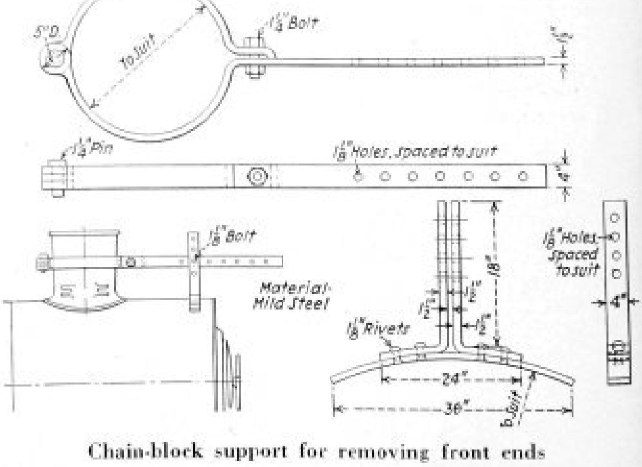
The boiler, which is of fairly large size, as indicated in the illustration, is upended on a concrete foundation, the firebox lined with firebrick and the upper half of it provided with a swinging door on a level with a substantial platform and inclined runway up which the refuse and scrap materials are trucked. The lower half of the firebox is open, being reached by a concrete-paved runway for the removal of ashes and any material which will not burn and falls through the grates. The latter are constructed of iron crossbars, generously spaced. The top of the incinerator is carefully screened to prevent the emission of sparks. The brick sidewall, stayed with rods as shown in the illustration, is provided simply as a medium of insulation to prevent the scorching of varnish on any cars which may happen to be on the adjacent track.

Scrap locomotive boiler used as an incinerator

The advantages of this incinerator, located where it is, may be summarized as the complete and ready disposal of passenger-car yard refuse at the point where it originates.

Device for Removing Front Ends

In small locomotive shops where crane facilities are not available, difficulty is often encountered in removing front ends due to the lack of a place for properly securing a chain block. This difficulty is overcome, and the removal and application of front ends is facilitated by the use of the device shown in the drawing. It consists of a clamp and a support, both made by 1½-inch by 4-inch bar steel.



Chain-block support for removing front ends

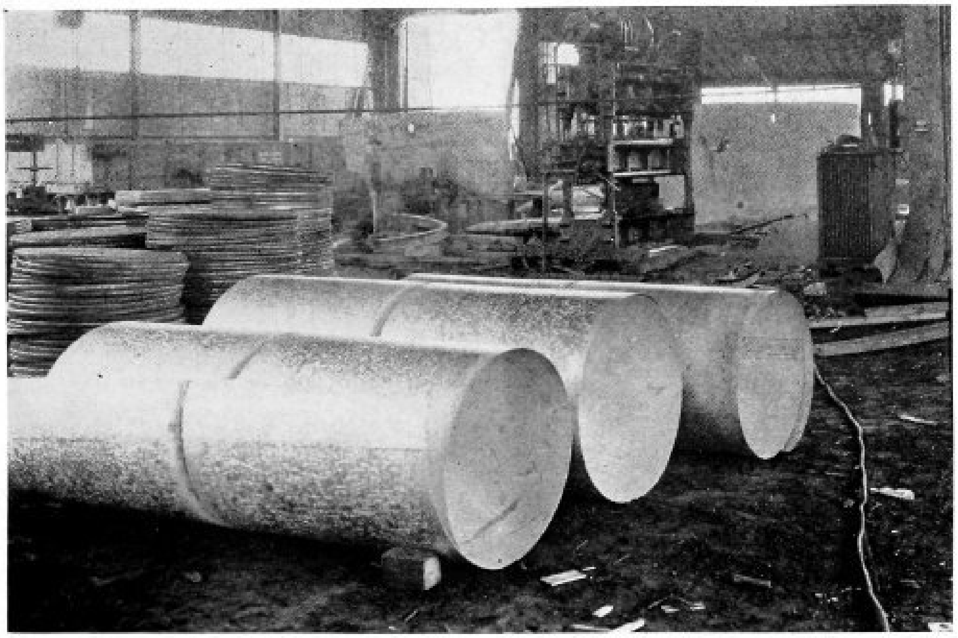


Fig. 1.—Lap-welded sheets after welding and rolling

Manual Welding of Tanks[*]

By G. G. Landis [†]

To those unfamiliar with welding, we have often answered the question "How do we begin?" with the advice to start at the bottom. This is figurative. We mean start with something small and, as the welding process becomes familiar, its possibilities will reveal themselves. So these articles will not attempt to set up hard and fast rules, but will rather suggest potential applications. While we may seem to be stressing primary fundamentals, full and complete knowledge of those fundamentals is necessary to the successful application of the process.

Our first experience with arc welding in production was in the redesign for arc-welded steel construction of a cast-iron compensator can used on our motors. This simple unit presented no major difficulty and since that time we have redesigned our entire line to eliminate costly castings and expensive rivets wherever possible.

Consider the welding of a small tank used, let us say, for storage. It will consist of a wall and head with fittings for filling and emptying the tank. The specifications will determine the thickness of the plate and the number of fittings, and the manufacturer's choice of arc welding will be based upon the low cost of such construction.

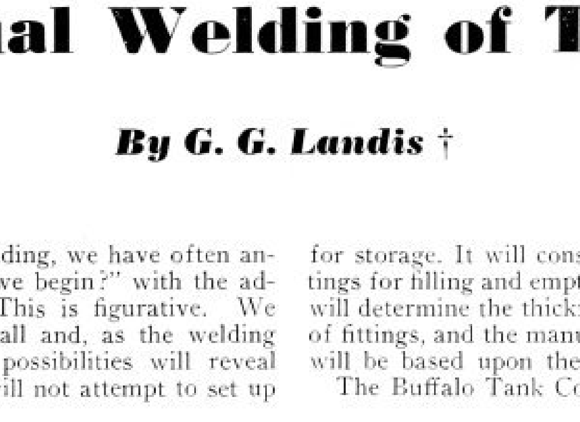


Fig. 2.—Tack welding longitudinal seams, the steel bar being used to press edges flush

The Buffalo Tank Corporation of Lackawanna, New York, equipped to fabricate all types of tanks, uses the following general procedure, depicted graphically in the illustrations.

The raw material is flat plate, some of which is cut for the heads of the tank. The sheets are first lap welded in order to make a tank of proper height. This shop is divided into sections and each section is grounded to a welding machine so that the various steps in production do not require continual hooking up of the unit to be welded with the machine. While this is a small item, it is one of those kinks which reduce handling and cut production costs.

In Fig. 1, sheets for galvanized tanks are shown after rolling into shape. This is 12-gage metal and the lap welds have been

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[*] Second article of series dealing with modern arc welding of tanks and pressure vessels.

[†] Chief engineer, The Lincoln Electric Company, Cleveland, O.

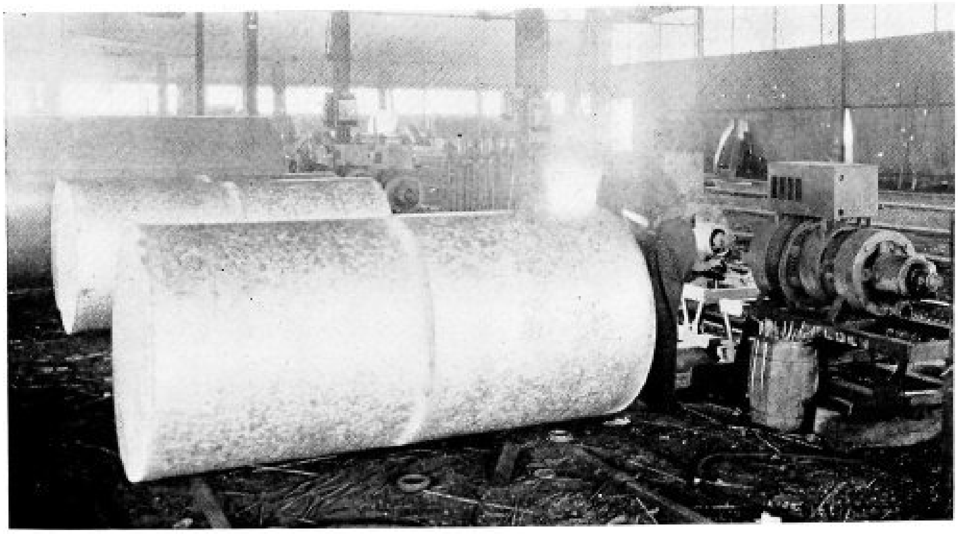

Fig. 3.—Completing the welding of longitudinal seams. For girth seams a rotating fixture is used in the welding operations

made on both sides. Note the press for stamping out heads in the background. Galvanized heads are fitted into this steel and the whole is rolled on to a grounded fixture where the heads are tack welded into position.

A welder and two helpers are responsible for the tack welding. Fig. 2 shows tacking operations on the longitudinal seam of one of these tanks. This is a lap weld and the bar, rolled along over the joint, insures a close fit at the line of fusion.

The tanks are then rolled on to a fixture which is wired to another welding machine, and the longitudinal seam receives a full bead of weld metal. This operation is shown in Fig. 3. Note the large tank which follows the galvanized containers on the production line.

For the girth seams the tanks are moved to the rotating fixture shown in the foreground of Fig. 3. Two operators complete the head seams. The speed of rotation is so adjusted that the tank revolves less rapidly than the operator is able to weld, and this allows the operator time to change electrodes and flex his muscles. Time studies in this shop have shown this to be the best method of production. The speed of the rotating fixture is easily adjusted for larger tanks and heavier metal.

The use of this fixture allows the operator to weld on a horizontal plane at all times which allows greater speed and requires less skill.

This entire shop is an example of procedure with arc welding. This company does not use riveted construction. Shop welding of all types of small tanks is done economically by the manual process because of careful fitting of all the elements before welding is begun. Tedious handling has been eliminated. Costs are lowered by 30 percent and the product is improved in that it is leak-proof.

The last operation in the shop is the testing. The unit is tested under air pressure with soap and water. Pinholes, which are rare, are closed with the hammer; large leaks are rewelded. The tanks are then painted and shipped.

Such a tank is leak-proof. The various members, fused together with the arc, combine to form what is in effect an intricate stamping, one piece. The engineers at this plant figure the weld as stronger than the plate metal and when the shielded-arc process is used better physical qualities, tensile strength, ductility and resistance to corrosion are insured.

As another example of shielded-arc welding, the illustration on the front cover of this issue shows a pressure vessel to operate at 250 pounds per square inch undergoing fabrication. This vessel was tested at 680 pounds per square inch pressure, and meets the A. S. M. E. Boiler Code requirements for Class 2 welding as well as those of the Hartford Steam Boiler Inspection and Insurance Company. It was one of six vessels manufactured by The Ohio Machine & Boiler Company, Cleveland, using the manual shielded-arc process.

(To be continued)

Acetylene Association Elects New Officers

Greatly increased interest in the continued development of oxy-acetylene welding and cutting was evidenced by the record-breaking attendance at the Thirty-Second Annual Convention of the International Acetylene Association held at the Congress Hotel, Chicago, November 11, 12 and 13, 1931. Over 1400 engineers, architects and representatives from industrial plants, factories and technical schools attended the various sessions of the convention. Important technical sessions were devoted to: "Transportation—Air, Rail and Motor Vehicle"; "Welded Piping—Production, Building and Overland Piping"; "Welding in the Chemical Industry"; and "Welding Education."

An outstanding feature of the convention was the evening program Wednesday, November 11, on "Welds, Weld Tests and the Welder." Dr. S. Lewis Land, educational director, Heating and Piping Contractors National Association, spoke on "Opportunities for Welding-trained Men in the Metal Working Trades and Industries" and T. M. Jones, superintendent of welding, Illinois Steel Company, South Chicago, Ill., discussed "The Training of Welders."

The following officers were unanimously elected for 1932: President, W. C. Keeley, Jr., vice-president, National Carbide Corporation; vice-president, E. J. Hayden, The Linde Air Products Company, Chicago; secretary, H. F. Reinhard, chief Engineer, J. B. Colt Company, New York; and treasurer, W. C. Cotter, The Oxweld Railroad Service Company, New York.

The following directors were elected to serve for three years: G. B. Walker, vice-president, The Linde Air Products Company, New York; L. F. Loutrel, Shawinigan Products Corporation, New York; and W. D. Flannery, Harris Calorific Sales Company.

What Can the Boiler Department Do to Reduce Pitting and Corrosion?

The problem of boiler corrosion has been one of constant concern to the railroads of the country, and one in which the Master Boiler Makers' Association has from its inception taken a keen practical interest. At each of the meetings of this association during recent years, a special report on this subject by members has been a feature, and the subsequent discussion by outstanding experts from the ranks of the water service engineers has made the work of the association an important one in the solution of the difficulty. The boiler makers of the country play an important role in combating corrosion and pitting and the report prepared by the Master Boiler Makers' Association for 1931 should prove of general interest to our readers. In later issues the contributions to the discussion of the subject by noted water service engineers will be published

Pitting and corrosion of locomotive boilers and tenders have existed since locomotives were first placed in service, and indications are that this trouble will continue to be present to some extent wherever locomotives are used. The possible loss from such conditions has become more noticeable during the past few years due to the large increase in size and cost of the modern locomotives, which are now required economically to compete with other forms of transportation. The pitting and corrosion may develop at such a slow rate in certain localities as to be of no great importance, and vice versa, it may develop at such rapidity in other localities as to be not only aggravating but actually serious from the standpoint of economical operation. There is no question but that considerable improvement has been made in arresting or inhibiting corrosion during recent years, but there still remains a fertile field for future work along these lines.

Large central station power generating companies are not troubled as much with pitting and corrosion as are the railroads using steam locomotives. This is due to the fact that the large power generating companies have a limited number of boilers operating under strict supervision and control at all times, and using water which does not vary appreciably in chemical composition from day to day, and requiring only a small amount of fresh make-up water due to the condensing equipment used. Contrast with this the steam locomotives of the country, which are extensive in numbers, using several different waters each day, which vary through wide ranges in quality, and utilizing as make-up not more than about ten percent of the condensed steam, the remainder being raw or treated water. In view of these operating conditions, the pitting and corrosion of locomotive boilers are much more serious and difficult of solution than the boilers in stationary service. Research and development work to counteract these destructive agencies must necessarily be handled largely by the railroad employees directly concerned.

In years past there has been considerable difference of opinion as to the causes of corrosion and several different theories have been advanced. In recent years, however, the electrolytic theory has been quite universally accepted and this is described briefly as follows:

The salts present in boiler waters break down into ions and these carry either a positive or negative charge of electricity. It has been determined that the hydrogen ions are those largely responsible for corrosion as these are displaced by iron ions from the steel, which have passed into solution, thus permitting more iron ions from the steel to enter solution. This electro-chemical action is also promoted by a difference in composition of various parts of the steel. A difference of potential has actually been found to exist between different spots on iron and steel sufficient to cause current to flow from one spot through the water to another spot, which results in the eating away of the metal at the first spot. The salts commonly found in water are good electrolytes and some are more active than others in assisting progressive corrosion. The presence of oxygen is said to increase corrosion under some conditions as it combines with iron ions to form iron oxide, thus permitting more iron to pass into solution and thus fits in with the electrolytic theory.

The above only explains the conditions which occur when corrosion and pitting exist, and does not explain the underlying causes, which may be one or more of several that contribute to the trouble.

The supervision of the water quality to reduce or control the hydrogen ion concentration and dissolved oxygen is largely a function of the water engineers and chemists. The method for proper handling, either by external or internal treatment is a question of economics to be determined for the individual situation and district. Methods have been developed and are available, the most important and workable of which is the maintaining of proper alkalinity in the boilers at all times. The records made in effecting economies by complete treating plants have been stressed in previous reports and discussions. We are advised that on one railroad the average flue mileage has been increased from 30,000 miles to between 120,000 and 130,000 miles, with a maximum of 315,000 miles, by using carefully supervised internal treatment with frequent alkalinity checks made by the terminal boiler inspectors to insure sufficient compound being added to produce the desired effect.

The results secured on many railroads have definitely shown that success of any method for chemical treatment is directly proportional to the amount of competent attention and supervision given, and that a very gratifying reduction in pitting and corrosion can be made by the proper and persistent application of the well known chemical treatments.

The coating of the interior of boilers with corrosion-resisting materials is said to be meeting with some success where applied to other than heat transfer surfaces. Experiments have been made by spray coating the water exposed surfaces of the boiler with a metal better able to resist corrosion than ordinary iron and steel, but the results are not considered sufficiently definite to warrant specific recommendation. The use of boiler tubes, sheets, and plates of a composition other than those quite generally recognized as standard, has not progressed to a point where definite conclusions can be reached as to the merits of such materials inhibiting corrosion.

In addition to the chemical treatment, some success is reported with the use of an electro-chemical process, which is based on using a small amount of an arsenic salt in the boiler and plating this out on the metal by passing a small current of minute electromotive force between electrode in the boiler and the boiler tubes and shell, which is claimed to counteract the destructive effect of the hydrogen ions under some conditions.

The responsibility and work of the boiler department is largely confined to materials and their application. Uniformity of quality of the material used in construction is very important, as irregularities in the composition of the steel will assist in increasing corrosion. Care should be taken to secure materials, which not only fully conform to the standards of uniform chemical quality, but also complete absence of segregation, or other physical defects, which would tend to start electrolytic action, due to difference in potential when immersed in water.

The boiler department should insist that reasonable care be taken of boiler materials before application, in order to avoid rusting and incipient corrosion starting, which would otherwise continue and cause trouble later in the boilers. All boiler materials should be kept dry while in storage by adequate provisions for shelter, drainage, and free circulation of air. It is equally important in corrosion prevention to avoid dents, cuts, kinks, or other injuries as well as to avoid rusting.

Boiler design and construction should be carried out, insofar as practicable, to avoid all undue strains in the materials which might result in a rearrangement of the crystalline structure that would cause a difference of potential and start electrolytic action when in service. The flanging or bending of sheets without proper annealing sets up strains which change the character of the metal and this has a tendency to promote electrolytic action, particularly where boiler waters of poor quality prevail. Care should be taken to see that no excessive strains are set up by too close a fitting die. All flanged sheets and straight sheets, where possible, should be furnaced and normalized.

It is quite generally recognized that where corrosion exists, it becomes most active in those parts of the boiler where scale accumulates. A possible explanation for this is that when scale is allowed to accumulate around the tube sheet, staybolts, etc., to an appreciable thickness, the metal beneath the scale becomes overheated to such an extent that its crystalline structure is probably changed and an ideal condition is set up for electrolytic action. The boiler department can assist in eliminating this kind of corrosion to a large extent by frequent and careful inspections, and giving special attention to see that no extra accumulation of either scale or sludge material is permitted to remain after washouts and that scale formation is kept at a minimum, consistent with practical railroading.

When boilers are blown down while hot they should never be allowed to stand after being emptied as the heat of the sheets combined with the heat of the brick arch, which is close to the crown sheet, has a tendency to bake the precipitants, which are in sludge form and bring about hard scale formation. This will deteriorate the sheets and have a tendency to build up fillets around crown and staybolts, which promote corrosion. It is considered best practice to wash out boilers as soon as practicable after they are emptied. It is further desirable that boilers be filled with hot water before firing up so as to eliminate insofar as possible the internal strains which occur during expansion and contraction of the metal.

It has been found that one of the most important and effective means for improving boiler conditions is the establishment and carrying out of a definite and systematic blow-off schedule, both at terminals and on line of road. This regular blowing will not only assist in keeping the boiler clean by removing the sludge and suspended matter, but in many instances a schedule can be put into effect, which will also eliminate sufficient dissolved alkali salts and other impurities to permit operating the locomotive boilers the full thirty days between washouts. In this way, it is possible to limit the maximum movement caused by expansion and contraction while washing boilers to the thirty-day government requirement. Objection has been raised to this practice at some points due to claims that fuel is wasted. This has not proven to be the case, as the water blown down through the blow-off schedule merely compensates for the water blown out at former washouts and water changes, and the cleaning of the boiler, which is effected by the frequent blowing, results in better steaming conditions and ultimate saving in fuel.

It should be borne in mind that the water from the blow-off cock has not taken up the heat necessary to change it to steam at boiler pressure, and the fuel lost will not average over one pound of coal for three gallons of water blown out. With the smaller fireboxes the location of the blow-off cock is not so important, but on the larger sizes much better results and more thorough mud removal can be obtained with the blow-off cocks in the back mud ring corners. If located in the throat sheet or near the front corners it will be found advantageous to use an internal perforated pipe collecting system. Development of suitable mufflers or distributers for taking care of the blown-off water has been of much assistance in carrying out the blow-off schedules. Where this method is used, regularity and system are of first importance, and spasmodic attention will only result in discouragement. With proper handling very satisfactory results can be obtained both in raw and treated water districts, which applies to stationary as well as locomotive boilers.

Where pitting and corrosion is experienced in engine tanks it has been found that one of the most effective remedies is a coating of cement wash. The red lead and oil paints commonly used do not afford adequate protection where pitting conditions are bad, and asphaltic enamels are subject to cracking on account of the distortion and movement of the tank.

The advantage of the cement wash lies in the ease of application and low cost. A two-coat job should last two years without further attention, after which time any blisters or loose coating can be scraped off and a new coating readily applied.

(Continued on page 14)

Fig. 1.—Prossering superheater tubes in a locomotive boiler

Application of Tubes to Locomotive Boilers*

An essential feature of locomotive boiler construction is the proper application of tubes to the tube sheets. Any description of this process naturally includes the types of tools necessary for its performance.

In the compilation of this article, it has been the aim to collect information which will enable workmen, who are not familiar with the proper procedure of applying locomotive boiler tubes, to be correctly informed on this subject, as well as to recommend the proper tools necessary for the process.

The tubes used in locomotive boilers are divided into three classes—the ordinary boiler tubes, sometimes called body tubes; large tubes for containing the superheater elements, frequently called boiler flues; and finally, those tubes which support fire brick arches, and which are commonly known as arch pipes. There is also included in this article some general information on the causes of leaks in locomotive boiler tube joints. The correct diagnosis of the cause of trouble is frequently the most important step in repair operations. Executives and employees who are interested in maintenance of locomotive boilers will find the study of these causes and the proper methods of making necessary repairs to be a very essential part of locomotive maintenance economy.

Many new locomotives on first arrival at destination are found to have minor leaks at the tube ends which are not due to improper construction, but which are caused by methods of transportation and rough handling of an empty boiler which does not have its tubes properly supported by the medium of surrounding water. The ends of boiler tubes should be slightly rolled or expanded before the new locomotive is put into operation. This process will avoid delay and lengthen the life of the boiler. It should be remembered that the rolling should be light, as when once a tube is really tight, excessive working will not make it more secure.

The use of the roller or of the sectional expander on the firebox end of the tube is largely a matter of local selection according to the intelligence and skill of the labor employed. Generally speaking, a roller expander should not be used by inexperienced men, preference in this case being given to the ordinary type of sectional expander.

All boiler tubes and arch pipes furnished by The Baldwin Locomotive Works, Eddystone, Pa., are of materials in accordance with the standard specifications of the American Society for Testing Materials, whether they be of iron, steel, brass or copper. The usual Baldwin practice is to furnish tubes of steel, but customers' specifications govern this matter when other materials are preferred. The usual standard thickness of steel tubes can be stated as follows: Body tubes having a length less than 19 feet are 0.109 inch thick (B. W. G. No. 12). Body tubes of 18 feet length and over are 0.120 inch thick (B. W. G. No. 11). Superheater tubes are 0.148 inch thick (B. W. G. No. 9). Arch pipes are 0.180 inch thick (B. W. G. No. 7).

If tubes are safe ended, the material for the ends should be one gage thicker than the body thickness of the tube, for example: If a No. 12 gage tube is used, then No. 11 gage should be used for the safe end.

Ferrules for all boiler tubes, including those for body tubes and for superheater tubes, are made of soft copper ⅛ inch longer than the thickness of the tube sheet and of the same outside diameter as the nominal outside diameter of the tube. Ferrules are made $\frac{1}{16}$ inch thick.

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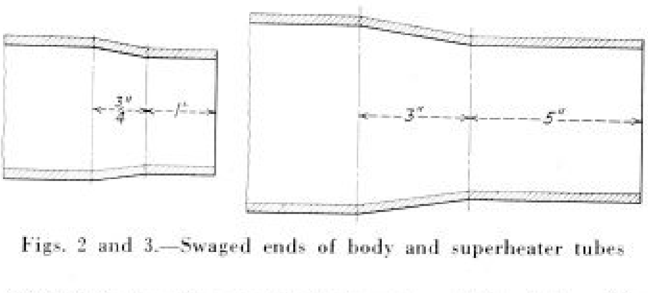
* Published through the courtesy of the Baldwin Locomotive Works, Eddystone, Pa.

In the preparation of firebox tube sheets, tube holes in the firebox tube sheet should be finished 1/64 inch larger than the outside diameter of the ferrule to be used. All holes in firebox tube sheets for tubes or arch pipes should be reamed with a straight reamer to exact size and the sharp edges removed. The edges of the hole should be finished to a radius of approximately 1/16 inch to prevent the cutting of the tubes. It is highly important that no sharp edges be left on the sheet, and both workman and inspector should see that the holes are properly prepared in this respect.

In the preparation of smokebox tube sheets, tube holes should be finished 1/32 inch larger in diameter than the outside diameter of the tube end and the tube holes must be cleaned and sharp edges removed in the same manner as recommended for firebox tube sheets. The contour of the edges is equally important for the smokebox tube sheet.

In the preparation of tubes, the firebox end of the body tube in ordinary Baldwin practice is swaged down toward its end, this swaging dimension being usually 1/8 inch below the nominal diameter of the tube. To permit the end of the tube to fit the ferrule neatly after the ferrule has been first tightened in the hole in the tube sheet, the swaged end of the tube exclusive of the tapered portion should not be less than 1 inch long and in general form as shown by Fig. 2. This length of swage allows 1/4 inch for beading beyond the edge of a tube sheet 1/2 inch in thickness. Any increase in the thickness of the tube sheet will necessitate a proportionate increase in the width of the ferrule and the length of the swaging. After swaging, the ends of the tubes should have all scale removed by light filing or grinding.



Figs. 2 and 3.—Swaged ends of body and superheater tubes

The firebox end of the superheater tube in ordinary Baldwin practice is swaged down to an average outside diameter of 4 1/2 inches, if the nominal diameter of the tube is 5 1/4 inches, 5 3/8 inches or 5 1/2 inches. Swaging for 5 inches nominal diameter tube is 4 1/8 inches outside diameter. The length of the swaging is always 5 inches irrespective of the thickness of the tube sheet to which it is applied. The details of superheater tube swaging are shown by Fig. 3. After the swaging operation, the end of the tube should have all scale removed by light filing or grinding. In setting the superheater tube at the firebox end it should be so gaged as to allow 1/4 inch for beading beyond the edge of the tube sheet. The general process of setting is similar to that used for the body tubes; the tube should be fitted neatly inside the ferrule, after the ferrule has been first tightened in the hole in the tube sheet.

Usually no preparation is required for the tube at the smokebox end other than cutting to correct length and the removal of scale by light filing or grinding. In special cases the tube may be expanded slightly at its front end, this expansion usually being limited to 1/8 inch increase above the nominal diameter. This is a matter of convenience in removing the body of the tube through the hole and not necessarily a feature of correct tube application. When finally set, the tube should be of sufficient length to project from 1/4 inch to 1/2 inch beyond the face of the smokebox tube sheet. Beading of all tubes at the smokebox end is not necessary. The practice of The Baldwin Locomotive Works is to bead from 10 percent to 15 percent of these for purpose of better support to the tube sheet.

Diagrams and descriptions are submitted showing the various operations required for the correct setting of locomotive boiler tubes. The description of these operations assumes that the holes in the tube sheet have already been properly prepared in accordance with instructions previously given.

In the first operation, ferrules must be placed in the holes in the firebox tube sheet as shown by Fig. 4, then tightened with a roller expander or lightly set with a sectional expander. The edge of the copper ferrule should be set 1/32 inch inside the face of the sheet.

In the second operation, the tube end should be placed in the hole in the firebox sheet by fitting it neatly into the already placed copper ferrule. The usual beading allowance is 1/4 inch, therefore the end of the tubes should project this amount beyond the surface of the tube sheet. It is suggested that the workmen employ a gage for this purpose so that uniformity can be obtained for the purpose of correct beading. After carefully setting into the ferrule, the tubes should be tightened with a roller expander or with a sectional expander as shown in Fig. 5.

Pneumatic operation of the roller expander will be found most convenient. An ordinary pneumatic drill of sufficient capacity can be used for this purpose, although a special driving tool can be procured if desired. The roller expander can also be operated by hand, the mandrel being turned carefully while maintaining even pressure on the expanding taper. Care should be used to not overwork the tube. Too much rolling is as bad as too little rolling. Operation of the sectional expander should be by a short-stroke pneumatic hammer or by a hand hammer. It is necessary to turn the expander in the tube between each consecutive driving in of the mandrel. One-quarter to one-half turn of the expander is recommended. Three or four driving operations are usually all that are required.

In the third operation, the front end of the tube should now be set. This operation is usually accomplished by the use of a roller expander as shown by Fig. 4. The projection at the front end is not so important as at the firebox tube sheet and exact gaging is not necessary, but the workmen should adhere to the limitations previously given.

In the fourth operation, all tubes (at both front and back ends) that are to be beaded should first be flared with a standard flaring tool. The tool recommended by The Baldwin Locomotive Works is shown by Fig. 6. This tool should be carefully used with a hand hammer or a short-stroke pneumatic hammer.

After completing the fourth operation, all tubes should be examined previous to beading, so that ends cracked in the expanding or flaring process may be detected. All tubes showing defects should be removed and replaced by sound ones.

The fifth operation of beading all tubes at their firebox end and the beading of selected tubes at the smokebox end should be carefully accomplished by means of a specially formed beading tool in the manner shown by Fig. 7. This should be done with a short-stroke pneumatic hammer or with a hand hammer. Care should be taken to see that nothing enters between the

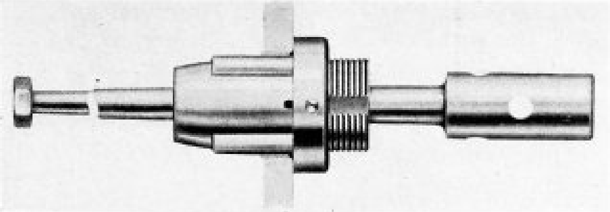


Fig. 4.—Roller expander

Fig. 6.—Standard flaring tool

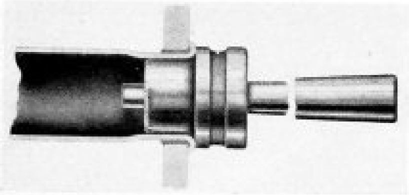


Fig. 5.—Sectional expander

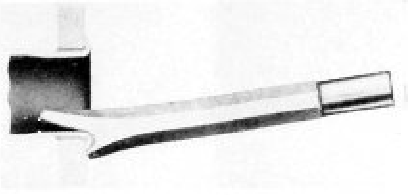


Fig. 7.—Beading tool

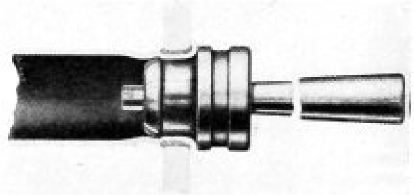


Fig. 8.—Prossering expander

bead and the tube sheet. The beading tool should be held so as to give the bead an outward set and force it down to the sheet without raising a fin on the inside, or nicking the outside of the sheet with the heel of the beading tool. Care should also be taken to properly center the beading tool inside the tube. It should be held radially and not slanted in one direction or the other. Skill is required for proper beading and extreme care should be taken to perform the operation perfectly.

In the sixth operation, steel or iron tubes as applied by The Baldwin Locomotive Works are usually prossered with a sectional expander as shown by Fig. 8. This prossering expander must be made to suit the exact thickness of the tube sheet, as the proper prossering operations must be done slightly beyond the inner edge to be effective. In prossering, the expander is carefully set and the mandrel is driven in a number of times; the expander being turned slightly each time the mandrel is withdrawn. This prossering operation is useful in making the tubes perfectly watertight and is considered a desirable practice. Care should be taken in the driving of the prosser mandrel so that even expansion may be obtained. If a pneumatic hammer is used, the short-stroke type is preferred.

In applying tubes to locomotive boilers it is highly desirable to use some systematic method. This can usually be left to the discretion of a careful workman. As a suggestion for good practice, a typical system is shown by Fig. 9, and description follows:

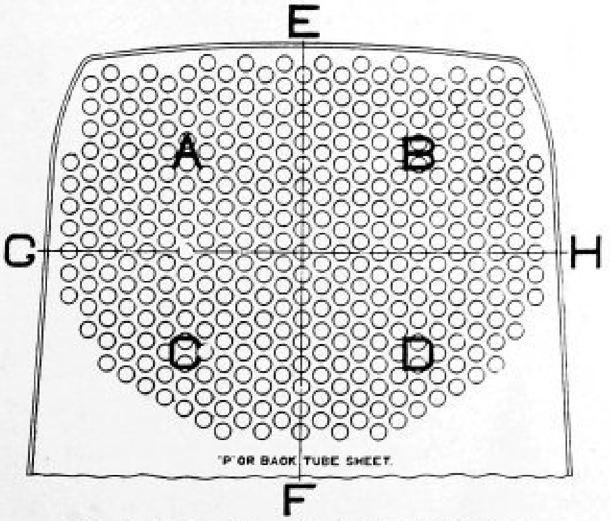


Fig. 9.—Systematic method of applying tubes

Expand all tubes on the lines *E-F* and *G-H* respectively, then work sections *A*, *B*, *C* and *D* successively; that is, all tubes in quarter section *A* should be properly set, then followed by those in quarter section *B*, and so on through quarter sections *C* and *D*. In performing these "sectional" operations, the best method is to commence at the outer edge of the sheet and work toward the center in the section of the line *E-F* and *G-H*. Better results will be obtained if two workmen are used, one at the front end and one at the back end, working the same tube simultaneously. The use of two men is of course obligatory in the first setting of the tube, as the holding in place to gage must be assured while the preliminary rolling or expanding is being done on the firebox end.

In many cases boiler tubes are sealed by electric-arc welding, and this operation is recommended by The Baldwin Locomotive Works for iron and steel tubes. Its accomplishment completes the last operation in the application of tubes.

After all operations for the application of boiler tubes are completed, the boiler should be filled with water and tested for leaks. The standard practice of The Baldwin Locomotive Works is to test by hydraulic pressure with warm water at 33⅓ percent above the working steam pressure required by the specification.

The application of large tubes in locomotive boilers fitted with superheaters follows exactly the same procedure as that for body tubes and the illustrations and operations will likewise govern. A roller expander, especially made for these larger size tubes, is used in the same manner as that for the body tubes. Prossering by the use of a sectional expander is always advised for the firebox ends of superheater tubes. In their application to the smokebox tube sheet, all superheater tubes should be beaded. Furthermore, the body tubes adjacent to the superheater tubes in the smokebox tube sheet should also be beaded to afford additional support.

In the application of arch pipes, The Baldwin Locomotive Works uses copper ferrules except when otherwise specified by the customer. Ferrules should be placed in sheets before the tubes are inserted and rolled tight with a tube expander. Arch pipes should be carefully placed in position and neatly fitted into their ferrules and tightened at both ends with the roller expander. In finishing the end of arch pipes, it is customary to bead them unless otherwise specified. Arch pipes are fastened in the interior firebox sheets and their setting must therefore be accomplished through the plug hole in the outside sheet which averages from 4 inches to 7 inches distant. When using the roller expander for setting arch pipes, it is necessary to remove the gage collar so that the tool can be applied through the plug hole. Beading and flaring tools for setting arch pipes must have extra long shanks for the same reason.

Many railways in the United States set arch pipes without the use of ferrules and without beading. In such case, the hole in the sheet should be carefully reamed true to size and the ends of the tubes rolled tight with a roller expander. After rolling, the ends should be flared with a flaring tool.

The causes of leaks in locomotive boiler tubes may be divided into two general classes: First, those leaks due to mechanical causes; second, those leaks due to variations in temperature.

Mechanical imperfections include improper tube setting in the first instance or improper maintenance in service. The firebox end is, of course, much more subject to mechanical defect, because its original application requires more careful and skillful treatment. The most prolific cause of trouble is due to hasty work in running repairs and its remedy is quickly stated, "Take time to do it well." Workmen should be instructed accordingly.

Another mechanical cause for leakage is vibration of the tubes. This need not be considered very important, as under ordinary operating conditions, a properly set tube is well supported by the medium of the water surrounding it. Vibration, however, will be the cause of loosening tube ends in an empty boiler transported long distances before being put into service. It is, therefore, advisable to lightly re-work the tube ends on all new locomotive boilers before they are put into service.

Another mechanical cause for leakage is the wearing of the tube ends by the abrasive action of the cinders. This is a matter well recognized by railways all over the world and is sometimes especially referred to as burnt off and cracked beads. The depth of the locomotive firebox, the location of the brick arch and other special variations in construction augment this trouble. It must be classed under the head of defective tubes, and these should be removed and replaced as soon as possible after the trouble develops. Such defective tubes may frequently be safe-ended and reapplied with a minimum loss of material.

Variations in temperature can be classed as equal or unequal. Little damage is done to the boiler tube ends when they are subjected to equal variations of temperature; that is, tube ends fitted to a tube plate can be heated and cooled a great number of times and not loosen, if all connected parts are heated and cooled uniformly. The unequal variations of temperature are really those which cause the leaks, and these should be avoided as far as possible. Careful firing can prevent temperature leaks to a great extent. Cold air should not be suddenly admitted to the firebox. When supplying fuel to the fire, the fireman should always make it a point to close the door after each shoveful is applied. Much discussion and difference of opinion exists regarding the effect of cold air in causing leaks in locomotive boilers. The closed fire door and intelligent firing are the best remedies no matter how slightly or how seriously variations in temperature may be considered as a cause for leakage.

The really important cause of leaks due to unequal variations in temperature is the varying change in the water temperature caused by the injection of feed water. The correct location of the boiler check is of prime importance in the locomotive boiler. It should be as far as possible from the firebox for convenient application. Checks located at the top of the boiler with a function for spraying the feed water at its first entry, are largely used in the United States, and may be considered as generally good practice.

By far the most prolific cause of leaking tubes is incrustation. This will, of course, vary with the quality of water supplied to the locomotive and the district in which the locomotive is operated. Its remedy is careful attention to washing and blowing off, and in extreme cases proper treatment of the feed water to reduce its propensity for scale formation.

As a general formula, it should be stated that the care of the tubes by methods of regular inspection for mechanical defects, by boiler washing and by intelligent firing, will be found one of the greatest economies in locomotive operation. The boiler is the source of power for the locomotive and should be carefully maintained in its entirety and especially at those points where the action of fire and water make for mechanical or chemical disturbances.

Boiler Pitting and Corrosion

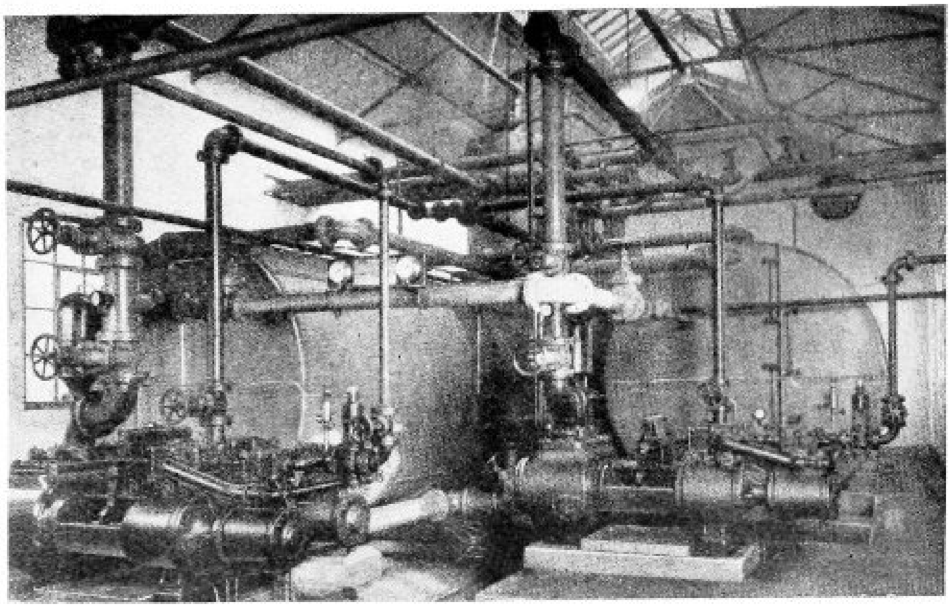
(Continued from page 10)

From the above outline it appears that insofar as the boiler department is concerned the work that it can best do to produce the greatest results towards reducing the losses from corrosion may be summarized as follows:

1. Utilize as far as possible material, which conforms to the best standards of chemical and physical quality.
2. Insist that proper care be taken of boiler materials prior to application.
3. Carefully watch and regulate the flanging and forming of sheets and plates, so as to avoid setting up internal strains in the metal which would tend to promote electrolytic action.
4. Thoroughly inspect all boilers at prescribed regular intervals and take necessary steps to insure scale formation is kept at a minimum.
5. Wash out boilers as soon as practicable after blown down, in order to avoid as far as possible the tendency for the scale to harden on the sheets and tubes. Fill boilers with hot water.
6. Equip boilers with facilities for easily handling blow-offs, and check the blow downs to insure the removal of as much objectionable matter as possible.
7. With present chemicals and treatment information available, the formation of heavy hard scale should be entirely avoided, and the attention of chemist or water engineer should be called promptly to any tendency for such formation, in order that proper chemical treatment may be made without delay.

This report was prepared by a committee composed of G. W. Buffington, chairman, M. V. Milton, and J. J. Davey.

J. K. B. Hare, whose headquarters with the Westinghouse Electric and Manufacturing Company, until recently have been in New York, has been appointed transportation manager of the Westinghouse Central District and will now be located in Pittsburgh, Pa.



Hot-water washing plant of the London and North Eastern Railway

British Locomotive Boiler Washing Plant*

The running sheds of the London and North Eastern Railway at Stratford form the main London depot for the Great Eastern section of the system, and have recently been equipped with a new boiler-washing plant, supplied and erected, to the specification of H. N. Gresley, chief mechanical engineer to the company, by Messrs. The Economical Boiler Washing Company, Limited, 34, Norfolk-street, Strand, W. C. The work undertaken included the re-arrangement and improvement of the water supplies; the installation of two stationary locomotive-type boilers providing power and heat for the hot-water washing-out plant, and a booster pump to deliver pressure water for the cold-water washing-out services.

The hot-water washing out plant is arranged on White's system, already fitted at a number of depots on the London and North Eastern Railway. There are three service mains from the central plant to various points in the sheds. Of these, the first serves to return to the system the water blown out of the boiler to be washed. The second provides the pressure hot water used for the usual washing out process, and the third service delivers clean hot water for filling locomotive boilers as required. The chief advantage of the system is that less time is required for the operation than if cold water were used, while it also facilitates the changing of boiler water when desired.

The building containing the main plant is situated between the main running shed and the smaller shed. The hot water services are arranged to be available at all of the twelve tracks of the main shed. There are six sets of distributing overhead pipe lines, each of which serves two tracks and in each line there are nine triple drop connections, spaced 45 feet apart. The smaller shed is served by two lines, which, between them, serve three tracks, and in this case each line has five triple drop connections similarly spaced. The central plant is sufficient to deal simultaneously with six locomotives on each of the three different services—that is, six can be blown down, six washed out, and six filled at any one time, and is stated to be the largest installation in the country. Its size was fixed on the basis of a 24-hour working day, and was necessary to cope with the very large number of engines stationed at this depot. The heavy traffic makes it most desirable that engines should be available promptly.

The central plant is housed in a brick building, which shelters two cylindrical hot-water tanks, of which the smaller, holding 15,000 gallons, acts as reservoir for the washing-out water, while the other, with a capacity of 20,000 gallons, supplies the filling water. Water from these tanks is delivered, under a pressure of 60 pounds per square inch, to the pipe lines by two horizontal duplex compound Worthington steam pumps, one connected to each tank. The exhaust from these pumps is passed into either or both of the tanks, as conditions may require. Water blown down from locomotives is treated in the following way. The blow-out pipe line enters a separator (or filter) and the water first strikes a baffle breaking it into a spray. Steam evaporated from the water on the reduction of pressure is led away and utilized in the heater by condensation and is fed into the filling tank. The remaining blown-down water passes through a filter containing broken coke to the wash-out tank. Sediment drains into a sludge tank which is blown to waste as required. The level in each tank is maintained by automatically controlled valves. Two locomotive-type stationary boilers, of which one serves as spare, are placed adjacent to the tank house.

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* Published through the courtesy of *Engineering*.

They provide steam at a pressure of 100 pounds per square inch to the two pumps, and act also as an auxiliary source of heat for the main tanks.

Three overhead mains connect up the central plant with the small shed, and four with the main shed, the blow out main being in duplicate. From each drop connection on the washing out and filling services, near the individual control valves, there is a small return main to the tanks, thus ensuring a complete circulation of hot water to every working point and maintaining its temperature. The two steam pumps, when no work is being done, run light at a slow speed, maintaining a pressure of 60 pounds per square inch and a slow circulation of the water. The opening of any valve in the shed on the washout or filling lines, reduces the pressure in the mains, and automatically brings the pump governor into action, speeding up the pump, which slows down again when the demand for water ceases. Both governors are set to maintain a pressure of 60 pounds per square inch in the mains, and each pump is of a sufficient capacity to maintain this pressure with a delivery of 30,000 gallons per hour. In case of emergency, it is arranged that each pump can draw from either tank and supply both services.

The working temperature of the water is maintained within predetermined limits, and these have been fixed as high as is practicable in the circumstances. The maximum temperature of the washing-out water used on this plant is 180 degrees F., and that of the filling water is 210 degrees F. These temperatures are automatically limited by thermostatically controlled valves. In the case of the wash-out water, the thermostat is placed in the pump delivery, and is set so that should the temperature rise above 180 degrees F., a normally closed diaphragm valve admits cold water to the pump suction, thus adjusting the temperature of the actual water delivered, while retaining the temperature of the tank water. The prescribed temperature limit for the filling water is obtained by placing a thermostat in the filling tank, which is connected to a normally open diaphragm valve, admitting live steam to the tank until the temperature reaches 210 degrees F., when the thermostat comes into operation and closes the valve. These thermostats can be adjusted to suit requirements. The method of pump regulation already mentioned, and the temperature control gear just described, enable the plant to work automatically.

The tank house also contains a 13-horsepower motor-driven centrifugal pump supplying the cold water washing out service. This unit has a capacity of 13,200 gallons per hour when pumping against a pressure of 43 pounds per square inch. Existing mains, previously used for washing out have, with certain alterations, been used for this service. This pump obtains its supply under a positive head, and a by-pass has been arranged for use in emergency, so that washing out can still take place during a breakdown of the pump. Such an arrangement insures flexibility of operation.

At the same time as the work already described was undertaken, rearrangements were made in the water supplies. Originally, water from several sources was mixed in a service reservoir located close to the Stratford running sheds and having a capacity of 110,000 gallons. The pressure of water available for any purpose was then strictly limited to about 13 pounds per square inch. The service to this tank was formerly interconnected with the general water supply of the locomotive works adjacent, which is derived partly from two wells and partly from the Metropolitan Water Board mains. The well waters are of inferior quality. Town water only is now fed to the tank and thence to the water cranes and the hot washing out plant, and it is now the only water used for filling locomotives. A new 8-inch town water main has been laid to the service tank ensuring an adequate supply of water at all times for every purpose. The mixed water supply has been separated and is used for both cold and hot washing out purposes, but if emergency arises the other source can, in addition, be drawn upon. By these modifications, the Stratford locomotive sheds have now a full supply of water at adequate pressure for all purposes and sufficient for all contingencies.

For the successful operation of the hot-water washing plant described, locomotives must, after the fire is dropped, come into the shed with a steam pressure of at least 40 pounds per square inch and be blown down with as little delay as possible. If this is done the greater part of the heat in the boiler water is recovered. The time occupied is about half an hour and the operation exercises a considerable cooling effect on the boiler so that hot washing out can be started immediately. Connection is made to the drop connections by a length of 2-inch armored hose pipe with unions which are suitable for fixing to the various hydrants. The time taken for the whole operation of blowing down, washing out, refilling and steaming varies with size of the locomotive. For a large engine of the Sandringham class (4-6-0), this is a minimum of six hours, which compares very favorably with the twelve hours required with the previous low-pressure cold water method. The fuel saving is substantial. Less is required for lighting up, engines are more quickly available, and the condition of the boilers improved. There is also a saving of water since the water in the locomotive boiler is filtered and used again for washing out purposes, and also because the make-up water used is inferior water, which, while not being suitable for boiler feed, is satisfactory for washing-out purposes.

Work of the A.S.M.E. Boiler Code Committee

The Boiler Code Committee meets monthly for the purpose of considering communications relative to the Boiler Code. Any one desiring information as to the application of the code is requested to communicate with the secretary of the committee, 29 West 39th St., New York, N. Y.

The procedure of the committee in handling the cases is as follows: All inquiries must be in written form before they are accepted for consideration. Copies are sent by the secretary of the committee to all of the members of the committee. The interpretation, in the form of a reply, is then prepared by the committee and passed upon at a regular meeting of the committee. This interpretation is later submitted to the council of The American Society of Mechanical Engineers for approval, after which it is issued to the inquirer and published.

Below are given records of the interpretations of the committee in Cases Nos. 680 (Reopened), and 695 to 698, as formulated at the meeting on October 23, 1931, all having been approved by the council. In accordance with established practice, names of inquirers have been omitted.

CASE NO. 680 (Reopened)—*Inquiry:* Will a steam-gage siphon which is formed to provide a water pocket between baffles in the body of the device, meet the requirements of the Code for Low-Pressure Heating Boilers if it is made of ferrous metal, if its water capacity is not sufficient to insure that the gage tube will be kept full of water under all conditions of normal operation, if the connection between it and the steam gage is of ¼-inch ferrous pipe, if the minimum area through the siphon is less than the equivalent of ½-inch pipe size, and if it is installed so that the water pocket is immersed in the steam space of the boiler?

Reply: The fact that the siphon is made from ferrous metal does not, in the opinion of the committee, violate the requirements of Pars. H-55 and H-108. The siphon as described does not meet the requirements of Pars. H-55 and H-108 if the water capacity is not sufficient to keep the gage tube filled with water under all conditions or normal operation, nor unless all parts of the piping less than ½-inch pipe size are of brass, copper, or bronze composition. It was not the intent of the code to prohibit the use of a ferrous siphon of the type described with an area less than the equivalent of ½-inch pipe size, nor to sanction the installation of a siphon with the water pocket immersed in the steam space of the boiler. A revision to incorporate the details covered by this interpretation is under consideration.

CASE NO. 695—*Inquiry:* When attaching to a dished head a cast or forged-steel nozzle which exceeds 6 inches in any dimension, it is necessary to increase the thickness of the head by ⅛ inch over that required for a blank unstayed dished head, provided the nozzle is properly reinforced as is required if the nozzle were placed in the shell? It is noted that under Par. U-36 no increase in thickness is necessary if the nozzle is placed on an elliptical head, but no mention is made whether or not this rule would also apply to a head that is dished to a segment of a sphere.

Reply: It is the opinion of the committee that when a cast or forged-steel nozzle is attached to a head dished to a segment of a sphere and is properly reinforced in accordance with the rules referred to in Par. U-59, no increase of thickness over that required for a blank unstayed dished head need be made.

CASE NO. 696—*Inquiry:* Would it be permissible to use plates of less thickness than ¼ inch in the fabrication of hot-water heaters of 24 inches or less in diameter, carrying a pressure of 100 pounds or less, as specified in Par. H-12 of the code?

Reply: It is the opinion of the committee that for such construction the minimum plate thickness is ¼ inch as specified in Par. H-12.

CASE NO. 697—*Inquiry:* Is it the intent of Par. U-73*c* of the code that dished heads inserted into shells as shown in Figs. U-15b or U-15c are permitted only in Class 3 vessels?

Reply: It is the opinion of the committee that inserted heads as shown in Figs. U-15b and U-15c of the code can be used only in Class 3 vessels.

CASE NO. 698—*Inquiry:* In using Table P-11 of the code to determine the minimum size of boiler outlets for safety valve connections on firebox boilers, is it permissible to interpolate between the values given therein for intermediate pressures, or shall relieving capacity for the next lower or next high pressure be used?

Reply: The discharge capacities given in Table P-11 may be interpolated to determine the values for intermediate pressures.

Preliminary Tests of Boiler for Panama Mail Liners

To provide the increased boiler pressure, higher efficiency and decreased maintenance essential for modern steam propulsion, the Foster Wheeler Corporation, New York, has developed a new steam generator for marine service, the first units of which will be installed in two of the four Panama Mail Liners, the *Santa Paula* and *Santa Elena*, being constructed by the Federal Shipbuilding & Dry Dock Company, Kearny, N. J.

The new boiler is of the three-drum, bent-tube, "A" design containing 4910 square feet of heating surface and operating at a pressure of 400 pounds per square inch. It is equipped with superheaters, water walls and economizers and will burn oil fuel. Four of these boilers will be installed in each of the two vessels.

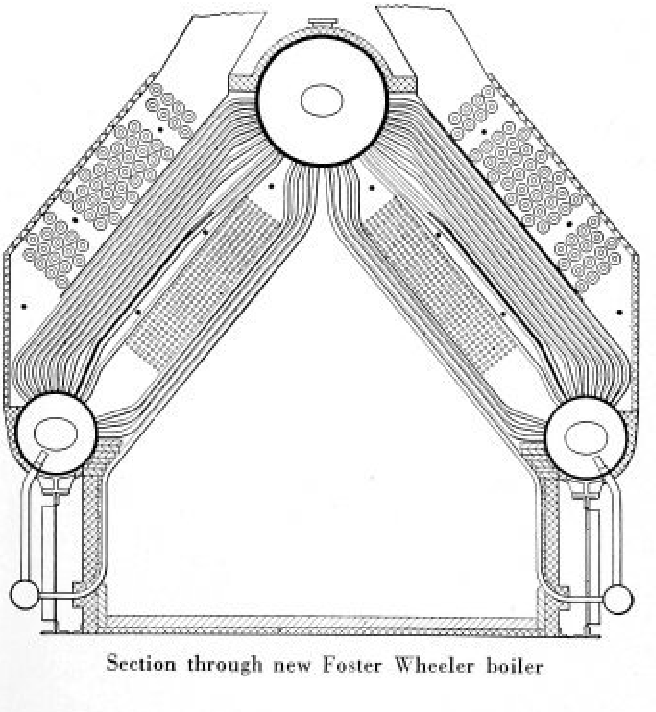
With the higher boiler pressures now prevailing free water circulation is of prime importance. This led to the choice of the three-drum, bent-tube type of boiler with its large furnace volume and compact arrangement of heating, superheating and economizer surfaces. Instead of supporting the refractory side walls in the furnace with tie bolts the fire rows of tubes have been extended down past the lower drums to headers on a level with the floors of the furnace.

The superheater is of the convection type placed close enough to the floor to give a fairly constant steam temperature regardless of the boiler rating. The economizer sections are installed at the back of the tube banks instead of above the boiler thus permitting the height of the furnace to be increased without projecting the economizer above the fireroom space limits.

The air required for combustion of the oil is blown downward from the top of the rear furnace wall in a double steel casing, thence under the boiler furnace floor and up over the front wall.

The boiler characteristics are as follows:

Boiler heating surface	4910 square feet
Economizer heating surface	3024 square feet
Superheating surface	1955 square feet
Total heating surface	9889 square feet
Furnace volume	1120 cubic feet
Maximum designed boiler pressure	450 pounds per square inch
Pressure at superheater outlet	400 pounds per square inch
Steam temperature	750 degrees F.



Section through new Foster Wheeler boiler

Upon completion one of the boilers was set up with a stack and accessories at the end of the pier at the plant of the Foster Wheeler Corporation, Carteret, N. J., for complete tests. The results of the tests show remarkable flexibility and efficiency. The boiler has been operated at ratings varying from 10,000 to 40,000 pounds of steam per hour and the final steam temperature has not varied more than 10 degrees.

The results of two of the preliminary tests run on December 11 are given in the following table:

Number of test	8	9
Steam pressure in drum, pounds per square inch	421	421
at superheater outlet, pounds per square inch	404	392
Final steam temperature, degrees F.	745	755
Superheat, degrees F.	296	310
Steam per hour, pounds	28,700	36,250
Oil per hour, pounds	2,146	2,770
Pounds steam per pound of oil	13.3	13.1
CO_2 in uptake, percent	14.2	14.4
Temperature gases in uptake, degrees F.	324	309
Temperature of feed water entering economizer, degrees F.	185	153
Temperature of feed water leaving economizer, degrees F.	278	271
Moisture in steam, percent	0.8	0.8
Heat added per pound of water, B.t.u.	1,221	1,262
Heat added per pound of oil, B.t.u.	16,400	16,600
Heat added per hour (in 1000 B.t.u.)	35,200	45,600
Heat in fuel per hour (in 1000 B.t.u.)	40,600	52,200
Heat in fuel per hour per cubic foot combustion space	37,100	47,700
Calorific value of fuel oil, B.t.u. per pound	18,900	18,900
HEAT BALANCE		
	Percent	Percent
Overall thermal efficiency	86.5	87.4
Sensible heat loss	5.6	5.2
Latent heat loss	5.4	5.4
Radiation and unaccounted for	2.5	2.0
	100.0	100.0

When installed on the ships the boilers are expected to produce under normal conditions 35,000 pounds of steam per hour. They are guaranteed for an overall efficiency of 85.67 percent at this rating and the preliminary tests indicate that even better results will be secured. The flue gas temperature leaving the economizer is approximately 300 degrees F.

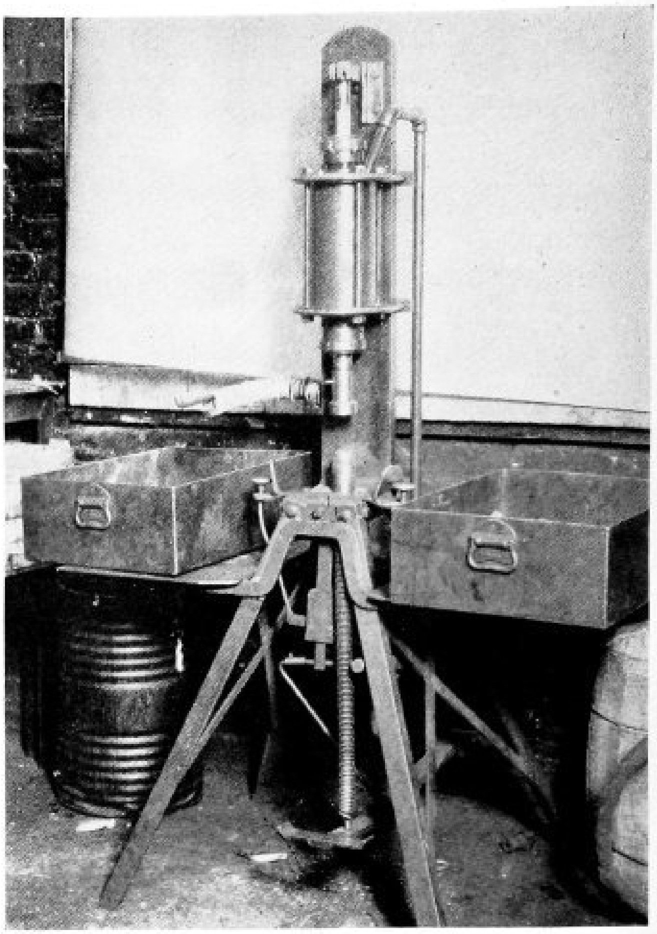
Cap Gasket Fitting Machine

Among the ingenious shop-made devices found in boiler shops of this country is a pneumatic tool, known as a flexible staybolt cap gasket fitting machine, developed by the Norfolk & Western Railroad, Roanoke, Va. As shown in the illustration, the machine consists mainly of a pneumatic plunger with properly shaped mandrels, a safety control, and a mandrel tripping device mounted in a sturdy frame.

The machine operates with a foot control which employs a dual safety device. This dual safety device consists of two pawl-like pieces fitting in notches on the control rod which is actuated by the pedal. Each of these pawls is disengaged by pressure on a knob at each side of the plunger. Unless both safety devices are released, employing both hands, the pedal cannot be moved. A guard plate protects each of the safety knobs so that, with one hand on each knob, there is no danger of mashed fingers as a result of the plunger operation.

The pneumatic tool itself has a spring to throw the plunger upward when the air pressure is removed.

An interesting feature of the machine is the tripping device to loosen the staybolt cap from the mandrel. This device is located at the rear of the pneumatic tool support and is not visible in the illustration. It consists of a lever and rod arrangement actuated when a roller on the upper end of the plunger rod passes over a cam. This motion, transmitted through a rod and levers, causes the mandrel to be depressed, thus releasing the staybolt cap.



Device for fitting staybolt cap gaskets

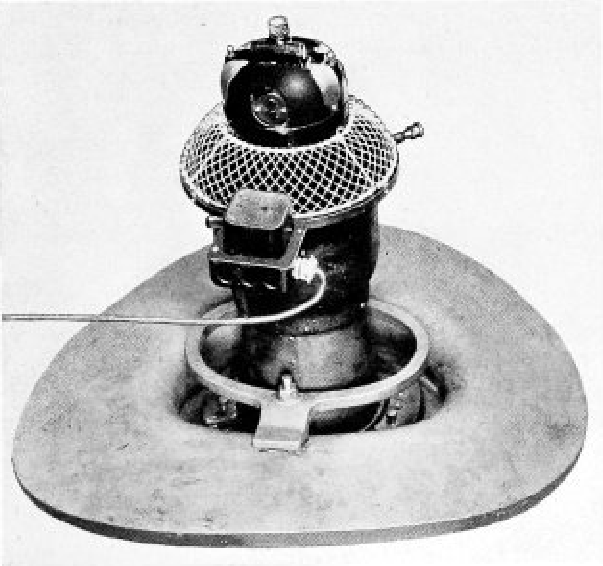
The entire machine is constructed from material generally found around a boiler shop. The stand is of welded angle and plate construction on which are built two shelves. In operation, a tray of caps to be gasketed is placed on one side, the gaskets on a hook to the left of the operators and the finished caps in the tray at the other side.

Small Capacity Boiler Manhole Blower

The Coppus Engineering Corporation, Worcester, Mass., announces the development of a smaller boiler manhole blower which is designated as Size No. 150. Although it has a capacity of 950 cubic feet per minute, the weight is negligible being only 48 pounds plus that of a detachable manhead and yoke weighing 20 pounds. The manhead and yoke on the blower and exhauster will fit manhole sizes 11 inches by 15 inches and 12 inches by 16 inches.

This boiler manhole blower is built around the Coppus Vano blower, a highly efficient design with non-overload characteristics. Driven by a ¼-horsepower, 110- or 220-volt universal motor, the unit is operable on either alternating or direct current. The outfit comes complete with a combined switch and circuit breaker, 12 feet of cord, and plug.

For use as an exhauster this unit may be supplied having a special inlet thimble and motor-cooling pipe.



Manhole blower weighing only 48 pounds

The flange on the inlet thimble is then identical with that on the diffuser, permitting the machine to be reversed to serve as a blower.

These blowers and exhausters are much used for rapid cooling and ventilating to facilitate repair or cleaning of boilers, furnaces, kilns, retorts, cracking stills, coke ovens, dehydrating pots, tanks and similar equipment.

The special advantages of the new size are lower price, lighter weight and extreme portability.

Multi-V Belt Grinder

The Hammond Machinery Builders, Inc., Kalamazoo, Mich., has recently developed a new machine for tool, casting and general-purpose grinding. This machine differs from many types of electrically driven grinders in that the power is transmitted from the motor to the spindle by means of multi-V belts. The motor is mounted at the rear of the pedestal. This construction permits a spindle housing of small diameter, allowing considerably more clearance between the grinding wheel and the spindle housing. The machines can be used in tandem and operated at different speeds inasmuch as the speed of the spindle is not limited to the speeds of the alternating current motors. The grinder can be supplied with 10-, 12-, 14-, or 16-inch diameter wheels in one-, two-, three, and 5-horsepower capacity. A feature of this machine is the fact that the whole spindle assembly can be removed from the pedestal without disturbing any mechanical part. This is an advantage when it is necessary to renew belts. The spindle is mounted in oversized ball bearings which run in a bath of oil and are protected from dirt and grit by means of a double labyrinth seal. Universal adjustable wheel guards are standard equipment. These are adjustable to the wear of the wheel and a clearance between the grinding wheel and the top of the guard can be maintained at a safe dimension as the wheel wears. Shatterless glass adjustable eye shields are also standard equipment. The motor on this machine is of the completely enclosed type, ventilated by means of a patented motor air cleaner.

General purpose grinder using V-belt power transmission

Lukens Steel Company Notes

J. Frederic Wiese, assistant to the general manager of sales of the Lukens Steel Company, Coatesville, Pa., has been appointed assistant to F. H. Gordon, vice president in charge of sales. Mr. Wiese, upon graduation from Swarthmore College in 1921, joined the Parkesburg Iron Company, Parkesburg, Pa., as mill representative with headquarters in Chicago, covering the mid-west, southwest and northwest. In this work, his principal contact was with the railroads. In 1926, Mr. Wiese joined Lukens Steel Company as assistant to general manager of sales, and two years later was placed in charge of the sale of Lukens products to the railroads. In his new capacity, Mr. Wiese will continue his supervision over railroad sales, and will be located, as before, at Coatesville.

J. Frederic Wiese

The Lukens Steel Company, Coatesville, Pa., has established in the Mellon Institute of Industrial Research, Pittsburgh, Pa., an industrial fellowship whose purpose is the scientific investigation of processes employed in the manufacture of steel plates.

Lukenweld, Inc., (division of Lukens Steel Company, Coatesville, Pa.) which is engaged in the design and manufacture of parts for machinery and equipment by gas-cutting, press-forming and arc-welding of rolled steel, has appointed two new representatives to handle the sales and service of its products. In the state of Wisconsin, Lukenweld, Inc., is now represented by Welding Engineering Company, 2872 North 41st Street, Milwaukee. In the Buffalo, N. Y. territory Marvine Gorham, Jackson Building, Buffalo, is the new representative.

The Boiler Maker

Volume XXXII Number 1

PUBLISHED Monthly by the Simmons-Boardman Publishing Company, 34 North Crystal Street, East Stroudsburg, Pa., with Executive Offices at 30 Church Street, New York City.

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Chicago: 105 W. Adams Street. Washington, D. C.: 17th and H Streets, N. W. Cleveland: Terminal Tower. San Francisco, Cal.: 215 Market Street.

Subscription Price: United States and Mexico, $2.00, Canada, $3.50, including duty. Foreign Countries, $3.00 a year. Single Copies, 35 Cents.

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The Boiler Maker is a member of the Associated Business Papers, Inc. (A. B. P.), and the Audit Bureau of Circulations, (A. B. C.)

Editorial Staff: H. H. Brown, Editor. L. S. Blodgett, Managing Editor. W. Lumbard, Associate Editor.

Contents

Communication

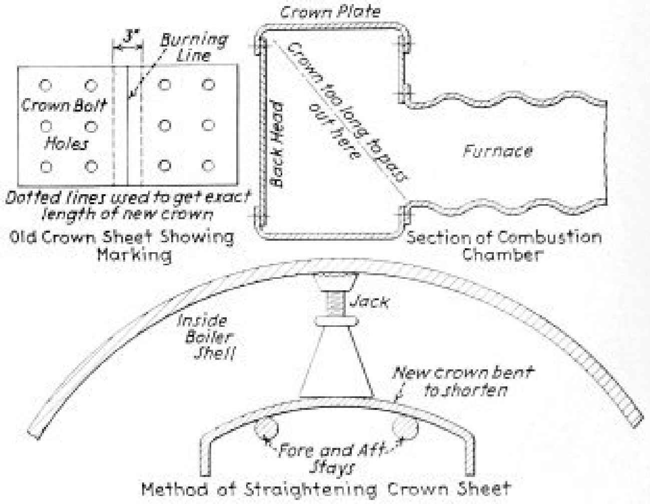
An Unusual Repair Job

To the Editor:

Recently, a steamship came into Montreal harbor with only one boiler in service, the other being shut down because of a peculiar accident to one crown sheet of the combustion chamber. The crown sheet was buckled and torn away from some of the crown dog staybolts and the plate was cracked between a number of the bolt holes. As the result of survey a new crown sheet was specified for installation.

The firm which received the contract for renewal set to work and burned off and numbered all of the dog stays. They burned out all of the seam rivets and then dropped the crown sheet to the bottom of the combustion chamber. When they tried to get the crown sheet out of the boiler through the furnace, however, the boiler maker found that it was too long and the sheet became jammed between the backhead and the bottle neck of the furnace. They stood the plate up against the backhead and struck a chalk line midway between two rows of crown bolts. Two additional lines were then marked parallel to the first line, these being 1½ inches on either side of the center line. These lines were center-popped and were used as guides to obtain the correct length of the new crown plate, as the burning of a plate is irregular and the gap is too wide to allow for correct measurement. The crown sheet was then cut in two parts across the center line and the plates were pulled out through the furnace, the metal front being removed from the furnace mouth while the burner was busy cutting the old sheets.

Because of the trouble in removing the old plate, the manager suggested making a new crown sheet in two plates joined with a seam of rivets. The foreman boiler maker, however, insisted that he could put back a new crown sheet in one plate. Eventually he was allowed his way, and the following is a description of his methods in completing the job:



Method of replacing a crown plate Scotch boiler

He laid out the old plate on a surface table and marked off the new sheet, being sure that the old sheet was set such that the lines on each side of the burned center line were 3 inches apart. This gave him the correct length of the plate. The new sheet was then punched, the crown bolt holes drilled and the plate was flanged on two ends.

In his next step he took a piece of gage iron, the same thickness as the new plate, and flanged this similar to the old plate. With this he went into the back end of the boiler and bent the strip so that it had sufficient clearance to pass from the furnace up through the combustion chamber. The new plate was then passed through bending rolls and bent similar to the gage iron. Shaped in this manner the crown sheet was then pulled through the furnace up into the boiler and placed across two of the large fore-and-aft stays. A jack was then placed on top of the plate and jammed between the inside of the shell plate and the crown plate. The crown plate was heated with two burning torches and the jack pressed the bend out of the plate so that it became straightened.

The plate was then bolted up, the only difficulty being that the two flanges had to be wedged from the other box, their location being too low to permit the use of a hammer. This was due to the small space between the boxes and its location in the water space. The job

(Continued on page 22)

Questions and Answers Pertaining to Boilers

This department is maintained for the purpose of helping those who desire assistance on practical boiler shop problems. Inquiries should bear the name and address of the writer. Anonymous communications will not be considered. The identity of the writer, however, will not be disclosed unless the editor is given special permission to do so.

By George M. Davies

Pressure on a Vacuum Tank

Q.—We have an order at my place of employment for a tank 126 inches diameter by 168 inches long, overall, to be designed and built for 28 pounds of vacuum. I would like to know what internal pressure this would be equal to? I also would greatly appreciate the formula or methods used in designing this kind of work. C. W. E.

A.—The question states that the tank is to be designed and built for 28 pounds of vacuum. I am assuming that what is intended is a 28-inch vacuum.

A vacuum is an enclosed space in which the pressure is less than that of the atmosphere and greater than absolute zero; that is, zero pressure of a perfect vacuum.

At sea level the pressure of the atmosphere is ordinarily 14.74 pounds per square inch measured above absolute zero. With a perfect vacuum at sea level the barometer would read 30 inches.

$$\frac{14.74}{30} = 0.4916 \text{ pound pressure per inch of vacuum.}$$

The atmospheric pressure exerted upon the tank due to a 28-inch vacuum would be $28 \times 0.4916 = 13.7648$ pounds per square inch. The absolute pressure in the tank would be the difference between 14.74 and 13.7648 or 0.9752 pound per square inch.

The pressure on the tank due to the 28-inch vacuum is a variable figure being influenced by the increasing elevation above sea level and weather conditions. The absolute pressure in the tank remains constant. These differences in pressure, however, do not need to be considered in this problem.

The pressure on the tank would be the difference between the atmospheric pressure and the absolute pressure in the tank due to the 28-inch vacuum or $14.74 - 0.9752 = 13.7648$ pounds per square inch pressure on the tank, the pressure for which the tank should be designed.

There are various formulas for computing the collapsing strength of thin cylindrical shells, the following being a typical one:

$$P = c\left(\frac{t^{2.25}}{L \times D}\right)$$

Where, P = collapsing pressure of a cylindrical shell in pounds per square inch.

t = thickness of shell in inches.
D = outside diameter of the shell in inches.
L = length of cylinder in inches.
c = a constant determined from experiments and given as 11,600,000.

The above formula is for tanks whose lengths do not exceed six diameters.

In using this formula when L is greater than six diameters, the value of L may be taken as $(6 \times D)$.

Dimension L refers to the length of the plain section of the shell which may be either the center-to-center distance between the circumferential seams or else the distance between the reinforcing bands, whichever the case may be.

From the above we infer that any cylindrical vessel subjected to external pressure may actually be greater in length overall than six diameters, provided the shell be built up in sections and reinforced at intervals by means of angles or bands riveted to its circumference.

For the required factor of safety of 5, the constant c in the formula reduces to

$$\frac{11{,}600{,}000}{5} = 2{,}320{,}000.$$

Values to be used in solving problems by means of the formula are as follows:

TABLE 1

Thickness of Shell Plate—Inches		Corresponding Values
Fraction	Decimal Equivalent	of $t^{2.25}$
1/4	.25	.0442
9/32	.28125	.05761
5/16	.3125	.07301
11/32	.34375	.09048
3/8	.375	.11004
13/32	.40625	.13173
7/16	.4375	.15566
15/32	.46875	.1818
1/2	.5	.2103
17/32	.53125	.2409
9/16	.5625	.2740
19/32	.59375	.3094
5/8	.625	.3473
21/32	.65625	.3876
11/16	.6875	.4303
23/32	.71875	.4756
3/4	.75	.5234

Fractional powers of numbers like the expression $t^{2.25}$ are most conveniently found by means of logarithms. For convenience Table 1 gives the value of $t^{2.25}$ for plate thickness from ¼ to ¾ of an inch varying by thirty-seconds of an inch.

In the formula,

$$P = c\left(\frac{t^{2.25}}{L \times D}\right)$$

where $c = 2{,}320{,}000$,

$$P = 2{,}320{,}000\left(\frac{t^{2.25}}{L \times D}\right)$$

$$\text{Then, } t^{2.25} = \frac{P \times L \times D}{2{,}320{,}000}$$

Substitute the values given in the question. Using 13.7648 pounds for the pressure, solve for $t^{2.25}$.

After solving for $t^{2.25}$, refer to Table 1 to determine the thickness of plate required.

In using this formula, allowance should be made for the fact that the efficiency of riveted seams or welds has not been taken into account.

The thickness required in an unstayed dished head with the pressure on the concave side, when it is a segment of a sphere shall be calculated by the following formula:

$$t = \frac{8.33 \times P \times L}{2 \times TS}$$

Where, t = thickness of plate in inches.
P = maximum allowable working pressure, pounds per square inch.
TS = tensile strength, pounds per square inch originally stamped on the plate used in forming the head.
L = radius to which the head is dished, measured on the concave side of the head, inches.

The radius to which the head is dished shall not be greater than the diameter of the shell to which the head is attached.

Where two radii are used, the longer shall be taken as the value of L in the formula.

Unstayed dished heads with the pressure on the convex side shall have a maximum allowable working pressure equal to 60 percent of that for heads of the same dimensions with the pressure on the concave side.

When a head, dished to a segment of a sphere, has a manhole or access opening that exceeds 6 inches in any dimension, the thickness shall be increased by not less than 15 percent of the required thickness for a blank head computed by the above formula, but in no case less than ⅛ inch additional thickness over a blank head. Where such a dished head has a flanged opening supported by an attached flue, an increase in thickness over that for a blank head is not required. If more than one manhole is inserted in a head, the thickness of which is calculated by this rule, the minimum distance between the openings shall be not less than one-fourth of the outside diameter of the head.

Where the radius L to which the head is dished is less than 80 percent of the diameter of the shell, the thickness of a head with a manhole opening shall be at least that found by making L equal to 80 percent of the diameter of the shell. This thickness shall be the minimum thickness of a head with a manhole opening for any form of head. *No head shall be of a lesser thickness than that required for a seamless shell of the same diameter.*

The tank should be fabricated by means of welding unless otherwise specified. The construction of a welded tank is illustrated in the December, 1931, issue.

Definitions

Q.—Would you please interpret the meaning of No. 10 tubes, B. W. G.; and could you give me the answers to these two questions:
(1) In a boiler having more than two banks of tubes, what is the bank nearest the fire called?
(2) What one word used would mean valves and gages? F. R.

A.—The statement No. 10 tubes, B.W.G. is no doubt referring to the thickness of the tubes and would be more correctly stated as, Tubes, No. 10 B.W.G. Boiler tubes are manufactured to the Birmingham Wire Gage (B.W.G.) for thickness. A No. 10 B.W.G. tube would be 0.134 inch thick.

(1).—The question could be more definitely answered had the type of boiler been specified. In a firetube boiler of the locomotive type having two or more banks of tubes, the tubes leading from the firebox would be the firebox tubes and those leading to the smokebox, the smokebox tubes.

In a watertube boiler, as the Stirling boiler, having two or more banks of tubes, the bank of tubes nearest the firebox is often referred to as the first pass; the next bank of tubes, the second pass; and the next the third pass. This is the order in which the hot gases pass between them.

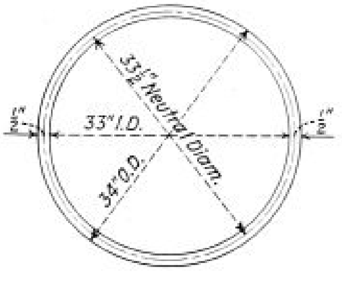
(2).—A collective name for those attachments to a boiler, necessary for its proper use, including grates, smokeboxes, uptakes, dampers, funnels, casings and the like, as well as those parts which are more commonly termed fittings by the engine room staff; namely, the various cocks, gages, valves, etc., are called boiler mountings. The brass mountings are more properly termed fittings; in some localities they are designated as trimmings.

Mounting is defined as that by which anything is prepared for use; equipment, as the mountings of a steam boiler, meaning the safety valves, water gages, steam gages, etc.

Pitch of Rivets in a Tank

Q.—Kindly explain the simplest way to find the pitch of rivets in a tank having a diameter of 33 inches with 5/8-inch rivets spaced 2 inches apart. R. J. T.

A.—The pitch of the rivets is the distance between the centers of the rivet holes, thus rivets spaced two inches apart have a 2-inch pitch. The rivets are usually laid out on the flat plate to the correct pitch.



Section of tank showing dimensions

Assuming that the 33-inch diameter of the tank, given in the question, is the inside diameter and that the tank plant is ½-inch thick, then the neutral diameter of the tank would be 33½ inches.

The rivet pitch should always be computed with the neutral diameter as a basis. The circumference of the neutral diameter is 105.243 inches. The tank would have 52 rivets in the circumferential seam spaced $\frac{105.243}{52}$ or 2.023 inches pitch. The holes are then punched or drilled as required before rolling.

The reason for computing the pitch from the neutral diameter is because the neutral diameter neither gathers nor stretches when rolling the sheet, but remains neutral, retaining its original length.

Unusual Repair Job

(Continued from page 20)

was then re-riveted and calked. Crown bolts were replaced and calked and dog stays or plate stays were replaced. New staybolt nuts were applied and the boiler was filled with water. The job tested all right and the steamship sailed away on time.

Montreal, Canada JAMES WISON.

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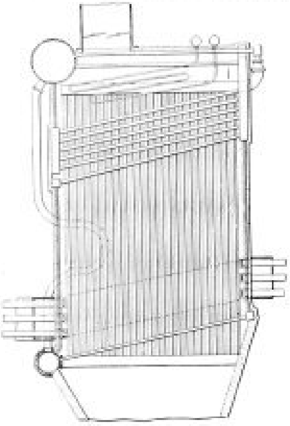
Selected Boiler Patents

Compiled by Dwight B. Galt, Patent Attorney, Washington Loan and Trust Building, Washington, D. C. Readers wishing copies of patents or any further information regarding any patent described, should correspond directly with Mr. Galt.

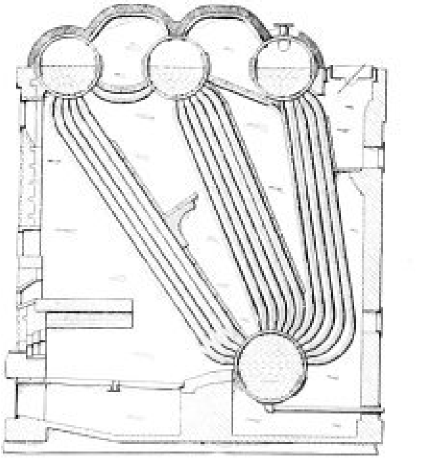
1,795,383. BOILER FURNACE. JOHN VAN BRUNT, OF FLUSHING, N. Y., ASSIGNOR TO INTERNATIONAL COMBUSTION ENGINEERING CORPORATION, OF NEW YORK, N. Y., A CORPORATION OF DELAWARE.

Claim.—A boiler furnace comprising in combination a boiler of the cross-drum type having a bank of substantially horizontal tubes each vertical row of which is connected into a front and rear header, a combustion chamber there-below largely defined by tubular water walls comprising a front wall formed of finned tubes and a header below the aforesaid front headers said tubes connecting said lower header and said front headers, a rear wall formed of finned tubes, an upper cross header therefor adjacent the aforesaid rear headers, and a lower cross drum therefor, side walls formed of finned tubes, upper cross headers therefor opposite the cross drum boiler and lower cross headers therefor, and a bottom screen of spaced tubes connecting the lower header of the front wall with the lower drum of the rear wall, means connecting the upper header of the rear wall with the front heaters of the cross drum boiler, a downcomer connection from the cross drum boiler to said lower drum and an upcomer connection from the upper headers of the side walls to the boiler, the lower headers of the side walls being connected with said drum, and means for introducing fuel to be burned in suspension in said combustion chamber. Five claims.



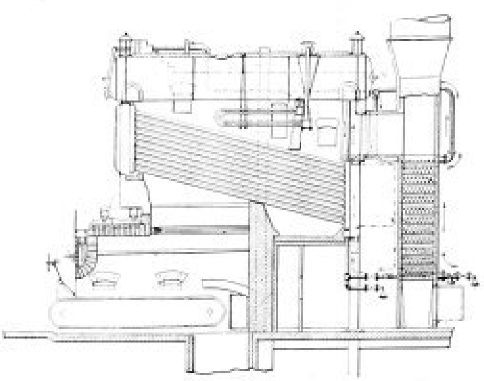
1,753,267. STEAM BOILER. DAVID S. JACOBUS, OF JERSEY CITY, N. J., ASSIGNOR TO THE BABCOCK & WILCOX COMPANY, OF BAYONNE, N. J., A CORPORATION OF NEW JERSEY.

Claim.—In a high-pressure steam boiler, two spaced drums each provided with openings, a bank of watertubes the ends of which are adapted to enter the openings of the respective drums in a plurality of circumferential and longitudinal rows, the portions of the drums entered by said tubes being of substantially uniform thickness, each tube being reduced in external diameter at one end at least where it enters the drum sufficient in extent lengthwise of the tube to permit sufficient movement of the reduced end of the tube through its opening in the associated drum to permit of the insertion of the opposite end of the tube in its opening in the associated drum and its removal therefrom, the intermediate portions of the tubes being relatively large to provide a relatively small gas flow area therebetween. Three claims.



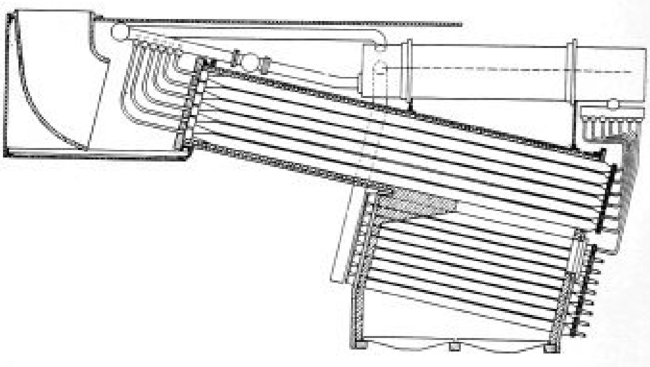
1,753,266. STEAM BOILER AND ECONOMIZER AND METHOD OF OPERATING THE SAME. DAVID S. JACOBUS, OF JERSEY CITY, N. J., ASSIGNOR TO THE BABCOCK & WILCOX COMPANY, OF BAYONNE, N. J., A CORPORATION OF NEW JERSEY.

Claim.—The method of operating a steam boiler and its economizer connected thereto, which consists in normally feeding water forwardly through the economizer to the boiler without circulation, and in circulating the water by gravity between the boiler and economizer in the direction opposite to that in which it normally flows during a layover period when the temperature of the gases passing over the economizer is subnormal and while the water in the boiler is hot. Thirteen claims.



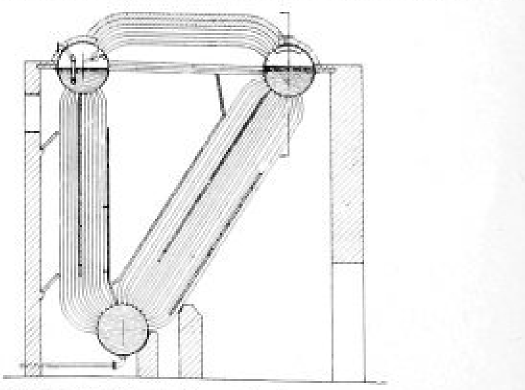
1,795,537. WATERTUBE BOILER PARTICULARLY FOR LOCOMOTIVES AND LOCOMOBILES. ULRICH BARSKE, OF HANOVER-RICKLINGEN, GERMANY, ASSIGNOR TO HANNOVERSCHE MASCHINENBAU-ACTIENGESELLSCHAFT, VORMALS GEORG EGESTORFF, OF HANOVER-LINDEN, GERMANY.

Claim.—A watertube boiler particularly for locomotives and locomobiles comprising a set of inclined straight tubes for conducting water, steam or a water-steam mixture extending through the space forming the smoke box and the upper part of the firebox, similar shorter tubes arranged on the side walls of the firebox, a detachable connecting tube for each end of the inclined tubes, and collector chambers to which with said inclined tubes are connected by said connecting tubes. Four claims.



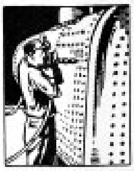
1,753,687. BOILER. GEORGE Y. BONUS, OF CHICAGO, ILL., ASSIGNOR TO ERIE CITY IRON WORKS, OF ERIE, PA., A CORPORATION OF PENNSYLVANIA.

Claim.—In a boiler, the combination of a front drum; a rear drum, said drums being in the plane of the water level; a lower drum; a front bank of tubes connecting the lower and front drum; a rear bank of tubes leading to the rear drum in circuit between the lower drum and the rear drum; an upper bank of watertubes between the front and rear drums, said drums and banks of tubes forming an unobstructed ring path from the lower drum to the front drum, from the front drum to the rear drum and from the rear drum to the lower drum forming the main circuit for the water; a furnace; and baffles comprising a plurality of baffles operating on the front bank of tubes controlling the application of heat to the tubes compelling an upflow of water in and throughout the front bank of tubes, a rearward flow of water in and throughout the upper bank of watertubes, and a downward flow of water in and throughout the rear bank of tubes. Six claims.



The Boiler Maker

Reg. U. S. Pat. Off.



Modern Marine Steam Generation

In ship operation, the generation of propulsive power is one of the costliest items. The engineering branch of the shipbuilding industry has in recent years more than ever before studied exhaustively the problem of increasing the efficiency of machinery and reducing fuel costs. Logically the first place to make such savings is in the steam-generating plant.

When marine practice turned from the old reliable Scotch boiler to the watertube type, a moderate increase in pressure followed, with a consequent increase in overall efficiency. Later, as in the power field, advantage was taken of still higher pressures and temperatures and higher fuel rates to realize yet more improvement.

An outstanding example of the trend in marine boiler practice is the steam generator described in this issue, which in test has given an overall thermal efficiency of 87.4 percent. This boiler operates at 450 pounds per square inch working pressure, and 750 degrees F. total temperature.

Corrosion and Pitting

Although the boiler department cannot of itself eliminate the problem of pitting and corrosion, it is in a position to assist materially in reducing the ravages of this most costly item of boiler maintenance by close co-operation with the water service department. It was to this end that the officers of the Master Boiler Makers' Association several years ago made the study of this subject one of major importance in the work of the organization.

During this period much has been learned of the causes and potential cures for pitting and corrosion, and the annual meetings of the association have made possible the greatest co-operation between the master boiler makers of the country and the engineers having the problem in direct charge. From the association report, which appeared in the January issue, it is evident that while the major work comes outside their scope there are, nevertheless, certain precautions which the boiler maker can observe during construction and maintenance of locomotive boilers to lessen the effects from bad water. Uniformity of the quality of materials used in construction is extremely important; for irregularities in the composition of steel will greatly aggravate the trouble. Care should be taken that materials during storage are protected against rusting which gives corrosion a start. Dents, cuts and kinks in the metal during storage or handling should be avoided as these are possible points of attack for corrosion later on. The design of a boiler and its actual construction should be carried out so that no undue strains in the material occur; for, when the crystalline structure of the metal is changed, electrolytic action in the water becomes possible, particularly where inferior waters are used. This action, which is briefly explained in the report, is undoubtedly the fundamental cause of both corrosion and pitting.

A practical example of potential difficulty is in the bending or flanging of sheets without annealing them afterwards to remove local strains in the metal. Too close fitting dies will cause strains. Scale accumulation on sheets, around staybolts, tube sheets and elsewhere in the boiler promotes overheating which, through the change of crystalline structure of the metal, also brings on active corrosion.

By frequent and careful inspection and removal of sludge and scale, the boiler department can do much to reduce corrosion to a minimum. A discussion of the part the boiler maker can play in combating this problem by prominent water service engineers is published in this issue.

Power Boilers

Statistics of the production of power boilers in the United States during 1931, which have recently been issued by the United States Department of Commerce, Bureau of the Census, show a drop of nearly 40 percent in the number built as compared with the year 1930. The total heating surface of the boilers built is approximately 46.3 percent less than in 1930. In actual numbers there were 8017 boilers of all types built having a heating surface of 7,229,618 square feet as compared with 13,166 boilers of 13,469,893 square feet in 1930, and 18,526 boilers of 19,468,534 square feet in 1929, the peak year since the statistics have been compiled.

This condition in the boiler field naturally reflects that of industry in general. It means that a great number of potential projects requiring steam power have been delayed and that, with the return of a more nearly normal level of industrial and building activity, the boiler manufacturers will have their share in any future prosperity.

Fig. 1.—Casing assembly of new marine boiler

New Small-Tube Steam Generator Introduced to the Marine Industry

Tests of a new type marine steam generator, developed by the Foster Wheeler Corporation, New York, which were reported in the January issue, indicated a boiler efficiency in excess of 87 percent, which is a particularly remarkable achievement when it is recalled that the tests were conducted in the open without cover or protection, during the month of December. The unit used in the tests is one of an order of eight for installation in two Panama Mail liners, the *Santa Paula* and *Santa Elena,* now under construction by the Federal Shipbuilding & Dry Dock Company, Kearny, N. J.

The steam generation requirements for each boiler set forth in the original specifications, call for a normal steam production of 26,100 pounds per hour with maximum requirements of 35,000 pounds of steam per hour. The operating pressure was given as 400 pounds per square inch with a final steam temperature of 750 degrees F. The customary limitations of space in a ship govern the consideration of the type to be employed, while light weight, ready accessibility, reliability, high efficiency, low draft loss, and the like, are all vitally important factors determining the availability of any type for marine service.

In developing the design to meet the rigid demands imposed by modern marine practice, the Foster Wheeler

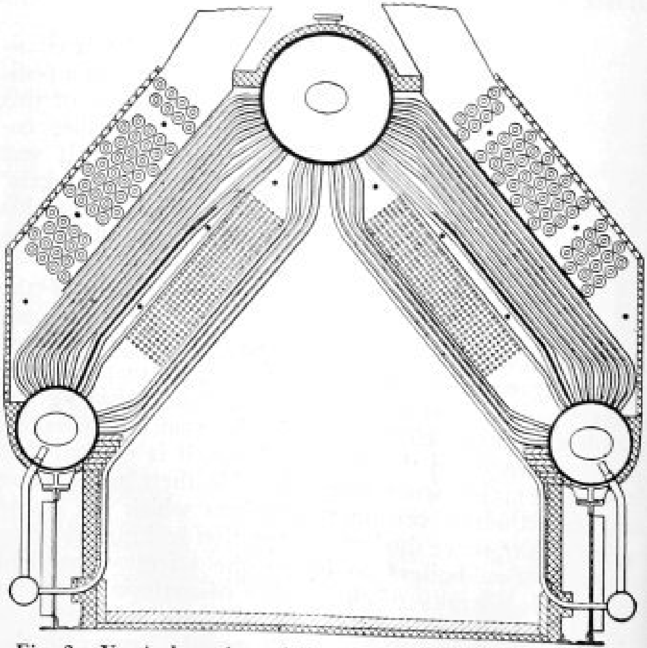


Fig. 2.—Vertical section of Foster Wheeler steam generator

Corporation departed from the tradition that furnace volume in which to burn fuel was relatively unimportant, and so was usually the last item considered. A compromise was reached between this practice and the recent trend in stationary steam-generator design, in which the furnace volume is first fixed and then the heat-absorbing elements suitably arranged above. The full principle of stationary steam-generator design could not be applied; for the space limitations in a ship, both as to boiler head-room and floor area, require a minimum size generator of maximum capacity to conform with these conditions.

As originally designed, each vessel called for six boilers, but it was decided to decrease the number to four and to increase the capacity of each unit. If a saving of weight and space were to be accomplished, the problem required solution by the development of a type different than the usual cross-drum boiler, and so the present three-drum bent-tube "A" design was evolved. Whereas the original design called for boilers of approximately 470 cubic feet of furnace volume each, the new design has a furnace volume of 1100 cubic feet. By efficient arrangement of heat-absorbing elements above the furnace, the head-room required by the present design is no greater than for the original boilers of the header type. The floor area is of course considerably less.

The capacity of the new generator is sufficient to maintain steam for full-speed service on three of the four units. An excellent idea of the comparative sizes of the installation originally planned and the present one may be gained from the fact that the four boilers of the new design occupy 19,620 cubic feet of volume as compared with 31,836 cubic feet for the original six header-type boilers.

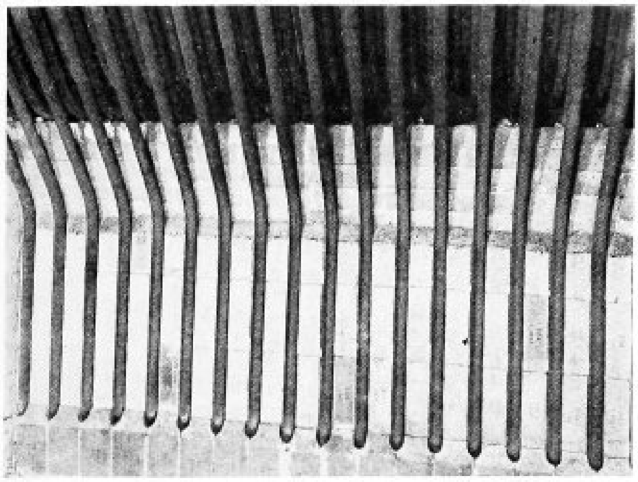


Fig. 3.—Water wall and brickwork after test

In some detail, the features of the generator are that the furnace is water-cooled on the sides while the rear wall and front wall are of special tile construction cooled by the circulation of the air required for combustion. Fig. 2 is a vertical section of the generator showing the furnace, water walls, boiler tubes, superheater and economizer sections. The baffles of steel plate are shown in solid black and the soot blower elements are represented by black dots. The water wall circulators take water from near the center of the mud drums.

Unlike the practice of utilizing combined forced and induced draft, as is customary on most high-efficiency boilers, the new generator employs forced draft with

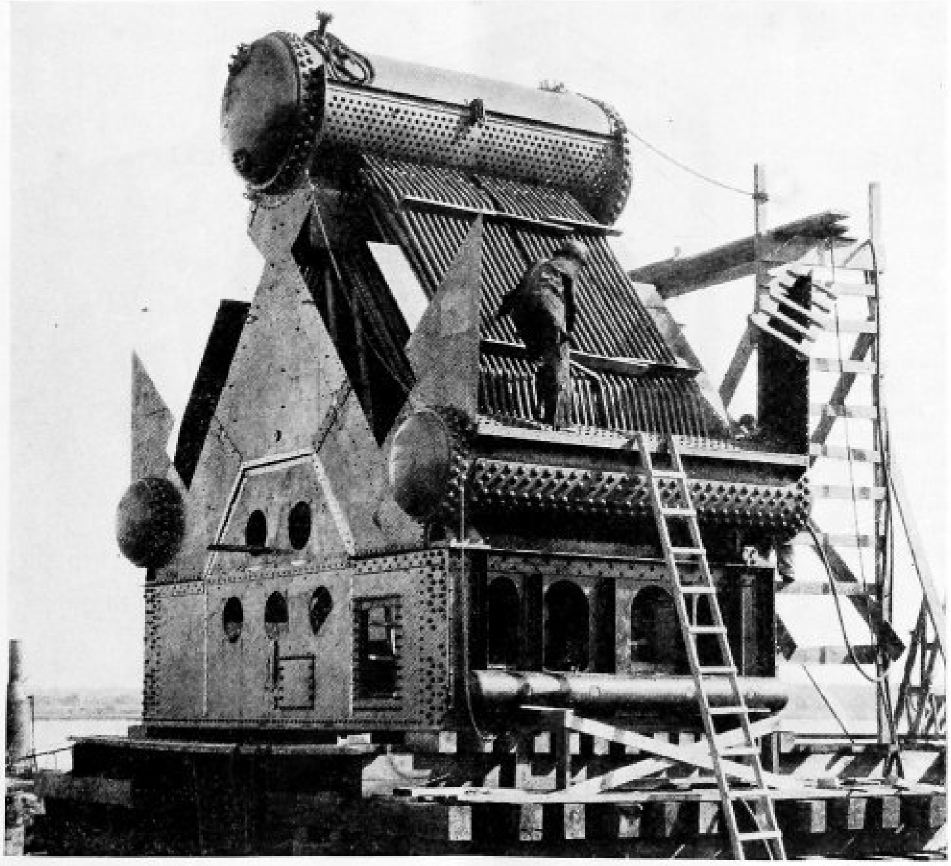
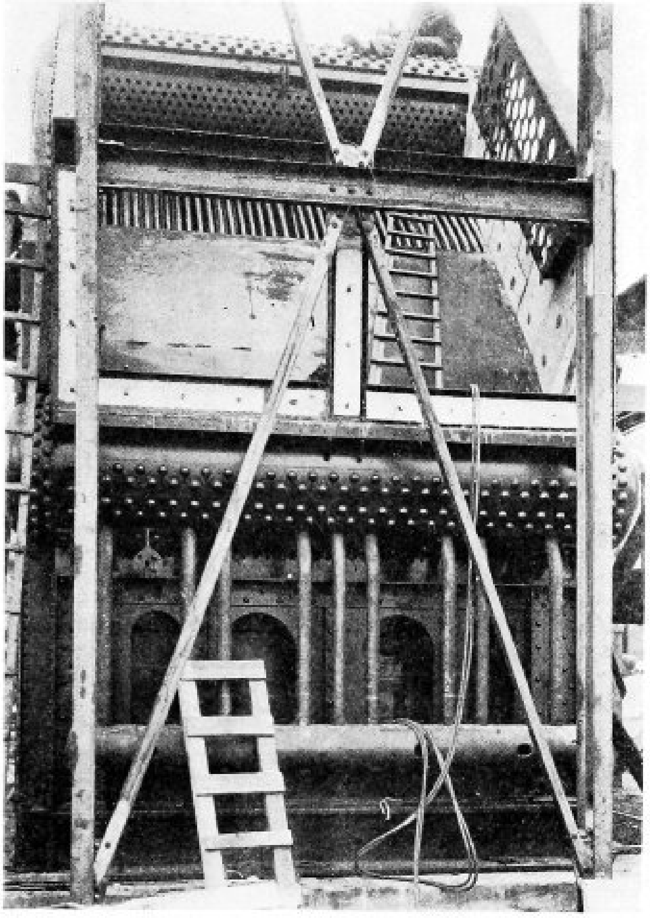


Fig. 4.—View of generator with drums in place



pressure in the furnace. This feature was made possible by air encasing the boiler; the pressure at the rear wall under the floor and the front wall being higher than that in the furnace. In operation, a blower forces air into this casing, or jacket, at the top of the rear wall, flowing downward over the rear wall and under the floor of the furnace to the front and up over the front wall to the burners. This air casing serves to conserve heat losses which would ordinarily occur through the boiler casing.

The main banks of tubes are 1½ inches diameter, the last pass of the boiler being made up of nine rows of 1½-inch tubes. In the case of the first rows comprising the water wall and the second row from the furnace, 2-inch diameter tubes are used. In front of the main banks of boiler tubes, the water wall tubes connect to the bottom of the steam drum and extend downward on either side of the furnace. Where they would ordinarily be expected to connect to the mud drums they turn vertically downward inside until reaching the floor of the furnace, thence passing out through the side walls and connecting to suitable headers. These headers are supplied with water by means of 3¼-inch diameter tubes extending into the mud drums.

For the side walls, firebrick tile is fitted behind the watertubes and is thus prevented from falling out of place. The front and rear walls of the furnace are vertical so the problem of retaining firebrick in these locations was a serious one. For this service firebrick tile of a special design has been developed to be self-supporting under the strenuous conditions experienced in marine boiler service. These tiles of interlocking design are shaped to fit over high-temperature-resisting metal clips, so that each tile is held securely in place.

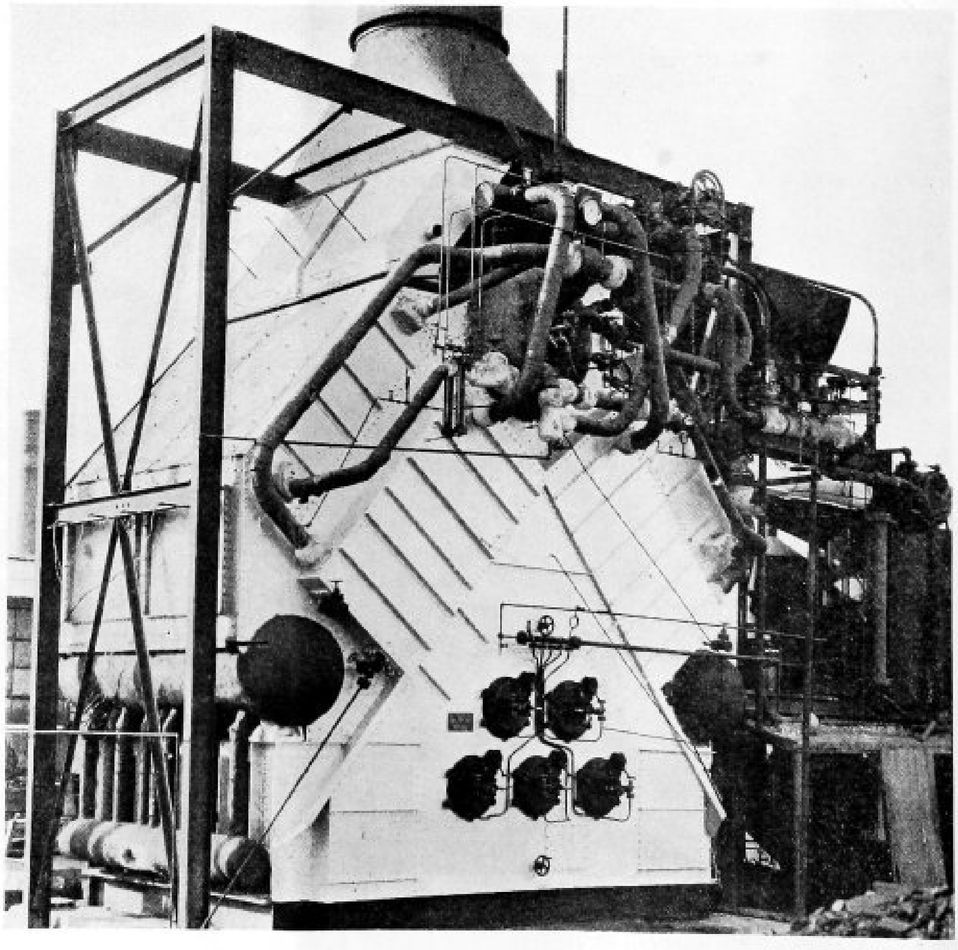


Fig. 5.—Above is a side view of the generator during construction, showing drums, lower tube bank, baffle plate, mud drum tube connections, magnesia block insulation and the like. Fig. 6.—At the right is the completed generator as it appeared when set up for testing

Insulation is applied as backing for the tile, the entire wall being secured by means of the clips to the casing.

During the study of preliminary designs 24-inch diameter mud drums and 1¼-inch diameter boiler tubes were considered. It was finally decided, however, to use 30-inch mud drums and 1½-inch tubes, the moderate increases in weight and cost being more than offset by operating advantages. One of the most important of these is the provision of a better manhole head construction in case of the 30-inch drums which gives ample room for access. The decision to use 1½-inch tubes was reached because they may be more effectively cleaned than the smaller size. Steadier steaming conditions also result through the greater water storage provided in the generator. It was evident during the conducting of the tests that the large drums were of considerable advantage, since it was necessary to administer a large amount of chemical treatment to raw water from the city mains used for boiler feed in preventing the formation of scale in the tubes. As might be expected, when the drums were opened during the shut-down periods of the boiler, an accumulation of sludge was found in the bottom of the mud drums. With the smaller drums this accumulation very likely would have been carried into the tubes.

In the economizers the heating surfaces are of 2-inch seamless-steel tubing on which has been shrunk cast-iron extended-surface gilled rings. Almost six times the heat-absorbing surface is provided by this form in contrast with the plain tube. Return headers are of forged steel. Handhole plates are located opposite each tube connection. The economizers, while occupying a comparatively small space, provide a large heat-absorbing area. They operate upon a positive counter flow of water and gases, the water passing down through the economizer while the gases go upward to the stack.

The illustrations indicate some of the more important construction features, Fig. 1 showing a partially assembled generator from which an excellent idea of the steel casing construction, the location of drums, superheater and economizer may be gained. The casing is entirely airtight, asbestos gaskets being used at all air-casing joints. This feature is of extreme importance because under the system of forced draft which is employed, the air in the casing is at a higher pressure than that in the furnace. In this view the superheater tube sheets will be seen to have holes at the left side and slots at the right hand side through which they may be removed. The tubes are rolled into the economizer tube sheets in the small holes at the left, while large holes at the front permit the removal of the economizer elements which, as described, are of the extended-surface, gilled-ring type.

Another method of erection is illustrated in Fig. 4 in which the three drums are in position. The upper drum is first held in place by a removable jig, which is used to establish accurate alinement. The casing is then built up to receive the drum, which is fitted into its support. As soon as the proper alinement has been established, the jig is removed. All boiler tubes in the generator are bent, as will be noted in this illustration, no straight tubes being used. The tubes also enter the drums radially. Below and to the right of the oil-burner openings an access door is indicated. The removal of the casing plate at this point permits ready entry to the furnace through the opening without disturbing the oil burners.

Fig. 5 is a side view of the boiler during construction, showing the steam drum at the top with boiler tube connections. At the bottom will be noted the mud drum with circulating tube connections to the water wall header. The construction of the drums with double-riveted butt joints, in which 1 15/32-inch steel rivets are used is of interest. Magnesia block insulation is applied in all side channels of the casing structure.

The generator as it appeared during tests at the Carteret, N. J., plant of the Foster Wheeler Corporation is shown in Fig. 6 with complete thermocouples, gas sampling and pressure-gage connections in operating condition. One particular feature is demonstrated in this illustration, namely, the compact and readily accessible arrangement of piping. This is an extremely important feature in marine installations, where the ability to tighten and service each pipe joint and valve connection is essential.

Two Explosions from Ineffective Safety Valves*

A safety valve in which the spring had been screwed down so far that the coils were tightly compressed led recently to a boiler explosion that caused considerable damage at a laundry in a mid-western city. It is supposed that someone screwed the valve down in order to apply a hydrostatic test to the boiler, and then forgot to reset the valve before again placing the vessel in service.

An examination made after the accident disclosed no excessive corrosion, or old cracks; in fact, the vessel appeared to have been in good condition and entirely suitable for the approved working pressure of 50 pounds. There seems to be no question but that the accident would not have occurred had the safety valve been able to function.

The boiler was of the locomotive type. Although no one was in the boiler room at the time, the blast injured one man and showered debris on five other persons who happened to be in another part of the building. The front of the building was blown out and a store next door was considerably damaged.

A safety valve in which the spring had not been screwed down but which in another way had been made just as ineffective as the one mentioned above, figured in still another laundry explosion that occurred a few weeks ago in Texas. In this case the safety valve had been installed with a stop valve between it and the boiler. The owner intended, of course, that this valve should be kept open. In some way it became closed with the result that the locomotive-type boiler dropped its crown sheet, tore in two at the throat sheet, and traveled abruptly up the main street of the town. The barrel shot forward 300 feet to cut down two gasoline pumps in front of a filling station. The firebox end was hurled backward against a telephone pole. Nothing was left of the boiler house.

After the accident it was found that the safety valve spring had corroded almost in two. Thus weakened, the spring probably allowed the valve to leak at a pressure somewhat less than the 85 pounds at which it was set. The supposition is that in order to lessen the annoyance someone closed the stop valve which, although it should never have been there, offered a convenient way of stopping the leak.

Even though we can think of no good reason why anyone would want to place a valve of any sort between a safety valve and the boiler, the insurance company's inspectors often find installations with that fault.

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* Published through the courtesy of *The Locomotive*, of the Hartford Steam Boiler Inspection and Insurance Company.

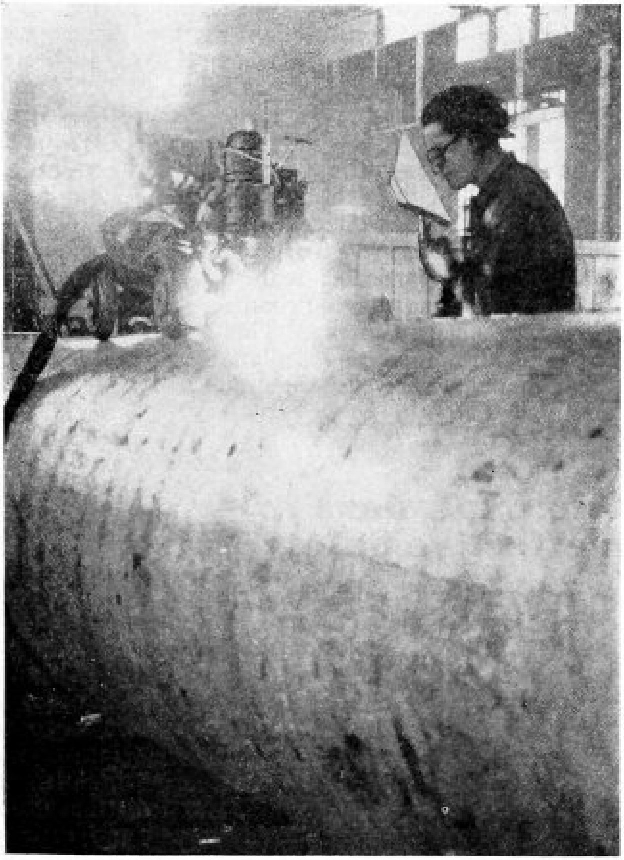


Fig. 1.—Self-propelled tractor-type electronic-tornado welder

Automatic-Arc Welding*

By G. G. Landis†

Where continuous similarity of the welding operation is specified by the demands of mass production, it is customary to use automatic-arc welding equipment. Manual welding allows a mobility and versatility on the part of the operator which cannot be rivaled by even the best automatic machines. The manual process, however, cannot hope to match the automatic method for speed and uniformity of quality in high-speed mass production.

The main element in automatic welding equipment, is of course, the welding head. This may be mounted on various set-ups or may be used with a movable tractor. There are two types of automatic welding heads. One feeds a wire electrode over the line of fusion and is merely the metallic-arc process of welding fixed to a feeder attachment, which allows the use of wire as electrode.

Carbon-arc welding has always proven very adaptable to automatic welding and one process in particular, the electronic tornado, possesses certain features which must be mentioned. An electronic tornado tractor mounting is shown in Fig. 1. The carbon is surrounded by a magnetic field which focuses the arc on the line of fusion. The head is water-cooled and the carbon is fed automatically so that the proper arc length is held at all times.

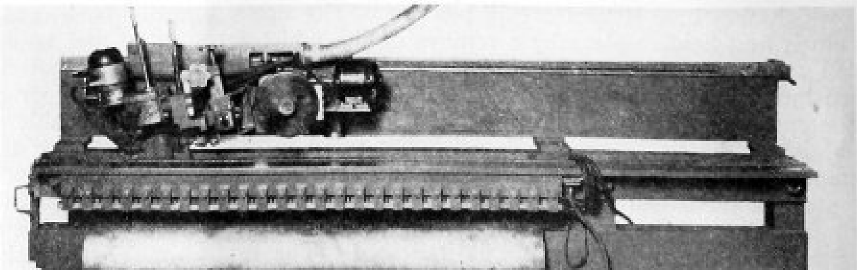
The spool on top of the tractor contains a fibrous autogenizer, which, fed into the arc, burns and gives off a protective gas, shielding the arc from the harmful effects of the air. This tractor is self-propelled and may be adjusted to any speed of travel in any horizontal direction.

With such a process, welds of 65,000 to 80,000 pounds per square inch tensile strength, extreme ductility and resistance to corrosion may be secured on metal from 18 gage to ¾ inch in thickness.

With the carbon-arc method of automatic welding, wire-feeding attachments are available, but the operations are simplified if the work can be prepared to provide additional metal at the line of fusion. On light gage material the edges can be up-struck or coined to meet the arc, and on heavy material it is customary to insert a filler strip between the butted edges of the metal. On most lap joints and edge joints this is not even a consideration. For certain applications a special mounting which feeds both the autogenizer and additional metal (in the form of flat wire) into the arc, is used.

In the production of range boilers automatic welding has become widely accepted. There is no comparison with riveting in this field as the use of welding allows the seams to be made in three operations. Fig. 2 shows a longitudinal seam-welding unit. Boilers up to 65

Fig. 2.—Side-seam mounting for welding range boilers automatically



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* Third article of series dealing with modern arc welding of tanks and pressure vessels.

† Chief engineer, The Lincoln Electric Company, Cleveland, O.

Welding Speeds with Electronic Tornado Process

Butt Joints

Thickness of Metal	18 ga.	16 ga.	14 ga.	12 ga.	10 ga.	$\frac{3}{16}$ inch
Welding Speed, Feet per Hour	170A	160A	150A	110A	95A	75A
Thickness of Metal	$\frac{1}{4}$ inch	$\frac{5}{16}$ inch	$\frac{3}{8}$ inch	$\frac{1}{2}$ inch	$\frac{5}{8}$ inch	$\frac{3}{4}$ inch
Welding Speed, Feet per Hour	60A	45A	35B	25B	20B	12B

Lap Joints

Thickness of Metal	16 ga.	14 ga.	12 ga.	10 ga.	$\frac{3}{16}$ inch	$\frac{1}{4}$ inch	$\frac{5}{16}$ inch	$\frac{3}{8}$ inch
Welding Speed, Feet per Hour	160	150	110	100	80	65	50	40

Edge Welds

Thickness of Metal	18 ga.	16 ga.	14 ga.	12 ga.
Welding Speed, Feet per Hour	200	180	150	125

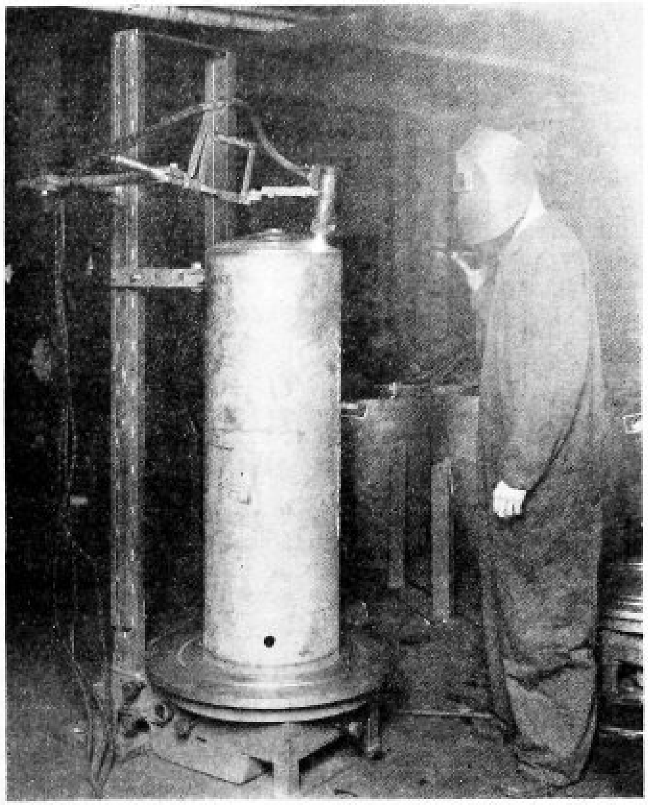
A—Welded one side with built-up bead on one side.
B—Welded from two sides with built-up bead on two sides.

inches long and of any diameter over 10 inches may be handled on this machine. The plates from 14 gage to $\frac{1}{4}$ inch in thickness can be either butt or lap welded on this unit. For butt welding, 14, 12 and 10 gage material, one edge should be turned up to provide additional material at the line of fusion. Heavier plate is fused together with a filler strip.

Head and bottom welding should be done on one fixture with a semi-automatic welding head. Fig. 3 shows such a mounting in action. In this case the operator guides the arc with a hand lever and speed of welding is maintained. The procedure for welding of heads and bottoms should be determined largely by the method of fitting the head to the side shell, the only factor being speed, as automatic welding is successful on all designs.

The fixture illustrated in Fig. 3 may be reversed to weld the bottoms on a tank. In this case, it is inclined in order to keep the weld horizontal, the bottom being fitted so that a lap weld is required. On a 60-inch range boiler 12 inches in diameter of 12 or 14 gage metal the average production speed is $12\frac{1}{2}$ units per hour for longitudinal seams. The welding time for both top and bottom seams is 90 seconds each.

Larger tanks can also be fabricated by the electronic



tornado. A typical installation is shown in Fig. 4. This equipment consists of rolls for revolving the tank with an arm for carrying the head. This arm can be raised or lowered to suit the diameter of the tank. Automatic travel is provided for the head when it is used to weld longitudinal seams. Thus, both circumferential and straight seams are welded on the same fixture.

On this tank all joints are designed for lap welding and on a unit 8 feet in diameter, 20 feet long, built of $\frac{3}{16}$-inch plate, welds are made at the rate of 11 inches per minute or 55 lineal feet per hour. On a similar unit day-in and day-out welding time on 550-gallon

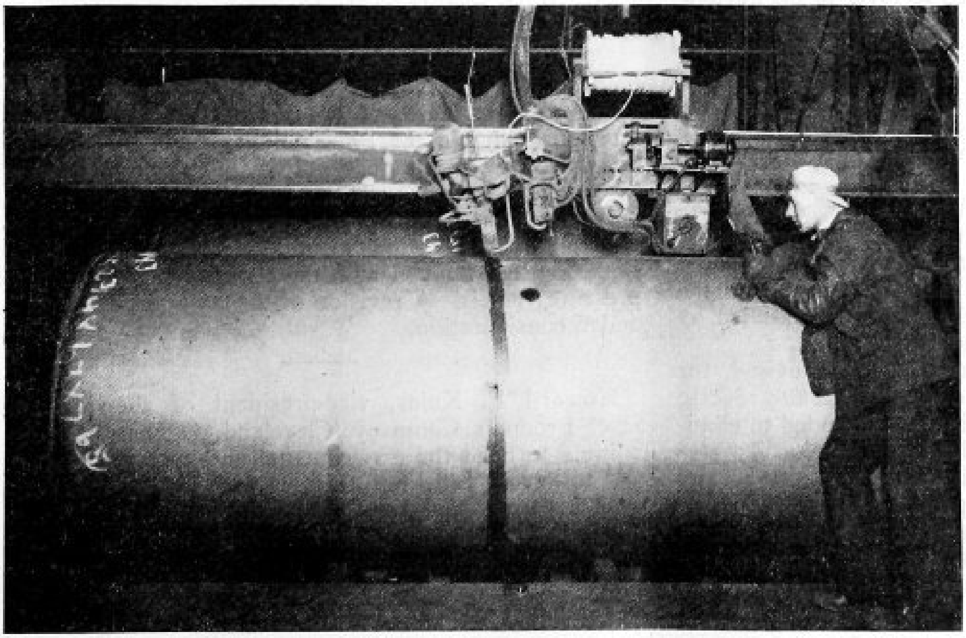


Fig. 3.—(Top) Semi-automatic fixture for fusing heads and bottoms in range boilers. Fig. 4. — (Left) Automatic welder for circumferential and longitudinal seams on large tanks

tanks made of $\frac{3}{16}$-inch plate was 16 minutes for the 31½ feet of welding required on each unit.

Reviewing the process of welding we find the major efficiency, "greater strength at lower cost" has determined its use in many boiler and tank shops. Butt welds so made are considered stronger than the metal which they fuse together, giving an absolutely leakproof construction which is highly resistant to shock and corrosion. A test of any fabrication process is whether or not it improves the product. Automatic-arc welding certainly does that.

(To be continued)

Work of the A.S.M.E. Boiler Code Committee

The Boiler Code Committee meets monthly for the purpose of considering communications relative to the Boiler Code. Any one desiring information as to the application of the code is requested to communicate with the secretary of the committee, 29 West 39th Street, New York, N. Y.

The procedure of the committee in handling the cases is as follows: All inquiries must be in written form before they are accepted for consideration. Copies are sent by the secretary of the committee to all of the members of the committee. The interpretation, in the form of a reply is then prepared by the committee and passed upon at a regular meeting of the committee. This interpretation is later submitted to the council of The American Society of Mechanical Engineers for approval, after which it is issued to the inquirer and published.

Below are given records of the interpretations of the committee in Cases Nos. 675 (Reopened), 693, 699, 701, 702 and 703, as formulated at the meeting on December 4, 1931, all having been approved by the council. In accordance with established practice, names of inquirers have been omitted.

Case No. 675 (Reopened).—*Inquiry:* Does the reply to section *a* of Case No. 661 permit the use of fusion welding of plain and wholly unstayed boiler furnaces under the conditions set forth?

Reply: The reply is intended to refer to the particular furnace under consideration and does not apply to plain and wholly unstayed boiler furnaces. If the welds are stress relieved and bend tests are made of the welds as specified in the welding rules, it would be sufficient for the particular joint in question which is under compression. A revision to cover welds under compression is contemplated.

Case No. 693.—*Inquiry:* When the carbon-arc process of welding is used, is it necessary, in making double-welded butt joints without V'ing or scarfing the plates, to chip, grind, or melt out the metal at the base of the weld from the second side of the plate to be welded, as is required by Par. U-72 revised?

Reply: The requirements in Par. U-72 revised, for chipping, grinding, or melting out metal on the reverse side of a double-welded butt joint, are intended to eliminate impurities and obtain a clean surface for the second side welded, and are not intended to apply to any process of welding by which proper fusion and penetration are otherwise obtained and no impurities remain at the base of the weld. The same qualification test specimens and test requirements must be made as for the other methods of welding.

Case No. 699.—*Inquiry:* What is the maximum length of a pipe connection for a steam gage before it shall be classed as a "long pipe" mentioned in the last section of Par. P-296?

Reply: It is the opinion of the committee that a length of pipe longer than 10 feet should be classed as "long" within the intent of Par. P-296.

Case No. 701.—*Inquiry:* An interpretation is requested of the requirements for test plates for longitudinal joints in Par. U-68 of the code as follows:

a Can these test pieces be cut from any portion of the plate as rolled?

b Can these test pieces be cut longitudinally or transversely to the rolling direction of the plate as best suits the mill's convenience?

c When two or more plates are ordered of the same thickness, can the tests for all of the plates be cut from one plate as rolled?

d Should fabricators specify the size of test pieces required on the assumption that different shops might not have the same methods for cutting the test pieces into various sections required?

Reply: a, b, and *c.* It is the opinion of the committee that the plates for test samples may, under the requirements of Par. U-68, be taken from any part of one or more plates that conform to the specifications for the material that is used in the fabrication of the welded vessels, and without reference to the direction of the mill rolling.

d The sizes of test specimens as given in the code will determine the size of the test plates from which the test specimens are to be cut, which sizes would presumably be given by the purchaser.

Case No. 702.—*Inquiry:* Under the new rules for fusion welding, what is the tolerance for the corner radius of the specimen for free bend tests? This corner radius is specified as 1/10 *t*.

Reply: It is the opinion of the committee that the corners should be rounded to a radius not to exceed 1/10 *t*. This measurement was intended as a maximum and not a mandatory fixed amount.

Case No. 703.—*Inquiry:* Is it necessary, in the construction of an ammonia condenser in which non-ferrous tubes are used, to follow the requirements for tubes in Table P-5 of the Power Boiler Code? The tubing can readily be made to meet the requirements of this table, but the question is raised as to why the Power Boiler Rules should apply to constructions covered by the Code for Unfired Pressure Vessels?

Reply: Where there are no rules in the Code for Unfired Pressure Vessels to cover a particular part of a construction, the committee would recommend that if there are rules in the Power Boiler Code which cover this part, they should apply. It is further the opinion of the committee that such a vessel, in which all of the parts meet the requirements of the Code for Unfired Pressure Vessels supplemented by the Power Boiler Code, can be stamped with the code symbol for unfired pressure vessels. A revision of the code to this effect is under consideration.

Howard F. Kulas, vice-president of the Midland Steel Products Company, Cleveland, O., has been appointed to head the experimental and research division and to devote full time to its work. The company has been active in development work in recent years with resulting advances and improvements in its various products. At its Cleveland plant the Midland Company has one of the most modern industrial research laboratories in the country.

What Can the Boiler Department Do to Reduce Pitting and Corrosion?

In the January issue a committee report on this subject, as prepared for the 1931 proceedings of the Master Boiler Makers' Association, was published. Contributing to the discussion, water service engineers of leading railroads have reviewed the practical steps taken to combat the ravages of this boiler disease. The articles on this subject published in this issue denote considerable progress in developing corrosion and pitting preventatives. Further articles on the subject will appear later

As the first to contribute to the discussion of pitting and corrosion in the 1931 proceedings of the Master Boiler Makers' Association, R. C. Bardwell, superintendent, Water Supply of the Chesapeake & Ohio Railway Company, outlined the subject as follows:

The slow progress of pitting in locomotive boilers which usually extends over a period of many months or even years, is a serious handicap to a careful and unbiased investigation as to the cause, and also delays definite determination as to effectiveness of remedies applied. During such periods it is rare that the above water quality does not change either due to variation at the source or in the amount used from the different stations. Also, the work done by the locomotive and changes in tonnage conditions will vary, all of which makes comparisons somewhat uncertain. Also, the average railway organization is not usually equipped to follow up extended tests of this character.

It is therefore believed well to add a word of caution against jumping to conclusions which may not be borne out by later developments. Continued observations and records are essential. The application of methods and treatment already reported are subject to a much wider use than now in effect.

Although any one individual case of pitting and corrosion may be due to a variety of causes, it is believed that possibly the most applicable remedy, which will produce favorable results in the majority of cases is the over treatment of the water to maintain the predetermined safe alkalinity at all times. This can be controlled by simple tests regardless of whether the internal or external methods of treatment are used. Such conditions result in reducing the hydrogen ion concentration and nullifying the action of those ferocious little molecules which have been so often described by Dr. Koyl at previous conventions. Objection is sometimes made to this treatment due to the complaint that foaming is increased which directly affects engine performance. It is true that such foaming will aggravate the foaming tendency due to loosening of old scale particles and increasing dissolved solids but by use of the blow-off cock this situation can be controlled and such blowing will result in cleaner boilers, lower fuel consumption, and less maintenance.

My answer to what can best be done by the Boiler Department to relieve pitting and corrosion conditions is to work and co-operate closely with the water chemists, calling their attention promptly to any trouble and insisting that it be taken care of to improve the situation without delay wherever the water quality is subject to correction, and following up the blow-off instructions recommended to insure that they are carefully carried out. I have had the pleasure and honor of working with some very high type members of your association during the past twenty-two years and it is believed that we were jointly able to effect very marked improvement by using the methods which have been found to be most serviceable.

Following Mr. Bardwell's review in the proceedings is one by R. E. Coughlan, supervisor of water supply, Chicago and Northwestern Railway Company, which follows:

During the past year the studies of boiler corrosion and pitting have shown a decided tendency to specialize along the lines of the individual problems which all students have agreed are wholly or in part responsible for this trouble. These studies are now progressing under the guidance of trained research workers either in the employ of the various railroads or at the various universities, thus combining technical knowledge with practical application. The subject is one of such vast proportions that progress must necessarily be slow, as technical solution without practicability is of no value. For this reason we find each group of workers along the various methods of solutions, patiently devising the necessary facilities and methods that in their opinion will aid in the elimination of this problem. The methods devised and proven must be such that not only can results be duplicated but must also be of practical application.

The decided difference in the operation of a locomotive boiler as compared to a stationary boiler of an industrial plant is accepted by all but still it is just one of the difficulties that is encountered when laboratory methods are tried on a locomotive boiler under operating conditions, as the factors of operation and the work allotted to such a boiler leave very little time or opportunity to do very much research work.

We have all heard of the excellent results obtained by the elimination of the greater percentage of oxygen, but still we also have observed the poor results of the same method in other cases. We also have seen the excellent results of the use of excess chemical treatment where the railroad is equipped with facilities to put such a practice into effect but we have also seen corrosion and pitting occur in water of a similar nature, showing decidedly that some other factor is also the cause.

It has been my privilege during the past year to observe a case of almost complete elimination of pitting and corrosion by means of the counter electrical device on a yard engine using water carrying excess treatment, where excess treatment alone does not stop corrosion. It has also come to my attention of another installation of this device using water from a lime and soda ash plant where the corrosion has been about 90 percent eliminated, even though the water supplied also contains an excess of chemical treatment.

There is no doubt that this problem of corrosion is due to a combination of factors so that the methods applicable to one district are not of much value in another location. The open feed-water heater will help in preventing pitting and corrosion if the open feed-water is functioning 100 percent. In order to reach this degree of efficiency, it requires the complete co-operation of the engine crews, round-house forces, and supervising offices. If any of these factors are missing, the feed-water heater is no better than many of the devices in the past which have now been removed from the locomotives. Excess treatment will also help, but excess treatment means almost 100 percent operation of water softening plants, education of engine crews in use of light water, and co-operation of operating departments in sympathy with the best interests of the railroad companies. If any of these factors are lacking, excess treatment means a succession of foaming complaints, increased engine-house expense and general dissatisfaction among the mechanical men responsible for the operation of the power. The counter electrical device will be of great help where it is properly operated and maintained.

In my opinion, no matter what devices or methods are finally developed in the solution of boiler corrosion and pitting, this problem will always be a joint one of the mechanical department and the technical men in the employ of the railroads.

The subject of tender corrosion in many cases is now successfully handled by means of protective coatings as the temperature being the temperature of the surrounding atmosphere and sometimes below this temperature, no chemical reaction is taking place in the water, so that it is mainly a matter of rust prevention on this material. Various forms of petroleum products, or asphaltic compounds seem to retard rusting in a satisfactory manner.

On this subject C. H. Koyl, engineer water service, C., M., St. P. & P. R. R., contributed the following:

Practically all that is known of the causes of boiler corrosion and means for its prevention is stated or outlined in the committee report, and the paragraphs on materials, care in storage, design and construction, and inspection, tell the part that the boiler maker can play in the preservation of the boiler.

All that I can do is to re-state the argument in slightly different terms.

The prevention of boiler corrosion is principally the job of the water engineer or water chemist. This is because in pure water, or properly treated water, the ordinary steel of commerce will not corrode enough to be noticeable, and the preparation of the water is comparatively cheap; whereas in bad water the only metals safe from corrosion are copper and some costly alloys, and the expense of constructing boilers of such material is out of the question.

As a matter of fact, the purest iron that can be made will dissolve slowly in the purest water that can be made, but the water cannot hold much iron in solution and soon becomes saturated, and the amount of iron dissolved is of no consequence unless the water holds free oxygen in solution (oxygen dissolved from the air) which, if present, combines with the dissolved iron to form iron oxide, plain iron rust, which is insoluble in water and settles to the bottom, and thus leaves room in the water for the solution of more iron.

When the steel or iron is not perfectly pure but contains on its surface the inequalities of chemical composition or physical condition mentioned in the committee's report, this tendency to solution is greater at some points than others and accounts for the pits on flues and for the deep rings on flues just back of the front flue-sheet where the metal has been strained by the expanding tool.

This solution of metals in water is really an electrical action and each atom of iron which leaves a flue to pass into the water carries its positive electric charge with it. But nature has so arranged that the positive and negative charges carried by the different ions dissolved in any given water are just equal in amount, and any iron ion (positively charged iron atom) can get into such water only by being able to force out some other positively charged atom (ion) which has a slightly less electromotive force or tendency to remain dissolved in the water.

And this is what gives the hydrogen ion its importance in the corrosion of iron. There are plenty of substances which have weaker solution tendencies than iron, but the only one found dissolved in ordinary boiler water is hydrogen, and therefore it is true in a general way that the pitting of iron or steel under water is in proportion to the number of hydrogen ions in the water.

This at once gives us the key to the prevention of pitting. If we can get all the hydrogen ions out of boiler water before it reaches the boiler, or if we can so fix the water that these ions cannot be forced out, we can prevent pitting because there is nothing else in ordinary water that can be forced out by iron, and therefore the iron cannot dissolve.

I will omit the chemical demonstration but it so happens that caustic soda in the water, or even soda-ash, (in amount according to the kind of water) will banish the hydrogen ions and prevent pitting. This method can be used only in soft water, but has been an effective preventive of pitting for years on the Great Northern Railway and on the Chicago, Milwaukee, St. Paul & Pacific, and doubtless has been used on other roads.

When the hydrogen ions are forced out of solution by the iron ions, they give up their electric charges to the parts of the boiler interior which are not being pitted and if this is prevented the hydrogen ions cannot escape, and pitting is ended. This is the object of the Gunderson method which coats the boiler interior with metallic arsenic, which keeps the hydrogen ions away from the iron.

Then there is a third method which is in use on the C., M., St. P. & P. R. R.

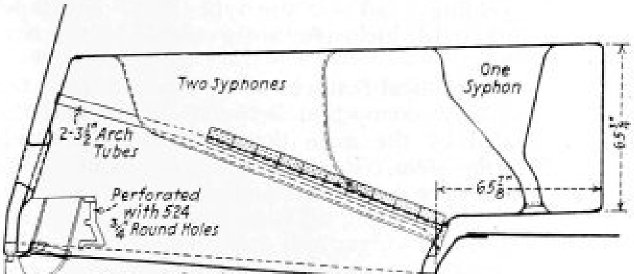
You will remember the statement that pitting is prevented by excluding the oxygen of the air from the water in the boiler, and this third method accomplishes this by passing all the water through an open feed-water heater attached to the side of the locomotive. Such a boiler on the worst pitting district of the C., M., St. P. & P. R. R. for four years had no pits and the flues were put back for another four years' work. The striking advantage of this method over any other is that while it prevents pitting it also saves 10 percent of the coal used in transportation.

In the next issue various phases of this important subject will be discussed by other experts among the water service engineers and boiler specialists, connected with railroads in widely separated parts of the country.

Santa Fe locomotive No. 5000 tested on the Pecos division

Santa Fe Locomotive Gives Excellent Performance Record

The Atchison, Topeka & Santa Fe received a modern 2-10-4 type locomotive, No. 5000, from the Baldwin Locomotive Works in December, 1930, and placed it in service at Clovis, N. M. Subsequently, during the months of July and August, 1931, this locomotive was tested with a dynamometer car in freight service on the Pecos division between Clovis, N. M., and Belen. Since Locomotive 5000 was the only one involved in this test, comparisons can be made only with previous tests of other locomotives. As compared with Santa Fe 2-10-2 type locomotives of the 3800 class, for example, tested in the summer of 1930, the new locomotive will handle approximately 15 percent more tonnage in 9 percent less time and with 17 percent less coal per 1000 gross-ton miles.



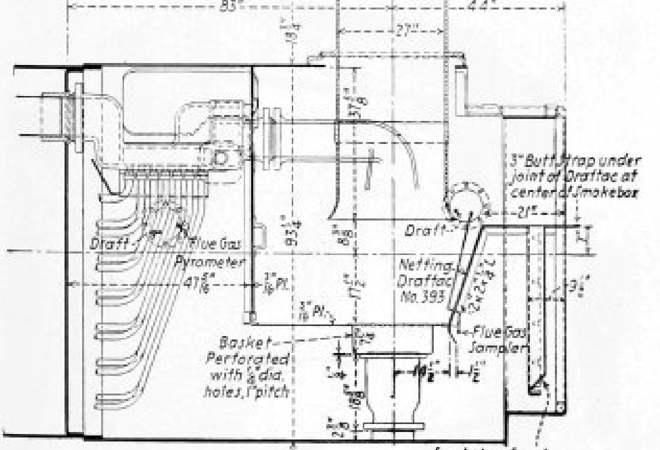
Cross section showing the large firebox and combustion chamber equipped with three Thermic Syphons

When compared with motive power which was considered strictly modern only about five years ago, therefore, the new Santa Fe 5000-class locomotive promises not only to effect important savings in fuel but to permit handling heavier train loads on shorter schedules. The resultant marked reduction in train-hours per ton handled, tendency to eliminate overtime, and minimizing labor and fuel costs due to delays will have a highly favorable effect upon operating expenses. In addition, the important objective will be achieved of giving better service to shippers.

Locomotive 5000 is designed with a greater ratio of boiler capacity to tractive power than is generally used. It is of the 2-10-4 type and carries 300 pounds per square inch boiler pressure.

This locomotive had an extraordinary capacity for sustained power at high speeds, which is reflected by power performance curves. It exerted a drawbar pull of 50,000 pounds at 33 miles per hour, equivalent to 4350 drawbar horsepower, at which it had a machine efficiency of 84 percent. With a drawbar pull of 82,500 pounds at 15 miles per hour, the machine efficiency was 90.0 percent.

The tractive force of the locomotive is calculated to be 93,000 pounds, which, with a weight on the drivers of 348,000 pounds, gives a factor of 3.75. The locomotive has shown over 93,000 pounds at the drawbar.



Front-end arrangement of locomotive 5000

Welded Bending Brakes

The Steelweld Machinery Company, Cleveland, O., manufacturers of the Battleship line of bulldozers, has developed a new line of rolled steel welded bending brakes. Following conventional design in brake and press manufacture, whereby power is applied through overhead eccentrics to the movable and adjustable ram, the machine is unique in certain details The housings, in place of being cast iron or steel castings, or cut from rolled plate, are of built-up welded sections. The stress members in the throat are very heavy and so arranged that deeper throats can, with safety, be supplied.

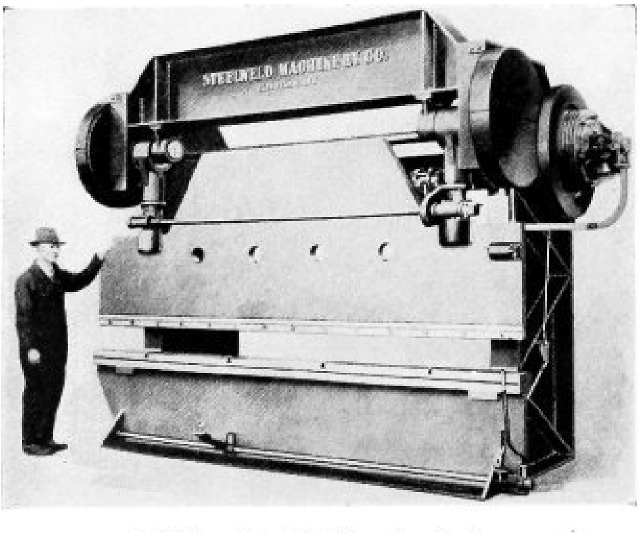
Wherever practical, the manufacturers prefer to ship the machine whole, except the bed, claiming thereby to eliminate bolts and rivets, and supply a simple monolithic all-welded frame of unusual strength and rigidity. In this machine, both the ram and bed are of ample depth and are overhung at one end, giving great advantage in jobbing work as well as longer die surface for narrow work.

The clutch is a twin disk. The operating levers move on ball bearings, eliminating much friction and reducing operator fatigue. It is claimed that the tool operates with unusual ease and speed. All the gears in the machine are steel, high-speed gears being of the Sykes herringbone type. Ample bearing area is provided on both sides of the eccentrics, giving a double bearing to each gear.

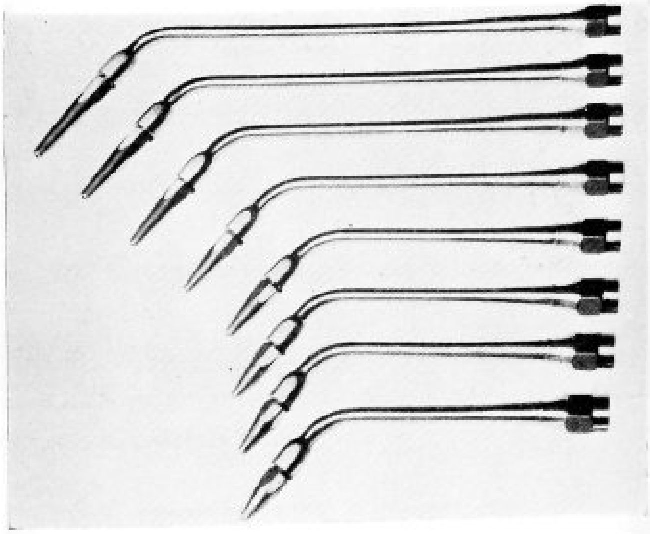
A feature of advantage is stressed by the manufacturers in the ball-joint connection, which is claimed to be unbreakable in service. No ordinary cast iron is used in the tool, the slide bearings and eccentric straps being of Meehanite iron castings. The flywheel is of rolled steel mounted on Timken bearings. Flywheel shaft is a high-carbon alloy steel with the pinion forged integral. Eccentrics are forged solid with the main shafts. Adjusting screws are heat-treated high-carbon steel.

The power elevation to the ram is electrically operated, self-locking and practically fool-proof. The motor is arranged to stall before damage occurs to mechanism. The manufacturers equip the machine with pressure grease lubrication from a central location.

Longer strokes, deeper throats, greater die clearances or width of bed and ram are readily obtainable, without delay in production.



Rolled-steel, welded bending brake



Welding heads with detachable tips

Welding Heads

A new series of welding heads with detachable tips has been introduced by The Linde Air Products Company, 30 East 42nd Street, New York, for use with the Oxweld type W-17 welding blowpipe. These will supplement the one-piece style heads for this blowpipe, so that the user will have a choice of two types of welding heads.

The new detachable tip heads were developed particularly for production operations. The new tip produces the same type of flame as that produced by the one-piece welding head for the type W-17 blowpipe. Sizes Nos. 6 to 13, inclusive, are available in the new welding heads.

One of the practical features of the new design is the very satisfactory connection between the tip and the stem provided by the male thread on the tip which screws into the stem. Removal or replacement of the welding tip is thus greatly facilitated.

Cromaloy Flux

Cromaloy flux, one of the important items in the line of Oxweld welding fluxes distributed by The Linde Air Products Company, 30 East 42nd Street, New York, has been developed especially for use in welding the chromium-containing alloys more generally known commercially as stainless steels or rustless irons. The ordinary fluxes used for welding or brazing are not satisfactory in welding stainless steel or rustless iron because they will not dissolve the infusible oxides, consisting chiefly of chromium oxide, which tend to form on the molten surface of these alloys. A satisfactory flux for use in welding these alloys must be sufficiently fireproof to protect the molten metal and hot metal adjacent to the weld from oxidation, and at the same time correctly compounded to dissolve the refractory chomium oxide with ease.

Because of its high solvent power for chromium oxide, and its high resistance to heat, Cromaloy flux is especially prepared for this type of work, and its use will insure best results in welding these special chromium-iron alloys.

Suggestions to Follow in Applying the New Welding Code ▲ ▲ ▲

By F. G. Sherbondy

The author, who is vice-president of The Biggs Boiler Works Company, Akron, O., writing in *The Welding Engineer* gives sane advice to companies engaged in the manufacture of pressure vessels by means of fusion welding. Since some concerns have determined through experience the best procedure to follow in production to meet new conditions imposed by welding, the companies not so far advanced in technique may save many arduous steps by profiting from their experience. The suggestions made by the author, covering the interpretation of the classes of welding permitted by the code, and the problem of stress relieving, come within the category of experience in pressure vessel production.

It is to be expected that the 1931 A. S. M. E. Boiler Code permitting as it does the construction without limitation of any type of pressure vessel, may present problems of classification and interpretation which are not always understood by the management of industries in general. Likewise, being the result of research and data supplied from so many different sources, the new code may contain conditions which, when combined under a uniform procedure, are likely to be misunderstood and misapplied.

To the industrial engineer preparing specifications for the construction of welded vessels, as well as to the management of industry in general, I should like to point to a feature that is of paramount importance. The classification of the accepted methods of welding in the 1931 code, under Classes 1, 2 and 3, is to be considered solely as—"A method of describing the recognized and accepted practices for welded construction" and *not* as a means of qualifying or grading respective types or methods of welding as to their *superiority* one over the other. For example—Certain definite types of vessels, or to be more exact, vessels of certain dimensions and for certain working conditions and pressures, are to be built under Class 1; certain other types of vessels are to be built in conformity with Class 2, and certain other types are to be built in conformity with Class 3 welding. These various classifications are the result of several years of painstaking research, countless tests and a collection of countless volumes of data covering the entire subject of welded vessel construction. From such efforts the classification referred to has developed. The result is that each class is distinctly within itself and is the recognized and accepted standard of construction for vessels of the type which fall into the respective classes.

There may exist a tendency, by some engineers who are responsible for the successful operation of various individual processing units, to place a capital burden on the management by not thoroughly analyzing the full meaning of the new code, and by arbitrarily specifying a type of construction entirely unnecessary for the particular type of vessels under consideration, feeling that in doing so they have played safe and relieved themselves of any individual responsibility. This theory, fortunately, is not often encountered, especially when dealing with engineers who are thoroughly conversant with welded construction and the details of code requirements.

There does exist, however, in the mind of an occasional engineer, the feeling that in order to be on the safe side, he must specify Class 1 construction, irrespective of conditions of service his individual unit is to operate under, whereas, as a matter of fact, Class 2 or possibly Class 3 construction would in all probability give him a higher degree of safety than he has ever enjoyed before on similar plant apparatus.

It is needless to say that such a policy is economically wrong and such an engineer is not being fair with his management or to his profession.

I doubt if any one subject that the code committee has had to consider was the subject of more discussion than that of stress relieving. Space will not permit details with reference to the various arguments pro and con. Perhaps it is sufficient to say that if a theory exists that stress relieving is a cure-all for all welding ills, the feature is very much overrated and its benefits are very much misunderstood. Stress relieving, like many other requirements of the specifications is to be recognized or perhaps it would be more exact to state that the *construction* of vessels that are free from measurable or objectionable stresses is a very important factor, comparable with ductility, uniformity and strength. However, the specifications applying to stress relieving for Class 1 vessels, as well as their application to a limited extent, to Class 2 vessels, is the result of a compromise from the original thoughts on the subject; the compromise no doubt being largely effected by the favorable experience records of the products of certain welding procedures, wherein thermal stress relieving was *not* employed.

Thermal stress relieving, as required by the code, does not, nor was its function ever intended to improve the tensile strength, ductility or soundness of the weld. Its requirements under the code are, I feel, due largely to a broad interpretation of welding, on account of the fact that individual ability or procedures cannot be recognized. Stress relieving on vessels of extremely heavy wall thickness, however, is undoubtedly a good practice, if not an essential requirement.

It has, however, been the experience of our company and many others that on vessels constructed of plates up to 1½ inches in thickness, thermal stress relieving is unnecessary and is regarded as an economic waste and this contention is shared by leading insurance companies who have had a background of several years' experience with vessels built under such welding procedures. On the other hand, other welding procedures may lock up stresses that must be thermally relieved, so the rules adopted by the code committee are justifiable for they naturally apply to the composite picture. However, in no case should thermal stress relieving be considered necessary in the production of welded vessels beyond the code requirements. Or, if employed such vessels should not be considered superior to vessels constructed under a procedure wherein the procedure itself is so controlled that measurable stresses are not originally set up.

It should be the object of every manufacturer of welded pressure vessels to protect his investment as well as the position that has been accorded welded construction, by constantly demanding the full and complete observance of the fundamentals of good welding. It should likewise be the duty of industries in purchasing welded equipment to inquire into the individual ability of the manufacturer, his experience and record of performance. The buyer should further earnestly consider the type of service in which the vessels under consideration are to be used and be guided by the proper interpretation of the A. S. M. E. code as to specifications.

Our company has been manufacturing welded high-pressure vessels for several years for the various process industries and the continuous and satisfactory operation of such units prompts me to say that notwithstanding the strides that have already been made in welded construction, it is still in its infancy; but it is essential that the construction must not be unduly handicapped by improper application of fundamentals.

Welded construction has definitely proven to be superior to other types of construction, and while in some instances welded vessels may prove to be slightly lower in initial cost, due to the saving of material involved, I think this is entirely the wrong attitude to hold with respect to this class of construction. The saving to be effected due to decreased maintenance, general utility, prevention of loss in production due to repairs, certainly favors welded construction as to ultimate cost, irrespective of its initial or first cost.

Staybolt Problems

By George M. Davies

The following questions have been submitted by a reader for consideration:

(1) What methods and machines are used for producing radial combination bolts with the threads in the tram?
(2) What methods are used for putting the threads in the tram of bolts that are threaded on ordinary bolt cutters?
(3) What are some efficient methods of properly installing radial combination bolts in locomotive boilers?

The answers to these questions are as follows:

A.—Radial staybolts are best threaded in a turret lathe especially designed for the purpose of threading staybolts; these lathes are provided with a lead screw to maintain an accurate thread lead.

Staybolt threading machines are equipped with dies of suitable shape for cutting the taper threads on the large end of the radial staybolts at the fit in the firebox crown sheet. An example would be a No. 4 universal turret lathe with staybolt attachments.

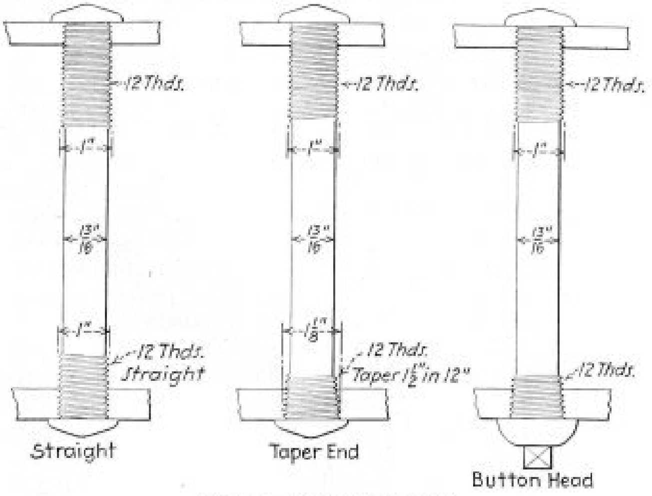


Fig. 1.—Radial staybolts

There are various types of radial staybolts such as straight-end, tapered end, buttonhead, and flexible staybolts. The flexible staybolt is most commonly used in locomotive boilers. However, this type of bolt is special, and does not enter into this problem. The straight-end, tapered-end, and buttonhead staybolts, illustrated in Fig. 1, are threaded in the same manner. The buttonhead staybolt, however, is used as an example in this case because of the additional operations due to the head.

The threading of the buttonhead radial bolt (or any other type) is well accomplished in a turret lathe by the following method: The body of the bolt is first gripped with the assistance of sectional brass sleeves, secured in the clutch of the machine. Then the two ends which require threading are turned to their proper dimensions. At this time, the under side of the buttonhead is faced

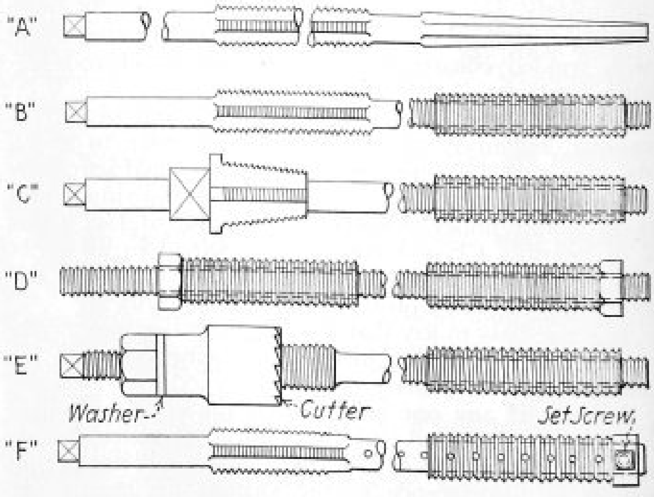


Fig. 2.—Taps and faring tools for radial staybolts

and grooved to prevent overhanging threads from interfering with the head when it is squeezed to a tight joint against the crown sheet. At the same time a short portion of the body of the bolt is trued to facilitate future gripping.

For the purpose of threading to lead, a specially arranged telescopic chuck, with dies complete, is fitted to the rear end of the turret; and another set of ordinary dies is secured in position in the front end of the turret. A master threaded bolt is then brought into use for the purpose of setting the telescopic dies to the proper lead and length. The set screws on the telescopic arrangement are then secured against movement, after which the bolt is secured by the chuck gripping the squared end of the bolt. According to the arrangement of the dies, the plain end of the bolt starts threading in the telescopic die. Before the plain end is completely threaded, the buttonhead end has also started, thus allowing the threading at the plain end of the bolt to be completely threaded before the opposite die has run up to the buttonhead, where it is prevented from injuring the facing on the under side of the head by the use of a geometric self-opening die-head.

Radial stays are the supporting stays which join the outside firebox of a boiler, which has a semicircular roof sheet, to an inside firebox crown sheet of any desired transverse shape or longitudinal slope.

They are usually upset at both ends, and always at one end, thus avoiding the necessity for threading their entire length. They are intended to enter and follow the threaded holes which have been prepared for them without stripping the threads either on the bolts or in the holes.

The art of good radial staybolting largely depends on the design which is developed on prints and supplied to the boiler shop; therefore, good results are expected from radial staybolts when the inside firebox crown-sheet holes are planned so that the bolts are at right angles to the inside sheet and in proper alinement with the holes of the outside sheet.

Practical experience has demonstrated that, to obtain immunity from leakage of radial stays at the crown-sheet end, the threads on the bolts should not run out. However, it is impossible to design a semicircular roof sheet and a different shaped crown sheet without such results. It is absolutely necessary that the crown sheet-holes be kept free from this objection; and the running out of threads should be confined to the roof sheet when the radial staybolts are not subject to the action of the ire.

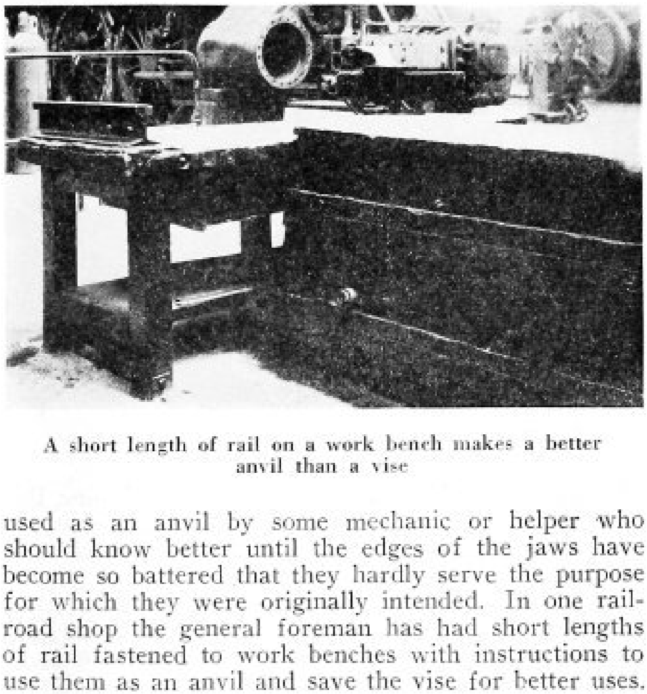
When tapping holes in new roof and crown sheets, the long tap *A*, Fig. 2, should be run through from the outside end with a motor, after which the crown sheet-end of such holes that are intended to receive button-head bolts should be faced with a facing tool *E*, Fig 2. This tool is provided with a spindle, introduced into the roof and crown-sheet hole, which guides the facing tool to produce a true surface on the crown sheet to correspond with the head of the bolt. If taper-end bolts are applied, the long tap *A*, Fig. 2, should be run through n the same way and followed up with the use of a suitable tap *C*, Fig. 2, at the crown-sheet end. A telescopic sleeve should be set to guide the tap with the roof-sheet hole so that all such holes may be tapped to the same size. It is essential that a controlling shoulder on the shank of the tap next to the motor be used.

When tapping holes in new crown sheets, where the old roof sheets are used, it is not necessary to run through the long tap which is used for new construction. Good substitutes are found in those taps having elescopic sleeves, as shown in *B*, *D*, and *F*, Fig. 2. These will tap the crown-sheet holes true to lead if correctly made and judiciously used. Suitable taps shown as *B* and *D*, Fig. 2, have sleeves which are threaded internally as well as externally to the same lead as the tap. Therefore, they cannot connect and thread to lead and are not dependent on small set screws and centers to keep them true to lead, as is the case shown in *F*, Fig. 2.

It is absolutely necessary to give radial stays the same lead as the tap with which the holes are tapped in order to guard against the threads being out of tram thus causing the threads on the bolts or in the holes in the plate becoming stripped.

For Use Instead of a Vise

On the shop or enginehouse work bench one of the handiest of all tools is the vise. So adaptable is it to a variety of uses that many men overstep the line and use it for purposes that it was never intended for. Almost anyone can recall the appearance of a vise that has been used as an anvil by some mechanic or helper who should know better until the edges of the jaws have become so battered that they hardly serve the purpose for which they were originally intended. In one railroad shop the general foreman has had short lengths of rail fastened to work benches with instructions to use them as an anvil and save the vise for better uses.



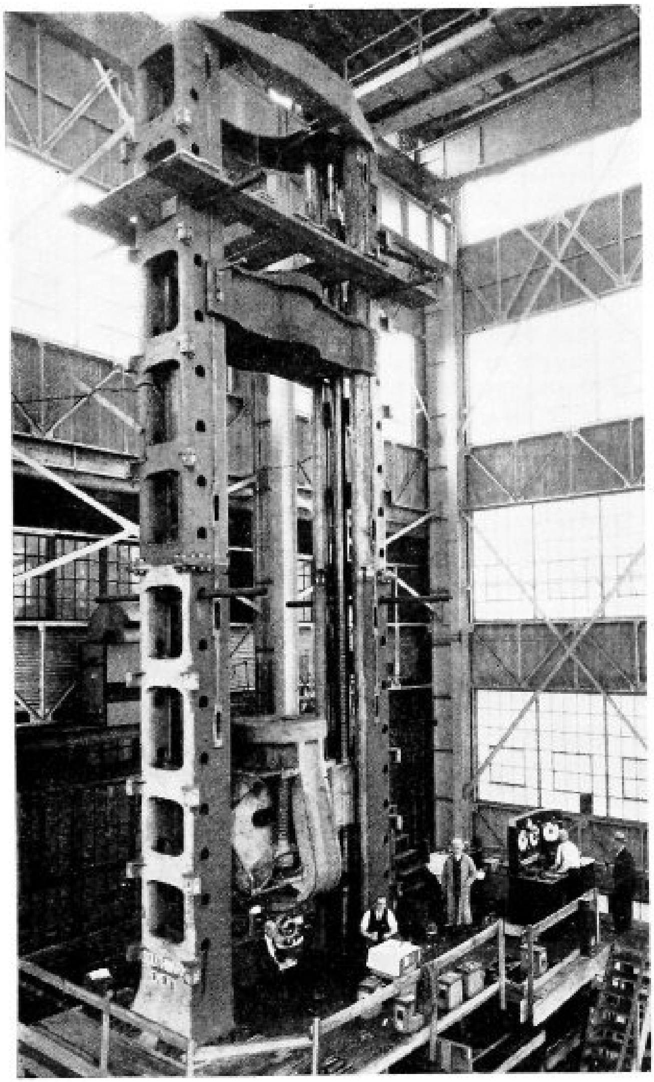
A short length of rail on a work bench makes a better anvil than a vise

Harnischfeger Corporation To Make Hansen Welders

A development of considerable importance in the rapidly expanding uses of welding is seen in the acquisition of the manufacturing rights for the well known Hansen arc welder by the Harnischfeger Corporation, Milwaukee, Wis. The Hansen Welder was developed by the Northwestern Manufacturing Company, and has been manufactured and marketed by that concern for some years.

The Harnischfeger Corporation announce that in taking over this new line, they will continue to produce the Hansen arc welder in the same range of sizes, and with the same structural features as previously—notably, the single current control, the self-exciter feature, and the internal stabilizer.

World's Largest Testing Machine Constructed



Southwark-Emery testing machine under calibration

The Baldwin-Southwark Corporation, Eddystone, Pa., has just completed and delivered to the University of California, Berkeley, Cal., the largest testing machine in the world. In point of dimension it is by far the largest.

It is able to test columns up to 33 feet 6 inches long in compression and up to 4,000,000 pounds load. In tension, it accepts specimens up to 33 feet 6 inches diminished by the machine stroke, and will apply loads up to 3,000,000 pounds. The spread between the columns is 10 feet and the table is 12 feet long. Since the table is level with the laboratory floor a truck may drive between the columns, and the specimen may be lifted directly from the truck body by the testing machine itself.

There are many unique features of this Southwark-Emery testing machine in addition to its great size and unprecedented capacity. Since the principal load application is hydraulic, it is necessary to provide a cylinder, a ram and a pressure-producing fluid pump. The ram on this testing machine is 46 inches in diameter and the hydraulic pressure is about 2500 pounds per square inch when operating at capacity.

The table of the testing machine is stationary, being secured directly to the foundations. The ram is fastened to this table. The cylinder, therefore, is the movable element and to it is fastened, by lug extensions, two vanadium-steel screws 12 inches diameter and 56 feet 8 inches long, threaded double 1/3 square threads per inch. By means of long bronze nuts which are bedded in the lower crosshead (which carries also the weighing system and, in a yoke, the lower grips for tension testing) the loads are transmitted from the screws to the crosshead. The purpose of the threads, however, is not to produce the load as has been the practice in some types of testing machines. During loading there is no movement of nut and screw relative to one another. The function of the screw and nut is to raise or lower the lower crosshead so that the space available for testing may be changed to accommodate the specimen to be tested.

Compression tests are carried out between the lower crosshead and the table, while tension tests are made between the upper crosshead and the yoke around the lower crosshead. The total stroke of the lower crosshead (load stroke and not movement on the screws) is 48 inches which is ample for the extension of practically any tension specimen which probably will be tested in this machine. The total adjustment for change of specimen length is 33 feet 6 inches. This latter change is effected by the rotation of the screws (while unloaded except for weights of parts) by means of a geared motor controlled from the control board which is located at some distance from the machine in a glass-inclosed room. Communication between the operator in the control room and the operator at the machine is obtained by telephone. In addition, loud speakers may be brought into action so that a group may be informed of the progress and technique of the test.

The materials of construction are, in general, high-grade steel castings, alloy-steel forged parts and high-grade Cramp bronzes. The rams are of close-grained cast iron and are ground to a final finished diameter. The weight of the main ram (48 inches in diameter) is about 28,000 pounds and that of the cast-steel cylinder is 40,000 pounds. The grips for tension testing are chrome nickel, heat-treated, machine-cut. They weigh more than 3000 pounds, and occupy recesses in the cast steel crossheads which are about 12 inches by 24 inches. They are handled by air cylinders as they are too heavy for manual manipulation.

The cast-steel table to which the main cylinder ram is fixed and on which the cast-steel columns rest is 17 feet 3 inches by 6 feet 6 inches in plan and weighs nearly 50,000 pounds. Its depth is 35 inches.

This machine stands 46 feet 9 inches above the floor line; the foundations extend about 25 feet below. The depth of the pit is 19 feet. This is a concrete tank of nearly 1000 barrels capacity. The total weight on the foundation is about 475,000 pounds.

In both compression and tension testing, the lower crosshead moves downwards. In order that these heavy weights (about 200,000 pounds) may be returned to position preparatory for a new test, it is necessary to provide two 13-inch diameter pull-back cylinders which are supplied with pressure from the same source as that for the main cylinder and ram. These pull-back cylinders have another very important function. In the failure of specimens either in tension or in compression, there is an enormous store of energy released in a very small fraction of a second. These cylinders and rams, together with four nests of heavy springs, absorb this "shot" of free energy and damp it out without damage to the structure, acting like gun recoil chambers.

Because the height of this machine above the floor is equivalent to more than that of the third floor of an average building, provision has been made for the application of an elevator. This shall permit the presence of assistants at any level for extensometer readings, for examinations of the specimens or for inspection of the grips.

The pressure-producing mechanism is a multiple rotating cylinder pump of 15 gallons-per-minute capacity driven by a 20-horsepower motor. The motor for moving the lower crosshead is rated at 50 horsepower and the head speed is 24 inches per minute. All controls are located at a welded-steel power-plant type instrument board with a ledge for a desk having a sheet Bakelite covering.

The heart of the weighing system is the Emery support or hydraulic capsule. The pressure in the main ram is not measured. In the path of the forces coming from the main ram to the specimen is the hydraulic capsule. This is a very heavy inelastic cylinder having a loose fitting piston the end of which is covered by a diaphragm. The movement of this ram is only a few thousandths of an inch at most, and the oil (with which the capsule is filled), displaced by this micrometer motion, causes a change of shape of the elastic tubes (with which the capsule is connected) in the indicating instruments of which there are four on the instrument board. The ranges of these indicating instruments are: 0 to 4,000,000; 0 to 2,000,000; 0 to 525,000 and 0 to 200,000 pounds.

The care with which calibration is carried out can only be appreciated by witnessing the process. The calibration of this machine required the services of three men, all experts, for one week. This calibration is effected by the use of Morehouse proving rings which are manufactured under the Whittemore-Petrenke patents and are calibrated at the Bureau of Standards, Washington, in a dead weight machine in which weights up to 100,000 pounds may be applied by increments. Baldwin-Southwark Corporation and their associate the A. H. Emery Company own and use the largest collection of rings, both in numbers and in total capacity, in this country, not being exceeded by the Bureau of Standards.

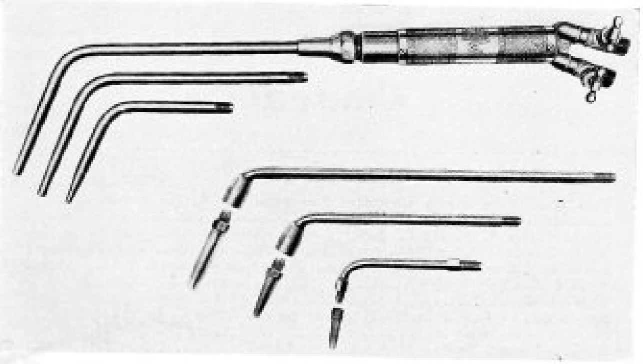
One of the unusual and unique features of this machine is the highly refined control which is incorporated. Devices are mounted for maintenance of constant load over an indefinite period; devices for the application of load at a constant rate of load increment, start, stop and inching buttons for motor control, limit switches for safety and cutouts for same under specific conditions are also fitted. A Telechron clock and indirect lighting for the gages as well as pilot lights for motor circuits are unusual refinements.

This machine will be installed shortly in the laboratory of the University of California. There, under the personal supervision of Professor R. E. Davis, in the course of the next few months, testing history is due to be made.

Welding Torch

The Alexander Milburn Company, Baltimore, Md., has recently placed on the market an entirely new welding torch, known as the type HMS, which is recommended for all types of welding. Copper swaged welding tips are usually furnished but they may be removed from the torch handle and replaced by extensions of various lengths.

These extensions are light, but unbreakable, nickel-silver tubes utilizing Milburn standard type FX and UB welding tips. The extensions are also interchangeable with Airco Davis-Bournonville welding torches.



New type welding torch with tips and extensions

By means of the type HMS torch handle, swaged tips and extensions, the user has at his disposal a complete, versatile welding assortment capable of handling any type of welding work from the lightest to the heaviest.

Business Notes

S. P. Goodloe, Mutual Building, Richmond, Va., has been appointed Southeastern sales agent for the Globe-Steel Tubes Company, Milwaukee, Wis.

B. F. McIntyre is now connected with the Chicago office of The Lincoln Electric Company, manufacturers of Linc-Weld motors and Stable-Arc welders, in charge of motor sales for the Chicago district.

The Cleveland office of the Independent Pneumatic Tool Company, Chicago, manufacturers of Thor electric and pneumatic tools and air compressors, is being moved from the Union Trust Building to 1740 East 12th Street. Hayden F. White is the manager.

J. F. Mehlhope, formerly Chicago district sales manager for the Central Alloy Steel Corporation and later for Newton Steel Company, has joined the sales force of the Chicago Steel Service Company, Chicago, warehouse distributors for toncan iron and enduro stainless steels, products of the Republic Steel Corporation, Massillon, O.

The Boiler Maker

VOLUME XXXII NUMBER 2

PUBLISHED Monthly by the Simmons-Boardman Publishing Company, 34 North Crystal Street, East Stroudsburg, Pa., with Executive Offices at 30 Church Street, New York City.

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Chicago: 105 W. Adams Street. Washington, D. C.: 17th and H Streets, N. W. Cleveland: Terminal Tower. San Francisco, Cal.: 215 Market Street.

Subscription Price: United States and Mexico, $2.00, Canada, $3.50, including duty. Foreign Countries, $3.00 a year. Single Copies, 35 Cents.

Request for change of address should reach us on or before the 15th of the month preceding the issue with which it is to go into effect. It is difficult, and often impossible, to supply back numbers to replace those undelivered through failure to send advance notice. In sending us change of address, please be sure to send us your old address as well as the new one.

THE BOILER MAKER is a member of the Associated Business Papers, Inc. (A. B. P.), and the Audit Bureau of Circulations, (A. B. C.)

EDITORIAL STAFF: H. H. Brown, Editor. L. S. Blodgett, Managing Editor. W. Lumbard, Associate Editor.

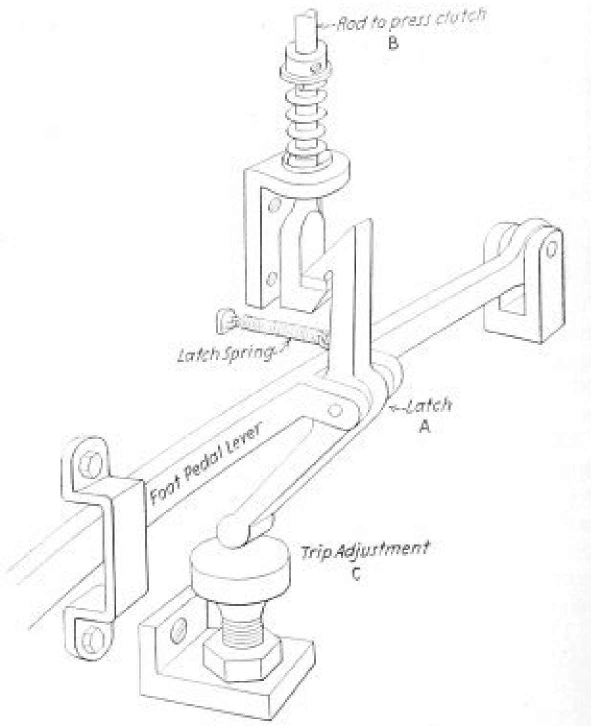
Contents

Communications

Press or Power Shear Safety Trip

TO THE EDITOR:

Noting your interesting shop safety devices shown and described in recent issues of THE BOILER MAKER and knowing that you are looking for other good ideas along that line, I am submitting herewith a safety trip that can be adjusted to release the clutch trip rod on a press or power shear the moment that the press has started on its revolution. When the foot pedal is depressed, withdrawing the clutch finger and allowing the driving dog to fly into the press fly-wheel, the latch *A* is tripped off the reach rod *B* by contact with the trip adjustment stop *C*. The reach rod *B* then being free, the finger immediately is let back in place and will be in place to pull the dog out of the fly-wheel before the completion of the revolution.



Safety device for power press trip

The whole success of the device is in the correct adjustment of the trip stop to trip the latch as quickly as the finger has been pulled. This device thus permits only one stroke of the press when the foot pedal is depressed and the machine cannot repeat the cycle until the foot is raised and the latch is again hooked. The coil spring pulls the latch in place when the foot is raised.

Penacook, N. H. CHARLES H. WILLEY.

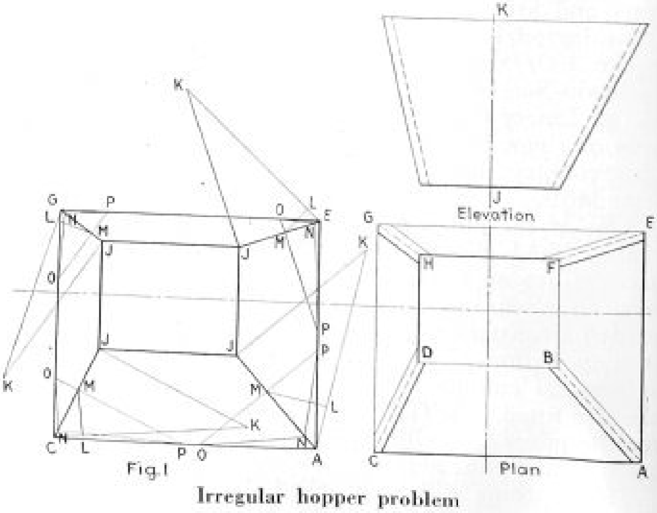
Bevel of Corner Angle

TO THE EDITOR:

In your October issue, one of your correspondents sought, through your column, a formula for determining the degree of corner bevel on an irregular hopper, and the answer afforded him was not, in the writer's opinion, a correct answer to the question. I submit herewith a sketch and explanation which I believe to be a solution to your correspondent's problem.

The sketch shows an irregular hopper, each corner

(*Continued on page 44*)



Irregular hopper problem

Questions and Answers Pertaining to Boilers

By George M. Davies

This department is maintained for the purpose of helping those who desire assistance on practical boiler shop problems. Inquiries should bear the name and address of the writer. Anonymous communications will not be considered. The identity of the writer, however, will not be disclosed unless the editor is given special permission to do so.

Elliptical Heads

Q.—Will you please publish the mathematical reasoning proving that an elliptical head is self-supporting?—A. B. D.

A.—Elliptical heads are self-supporting in the sense that they do not need to be braced or stayed up to certain limits of pressure, diameter, and radii. Over and above these limits they require bracing or staying.

The A.S.M.E. code gives formulas for determining the thickness, etc., of dished heads and also the limitations to which these heads can be built without staying.

These formulas are the results of tests determining the limit to which dished heads can be constructed without staying or bracing and are not the results of a mathematical formula.

Strength of Boiler Braces

Q.—What is the total load that can be carried by the brace as shown in Fig. 1?—G. M.

A.—The A.S.M.E. Boiler Construction Code, Section 1, has the following requirements for determining the design of boiler braces and brace connections:

(1) Determine the required cross-sectional area of the brace by first computing the total load to be carried by the brace, and dividing the total load by the value of allowable stress for unwelded stays or braces given in Table P-7.

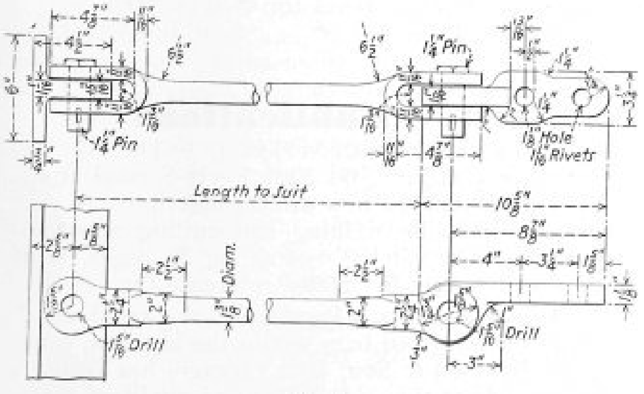


Fig. 1

According to Table P-7, the allowable stress for unwelded stays or braces and unwelded portions of welded stays or braces is as follows: For lengths between supports not exceeding 120 diameters, use 9500 pounds. For lengths between supports exceeding 120 diameters, use 8500 pounds.

(2) Design the body of the brace so that the cross-sectional area shall be at least equal to the required cross-sectional area of the brace for unwelded stays or braces. Where the stays or braces are welded, the cross-sectional area at the weld shall be at least as great as that computed for a stress of 6000 pounds per square inch.

The sketch of the brace submitted with the question is shown in Fig. 1 and indicates that the brace is fabricated by welding two drop forged jaws to the brace rod to obtain the required length. The sketch does not indicate just how the welding is done, no increase in diameter for the weld being shown.

In determining the total load that can be carried by the brace, it is necessary to assume that the actual cross-sectional area of the brace rod is the required cross-sectional area of the brace and to check the area on this basis.

In the brace rod shown in Fig. 1, the diameter through the weld is 1⅜ inches and the load is computed in accordance with paragraph (2) with an allowable stress of 6000 pounds per square inch.

Area having 1⅜ inches diameter = 1.4849 square inches

1.4849 × 6000 = 8910 pounds, total load carried by brace rod shown in Fig. 1.

The brace rod shown in Fig. 2 is the same brace rod as shown in Fig. 1 with the exception that the diameter of the rod through the welded portion at both ends has been increased from 1⅜ inches to 1¾ inches.

This brace rod can now be computed as indicated by paragraphs (1) and (2) first through the unwelded portion and then the welded portion, to determine the load that can be carried by the brace. The diameter of the brace rod through the unwelded portion is 1⅜ inches, the area being 1.4849 square inches. Assuming the brace rod to be over 120 diameters in length, then the allowable stress would be 8500 pounds per square inch.

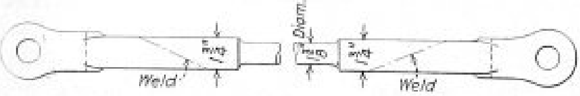


Fig. 2

1.4849 × 8500 = 12,620 pounds, total load carried by the brace rod through the unwelded portion. The diameter of brace rod through the welded portion is 1¾ inches, the area being 2.4053 square inches. With an allowable stress of 6000 pounds,

2.4053 × 6000 = 14,430 pounds, total load carried by brace rod through the welded portion.

The least of these two loads, being 12,620 pounds, is the total load that can be carried by the brace shown in Fig. 1 modified as shown in Fig. 2.

After determining the total load that can be carried by the brace based on its cross-sectional area, it is then necessary to check the design of the brace rod, to see that the various sections are in accordance with the requirements.

(3) Make the area of the pins to resist double shear at least three-quarters of the required cross-sectional area of the brace.

Required cross-sectional area = 1.4849 square inches.

1.4849 × .75 = 1.1136 square inches required cross-sectional area of pins to resist double shear.

Diameter of brace rod pins in double shear = 1¼ inches.

Area of brace rod pins in double shear = 2.2272 square inches actual cross-sectional area of pins to resist double shear.

(4) Make the combined cross section of the eye at the side of the pin (in crowfoot braces) at least 25 percent greater than the required cross-sectional area of the brace.

Required cross-sectional area = 1.4849 square inches.

1.4849 × 1.25 = 1.8561 square inches, required cross-sectional area of the eye at each side of the pin.

Actual combined cross-sectional area through eye at each side of the pin of the brace rod is as follows:

$$3\tfrac{1}{4} - 1\tfrac{5}{16} = 1.9375$$

$$1.9375 \times 1.375 = 2.664 \text{ square inches.}$$

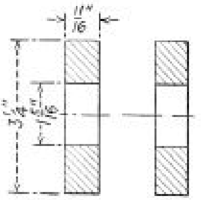


Fig. 3

Actual cross-sectional area of the brace foot at the pin is as follows:

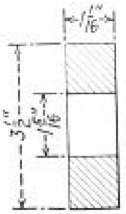


Fig. 4

$$3\tfrac{1}{2} - 1\tfrac{5}{16} = 2\tfrac{3}{16}$$

$$2.1875 \times 1.0625 = 2.324 \text{ square inches.}$$

(7) Make the net sectional areas through the sides of the crow-foot, tee irons, or similar fastenings at the rivet holes at least equal to the required rivet section; that is, at least equal to 1¼ times the required cross-sectional area of the brace.

Required cross-sectional area = 1.4849 square inches.

1.4849 × 1.25 = 1.8561 square inches, required net sectional area through brace foot at the rivet hole.

Actual cross-sectional areas through the brace foot at the rivet hole is as follows:

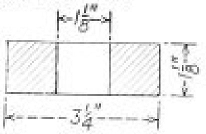


Fig. 5

$$3\tfrac{1}{4} - 1\tfrac{1}{8} = 2\tfrac{1}{8}$$

$$2.125 \times 1.125 = 2.39 \text{ square inches.}$$

The cross-sectional area through the brace at the second rivet is not figured, due to the fact that this area is also supported by one rivet in single shear. The value of the cross-sectional area through the rivet, plus the strength of one rivet in single shear should be greater than the load carried by the body of the stay.

(8) Make the combined cross-sectional area of the rivets at each end of the brace at least 1¼ times the required cross-sectional area of the brace.

Required cross-sectional area = 1.4849 square inches.

1.4849 × 1.25 = 1.8561 square inches, required combined cross-sectional area of rivets at each end of the brace.

Actual cross-sectional area of rivets in the brace foot is found as follows:

Diameter of rivet after driving = 1⅛ inches.

Area of 1⅛-inch diameter rivet is .994 square inch.

.994 × 2 = 1.988 square inches.

The various sections of the brace and brace foot having the desired proportions, the brace rod shown in Fig. 1, modified as shown in Fig. 2 will carry a load of 12,620 pounds.

Bevel of Corner Angle

(Continued from page 42)

of which presents a different degree of bevel to the angle. The elevation is also shown to give an idea of the height.

Fig. 1 shows the method for finding the bevel. Lay down the outside or top lines of plan *A-C-G-E* and in its relative position the small or bottom lines *J-J-J-J*. Next connect lines *J-A*, *J-C*, *J-G* and *J-E*.

At right angles to *J-A*, draw line *J-K*, equal in length to *J-K* on the elevation, Fig. 1. Connect *A* and *K*.

At any point, say *L*, at right angles to *A-K* draw *L-M*. With dividers set at *L-M* and using *M* as a center, locate *N* on line *A-J*.

At point *M* and perpendicular to *A-J*, draw line *O-P*. Now connect *O-N* and *P-N*, and it will be found that *O-N-P* represents the bevel for the angle for that particular corner.

It will be readily seen that I have determined the bevel for the lower right-hand corner. The same procedure followed on the other three corners will present, as is shown, a different bevel for each corner.

East Boston, Mass. SAM J. RANDALL.

Trade Publications

WELDING AND CUTTING APPARATUS.—The Air Reduction Sales Company, 60 East Forty-Second street, New York, has issued a new catalog descriptive of Airco-Davis-Bournonville welding and cutting apparatus and supplies. The catalog, containing 26 pages, is of pocket size, indexed for ready reference.

COLD FINISHED BARS.—Because of the rapid development of cold finished bars within the last few years, Joseph T. Ryerson & Son, Inc., Chicago, has issued a bulletin describing the wide variety of shafting, screw stock, and open-hearth case-carburizing steels in use today.

OPEN-HEARTH SHEET-STEEL PRODUCTS.—The new edition of the booklet "Inland Open-Hearth Sheet-Steel Products," issued by the Inland Steel Company, Chicago, contains up-to-date information on the ordering of steel sheets. Standard extra and differentials, sheet weights, and bundling tables, standard commercial tolerances and trade customs and practices are included, as well as concise description of Inland sheet-steel products.

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Michigan
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New Jersey
New York
Ohio
Oklahoma
Oregon
Pennsylvania
Rhode Island
Utah
Washington
Wisconsin
District of Columbia
Panama Canal Zone
Territory of Hawaii

Cities

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Detroit, Mich.
Erie, Pa.
Kansas City, Mo.
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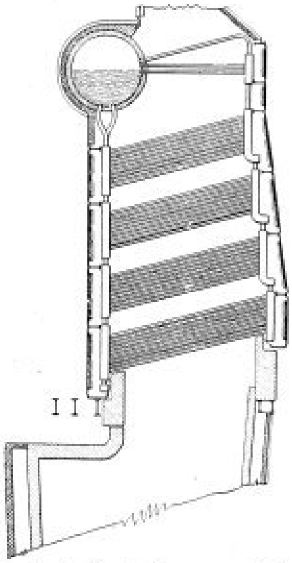
Selected Boiler Patents

Compiled by Dwight B. Galt, Patent Attorney, Washington Loan and Trust Building, Washington, D. C. Readers wishing copies of patents or any further information regarding any patent described, should correspond directly with Mr. Galt.

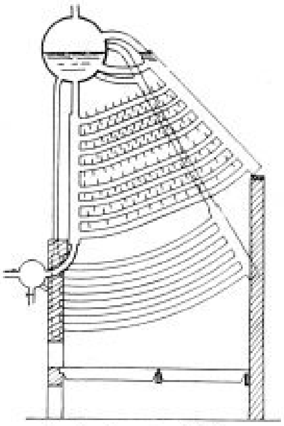
1,754,350. STEAM BOILER. HORACE E. BUNKER, OF PLAINFIELD, N. J., ASSIGNOR TO THE BABCOCK & WILCOX COMPANY, OF BAYONNE, N. J., A CORPORATION OF NEW JERSEY.

Claim.—In a steam boiler having a tapered pass, a plurality of banks of tubes disposed in said pass and spaced apart longitudinally thereof, each bank comprising a plurality of tubes of substantially equal length disposed in rows spaced apart longitudinally of the pass and the length of tubes in the successive banks decreasing from one end of the pass toward the other, vertically extending substantially straight water chambers disposed at opposite ends of the tubes in each bank and connected thereto, the chambers on at least one side of the pass being stepped with respect to each other, means connecting the chambers of successive banks to form continuous chambers at opposite sides of the pass, a steam and water drum, and connections from the upper ends of said chambers to said drum. Three claims.



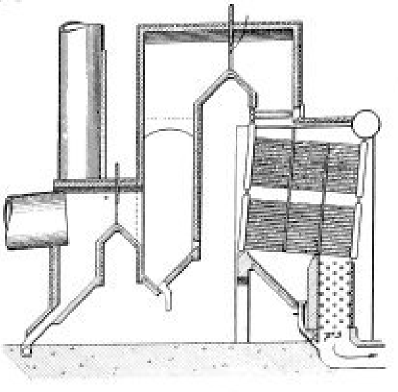
1,796,512. BOILER WITH HIGH EVAPORATIVE CAPACITY. ALFRED DHOME, OF ST.-DENIS, FRANCE, ASSIGNOR TO SOCIETE ANONYME DES ETABLISSEMENTS DELAUNAY BELLEVILLE, OF ST.-DENIS, FRANCE, A FRENCH CORPORATION.

Claim.—A watertube boiler with high evaporative capacity, comprising in combination substantially parallel evaporation tubes arranged in parallel groups, headers connected to the ends of said tubes, arranged parallel to the general flow of gases escaping from the furnace, an upper drum connected to said headers, the headers to which are connected the ends of the tubes converging so as to form an acute angle towards said upper drum, whereby the lengths of the said tubes are decreased towards said drum, means for feeding water to the headers at the lower end and transverse heat transmitting fins provided on said tubes and located more closely together as said tubes are more and more distant from the furnace, whereby the hot gases flowing through the boiler effect only one travel in the same general direction approximately parallel with the surface of said fins.



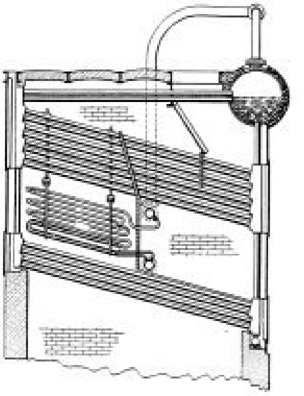
1,753,589. WASTE-HEAT BOILER. JOHN E. BELL, OF BROOKLYN, N. Y.; LOLA R. BELL EXECUTRIX OF SAID JOHN E. BELL, DECEASED.

Claim.—The combination in a waste-heat boiler of the horizontal watertube type transversely baffled to provide a plurality of transverse passes for the heating gases, of means providing an inlet for the heating gases opening downward to the upper end of the transverse pass adjacent one end of the boiler, and common means for collecting dust separating out of the heating gases in said pass and the pass immediately adjacent thereto. Six claims.



1,794,507. FLUID HEATER. WILBUR H. ARMACOST, OF NEW YORK, N. Y., ASSIGNOR TO THE SUPERHEATER COMPANY, OF NEW YORK, N. Y.

Claim.—In a fluid heater, the combination with means defining a pass for combustion gases, of a plurality of multi-loop tubular heating units disposed in said gas pass, each of said units comprising an inlet portion formed of a single tube and an outlet portion serially connected therewith formed of tubing of the same external but smaller internal diameter than said inlet tube. Seven claims.

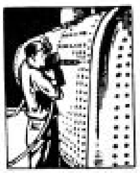


1,752,673. WASTE-HEAT BOILER. GEORGE T. LADD, OF PITTSBURGH, PA., ASSIGNOR TO LADD WATERTUBE BOILER COMPANY, OF NEW YORK, N. Y., A CORPORATION OF DELAWARE.

Claim.—In combination, a furnace from which waste heat is to be derived, an off-take therefor having a throat, a boiler in the off-take above the throat including an upper drum and a lower drum and connecting vertically disposed tubes, downcomer tubes within the throat and leading from the lower drum downwardly, a header for said tubes located exterior of the throat, upcomer tubes within the off-take connected to the upper drum at one end, a header to which the other ends of said tubes are connected, and means for connecting the headers.

The Boiler Maker

Reg. U. S. Pat. Off.



Adoption of A. S. M. E. Code

With the recent adoption of the A. S. M. E. Boiler Construction Code by the City of Houston, Tex., the American Uniform Boiler Law Society reports a total of twenty-one states and territories and seventeen cities that have adopted the A. S. M. E. Code either in whole or in part. While a number of states and cities are enforcing the code without reservation, there are a few which have adopted or accepted only specific sections of the code.

During the past few months eight states and eleven cities have taken action on Section I to VIII inclusive of the 1931 Boiler Code as follows: The State of Pennsylvania has Section VIII (Unfired Pressure Vessels) under consideration, while the State of Delaware will report on this section at a later date. Ohio has adopted Section III (Boilers of Locomotives), with a factor of safety of 5, and has also adopted Section VIII. Sections I (Power Boilers) and II (Material Specifications) have been accepted by Missouri and this state will act on the remainder of the code at a later date. The State of Washington has accepted the entire code, while California will take action on Sections VI (Rules for Inspection), VII (Suggested Rules for the Care of Power Boilers) and VIII at a later date. The Territory of Hawaii has accepted Section VIII and the Canal Zone, while it has followed the code in the past, will act upon its adoption in the future.

While the cities of Philadelphia, Pa., Memphis, Tenn., Kansas City, Mo., Chicago, Ill., and St. Joseph, Mo., will report later on the adoption of such sections of the code as have not been approved to date, Scranton, Pa., has Section VIII under consideration. Seattle, Wash., has no jurisdiction over miniature boilers as covered by Section V, and St. Louis, Mo., has adopted Sections IV (Low Pressure Heating Boilers) and V, but has no jurisdiction over unfired pressure vessels as covered by Section VIII. Erie, Pa., has adopted Section VII and Los Angeles, Cal., has adopted Section VIII. Houston, Tex., the latest addition to the list of code cities, has adopted Sections I, II, IV and V, has accepted Sections VI and VII, but has no jurisdiction over locomotive boilers or unfired pressure vessels.

Progress is being made by the American Uniform Boiler Law Society in encouraging the adoption of the Boiler Code of the American Society of Mechanical Engineers by the various cities and states in the country. The ultimate desire, however, is the adoption of this code by all states and cities. Diversity of boiler construction requirements tends to confusion on the part of manufacturers; uniformity, on the other hand, simplifies not only inspection and insurance requirements, but enables the boiler manufacturer to standardize his construction methods and inspection procedure.

Consider the Foreman

With the decline in their business, certain railroads have adopted a policy of classifying department foremen on a basis little higher than shop labor. When a shop is shut down or running on part time, the foremen are subject to the same curtailment in employment as is accorded other classes of shop employees.

While recently, master boiler makers, having records of many years of loyal service, have been subjected to such treatment, the heads of other departments in the shop have not been immune.

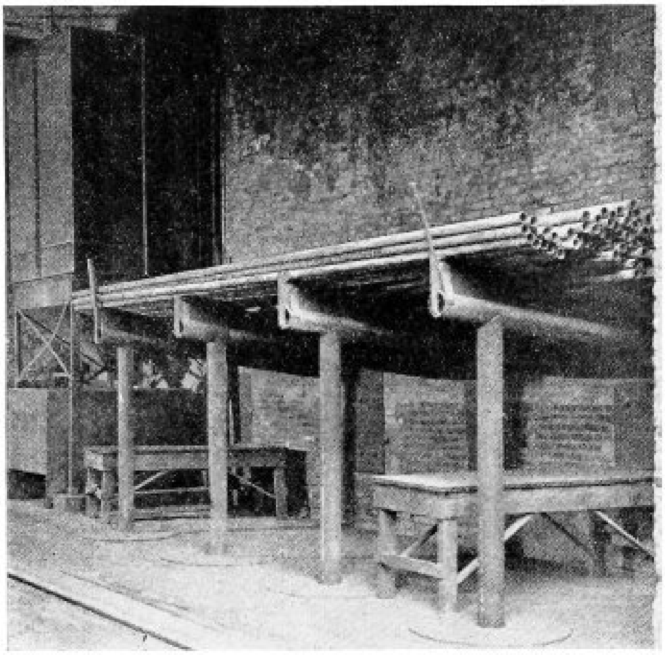
The effects of this policy, if pursued by many of the railroads, may conceivably have a reaction in the future that will far outweigh any passing momentary saving. Entirely aside from the economic phase of the problem, there is the question of morale which is fundamental to the railroad as to every other industry.

The organization of industry is no different than that of an army and its function is similar in that it is designed to wage an economic battle. Of highest military importance is the morale of the men. While the personnel of an army at times may be reduced, the roster of officers is kept intact and, of utmost importance, the non-commissioned officers' ranks are not emasculated. There is always the nucleus of a trained fighting force about which the army may be reformed.

The present state of the railroads is no different than this. If there is to be a future for the railroads of this country, the return of business must be built on the morale of the officers and the foremen. To destroy the prestige and weaken the spirit of loyalty in the minds of either class at this time is to make extremely difficult a rapid recovery when the turn comes.

If, in the light of the grave possibilities from the pursuance of this policy, the railroads continue to apply it to their foremen, it is certain that one of the most valuable assets of the railroads—the morale of the "top sergeants"—will be destroyed. The officers of any railroad, having an interest in its future economic welfare, cannot afford at this time to neglect those in its organization who can most quickly and effectively rebuild the shattered ranks of its personnel. Considerate treatment of the department heads and foremen at this time will do much to insure the re-establishment of efficient shop forces when the need arises, as it inevitably will with more normal business.

Milwaukee Progressive Flue Repair Shop ▲ ▲ ▲



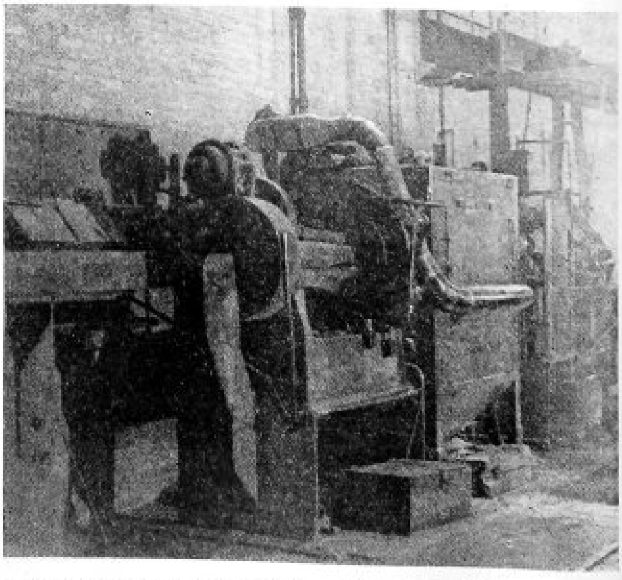
Tube or flue receiving rack

The operations are organized for progressive movement without back travel—Both power and gravity movements reduce manual handling to a minimum—Scale is removed by passing the flues through cracking rolls and the cleaning finished by sandblasting

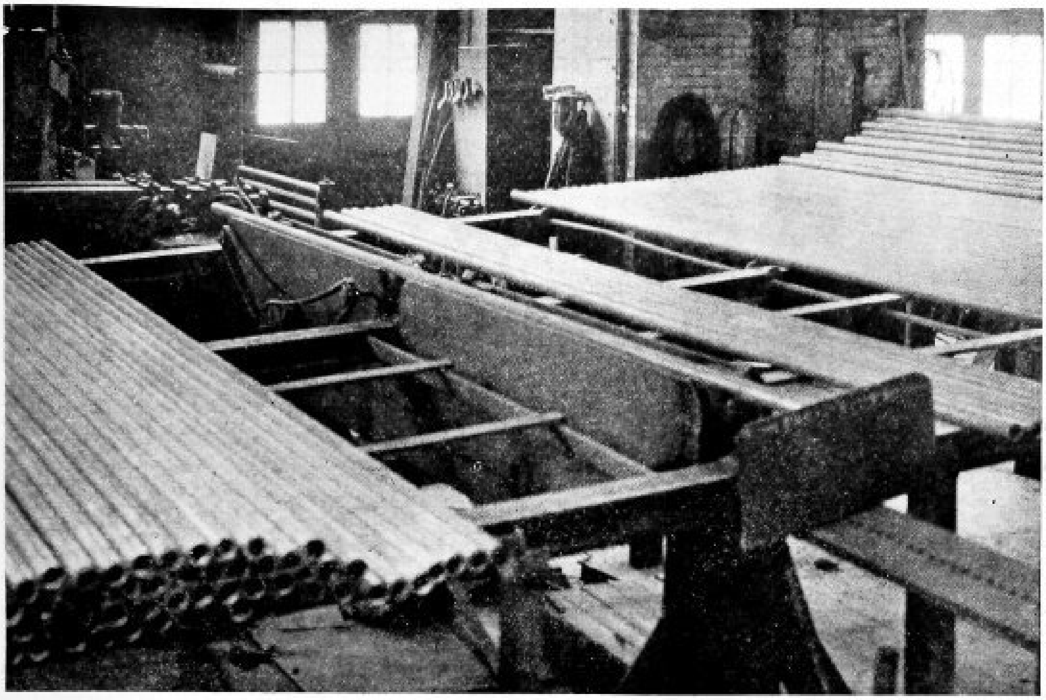
Both locomotive and car maintenance work, particularly heavy repairs, on the Chicago, Milwaukee, St. Paul & Pacific have been concentrated largely at the main repair shops, Milwaukee, Wis. This is especially true with respect to heavy boiler repairs, including the application of safe ends and general maintenance of boiler tubes and superheater flues. A thoroughly-revised and modern flue shop was completed early in 1930, one of the unusual features being the provision of cracking rolls and sandblast equipment for cleaning tubes and flues. Dual friction cut-off saws and an electric flue-welding machine were installed, and the entire machinery and equipment arranged for the progressive movement of flues through the various repair operations without back travel and with a minimum expenditure of manual labor.

The fundamental soundness of the Milwaukee flue shop layout has been amply demonstrated in about two years of operation, during which the cost of repairs, including charges for labor, material and shop overhead, has been reduced to an average of about 2½ cents a foot for small tubes and 9 cents a foot for superheater flues. With an average force of three boiler makers and five helpers, working 40 hours a week, an output of about 6000 to 6500 tubes, plus 150 to 350 superheater flues, can be obtained.

The Milwaukee flue shop machinery layout is shown in the large drawing, important units in the repair equipment being indicated by letters and the general movement of tubes and flues by arrows. A detailed record of the machines installed, individual power requirements and manufacturers' names, is given in a table elsewhere in this article. Tubes and flues are received from the erecting shop or from the stores department, as the case may be, in one bay of the boiler shop building, separated from the flue shop proper by a brick firewall. Tubes received from the erecting shop are usually loaded on push cars and moved to the boiler shop via the transfer table. Tubes received from other points on the Milwaukee for repairs are handled on



Ryerson scale cracker and New Haven sand-blast machine



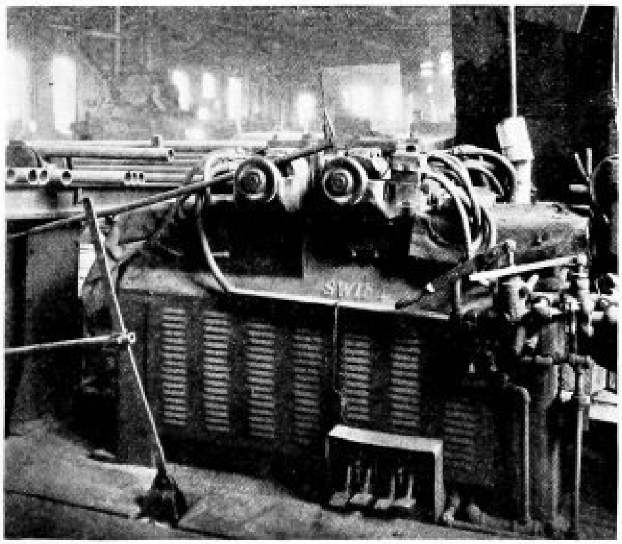
For handling tubes to the electric-welding machine a skidway and air-operated device are used to save labor. Below is the Swift electric flue welder showing the levers for operation

Blue Bird three-wheel trailers of rugged design. In either case, an entire set of tubes is lifted by means of a steel-cable sling and Elwell-Parker crane truck to the receiving rack *A*. This is an inclined rack or skidway of substantial construction, made of steel I-beams and scrap superheater flues, welded together to form a rigid unit, supported on circular steel floor plates of generous size.

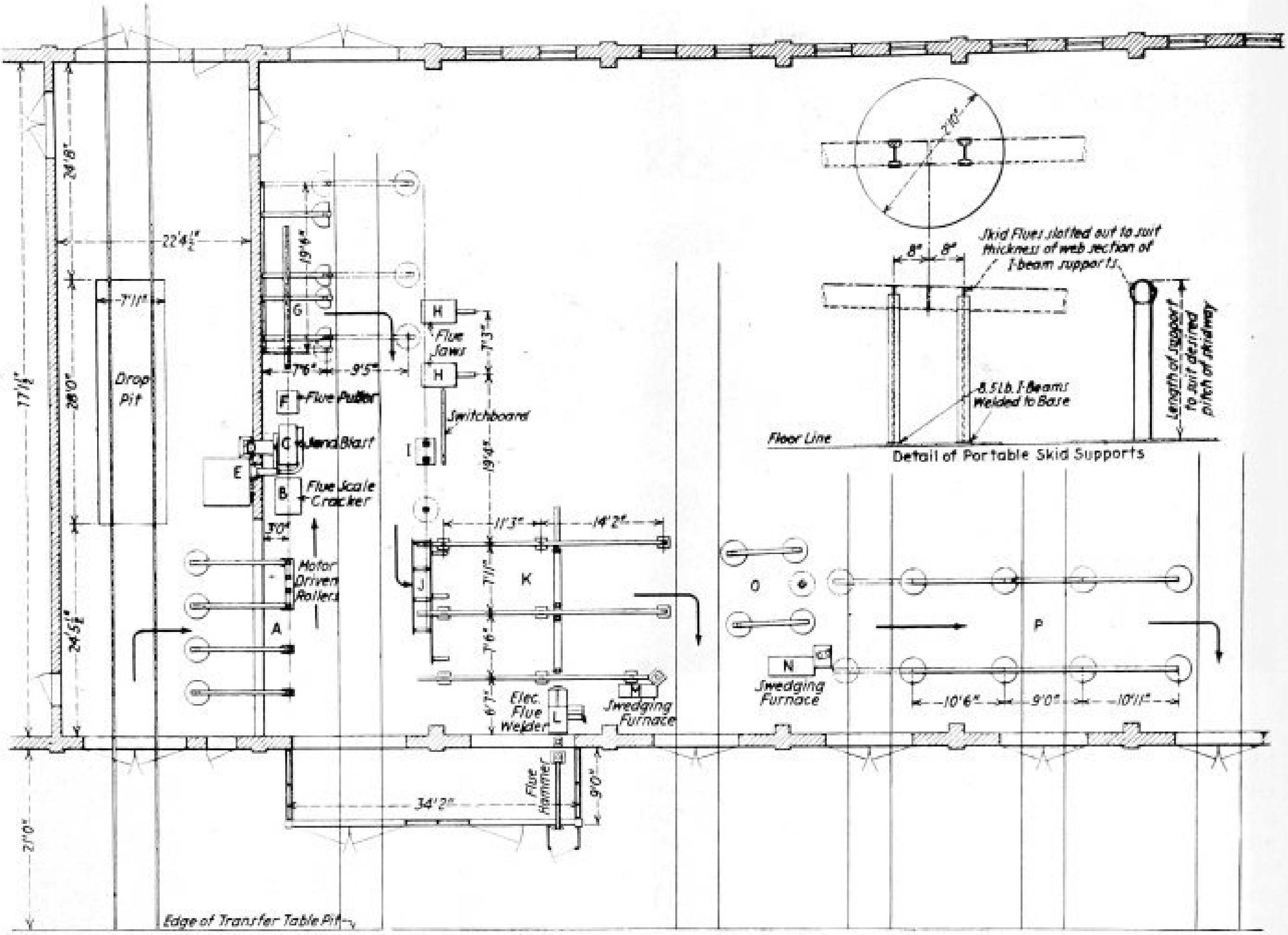
The tubes pass through a narrow opening into the flue shop, where they are tripped one at a time into the cleaning line and delivered by motor-driven rollers to the Ryerson flue scale cracker *B*. The three power-driven, knurled cracking rolls, adjustable by an eccentric to accommodate any size from 2-in. to 5½-in. tubing, are set at an angle, and serve not only to crack off the dry scale but to propel 2-in. tubes into the sandblast machine *C* at a rate of 18 ft. per minute and 5½-in. flues at a rate of 9 ft. per minute.

The sandblast machine is equipped with seven easily-replaceable cast-iron nozzles, through which dry silica sand, drawn from the bottom of the machine by suction, is discharged directly on top of the revolving tube or flue. Air for this operation is taken from the shop air-line at a pressure of about 90 lb. per. sq. in. Motor-driven fan *D* draws air, fine dust and scale from the machine, delivering it to the New Haven dust arrester *E*. In this large dust arrester, baffles serve to release the dust from the air, permitting it to drop to the bottom of the arrester where it is drawn off through a small hopper slide opening into a lift-truck skid for easy removal. The seven sandblast nozzles are evenly spaced on about 6-in. centers, along the section of tube or flue which is in the sandblast machine. While in some cases the nozzles have only a short life, they are relatively inexpensive to replace. The sand is used over and over again until its cutting qualities are practically lost, at which time it passes to the dust arrester in the form of a fine powdery material. Approximately one cubic yard of new sand is used for replacement each day under normal operation.

As the boiler tube or flue passes through the sandblast machine, it is engaged by the three rolls of the flue puller *F*, only one of these rolls being power-driven. The puller rolls are set to move the tube or flue at the same speed as that given by the cracker. Passing through the flue puller, the forward end of the tube or flue engages a carrier which moves along an inclined I-beam rigidly supported from the flue-shop wall. When this carrier reaches the end of its stroke, a trip is operated which permits the tube or flue to drop to the skidway *G*. This skidway is constructed of straight lengths of scrap superheater flues slotted at one end to engage a Z-bar bolted to the wall and at the other end resting on short upright sections of flues welded to movable floor plates. It is obvious that this construction permits moving the inclined straight lengths of flues constituting the skidway to any point desired along the Z-bar, so as to accommodate tubes and flues of varying lengths.



From skidway *G*, the tubes or flues roll by gravity onto temporary superheater-flue extensions across the



Shop arrangement for repairing boiler tubes and flues at the Milwaukee shops of the Chicago, Milwaukee, St. Paul & Pacific

List of Flue-Shop Machinery and Equipment

Kind of machine	Size of motor	Manufacturer
No. 13 fan blower....	5 hp.	Clarage Fan Company, Kalamazoo, Mich.
Universal scale remover	5 hp.	Joseph T. Ryerson & Son, Inc., Chicago
Universal sandblast....	......	New Haven Sand Blast Company, New Haven, Conn.
Flue puller	3 hp.	New Haven Sand Blast Company, New Haven, Conn.
Dust arrester	......	New Haven Sand Blast Company, New Haven, Conn.
Two friction saws.....	10 hp.	Joseph T. Ryerson & Son, Inc., Chicago
No. 35 electric welder.	......	Swift Electric Welder Company, Detroit, Mich.
Two swedging furnaces	......	Shop made
Three flue hammers...	......	Joseph T. Ryerson & Son, Inc., Chicago

track, ready for handling into the cut-off saws. It is here that the initial inspection of the tubes or flues takes place. As they are rolled across the track, before passing to the cut-off saws, they are examined, any observed to be thin or excessively pitted or corroded being rejected and placed on a trailer for delivery to the scrap dock. Those suitable for reclamation are passed on to the two Ryerson friction saws *HH*, one of which is stationary and the other adjustable, to permit cutting the tubes and flues to the correct length, both ends being cut simultaneously. These cut-off saws are of the hobbed-tooth friction type, water-cooled and operated at 3600 r.p.m., each by a 10-hp. motor.

After being cut to length, each tube or flue is lifted slightly from its supports by an air-operated unit equipped with power rolls which give the tube or flue an impulse over intermediate rolls *I* to roller-equipped table *J*, from which it rolls by gravity to a portable rack or sling, subsequently elevated by a crane truck onto the upper end of skidway *K*. The details of this skidway, as well as the air-operated unit for delivering flues to the electric welder, are clearly shown in detail on another drawing. Inverted inclined rails are welded to vertical supports of the required length made of scrap flues. A flue carrier, equipped with power rolls driven from a reversible air motor, is provided with an air cylinder arrangement to elevate each tube or flue, as it is tripped, and propel it into the electric welding machine, where it is stopped by the safe end previously gripped in the dies. After being welded, operation of the power rolls moves the tube or flue through the dies and into the required position under the flue hammer where the weld is reduced to the proper dimensions.

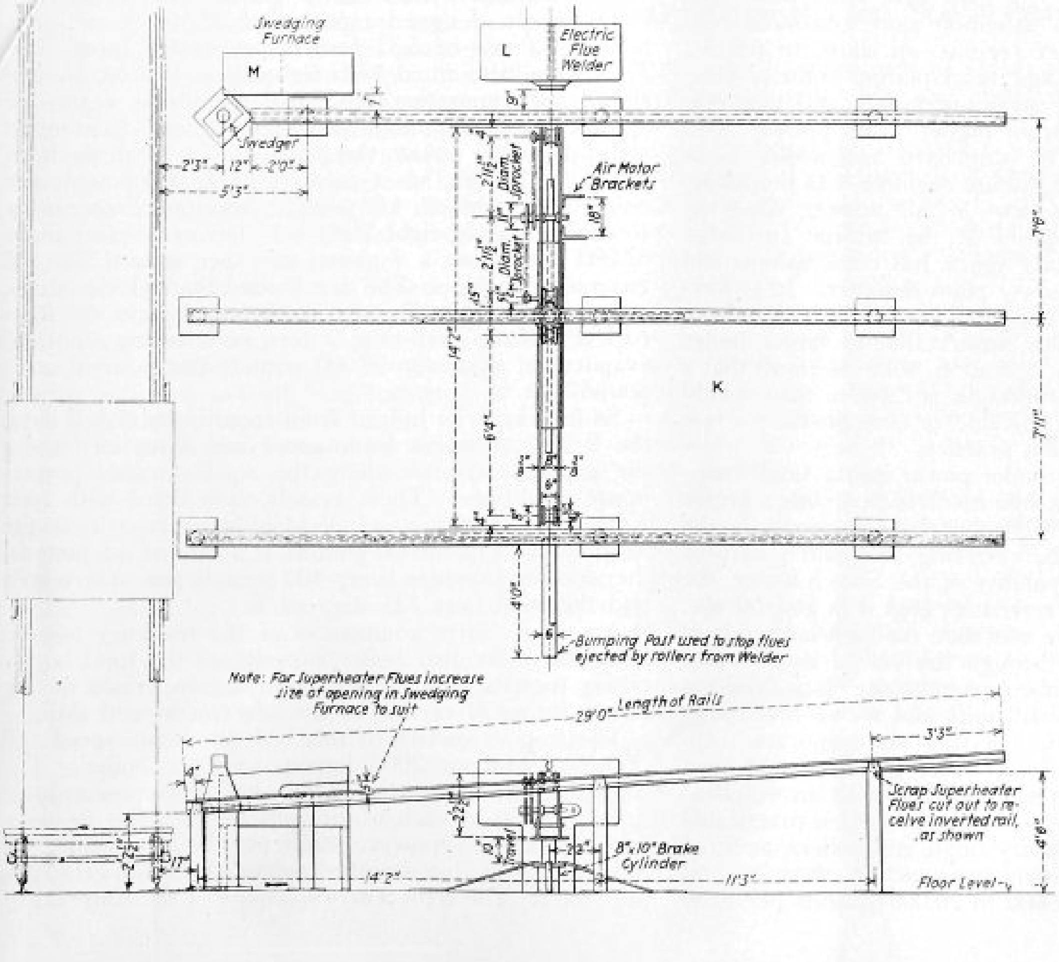
After hammering of the weld, the tube or flue is returned by the reversible rolls to skidway *K* and rolled by gravity into the swedging furnace *M*. This furnace is of the open-side type, 18 in. wide by 4 ft. 6 in. long, which permits the gradual heating of the tubes until they are removed, one at a time, from the hottest end of the furnace for the swedging operation under the hammer. After swedging, the tubes are loaded on a push car and, if not pitted, are ready to be blown out and delivered to the erecting shop or stores department through the adjacent shop doorway. If pitted, they are moved to storage rack *P* for final inspection, welding of pitted spots by the acetylene torch, etc. Superheater flues are rolled across a portable skidway to position *O*, where they are swedged in furnace *M* of similar design to *N*, but large enough to accommodate the larger flues. The superheater flues, also, are elevated to storage rack *P* where any pitted spots are filled in by welding. After all repair operations are completed, either tubes or flues are blown out and are then ready for delivery in complete sets through the nearest flue-shop door to the erecting shop, or to the storehouse. No tubes or flues are tested subsequent to reclamation, as experience has shown less than 1 percent of failures after application in the boilers.

Under normal conditions, two helpers operate the entire line of flue-cleaning equipment, one on the day shift and one at night, in order to keep up with this part of the work. One man operates the friction saws, with a helper when handling superheater flues. One man operates the electric welding machine and flue hammer, and another the swedging furnace *M*. The latter, when swedging large superheater flues in furnace *N*, has an additional helper to assist in revolving the flues. One or more welders, as required, are used in filling in pitted holes with the oxy-acetylene welding torch. Therefore, when the flue shop is running at what might be called normal capacity, a force of three boiler makers, five helpers and one or two acetylene welding operators, depending upon the severity of the pitting condition, is employed.

The only part of the flue shop operating on more than one shift is the cleaning line which must usually be operated one extra shift in order to keep up with the other part of the work. Whereas the capacity of the cleaning line is at the most about 60 two-inch tubes an hour, the friction saws have at least twice that capacity and the electric welding machine can apply safe ends 18 in. long or less to 2-in. tubes at the rate of 80 to 86 an hour. The swedger has a capacity of about 150 two-inch tubes an hour.

The reclamation of boiler tubes and flues constitutes exceptionally severe service, and all machine bearings must be fully protected from dust and dirt and designed to stand severe vibration. As a result of experience, certain precautions must be observed to secure the best results in cleaning tubes and flues. For example, the scale cracker, while designed to operate efficiently with all kinds and thicknesses of scale, will do this only if the scale is dry. Otherwise, the knurled projections on the wheels become clogged with wet scale in a powdered form and are rendered useless for the purpose intended. Occasionally, even with dry scale, it is not all removed and a set of flues may have to be put through the cracker and sandblast machine twice. All flues must be entered in the scale cracker from the front ends, which are relatively smooth and without burrs, to avoid difficulty in engaging the cracking rolls. Pretty much standard practice is followed in the operation of the friction saws except that in this case they are provided with pneumatic feed to permit one-man operation. Hand-feed is also available for use in case of necessity. No special machine or belt sanding of the ends of the flues or of the safe ends is necessary in preparation for the welding operation as the small amount of thin flash left by the friction saws is quickly melted into the weld and subsequently hammered into a uniform smooth surface. The operation of the welding machine is fully safeguarded, with all operating levers and handles conveniently located, and the inherent characteristics of the machine such that it is in general easy to operate at the rate of speed necessary for the best results, from the point of view of maximum production of satisfactory welds.

In these days, modernization of production processes is a paying investment. The results obtained by the Milwaukee demonstrate the value of such work.



General arrangement of device for handling tubes and flues to the electric welder

Modern Trend of Marine Boiler Installations ▲ ▲ ▲

By James Swan

The use of turbines in power stations ashore and for the propelling machinery of all steamships of medium and large power and speed may now be considered as firmly established standard practice. While it is true that, in certain somewhat exceptional instances, high-pressure reciprocating engines are used ashore in connection with exhaust-steam utilization for various manufacturing processes, and, further, that the combination of reciprocating engines and low-pressure turbines for moderate powers is productive of most satisfactory results afloat, the straight turbine plant has become practically universal for large steam power units both on land and sea.

One of the first results of the introduction of turbines was the added importance given to condensers, because of the greater demands made upon them and the vastly increased necessity of care both in condenser design and operation, and in the selection of tube material. It is perhaps too much to say that all of the new condenser problems have been entirely and satisfactorily solved, but it is certainly true that such progress has been made that the salting of boiler feed water by condenser leakage and tube failure is far from being the bugbear which constantly threatened sea-going engineers but a few years ago.

With the low-pressure end of the expansion curve satisfactorily cared for, the past few years have seen the designing engineers' attention more and more concentrated upon its upper regions—in short, to the development of the boiler and its adaptation to the production of ever-increasing steam pressures and temperatures. Although the use of higher steam pressure had its practical inception on shipboard and while, for a considerable period, the marine engineer was the leader in this development, the lead in this respect, since the virtually universal adoption of the turbine for large powers in the past twenty years, has come ashore and rests with the modern power plant designer. In power-house plants, increased pressures and temperatures have been accompanied by the construction of larger boiler units and more intensive operation, with the result that a vastly greater power generation per boiler than would have been considered practicable, or even possible, a few years ago is now common practice.

The concentration of boiler power into a small number of large units rather than its division among a larger number of small ones is a natural development in the use of watertube boilers. So long as steam pressures were confined to the capability of the Scotch boiler, the latter's many good characteristics kept it in general use, but as its practicable size was soon reached, large power could be attained only through the use of more boiler units. With the watertube type, pressure alone does not limit the size of individual units and so we find today single power-house boilers having an evaporation capacity of 500,000 pounds of water per hour and even more during peak loads as compared with an evaporation of 35,000 to 40,000 pounds in the largest practicable Scotch boiler, while ordinary single-end boilers, as fitted in merchant ships of recent construction, show a total evaporation of from 17,000 to 20,000 pounds per hour at a rate of 5 to 6 pounds per square foot of heating surface.

In the light of the rapid and well nigh revolutionary development in engineering during the past few years, one can hardly dare prophesy a limit to the steam pressures which may be used on board ship. It seems, however, unlikely that pressures of much over 500 pounds per square inch, or a total steam temperature of over 750 or 800 degrees F., will be found in very general use for some time to come. These limits are, of course above the ability of the return tubular boiler to attain, without decrease in diameter and consequent reduction of capacity.

The boilers fitted in certain recent ships show a marked tendency to follow power-house practice, not only as to increased steam pressures and temperatures, but in individual evaporative capacity as well. It is interesting to note, however, that the increased capacity is the result mainly of larger generating surface, the evaporation per square foot of heating surface being kept within conservative limits. Thus, the Babcock & Wilcox boilers of the recently constructed Export Line ships have a designed capacity of 22,500 pounds per boiler at a rate of 5.33 pounds per square foot. The *Statendam,* also fitted with Babcock & Wilcox boilers, shows an evaporation of 35,000 pounds at a rate of approximately 5 pounds per square foot. Coming to ships of greater power, the *Bremen,* with 20 three-drum watertube boilers, has a capacity of 45,000 pounds per boiler at a rate of 4.9 pounds, and the *Empress of Britain* in each of eight Yarrow boilers evaporates about 62,000 pounds at a 5-pound rate per square foot of generating surface. The new United States Lines ships, now under construction at Camden, each have six Babcock & Wilcox small-tube boilers, each having a normal evaporation of about 50,000 puonds per hour at a 5-pound rate.

So far as can be judged from recently published data, the British steamers *Strathnaver* and *Strathaird* show the greatest advance along the road towards power-house conditions. These vessels, each fitted with four Yarrow boilers, have an individual boiler capacity in the neighborhood of 69,000 pounds at a rate of 5.5 pounds, the working pressure being 400 pounds per square inch and the total heat 725 degrees F.

An even clearer conception of the tendency toward increased individual boiler capacity as the total horsepower mounts may be had from a comparison of the power rating of certain of the aforementioned ships.

The Export vessels, of relatively moderate speed and power, get about 2000 horsepower per boiler. The *Statendam,* for her rated 22,000 shaft horsepower, must develop 3666 in each of her six boilers; the *Bremen,* for 92,500 horsepower, 4625 per boiler; while the *Strathnaver,* with a designed power of 28,000, gets 7000 per boiler. The eight Yarrow boilers of the *Empress of*

Britain produce from 7000 to 7500 each, while her single Johnson boiler, the first of its type, we understand, to be put into actual service at sea, is said to be capable of generating 10,000 horsepower, although this would seem to be possible only at the expense of forcing to a much higher evaporative rate than is usually considered safe for merchant ships.

As showing very clearly the improvement in overall efficiency which results from the use of higher pressures and temperatures, the performance of the Panama-Pacific liners *California* and *Virginia* are particularly interesting. These vessels are practically duplicates, with the single important exception of their boiler plants. Both have Babcock & Wilcox boilers, those of the *California* operating at 275 pounds per square inch pressure and 100 degrees superheat, while the corresponding figures for the *Virginia* are 300 pounds and 200 degrees. The fuel consumption of the *California* is 0.76 pound per horsepower hour and of the *Virginia* 0.70 pound. Moreover, the *Virginia* generates her power in eight boilers as against the twelve of the *California*. The *Virginia* uses 110 tons less fuel per round voyage between New York and the West Coast than her sister vessel. The *Virginia's* boilers are, however, individually larger than those of the *California*, having 5461 square feet of heating surface each, as compared with the latter's 4598 square feet.

Although the modern high-pressure boilers are markedly more efficient than their predecessors, the great increases in individual power naturally make necessary increase in absolute size and space occupied. With the great power-house boilers, this increase in size has been very largely vertical. Not only is firebox volume enormously greater than in the older types, but, to ensure efficient operation, air heaters and economizers are added above the actual steam-generating portion of the unit with a resulting overall height of boiler which, to one unfamiliar with the present-day electric power plant, seems quite unbelievable. It does not seem possible that the marine engineer can follow his stationary brother to such "heights" of boiler construction, but a comparison of evaporative capacity per square foot of fireroom floor shows the effect of concentration of boiler power into a few large and relatively lofty units. The *Statendam,* with large-tube Babcock & Wilcox boilers, evaporates 49 pounds per square foot of fireroom floor per hour; the *Bremen,* with small-tube, three-drum boilers, 54 pounds; the *Empress of Britain* and the *Strathnaver,* both with small-tube boilers, evaporate respectively 64 pounds and 78 pounds. The concentration into small floor space does not, however, necessarily indicate a corresponding increase in efficiency, for the *Statendam* with an overall fuel consumption of 0.61 pound per shaft horsepower hour for around 20,000 total horsepower compares most favorably with the *Empress of Britain's* 0.57 pound at 55,000 to 60,000 horsepower.

Just what line future marine boiler design will follow seems difficult to predict. So long as the specific evaporative rate is not materially increased, and it will be noted that these rates for the various ships which have been mentioned do not vary materially, the type of boiler to be used and, dependent upon this, the number of boilers may well be influenced by the size and arrangement of the ship which is to receive them. Concentration of boiler power into a few units involves greater height of boiler and uptake connections and a corresponding reduction of available commercial space, for either passengers or cargo on the decks in way of the boiler compartments. On the other hand, the division of power among smaller boiler units means longer and more complicated steam piping, and a greater number of valves and other similar fittings to be looked after and maintained in good condition.

Of course, higher rates of evaporation than those mentioned are entirely possible and are common on high-powered naval vessels, but it seems doubtful if any great increase in specific evaporative rates are practicable for the boilers of commercial vessels which have to operate for long periods at or near to the designed maximum power. With the high pressures and temperatures now coming into use, only the greatest care makes satisfactory and sustained steam generation possible. Feed water must be as near to pure distilled water as it is possible to make it, and oil must be kept out—the boilers outside as well as in, must be kept clean. Every increase in specific evaporative rate, every increase of "forcing" the boiler, makes more difficult and uncertain the maintenance of this clean condition for any but the shortest periods.

The successful design of marine machinery as well as of ships themselves is largely dependent upon a correct evaluation of more or less conflicting requirements and limitations. Boilers, in common with all other members of a marine power plant, are subject to these conditions and the final choice must be in the nature of a compromise. Pressures and temperatures will undoubtedly continue to increase and, particularly for high powers, individual boiler powers will be larger. While the fuel rates of the enormous power-house units may never be attained by marine plants, it now appears fairly certain that eventually they will be sufficiently closely approached to justify the continued use of steam for the propulsion of large vessels.

Two Loeffler Boilers Ordered for Moscow

The Vitkovice Works of Czechoslovakia has recently received an order from the Technical Institute of Moscow for two complete Loeffler boilers to operate in Europe's largest high-pressure power plant. Each unit will supply 285,000 pounds of steam per hour in normal operation and a maximum of 350,000 pounds for short peak loads. The steam pressure will be 1900 pounds per square inch and the total steam temperature will be 932 degrees F.

The Moscow station, besides being the largest in size, is the only European power plant which operates at such high steam temperature and pressure. Russian engineers spent many weeks in Vitkovice studying the four Loeffler boilers in operation there. The following gives technical data relative to the boilers ordered for Moscow.

The feed water is brought to a temperature of 410 degrees F. in bleeder heaters, where it is raised to 485 degrees in economizers before it enters the vaporizer drums.

Saturated steam, leaving the vaporizing drums at 1900 pounds and 628 degrees, is delivered by the booster circulating pump to a radiant superheater consisting of U-shaped tubes. These line two adjacent furnace chambers on all sides with the exception of the ceilings.

It is planned to burn "Donez" coal of 12,200 British thermal units per pound lower heating value at first and to fire later on Moscow coal of 4900 British thermal units per pound lower heating value. These Loeffler boilers will burn the coal in pulverized form and means are provided to dry the raw coal before it is pulverized.

The pulverized coal is first brought into a bunker above the furnaces, whence the fuel is supplied to two rows of eight burners each, where it is thoroughly mixed with air of 485 degrees F. temperature. The double furnace chamber has a cubical content of 21,180 cubic feet, the radiant-superheater surface being 3120 square feet. After leaving the furnace chamber, the gases branch off at right and left and sweep in their second path the other heating surfaces of the boiler.

There they pass the resuperheaters (convection type), each of 10,760 square feet. These have 48 flat tube coils. Each tube is of 2.75 inches outside diameter. Steam leaving the radiant superheaters at 735 degrees is here raised to 932 degrees, after which about one-third goes to the turbine and two-thirds to the vaporizer drums.

Above the resuperheaters are the reheaters, which reheat steam at a pressure of 380 pounds from 535 degrees F. to 752 degree F. The heating surface of these reheaters is constructed similarly to that of the resuperheaters. Each measures 6886 square feet and consists of 32 flat pipe coils of 1.87 inch outside diameter. These are similar to those of the convection superheater.

Above the reheaters are the two economizers, each of 6080 square feet. Above these, in turn, are air-reheaters of Vitkovice Thermorex design.

Above the air preheaters, is the induced-draft plant formed by two suction fans, each designed for a maximum of 140,140 cubic feet of gas per minute with gas temperature of 410 degrees F. The delivery ducts of these fans enter a gas filter, whence the gases leave through short stacks.

Each boiler has five vaporizer drums 26 feet long with a diameter of 43 inches, besides one mud drum of the same diameter but only 6 feet 6 inches long.

For the circulation of the steam, each boiler has a single-stage, turbine-driven, centrifugal booster pump. It is able to move 1,100,000 pounds of saturated steam per hour.

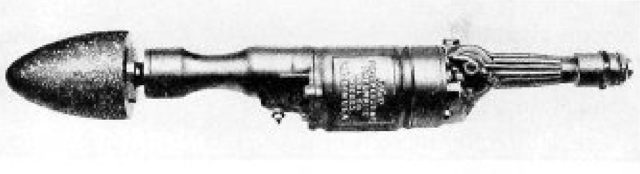
A more complete description of the Loeffler boiler appeared in an article by Arthur J. Herschmann published in THE BOILER MAKER, for November, 1931.

Cone-Shaped Rotary Grinder

The Independent Pneumatic Tool Company, Chicago, has added a high-speed rotary air grinder to its line of pneumatic tools. This size will be known as 255-X and is designed to operate cone-shaped grinding wheels.

This type of grinder is used by foundries and machine shops for cleaning castings of all descriptions and for grinding castings and forgings where the use of the ordinary emery wheel is not practical.

The speed of this Thor grinder is 10,000 revolutions per minute and it will operate cone wheels measuring 2¾ inches outside diameter. The weight of the machine is 8¾ pounds; length overall is 19¼ inches.

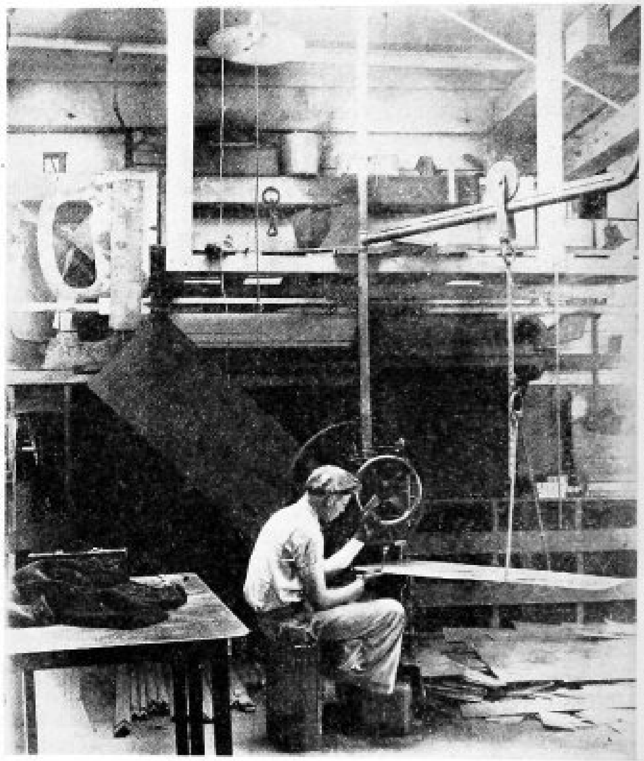


Independent high-speed rotary grinder

The driving collar of the 255-X is provided with right-hand thread so that the cone wheels now in use in shops and foundries may be used without requiring any changes or attachments.

Light Jib Crane for Punch Press

Shown in the illustration is a light jib crane which has proven useful, and in many instances saved the time of a helper, in the operation of the small punch press. The mast of the crane, made of 2½-inch pipe, is secured to the press by stirrup straps of ¼-inch by 3-inch steel. The end of the pipe fits over a 1-inch by 2-inch vertical stud, which is bolted to the top of the machine. The top end of the mast, is pivoted in a channel-iron section secured to the roof structure.



A jib crane of light construction is a useful facility for the operator of a small punch press

The arm is secured to the mast by means of a tee. It is turned up at the outer end, and a section of 1-inch angle iron is welded along the top of the arm to serve as a track for the trolley wheel. The outer end of the boom is braced and secured to the top of the mast by a light iron rod.

The trolley pulley is made of three circular disks of steel, welded together. The two outer disks are made slightly larger in diameter than the middle disk to form the flanges of the wheel. The trolley wheel is carried on the vertical leg of the angle. The trolley wheel stirrup is of ¼-inch by 2-inch bar with an eye bolt at the lower end. Secured to this eye bolt, is a ¼-inch by 1½-inch bar 3 feet long, which is punched with holes its entire length. An S hook is provided which can be inserted in any one of these holes to suit the height of the work. This feature is very aptly shown in the illustration.

What Can the Boiler Department Do to Reduce Pitting and Corrosion?

Following articles on this subject, appearing in the January and February issues, the present article covers further discussion of this problem by consulting engineers and metallurgists. In the January issue a committee report on pitting and corrosion, as prepared for the 1931 proceedings of the Master Boiler Makers' Association, was published. Discussion, in the February issue, by water service engineers of leading railroads, reviewed the practical steps taken to combat this trouble. The present article is from the steel maker's standpoint and from that of the heat exchange expert.

As the fourth to contribute to the discussion of pitting and corrosion in the 1931 proceedings of the Master Boiler Makers' Association Howard L. Miller, metallurgist, Republic Steel Corporation, Massillon, Ohio, outlined the subject as follows:

Since our last meeting at the Pittsburgh convention, the writer has visited many of the members in their own shops. There are some interesting experiences encountered in boiler corrosion, and the success in resisting it, that has come to our notice, would be interesting to the members of the Master Boiler Makers' Association.

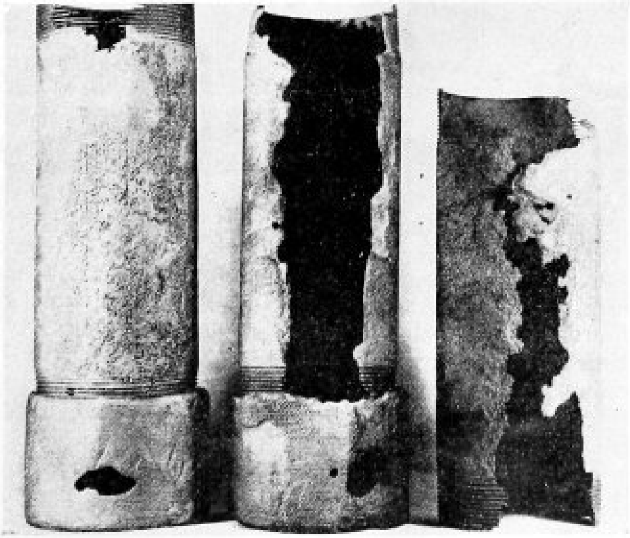
The railroads in the New England States have experienced failures of side sheets by cracking on the water side in the second or third row of staybolts above the mud ring. Grooving at the mud ring line is also encountered. These New England railroads use a large amount of surface water in their boilers. We believe that organic salts or acids from rotting pine needles and oak leaves or other vegetation is responsible for this peculiar type of corrosion cracking. This type of failure is distinctly separate from the cracking on the fire side due to overheating and unequalized expansion. Sheets which appear to be perfect on the fire side show bad cracks running from the staybolt holes on the water side of the sheet.

One of these railroads has experimented with nickel-steel sheets and also with copper molybdenum iron sheets. The copper-molybdenum iron sheets seem to have solved the difficulty as the oldest sheets in point of service have passed through their second shopping and are still going strong. About 30 locomotives on this particular road now are equipped with side sheets of this material and are showing up very well in point of service. Two other roads are applying side sheets of this same analysis.

In regard to corrosion of plates in tenders, two distinct types of trouble are encountered. In the coal bin section, the corrosion is general in character and is caused by the action of the hot water from squirt hose coming in contact with the sulphur in the coal, and with oxygen from the air, forming sulphuric or sulphurous acid. This acid in the water runs often as high as 5 per cent in drippings from coal cars on sidings and from tenders. This acid eats away at the shovel sheets, front sheet and side sheets of coal bins and causes frequent renewals of these plates, especially where high sulphur soft coals are used.

The copper-molybdenum iron has been used for drain piping in coal mines for years and is the only commercially practical material that will withstand the action of dilute sulphuric acid in mine waters.

Three pieces of 4-inch pipe were laid in a ditch along the W. & L. E. tracks leading from a mine at Coshocton, O., where drainage from a mine flows through the ditch. In eight months time the wrought iron pipe and the common steel pipe were in the condition on the bottom as shown in the illustration. The copper-molybdenum iron pipe was thinned down but still able to hold water.



Samples of copper-molybdenum iron, wrought iron and common steel pipe after corrosion tests

One of the New England railroads has several tenders built complete from plates of this composition. Last reports indicate the life of the plates will more than compensate for the increased cost of the plates.

There is no one steel that will resist all types of corrosion, nor can we find any one chemical that will successfully treat all types of water. each location or condition must be studied separately and where conditions seem to be parallel with conditions in some other place, then the method successful in the other case may prove also successful in the case considered. If not, other methods of attack must be tried.

In regard to corrosion and pitting of boiler tubes, where corrosion is general and covers large surface areas on the tubes, tubes should be selected that contain the proper alloying elements to increase the general resistance of the metal to the corrosive elements in the water.

Where spot corrosion or pitting is encountered, this is, we believe, due to imperfections in the surface of the metal, such as mill scale rolled into the surface, or iron oxide in the surface metal, carried down from the original ingot casting.

We have perfected a process that includes pickling the finished tube in acid to remove all mill scale and this process produces a tube that is free from included oxides in the surface and presents a practically uniform surface condition to resist the general corrosion and prevent localized electrolytic reaction, which results in pin holes. These extra precautions are showing up in field tests, in bad water districts, to be well worth the added effort and expense.

We are extremely sorry that the Master Boiler Makers' Association could not meet this year and we desire to express the conviction that the importance of these meetings will again be recognized and that 1932 will witness one of the best meetings from a technical standpoint that will ever be held. It has been our pleasure to be with you at two conventions, and the interchanging of experiences and broadening of viewpoint of the industry as a whole, have been of very great benefit to us. We feel that whatever of benefit we have been able to present to you as steel makers has been more than balanced by the insight we have gained of the troubles and difficulties of the men on the firing line, the master boiler makers themselves.

Following Mr. Miller's review in the proceedings is one by C. A. Seley, consulting engineer, Locomotive Firebox Company, which follows:

The above subject has received much attention from this body, as well as from other railroad associations, clubs, and the press. The invitation of your secretary to contribute another paper for this year's proceedings is much appreciated, although it is difficult to offer anything new on the subject. At the present time I am engaged in test work on a Lake ore carrier, in research work on water circulation in Scotch marine boilers, and therefore am away from my office library and files of information.

To what extent the boiler maker can aid in reducing corrosion and pitting troubles is, in my opinion, limited principally to the craftsmanship in handling the materials employed in proper fashion, so that, whether in manufacture or repair, the character of the material used will not be altered, by the fabrication methods or working, as to deteriorate its quality and suitability for its purpose. He may, however, indirectly help the matter along for more favorable conditions and care of boilers when they are not directly under his charge and control.

Roughly, boiler life may be divided into three eras. First, while in the shop or house for work or repair; second, while in operation on the road; and third, while being handled at the terminal after service.

Taking the first era, that in shop or house; evidently corrosion and pitting does not get in its work at that time unless there is extensive laying by, with the boiler not properly drained or protected. In such a case, the boiler maker may be asked for help or advice, to secure proper care of the boiler.

The second era, while in operation, is the period when corrosion and pitting are most active, due to the physical and chemical reactions brought about by general operating or handling conditions, and the character of the various water supplies used in producing steam This branch of the subject has been most extensively gone into and covered by papers and discussions too numerous and extensive to be included at this time.

From all these papers and discussions many good suggestions and ideas have extended knowledge and better practice in the matters of water purification methods, the advantage of proper water circulation in the boiler, means of extraction of dissolved gases in the water and of protection from those remaining, etc.

The third era, the terminal handling after service, while not much concerned in corrosion and pitting, is on many railroads the hardest period of boiler life. Except at relatively few terminals, where there are hot water washing and steaming-up facilities, the frequent handling at ash pits, blowing down, washing, filling and firing-up are a succession of temperature changes that are more severe in their effect on the boiler structure than they receive in service. On the theory that time is the most precious commodity on a railroad, the boiler is generally put through the above operations as quickly as possible, regardless of the fact that steel is one of the most sensitive of metals and the structure of the boiler is devoid of the elasticity safely to receive such treatment. The consequences are later developed in broken staybolts, leaky flues, incipient cracks and internal stresses, and it is no wonder that the locomotive boiler is the most inspected and repaired steam generator in existence.

The boiler maker, of all men on the railroad, knows most of the results of such treatment and is generally not in position to avoid the excessive range of temperature changes incident to terminal handling but wherever he can influence proper care and the introduction of methods and means to that end, it is his manifest duty to do so.

Recent Failures of Stationary Boilers*

The engineer of a large modern laundry in Hollywood, California, was severely injured recently when a fuel explosion in the furnace of a gas-fired power boiler wrecked the setting and blew out a brick building wall. He was not only burned about the head and hands by the sheet of flame that swept through the boiler room, but was hurled twenty feet through a window.

The steel columns supporting the boiler and the 18-inch steel girders between these columns were so badly bent and twisted that an entire new supporting structure had to be installed. Deflection of the support caused the boiler to drop about 3 inches, breaking the blow-off pipe, the safety valve "Y," and several smaller pipe connections. A companion boiler also was badly damaged, as the explosion ripped open the breeching and carried away the safety valve. It took thirty-six hours to place the spare boiler in operating condition. The property damage was estimated at $6000.

According to the engineer, he closed the fuel control valve at the boiler as well as the valve at the meter outside the building when shutting the boiler down the night before. In some way gas leaked through these valves during the night and filled the furnace and gas

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* Published through the courtesy of *The Locomotive*, of the Hartford Steam Boiler Inspection and Insurance Company.

passages. The explosion took place when he attempted to light the pilot.

A still more serious furnace explosion occurred in California last summer when the engineer of a paper products plant in Los Angeles lost his life. As in the case just described, the breeching was destroyed and the brickwork of the setting knocked down. Caught under the brick wall as it fell, the engineer was crushed. The property damage was considerable. An estimate placed the plant's financial loss at $10,000.

It was thought that perhaps the accident was caused by the engineer permitting the gas pilot to flow too long before igniting it, but as the victim was alone in the boiler room at the time this supposition could not be checked. It seems more probable, however, that there may have been a leak in the gas or oil lines that resulted in the furnace filling up with combustible vapor, as in the case previously described.

To minimize the danger at the time of lighting the pilot on either gas or oil-fired furnaces, the operator should take particular pains to see that the damper is open. Before approaching the furnace with a torch he should make sure that the furnace and combustion passages have been well ventilated. With oil-fired furnaces he should see that there is no accumulation of oil that may have leaked into the combustion chamber. Tips and nozzles of oil burners should be looked over to see that they are clean and securely connected to the oil lines, and with installations where there is an arrangement for preheating the oil, this temperature should be right before the burner is lighted.

Recently, when asked to issue insurance on an oil-fired heating boiler in a theater, the company found that the boiler had sustained several furnace explosions. An inspector reported that because of the presence of a deep stage between the boiler and the stack, the smoke flue had been installed with a deep dip so as to pass below the lowest point of the stage foundation. Thus right at the boiler there was a high pocket in this flue that permitted the accumulation of combustible gases. Such an installation, which, fortunately, is unusual, is admirably arranged to produce furnace explosions not only because of the pocket where gas can collect but because the depression in the flue prevents the stack from creating a draft sufficient to serve the furnace. In this particular case it was impracticable to relocate the flue so that it would pass in a horizontal plane from the boiler to the stack, but the danger was minimized by the installation of a small motor-driven induced draft fan.

* * *

The boiler of what was purported to be the largest bootleg whisky plant ever discovered in Southern New Jersey exploded shortly before daylight on December 2, killing two men instantly and, like a huge projectile, passing through and demolishing a nearby farm house in which several persons narrowly escaped injury.

Investigators had no difficulty in agreeing on the cause of the accident. A large steel heating boiler of the firebox type, intended for a pressure of not more than 15 pounds, had been used to furnish steam for two stills at a pressure of from 60 to 80 pounds per square inch. At the time of the accident the pressure may have been even greater, for although the boiler had a steam gage on which pressures up to 200 pounds could be read, there was not a sign of a safety valve. It seemed evident that the persons responsible for the installation had little knowledge of boilers.

Even though its speed must have been checked when it encountered the house, the boiler traveled 500 feet before coming to rest. The shell was found intact, but the furnace sheet and front head had been torn out completely, stripping the staybolt threads.

The distilling plant had two very large stills. One was rated at 15,000 gallons capacity and the other at 10,000 gallons. Federal prohibition officers estimated the plant cost between $50,000 and $75,000. The stills were in a barn, concealed behind a wall of hay, but the boiler was in a shed some distance away. Apparently the victims were in the boiler shed at the time of the blast. One was blown 50 feet, and the other was hurled into a field 200 feet away.

The boiler tore a hole 20 by 25 feet in the upper story of one wing of the T-shaped farmhouse. It passed directly over the bed in which a farm hand was asleep, miraculously sparing his life but taking with it a pair of trousers in which he had secreted a roll of money representing his savings. These trousers, with the money still in the pocket, were found later hanging to the topmost limb of a tree some distance from the house.

After passing through the wing, the boiler sliced away the wall of the second-story hallway adjacent to rooms in the other wing in which six children were sleeping. They were hurled from their beds but were not seriously injured.

* * *

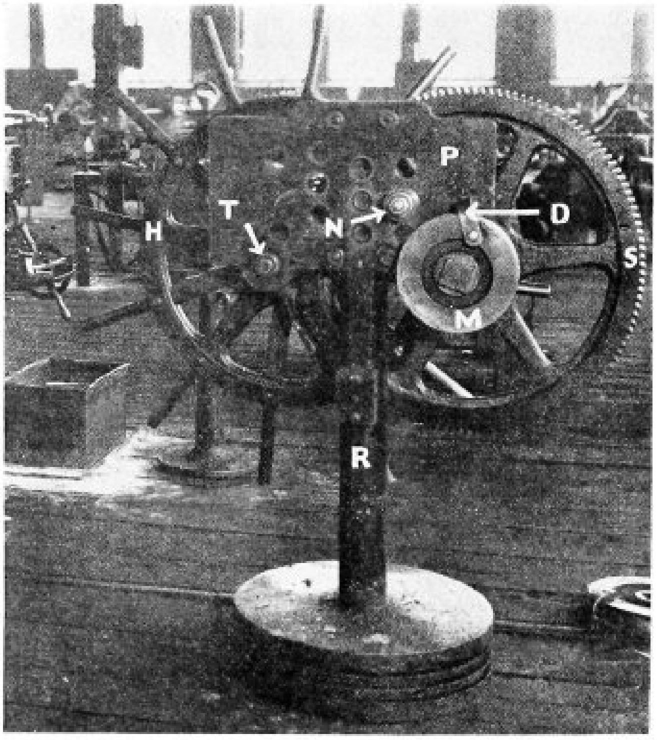
Two significant factors figured in the explosion of a small digester that recently came to our attention. The vessel was built with a disk head, the edge of the saucer-like disk being welded to the edge of the shell in such a way that the strain caused by breathing of the head was made to concentrate at the weld. To make matters worse the vessel had no safety valve to guard against a pressure greater than that for which it was intended. Full dependence was placed on the effectiveness of a reducing valve which, moreover, was provided with a by-pass for use when starting the cock.

The explosion occurred when this by-pass was accidently left open too long. However, in the absence of a safety valve, the accident might have come about with the by-pass closed, for reducing valves have often been known to get out of order.

Pipe-Bending Machine

A rugged, portable, pipe-bending machine, designed and built at the Milwaukee shops of the Chicago, Milwaukee, St. Paul & Pacific, is shown in the illustration. It consists of a scrap piston head and rod with the nut burned off to give the head a flat bearing on the shop floor, the pipe-bending mechanism being mounted on the upper end of the rod.

Piston rod *R* is milled on one side to give a flat bearing surface for the forged-steel adjusting plate *P*, 14 inches by 27 inches by 1½ inches thick, drilled with 1¾-inch holes to accommodate the idle roller *N* in various positions as required, depending on the size and location of the pipe bend to be made. The adjusting plate is firmly secured to the piston rod by means of two U-clamps, the four recessed nuts on the clamp ends being shown in the illustration. Large roller *M*, provided with a 2-inch half-round groove in its circumference, is keyed to a 2½-inch shaft, which has a well-lubricated bearing in adjusting plate *P* and is keyed to the cast-iron spur gear *S*. This is a 4-pitch gear, with 134 teeth, 3-inch face, outside diameter of 34 inches and pitch di-



A convenient, portable pipe-bending machine

ameter of 33½ inches. A 4-pitch, 16-tooth pinion, of 4½-inch outside diameter, engages gear *S*, being pressed into hand wheel *H* and revolving freely on handle post *T*. Wheel *H* is built up of a cast-iron wheel with 12 pipe handles riveted in place as shown. Strap *O*, attached to large roller *M*, engages the pipe, holding it against the roller during the bending operation.

Ample power is available in this machine, with a gear ratio of 33½ to 4, to bend pipe up to 2 inches in diameter. Pipe sizes up to 1½ inch can be bent cold to a minimum radius of 4¾ inches. An important feature in the satisfactory bending of pipe with this type of machine is to make the groove in roller *M* slightly deeper (about ⅛ inch) than one-half the diameter of the pipe. In other words, the groove is slightly elliptical in shape and, when maximum pressure is exerted tending to flatten the pipe, it is simply forced further into the groove without being deformed. By means of special jigs, made to hold superheater unit ball joints, superheater pipe can also be bent cold on this machine and to a very desirable degree of uniformity. The pipe bender is portable and can be readily moved by means of a crane or truck to any point in the pipe or erecting shops most convenient for the work in hand.

Methods of Welding Boiler Drums

The following abstract of a paper by Dr. Comfort A. Adams of the School of Engineering, Harvard University, published in *Mechanical Engineering*, describes various methods of welding with particular reference to those employed in boiler manufacture.

The author shows that flash welding is not properly suited to the manufacture of large boiler drums, because of the enormous cost of equipment. Spot welding is difficult to apply to heavy plates, and is rarely used with plates thinner than ⅜ or ½ inch. Projection welding is likewise rejected. Resistance welding is used for tube-welding manufacture, particularly for thin-walled tubing.

The author discusses next the various forms of arc welding and the various methods of protecting the metal from oxidation and absorption of oxygen. This latter is done by the use of coated or covered electrodes, or by protecting the metal in its passage across the arc by a stream of combustible gas around the arc, usually hydrogen or of the hydrocarbon variety. The oxy-acetylene flame when properly adjusted produces its own protective elements in the form of the products of combustion. As regards the strength of the weld, the following is stated:

The physical properties of a piece of steel depend largely upon its chemical composition and upon its heat history. All metal is crystalline in its structure. When steel cools from the molten state it crystallizes, and these crystals combine in groups to form what are commonly known as grains of varying shapes and sizes, and the size of these grains is largely a question of their heat history, that is, the temperature to which the metal has been raised, the time it has been held there, the rate of cooling, etc. Under the conditions of cooling in the ordinary arc weld the grains are very coarse, and, although the metal may have ample tensile strength, it is almost universally brittle. By proper heat treatment such metal may be refined as to its grain size, with an accompanying change of properties, particularly as to its ductility.

It just happens that in the making of arc welds in heavy plates it is practically necessary to deposit the metal in layers, in which case each layer puts the layer below it through an ideal heat treatment as far as grain refining is concerned. This is particularly true in heavy-plate welding where large currents are used and where sufficient heat is thereby transmitted to the underlying layer to carry it through this refining process. It is obvious, however, that the depth of penetration of the refining heat is limited, and therefore that it is necessary to restrict the thickness of the layers to that point.

The author brings up the following important question: When is the slightly lower ductility of one of these welds without grain refinement any serious handicap to the safety of the resulting structure, assuming of course, that the tensile strength of the joint is equally high in both cases (and this is a fact)? He does not answer the question completely, but states as his personal opinion that there are very few structures in which this lower ductility has any influence on their satisfactory service. In the case of arc welding with either bare or covered electrodes, great care is necessary to avoid gas or slag pockets or the finer type of porosity, with the consequent danger of fatigue failure starting at this point. The following passage is quoted verbatim: "You have doubtless been told that welds are being made which are superior in every respect to the parent metal, and this is in general true, but I wish to assure you that this is not the whole story, and that it is not a simple matter to produce a perfectly homogeneous non-porous weld metal. If the pores are very small and uniformly distributed, they will probably not affect the satisfactory functioning of the resulting joint in any way, but the larger pockets are distinctly dangerous, and these do occur occasionally in the best-regulated families."

The author refers in this connection to Professor Moore's report on breathing tests of cylindrical shapes, and reference to the A.S.M.E. Boiler Code Committee requirements of X-ray tests should prove the absence of slag or gas pockets in the welded joints.

Arc Welding for Pressure Vessels*

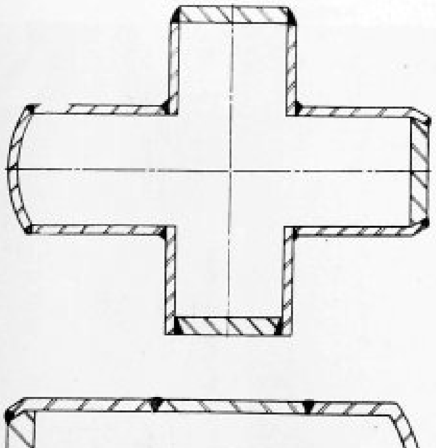
By G. G. Landis†

The interest reflected in arc welding by the Boiler Code Committee of the American Society of Mechanical Engineers demands that these articles consider the welding of pressure vessels. We have already seen that high-pressure piping and refinery equipment is largely welded and many boiler makers, power plant engineers and heating experts have used welding extensively.

So great are the economies of arc welding that in many instances the process has been used on pressure vessels without considering the insurance company's or the A. S. M. E. requirements. Arc welding with proper precautions and care has proven successful in these applications.

The requirements for successful welding in any field or application are:

1. A good design by an engineer who thoroughly understands welding. Actual shop experience is recommended.
2. Good welding equipment which insures proper fusion, even deposition and easy operation.
3. A good welding crew under the supervision of an intelligent, conscientious and experienced foreman.
4. The proper welding process and procedure for each job. This will of course be reflected from the other three factors.



Figs. 1 and 2.—Types of electric welded pipe used for test

Fig. 3.—Crossed pipe sample failed at 3100 pounds

Fig. 4.—Pipe with welded heads failed at 5100 pounds

While these points may seem obvious, it is our experience that they are often neglected in whole or in part, due to prejudice, tradition or economy. Failure of any welded unit can be traced to one of these factors, all of which can be controlled.

Shop experience, a few weeks in a good welding school, should be part of the training of every welding engineer. The possibilities and limitations of the process, inspection of welds and so forth are best learned in that manner. We do not suggest that every engineer become an expert welder operator, but such basic knowledge is advantageous. A striking example of this occurred when Maurice Taylor, of our technical staff, went to Kobe, Japan, for the Kawasaki Dockyard Company.

The Japanese are great believers in welding. Japanese engineers have two or three engineering degrees and a profound knowledge of all the factors bearing up the use of the electric-arc process of fabrication. Economy

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* Fourth article of series dealing with arc welding of tanks and pressure vessels.

† Chief engineer, The Lincoln Electric Company, Cleveland, O.

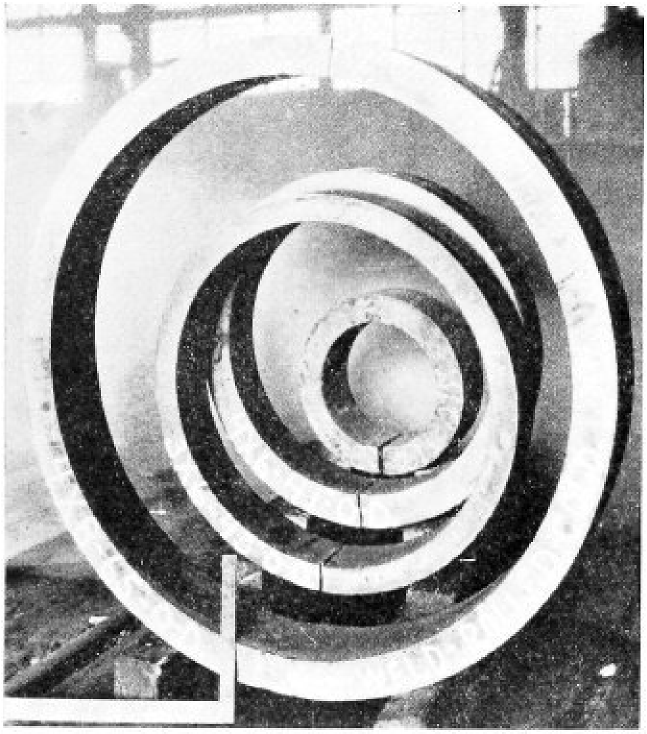


Fig. 5.—Rings welded in protective atmosphere

and efficiency are hampered, however, by their inability to get into the shop and actually see how and why certain operations should be varied.

In this country arc welding was first propagandized by the welding operators, who spread the advantages of the process to foremen, superintendents and engineers. As editor of a publication devoted exclusively to welder operators it has been a pleasure to meet and know thousands of craftsmen. No other type of workman is so conscientious, so interested in his process and so enthusiastic over learning more about it. While the human factor is still important in welding, the general class of operators is so high that we usually do not consider this human factor at all. Operators should be tested and their work inspected periodically of course, but we have found that the first report on a flaw or improper weld usually comes from the craftsman himself.

For the welding of pressure vessels the shielded arc process should be used. Engineers, metallurgists, chemists, welding technicians and research engineers seem agreed on this one point at least. It is phenomenal that with this process, and the proper technique, arc-depositing metal has better physical characteristics than the welding rod or base metal.

The proper deoxidizing gas used to shield the arc will actually reduce the oxide content of the molten steel, resulting in corrosion.

Tensile strength of 65,000 to 80,000 pounds per square inch; ductility of 25 percent elongation in 2 inches; better resistance to corrosion than rolled steel, all of these are assured with the proper application of the shielded-arc process.

Another depreciating statement is that the weld made be strong but that the plate is weakened because of the heat of welding. The following is taken from a report of a demonstration made before representatives of the Hartford Steam Boiler Inspection and Insurance Company.

The tests were made on 8-inch welded steel fittings, welded by The Lasker Boiler and Engineering Corporation, and consisted of hydrostatic bursting tests. This company submitted one 8-inch electric welded cross with various types of heads welded in the outlets as per Fig. 1; also one piece of 8-inch pipe with welded heads as shown in Fig. 2. Fittings were tested to destruction in an attempt to determine a weakness in the design of heads.

Pipe used in these test specimens was 8-inch extra strong wrought steel pipe ½ inch thick, having 0.12 percent carbon, 0.30 percent manganese, 0.045 percent phosphorous, 0.06 percent sulphur content. All welds were made by the electric-arc method using shielded-arc electrodes.

Specimens were tested under hydrostatic pressure to destruction.

Microscopic examination was made to determine whether the pipe metal was in any way affected by the welding temperature. There was practically no difference in the grain structure of specimens taken adjacent to the weld and at the farthest point away from the weld, thus showing that the extensive welding had no effect on the strength of the pipe.

The proportional limit of the welded steel cross was reached at 1700 pounds per square inch pressure. Failure occurred at 3100 pounds pressure, see Fig. 3. Pipe with welded heads failed at 5100 pounds pressure, see Fig. 4. Falures in both cases occurred in pipe material. The welds did not show any signs of distress.

Microscopic examinations of sections in the welded fittings indicated that pipe material was not affected by the welding temperature.

This test, which was primarily made to determine the best design of welded head, did not reveal any light in this respect as none of the welds failed.

On heavier units than these small fittings, microscopic studies of grain structure show little if any difference caused by the heat of welding. Test coupons cut from thick samples usually fail far from the weld. An interesting example of the fine quality of metal deposited in a protected atmosphere is shown by Fig. 5. These rings are composed of four inch steel bars, butt welded and then rolled. A weld of this size is very large and numerous beads of weld metal must be de-

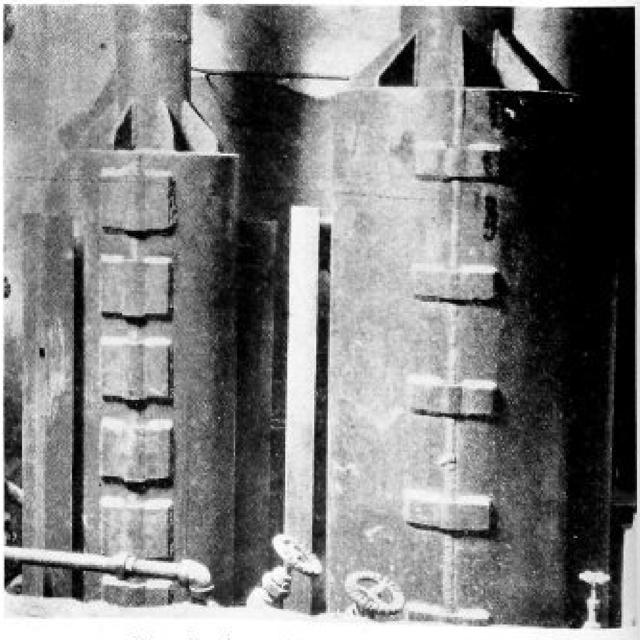


Fig. 6.—Arc-welded steam separators

posited. Unless practically perfect fusion had occurred the welds could not have stood this test.

Fig. 6 shows an interesting application of arc welding on pressure vessels. These steam separators were built on rush by the United District Heating Company. The separator on the left is constructed of ¾-inch plate to carry 375 pounds pressure of steam at 550 degrees F. All the baffles and fittings are fused into place by arc welding. In reference to the building of these units E. R. Benedict of the United District Heating Company says:

"We have used welding for 12 years. When the need for these separators arose welding was the obvious solution. Personally, we would not have used the reinforcing straps. Those welds will hold."

The shielded-arc process is used manually with a modern, variable voltage welder and a heavily coated electrode. This coating of special composition burns in the arc less rapidly than the electrode melts, forming in effect a crucible which protects the arc for almost its entire length. As the edges of this coating burn in the arc a gas is given off which serves (1) to protect the molten metal from the harmful effect of the oxygen and nitrogen in the air, and (2) to reduce the oxide content of the metal at the point of fusion. The residue from this coating forms an easily removable slag which protects the hot weld metal while in the molten state. Higher welding currents and voltages are used with this electrode than so-called bare electrode, and welding speeds increased proportionately. The increase in speed alone justifies the slightly higher cost of this process, while there is no comparison in physical characteristics.

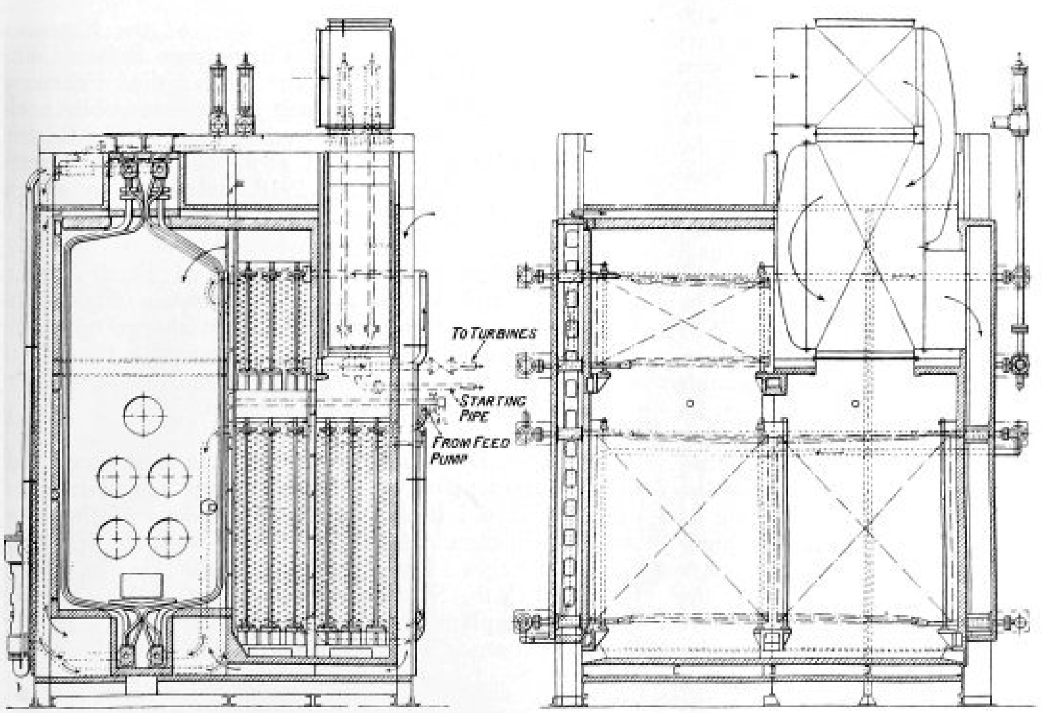
It is perhaps beyond the scope of this article to give an exact procedure of welding on high-pressure work as this will vary, but it is sufficient to say that shielded arc welds made under proper procedure will fulfill the most exacting requirements.

The use of arc welding is bound to spread in the construction of tanks and boilers for the two reasons which determine the adoption of any new process . . . an improved product . . . at lower cost. Arc welding fulfills these requirements in every tank and boiler application, from range boilers, seamless and leakproof, produced at the rate of over 12 an hour with three operations; 1000 gallon tanks welded with a labor cost of $4.35; to high-pressure boiler drums of thickness too great for riveting and too large and costly for the forging and piercing mills.

Super-Pressure Boiler Installation of the Steamship Uckermark *

Since 1926 the Hamburg-American Line has been interested in the subject of increasing the steam pressures utilized on its vessels. The repowering of ships of the *Albert Ballin* class with a steam pressure of 411 pounds per square inch at 707 degrees F. temperature was the first step in this direction made by marine engineers in Germany.

Efforts in this direction were so successful that recently experiments have been made with an installation of super-pressure boilers of the Benson type, which have been installed on the steamship *Uckermark*. In order to limit the risk in connection with the trial of the Benson boiler, it was decided to retain the old boil-



Front and side elevations through Benson boiler installed on steamship Uckermark

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* Prepared from a description of the complete machinery installation of the steamship *Uckermark*, by Dr. Ing. E. Goos, appearing in *Shipbuilding and Shipping Record*.

er and turbine plant from one of the *Albert Ballin* class vessels operating at 220 pounds per square inch, replacing only one double-ended boiler with the Benson boiler, which was designed to generate all the steam needed by the main turbine.

The principle of the Benson boiler is the raising of the pressure and temperature of the steam beyond the critical stage. It has been found by research and calculation that if water with a pressure of 3230 pounds per square inch is heated to about 707 degrees F. its state of aggregate is perfectly changed. Beyond this limit the steam exists only as gas. After having passed the above limit we are sure that this gas does not contain any molecules or drops of water.

Sections through the boilers are shown. As to the materials used for the boiler, it should be mentioned that the pipes of the radiating part exposed to the highest temperature of the fire gases consist of steel containing 0.4 percent molybdenum. The 32 headers of the radiating part are made from solid ingots of Siemens-Martin steel and the pipes have been welded on to them, the headers being connected by the pipes of the downward water circulation, which are also welded on. All headers are connected in series, the 11 pipes of each header run parallel and their number is fixed according to the steam capacity of the boiler. The length of the water way in the pipes depends upon the number of headers connected in series and must be long enough to ensure the transformation of every water particle into the form of gas. The pipes for the steam-superheater and economizer are made of seamless solid-drawn Siemens-Martin steel. The air heater of the pocket type is made of sheet iron and the plates have been welded together with particular care to avoid the loss of combustion air. The flue gases are drawn off by a special fan electrically driven and placed above the air-heater. The combustion air is supplied by a centrifugal fan drawing its supply from the engine and boiler casing. It is then forced through the heater and by special ducts alongside the boiler is led to the oil burners.

The arrangement of 10 burners was found on the first voyage to be little suited for ensuring satisfactory combustion and the equal distribution of stresses on the pipes in the radiating part. The front walls of the boiler were therefore altered to make four burners on each front. The downcast watertubes of the radiating part are protected by nichrotherm plates, these having been found to withstand the highest temperatures efficiently. The lower headers for the pipes in the radiating part, as well as the two front walls, are protected against the damaging effects of high temperatures by bricks possessing a high melting point.

The water delivered by the high-pressure feed pumps passes, firstly, part 1 of the economiser; secondly, part 1 of the radiating part; thirdly, part 2 of the economiser; and fourthly, part 2 of the radiating part. The critical point of the water will be reached in part 2 of the economiser. The steam possesses at the end of part 2 of the radiating part a pressure of 3300 pounds per square inch and a temperature of 752 degrees F., and therefore is somewhat superheated. After this it is passed through a steam-reducing valve to be reduced to nearly the working pressure of the high-pressure turbine, passing then through a superheater consisting of several coils. On its way to the nozzles of the high-pressure turbine a drop of about 147 pounds per square inch pressure takes place, so that the steam enters the turbine with a pressure of about 882 pounds per square inch and 824 degrees F. temperature.

At first the division of the several coils or groups of the pipes was carried out in such a way as to have the critical point of the water located in part 1 of the radiating part, but it was found that those pipes in which the transformation from water into gas took place were not properly cooled and, therefore, became overheated and burst. This bursting of the pipes could not be regarded as an explosion as only the oil burners were blown out and no other harm was done. Owing to the very small amount of water that could only gradually flow to the opening caused by the bursting of the pipes, the steam spontaneously generated had ample time to escape through the funnel.

The special characteristic of the Benson boiler is to generate the steam in pipe systems connected in series without having large water drums, and, therefore, secures the safety of the boiler from explosion. Since the alteration in the connection of the pipe systems not a single pipe has burst, a fact which proves that the thoughtful consideration leading to the alteration mentioned before was on sound lines.

The weight of the Benson boiler complete with all accessories is about 100 tons, of which the boiler itself is about 32 tons; the grids of angle iron for fastening the boiler and casing are of 35 tons weight; and the brickwork, furnace fronts, insulation, valves, etc., are about 32 tons weight. The water or steam content altogether weighs only about 1 ton. The double-ended boiler which has been taken out in order to provide the space for the Benson boiler is of about 160 tons weight with water, whereas its steam generating capacity is only one-third of that of the Benson boiler.

The total heating surface of 5546 square feet is divided as follows:

Radiating parts 1 and 2	1348 square feet
Economizer parts 1 and 2	3400 square feet
Superheater	798 square feet

The floor space of the Benson boiler is about 19.7 feet by 14.8 feet; its height is about 19.7 feet, and it is able to generate steam equivalent to about 6000 shaft horsepower.

Business Notes

Raymond Newcomb, vice-president of the Kewanee Boiler Company and of the Fitzgibbons Boiler Company of New York and Oswego, N. Y., died February 16, at Selma, Ala., as the result of an automobile accident near that city. Mr. Newcomb was born in Boston in 1898 and was graduated from the Massachusetts Institute of Technology in 1918. He was an ensign in the navy during the later months of the war.

Fred Archer, formerly president of Fred Archer, Inc., has been appointed to the Philadelphia office of the Lincoln Electric Company. He is in charge of motor sales for this office. Mr. Archer was with the White Motor Company and the Kelly-Springfield Tire Company before forming his own concern.

Joseph T. Ryerson & Son, Inc., has announced that the management of its St. Louis plant has been taken over by Harold B. Ressler, vice-president of the company at Chicago. Mr. Ressler, who was the St. Louis plant manager for 15 years, will be in direct contact with the St. Louis organization, making regular visits to the plant and the St. Louis territory. In Mr. Ressler's absence, R. B. Wilson, manager of sales of the Ryerson St. Louis plant, will be the senior resident executive.

Care and Maintenance of Locomotive Boilers*

By J. W. Hobson

Like the human body, a locomotive is subject to petty ailments and diseases which may become chronic. Some of these may in certain cases be inherent defects due to faulty design and bad workmanship when first constructed, and in consequence may be incurable unless the expensive operation of reconstruction is performed. Others are without doubt brought about by neglect and unsatisfactory working conditions, which, in many instances, can be improved and a cure effected. Further, locomotives are subject to injury by accident and become normally worn out as age creeps on; but here the simile ends, as, by judicious overhaul and renewal of parts where necessary, rejuvenation is achieved and further period of useful working life obtained.

The shell of the boiler is liable to various defects, such as internal (and sometimes external) corrosion, blisters, grooving, and cracks. The most common is, of course, internal corrosion or pitting, which may be caused by gases, carbonic acid and oxygen, being dissolved in the feed water, or by waters which contain acids, or again by others in which there are corrosive mineral constituents. Signs of pitting are frequently evident in the bottom of the boiler barrel. Grooving may be looked upon as a form of internal corrosion. It is a serious trouble, generally occurring in close proximity to the seams or at the foundation ring near the corners.

External corrosion also frequently occurs, due to slight leakages and the lagging holding the moisture. Severe deterioration is also experienced in the vicinity of mud doors, etc., in addition to wasting round tube holes in the smokebox tube plate where leakage has taken place. These troubles, being largely local, can be remedied by thoroughly scaling and cleaning, followed by electric welding, by means of which the parts affected can be restored to their original thickness and strength.

In many cases, the application of a carbon compound paint applied to the interior of the boiler has been found to resist pitting and the adherence of hard scale successfully. Electrolytic action sometimes accounts for the wasting of steel in proximity to copper, brass, or gunmetal. In isolated bad cases, it has been found advantageous to substitute steel fireboxes and tubes in place of copper and brass respectively.

An interesting instance of this nature was recently brought to the author's notice. A rapid wastage of the rivet heads inside the barrel occurred at the point of attachment of the smokebox rings in proximity to screwed brass mud plugs in the tube plate. Steel plugs were substituted by the owner's engineer. This is a practice to be deprecated, as such plugs may rust in if not frequently removed, and thereby cease to fulfill their legitimate purpose. It might be added that, in this instance, it was not a question of neglect, as the boiler was thoroughly washed out at intervals never exceeding a fortnight, and on examination the strange feature revealed was that the steel rivet heads alone had wasted, the steel plate had not been attacked in the least.

As may be expected, firebox troubles are many and varied, and of these probably that of broken stays is most frequently encountered. It should be appreciated that when a stay breaks extra load is thrown on the adjacent stays; hence the necessity of keeping a watchful eye on such an eventuality and replacing the broken stay without delay. To ensure prompt attention, some users have the firebox stays drilled with a small hole up the center from the outer end, as it is usual for these stays to break near the outer casing plate.

A large bulge on the inside firebox plate proves beyond doubt that stays are broken, whereas small bulges between the stays indicate overheating often due to deposit on the waterside.

Direct stays are subject to corrosion (Fig. 1), and when the water is very bad, steps should be taken to protect them. A well-known method much to be recommended is to encase stays in piping filled with cement (Fig. 2). In many cases wastage occurs within an inch above the firebox crown; and gunmetal sleeves or ferrules about 4 inches long screwed down the lower portion of the stay, making a close joint with the copper, have been adopted in such instances (Fig. 3). The former method, moreover, is probably the more effective.

When severe wastage occurs in the firebox, cover patches should not be used, but the defective part

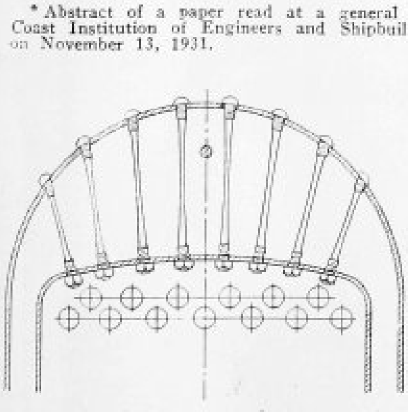

Fig. 1.—Direct stays showing effect of corrosion

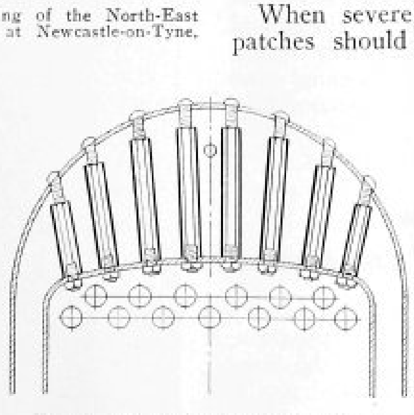

Fig. 2.—Protection of direct stays by piping

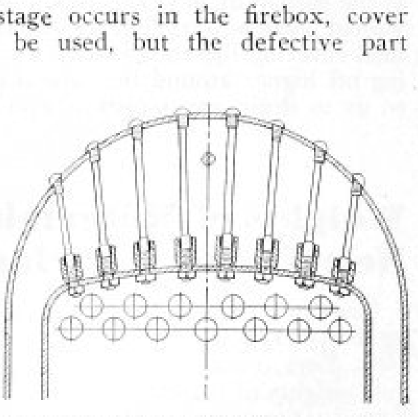

Fig. 3.—Protection of stays by gunmetal ferrules

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* Abstract of a paper read at a general meeting of the North-East Coast Institution of Engineers and Shipbuilders, at Newcastle-on-Tyne, on November 13, 1931.

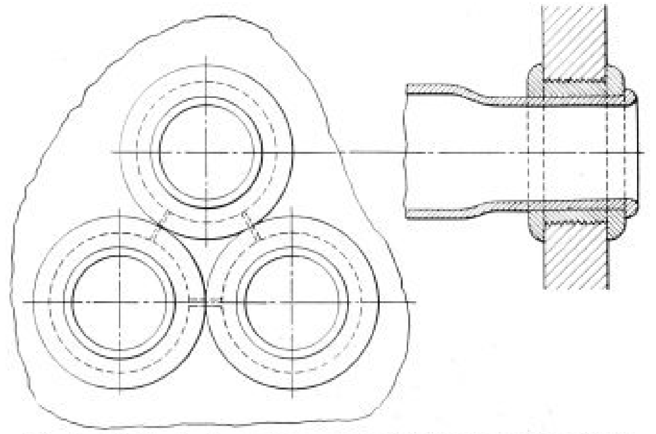


Fig. 4.—Method of repairing cracked bridges of tube plates

should be cut out and a riveted patch put on; otherwise the patch should be bedded on carefully and secured by set bolts about 2 inches pitch.

It is often necessary to repair firebox tube plates, due to the bridges between the tubes being cracked, through over expanding. To perform this operation satisfactorily, the tubes are removed and the tube holes affected are opened out a little larger and tapped with a fine thread; bushes are then screwed in and beaded over on each side (Fig. 4). If done properly, this is a perfectly satisfactory repair, and no further attention is required throughout the remainder of the life of the tube plate. The tubes, of course, should be reduced in diameter at the ends to correspond to the size of the bush.

When tubes appear to require frequent expanding the cause should be ascertained. There are numerous cases of tube plates being irreparably ruined by the frequency of expanding, due to leaky tubes, which may be caused by suddenly opening the firedoor repeatedly, constantly overloading the engine, too frequently using the blower, or by cooling down the boiler too rapidly for cleaning-out purposes.

In the case of extensive repairs, the boiler should, after completion, be subjected to a hydraulic test pressure of 50 percent above the intended working pressure, and careful gagings should be made before, during, and after such tests, particularly as regards the deflection of the firebox crown.

There are two essential features to ensure smokebox efficiency and to avoid deterioration, the first being airtightness, and the second cleanliness. Wastage frequently occurs in the lower part of the tube plate, due to moisture from the exhaust steam and leakage from mud doors, etc., combining with the fine ash, which often contains traces of sulphur and which accumulates at the bottom of the smokebox. The remedy is for the wasted portion to be built up by electrically welding and then covering the bottom with Portland cement, finishing off higher around the tube plate and sloping same so as to drain any moisture which might accumulate.

Weights of Materials in New Foster Wheeler Generator

Since the publication on page 26 of the February issue of details of the new Foster Wheeler Corporation, New York, marine steam generator, data on the sizes and weights of materials entering into the construction have been released. The particulars which should be referred to the original article, are as follows:

Weights and Character of Materials for One Boiler

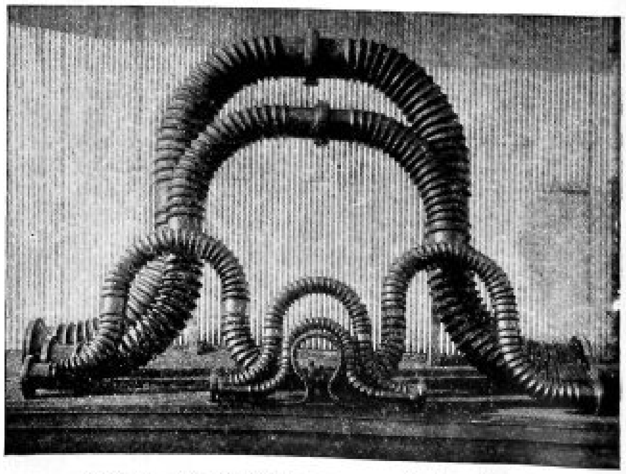
Character	No.	Size inches	Quality	Thickness inches	Quantity
Drums	1 2	48 30	Marine Steel	1 9/16 1 3/32	35,000 lbs.
Headers Superheater Water wall		6 8 10	Open Hearth Seamless	5/8 7/8 7/8	6,000 lbs.
Rivets (Drums)		1¼ 1 15/32	Marine Steel		2,000 lbs.
Tubes					
Superheater	88	1⅛	Chrome vanadium	9 ga.	1,300 ft.
Superheater	292	1¼	Seamless steel	9 ga.	4,670 ft.
Boiler	988	1½	Seamless steel	10 ga.	11,280 ft.
Economizer	180	2	Seamless steel	9 ga.	2,060 ft.
Water wall	18	3¼	Seamless steel	7 ga.	60 ft.
			Total weight		52,000 lbs.
Special Alloy Plates, superheater supports			for 1,500 deg. F.	3/8	1,500 lbs.
Structural Channels Angles Bars			Standard Structural Steel		26,000 lbs.
Casing Plates			Blue Annealed Steel	10 ga. 1/4 3/8	10,000 lbs.
Rivets and bolts		⅜ to 1	Standard material		1,500 lbs.
Castings					
Steel					5,000 lbs.
Iron					29,000 lbs.
Total weight of metal per boiler					168,000 lbs.

Expansion Bends for Steel Pipe and Tubing

The illustration shows an English development as applied to steel pipe and tubing in the familiar form of expansion bends. The corrugating of plain pipe adds several very material advantages, the most important of which is flexibility. This flexibility is available in all directions permitting safe connections in close quarters and where vibration and weaving are factors.

While new to the American market, Aiton-Process corrugated pipe bends are standard equipment in England and on the Continent for power-house, marine and special service. This pipe detail is now available to the American consumer, as the Standard Piping & Fitting Company, Philadelphia, Pa., has been granted sole license to manufacture these bends in the United States.

Preparations are now being made to produce these bends in this country, and American-made Aiton-Process bends should be available about July 1. In the meantime the Standard Piping & Fitting Company is filling the demand by importing English bends with plain ends, supplying and attaching American flanges at the Philadelphia plant.



Typical Aiton-Process corrugated pipe bends

The Boiler Maker

OLUME XXXII NUMBER 3

PUBLISHED Monthly by the Simmons-Boardman Publishing Company, 34 North Crystal Street, East Stroudsburg, Pa., with Executive Offices at 30 hurch Street, New York City.

amuel O. Dunn, Chairman of Board. Henry Lee, President. C. R. Mills, Vice-resident. L. B. Sherman, Vice-President. Roy V. Wright, Vice-President and ecretary. F. H. Thompson, Vice-President. E. T. Howson, Vice-President. F. C. och, Vice-President. John T. DeMott, Treasurer. George Slate, Vice-President d Business Manager.

hicago: 105 W. Adams Street. Washington, D. C.: 17th and H Streets, N. W. leveland: Terminal Tower. San Francisco, Cal.: 215 Market Street.

ubscription Price: United States and Mexico, $2.00. Canada, $3.50, including uty. Foreign Countries, $3.00 a year. Single Copies, 35 Cents.

equest for change of address should reach us on or before the 15th of the onth preceding the issue with which it is to go into effect. It is difficult, nd often impossible, to supply back numbers to replace those undelivered rough failure to send advance notice. In sending us change of address, lease be sure to send us your old address as well as the new one.

HE BOILER MAKER is a member of the Associated Business Papers, Inc. A. B. P.), and the Audit Bureau of Circulations, (A. B. C.)

DITORIAL STAFF: H. H. Brown, Editor. L. S. Blodgett, Managing Editor. W. umbard, Associate Editor.

Contents

Communications

Reducing the Size of Nuts

TO THE EDITOR:

It oftens happens, on marine repair jobs where poorly threaded bolts have been reduced, that the boiler maker has only nuts of large size which slip on the bolts and are useless for the job. Here is a way of getting over that difficulty.

Take the large nuts to the blacksmith and have him heat them. He will then place them on the anvil, edge up not face up; apply his flattener, his helper striking the same, and close in every side so that it may be tapped out to a smaller size of thread.

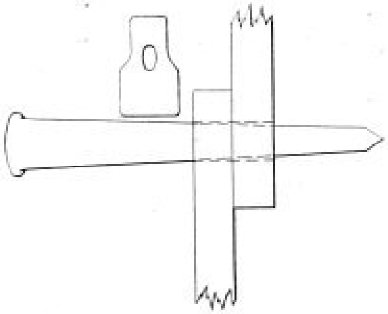
As this thins the material somewhat, it is not as good as a new nut. However, if enough of the original bolts around the defective one are full size and have good threads, the job will do until the ship goes where she can get new nuts.

Montreal, Canada. JAMES WILSON.

Removing Drift Pins

TO THE EDITOR:

To overcome the difficulty often experienced in removing tightly driven drift pins, sharp strokes of the hammer from a direction transverse to the entrance of the pin and as close to that entrance as possible will be found helpful. The close-in blow is not only effective in smoothing out the hole but is not so likely to break or bend the pin.



Method of removing drift pins

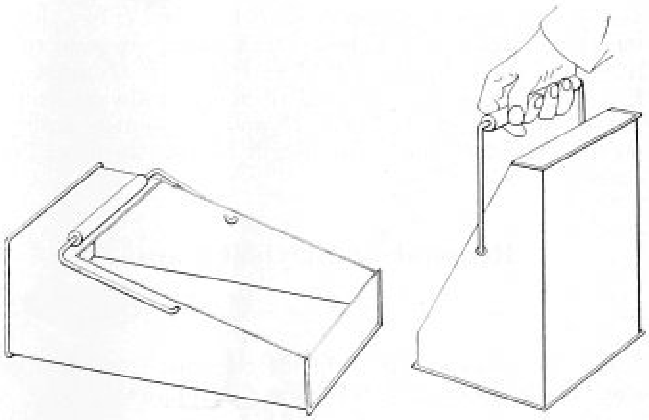
Alhambra, Cal. C. B. DEAN.

Non-Spillable Drip Pan

TO THE EDITOR:

The illustration shows a very useful way of converting an empty five-gallon container into a simple and practical non-spillable drip pan or drain pan.

By shearing the can to the diagonal slant as shown and then fitting it with a bail and handle at the point shown, you will have a drip pan that can be picked up with one hand, as shown, and carried conveniently without spilling.



Drip pan made from five-gallon container

Penacook, N. H. CHARLES H. WILLEY.

Questions and Answers Pertaining to Boilers

This department is maintained for the purpose of helping those who desire assistance on practical boiler shop problems. Inquiries should bear the name and address of the writer. Anonymous communications will not be considered. The identity of the writer, however, will not be disclosed unless the editor is given special permission to do so.

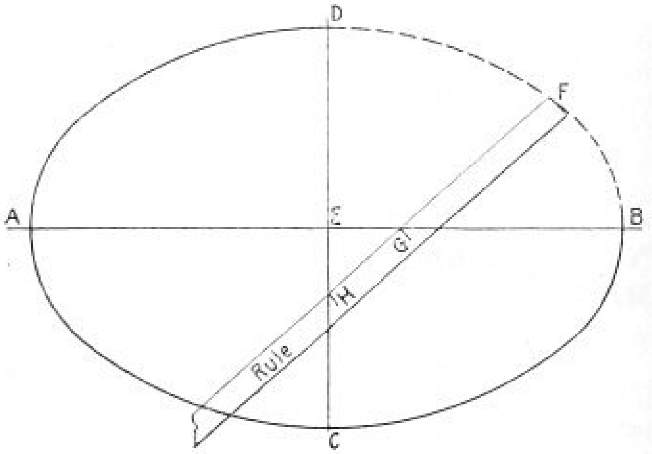
By George M. Davies

Layout of Ellipse

Q.—Will you tell me through the Questions and Answers Department of The Boiler Maker an accurate and simple way to draw an ellipse, such as a 10-inch by 15-inch manhead? H. H.

A.—An ellipse is a conic section. No part of an ellipse is a part of a circle, therefore it cannot be drawn with the ordinary compasses. A quick and accurate method of drawing an ellipse is as follows:

Draw the lines *A-B* and *C-D* at right angles to each other. Set off from *E* to *B* and *E* to *A* half the major axis, or 7½ inches; set off from *E* to *C* and *E* to *D* half the minor axis, or 5 inches. Now mark on your rule the length *F-G* equal to *E-D* and mark *F-H* equal to *E-B*, then by moving the rule around and always keeping the point *H* on the line *C-D*, and the point *G* always on the line *A-B*, the point *F* will be exactly upon the curve of a true ellipse.



Method of laying out a true ellipse

Removal of Staybolt Caps

Q.—What are the reasons for removing the caps from flexible staybolts on a locomotive? I would like to have all the reasons you know of.—J. M. D.

A.—The reasons for removing the caps from flexible staybolts on locomotive boilers are as follows:

(1) To renew a broken staybolt.
(2) To renew a fractured sleeve.
(3) To inspect the staybolt.
(4) To remove and reapply staybolts when applying new firebox sheets.
(5) To replace a defective cap or cap gasket.
(6) To comply with the following I.C.C. Rules.

Rule 23. Method of testing flexible staybolts with caps.

Except as provided in paragraph (b) all staybolts having caps over the outer ends shall have the caps removed at least once every two (2) years and the bolts and sleeves examined for breakage. Each time the hydrostatic test is applied the hammer test required by rules 21 and 22 shall be made while the boiler is under hydrostatic pressure not less than the allowed working pressure.

(b) When all flexible staybolts with which any boiler is equipped are provided with telltale holes not less than 3/16 inch nor more than 7/32 inch in diameter, extending the entire length of the bolt and into the head not less than one-third of its diameter, and these holes are protected from becoming closed by rust and corrosion by copper plating or other approved method, and are opened and tested, each time the hydrostatic test is applied, with an electrical or other instrument approved by the Bureau of Locomotive Inspection, that will positively indicate when the telltale holes are open their entire length, the caps will not be required to be removed. When this test is completed the hydrostatic test must be applied and all staybolts removed which show leakage through the telltale hole.

The inner ends of the telltale holes must be kept closed with fireproof porous material that will exclude foreign matter and permit leakage of steam or water, if the bolt is broken or fractured, into the telltale hole.

When this test is completed the ends of the telltale holes shall be closed with material of different color than that removed and a record kept of colors used.

(c) The removal of flexible staybolt caps and other tests shall be reported on the report of inspection Form No. 3 and a proper record kept in the office of the railroad company of the inspections and tests made.

(d) Firebox sheets must be carefully examined at least once every month for mud, burn, bulging and indications of broken staybolts.

(e) Staybolt caps shall be removed or any of the above tests made whenever the United States inspector considers it desirable in order to thoroughly determine the condition of staybolts or staybolt sleeves.

Requirements of Rule 23 may be interpreted as follows:

(a) When locomotives are stored for one or more full calendar months, the removel of flexible staybolt caps for the purpose of inspection will be due after 24 calendar months service, provided such service is performed within three consecutive years. Portions of calendar months out of service will not be counted. Time out of service must be properly accounted for by out-of-service reports filed with the United States inspector and notation of months claimed out of service must be made on the back of each subsequent inspection report and cab card.

(b) A locomotive stored when flexible staybolt cap removals are due need not be given such test until immediately prior to being returned to service.

(c) When locomotives are being given their annual inspection and test, flexible staybolt caps should be removed at the time of making this inspection, if they become due for removal before another annual inspection is due, and their removal shown on Form 3. One of the principal reasons urged for extending the time for removal of flexible staybolt caps from 18 months to two years was for the purpose of bringing the test due at every second annual inspection.

Thickness of Stayed Surfaces

Q.—In the case of a crown sheet, I have figured the safe working pressure on the stayed surfaces but the thickness of the steel is not used. If a general pitting of ⅛ inch in depth is around the radials, this does not seem to be considered at all. Is there a method of figuring the factor of safety by using a constant? The Government inspectors do not say to what extent a crown sheet may pit before they will take exception to that sheet, although in figuring a ¾-inch boiler sheet which has a general pitting of ⅛ inch, they will assume the sheet thickness to be ⅝ inch.

The crown sheet in question has the following particulars: Patch of radials varies 4 inches; boiler pressure is 200 pounds per square inch; diameter of radials through crown sheet, 1 inch; thickness of sheet not used by mechanical engineers.

I have been instructed that I can run a firebox until the pitting is 3/16 inch around the radials. If you have any information on this problem, I would be glad to receive it. C. H. V.

A.—In determining the allowable working pressure for stayed surfaces there are two things to be considered.

First: The maximum allowable working pressure based on the strength of the staybolt.

Second: The maximum allowable working pressure based on the thickness of the plate.

The maximum allowable working pressure based on the strength of the staybolt is covered by the A.S.M.E. Boiler Construction Code as follows:

Par. P-220 (a)—The full pitch dimensions of the stays shall be employed in determining the area to be supported by a stay and the area occupied by the stay shall be deducted therefrom to obtain the net area. The product of the net area in square inches by the maximum allowable working pressure in pounds per square inch gives the load to be supported by the stay.

(c) The maximum allowable stress per square inch at point of least net cross-sectional area of staybolts shall be as given in Table P-7. In determining the net cross-sectional area of drilled or hollow staybolts the cross-sectional area of the hole shall be deducted.

Table P-7 (a)—Unwelded or flexible staybolts less than 20 diameters long, screwed through plates with ends riveted over —maximum allowable stress, pounds per square inch = 7500.

The formula would be:

$$P = \frac{7500 \times 2}{A - a}$$

where P = maximum allowable working pressure in pounds per square inch.
a = least net cross-sectional area of staybolts, square inches.
A = area to be supported by staybolt using full pitch dimensions, square inches.

In this formula the thickness of the plate is not considered.

The maximum allowable working pressure based on the thickness of the plate is covered by the A.S.M.E. Boiler Construction Code as follows:

Par. P-199—The maximum allowable working pressure for various thickness of braced and stayed flat plates and those which by these rules require staying as flat surfaces with braces or staybolts of uniform diameter symmetrically spaced, shall be calculated by the formula,

$$P = C \times \frac{T^2}{p^2}$$

where, P = maximum allowable working pressure in pounds per square inch.
T = thickness of plate in sixteenths of an inch.
p = maximum pitch measured between straight lines passing through the centers of the staybolts in the different rows, which lines may be horizontal, vertical or inclined, inches.
$C = 112$ for stays screwed through plates not over 7/16 inch thick with ends riveted over.
$C = 120$ for stays screwed through plates over 7/16 inch thick with ends riveted over.
$C = 135$ for stays screwed through plates and fitted with single nuts outside of plate or with inside and outside nuts omitting washers (see Par. P-203).
$C = 150$ for stays with heads not less than 1.3 times the diameter of the stays, screwed through plates or made a taper fit and having the heads formed on the stays before installing them and not riveted over, said heads being made to have a true beaming on the plate.
$C = 175$ for stays fitted with inside and outside nuts and outside washers where the diameter of washers is not less than $0.4p$ and thickness not less than T.

When two sheets are connected by stays and but one of these sheets requires staying, the value of C is governed by the thickness of the sheet requiring staying.

Transposing this formula for T, we have:

$$P = C \times \frac{T^2}{p^2}$$

$$P \times p^2 = C \times T^2$$

$$\frac{P \times p^2}{C} = T^2$$

$$\sqrt{\frac{P \times p^2}{C}} = T$$

Substituting the values given in the question, we have:

$P = 200$ pounds
$p = 4$ inches
$C = 112$ taken from Par. P-199.

$$T = \sqrt{\frac{200 \times 4^2}{112}}$$

$$T = \sqrt{\frac{200 \times 16}{112}}$$

$$T = \sqrt{28.5}$$

$T = 5.3$ sixteenths of an inch or .33125 inch minimum thickness of plate permissible.

Adjustment of Steam Gages

Q.—Will you please tell me through the Questions and Answers department of THE BOILER MAKER the correct way to adjust steam gages? I notice that most boiler inspectors simply set the gage hand back the amount it was off. This doesn't appear to me to be the correct way. I hope this question will not be out of place in THE BOILER MAKER.—H.H.

A.—The method of adjusting a steam gage depends upon the trouble experienced and upon the accuracy desired. If upon testing the gage it was found to be out a definite number of pounds and this discrepancy was about constant throughout the entire scale the resetting of the hand is all that is necessary. This is accomplished by removing the gage hand with a gage hand puller as illustrated in Fig. 1 and afterwards refitting it to the spindle by very light hammer blows.

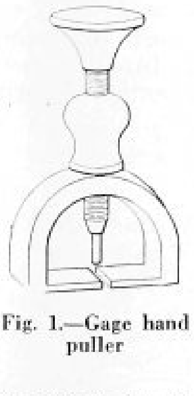


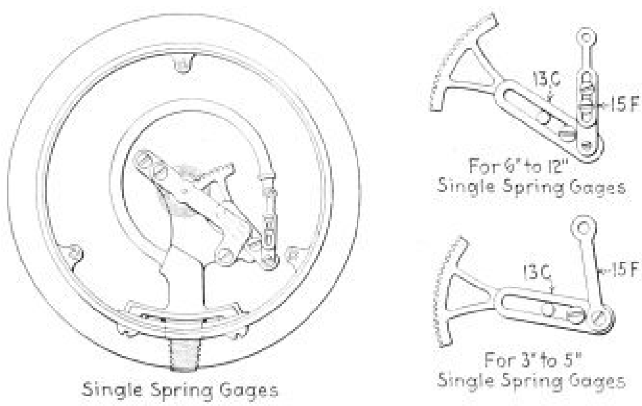
Fig. 1.—Gage hand puller

However, should there be found to be a progressive gain or loss, the adjustment in the mechanism is to be resorted to. This part, of course, is naturally more troublesome except to an expert, although it may be done by any one using reasonable care.

There are various makes of steam gages but all are, more or less, based on the same linkage principle.

Fig. 2 illustrates a typical single spring gage and is adjusted as follows:

Attach a test gage and the gage to be adjusted to the test pump; or if a gage tester is used, attach the gage to be adjusted in the usual way. In either case run the pressure up to the first or second figured graduation, either by the pump or by applying weights upon the gage tester so that there shall be a pressure on the gage, and set the hand of the gage on the point on the dial for the pressure applied. Then run the pressure up to the limit of the pressure shown on the dial of the gage, to be adjusted to see if the



For 6" to 12" Single Spring Gages

For 3" to 5" Single Spring Gages

Single Spring Gages

Fig. 2.—Single spring gage

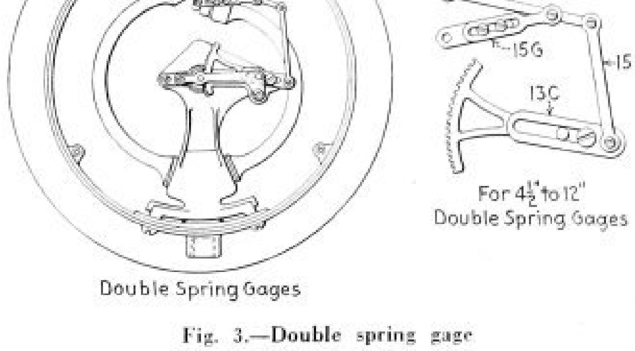
mechanism moves freely. If a gage tester is used, this can be easily done by pressing upon the weight tray. Afterwards, apply gradually increasing pressures and if the hand of the gage differs in its movement from that of the test gage or does not correspond with the weights of the gage tester, the gage must be adjusted. If the gage hand moves increasingly too fast, *lengthen*, or if it moves increasingly too slow, *shorten* the sector slide *13C*, which connects the sector of the movement with the slide link, *15F*, as shown in Fig 2.

If the hand of the gage to be adjusted should read fast, or show a gain over the first half of the pressure range on the dial, and slow, or show a loss on the last half of the pressure range shown on the dial, for example, shorten the slide link *15F* slightly, that is, from 1/32 inch to 1/16 inch. This will make the hand move slower at the middle pressures and faster at the lower and higher pressures; and if it then moves too fast, slightly lengthen the sector slide *13C*.

If the gage should read slow or show a loss over the first half of the pressure range on the dial, lengthen the slide link *15F*. This will make the gage hand move faster at the medium pressures and slower at the higher and lower pressures; and then if too slow shorten the sector slide *13C*. It is necessary to adjust both the sector slide *13C* and the slide link *15F* to adjust the gage correctly. Be careful not to bend the links by too much pressure; but tighten the slide screws firmly after the gage is adjusted.

Fig. 3 illustrates a typical double spring gage and is adjusted as follows:

Attach a test gage and the gage to be adjusted to the test pump; or if a gage tester is used, attach the gage to be adjusted in the usual way. In either case run the pressure up to the first or second figured graduation, either by the pump or by applying weights upon the gage tester, so that there shall be a pressure on the gage, and set the hand of the gage on the point on the dial for the pressure applied. Then run the pressure up to the limit of the pressure shown on the dial of the gage to be adjusted to see if the mechanism moves freely; if a gage tester is used, this can be easily done by pressing upon the weight tray. Afterward, apply gradually increasing pressures and if the hand of the gage differs in its movement from that of the test gage, or does not correspond with the weights of the gage tester, the gage must be adjusted. If the gage hand moves increasingly too fast, *lengthen*, or if it moves increasingly too slow, *shorten* the sector slide *13C*, which connects the sector of the movement with the connecting link *15F*, as shown in Fig. 3.



For 4½" to 12" Double Spring Gages

Double Spring Gages

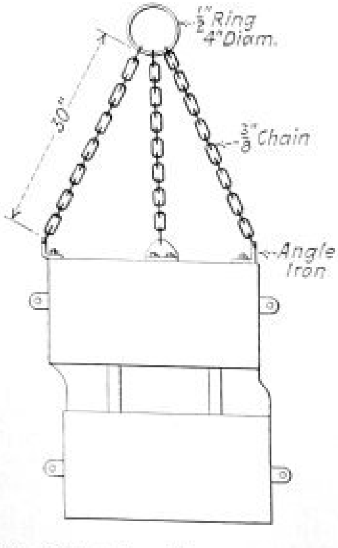
Fig. 3.—Double spring gage

If the hand of the gage to be adjusted should read fast, or show a gain over the first half of the pressure range on the dial, and slow or show a loss on the last half of the pressure range shown on the dial, for example, lengthen the slide link *15G* slightly, that is, from 1/32 inch to 1/16 inch. This will make the hand move slower at the middle pressures and faster at the lower and higher pressures; and if it then moves too fast, slightly lengthen the sector slide *13C*.

If the gage should read slow or show a loss over the first half of the pressure range on the dial, shorten the slide link *15G*. This will make the gage hand move faster at the medium pressures and slower at the higher and lower pressures; and then if too slow, shorten the sector slide *13C*. It is necessary to adjust both the sector slide *13C* and the slide link *15G* to adjust the gage correctly.

A Safe Way to Handle Compressors

Shown in the sketch is a simple and safe way to handle air compressors around a shop by overhead crane or crane truck. The feature of this chain and ring arrangement is the three pieces of angle which are secured to the end link of each chain. The horizontal leg of each angle is bolted to the top end of the compressor, as shown. The T-head bolts and nuts used to secure the top head to the compressors are used to bolt the angles.



This chain and angle arrangement ensures safe handling of air compressors

Associations

Bureau of Locomotive Inspection of the Interstate Commerce Commission

Chief Inspector—A. G. Pack, Washington, D. C.

Assistant Chief Inspectors—J. M. Hall, Washington, D. C.; J. A. Shirley, Washington, D. C.

Steamboat Inspection Service of the Department of Commerce

Supervising Inspector General—D. N. Hoover, Jr., Washington, D. C.

American Uniform Boiler Law Society

Chairman of the Administrative Council—Charles E. Gorton, 253 Broadway, New York.

Boiler Code Committee of the American Society of Mechanical Engineers

Chairman—Fred R. Low.

Vice-Chairman—D. S. Jacobus, New York.

Secretary—C. W. Obert, 29 W. 39th Street, New York.

National Board of Boiler and Pressure Vessel Inspectors

Chairman—C. D. Thomas, Salem, Oregon.

Secretary-Treasurer—C. O. Myers, Commercial National Bank Building, Columbus, Ohio.

Vice-Chairman—William H. Furman, Albany, N. Y.

Statistician—L. C. Peal, Nashville, Tenn.

International Brotherhood of Boiler Makers, Iron Ship Builders and Helpers of America

International President—J. A. Franklin, suite 522, Brotherhood Block, Kansas City, Kansas.

Assistant International President—J. N. Davis, suite 522, Brotherhood Block, Kansas City, Kansas.

International Secretary-Treasurer—Chas. F. Scott, suite 506, Brotherhood Block, Kansas City, Kansas.

Editor-Manager of Journal—John J. Barry, suite 524, Brotherhood Block, Kansas City, Kansas.

International Vice-Presidents—Joseph Reed, 1123 E. Madison Street, Portland, Ore.; W. A. Calvin, 1622 Glendale Street, Jacksonville, Fla.; Harry Nicholas, 6215 S. Benton Blvd., Kansas City, Mo.; W. E. Walter, 637 N. 25th Street, E. St. Louis, Ill.; J. H. Gutridge, 910 N. 18th Street, Milwaukee, Wis.; W. G. Pendergast, 26 South Street, New York, N. Y.; W. J. Coyle, 424 Third Avenue, Verdun, Montreal, Quebec, Can.; A. M. Milligan, 262 Trent Avenue, E. Kildonan, Man., Can.; J. F. Schmitt, 28 S. Roys Street, Columbus, O.; William Williams, 502 Labor Temple, Portland, Ore.

Master Boiler Makers' Association

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First Vice-President—Franklin T. Litz, general boiler foreman, Chicago, Milwaukee, St. Paul & Pacific Railroad, Milwaukee, Wis.

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Treasurer—George R. Boyce, A. M. Castle & Company, Chicago, Ill.

Secretary—Frank C. Hasse, Oxweld Railroad Service Company, 230 N. Michigan Ave., Chicago, Ill.

American Boiler Manufacturers' Association

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Vice-President—S. G. Bradford, Edge Moor Iron Company, Edge Moor, Del.

Secretary-Treasurer—A. C. Baker, 801 Rockefeller Building, Cleveland, O.

Executive Committee (three years)—Homer Adams, Fitzgibbons Boiler Company, Inc., New York; George W. Bach, Union Iron Works, Erie, Pa.; G. S. Barnum, The Bigelow Company, New Haven, Conn. (Two years)—Owsley Brown, Springfield Boiler Company, Springfield, Ill.; F. W. Chipman, International Engineering Works, Framingham, Mass.; W. C. Connelly, Foster Wheeler Company, Cleveland, O. (One year)—J. G. Eury, Henry Vogt Machine Company, Louisville, Ky.; A. C. Weigel, Combustion Engineering Corporation, New York; E. G. Wein, E. Keeler Company, Williamsport, Pa.; *Ex-officio*, H. E. Aldrich, Wickes Boiler Company, Saginaw, Mich.

States and Cities That Have Adopted the A.S.M.E. Boiler Code

States

Arkansas
California
Delaware
Indiana
Maryland
Michigan
Minnesota
Missouri
New Jersey
New York
Ohio
Oklahoma
Oregon
Pennsylvania
Rhode Island
Utah
Washington
Wisconsin
District of Columbia
Panama Canal Zone
Territory of Hawaii

Cities

Chicago, Ill.
Detroit, Mich.
Erie, Pa.
Kansas City, Mo.
Los Angeles, Cal.
St. Joseph, Mo.
St. Louis, Mo.
Scranton, Pa.
Seattle, Wash.
Tampa, Fla.
Evanston, Ill.
Memphis, Tenn.
Nashville, Tenn.
Omaha, Neb.
Parkersburg, W. Va.
Philadelphia, Pa.

States and Cities Accepting Stamp of the National Board of Boiler and Pressure Vessel Inspectors

States

Arkansas
California
Delaware
Indiana
Maryland
Minnesota
Missouri
New Jersey
New York
Ohio
Oklahoma
Oregon
Pennsylvania
Rhode Island
Utah
Washington
Wisconsin
Michigan

Cities

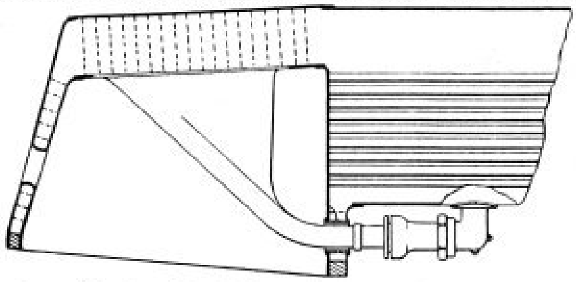
Chicago, Ill.
Kansas City, Mo.
Memphis, Tenn.
Erie, Pa.
St. Louis, Mo.
Scranton, Pa.
Seattle, Wash
Tampa, Fla.
Detroit, Mich.
Nashville, Tenn.
Omaha, Neb.
Parkersburg, W. Va.
Philadelphia, Pa.

Selected Boiler Patents

Compiled by Dwight B. Galt, Patent Attorney, Washington Loan and Trust Building, Washington, D. C. Readers wishing copies of patents or any further information regarding any patent described, should correspond directly with Mr. Galt.

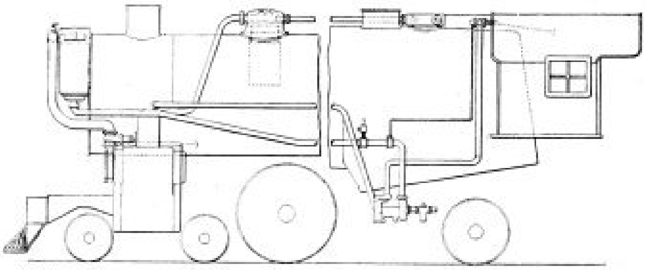
1,801,434. LOCOMOTIVE BOILER. BERT E. LARSON, OF CHICAGO, ILL., ASSIGNOR TO LOCOMOTIVE FIREBOX COMPANY, OF CHICAGO, ILL., A CORPORATION OF DELAWARE.

Claim.—A locomotive boiler including a firebox, a flat hollow thermic syphon fixed at one end in the crown sheet of the firebox and having a neck extending through and forwardly beyond the front water leg of the boiler and means connecting said neck with the boiler at a point in advance of said water leg and including a ball joint fitting. Four claims.



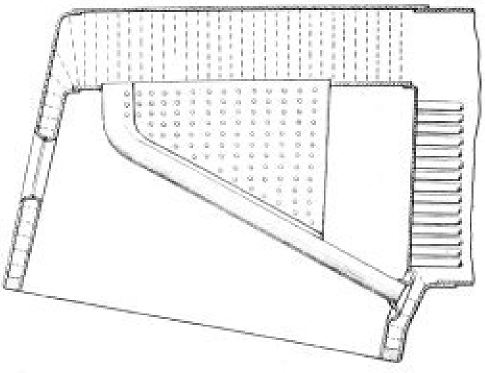
1,802,125. FEED WATER-HEATING DEVICE, EUGENE L. SCHELLENS, OF MILLBURN, N. J., ASSIGNOR TO C-S ENGINEERING COMPANY, OF ENGLEWOOD, N. J., A CORPORATION OF DELAWARE.

Claim.—In a heater of the character described, the combination of a casing adapted to be secured to a locomotive boiler, said casing having an open bottom adapted to be in communication with the boiler, a detachable cover member having a water inlet and a steam outlet, a spray valve carried by said cover member for delivering the water from said inlet into said casing, and a series of superimposed baffling members suspended from said cover in spaced relation, said baffling members being removable as a unit upwardly through said casing when said cover is removed. Three claims.



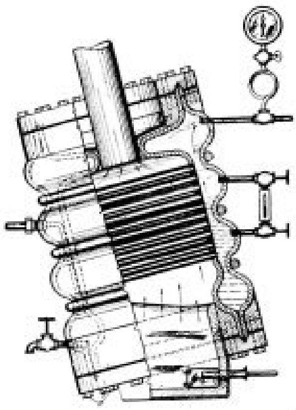
1,800,506. THERMIC SYPHON AND METHOD OF MAKING THE SAME. GUST J. CHRISTENSON, OF CHICAGO, ILL., ASSIGNOR TO LOCOMOTIVE FIREBOX COMPANY, OF CHICAGO, ILL., A CORPORATION OF DELAWARE.

Claim.—The method of making thermic syphons which consists in providing an open top, flat hollow syphon body having side walls, in slitting a seamless metallic tube throughout a portion of its length beginning at one end thereof, opening up portions of the tube upon opposite sides of the slit to provide parallel flanges thereon, forming the tube so that the edges of its flanges fit the desired edges of the side walls of the body and terminates at one end in substantially the plane of the open top of said body and then welding the fitted edges of the body side walls and tube flanges together, the unslitted end of said tube projecting beyond the syphon body to form an inlet neck therefor. Eleven claims.

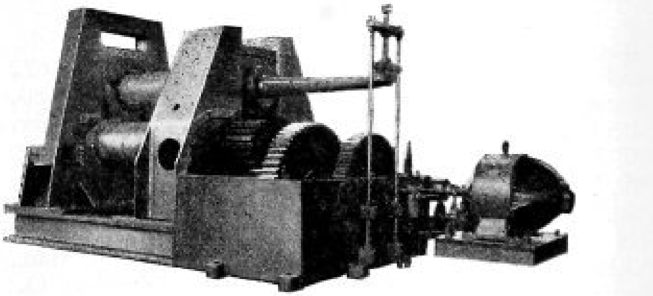


1,794,801. HIGH-PRESSURE BOILER. EDWARD W. SOUTER, OF SAN LUIS OBISPO, CAL., ASSIGNOR OF ONE-HALF TO GEORGE F. MOERMAN, OF SAN LUIS OBISPO, CAL.

Claim.—In a boiler, in combination, inner and outer round shells one within the other and of corrugated form to form water space therebetween, the opposite ends of said shells fitting closely together one within the other and having rings welded to their outer sides with said ends between said rings and projecting beyond the same, said projecting portions being beveled on their exterior sides, end rings or plates having grooves in their inner faces receiving the projecting ends of said shells and shaped for wedging them tightly together, screw bolts for securing said end rings to the rings welded to the ends of said shells, watertubes in said inner shell, a firebox under said inner shell and said watertubes, and a flue from the upper end of said inner shell.

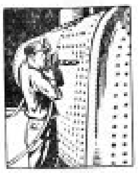


Classified Advertisement



The Boiler Maker

Reg. U. S. Pat. Off.



Master Boiler Makers' Convention

At a general meeting of railway mechanical department associations' representatives, held recently in Chicago, it was unanimously decided after a thorough discussion of the problems involved that no meetings be held during 1932. Since the Master Boiler Makers' Association comes within the category of railway mechanical organizations, this decision definitely settles the possibility of a meeting being held this year. Two representatives of the association were present at the meeting.

It was further decided that when meetings of the mechanical associations are resumed nine of them will convene at the same place and during the same week. This group includes the fuel association, traveling engineers, car department officers, air brake department, general foremen, tool foremen, blacksmiths, equipment painters and master boiler makers.

The question of whether the period to be included would be a calendar week or would include Thursday, Friday and Saturday of one calendar week and Monday, Tuesday and Wednesday of the following calendar week was discussed. It seemed to be the general consensus of opinion that the last three days of one week and the first three days of the following week would be the profitable arrangement, particularly with respect to one group checking out of the hotels and the following group coming in.

The representatives of the various associations present also agreed to take under advisement the possibility of further co-ordination and consolidation to avoid any overlapping in the work of their associations. The decision was made that the question of dates, place, grouping and the like, of such meetings, when conditions are such that conventions again can be held, would be left in the hands of the general committee of the Mechanical Division, American Railway Association to be handled through a sub-committee of that division.

While the temporary halt in the progress of the Master Boiler Makers' Association is to be deplored, it is very possible that under the proposed scheme of conventions the change will bring opportunities for greater expansion than ever before. Certainly exhibits of supply companies will be more complete, and the closer contact with associations made up of men from other departments will broaden the viewpoint of representatives of the boiler division.

Until the first conventions under the new scheme are carried out, it will be impossible to predict the benefits to be gained from this plan of co-ordination. In the meantime members of the Master Boiler Makers' Association should strive to retain their interest in its welfare and belief in its future.

With other conditions so serious, but little enthusiasm can be expected over such considerations as conventions. Nevertheless, the value of organization work has definitely been proven in the past, and the need of it does not cease to exist simply because of a temporary setback. It should be the duty of all the officers, as it is in some individual cases, to safeguard the welfare of the association through this trying period, and to lay such plans as will make possible the renewal of its work with a minimum loss in membership and interest.

Justice for the Shop Staff

Mention has been made in these pages of an unfair and demoralizing policy towards department heads and foremen which in recent months has been adopted by some of the railroads of this country. In these instances, curtailments in employment and privileges have been indiscriminately applied to individuals regardless of their rank or service. While temporary savings may be made, the evils of such a course will be felt long after the need for retrenchment has ceased.

So far as the highly trained shop staff is concerned, the problem is no different. Even under present adverse conditions, there is an equitable basis on which a railroad can establish its employment policy. With justice weighing its decisions, it is possible to determine the course which will work the least hardship on those individuals, who in other times have proven their efficiency as artisans and their loyalty to their employers.

In great measure, the rebuilding of the railroads' estate will depend upon such men. Their experience and ability, developed over a long period of years, are among the most important of the assets of any railroad, and at this time should be considered in this light.

Having served as the framework on which the greatest nation on earth could be built, should the railroads now cease as the keystone of the transportation arch, a period of disintegration might be expected. However, the facts point to quite a different fate. Constructive plans for the future, covering methods and efficiency of operation and of financing, are being instituted. The same energy and thought on the part of the railroad executives should be devoted to the welfare of the men upon whom the success or failure of this future will largely depend.

Checking-Up on the Welder

By H. L. Miller*

The most important work that the welder is called upon to do in the construction and maintenance of the locomotive is the welding of the seams in the firebox. While the welds are supported to some extent by the staybolts and, under normal conditions of operation, a poor job of welding may still hold water, when low water conditions occur, a poor weld will often rip for a long distance. The resultant explosion becomes many times worse than it would be if the strength of the weld were up to the strength of the sheet.

Upon the welding foreman's shoulders rests the responsibility for the safety of many lives as well as thousands of dollars of property. Much has been written regarding inspections and tests of welds after the weld is completed. Many complicated pieces of apparatus, such an X-rays and other electrical devices have been tested out, but due to the inaccessibility of the welds in a firebox as to getting apparatus for testing behind them in the water space, and due also to the cost and highly technical nature and as yet doubtful efficiency of these devices, it is doubtful whether the welding foreman will be able to consider their application in the boiler shop for a long time to come.

The welding art, is an art. The metallurgist and the chemist must do their part along with the electrician, but the welder, the man who melts the steel and fuses it into the plates and joins them together with a bond of metal, he is the man on whom the responsibility for the strength and integrity of his work lies. He should feel a pride in his work, as he is about the only man around the shop whose work is still an art. Even the painter now works with stencils and machines for spreading paint, but the welder is the only free-hand artist left and his is indeed a great responsibility.

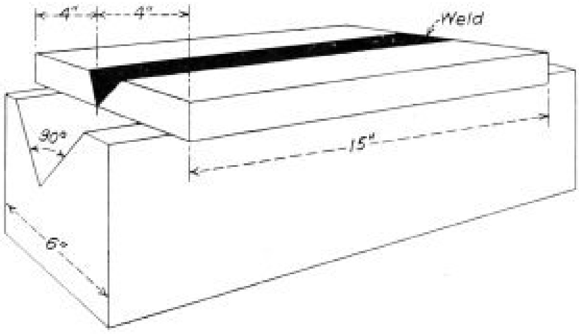


Fig. 1.—Test weld on V block for bend test

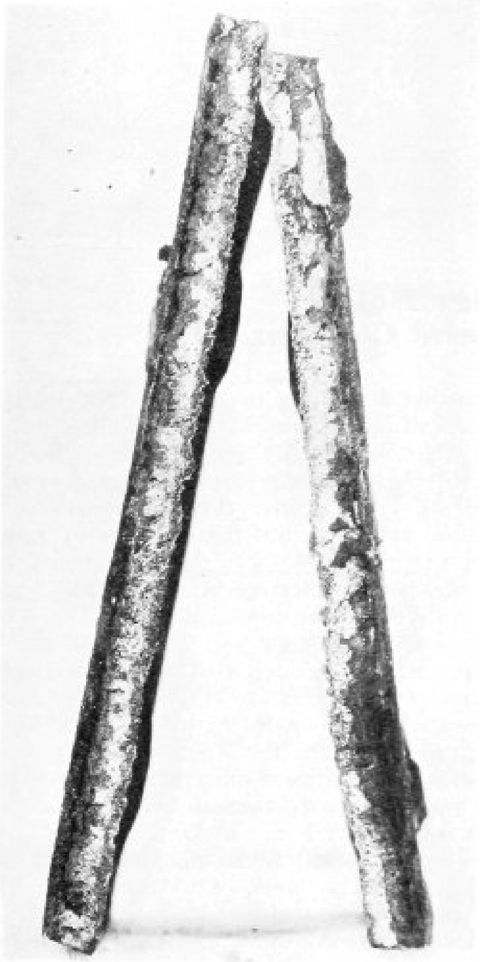


Fig. 2.—Side seam weld broken in test

But is it fair to hold a workman responsible for his product when he has so few chances to check his work in making tests? The writer has worked out a simple method for welding foremen to check up on their welders, that involves no machinery or special apparatus beyond what should be found in the blacksmith shop.

Test plates are cut about 4 inches by 15 inches of $\frac{3}{8}$-inch firebox plate. They are scarfed on one side for a single V weld on the 15-inch edge of the plate. The welds are made in the various positions as if they were in the firebox, that is down hand, vertical, horizontal seam and overhead. Welding is done from one side only as in repair work when the water side is inaccessible.

The conventional method of having a welder make a test weld once every three or six months and then send-

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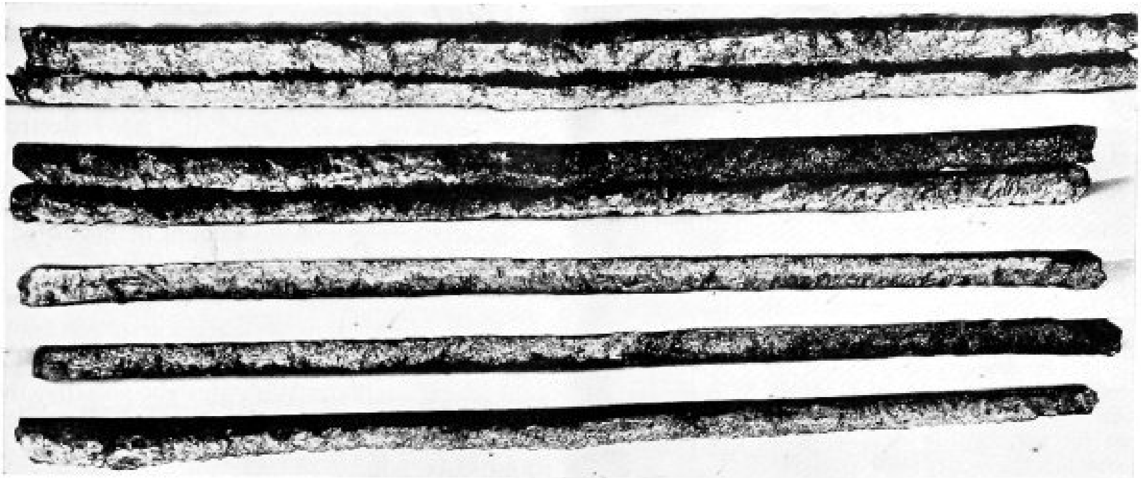


Fig. 3.—Examples of poor welds broken in V block

ing the tests away to be pulled in the tensile machine and seldom, if ever, sending the broken pieces back to the welder, is not, to our notion, any test at all. Most of these test welds are only 3 inches of running seam and are generally welded "down hand." Very little down-hand work is done in firebox welding, except on a new firebox when assembling on the floor. All work done in the firebox when in the boiler is in the vertical, horizontal side seam, or overhead, except the water-side seam on the crown sheet to flue-sheet joint, or the water side on the syphon to crown-sheet welds.

The welding technique for vertical, horizontal or overhead work is entirely different from down-hand work. Yet how many test welds are made in these directions of welding?

Suppose you are a welding foreman and a government inspector asked you how you could prove that your welders were doing good work on the job. How would you go about proving it? Let's go into this situation in this way. Here is an engine coming in for class repairs and requiring, let us say, new side sheets or a half door sheet, and maybe a new back flue sheet. In the case of the side sheet, the horizontal seam is the most difficult to weld and one that fools more welders and inspectors than all the other seams.

The horizontal seam in many explosions has been the reason for more bad rips than the vertical seams and is second only to the crown-sheet to flue-sheet seam in this regard.

Suppose we cut out the old side sheets, going up one row of staybolts above the weld. When the sheet is out, cut off a strip one row of bolts below the weld so that the weld is in the middle of a strip two rows of bolts wide and running the length of the weld. Burn the bolts off flush with the water side. Cut the strip on the shear into sections about 12 inches to 15 inches long and then lay them longways on a V block and break them under the steam hammer, or else hold under the clamp and break the weld down with a sledge. The water side of the plate should be on the outside of the bend.

The welds are placed on the V block as shown in Fig. 1, and are pounded down in the V block under the steam hammer, using a round bar or round-nosed fuller. Breaking against the weld shows up the fusion or lack of fusion on the water side. Most welders can make the top bead look pretty good, but unless the weld is clear through the thickness of the plate, the percentage of the weld is low.

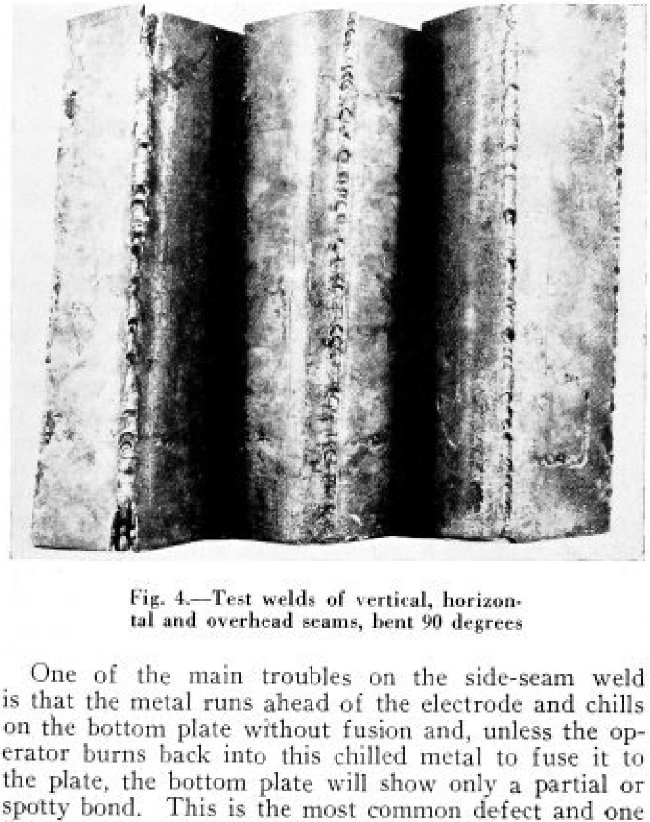


Fig. 4.—Test welds of vertical, horizontal and overhead seams, bent 90 degrees

One of the main troubles on the side-seam weld is that the metal runs ahead of the electrode and chills on the bottom plate without fusion and, unless the operator burns back into this chilled metal to fuse it to the plate, the bottom plate will show only a partial or spotty bond. This is the most common defect and one liable to cause the worst rips in low-water jobs.

One advantage of this test over the pull test is that a steady slow pull will not show up an overheated or crystallized weld. We have often seen welds that were brittle and would not bend but would break off sharp, yet when pulled in the tensile machine, would hold and pull the plate apart.

The strain on a weld in a low-water job is a hard and sudden bulging or bending strain and brittle or overheated metal will undoubtedly cause trouble under these conditions. But you say, an explosion will tear plate where there are no welds. True enough, but where the weld is efficient, it will tear the plate instead of the weld, which is what we are working for—efficiency in the joining of the plate equal to the strength of the plate itself.

The writer has seen this test performed on several old plates, from two to five years in service, and the average efficiency of the side-seam weld with regard to fusion on the plates is around 60 percent.

The fractured surfaces of the section of horizontal welded seam as shown in Fig. 2, were found in testing the welds on a pair of side sheets removed due to fire cracking at the staybolt holes after two years.

The weld had shown no trouble in service and the appearance from the fire side of the sheet was good. Yet when broken on the V block, the total metal holding the sheets together was not more than 50 percent of the thickness of the plate.

The bright spots and areas are the fractured surfaces and are mainly on the side next to the fire. In the illustration, the water surface of the plates is to the outside and the fire surfaces facing each other. You will note that along the back side of the weld there has been no penetration or fusion on the plate. The sharp edge of the scarf still remains.

When you consider that these edges shown should be solid metal, of the thickness of the weld instead of the small amount of metal actually holding, as shown by the bright areas, you can realize what might happen if the water had ever gone low in this engine.

Now unless you, as a welding foreman, are checking up on your welders' work by examining welds in such a manner after removal from the boilers, you cannot be sure at all of what condition your firebox seams are in. This particular job was not any worse than many others I have seen in tests of this sort in various shops. It went two years in service without trouble because it was held up by the staybolts.

Fig. 3 shows examples of bad test welds, broken in the V block. The two bottom pairs are badly overheated. They were done in the vertical direction on a piece-work rate. The middle piece shows good heat on the first bead but too much on the second. The next to top piece is an example of poor penetration on the first bead. This was a horizontal seam. The top piece is somewhat better but still brittle, although the plate bent quite a little before fracture occurred. The angle of the bend was about 20 degrees in the part to the right, before rupture.

Fig. 4 shows three test welds which passed inspection. From left to right, they were made in the vertical, horizontal and overhead positions. When this shop was on piece work, the man who did this work was too conscientious a workman to draw a good pay check.

The really fair way to rate welders on the pay roll is to take such weld tests, as we have shown, once a month and to grade each man according to his showing on his test welds. This method would work up plenty of spirit of craftsmanship among the welders, and the good men would welcome this sort of test as a means of showing their real ability.

The test welds should be made under operating conditions and should be tested as soon as cool. The welder should see his work tested, while the method he used on each weld is fresh in his mind, and he should be allowed to repeat his test immediately until any defects in his work can be ironed out to his own satisfaction. By quick checking on his practice in this manner, a welder can bring his workmanship up to the point where it should be a matter of pride both to him and to his foreman.

B. F. McIntyre, formerly with the Industrial Controller Company, Milwaukee, Wis., is now connected with the Chicago office of The Lincoln Electric Company, Cleveland, O.

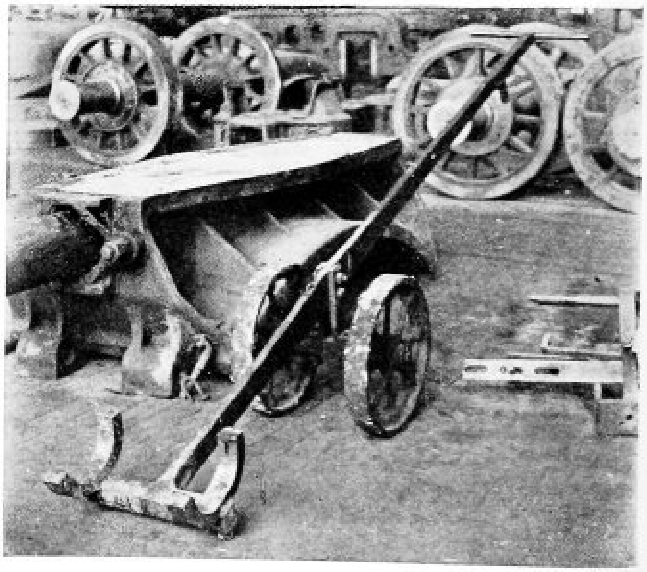
Hi-Tensile Electrodes

The Page Steel & Wire Company, Monessen, Pa., has developed a new type of coated electrode known as the Page Hi-Tensile electrode which gives extremely quiet operation with little or no sputter. Inasmuch as the arc is free-burning, the molten pool of metal readily can be observed and the amount of deposited metal can be accurately judged. The grain structure of the weld is essentially the same as that of mild steel and boiler plate base metal.

Unlike most coated electrodes, it is said that Page Hi-Tensile electrodes do not require reverse polarity for efficient operation. With straight polarity, the speed is equal to that of any other type of electrode using reverse polarity.

Binder Cart for the Back Shop

The binder cart shown in the illustration has proven advantageous for transporting and putting up binders on locomotives. It consists of a 10-foot section of 1-inch by 6-inch steel forged to form a lever and binder carrier. This is mounted on the axle as shown which in turn is carried on two 18-inch wheels of cast iron. The fulcrum point over the axle is located 4 feet from the binder carrier. The two vertical bars which fulcrum the arm to the axle are of 1-inch by 3-inch steel 8-inches high, bolted as shown. The carrier arms for the binder are made of 1½-inch by 3-inch bars, shaped as shown. These arms are welded to the ends of the cross piece which is forged from 3-inch by 3-inch steel stock, 18 inches long. This cross piece is welded to the end of the lever bar of the cart.



A handy two-wheel cart for putting up binders

A handle bar is welded to the opposite end of the lever to facilitate balancing the binder especially when lifting it into position on the locomotive.

William L. Austin, former president and chairman of the board of the Baldwin Locomotive Works, died at Rosemont, Pa., on March 10.

Failures in High-Pressure Boilers

The adoption of watertube boilers on board ship, in which steam is generated at pressures approaching 500 pounds per square inch, has introduced a number of circumstances tending to cause failure which were entirely absent in boilers of the usual return-tube marine type, in which pressures rarely exceeding 250 pounds per square inch are employed. Some idea of the diversity of these circumstances can be gathered from the perusal of a number of reports dealing with the causes and prevention of boiler failure which were presented at the annual meeting of the German Association of Boiler Owners. There has been in that country a tendency to concentrate attention on the condition of the feed water as leading to such failures as have occurred, a tendency which is shared by England, and it is of importance, therefore, to record that Professor Bauer, one of the greatest experts on the subject of steam engineering, suggests that this tendency is dangerous since it means that the whole question is being diverted along a side track while the real causes are overlooked. He presented a report dealing with the experience gained at the Government Material Testing Bureau at Dahlem during the course of an investigation into the failures on boilers at pressures up to 35 atmospheres (515 pounds per square inch), in which he stated that corrosion, with which the condition of the feed water is closely associated, is only of secondary importance, the condition of the material being the more important consideration.

The causes of failure in high-pressure boilers can, Professor Bauer suggests, be sought under four headings, viz., the material itself; the effect of the method of manufacture on the properties of the material; the defects occasioned by the construction and operation of the boilers; and the type of welding, this last being introduced on account of the fact that welding is being increasingly employed in Germany both for the construction and the repair of boilers. With regard to the effect of the material itself on the incidence of failure, he places the chemical composition first in its influence upon the properties of the material of plates and tubes, and for the best results he asserts that it must be as free as is feasibly possible from phosphorus, sulphur, arsenic, and other non-metallic impurities. Practically all the new boiler plates tested at the bureau in recent years have been satisfactory as regards chemical composition, although high phosphorus and high sulphur are often found in material taken from old boilers. It is of importance to note that while the usual amounts of arsenic, about 0.04 percent, have hardly any influence on the properties of the steel, an increase in the arsenic content leads to an increase in the detrimental effects of the phosphorus and sulphur which may be present. On the question as to whether soft or hard steels are preferable in boiler construction, the experience of the bureau is that no objections can be raised against the use of hard steels.

The question of the effect of the method of manufacture is obviously of considerable importance, since even the best material can be spoiled by improper treatment. The detrimental effect of cold bending and working is clearly revealed both by the results of tests and by micro-photographs, while other causes of failure of somewhat similar nature are the use of excessive riveting pressure with a crushing of the material around the hole, the drifting of rivet holes and excessive work in expanding the ends of the tubes. The failures arising in operation, apart from those due to corrosion, are caused by defective circulation causing local overheating and by excessive stiffness in the construction. In some cases, reinforcing plates riveted to drums have proved themselves to be sources of weakness rather than of strength, and this emphasizes the necessity for providing elastic deformation under the loads due to expansion on heating up. Finally, it may be mentioned that welding is not viewed with favor by the bureau owing to the extent to which the results are dependent upon the skill of the operator and the difficulty of testing the efficiency of the work. In particular, it is stated that the cutting out and welding of cracks is usually both worthless and dangerous, since the repair cannot remove the faults which cause the cracks. It is to be noted that while Professor Bauer relegated the problem of feed water to a position of secondary importance, some of his colleagues at the meeting had evidently experienced certain troubles as a result of corrosion, other papers being presented which dealt with this phase of the problem, and while impure feed may have been responsible for some of these troubles, it was shown the failure may be caused by corrosion due to steam trapped by the flow of water. This is a question which demands closer investigation.—*Shipbuilding and Shipping Record.*

Mikado Type Locomotives for Chinese Railroad

The Kailan Mining Administration, China, for working the Peiping Liao-Ning Ry., placed an order with the North British Locomotive Co., Ltd., for six Mikado 2-8-2 type engines with bogie tenders to suit the standard gage.

The leading particulars are:—Cylinders 21 in. diameter by 28 in. stroke; coupled wheels 4 ft. 6 in. diameter, leading truck wheels 3 ft. 1½ in. diameter; rear truck wheels, with outside bearings, 3 ft. 7 in. diameter; wheelbase, rigid, 15 ft.; total 31 ft. 4 in. Working pressure 180 lb. per sq. in. The heating surface is made up as follows:—Small steel tubes 1,256 sq. ft.; large tubes 705 sq. ft.; steel firebox 140 sq. ft.; two thermic syphons in firebox 42 sq. ft.; total evaporative surface 2,143 sq. ft.; superheater 583 sq. ft.; total 2,726 sq. ft. Firegrate area 41.4 sq. ft. The boiler feed is by an exhaust steam injector and a Hancock injector. Three 2½-in. Ross pop safety valves and a chime whistle, pneumatic cylinder cocks, "Cambridge" pyrometers, and a "Teloc" speed recorder, are included in the special fittings. A mechanical stoker with a stoker engine, is included in the equipment, with rocking grate and grate shaker, and on the sides of the firebox are fitted two Clyde superior tube cleaners. The superheater header is of the multiple valve type. Electric lighting equipment is provided, and pneumatic sanding gear. The automatic air brake on engine and tender and the steam heating apparatus have been furnished by the Westinghouse Brake Co., of London. Automatic couplers of the standard D pattern are fitted on the engine and tender. Estimated at 75% of the boiler pressure, the tractive force is 30,870 lb. In working order the engine weighs 84 tons 5 cwt., of which 62 tons 1 cwt. are on the coupled wheels. The bogie tender has wheels 2 ft. 9 in. diameter, with a wheelbase of 18 ft. 7 in. It carries 5,000 gallons of water and 10 tons of fuel (450 cu. ft.). Loaded, the tender weighs 59 tons 19 cwt. The total wheelbase of engine and tender is 59 ft. 1½ in.

Fig. 1.—Arc welding field connections on pipe line

Arc Welding for Pipe, Piping and Pipe Lines*

By G. G. Landis†

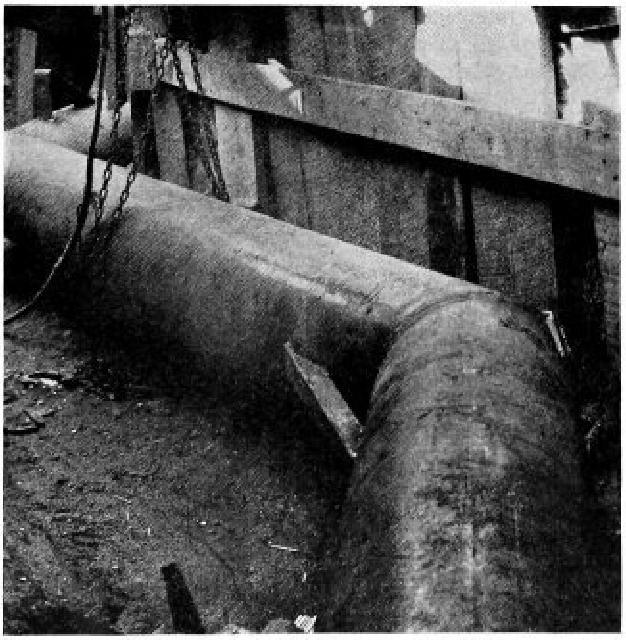


Fig. 2.—Offset in 16-inch steam line

Widely discussed these days, is the question, can arc-welded vessels withstand severe operating conditions satisfactorily? To the boiler maker such a consideration is of major importance. The users of welding believe that the process is entirely satisfactory and the large amount of pressure equipment made in the past has justified, in service, this faith.

Among the most expensive single units constructed today are the great pipe lines for the transmission of gas and petroleum. Leaks on such a line mean suspension of service and costly maintenance operations. It is not surprising to note that since the introduction of the shielded arc process in July, 1929, almost 4000 miles of pipe line, such as that shown in Fig. 1, have been constructed by this method. The use of the process has been stimulated by several factors:

It is the quickest way of making a permanent field connection, the welding time being less than one minute per inch diameter of pipe for two beads, cleaned.

Joints so made in 0.20 to 0.30 carbon-steel pipe have a tensile strength greater than the pipe metal (126 coupons averaged over 77,000 pounds per square inch).

Joints are ductile and resistant to corrosion.

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* Fifth article of a series dealing with arc welding in various branches of the boiler and pressure vessel field.

† Chief engineer, The Lincoln Electric Company, Cleveland, O.

Such construction insures a leak-proof joint under all conditions.

Many of these pipe lines are tested under 150 to 200 pounds pressure without a leak, and one contractor who has made over 350,000 joints with this process has not reported a failure.

The same pipe line procedure is used in the construction of high-pressure steam lines. Steam pipe lines used for district heating are now put into service arc welded, because, as one contractor put it—"a welded line is down for good." A 16-inch line operating at 250 pounds of steam at 450 degree F. of heat is shown in Fig. 2. An obsolete fire cistern necessitated the bend shown. The entire bend was fabricated in the field and at low cost. The structural brace across the angle and the strap on the joint are to prevent the pipe from buckling near the joint under temperature.

In St. Louis, over a mile of 20-inch high-pressure steam line was laid which is notable because of the field procedure. The connections are plain butt joints beveled at 30 degrees. No backing up ring or chill band was used, the beads of weld metal being run on the pipe as it was lined up, without rolling the pipe. This required overhead welding, which with the shielded-arc process permitted joints which tested stronger than the pipe itself.

Power plant engineers have long used the electric arc process on high-pressure headers and fittings. In these applications, for leak-proof security, arc welding has become almost standard procedure.

Large diameter water pipe, while not working under high pressure, offers a rigid test for arc welding. Such pipe is subject to great variations in temperature and many shock loads, due to water hammer. Such pipe is both manufactured and laid with the aid of arc welding, the former application being more widely used than the latter.

The automatic process is used in the production of water pipe. Fig. 3 shows the electronic tornado equipment used in the tank and boiler shop of Farrar and Trefts, Buffalo, N. Y. This is 66-inch diameter water pipe made from ½-inch plate in 40-foot sections for the city of Detroit. In Fig. 3, the tractor is shown on the inside of the section. The plates are rolled in half sections and fitted together with a filler strip between the butted plates. These filler strips are tack welded in place and the tractor-mounted welding heads set in action. After the first pass the pipe is rotated and the operation repeated. Each seam is welded from both sides, and with two tractors the speed per seam is 25 feet per hour for the double pass. Welded seams on this type of pipe are better than the pipe metal.

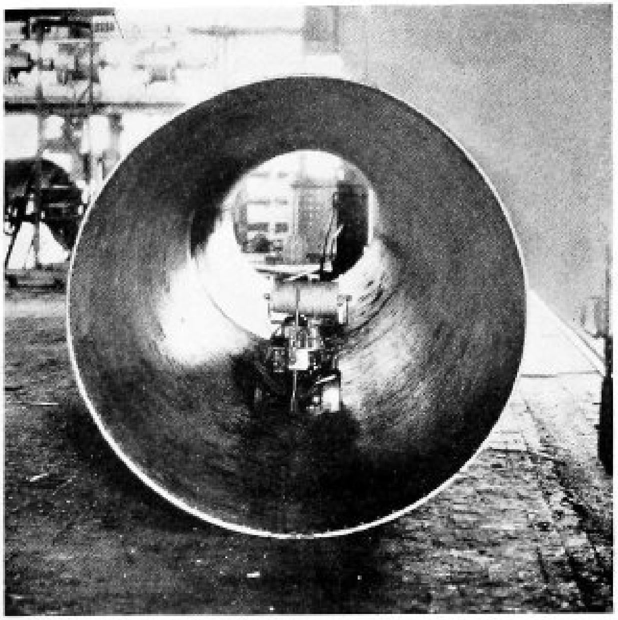


Fig. 3.—Welding inside seam of 66-inch pipe

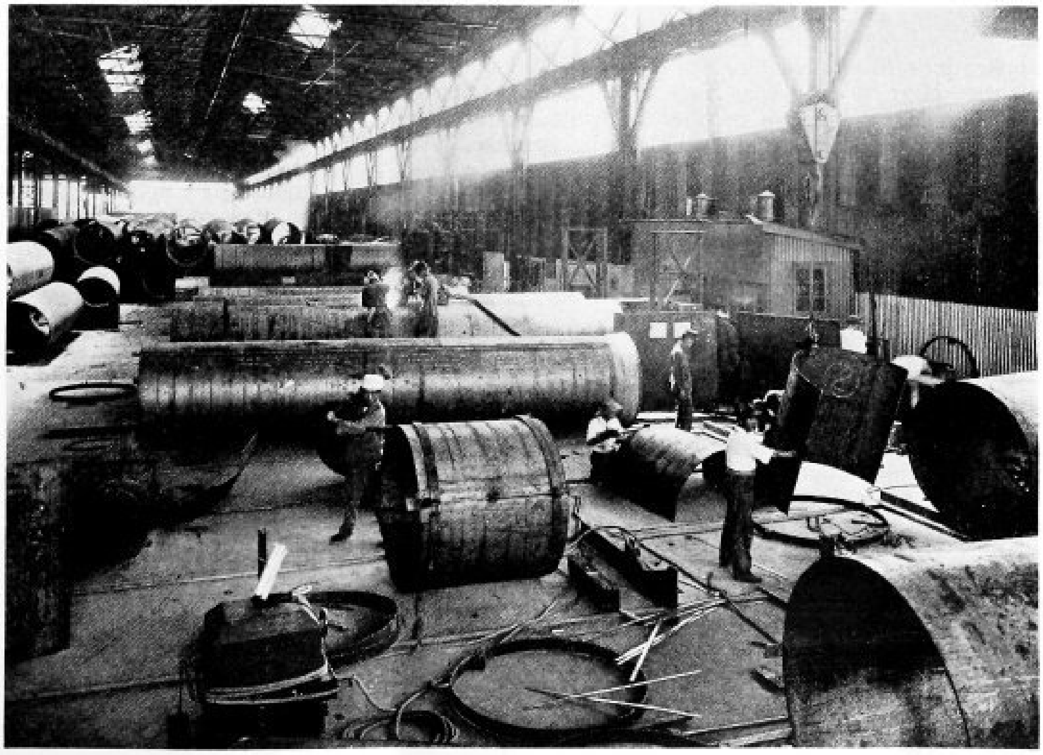


Fig. 4. — Arc - welded tanks and pipe are produced in Japan by the Kawasaki Dock Yard Company, Ltd.

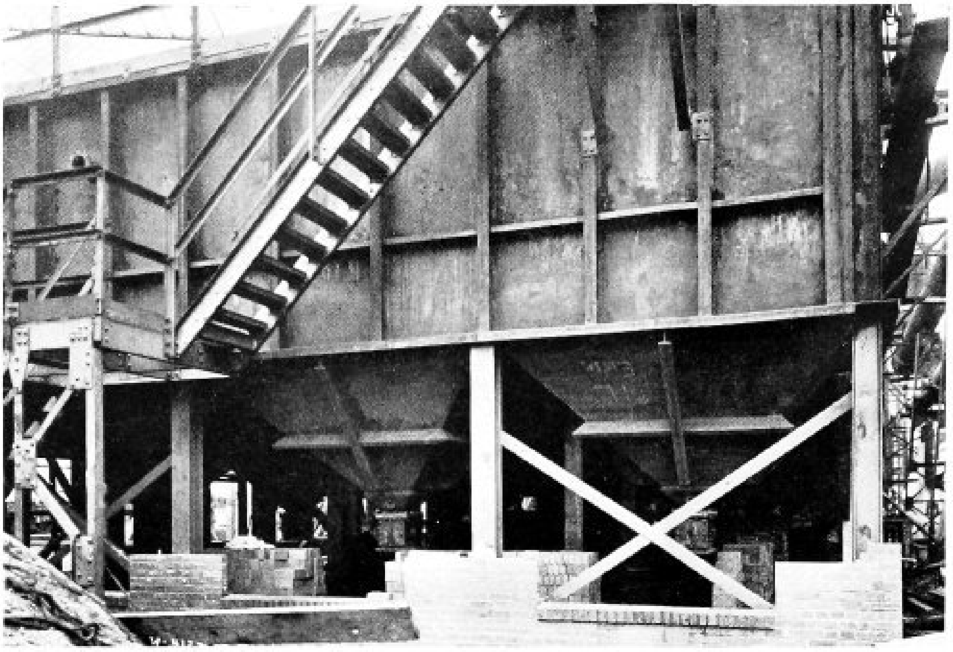


Fig. 5.—Lower side of 120-foot by 30-foot processing unit showing type of structural brace

An international view of arc welding economy is seen in Fig. 4, which shows the tank and pipe shop of the Kawasaki Dock Yard Company, Ltd. of Kobe, Japan, where the electronic-tornado process has just been installed. Here five miles of 2100 millimeter water pipe is being built for the new Tokyo water supply system. This pipe is connected in the field with arc welding. This requires manual welding overhead on the inside of the pipe, but with the shielded-arc process this type of connection has withstood all tests both from the time and cost factors.

The Kawasaki Dock Yard Company uses arc welding in a great number of high-pressure applications, especially on pipes and containers. Great believers in welding, they have utilized the economies of the process well.

While we are reviewing the uses of arc welding in the field of piping it is well to consider the numerous applications of arc welding in refineries and chemical plants. Here, pressures as high as 800 pounds must be considered and because of the materials handled the danger of disaster is of tremendous importance. Arc welding is used on steel containers, piping, purifiers, processing units and storage tanks. Where corrosion is a large factor, stainless steel units are welded with an 18-8 alloy electrode which carries the heavy extruded coating of the shielded-arc process. Engineers in many refineries, from their experience with arc-welded pipe lines, use the process without question. It is obviously these men who are most familiar with arc welding who reap its greatest benefits.

An interesting example of the use of arc welding in the construction of a processing unit is shown in Fig. 5.

This is a one-hundred-and-twenty-foot by thirty-foot tank built to withstand rugged use and service. The unit was recently put into operation at the Lackawanna plant of the Bethlehem Steel Corporation. The Koppers Construction Company and the Buffalo Tank Company designed and erected the unit. The only bolts in the tank itself are on the covers which must be removable. Lifting rings, manholes, etc., are all welded to the plate. Note the hoppers below the main body of the unit. Plate ¾ inch to 1 inch thick is used in these hoppers, which are stiffened with structural I beams and channels.

The tank is supported on 9-inch H sections anchored to the foundation. The plate used ranged from ⅜ inch to one inch in thickness. About 260 tons of steel were used and the job was completed with eight weeks of shop and field welding, both operations being done simultaneously.

Beside this new tank is the old unit which it replaces. Bolts, castings, expensive fittings complicate the discarded unit. The new tank was not only inexpensive to erect but, because of seamless construction, will be free of maintenance operations.

The Koppers Construction Company has specified over 2 miles of arc-welded gas pipe of large diameter in this mighty plant at Lackawanna. Today intelligent industrialists are seeking new applications for arc-welding economy.

Combustion Corporation Awarded Government Contract

Combustion Engineering Corporation announces the receipt of a contract from the United States Treasury Department, Washington, D. C., for four 2500-horsepower Walsh-Weidner sectional header boilers and four C-E multiple retort stokers, super-station type. This equipment is to be installed in the new Triangle heating plant, Washington, D. C., which plant will provide heat for the various buildings included in the new Federal building development.

In addition to boilers and stokers, the contract includes steel-encased boiler settings, water-cooled furnaces and forced and induced draft fans and drives.

The boilers are designed for 400 pounds per square inch pressure and are guaranteed for a production of 215,000 pounds of steam an hour continuous and 237,000 pounds maximum, from and at 212 degrees F.

The stokers will be 15 retorts wide, 45 tuyeres long, and will have a projected grate area of 480 square feet each. They will be equipped with double-roll clinker grinders.

The consulting engineers on this project are United Engineers and Constructors, Inc., Philadelphia.

Layout of Welt Strips

By William Morrison

The layout of welt strips is a subject that authors on laying out problems often ignore. In actual practice the layerout in the shop generally is furnished with a longitudinal seam drawing showing the pitch of the rivets and the circumferential distances between the rivet rows. These circumferential distances are the distances to be laid out on the shell. Referring to the sectional end view, Fig. 1, these circumferential distances between the rivet rows on the outside welt *increase* as x and x' and on the inside welt they *decrease* as z, z' and z^2. The layerout, therefore, must have some method of calculating this increase and decrease.

Some layerouts make a full-size sectional end view as shown in Fig. 1 and transfer these circumferential distances from the neutral axis as x, x', z, z' and z^2 to the flat plate. This method, although used for a good many years by layerouts to come within a certain degree of accuracy, is fast being discarded in modern shops. I do not believe any layerout would attempt to calculate the circumference of a shell from a full-size drawing; nor should we use this method for figuring these circumferential distances. We should have some method of calculating them.

Another method used by many layerouts is known as the principle of *constant increase* and *decrease* between the rivet rows in the layout of welt strips, disregarding the diameter of the shell and the thickness of the plates used. For example, in Fig. 1 the distance x would be 1/32 inch *larger;* x' would be 1/16 inch *larger;* z would be 1/32 inch *less;* z' would be 1/16 inch *less* and z^2 would be 3/32 inch *less* than that shown on the longitudinal seam drawing. Applying this constant method in laying out welt strips to a given case is not serious when the diameter is large in comparison to the plate thickness; but when the plate thickness is large and the diameter is small, a considerable error is introduced; and the increase and decrease is not constant.

According to the teaching of geometry, the length of an arc subtending a degree depends upon the diameter of the circle on which it is measured. To illustrate this principle, let the angle *R-O-S*, Fig. 2, subtend 5 degrees on the arc *P-N;* draw another larger arc, *R-S*, and lay off on *R-S* the points *1°*, *2°*, *3°*, *4°* and *5°* the same distances from *R* as *1°*, *2°*, *3°*, *4°* and *5°* are from *P* on *P-N*, giving the arc *R-T* the same length as the arc *P-N*. It is evident that the angle *R-O-T* is less than the angle *R-O-S*. By inspection *R-A*, which is a fifth of the arc *R-S*, is greater than *R-1°* or its equal *P-1°* on the arc *P-N*. These circumferential distances between the rivet rows of a welt strip are small arcs of a certain degree in length. As these arcs depend upon the diameter of the circle on which they are measured, and we know there is no such thing as a constant diameter, there is no such thing as a constant *increase* and *decrease* between the distances of the rivet rows.

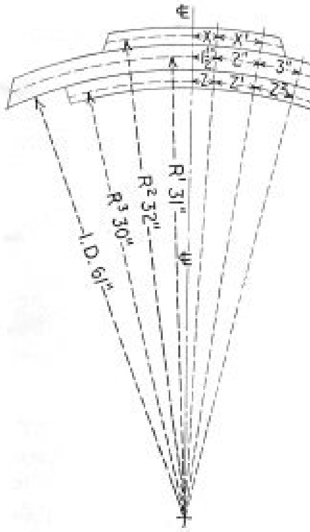


Fig. 1.—Sectional end view of welt strip

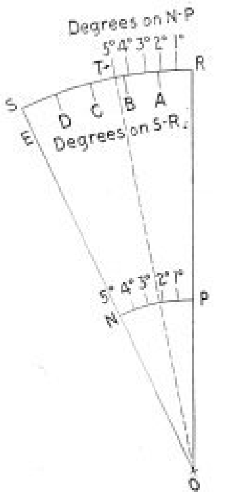
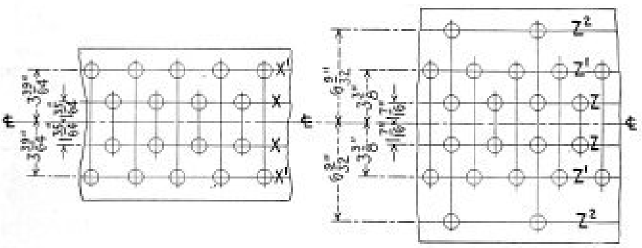


Fig. 2.—Error in using degree principle



Figs. 3 and 4.—Welt strip layout

Referring to Fig. 1, it can be plainly seen that on the neutral axis of the shell these circumferential distances are known, and from these we can solve this increase and decrease. We have, however, two problems before us. First, the neutral axis of the shell is known as a small arc with a known distance, and the neutral axis of the outside welt is known as the large arc with the unknown distances x and x' in laying out the outside welt strip. Second, in laying out the inside welt strip, the neutral axis of the shell is known as the large arc with a known distance and the neutral axis of the inside welt is known as the small arc with the unknown distances z, z', and z^2. Hence, we must have two rules; first, to find the length of the long arc from the length of the short arc, and second, to find the length of the short arc from the length of the long arc. The following two rules give a close degree of accuracy:

Rule No. 1.—To find the length of the long arc from the short arc, multiply the length of the short arc by the long radius and divide the product by the short radius.

Rule No. 2.—To find the length of the short arc from the long arc, multiply the length of the long arc by the short radius and divide the product by the long radius.

In the layout of the outside welt use rule No. 1. In the layout of the inside welt use rule No. 2. In addition, we must add these small arcs together as we work from the center line. Thus the length of the first arc is 1½ inches from the center line; the length of the second arc is 1½ + 2 = 3½ inches from the center line, etc. The conditions under which rules Nos. 1 and 2 are applied are indicated by the following example: Referring to Fig. 1, let the inside diameter of the shell be 61 inches and the plate thickness for the shell and welts be 1 inch. The circumferential distances between the rivet rows are as shown.

To layout the outside welt by applying rule No. 1:

$$\frac{1\tfrac{1}{2} \times 32}{31} = 1\ 35/64 \text{ inches} = x \text{ from the center line.}$$

$$\frac{3\tfrac{1}{2} \times 32}{31} = 3\ 39/64 \text{ inches} = x + x' \text{ from the center line.}$$

To layout the inside welt by applying rule No. 2:

$$\frac{1\tfrac{1}{2} \times 30}{31} = 1\ 7/16 \text{ inches} = z \text{ from the center line.}$$

$$\frac{3\tfrac{1}{2} \times 30}{31} = 3\tfrac{3}{8} \text{ inches} = z + z' \text{ from the center line.}$$

$$\frac{6\tfrac{1}{2} \times 30}{31} = 6\ \ 9/32 \text{ inches} = z + z' + z^2 \text{ from the center line.}$$

In rule No. 1, R^2 is the long radius, R' is the short radius and 1½ inches and $1\tfrac{1}{2} + 2 = 3\tfrac{1}{2}$ inches are the short arcs.

In the rule No. 2, R' is the long radius, R^3 is the short radius and 1½ inches, $1\tfrac{1}{2} + 2 = 3\tfrac{1}{2}$ inches and $1\tfrac{1}{2} + 2 + 3 = 6\tfrac{1}{2}$ inches are the long arcs.

Figs. 3 and 4 show the solutions as laid out from the center line of the welt strips from the foregoing example. Comparing rivet rows x and x' in Fig. 3 with z and z' in Fig. 4 again proves that the increase and decrease between the rivet rows are not constant.

As the dimensions for the longitudinal section of a welt strip do not change from those shown on a seam drawing, no discussion of the subject need be included here.

Work of the A. S. M. E. Boiler Code Committee

The Boiler Code Committee meets monthly for the purpose of considering communications relative to the Boiler Code. Any one desiring information as to the application of the code is requested to communicate with the secretary of the Committee, 29 West 39th St., New York, N. Y.

The procedure of the committee in handling the cases is as follows: All inquiries must be in written form before they are accepted for consideration. Copies are sent by the secretary of the committee to all of the members of the committee. The interpretation, in the form of a reply, is then prepared by the committee and passed upon at a regular meeting of the committee. This interpretation is later submitted to the council of The American Society of Mechanical Engineers for approval, after which it is issued to the inquirer and published.

Below are given records of the interpretations of the committee in Cases Nos. 688, 694, 700, 704, and 705-709 inclusive, as formulated at the meeting on January 15, 1932, all having been approved by the council. In accordance with established practice, names of inquirers have been omitted.

Case No. 668.—*Inquiry:* When used in the construction of the boiler pressure parts, can material made in accordance with Specification S-2 be considered as being in compliance with Pars. P-2 and P-3 of the code?

Reply: Revised A.S.T.M. Specification A89-31T provides for firebox-quality plate. The Boiler Code Committee recommends that, pending the adoption of revisions to Specification S-2 for Steel Plates of Flange Quality for Forge Welding to make them identical with A.S.T.M. Specification A89-31T, the use of this revised specification be permitted as being in compliance with Pars. P-2 and P-3.

Case No. 694.—*Inquiry:* Is it the intent of Par. P-314 of the code, which specifies that feed water shall not be introduced into a boiler in such a manner as to be discharged close to riveted joints of shell or furnace sheets, that this limitation shall apply to the longitudinal riveted joints in the barrels of locomotive-type boilers?

Reply: There are ordinarily no structural difficulties that would make it necessary to introduce feed water into a boiler close to a riveted joint, and as this practice is definitely prohibited by Par. P-314, it is the opinion of the committee that the point of feed entry should be kept removed from any riveted joint in all types of boilers.

Case No. 700.—*Inquiry:* Is it the intent of the requirement for the nick-break test in Par. U-69 of the code to absolutely prohibit more than six gas pockets per square inch of any size?

Reply: It is the opinion of the committee that the intent of the requirement for nick-break test in Par. U-69 will be met if there are more than six gas pockets per square inch, provided the total area thereof does not exceed the area of six gas pockets, each $\frac{1}{16}$ inch in diameter.

Case No 704.—*Inquiry:* What is the maximum allowable unit joint working stress for Class 3 vessels, with double-welded lap-type longitudinal joints for thickness of ⅜ inch or less under the provisions of Par. U-73 of the code? It is noted that this allowable working stress is not provided for in Par. U-70.

Reply: It is the opinion of the committee that the maximum unit joint working stress for double full-fillet lap welds for longitudinal joints should be 7000 pounds. It is therefore proposed to insert the following as the third section of Par. U-70:

> For double full-fillet lap welds for longitudinal joints and for material thickness of ⅜ inch or less, the maximum unit joint working stress ($S \times E$) shall not exceed 7000 pounds per square inch.

Case No. 705.—*Inquiry:* Is it necessary to limit the total length of the specimen and the distance between the supports as shown in Fig. U-13 of the code for the nick-break test?

Reply: It is the opinion of the committee that the dimension "about 6 inches" in Fig. U-13 is too restrictive and should be replaced by a reference letter A, with the definition that "A is to be about 3 inches for ¼-inch plate and as close together as practicable for greater thicknesses."

Case No. 707.—*Inquiry:* Request is made for a revision of that part of Par. P-314 of the code which provides for location of the feed pipe in an horizontal return tubular boiler above the central row of tubes when the shell diameter exceeds 36 inches, so that it will apply only to boilers over 40 inches in diameter.

Reply: It is the opinion of the committee that the present requirement for the discharge of the feed water in horizontal return tubular boilers in excess of 36 inches should be changed to apply to boilers in excess of 40 inches in diameter. It is therefore proposed to revise this requirement so as to make it apply to horizontal return tubular boilers in excess of 40 inches.

Case No. 708.—*Inquiry:* How is the allowable working pressure for a corrugated bend in seamless pipe to be determined under the code requirements?

Reply: In computing the allowable working pressure of a corrugated pipe, it should be figured for the original pipe from which the corrugated bend is made and the dimensions of which are the same as the end straight portions of the pipe where it is not corrugated, provided there is no thinning down in the corrugations.

CASE No. 709.—*Inquiry:* Will it be permissible under the requirements of the Power Boiler Section of the Code to use electric-resistance-welded steel pipe or tubing which meet the requirements as to chemical and physical properties of Specifications S-18 and S-17 in the construction of code boilers?

Reply: The Boiler Code Committee recommends that electric-resistance-welded steel pipe or tubing be accepted under the classification of lap-welded pipe (Specification S-18) and lap-welded tubes (Specification S-17) provided they meet all chemical, physical, and other requirements of lap-welded steel pipe or tubes in those specifications.

An Unusual Marine Boiler Repair

By G. P. Blackall

The largest patch that has ever been welded into the back-end plate of a marine boiler in the port of London has turned out a successful job and the boiler is now in regular operation. This boiler had four corrugated furnaces, the two lower furnaces having one common combustion chamber, the back-end plate of which was found to be cracked over a fairly large area between the screwed stays.

After survey, it was decided to cut out the cracked plate over an area equivalent to 3 feet 9 inches by 3 feet 9 inches of ⅝-inch original thickness. Suitable wooden templates were then made, marking off the stay ends within this area, and one row of stays beyond all around. This template was required again later before welding was completed.

At this point, the stays were removed by burning out the ends from the firebox, and also the back-end plate in the engine room. After being carefully marked off, the old plate in the combustion chamber was cut out by the same process and removed. The four edges of the remaining plate were prepared for welding by beveling at an angle (to meet the new plate) of 70 degrees. The four edges of the patch of 3 feet 9 inches by 3 feet 9 inches by ⅝-inch Lowmoor iron were then treated in a similar manner, and in addition all stay holes were marked off and drilled 1⅛-inch diameter to template.

The patch was then placed in position and temporary ⅞-inch staybolts were inserted through to the boiler back-end plate with suitable nuts and washers, screwed up only hand tight. These temporary stays allowed the patch to float in all directions depending upon the locality of the arc during welding. The actual welding was under constant supervision, and, as an instance of the precautions taken, it may be stated that sixteen points were marked off and perfectly trammeled before welding started. The trammel was constantly used to check the movement of the patch in any direction. This method controlled the speed of welding.

The following points are worth noting:

The old plate was cut through the stay holes and not through the solid plate between the stays.

Half stay holes were filled in, at the junction of the vee'd plates, as welding progressed.

After the V and half holes had been completed, it was decided to weld solid pads over every stay hole in the weld to take the place of the usual washers.

The presence of these solid pads 3⅛ inches by ¼-inch thick considerably reduced, to advantage, the length of weld exposed to furnace flames.

Steps were taken to avoid the use of pneumatic tools for this work, and in any case the weld metal required very little cleaning up.

A hydraulic test of 320 pounds per square inch and steam tests of 160 pounds per square inch were afterwards applied.

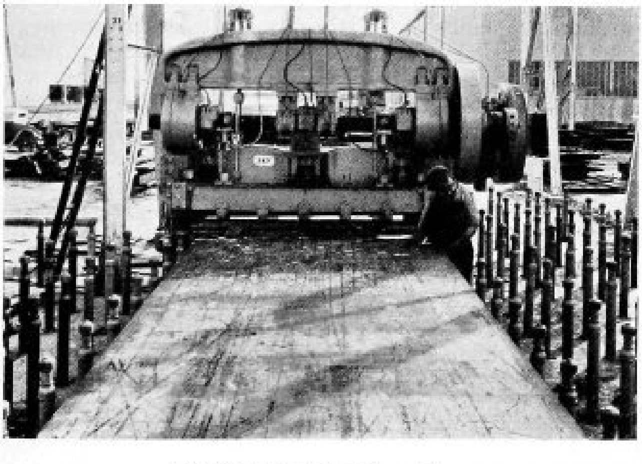
During the steam test the two low fires were drawn, planks and sacking placed along the firebars and the back end reached for survey, while the safety valves were blowing off steam from above.

The patch stood both tests successfully and the boiler is in regular and satisfactory operation.

A total of 76 working hours was required, which included fullering, chipping and filing, and the welding plant standing by for hydraulic and steam tests.

Ball Transfers for Plate Shears

The illustration shows an installation of Mathews ball transfers for facilitating the handling of large plates. In this particular installation several hundred heavy duty type-500 ball transfers were employed each of which was installed on a small pillar, as shown. The plate which is being handled is 1 inch thick, 8 feet wide, 30 feet long and weighs approximately 10,000 pounds.



Installation of ball transfers

This plate, which is to be trimmed in the power shears illustrated, was brought from storage by an overhead crane and pulled on to the ball transfer area by air-operated winches. The two operators on this shear easily manipulated this 10,000-ton plate.

Ball transfers are manufactured by the Mathews Conveyor Company, Ellwood City, Pa.

The Okadee Company and the Viloco Railway Equipment Company, formerly located at 14 East Jackson Boulevard, Chicago, Ill., have moved to 611 McCormick Building, 332 South Michigan Avenue, Chicago.

Revisions and Addenda to the A. S. M. E. Boiler Construction Code

It is the policy of the Boiler Code Committee, of the American Society of Mechanical Engineers, to receive and consider as promptly as possible any desired revision of the Rules and its Codes. Any suggestions for revisions or modifications that are approved by the committee will be recommended for addenda to the code, to be included later on in the proper place in the code.

The following proposed revisions have been approved for publication as addenda to the code. They are published below with the corresponding paragraph numbers to identify their locations in the various sections of the code, and are submitted for criticisms and approval from any one interested therein. Communications should reach the secretary of the Boiler Code Committee, 29 West 39th Street, New York, N. Y., in order that they may be presented to the committee for consideration.

Par. P-2. Revised:

P-2. *a* Steel plates for any part of a boiler subject to pressure and exposed to the fire or products of combustion shall be of firebox quality in accordance with Specifications S-1 or S-2.

b Steel plates for any part of a boiler subject to pressure and not exposed to the fire or products of combustion shall be of firebox or flange quality in accordance with Specifications S-1 or S-2.

c Minimum tensile-strength limits other than those specified for flange and firebox plates may be used provided a range of 10,000 lb. per sq. in. is specified, that the maximum limit does not exceed 65,000 lb. per sq. in., and that the steel conforms in all other respects to Specifications S-1 and S-2.

d Seamless steel drum forgings made in accordance with Specifications S-4 may be used for any part of a boiler for which firebox quality only is specified, or for which either firebox or flange quality is permitted.

Par. P-3. Revised:

P-3. Open-hearth-steel pipe or steel tubing in accordance with Specifications S-18 or S-17 may be used for a boiler drum or other pressure part exposed to the fire or products of combustion, provided the nominal diameter of the pipe or tubing is not greater than 18 in.

Par. P-9. Delete Second Sentence of Second Section of This Paragraph.

Par. P-21. Revised:

P-21. Generating tubes for boilers expanded into tube seats shall comply with Specifications S-17 for Lap-Welded and Seamless Steel and Lap-Welded Iron Boiler Tubes, or Specifications S-22 for Seamless Copper Boiler Tubes.

Open-hearth-steel pipe or wrought-iron pipe, not to exceed 1½ in. pipe size, which meets the Specifications for Steel or Wrought-Iron Pipe, may be used for watertube boilers for a working pressure not to exceed 250 lb. per sq. in., when screwed in the sheet or fittings, provided the wall thickness is at least 50 percent greater than the minimum wall thickness required by Table P-2 or P-3.

Par. P-22. Revised:

P-22. The maximum allowable working pressures for steel or wrought-iron tubes or nipples used in water-tube boilers shall be as given in Table P-2 or P-3. The minimum thicknesses of tubes used in fire-tube boilers, for various maximum allowable working pressures, shall be as given in Table P-5.

The maximum allowable working pressure for copper tubes or nipples used in water-tube or fire-tube boilers, shall be as given in Table P-4, but they shall not be used for pressures exceeding 250 lb. per sq. in. nor for temperatures exceeding 406 F.

Tables P-2, P-3, and P-5. Add the Words:

"Conforming to the Requirements of Specifications S-17," to the titles of these tables.

Table P-4. Add the Words:

"Conforming to the Requirements of Specifications S-22," to the title of this table.

Par. P-25. Add Following Sentence:

Galvanized pipe shall not be used.

Par. P-186. Add Following Sentence to Third Section of This Paragraph.

Furnaces subjected to compression stresses may be fusion-welded provided the welds are stress-relieved in accordance with Par. P-108 and bend tests of at least one sample for each furnace of the welding used meet the requirements of Par. P-102.

Par. P-273. Insert the Following as Second Sentence of This Paragraph:

The maximum allowable pressure at which the safety valve may be operated shall be cast or stamped on the body of the valve.

Par. P-288. Revised:

P-288. *a* Every attached superheater shall have one or more safety valves near the outlet. The discharge capacity of the safety valve, or valves, on an attached superheater may be included in determining the number and size of the safety valves for the boiler, provided there are no intervening valves between the superheater, safety valve, and the boiler, and provided the discharge capacity of the safety valve or valves on the boiler, as distinct from the superheater, is at least 75 percent of the aggregate valve capacity required.

b Every independently fired superheater shall have one or more safety valves having a discharge capacity based on Par. P-270, provided that valves are fitted which permit the superheater to become a fired pressure vessel. If communication between the superheater and the boiler cannot be closed, safety valves shall be provided as prescribed in Par. P-288*a*.

c A soot-blower connection may be attached to the same outlet from the superheater that is used for the safety-connection.

Par. P-296. Revise First Sentence of Second Section to Read.

Where the use of a pipe longer than 10 ft. becomes necessary, an exception may be made to the rule that the gage must be arranged so that it cannot be shut off except by a cock placed near the gage, and a shut-off valve or cock arranged so that it can be locked or sealed open may be used near the boiler.

Par. P-303. Revised:

P-303. When boilers are connected to a common steam main, the steam connection from each boiler having a manhole opening shall be fitted with two stop valves having an ample free-blow drain between them. The discharge of this drain valve must be visible to the operator while manipulating the valve. The stop valves shall consist preferably of one automatic non-return valve (set next the boiler) and a second valve of the outside-screw-and-yoke type; or, two valves of the outside-screw-and-yoke type shall be used.

Par. P-308. Revised:

P-308. Each boiler shall have a bottom blow-off pipe fitted with a valve or cock in direct connection with the lowest water space practicable.

The minimum size of pipe and fittings shall be 1 in., and the maximum size shall be 2½ in., except that for boilers with 100 sq. ft. of heating surface or less, the minimum size of pipe and fittings may be ¾ in.

Straight-run globe valves of the ordinary type as shown in Fig. P-23, or valves of such type that dams or pockets can exist for the collection of sediment, shall not be used on such connections.

Straight-way "Y"-type globe or angle valves may be used in vertical pipes, or they may be used in the horizontal runs of piping provided they are so constructed or installed that the lowest edge of the opening through the seat shall be at least 25 percent of the pipe diameter below the center line of the valve.

Return connections of the same size or larger than the size herein specified may be used, and the blow-off may be connected to them. In such case the blow-off must be so located that the connection may be completely drained.

All water walls or water screens forming parts of a steam boiler shall be equipped with drain or blow-off valves conforming to the requirements of this paragraph and Par. P-311 for boilers.

Par. P-310. Add Following Sentence:

Galvanized pipe shall not be used.

Par. P-311*a*. Add Following Sentence:

The drain or blow-off valves for water walls or water screens forming parts of a boiler shall conform to the requirements of this paragraph and also of Par. P.-308.

Par. P-314. Revised:

P-314. *Feed Piping.* The feedwater shall be introduced into a boiler in such a manner that the water will not be discharged directly against surfaces exposed to gases of high temperature or to direct radiation from the fire, or close to any riveted joints of the furnace sheets or of the shell. Feedwater, other than condensate returned as provided for in Par. P-308, shall not be introduced through the blow-off except for boilers having over 2500 sq. ft. of heating surface, where it may be used as an emergency feed connection only in addition to the regular feed connection.

When a horizontal-return tubular boiler exceeds 40 in. in diameter, the feedwater shall discharge at about three-fifths the length from that end of the boiler which is subjected to the hottest gases of the furnace (except a horizontal-return tubular boiler equipped with an auxiliary feedwater heating and circulating device), above the central rows of tubes. The feed pipe shall be carried through the head or shell furthest from the point of discharge of the feedwater in the manner specified for a surface blow-off in Par. P-307, and can be securely fastened inside the shell above the tubes.

In vertical tubular boilers, feedwater shall be introduced at a point not less than one-fourth the length of the tube for tubes 4 ft. and less in length above the lower tube sheet or crown sheet. For tubes more than 4 ft. in length, the feedwater shall be introduced at a point not less than 12 in. above the crown sheet.

Table P-11. Insert the Following Notation Under This Table:

The discharge capacities given in the above table may be interpolated to determine the values for intermediate pressures.

Par. P-332. First Section Revised:

Each boiler shall conform in every detail to these Rules, and shall be distinctly stamped with the symbol as shown in Fig. P-25, denoting that the boiler was constructed in accordance therewith.

Revised First Sentence of Second Section:

After obtaining the stamp to be used when power boilers are to be constructed to conform with this section of the A.S.M.E. Boiler Code, a state inspector, or a municipal inspector of any state or municipality that has adopted the code, or an inspector employed regularly by an insurance company and who is qualified under the rules of such states or municipalities to inspect boilers, is to be notified that an inspection is to be made; and he shall inspect such boilers during construction and after completion.

Revised Last Sentence of Second Section to Form a Separate Section:

This data sheet, together with the stamping on the boiler, shall be a guarantee by the manufacturer that he has complied with all the requirements of this section of the code.

Specifications S-2.

Revisions have been proposed in Pars. 1, 2, 3, 6, 7, 9, 10, 11, and 14 of this specification to make them correspond with revised paragraphs in identical A.S.T.M. Specifications A89-31T. For detail schedule of the revisions, communicate with the secretary of the Boiler Code Committee.

Specifications S-3.

The following paragraphs have been proposed for revisions:

Insert a paragraph at the beginning of the specifications to read:

Scope. These specifications cover steel plates or sheets under ¼ in. in thickness to be used for brazing.

Par. 5. Revise to read:

Two tension tests and two bend tests shall be taken from each heat; each test from a different sheet or plate as rolled.

Par. 6. Revise to read:

6. *Marking.* The name or brand of the manufacturer and the melt number shall be legibly stenciled on each sheet or plate.

Specifications S-9.

Revisions have been proposed in Pars. 1, 10*a*, and 12 of these specifications to make them identical with revised paragraphs in A.S.T.M. Specifications A96-31. For detail schedule of the revisions, communicate with the secretary of the Boiler Code Committee.

Specifications S-19.

A revision has been proposed in Par. 13 of these specifications to make them identical with corresponding revised paragraph in A.S.T.M. Specifications A72-31. For detail schedule of the revision, communicate with the secretary of the Boiler Code Committee.

Exemption Clause Preceding Par. L-1. Revise Exemption Clause Preceding Par. L-1 to Read:

These rules are not intended to apply to boilers of locomotives which are subject to Federal inspection and/or control.

Par. L-82. Revise First Sentence of Second Section to Read:

After obtaining the stamp to be used when locomotive boilers are to be constructed to conform with this section of the A.S.M.E. Boiler Code, a state inspector, or a municipal inspector of any state or municipality that has adopted the code, or on inspector employed regularly by an insurance company and who is qualified under the rules of such states or municipalities to inspect boilers, is to be notified that an inspection is to be made; and he shall inspect such boilers during construction and after completion.

Revise Last Sentence of Second Section to Read:

This data sheet, together with the stamp on the boiler, shall be a guarantee by the manufacturer that he has complied with all the requirements of this section of the code.

Par. H-35. Revised:

H-35. Boilers of the wet-bottom type having an external width of over 36 in. shall have not less than 12 in. between the bottom of the boiler and the floor line, with access for inspection. When the width is 36 in. or less the distance between the bottom of the boiler and the floor line shall not be less than 6 in., and when the width of the wet part of the bottom is 24 in. or less, it shall not be less than 4 in.

Pars. H-46 and H-99 Revise First Sentence to Read:

Safety valves shall be connected to the boilers independent of other connections, and be attached directly or as close as possible to the boiler without any unnecessary intervening pipe or fitting except the Y-base forming a part of the twin valve or a steam equalizing pipe between boilers.

Pars. H-55 and H-108. Revise Third Section to Read:

Connections to steam-gage siphons shall be of nonferrous metal when smaller than 1-in. pipe and longer than 5 ft. between the siphon and point of connection of pipe to boiler, and also when smaller than ½-in. pipe and shorter than 5 ft. between the siphon and point of connection of pipe to boiler.

Par. H-66. Add the Following Sentence:

After obtaining the stamp to be used when welded heating boilers are to be constructed to conform with this section of the A.S.M.E. Boiler Code, a state inspector, or a municipal inspector of any state or municipality that has adopted the code, or an inspector employed regularly by an insurance company and who is qualified under the rules of such states or municipalities to inspect boilers, is to be notified that an inspection is to be made; and he shall inspect such boilers during construction and after completion.

Par. H-81. Revise Last Sentence to Read as Follows:

The inspector shall be a state inspector, or municipal inspector of any state or municipality that has adopted the code, or an inspector employed regularly by an insurance company and who is qualified under the rules of such states or municipalities to inspect boilers.

Par. M-20. Add the Following Sentence to the Second Section:

After obtaining the stamp to be used when miniature boilers are to be constructed to conform with this section of the A.S.M.E. Boiler Code, a state inspector, or a municipal inspector of any state or municipality that has adopted the code, or an inspector employed regularly by an insurance company and who is qualified under the rules of such states or municipalities to inspect boilers, is to be notified that an inspection is to be made; and he shall inspect such boilers during construction and after completion.

Add Following Section to This Paragraph:

This data sheet, together with the stamping on the boiler, shall be a guarantee by the manufacturer that he has complied with all the requirements of this section of the code.

Par. CA-5. Revised:

CA-5. Cracks which are generally transcrystalline (across the grains) in character and may occur both internal and external to the riveted, welded, and expanded joints in any section of the boiler, are attributed to one or more of the following causes: Steel of unsuitable quality, excessive internal stresses, imperfect thermal or mechanical treatment during fabrication, unskilled or abusive treatment during repairs, or to corrosion and certain other severe operating conditions.

Cracks which are generally intercrystalline (around the grains) in character occur in boilers only in joints and seams where in certain boiler waters dissolved solids may become highly concentrated. These cracks are generally attributed to the effect of caustic soda on highly stressed metals within these parts.

This action, which is usually termed "caustic embrittlement," has taken place in cases where the boiler water contains but little sodium sulphate in proportion to the total sodium hydroxide and sodium carbonate alkalinity.

In cases of boiler failures from cracks, competent advice should be secured to determine the cause for such failures.

Operating evidence supplemented by laboratory work indicates that if not less than the following ratios of sodium sulphate to total sodium hydroxide and sodium carbonate alkalinity calculated to equivalent sodium carbonate are maintained in the boiler water, "caustic embrittlement" will be inhibited:

Working pressure of boiler, lb. gage	Sodium sulphate		Total sodium hydroxide and carbonate alkalinity calculated to equivalent sodium carbonate
0 to 150[a]	1	to	1
150 to 250	2	to	1
250 and over	3	to	1

[a] For cases where this ratio should be higher, see footnote No. 1 below.

Laboratory research work has indicated that there may be other means of inhibiting "caustic embrittlement;" however, there are insufficient operating data available at the present time to prove the value of these preventive measures.

Pending further operating data from boilers in service, it is recommended that the requirements of Par. I-44 of Section VI of the code be extended to all riveted, welded, or expanded joints, and that careful examination of all joints be made if leaks occur and do not remain tight after proper calking.

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[1] Bulletin No. 216 of the Engineering Experimental Station, University of Illinois, entitled "Embrittlement in Boilers," 1930, pp. 76-78.

Report of The American Uniform Boiler Law Society

By Charles E. Gorton*

During the fiscal period 1930-1931 code bills were presented to four state legislatures, looking to the adoption of the A.S.M.E. Boiler Code as their state standard. It was necessary in several instances to have bills amended after their introduction, in order to meet certain requirements and suggestions from members of the society. Owing to the unsettled condition of affairs during this period, we could not say what the outcome would be, but at the end of the sessions the bills in the main were still on the calendars, not having been acted upon.

So far as the society was concerned, the necessary work, including traveling, conferences and the preparing or suggesting the class of legislation, had been accomplished.

While the country has been and is passing through a period of general business depression, and the inclination has been to reduce traveling expenses, etc., wherever possible, we can assure you that this has not been the case with the society, for during the past two years we have been called upon to render greater service to our membership, which necessitated more traveling than during any period of the history of the society.

Some nine or ten bills were introduced in different legislatures, which members of the society felt were inimical to the best interests of industries. It has been necessary and will continue to be necessary, to keep in touch with the several capitols when this kind of legislation is introduced. We are compelled to feel at times that the country and industry should not be burdened with this class of suggested legislation.

We found that the tendency in different localities was to create new bureaus or offices which of necessity called for new or increased appropriations, which seemed to be contrary to the feeling of retrenchment on the part of the legislatures, and this feeling naturally had its effect upon code legislation.

The society has endeavored to keep in touch with the authorities having the administration of the code in hand, and has been of service to them when called upon for advice or suggestions, and we wish to say that in the main, uniformity of administration has been pretty well adhered to during the past, although we wish to call attention to certain tendencies which if carried very far would have their effect upon uniformity insofar as code rules are concerned.

We have noticed an inclination to incorporate in the rules, subject matter that departments feel would be of benefit to their local conditions. Care must be taken to see that this kind of matter is not contrary to or different from the code rules. Where there is any question on this point, it would be in order to refer matters of this kind to the Boiler Code Committee for their study and opinion and if favorably passed upon, they could be incorporated in the code rules, and be used by all code states.

Our society has had co-operation from the different sections and many of the members of the society during the past year, but we wish to emphasize that in unity of purpose there is strength, especially when it applies to the work the society is doing in the interests of all its membership by having the A.S.M.E. Boiler and Unfired Pressure Vessel Rules generally adopted, which in themselves wherever adopted, assure to the general public a reasonable degree of safety to life and property.

The society has gone to a considerable expense in keeping those of its membership informed who were or might be interested in the Unfired Pressure Vessel Rules and Paragraphs of the Boiler Code that were under revision, or where new matter was added, by sending out reprints of the suggested rules that appeared in *Mechanical Engineering* from time to time, requesting that constructive suggestions or criticisms of the suggested rules be sent to the Boiler Code Committee for their consideration.

It has been our purpose to attend, as far as possible, meetings of sister organizations, members of which go to make up our membership, in order that we may be in close touch with their problems that affect our work. We have acted and will continue to act as a clearing house on questions of vital interest to the membership—other than giving out any information that could be construed as an interpretation of any paragraph of the code. Matters of this kind should be sent direct to the secretary of the Boiler Code Committee, 29 West 39th Street, New York.

Very few of the code states have jurisdiction over unfired pressure vessels, but now that the Boiler Code Committee has issued a complete set of rules it will be a part of the society's work in future, to suggest legislation in order that these rules may become effective in the states that now use the Boiler Code and in future code states.

The A.S.M.E. Boiler Code Committee, at its June, 1931, meeting, took final action on the revisions of the Unfired Pressure Vessel Code (Section 8, of the Boiler Code), as well as on the Fusion Welding of Drums or Shells of Power Boilers. The rules as recommended by the Boiler Code Committee, were adopted by the council of the A.S.M.E. on July 7, 1931, and the Unfired Pressure Vessel Code and Boiler Revisions were issued in printed form during the month of August.

Since the issuance of the 1931 Unfired Pressure Vessel Code and Rules for the Fusion Welding of Drums or Shells of Power Boilers, we have had numerous inquiries as to what action, if any, the code states have taken in regard to the Unfired Pressure Vessel Code and especially the rules covering Fusion Welding of Drums and Shells of Power Boilers.

We have definite information from W. E. Smith, chief inspector of the Hawaiian Sugar Planters' Association, that the Boiler Code Committee of the association in the Territory of Hawaii, who use about 95 percent of the boiler horsepower in the islands, has approved the section on the Welding of Power Boilers, so that welded drums are acceptable in this district when

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* Chairman of The American Uniform Boiler Law Society, New York.

built according to the latest edition of the Power Section of the A.S.M.E. Code.

Rhode Island will accept boilers built in accordance with the latest Boiler Code Rules.

We understand that Oregon will accept boilers and unfired pressure vessels when built to code rules.

New York at the present time, will accept fusion welded drums when boilers are built to the latest rules of the boiler code, provided Section No. 334 of the New York State Industrial Code, Bulletin No. 14, is rigidly complied with. New York is now revising its rules to conform with the latest A.S.M.E. Boiler Code. The state has no jurisdiction over unfired pressure vessels at the present time, but steps are being taken to have the necessary legislation introduced if possible, during the next legislative session.

We hope to be in a position in the near future, to give the members more detailed information as to when code states will put into effect the latest Boiler Code Rules. We have always taken it for granted that a code state will accept the revisions and new matter added from time to time by the Boiler Code Committee, the only question being at what early date they will accept equipment built under the latest rules.

The state of Maine passed laws which require that all boilers coming into the state are to be figured by the A.S.M.E. formula for construction. This piece of legislation is only the beginning to get the people acquainted with the workings of the code. The next step will be to ask the legislature to adopt the A.S.M.E. Code as the state standard.

The Boiler Code Committee, some time ago, appointed a sub-committee of its members to meet with the Massachusetts Board of Boiler Rules. This committee held a meeting in Boston and many of the differences between the code and Massachusetts rules were ironed out and satisfactorily adjusted. It is our intent to suggest to the Boiler Code Committee that these two agencies be called together within a reasonable time, in order that they may take up the remaining differences of the two sets of rules and at the same time go over the Rules for Unfired Pressure Vessels.

We understand that at the present time the Massachusetts law applies only to air tanks used for the operation of pneumatic machinery. Recently garages and automobile repair places began using these to elevate automobiles, which brought them under the provisions of the law. They found that so many cases did not conform to the air tank regulations that they were forced to adopt regulations governing these tanks, less restricted than the old form. Massachusetts Form J (Air Tank Regulation) is the latest development and seems to be consistent with the Unfired Pressure Vessel Code in-so-far as it applies to such tanks.

The American Uniform Boiler Law Society was formed in 1916 to obtain the general adoption of the codes of the A.S.M.E. Boiler Code Committee, in order to secure uniformity in the rules and regulations of the various states and municipalities. The codes that have been formulated by the Boiler Code Committee now include a wide range of pressure containers other than boilers, and they may from time to time formulate still other rules. The term boiler is used in the same broad sense of including all pressure vessels as in the case of the Boiler Code Committee itself.

The society has made no attempt to influence the requirements of the code rules and regulations. It has always held that uniformity can best be secured and the interest of user and public best be protected by placing the formulation as well as interpretations of such requirements in the hands of the Boiler Code Committee.

Recent revisions of the Power Boiler Code to permit the more extended use of fusion welding, and the almost complete revision of the Unfired Pressure Vessel Code, are resulting in a large number of manufacturers not formerly identified with the society, taking an active interest in the adoption of these revised codes. This naturally will require increased activity which will be facilitated by extended participation of the council in the work and an enlarged executive committee to meet with the chairman at frequent intervals.

A new constitution and by-laws has been adopted which provides for a council consisting of thirteen elected members and the chairman. This was done in order to make provisions for the representatives of interests not formerly identified with the society. The society has closely co-operated with the National Board of Boiler and Pressure Vessel Inspectors, whose members are representatives of the several governmental divisions which have legally adopted the boiler code, and it was felt advisable to include that organization in the membership of the society. The following sections will have representatives on the council:

Power boiler manufacturers	2 members
Cast iron heating boiler manufacturers	1 member
Steel plate heating boiler manufacturers	1 member
Locomotive manufacturers	1 member
Unfired pressure vessel manufacturers	2 members
Material manufacturers	1 member
Accessory manufacturers	1 member
Welding equipment manufacturers	1 member
Insurance companies	1 member
Boiler and pressure vessel users	1 member
National Board of Boiler and Pressure Vessel Inspectors	1 member

It was found necessary to place the finances of the society upon a more stable basis by the establishment of fixed annual dues rather than contributions. The amount of the dues will bear some equitable relationship to the interest that each member has in the activities of the society.

Latent Energy in Locomotive Boilers

Not everyone realizes how much latent or stored-up energy there is in a modern steam locomotive of large size, carrying heavy boiler pressure. According to an interesting letter recently received from a correspondent, a heavy 2-10-4 type locomotive on his road, carrying 250-pounds pressure, would have 81.5 million foot-pounds of energy stored in the steam under normal operation, and 6960 million foot-pounds of energy in the water, or a grand total of slightly over 7 billion foot-pounds of kinetic energy to be liberated, in the event of boiler rupture. For purposes of comparison, a 10-inch projectile, weighing 400 pounds, requires only about 25 million foot-pounds of energy to give it a muzzle velocity of 2000 feet per second. In other words, when a locomotive boiler of the size mentioned explodes, the potential destructive effect would be equivalent to that of 280 of these 400-pound, 10-inch shells.

Records of the Bureau of Locomotive Inspection show that fully 90 percent of all fatal accidents caused by the failure of locomotive parts and appurtenances are due to boiler explosions as a result of crown sheets failing, and these failures are increasingly hazardous with the increased size of modern boilers and the higher steam pressures carried. In one case, according to Chief Inspector Pack's recent report, "The force of the explosion tore the boiler from the frame and hurled it forward 429 feet. The boiler alighted on the track and then slid forward for some distance, where the locomotive running gear and train collided with it, resulting in the derailment of the running gear, tender and 14 freight-train cars, 8 of which caught fire and were destroyed." In another instance, "The locomotive was moving backward at the time of the accident. The force of the explosion tore the boiler from the frame and hurled it forward, or in the opposite direction from which the locomotive was moving. The boiler first alighted on the back head, then bounded and alighted on the front end, after which it rolled over and came to rest 155 feet east and 35 feet north of the point of the explosion."

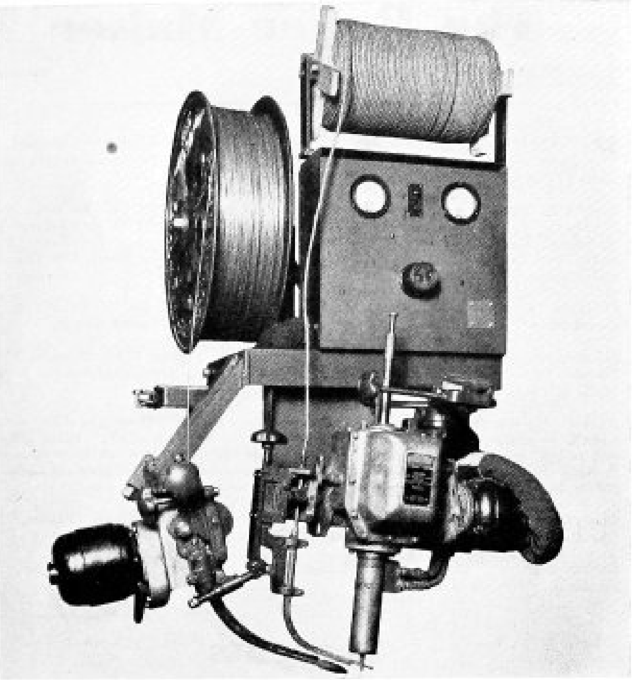
It is probably safe to say that no greater accomplishment in railroading has been recorded in the past two decades than the reduction of locomotive boiler explosions due to crown-sheet failures, from 92 in 1912 to 13 in 1931. In spite of this excellent performance, it must be admitted, however, that an examination of the records show two more crown-sheet failures in 1931 than in 1930, four more people killed and 27 more people injured. In fact, in recent months, boiler failures have been occurring all too frequently, and the necessity of redoubled vigilance in every phase of locomotive boiler construction, maintenance, equipment and operation is apparent if a continued reduction in the number of boiler failures is to be made.

While railroad travel is the safest known means of transportation and the railroads have achieved a notable success over a period of years in safeguarding boiler operation, there is still room and, in fact, an urgent necessity for further improvement. Probably man-failures in this connection can never be entirely eliminated, but the question may well be asked if everything possible has been done to minimize their effect and if all practicable mechanical safeguards have been applied to forestall failures of the human element or at least limit the attendant damage. The results of boiler explosions, largely due to crown-sheet failures, in loss of life and personal injuries, with accompanying damage claims, damage to equipment and lading, and particularly loss of prestige and public confidence, are difficult to evaluate. It may safely be said, however, that the actual out-of-pocket expense of a single boiler explosion may well reach several hundred thousand dollars. It is equally essential to provide full mechanical precautions and safeguards against failure of the human element and to exercise the greatest care in checking boiler maintenance and operating conditions if the railroads are to safely employ the tremendous amount of energy stored in locomotive boilers.—*Railway Mechanical Engineer.*

Automatic Welding Head

A wire feeding head which allows automatic arc welding with the shielded arc process has been developed by the Lincoln Electric Company of Cleveland, Ohio. The result claimed for this head is very high speed operation on either butt, fillet or building up welding, the deposited metal having physical characteristics equal to or better than mild steel.

The shielded arc technique is secured through the use of the electronic tornado welding head and a fibrous autogenizer which is fed into the arc flame. This autogenizer burns and forms a gas excluding the oxygen and nitrogen of the atmosphere while the metal is molten. The welding head utilizes a carbon arc around which is superimposed a magnetic field which directs the arc stream on the line of fusion.



Wire-feeding automatic arc welding head

Continuous filler metal is obtained from a reel mounted on the head and fed into the arc just in front of the arc travel. Means are provided for varying the rate at which the filler metal is fed into the arc so that the amount of metal deposited can be varied to meet the conditions demanded by the speed of the head and the type of weld being made. Since the rate at which the filler can be fed is varied independent of the speed of the head travel, only one size of filler metal is necessary for any type of work within the range of the machine. The filler metal is usually in strip form which insures accuracy in feeding the metal in the arc.

The only factors determining the most desirable size and form of the filler metal are ease of handling, compactness of the coils and minimum disturbance of the arc. In view of these factors filter strip $\frac{3}{16}$ inch wide and $\frac{1}{16}$ inch thick has been adopted as standard and is furnished in 70-pound coils 12 inches inside diameter and about 18 inches outside diameter, and 3¼ inches traverse. These coils are securely tied and treated to prevent corrosion during storage. When being used the coil of filler is mounted on a steel reel with detachable head.

The filler metal carries no current and its size is independent of the current used for welding. One size of strip can be used for any current from 75 to 500 amperes. As the filler does not carry current, spatter and waste are largely eliminated. The filler strip does not pass through the arc as in metallic welding, but beneath it, and is therefore not subject to the extremely high temperature of the arc stream. It makes possible a practical method of shielded-arc welding where it is desirable or necessary to add filler metal.

Since the welds are made in a shielded arc, the physical characteristics are similar to those obtained by other shielded arc methods. It is claimed a tensile strength ranging from 65,000 to 80,000 pounds per square inch is obtained, with a ductility of 20 percent or more elongation in 2 inches.

The Boiler Maker

Volume XXXII Number 4

PUBLISHED Monthly by the Simmons-Boardman Publishing Company, 34 North Crystal Street, East Stroudsburg, Pa., with Executive Offices at 30 Church Street, New York City.

Chicago: 105 W. Adams Street. Washington, D. C.: 17th and H Streets, N. W. Cleveland: Terminal Tower. San Francisco, Cal.: 215 Market Street.

Subscription price in United States and possessions, 1 year, $2.00, 2 years, $3.00; Canada, including duty, 1 year, $2.50, 2 years, $4.00; foreign countries, 1 year, $3.00, 2 years, $5.00. Single copies, 25 cents each.

Request for change of address should reach us on or before the 15th of the month, preceding the issue with which it is to go into effect. It is difficult, and often impossible, to supply back numbers to replace those undelivered through failure to send advance notice. In sending us change of address, please be sure to send us your old address as well as the new one.

The Boiler Maker is a member of the Associated Business Papers, Inc. (A. B. P.), and the Audit Bureau of Circulations, (A. B. C.)

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Contents

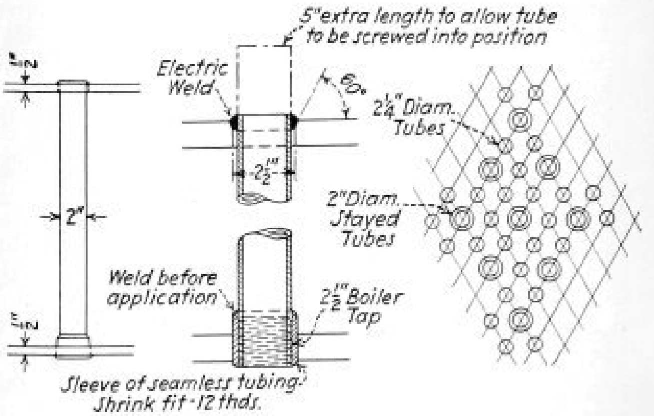
Communications

Applying Stay Tubes

To the Editor:

The sketch shows a simple and effective method of applying stay tubes in a large vertical or upright boiler. In a case where such a method was applied on a 60-inch diameter boiler we were unable to keep the machine on the road more than a week at a time because of badly leaking flues prior to the application of these stay tubes. Since applying the stay tubes, however, the trouble has been entirely eliminated. The machine in question employed a crew of seven men and it meant a serious loss when it was tied up.

Our experiment has been in use now more than a year and is more than successful. Alterations which included the addition of nine stay tubes in the firebox required that nine of the original 2¼-inch diameter tubes be removed. The holes were then countersunk and nine 2-inch diameter stay tubes were inserted through the bottom, screwed into position, welded at the top and rolled and beaded at the bottom. A sleeve of seamless tubing was shrunk and welded to the lower end of the stay tubes and threaded twelve threads to the inch. No special tubes or special tools were required.



Method of applying stay tubes to a vertical boiler

In application 5 inches of extra length was provided to allow the tube to be screwed into position. When in position the tube was welded at the top and the extra length of tubing was cut off. In this case the firebox was 72 inches in diameter, had two hundred and sixty-nine 2¼-inch diameter tubes and nine 2-inch diameter stay tubes.

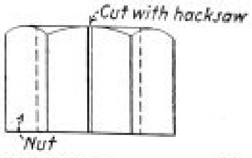
Carnegie, Pa. A. P. McMillan.

Slotted Nuts for Marine Use

To the Editor:

Any engineer or boiler maker who has gone inside a boiler to remove zinc plates knows the difficulty of starting nuts which are salted and scaled to the bolt. Often such difficulty causes the men to become impatient and use a hammer on the nuts, resulting in breaking the bolt. This bolt being tapped into the furnace causes many delays, as it is often necessary for a boiler maker to repair the damage and requires another test on the boiler.

Here is a method of avoiding this trouble: Take off all of the zinc nuts, place in a vise one at a time and with a hacksaw cut the nut through one side parallel with the hole, as shown in the sketch. This allows the nut to open up when taking it off or putting it on and saves breaking studs since no hammer has to be used.



Slotted nut for easy removal from bolt

It is also a good idea to cut all the baffle nuts inside the furnace in this manner.

Montreal, Canada. James Wilson.

Questions and Answers Pertaining to Boilers

This department is maintained for the purpose of helping those who desire assistance on practical boiler shop problems. Inquiries should bear the name and address of the writer. Anonymous communications will not be considered. The identity of the writer, however, will not be disclosed unless the editor is given special permission to do so.

By George M. Davies

Stay Tap Breakage

Q.—I was interested in the article on stay taps in the December, 1931, issue, and I should like to ask your expert advice and opinion on causes of stay tap breakages. This seems a tall order, but any information concerning proper usage of stay taps, both ground and carbon, primary causes of breakage, things to guard against, etc., would be appreciated. I have had the experience of having several taps break when using pneumatic power (both ground and carbon). The taps were in good condition; in fact, several were new; the taps had run through both steel and copper, and when the taps had protruded through about 10 inches this end (front end) fell off. The taps were running easily at a normal speed and had not jammed or caught up in any way. On examination, there was no appearance of flaw. I am unable to offer any explanation of the cause of the breakages. The ground taps were of high-class manufacture, 5-fluted (straight), and the carbon taps were 4-fluted and 5-fluted.—C. H. W.

A.—There are numerous causes for the breakage of staybolt caps, the principle of these being:

(1)—The taps are too hard, due to improper tempering of the steel.

(2)—The taps are run at too high a speed.

The Master Boiler Makers' Association discusses this subject at the 1927 convention and the following extracts from the discussion may be of some assistance to you.

"The question arises as to the proper kind of taps. Practically all tap manufacturers have their own idea as to the size, amount of taper, and amount of full treads for the taps of their own make. Also, much has been written as to the relative merits of a straight or fluted tap. On a great many stock taps, both straight and spiral fluted, the maximum reamer diameter is so much smaller than the root diameter of the sizing treads, that the threaded section of the tap must do a certain amount of reaming, and, as a result, the cuts are very fine, and, being very light, do not clear themselves, but wad up in the flutes, or gum up between the teeth. The heat generated when this occurs is sufficient to draw the temper of the teeth causing the tap to tear instead of cut.

"The most nearly perfect tap will have a maximum reamer diameter of not more than .025 inch below the root diameter of the sizing threads, the root of the sizing threads to have the same taper as the reaming portion of the tap.

"In re-tapping a hole 1/16 inch large, the teeth, on the first few revolutions of the taper section have the same form as the old thread and have a tendency to act as a lead screw, so that the new thread will follow the old. The remainder of the taper section will cut out part of the old thread, and form a means of steadying the tap in the hole before the new root diameter is reached, and prevent oversize reaming.

"A great many railroads have their own specifications for staybolt taps, others use what is known as the standard stock tap, supplied by the different tap manufacturers.

"In selecting taps, the proper procedure is to purchase taps from a number of tap manufacturers and test each make of tap and from the test made, the tap which gives the best service should be adopted as standard.

"In making a tap test, laboratory methods should be followed as nearly as possible. Tapping variables, such as boiler plate hardness and thickness, distance between sheets, size of drilled holes, air pressure, tapping speed, and kind of lubricant should be strictly adhered to. As far as practical all tests should be made under actual working conditions; that is, a locomotive or boiler that is undergoing repairs should be used for the tests. An old sheet, from which staybolts have been removed, will give a more accurate test than two new sheets. Staybolt taps, while on test, should be carefully checked with a ring gage, as to size, and the tapped holes checked with a plug gage, and should be sharpened whenever necessary, and used until allowable under size is reached.

"Staybolt taps of standard manufacture, unless made to a railroad company's specification, have an allowable oversize of approximately three one-thousandths inch.

"All new staybolt taps, when purchased, should be gaged, and taps of the same size should be marked and used together, so as to have all tapped holes approximately the same size.

"The question of lubricant used for staybolt tapping and threading rests entirely with the boiler shop foreman. A lubricant should be selected that will give the longest tapping service, cut the cleanest threads, and that will not harden if allowed to stand for some time before the bolts are applied. With certain makes of lubricant, the cuttings will harden on the foundation ring if allowed to stand for any length of time.

"The question arises 'When tapping foi crown bolts, should a tap be used with continuous threads the full length of the water space, or a short tap with long threads?' If the crown bolts are threaded on a turret lathe, threading both ends at the same time, it is better to use a continuous thread tap. If each end of the bolt is threaded separately, it will make no difference if the short tap is used, as the threads on one end may not line up in respect to those on the other.

"At a recent test of staybolt taps, and motors, it was proven conclusively that a reversible motor, with speed governor having a speed of approximately 300 revolutions per minute, was the most satisfactory, and that a spiral fluted tap was superior to a straight flute; a spiral fluted tap with dimensions as mentioned elsewhere in this report proves to be the nearest perfect tap obtainable."

Locomotive Draft Regulation

Q.—Describe briefly the draft regulation device of the smokebox of a locomotive. G. M.

A.—The diaphragm is used to deflect the gases and cinders downward, and also to equalize the draft through the flues. The deflector plate is equipped with an adjustable damper by which the travel of the gases can be regulated so as to make the fire burn evenly on the entire grate surface.

If the deflector plate was omitted, the exhaust of the cylinder through the exhaust pipe into the stack would cause the upper flues to have the greatest draft. If the fire near the back end of the firebox indicates that the draft through the upper tubes is too great, the damper of the deflector plate should be lowered to choke the draft through the upper flues and equalize the draft through all the flues. If the draft through the lower tubes is too great, the damper should be raised sufficiently to lessen the draft through them, and thus equalize the draft and make the fire burn evenly. If the fire does not burn freely and the inside of the fire door becomes covered with soot, it indicates a choked draft. This is due to a number of tubes being stopped up, or to the deflector plate being too low or two close to the flue sheet.

In addition to the deflector plate, there is a pipe called a petticoat pipe, its duty is the same as that of the deflector in regulating the draft. Its movement is the opposite of the deflector plate. The damper of the deflector plate, as well as the petticoat pipe, is set by trial. In addition to allowing the hot gases to give up their heat to the water, they must draw the products of combustion from all parts of the firebox evenly. This in turn makes the draft of fresh air uniform through all parts of the grates. If many of the flues are stopped up, not only is the heating surface of those flues lost, but the fire will not burn evenly over all parts of the grates. For these reasons, the flues should be unobstructed, so that the hot gases can pass through them.

Efficiency of Circular Patches

Q.—Will you please give me the correct formula for calculating the efficiency of the patch shown in Fig. 1?—F. J.

A.—The efficiency of the pitch as shown in Fig. 1 may be computed as follows:

$$(1)\ \text{Efficiency} = \frac{S}{A}$$

where, S = net plate measured on an arc subtended by a cord A, Fig. 2.
A = a chord equal in length to the radius of the rivet circle, Fig. 2, for circles of 12 inches or under.

$$(2)\ \text{Efficiency} = \frac{\text{net section through } A\text{-}A}{\text{solid plate through } A\text{-}A}$$

Net section through A-A is taken as the combined net section of the shell and patch.

Solid plate through A-A is taken for the shell only.

$$(3)\ \text{Efficiency} = \frac{N \times a \times 44{,}000 + TS \times \text{net sectional area of shell through } A\text{-}A}{TS \times \text{solid shell plate through } A\text{-}A}$$

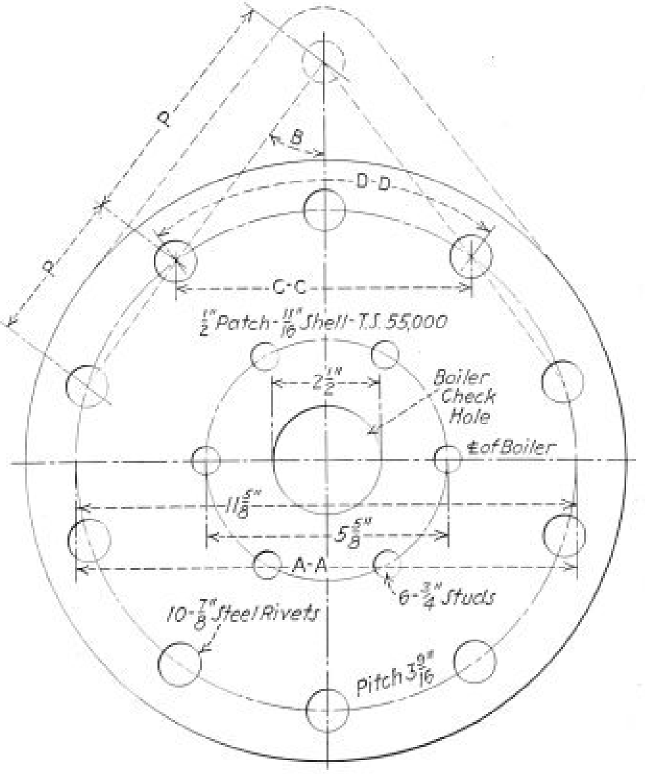


Fig. 1.—Circular patch for boiler

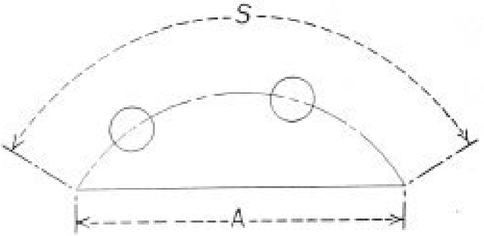


Fig. 2.—Section of patch for computation of efficiency

where, N = number of rivets in single shear (five)
TS = tensile strength of plate, 55,000 pounds per square inch.
a = cross-sectional area of rivet after driving, square inches.

Solid shell plate through A-A, Fig. 1 = 11⅝ inches × 11/16 inches.

The least efficiency obtained in (1), (2), or (3) should be taken as the efficiency of the patch.

Designing the patch as indicated by the dotted lines would effect the efficiency as follows:

$$(1)\ \text{Efficiency} = \frac{2(p - d)}{p\sqrt{3 \times \text{sine } B^2 + 1}}$$

where p = shortest pitch of rivets, in inches
d = diameter of rivet holes, in inches
B = angle in degrees, see Fig. 1.

(2) Same as for round patch.

(3) Same as for round patch, except N, number of rivets in single shear, would be six.

$$(4)\ \text{Efficiency} = \frac{1 \times a \times 44{,}000 + TS \times \text{net sectional area of shell through } D\text{-}D}{TS \times \text{solid shell plate through } C\text{-}C}$$

The least efficiency obtained in (1), (2), (3) or (4) should be taken as the efficiency of the patch.

In designing circular patches, the thickness of the patch should be made equal to the thickness of the shell.

Round or diamond patches should have no rivets on the longitudinal center line and the circumferential pitch should be spaced so that the patch can be well calked.

Oil in Boilers

Q.—On inspection of boilers how and where would you note the presence of oil in a Scotch Boiler? In a watertube boiler?—J.L.D.

A.—Many engineers are under the impression that, when examining a boiler for grease or oil, the only place grease can be found is around the water level, tops of tubes and combustion chambers in Scotch boilers, or the steam drum in watertube boilers. This is based on the idea that all grease will come to the surface of the water.

This is not always true, for under certain conditions oil may be precipitated and will adhere to the tubes and plates below the water level. Therefore, when inspecting a boiler, the entire water side should be carefully examined for signs of grease. Where scale deposits exist, such deposits should also be examined for signs of oil. Heating surfaces in direct contact with the fire should be very carefully examined. When grease is found or its presence is suspected, the boiler should be given a good boiling out. Tube failures in watertube boilers and the collapse of furnaces and combustion chambers tops in Scotch boilers can in many instances be attributed to the presence of oil.

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Assistant Chief Inspectors—J. M. Hall, Washington, D. C.; J. A. Shirley, Washington, D. C.

Steamboat Inspection Service of the Department of Commerce

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Secretary—C. W. Obert, 29 W. 39th Street, New York.

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States

Arkansas
California
Delaware
Indiana
Maryland
Michigan
Minnesota
Missouri
New Jersey
New York
Ohio
Oklahoma
Oregon
Pennsylvania
Rhode Island
Utah
Washington
Wisconsin
District of Columbia
Panama Canal Zone
Territory of Hawaii

Cities

Chicago, Ill.
Detroit, Mich.
Erie, Pa.
Evanston, Ill.
Houston, Tex.
Kansas City, Mo.
Los Angeles, Cal.
St. Joseph, Mo.
St. Louis, Mo.
Scranton, Pa.
Seattle, Wash.
Memphis, Tenn.
Nashville, Tenn.
Omaha, Neb.
Parkersburg, W. Va.
Philadelphia, Pa.
Tampa, Fla.

States and Cities Accepting Stamp of the National Board of Boiler and Pressure Vessel Inspectors

States

Arkansas
California
Delaware
Indiana
Maryland
Minnesota
Missouri
New Jersey
New York
Ohio
Oklahoma
Oregon
Pennsylvania
Rhode Island
Utah
Washington
Wisconsin
Michigan

Cities

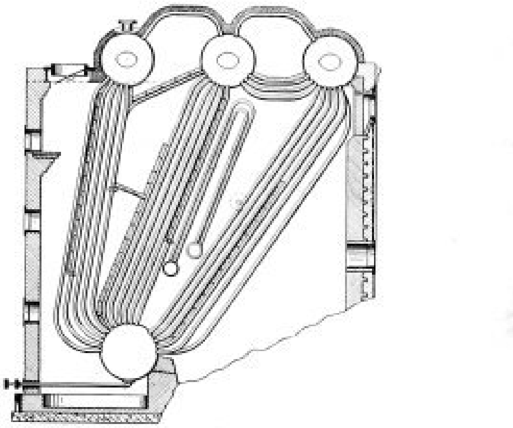
Chicago, Ill.
Detroit, Mich.
Erie, Pa.
Kansas City, Mo.
Memphis, Tenn.
Nashville, Tenn.
Omaha, Neb.
Philadelphia, Pa.
Parkersburg, W.Va.
St. Louis, Mo.
Scranton, Pa.
Seattle, Wash.
Tampa, Fla.

Selected Boiler Patents

Compiled by Dwight B. Galt, Patent Attorney, Washington Loan and Trust Building, Washington, D. C. Readers wishing copies of patents or any further information regarding any patent described, should correspond directly with Mr. Galt.

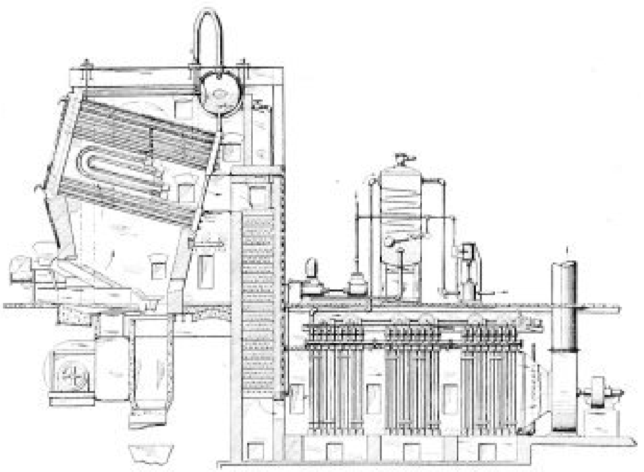
1,790,710. SLAG-SCREEN WATERTUBES. DAVID S. JACOBUS, OF MONTCLAIR, AND HOWARD J. KERR, OF WESTFIELD, N. J., ASSIGNORS TO THE BABCOCK & WILCOX COMPANY, OF BAYONNE, N. J., A CORPORATION OF NEW JERSEY.

Claim.—In combination, a watertube boiler having an upper transverse steam and water drum connected to a lower mud drum by a bank of tubes arranged in rows, the tubes in the first and second rows of tubes of said bank being spaced farther apart from each other than the tubes in the rear rows and entering each of said drums along a single row of holes. Twelve claims.



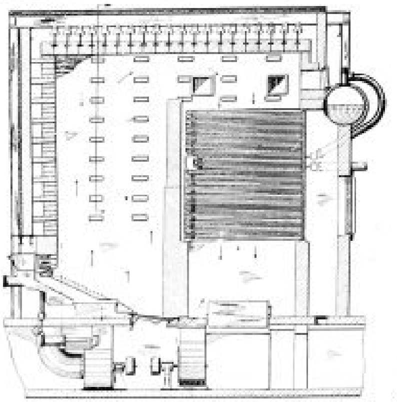
1,801,275. STEAM BOILER. DAVID S. JACOBUS, OF JERSEY CITY, N. J., ASSIGNOR TO THE BABCOCK & WILCOX COMPANY, OF BAYONNE, N. J., A CORPORATION OF NEW JERSEY.

Claim.—A watertube boiler comprising a steam and water drum and having uptake and downtake water compartments connected by two banks of horizontally inclined watertubes, the banks being spaced apart to provide a chamber between them, a furnace beneath the boiler, a superheater having the major portion of its tube-heating surface within that portion of the space between the watertubes adjacent the uptake water compartment, said superheater having tubes extending longitudinally of the watertubes and boxes or headers extending within the space between the watertubes transversely of the watertubes and located intermediate of the length of the watertubes and baffling comprising a longitudinal baffle above the lowermost tubes and extending from the low ends of the tubes to a point adjacent the superheater headers and a transverse baffle extending upwardly from the inner portion of the longitudinal baffle across the tubes of the upper bank and contiguous to the superheater headers and defining an upwardly decreasing flow area for the gases across the tubes of said upper bank, said superheater headers forming a part of said transverse baffle, and the baffle having portions located in the space between the banks of watertubes and above the longitudinal baffle arranged to protect a superheater header above the lowermost header from direct impingement of hot gases passing upwardly in the first pass while said headers remain a portion of the heating surface. Four claims.



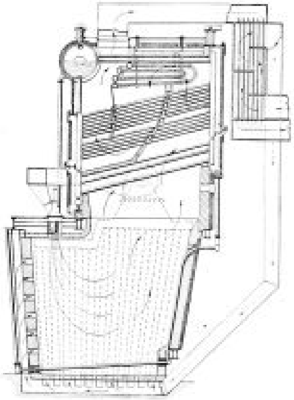
1,751,551. BOILER AND SETTING THEREFOR. DAVID S. JACOBUS, OF JERSEY CITY, N. J., ASSIGNOR BY MESNE ASSIGNMENTS, TO FULLER LEHIGH COMPANY, A CORPORATION OF DELAWARE.

Claim.—An inclined furnace wall comprising an inner wall and an outer wall, both of said walls being substantially flat, refractory members carried by one of the walls and resting against the other wall and slidable over the surface thereof for maintaining said walls in spaced relation while allowing them to expand relatively to each other in a direction substantially parallel to the planes of the walls. Seven claims.



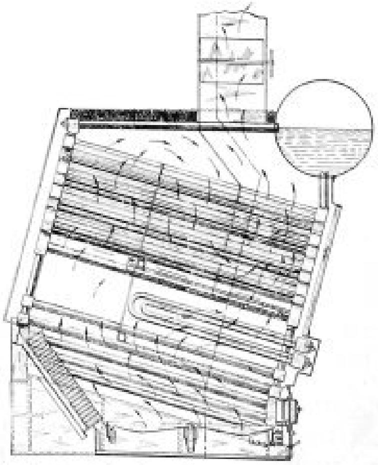
1,794,774. AIR-COOLED FURNACE. DAVID S. JACOBUS, OF JERSEY CITY, N. J., ASSIGNOR BY MESNE ASSIGNMENTS TO FULLER LEHIGH COMPANY, A CORPORATION OF DELAWARE.

Claim.—A furnace chamber having air passages in the walls thereof, an air heater through which the furnace gases pass, a fuel burner to supply fuel to said chamber, ducts to lead heated air from said air heater to said passages and from thence to said burner, and a fan in the duct between said passages and said burner. Eleven claims.



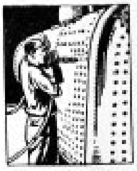
1,801,251. SUPERHEATER BOILER. PAUL A. BANCEL, OF NUTLEY, N. J., ASSIGNOR TO THE BABCOCK & WILCOX COMPANY, OF BAYONNE, N. J., A CORPORATION OF NEW JERSEY.

Claim.—In combination, a watertube boiler having horizontally inclined watertubes divided into banks, a superheater chamber between the banks, a superheater in said chamber, a main gas outlet and cross baffles forming passes to direct gases back and forth across the watertubes to the main gas outlet, said chamber having an inlet and an outlet arranged to permit some of the gases to flow through the chamber, and a flue disposed in one of said passes and connecting said chamber outlet to the main gas outlet. Sixteen claims.



The Boiler Maker

Reg. U. S. Pat. Off.



Boiler Manufacturers' Convention

The forty-fourth annual convention of the American Boiler Manufacturers' Association will be held at the Greenbrier Hotel, White Sulphur Springs, W. Va., May 23 to 25. While the formal program of the meeting has not as yet been issued, the general sessions of the convention will be devoted to matters of special interest to both active and associate members, with reports of standing committees, a review of the condition of the industry and the like. During the course of the three days' session, trade problems will be discussed.

The meeting is of particular importance this year as it will be held in conjunction with conventions of the Water Tube Boiler Association, the Stoker Manufacturers' Association and the Pulverized Fuel Association. The Steel Heating Boiler Institute will also be represented.

Since this is the first time that a joint meeting of what constitute the four important phases of the steam power supply field, the program arranged should develop some extremely interesting problems common to all. For this reason, the president of the American Boiler Manufacturers' Association and its secretary have been particularly desirous that the association be represented by at least one official of each of the member companies. Hotel accommodations and transportation facilities have been arranged so that the financial burden of the convention will be extremely moderate and should induce a large attendance.

Planning Ahead

Changing conditions of service and types of motive power have created new locomotive maintenance problems. Mechanical officials of a number of railroads have taken occasion, during the present period of low demand for power, to develop new rules and to revise old ones for the care of locomotives.

A case in point is the work in this direction by the superintendent of motive power of the Chesapeake & Ohio Railway Company, the boiler operating regulations of which appear in this issue. Foresight, not only in the modernizing of rules of this character dealing with day-to-day operation of locomotives but also in rearranging and planning in locomotive shops, will be of the utmost value when the railroad plant of this country is again in motion.

Applying this thought to the average locomotive shop, there is much to be accomplished. For the period from 1920 to 1930, operations followed in almost unchanging routine except in a few isolated cases. The Paducah, Kentucky, shops of the Illinois Central and the Huntington shops of the Chesapeake & Ohio are excellent examples of modern well-planned shops built during this period, the capabilities of which were designed to meet almost any production demand placed on them.

If the best talent, both engineering and practical, available now on the mechanical staff of every railroad were applied to a definite modernization of facilities program to meet its individual maintenance problem, not many months from now such foresight would bring returns in increased efficiency and lowered costs.

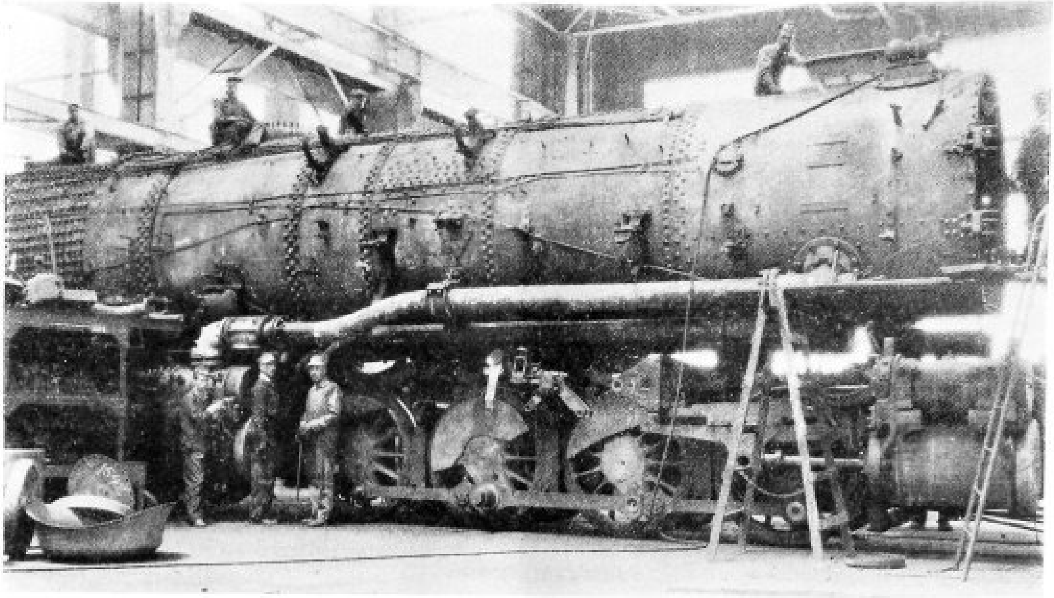
Constructive forward-looking measures planned now by both the operating department and the mechanical are essential to the future welfare of the railway industry. From the mechanical viewpoint, it is logical to provide for the rehabilitation of shops and equipment during times of low production requirements. The fact, that the present stagnation of the railroads' business is not an ordinary dull period but a serious crisis in their history, does not change this principle but only emphasizes it. The railroads still constitute the framework on which the industry of the United States is built, and any recovery of the latter will be reflected in a demand on transportation facilities. It is an opportune time for the railroad establishment to prepare for the future.

Weld Tests

With the increasing acceptability of welding as a process of pressure-vessel construction, the necessity for making weld tests becomes more important. Such tests are not only essential in determining the value of a new material, a new process, or of a new design, but should be employed to insure proper qualification of welders and to insure a satisfactory product.

In the case of development work and welder qualification, destructive tests should be used; but where tests of the finished product are required, it is far more desirable to use non-destructive tests than to examine coupons cut from the work. One of the simplest of the non-destructive tests is the visual examination. If this test can be used at the time the weld is being made, it has many advantages, no apparatus except a pair of dark glasses being required. With sufficient experience, a good estimate of proper penetration may be obtained.

(Continued on page 109)



Chesapeake and Ohio Rules for the Care of Locomotive Boilers

Through the courtesy of the superintendent of motive power of the Chesapeake and Ohio Railway Company, the requirements developed by the railroad for the care of locomotive boilers have been released for publication. The instructions cover the cooling of boilers, washing, inspection, filling and the like and represent what is considered the best practice developed to meet the conditions experienced in the territory in which this road operates. Supplementing the principal instructions, additional rules for blowing down are given as well as for testing.

The following instructions have been developed by the Chesapeake and Ohio Railway Company, covering the care of locomotives at terminals and on the road, with special attention to boiler requirements.

All boiler washing is to be performed according to the following rules. No changes allowed from these rules, except by written instructions from the proper authority.

All locomotives in service must have boilers washed at least once every thirty days, or more frequently if conditions require.

Boilers should be thoroughly cooled before being washed, excepting at points where improved hot-water washout systems are installed. When boilers are cooled in the natural way without the use of water the steam should be blown off, but the water must be retained above the top of the crown sheet and the boiler allowed to stand, until the temperature of the steel in the firebox is reduced to about 90 degrees F., or so that it feels warm to the hand. Then the water should be drawn off and the boiler washed. When the engine cannot be spared from service sufficiently long for it to be cooled in this manner before washing, proceed as follows:

When there is sufficient steam pressure, start the injector and fill the boiler with water until the steam pressure will no longer work the injector. Then connect the water-pressure hose to the feed pipe between the engine and tender and fill the boiler full, allowing the remaining steam pressure to blow through the syphon cock or some other outlet at the top of the boiler. Open the blow-off cock and allow the water to escape, but not faster than it is forced in through the check, so as to keep the boiler completely filled until the temperature of the steel in the firebox is reduced to about 90 degrees F., then remove all plugs and allow the boiler to empty itself.

When an engine is taken to the roundhouse with full steam pressure, the injector should be opened as outlined above and allowed to remain open until the steam pressure is reduced to 80 pounds; then the blowoff cock can be opened and the boiler emptied; then proceed to wash out with water at about 125 degrees F. temperature.

After washing has been completed, boiler should be filled with water at about 150 degrees temperature.

Before washing the boiler remove all washout plugs. Begin washing through the holes on the side of the boiler, opposite the front end of the crown sheet. Wash the top of the crown sheet at the front end. Then wash between the rows of crown bars and bolts at right angles to the nozzle, directing the stream toward the back end of the crown sheet. After washing through the holes near the front end of the crown sheet, use the holes in their respective order toward the back of the crown sheet. The object of this method is to work the mud and scale from the crown sheet toward the side and back legs of the boiler and prevent it from depositing on the back ends of the flues.

Next wash the crown sheet from the boiler head, revolving the nozzle so as to wash the top of the boiler and all radial stays or bolts as well as the crown sheet. All scale must be removed from the crown sheet; if necessary enter the boiler or thump crown sheet with large size shipping hammer.

Wash the back end of the flues through the holes in the connection sheet, revolving the nozzle so as to wash all foreign substance from the back end of the flues and flue sheet.

Wash the water space between the back head and the firebox door sheets through the holes in the back head, being careful to remove all scale and mud above and below fire door hole.

Arch tubes must be washed and scraped clean with scrapers or pneumatic cleaners, each time the boiler is washed. If scale is allowed to form in arch tubes, the metal becomes overheated and bulges are formed, and, if allowed to remain, the tube warps out of line with the holes, strains are set up and cracks develop, and the tube is very dangerous and liable to pull out or burst. Therefore, a locomotive should not be allowed to leave a terminal with dirty arch tubes, and all concerned are instructed to comply strictly with the rule.

The condition of an arch tube as to the scale on the water side can be readily determined by the presence of clinkers adhering to the fire side. If an arch tube is clean on the water side it will be clean and smooth on the fire side. The condition of the firebox sheets can usually be determined by similar evidence. It may be laid down as a general rule that clean fireboxes on the water side are clean and smooth on the fire side. Any clinker adhering or sand paper roughness on the fire side indicates scale formation opposite.

After arch tubes are washed and scraped, return to the holes on the side of the boiler opposite the crown sheet, using nozzles and revolve them so as thoroughly to wash down the side sheets and staybolts, making sure that all spaces on the side of the firebox are clear of mud and scale. Then wash through holes near check valves near front end of boiler.

After the above instructions have been complied with, wash the barrel of the boiler towards the throat sheet. Then use a straight nozzle directly against the flues, reaching as great space as possible in all directions. Then use the bent nozzle through the front hole in the bottom of the barrel, and also the straight nozzle in the same manner as above, until certain that the flues and spaces between the flues and barrel are as clean as it is possible to make them.

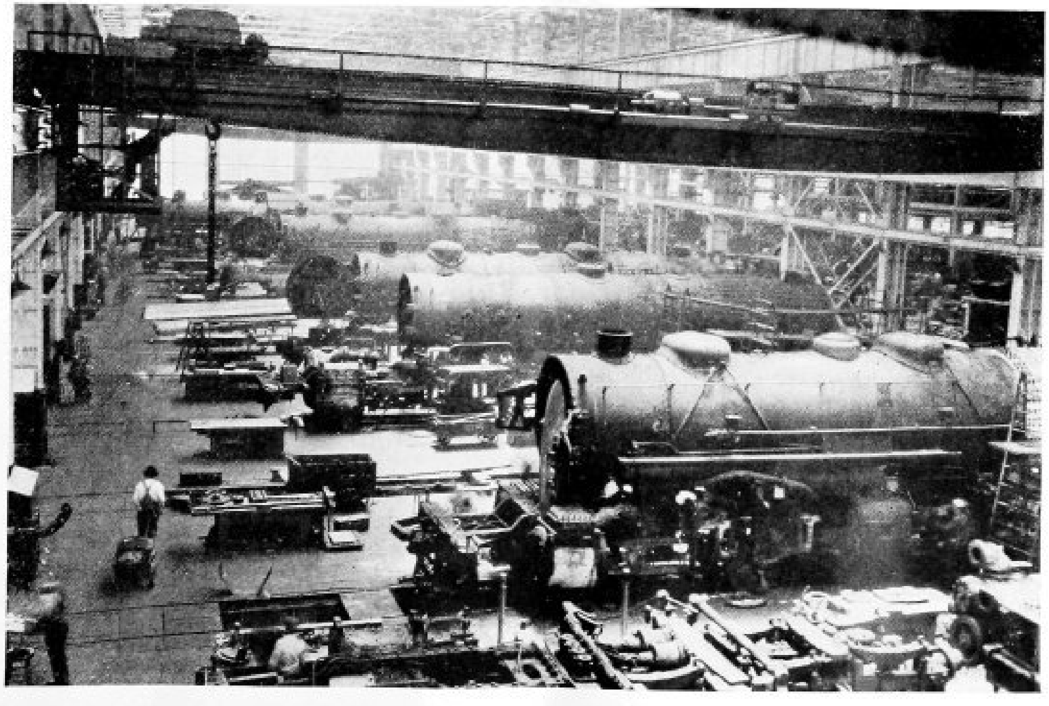
Next use the nozzles in the side and corner holes of the waterlegs, revolving them thoroughly to clean the side sheets, and finally clean off all scale and mud from the mud rings by means of straight nozzles in the corner holes.

It must not be assumed that because clear water runs through the holes that the boiler is cleaned; but all spaces must be examined carefully with rod and light, and, if necessary, use a pick, steel scraper or other tools, to remove accumulated scale.

When filling boilers of all engines, except those equipped with type-E superheaters, they must be filled through the blow-off cock; and for engines equipped with type-E superheaters, they must be filled through the branch pipe and the throttle.

At all points not equipped with the direct steaming method, the boilers must be filled with water of approximately the same temperature as the boiler plates.

Where the direct steaming method is used, the boiler



must be filled with water of approximately the same temperature as the boiler plates until it reaches the bottom row of flues; and on engines equipped with syphons, the water should be allowed to go into the boiler until it has reached a point just above the syphon necks. After the water reaches these levels, the direct steaming operation may be started.

On engines equipped with the circulating system, the water should be started into the boiler at approximately the same temperature as the boiler plates and allowed to run at this temperature until the water covers the circulating nozzle. When the water covers the circulating nozzle, steam may be turned on through the nozzle. Allow the water entering the boiler to be turned on until it reaches the top of the syphon neck; then the direct steaming system may be used.

It is important that the instructions regarding the temperature of water, as indicated above, be followed as nearly as possible so as to avoid any unusual stresses and consequent damage to boilers.

Supplementing the foregoing the following rules govern the blowing down of locomotives at terminals and on the road:

All locomotives having blow-off cocks in the back mud-ring corner with muffler attachment, are included in this shop practice and should be handled accordingly.

When an engine arrives at a terminal the fire should be put in good condition, the boiler filled with water to the top of the water glass and the injector shut off. The blow-off cock in the back mud-ring corner should be opened wide and allowed to remain open until a full glass of water has been discharged from the boiler. If the engine is equipped with blow-off cocks in both back mud-ring corners, one-half a glass should be blown from one which should then be closed and the other half blown from the other. Care should be taken with regard to using the blower with the fire in bad condition.

When leaving the terminal, before the engine is dispatched from the ready track, the fire should be put in good condition, the boiler again filled with water to the top of the glass, the injector shut off, and the blow-off cock opened wide until a full glass has been discharged from the boiler. If the boiler is equipped with blow-off cocks in both back mud-ring corners, one-half a glass should be blown from each side. Care should be taken with regard to using the blower with the fire in bad condition. This terminal blowing must be performed by the shop forces.

On line of the road, the engineer will blow the boiler every nine or ten miles by opening the blow-off cock wide three times, holding it wide open for a few seconds each time.

In addition to this, after the engine has been at rest for a period of time such as taking siding or making a water-stop, the blow-off cock should be opened wide for a period of not less than 10 seconds with the injector shut off, just before the leaving point of stop.

In case of foaming trouble interfering with the operation of the locomotive, the engineer will fill the boiler with water to the top of the water glass, shut off the injector, and open the blow-off cock wide until entire glass of water is discharged. If trouble is especially bad, it may be necessary to repeat this operation in order to remove sufficient dirty water from the boiler and replace it with fresh water from the tank to insure good operation.

Engines in switching and shifter service will be blown by the engineer approximately each hour at a convenient time, by opening the blow-off cock wide three times and holding it open for a few seconds each time.

In case of foaming interfering with operation of the engine, the boiler should be filled with water to the top of the glass, the injector shut off, and the blow-off cock opened wide until a full glass has been discharged, in order to replace sufficient dirty water in the boiler with fresh water from the tank to relieve the trouble.

In general the following instructions should be observed:

Do not fill the boiler or use injector while blow-off cocks are open.

In case of unusual water conditions which require special attention, the matter should be referred to the master mechanic or the general master mechanic, who will handle the matter with the chemist, furnishing him necessary information for additional instructions.

When locomotives are held with fires, waiting to be ordered at terminals or in work trains or other service, the fire should always be banked alongside sheets and across the front end of the firebox to prevent cold air striking the sheets.

If leaks develop at any screwed connection into the boiler, no attempt must be made to stop the leak by screwing in while the boiler is under steam pressure.

It is frequently necessary, on account of foaming conditions or other causes, to change water in locomotive boilers. When making a water change, the boiler must be filled with water as nearly as possible of the same temperature as the boiler.

When water hose is used to wash out any part of a boiler, it is considered a washout, and all plugs must be removed.

Every boiler, before being put in service and at least once after every twelve months' service, shall be subjected to a hydrostatic pressure 25 percent above the working pressure. The boiler should be filled with water as nearly as possible of the same temperature as the boiler. The temperature of the water, while the test is being applied, should not be lower than 90 degrees and not higher than 120 degrees F. In cases where boilers are filled with cold water, (which should be done only when the boiler is cold), the temperature should be raised to the above limits by using steam. While making the hydrostatic test, the pressure should fluctuate as little as possible.

Soda ash is only to be applied to boilers which have had flues removed and heavy repairs made. If not necessary to let water out of the boiler to make repairs after the hydrostatic test, let the water out to the desired amount to make the steam test and apply soda as follows:

Dissolve the soda ash in warm water. Remove the plug from the top of the boiler and pour the mixture through a funnel into the boiler. The boiler shall then be fired up and the steam pressure raised to not less than the allowable working pressure and the boiler and appurtenances thoroughly examined. All cocks, valves, seams, bolts, and rivets must be tight under this pressure and all defects disclosed must be repaired.

The boiler before being put in service should then be blown down and allowed to cool. After the boiler cools, it should be thoroughly washed out to remove any sediment caused by the use of soda ash.

Soda ash is placed in the boiler to form a protective alkaline film on the metal surfaces retarding pitting and corrosion and to remove grease and oil from the boiler.

The dome cap or inspection dome cap must be removed after making the hydrostatic tests, and the interior surface and connections of the boiler examined as thoroughly as conditions will permit.

The above instructions shall in no way conflict with

(Continued on page 103)

Determining the Size of Openings in Cylindrical Drums[*]

By D. S. Jacobus[†]

Much thought has been given by the A.S.M.E. Boiler Code Committee to the question of the largest size of an unreinforced opening that should be used in a cylindrical drum or shell of a boiler or pressure vessel without reinforcement. The problem is difficult as a correct solution involves the effect of the redistribution of the stresses in the shell after the most highly strained parts exceed the elastic limit of the material.

The distribution of stresses within the elastic limit around a circular opening, such as a tube hole, is well known, the stresses at the sides of the hole being over twice the average stress as ordinarily computed in determining the strength of a drum. Even should comprehensive experiments be made to show the redistribution of stresses when the elastic limit at some point is exceeded, there would still remain the problem of setting the stress that could be safely used in computing the strength of the drum.

It is well known that the so-called factors of safety used in the formulas for designing boilers and pressure vessels are factors for guidance in design and construction, rather than actual factors of safety. It has been pointed out that the ordinary rules for determining the factor of safety of a cylindrical tube sheet, which are based on the average stress in the ligaments between the tube holes, would differ greatly from those for a factor based on the maximum stress existing at the sides of the tube holes, and that the factors of safety now used are far from correct should they be defined as the ratio of the tensile strength of the material to the maximum stress in any part of the structure. The effect of the redistribution of the stresses after load is applied must be considered in any rules bearing on the strength of boilers and pressure vessels, and it is for this reason that the ductility of the material used in shells and other pressure parts is an important element.

Our rules for the construction of boilers and pressure vessels are the outcome of long experience through use in practical service. The factor of safety of 5 employed in the A.S.M.E. Boiler Code for the construction of power boilers and for pressure vessels for stationary service applies to shells pierced with holes and provides a reasonable margin for deterioration through corrosion or like causes. A pressure vessel having a shell which is not pierced with a hole and which contains no other stress raiser in the metal of the shell, such as a groove or deep scratch, could be used with safety at higher pressures than one which contains a stress raiser. Again, the class of service to which a pressure vessel is subjected must be taken into consideration as an intermittent pressure may lead to failure where a constant pressure would not. The effect of corrosion fatigue must also be considered. When it is appreciated how many elements enter the problem, the difficulty of obtaining

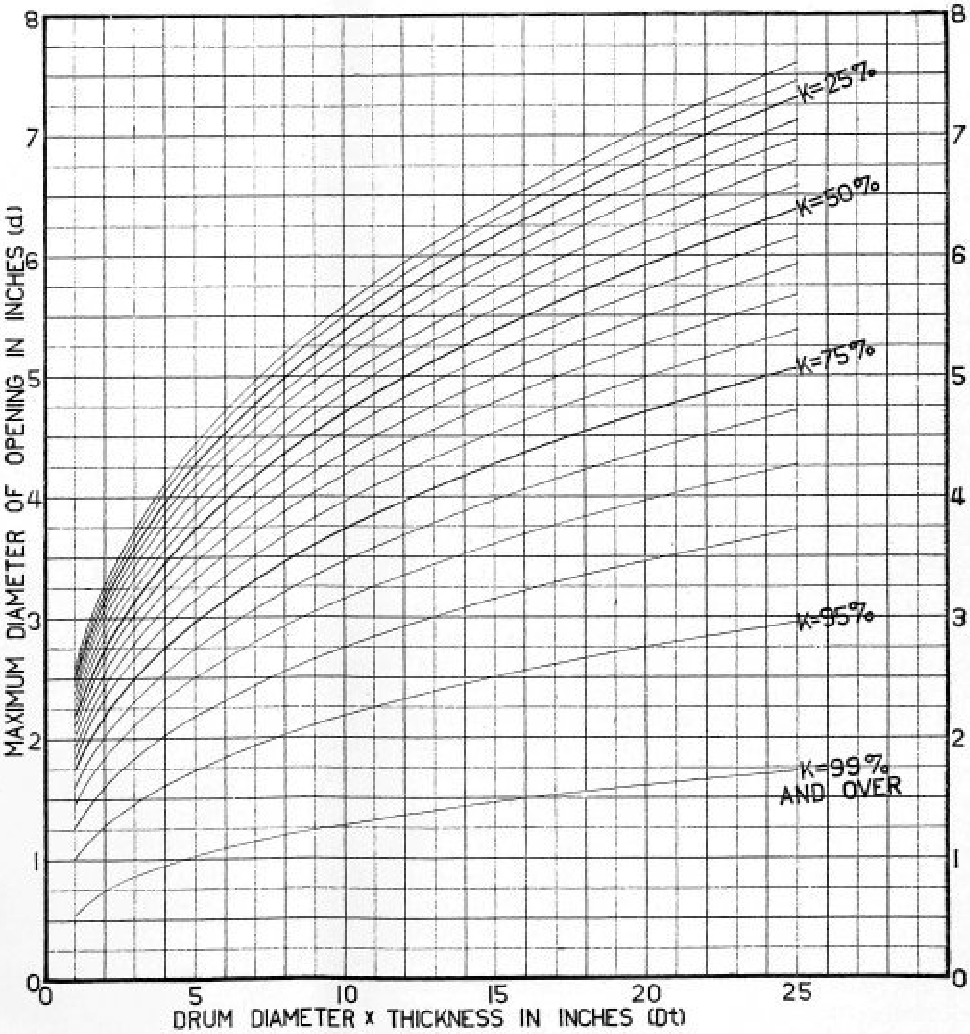


Fig. 1.—Maximum diameter of unreinforced openings in shells ($Dt \leqq 25$)

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[*] Published through the courtesy of *Mechanical Engineering*.

[†] Advisory Engineer, The Babcock & Wilcox Co., New York, N. Y. Past-President A.S.M.E.

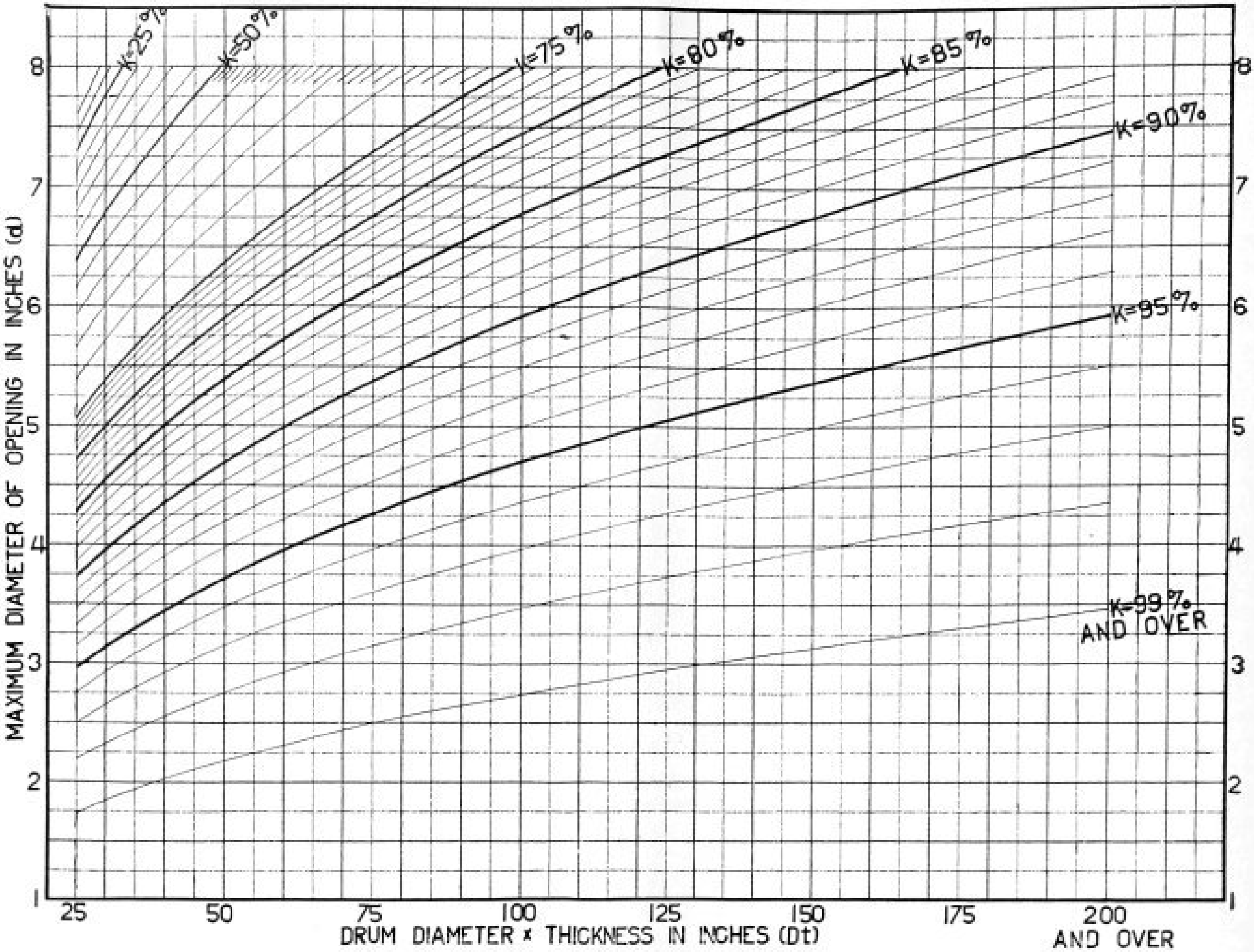


Fig. 2.—Maximum diameter of unreinforced openings in shells ($Dt \leqq 25$)

an exact solution will be apparent. The solution which follows is not an exact one, but has been developed to provide a rule which will give safe results without involving undue hardship. It is based on data secured through the use of certain constructions of which a great number have been used and which have given satisfaction in long-time service. It applies to circular holes in a cylindrical shell when used for tube holes or the like.

The rule is based mainly on the size of unreinforced openings that have been successfully used in long-time service for headers of superheaters and water walls. It applies to cylindrical drums constructed from steel of the quality specified in the A.S.M.E. Boiler Code. It was developed for shells 8 inches or more in diameter in which the shell thickness does not exceed one-fifth the diameter, and in which the largest hole does not exceed six-tenths the diameter of the shell.

If the law of geometric proportions would hold, and a 4-inch diameter hole in a 12-inch header ¾ inch thick has given safe and reliable results, it would follow that an unreinforced hole 20 inches in diameter could be placed in a drum 60 inches in diameter having a shell thickness of 3¾ inches, provided the working pressure was the same in both cases. In the rule proposed, the openings in the larger drums would be much smaller than sanctioned by the law of geometric proportions, as it is not necessary to provide as large openings as would be determined by this law; and, again, there are other elements, such as the cross-strains which come on connections to larger drums, which make it advisable not to sanction an unreinforced hole of too large a diameter.

The proposed rule is as follows:

$$d = 2.75 \sqrt[3]{Dt\,(1 - K)}$$

where d = maximum diameter of unreinforced circular opening, inches
t = thickness of shell plate, inches
K = ratio of computed stress in the solid plate to one-fifth of the minimum of the specified range of tensile strength of the steel forming the shell

$$= \frac{PD}{2St}$$

where P = maximum allowable working pressure, pounds per square inch
D = outer diameter of shell, inches.
S = one-fifth of the minimum of the specified range of tensile strength of the steel forming the shell, pounds per square inch.

A comparison of the data secured from practice with the proposed rule is given in Table 1. The results of the application of the rule are also shown in Figs. 1 and 2. As indicated in these figures, it is proposed to base

Table 1.—Comparison of Data Secured from Practice with the Proposed Rule

Efficiency percent	Drum diameter, inches	Thickness, inches	*d*, Practice inches	*d*, Proposed rule, inches
45	36	⅝	5⅞	6.36
50	12¾	1	4 17/32	5.10
50	11¾	1	4 1/32	4.96
50	9¾	1	3 9/32	4.58
70	42	⅝	4 17/32	5.47

the maximum diameter of opening to be used without reinforcement for a shell having a higher ligament efficiency than 99 percent on that for a shell having an efficiency of 99 percent. This will allow for placing a relatively small hole in a drum computed for an efficiency of 100 percent. Placing a hole in an otherwise seamless shell would greatly weaken the shell, but it should be remembered that the rule herein proposed is for use in connection with the so-called factor of safety of 5, which applies to drums pierced with holes.

In Fig. 2 the maximum diameter of opening is given as 8 inches with the idea that by so limiting the diameter no hardship will be involved and the rule may be made correspondingly safer.

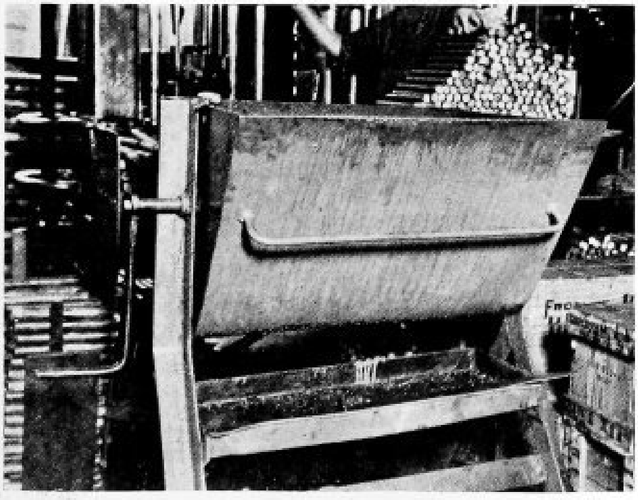
Device for Reclaiming Cutting Oil

An ingenious device for reclaiming cutting oil from chips and filings, after machining operations, has been developed by the Norfolk & Western Railroad at its Roanoke, Va., shops. Used in conjunction with the machining of staybolts, the device, as shown in the illustrations, consists of a completely welded hopper mounted in a welded angle framework. The hopper is open at the top and is free to revolve about the rod to which the handle at the left is attached. This allows the hopper to be dumped to remove chips and filings.

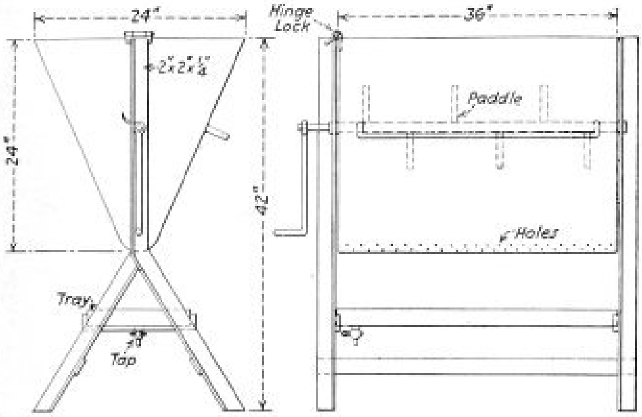
To keep the hopper in an upright position a hinge lock clamps over the upper section of the angle frame. The rod on which the hopper is suspended is fitted with a number of paddles so that the mass of oil and filings may be churned to hasten the filtering process. The bottom of the hopper is punched with numerous small holes to allow the oil to drop into a tray also supported on the stand. The tray is slightly tipped so that the oil will flow to the lower end at which point is located a petcock or tap by which the reclaimed oil may be withdrawn into smaller vessels.

The handle employed for churning the material is detachable and when not in use is hung on a hook at the side supporting angle. The entire device is of welded construction, the hopper being made of eighth inch sheet iron. It is approximately 36 inches in length, 24 inches in width and 42 inches in height.

Considerable savings over the cost of construction can be realized through the reclamation of the oil which, because of the presence of cuttings and other abrasives, was previously thrown away. In many cases such a reclaimer can be manufactured from materials found around the boiler shop. It is simple in construction and will save its cost many times over in cutting oil that may be reclaimed.



Oil reclaiming device in operation



Details of reclaiming hopper

Westinghouse Studies Corrosion of Welds

The effect of corrosion is becoming so important in welded boilers, pipes, containers for oil and chemicals that the Westinghouse Research Laboratories are studying it with special apparatus. In welded structures of rustless steel, alloys, or monel metal this problem is vital. The results of these studies may tell engineers how to fabricate structures so that all parts will be uniformly resistant to rust.

In a weld of low-carbon steel, corrosion may be expected to start in the zone where weld metal meets parent metal. At this point, according to the electrolytic theory of corrosion, a potential difference may exist which is responsible for an accelerated attack. Oxides and other heterogeneous particles, if present in the weld, tend to hasten corrosion by the formation of electrolytic cells. The soil corrosion of pipe line is an example. A more homogeneous weld should be expected with coated welding wire than with bare electrodes since the coating resists the entrance of foreign elements.

To corroborate many facts already known on this subject and to uncover others, a special corrosion device has been built which greatly hastens the slow process of rust. In it the test specimens are subjected to intermittent immersions in a corroding liquid. The apparatus suddenly immerses the samples for a definite period, leaving them at rest, and then removing and exposing them to air for a definite period. They are moving only when being lowered or raised, a time which is a very small fraction of the cycle.

Samples are suspended from a rack by glass hooks, horse hair, or silk, and a motor-driven crank shaft raises and lowers the rack. The driving motor is controlled by a timing device composed of a synchronous motor operating a contact, which causes the motor periodically to turn the crank shaft a half revolution. The timing can be set for any cycle of test operation. To obtain reproducible results the corrosive liquids are kept at a constant temperature, by circulating water of a constant temperature along the outside of the vessels containing the corrosive liquids.

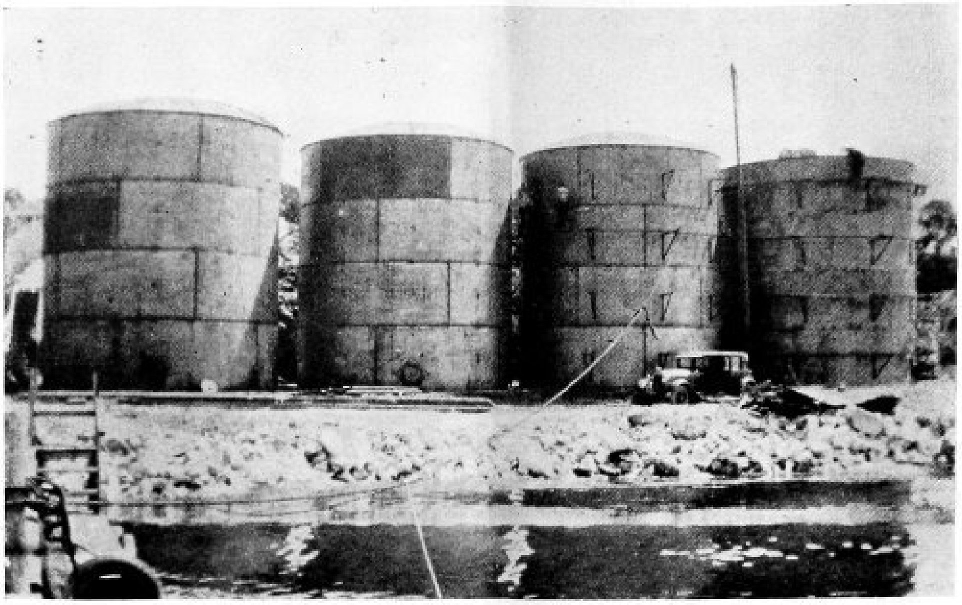


Fig. 1.—Tanks erected in thirteen days

Arc Welding Field Storage Tanks

*By G. G. Landis**

Inasmuch as most tank and boiler shops either engage in field erection of storage tanks or roll plates for field tanks, the application of arc welding to this type of construction merits discussion here. In field storage tank building the shielded-arc process, which allows a 300 percent increase in welding speeds, and results in welds of 65,000 to 75,000 pounds per square inch tensile strength, high ductility and extreme resistance to corrosion, should be used.

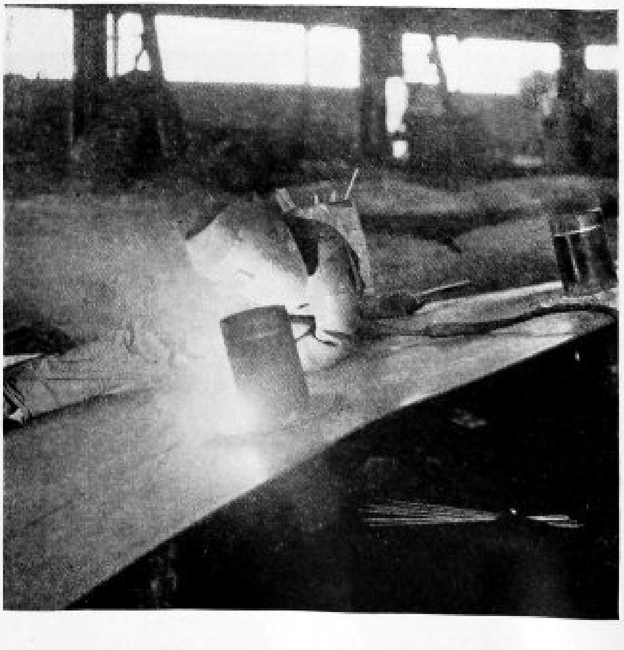
In the main there are two methods of erecting storage tanks in the field, with and without bolts. Obviously after the plates have been rolled in the shop and the fittings fused into place, the only problem is that of holding the tank in place for welding. It is a simple matter to punch the plates for temporary bolts in the shop and this practice is widely followed.

On the other hand, the exponents of arc welding have become so independent of bolts and rivets, so proud of rivetless construction that a method of building tanks without punching the plates has been devised. Both procedures are efficient, the margin of greater efficiency resting with the experience of the field crew.

Fig. 1 shows four 25- by 30-foot gasoline storage tanks erected by the Buffalo Tank Corporation, near Troy, N. Y. Plates for these tanks were bolted before welding. For all four tanks three days of shop work were required. In the field, two welder operators, two helpers, a foreman and two laborers, erected the first two tanks in five standard working days. The entire job was finished in ten days, less than two weeks of actual labor in shop and field. These tanks will not leak or sweat at the seams. They are in effect, one piece of steel. In Fig. 2 a detail of the shop work on this type of tank is shown.

Built without a bolt or rivet near the seams are the two new 80,000-barrel tanks erected at Texas City, Texas, by the Wyatt Metal and Boiler Works. Fig. 3

Fig. 2.—Welding plates for erection



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* Chief Engineer, The Lincoln Electric Company, Cleveland, Ohio.

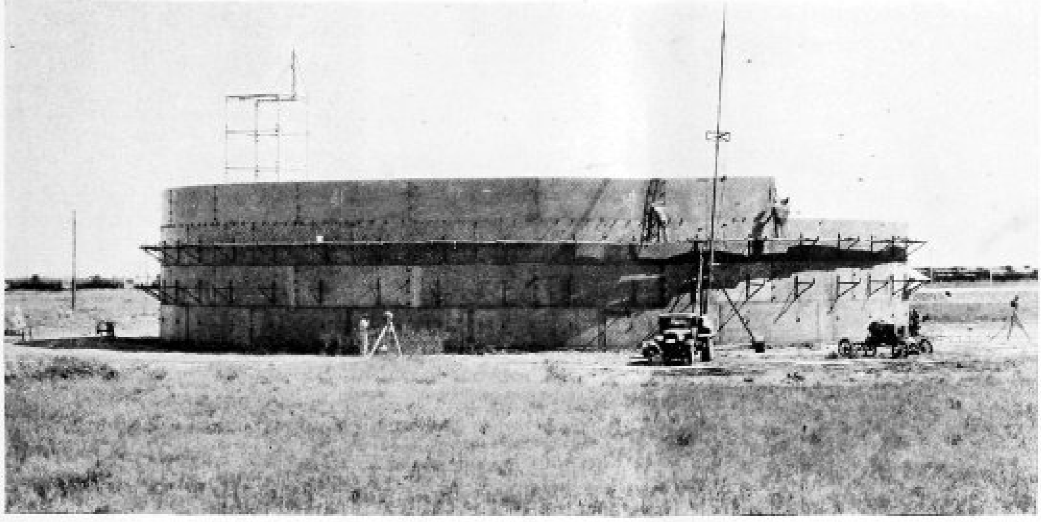

Fig. 3.—Storing tank of 80,000 barrels capacity, erected without bolts or rivets used at the seams

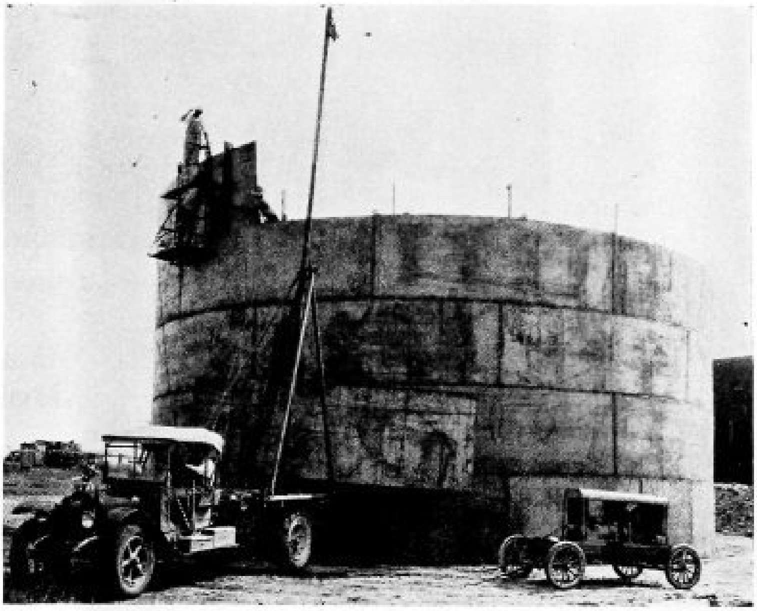

Fig. 4.—(Right) Lifting a top ring plate for 20,000 barrel tank. Fig. 5 (Above) Tack welds finished and welding operation started

shows one of these tanks under construction. Holes in the center of the plates carry the shelf angles which support the staging. A patented clamp is used to hold the plates in position.

The welding equipment required for such work is simple and inexpensive compared to the savings over other types of construction. Where power is available, a motor-driven machine of 300- or 400-ampere N.E.M.A. rating is now widely used. A machine of this capacity is needed to supply current for the shielded-arc process, which utilizes a greater welding heat and allows greater speeds. Modern efficient welding equipment includes a special electrical circuit for welding which should include variable voltage design, a stabilized-arc circuit, separate excitation, and so forth. These are principles which the manufacturers of arc-welding equipment and the operators in industry have found necessary to secure the proper welding current and arc stability.

The same principles can be incorporated in gasoline-motor driven welders for use where power is not available. Field storage tanks are erected by such units, which can be mounted on motor trucks or on separate trailers. Companies which specialize in tank erection usually equip themselves with a heavy truck carrying a light derrick, motor-driven welder and all other supplies. Such a concentration of tools may seem simple, but it is just such apparatus which reduces field erection costs. An automatic idling device on the gasoline-driven welder will idle the motor 10 seconds after the arc is broken and allow the motor to resume proper speed the instant the arc is re-struck. This device saves 25 percent or more in fuel.

Power for any efficient welder can be supplied at lower cost than the compressed air required in a rivet hammer required for the same work.

The economy of welding on large storage tanks was first realized in the roofing and re-roofing of large petroleum storage units. With the shielded-arc process, welding speeds of 40 feet per hour are easily attained. During welding the roof is blocked up a few inches above the supporting members, so that, on cooling, undesirable buckling is avoided.

A graphic portrayal of storage tank erection with arc welding is shown in Figs. 4 and 5. Two of these tanks, of 20,000 barrel capacity, were for the Eckerson Refining Company of Corpus Christi, Texas. Under test conditions these tanks did not show a single leak.

Fig. 4 shows the plate being hoisted and fitted into

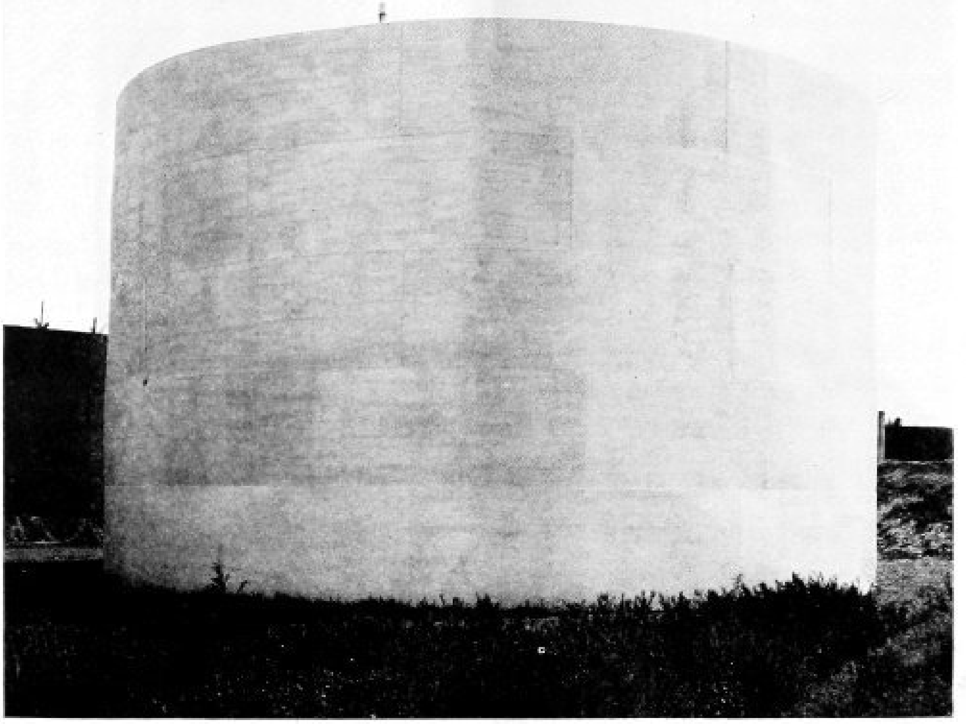

Fig. 6.—Completed 20,000-barrel tank

position. The gasoline-driven welder is in the foreground. The vertical supports showing above the fourth ring are part of the temporary staging inside the tank. After alining the section, the operator begins to tack weld the plates into position. This ring is inserted within the fourth ring and another welder operator tacks the plate on the inside of the tank. In Fig. 5, the operator has begun to weld the outside seam. The tack welds are plainly visible at the lower edge of the plate.

In Fig. 6, the completed tank, covered with aluminum paint, is shown. A welded tank with no rivets to sweat will have a better appearance, especially when this type of paint is used. Sweating from the rivets will cause a black streak or smudge below the rivets. The cost of protective aluminum paint is such that it is necessary to weld storage tanks if the economic advantages of this protection are to be secured and good appearance maintained.

A field storage tank, while not operating under high pressure is subject to other forces, corrosion and wind, especially. Welds made in a shielded arc, free from oxides and nitrides, are more resistant to corrosion than mild steel. Wind on one side of the tank cannot open the seams as the welded shell acts as a continuous unit and springing of the plates is impossible. Calking is, of course, eliminated.

Arc-welded field tanks are leakproof, and they cost less. On the two tanks of the type shown in Fig. 6 the welding contract was $1000, $750 less than the lowest bid on riveted construction, and the welding contractor realized a large profit.

The Bird-Archer Company has moved its offices from 1 East Forty-second Street to 90 West Street, New York City.

Work of the A.S.M.E. Boiler Code Committee

The Boiler Code Committee meets monthly for the purpose of considering communications relative to the Boiler Code. Any one desiring information as to the application of the code is requested to communicate with the secretary of the Committee, 29 West 39th St., New York, N. Y.

The procedure of the committee in handling the cases is as follows: All inquiries must be in written form before they are accepted for consideration. Copies are sent by the secretary of the committee to all of the members of the committee. The interpretation, in the form of a reply, is then prepared by the committee and passed upon at a regular meeting of the committee. This interpretation is later submitted to the council of The American Society of Mechanical Engineers for approval, after which it is issued to the inquirer and published.

CASE No. 688 (Reopened).—*Inquiry:* When used in the construction of the boiler pressure parts, can material made in accordance with Specifications S-2 be considered as being in compliance with Pars. P-2 and P-3?

Reply: Revised A.S.T.M. Specifications A89-31T provide for firebox-quality plate. The Boiler Code Committee recommends that pending the adoption of revisions to Specifications S-2 for Steel Plates of Flange Quality for Forge Welding to make them identical with A.S.T.M. Specifications A89-31T, the use of these revised specifications be permitted as being in compliance with Pars. P-2 and P-3. In line with the above recommendation the committee has under consideration the following proposed revisions of Pars. P-2 and P-3.

P-2*a* Steel plates for any part of a boiler subject to pressure and exposed to the fire or products of combustion shall be of firebox quality in accordance with code specifications S-1 or S-2.

b Steel plates for any part of a boiler subject to pressure and not exposed to the fire or products of combustion shall be of firebox or flange quality in accordance with code specifications S-1 or S-2.

c Minimum tensile-strength limits other than those specified for flange and firebox plates may be used provided a range of 10,000 pounds per square inch is specified, that the maximum limit does not exceed 65,000 pounds per square inch, and that the steel conforms in all other respects to Specifications S-1 and S-2.

d Seamless steel-drum forgings made in accordance with code Specifications S-4 may be used for any part of a boiler for which firebox quality only is specified or for which either firebox or flange quality is permitted.

P-3 Open-hearth steel pipe or steel tubing in accordance with Specifications S-18 or S-17 may be used for a boiler drum or other pressure part exposed to the fire or products of combustion, provided the nominal diameter of the pipe or tubing is not greater than 18 inches.

CASE NO. 704 (Reopened).—*Inquiry:* What is the maximum allowable unit joint working stress for Class 3 vessels, with double-welded lap-type longitudinal joints for thickness of ⅜ inch or less under the provisions of Par. U-73 of the code? It is noted that this allowable working stress is not provided for in Par. U-70.

Reply: It is the opinion of the committee that the maximum unit joint working stress for double full-fillet lap welds for longitudinal joints of plates less than ¼ inch in thickness should be 5600 pounds per square inch, and for plates ¼ inch and over in thickness, 7000 pounds per square inch of cross section. It is proposed to restrict the use of full-fillet welded tanks to certain classes of service.

CASE NO. 710.—*Inquiry:* Is it the intent of the provisions in Pars. P-107 and U-75 of the code that the thickness of the reinforcing pad shall not be less than ¾ inch, irrespective of the thickness of the shell or head to which it is fitted?

Reply: It is the opinion of the committee that the ¾-inch thickness of reinforcing pads given in Pars. P-107 and U-75 was intended to apply only to the thicker shells or heads, and that a thickness of 1 inch may be taken as the least thickness of shell or head to which the requirement for ¾-inch reinforcing pads applies. A revision of these paragraphs is contemplated.

CASE NO. 711.—*Inquiry:* Can Par. P-203*d* be applied to determine the maximum pitch between the centers of through rods, or does this section apply only to the distance between the inner surface of the shell and the centers of through rods?

Reply: Par. P-203*d* is intended to apply to the spacing in stayed surfaces between the inner surface of a shell to which the surface is attached at its boundary and the centers of the adjoining braces. It is the opinion of the committee that the spacing between the centers of through rods or other types of braces supporting a stayed surface should not exceed the limits specified in Par. P-199 or Table P-7.

CASE NO. 713.—*Inquiry:* Is it the intent of Par. P-303 of the code to require two stop valves in the connection from a boiler to a common steam main when any one of the battery of boilers does not have a manhole opening?

Reply: It is the opinion of the committee that if any one of two or more boilers connected to a common steam main is not fitted with a manhole opening so that entry to the interior is impossible, that boiler need not be fitted with more than one stop valve. A revision of Par. P-303 to this effect is under consideration.

CASE NO. 714.—*Inquiry:* When outlets are formed by welding to the shell or head of a vessel a forged steel nozzle having a saddle flange and neck of seamless construction, is it not permissible to compute the reinforcing effect of the saddle flange and a portion of the neck, in place of the rules given in Par. U-75 of the code for the design of such reinforcing rings and reinforced nozzles?

Reply: It is the opinion of the committee that the construction outlined in the inquiry may be considered as meeting the intent of the code requirements provided:

1.—The net cross-sectional area of the manufactured nozzle (having a saddle flange and neck of seamless construction) on a line parallel to the axis of the shell, shall not be less than the cross-sectional area of shell plate removed on the same line. This area shall be computed by multiplying the required shell plate thickness (calculated by the formula in Par. P-180 and/or Par. U-20, using $E = 1$) by the maximum length of the shell plate removed by the opening.

2.—The reinforcing value of such a nozzle shall consist of the cross-sectional area of the saddle flange plus the cross-sectional area of the neck to a height 3 times the thickness of the saddle flange.

3.—All welds shall be strength welds as provided for by detail 6 of Fig. U-16.

4.—In cases where the replacement value of the nozzle does not equal the above requirements, an inside reinforcing ring[1] may be used to provide the additional necessary replacement.

(Revisions of Pars. U-75 and P-107 of the code are now in the hands of the Boiler Code Committee. This Case is issued to be used until the final revisions are adopted.)

CASE NO. 716.—*Inquiry:* Can any pressure part of a boiler of circular cross section be fusion welded in accordance with the requirements of Pars. P-101 to P-111 of the code? The part in question is a superheater header 14 inches in diameter having a shell thickness of 1 inch.

Reply: In the opinion of the committee such a structure is properly classified as a drum or shell.

CASE NO. 718.—*Inquiry:* Is it necessary to post-heat the longitudinal joints of cylindrical shells of heating boilers in the same manner as preheating is required for gas welding in Par. H-80 of the code, or is such post-heating with electric welding optional? It is the impression of certain manufacturers that post-heating is not mandatory when electric welding is used.

Reply: In view of the advance in the art of welding since Par. H-80 was formulated, the committee considers the provisions for preheating or post-heating to be unnecessary.

Chesapeake and Ohio Boiler Rules

(Continued from page 96)

the Interstate Commerce Commission's laws, rules and instructions for the inspection and testing of locomotives, tenders and their appurtenances.

To prevent any damage being done by corrosion while engines are laid up out of service at any point during winter months, one half the amount of soda ash for the respective classes of boilers should be applied to the boilers through the bottom washout plug holes in the back head, syphoning the soda ash by the use of an air gun.

During summer months, engines laid up out of service will be filled with treated water, where available; and where treated water is not available, soda ash should be applied as above outlined.

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[1] The outside diameter of the reinforcing ring must at least equal the outside diameter of the saddle of the nozzle.

Comparative Tests on Types of Locomotive Crown Bolts

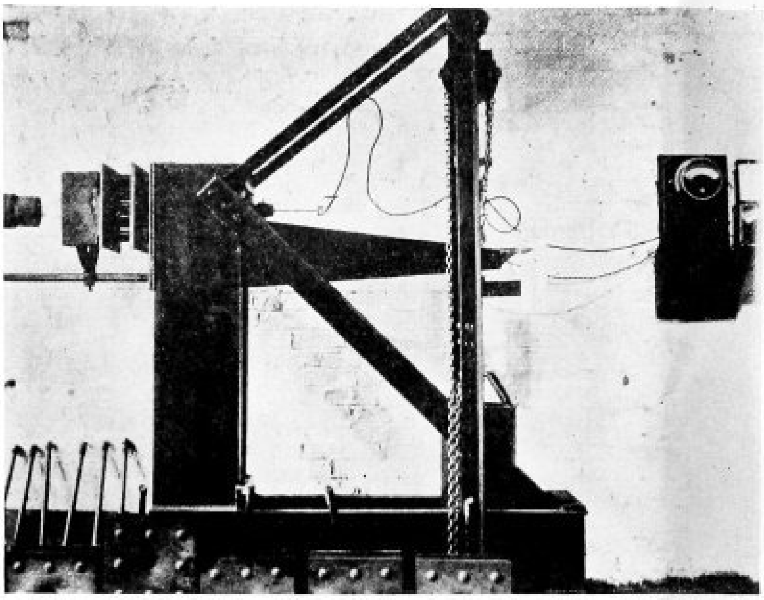

Fig. 1.—Machine used for testing crown bolts

In collecting and compiling data on the subject, "What is the best type of crown stay, buttonhead type, taper bolt with ends riveted over or straight bolt with ends riveted over," a questionnaire was sent to each member of the Master Boiler Makers' Association. The questionnaire pertained to type of crown bolts used in locomotive boilers on different railroads, advantages of the particular type used and disadvantages experienced.

It was gratifying, indeed, to receive replies from 109 members representing 60 railroad companies. Button-head type crown stays were used generally in radial stay fireboxes until about 15 years ago. Answers to our inquiries from 60 railroads disclosed that at the present time 77 percent of them have now adopted the hammered-head crown bolt while 23 percent continue to use the button-head type. The type of crown bolts standard on those railroads from which replies were received are shown in the following tabulation giving the kind of head and the degree of taper:

Crown Bolt Types Used by Railroads

Kind of Head	Taper to the Foot	Number of Railroads Using as Standard
Button head	straight	1
Button head	3/16"	1
Button head	½"	2
Button head	¾"	9
Button head	⅞"	1
Hammered head	straight	4
Hammered head	¾"	2
Hammered head	1¼"	1
Hammered head	1½"	23
Hammered head	2"	15
Hammered head	3"	1

Note.—Three (3) railroads use part installation of taper and part installation of straight hammered-head type crown stays.

As shown in the foregoing tabulation the degree of taper at the crown-sheet fit on both the button-head and hammered-head crown stay varies from straight to 3 inches in 12 inches. The most popular taper for button-head type crown stays is ¾ inch in 12 inches, while that for hammered-head crown stays is 1½ inches in 12 inches and 2 inches in 12 inches.

The use of hammered-head crown stays is becoming more and more general due to the fact that they are more economical to apply and are easier to maintain. Probably they originally came into general use because of the need of a smaller head than the button-head type in oil-burning locomotives. The large amount of iron in the button-head type becomes overheated and wastes away due to high temperatures of the fire causing the bolt in many cases to become weakened.

Experience has demonstrated that the hammered-head crown stay, while not as strong as the button-head type, has ample strength under ordinary working conditions; that is, with water over the crown sheet, as it will stand

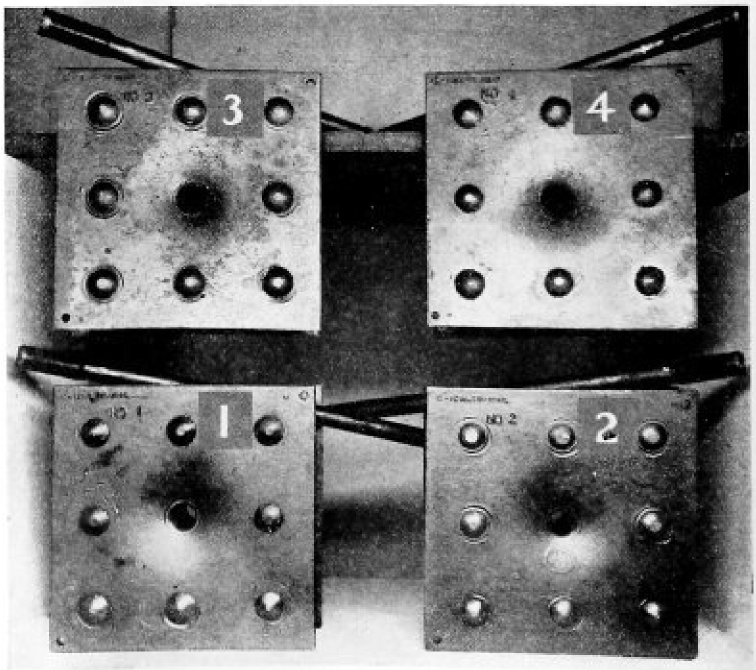

Fig. 2.—Tests on straight-threaded crown stays. Tests 1 and 2—1⅛ inc diameter. Tests 3 and 4—1¼ inch diameter

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* Committee report prepared for the twenty-second convention of the Master Boiler Makers' Association to be held at Chicago, Ill.

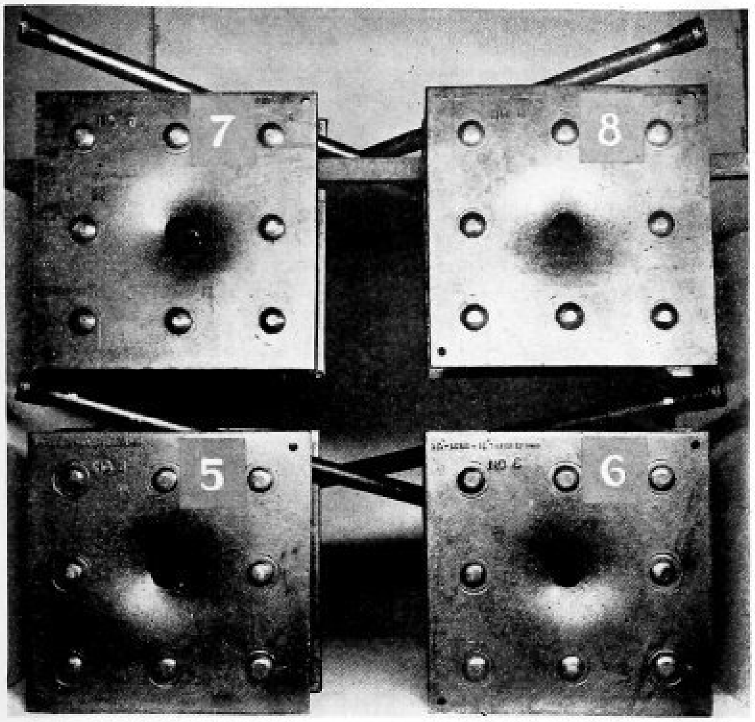

Fig. 3.—Tests on taper head crown stays—19/32-inch diameter at large end of taper

Comparative Tests

Demonstrating the Holding Power of All Standard Types of Crown Bolts in a Heated Sheet—Test Data with Corresponding Temperature and Deflections

Also Average Curve Readings from Which the Average Curve of Each of the Two Tests May Be Plotted

Tests No. 1 & No. 2—1⅛-inch straight U. S. thread	Table 1
Tests No. 3 & No. 4—1¼-inch straight U. S. thread	Table 2
Tests No. 5 & No. 6—1⅛-inch taper head stay—1½-inch taper of head	Table 3
Tests No. 7 & No. 8—1¼-inch taper head stay—2-inch taper of head	Table 4
Tests No. 9 & No. 10—1⅛-inch button head stay—¾-inch taper with 1 21/32-inch x ⅝-inch head	Table 5
Tests No. 11 & No. 12—1¼-inch taper head stay—1½-inch taper of head	Table 6
Tests No. 13 & No. 14—1¼-inch taper head stay—2-inch taper of head	Table 7
Tests No. 15 & No. 16—1¼-inch button head stay—¾-inch taper with 1 27/32-inch x 11/16-inch head	Table 8

Table 1

TEST NO. 1—1⅛-INCH STRAIGHT 12 U. S.

Time Min.	Pyrometer Reading	Actual Temp. Degrees F.	Deflection Readings	Actual Deflection Inches
0	2.00	100	0.56	0
1	5.15	208	0.56	0
2	12.00	432	0.56	0
3	15.25	537	0.57	0.01
4	21.00	726	0.57	0.01
5	25.00	856	0.58	0.02
6	28.50	968	0.60	0.04
7	30.50	1030	0.63	0.07
8	32.00	1070	0.70	0.14
9	33.10	1110	0.83	0.27
10	33.20	1114	0.92	0.36
11	33.20	1114	0.98	0.42
12	33.25	1115	1.02	0.46
13	33.40	1120	1.08	0.52
14	33.45	1121	1.21	0.65

TEST NO. 2—1⅛-INCH STRAIGHT 12 U. S.

Time Min.	Pyrometer Reading	Actual Temp. Degrees F.	Deflection Readings	Actual Deflection Inches
0	2.00	100	0.33	0
1	7.50	285	0.33	0
2	9.50	351	0.35	0.02
3	20.20	699	0.36	0.03
4	25.90	884	0.38	0.05
5	29.60	1001	0.53	0.20
6	31.50	1060	0.82	0.49
6¼	32.20	1082	1.00	0.67

Average Curve Readings

Temp.	Deflection
0	0
100	0
200	0
300	0
400	0.006
500	0.015
600	0.020
700	0.024
800	0.030
900	0.040
950	0.056
1000	0.095
1050	0.205
1100	0.625
1102	0.660

Table 2

TEST NO. 3—1¼-INCH STRAIGHT 12 U. S.

Time Min.	Pyrometer Reading	Actual Temp. Degrees F.	Deflection Readings	Actual Deflection Inches
0	2.00	100	0.33	0
1	9.50	350	0.33	0
2	16.00	560	0.33	0
3	22.50	774	0.34	0.01
4	28.25	959	0.36	0.03
5	32.50	1091	0.42	0.09
6	35.00	1169	0.63	0.20
7.1	36.40	1210	1.02	0.68

TEST NO. 4—1¼-INCH STRAIGHT 12 U. S.

Time Min.	Pyrometer Reading	Actual Temp. Degrees F.	Deflection Readings	Actual Deflection Inches
0	2.00	100	0.27	0
1	8.25	309	0.27	0
2	15.00	531	0.28	0.01
3	21.25	733	0.29	0.02
4	26.50	904	0.31	0.04
5	30.50	1030	0.34	0.07
6	33.25	1115	0.49	0.22
7	34.90	1165	0.74	0.47
7.56	35.50	1183	0.94	0.67

Average Curve Readings

Temp.	Deflection
0	0
100	0
200	0
300	0
400	0
500	0.004
600	0.008
700	0.012
800	0.018
900	0.028
950	0.038
1000	0.054
1050	0.077
1100	0.122
1150	0.225
1198	0.675

Average Time 7.33 Min.

Table 3

TEST NO. 5—1⅛-INCH DIA. 1½-INCH TAPER 12 U. S.

Time Min.	Pyrometer Reading	Actual Temp. Degrees F.	Deflection Readings	Actual Deflection Inches
0	2.00	100	0.30	0
1	9.75	359	0.31	0.01
2	18.00	627	0.32	0.02
3	25.00	856	0.34	0.04
4	30.60	1033	0.38	0.08
5	34.00	1138	0.60	0.30
6	35.90	1195	. . .	. . .
6.1	36.10	1201	0.90	0.60

TEST NO. 6—1⅛-INCH DIA. 1½-INCH TAPER 12 U. S.

Time Min.	Pyrometer Reading	Actual Temp. Degrees F.	Deflection Readings	Actual Deflection Inches
0	2.00	100	0.42	0
1	10.50	383	0.43	0.01
2	18.40	640	0.43	0.01
3	25.40	868	0.45	0.03
4	30.50	1030	0.48	0.06
5	33.75	1130	0.60	0.18
6	35.60	1186	0.84	0.42
7.04	36.70	1219	1.10	0.68

Average Curve Readings

Temp.	Deflection
0	0
100	0
200	0.004
300	0.008
400	0.012
500	0.014
600	0.016
700	0.021
800	0.030
900	0.040
950	0.046
1000	0.058
1050	0.082
1100	0.152
1150	0.294
1200	0.570
1210	0.640

Average Time 6.57 Min.

Table 4

TEST NO. 7—1¼-INCH DIA. 2-INCH TAPER—12 U. S.

Time Min.	Pyrometer Reading	Actual Temp. Degrees F.	Deflection Readings	Actual Deflection Inches
0	2.00	100	0.36	0
1	10.75	391	0.36	0
2	18.80	653	0.37	0.01
3	26.10	891	0.40	0.04
4	31.20	1052	0.46	0.10
5	34.40	1150	0.69	0.33
6	36.00	1198	0.90	0.54
6.4	36.30	1208	1.10	0.74

TEST NO. 8—1¼-INCH DIA. 2-INCH TAPER—12 U. S.

Time Min.	Pyrometer Reading	Actual Temp. Degrees F.	Deflection Readings	Actual Deflection Inches
0	2.00	100	0.40	0
1	9.40	347	0.40	0
2	16.60	581	0.41	0.01
3	23.60	810	0.42	0.02
4	29.50	998	0.46	0.06
5	33.40	1120	0.59	0.19
6	35.60	1186	0.86	0.46
6.75	36.70	1219	1.10	0.70

Average Curve Readings

Temp.	Deflection
0	0
100	0
200	0
300	0
400	0.001
500	0.006
600	0.012
700	0.017
800	0.022
900	0.037
950	0.050
1000	0.066
1050	0.094
1100	0.163
1150	0.315
1200	0.555
1212	0.720

Average Time 6.57 Min.

Table 5

TEST NO. 9—1⅛-INCH DIA. ¾-INCH TAPER 1 21/32 BUTTON

Time Min.	Pyrometer Reading	Actual Temp. Degrees F.	Deflection Readings	Actual Deflection Inches
0	2.00	100	0.34	0
1	8.75	326	0.34	0
2	16.30	372	0.35	0.01
3	23.25	798	0.36	0.02
4	29.40	995	0.37	0.03
5	32.20	1082	0.40	0.06
6	34.90	1165	0.46	0.12
7	36.90	1225	0.57	0.23
8	38.20	1263	0.69	0.35
9	39.20	1292	0.79	0.45
10	40.20	1321	0.88	0.54
11	40.80	1338	0.94	0.60
12	41.30	1352	1.00	0.66
13	41.70	1364	1.04	0.70
14	42.40	1384	1.09	0.75
15	42.50	1387	1.13	0.79
16	42.50	1387	1.17	0.83
17	42.30	1381	1.20	0.86
18	42.25	1380	1.23	0.89
19	42.20	1379	1.28	0.94
20	41.90	1370	1.31	0.97
21	41.50	1358	1.37	1.03
22	40.60	1353	1.45	1.11
22¾	39.90	1313	1.60	1.26

TEST NO. 10—1⅛-INCH DIA. ¾-INCH TAPER 1 21/32 BUTTON

Time Min.	Pyrometer Reading	Actual Temp. Degrees F.	Deflection Readings	Actual Deflection Inches
0	2.00	100	0.26	0
1	8.50	318	0.26	0
2	14.50	513	0.27	0.01
3	20.40	706	0.28	0.02
4	25.40	868	0.29	0.03
5	20.50	998	0.30	0.04
6	32.60	1094	0.32	0.06
7	34.80	1162	0.38	0.12
8	36.50	1213	0.48	0.22
9	37.60	1245	0.61	0.35
10	38.20	1263	0.71	0.45
11	38.90	1284	0.80	0.54
12	39.40	1298	0.87	0.61
13	39.80	1310	0.94	0.68
14	40.10	1318	0.99	0.73
15	40.20	1321	1.03	0.77
16	40.40	1327	1.08	0.82
17	40.50	1330	1.14	0.88
18	40.60	1332	1.18	0.92
19	40.60	1332	1.22	0.96
20	40.70	1336	1.26	1.00
21	40.70	1336	1.31	1.05
22	40.70	1336	1.37	1.11
23	40.70	1336	1.44	1.18
24	40.60	1332	1.50	1.24
24.8	40.30	1324	1.70	1.44

Average Curve Readings

Temp.	Deflection
0	0
100	0
200	0
300	0
400	0.004
500	0.009
600	0.014
700	0.019
800	0.023
900	0.029
950	0.031
1000	0.038
1050	0.048
1100	0.071
1150	0.107
1200	0.176
1250	0.340
1300	0.540
1350	0.750
1360	0.850
1320	1.350

Average Time 24.27 Min.

Table 6

TEST NO. 11—1¼-INCH DIA. 1½-INCH TAPER—12 U. S.

Time Min.	Pyrometer Reading	Actual Temp. Degrees F.	Deflection Readings	Actual Deflection Inches
0	2.00	100	0.30	0
1	9.50	351	0.31	0.01
2	17.00	595	0.31	0.01
3	23.80	816	0.32	0.02
4	29.60	1001	0.34	0.04
5	33.50	1122	0.43	0.13
6	36.10	1201	0.68	0.38
7	37.70	1248	0.93	0.63
7.2	38.10	1260	1.03	0.73

TEST NO. 12—1¼-INCH DIA. 1½-INCH TAPER—12 U. S.

Time Min.	Pyrometer Reading	Actual Temp. Degrees F.	Deflection Readings	Actual Deflection Inches
0	2.00	100	0.34	0
1	10.30	376	0.34	0
2	17.00	595	0.35	0.01
3	23.30	800	0.36	0.02
4	28.50	967	0.38	0.04
5	32.20	1082	0.42	0.08
6	35.00	1168	0.59	0.25
7	36.60	1216	0.82	0.48
7.75	37.50	1242	1.02	0.68

Average Curve Readings

Temp.	Deflection
0	0
100	0.001
200	0.003
300	0.005
400	0.007
500	0.009
600	0.010
700	0.014
800	0.020
900	0.027
950	0.032
1000	0.039
1050	0.058
1100	0.097
1150	0.196
1200	0.375
1250	0.690
1252	0.705

Average Time 7.47 Min.

Table 7

TEST NO. 13—1¼-INCH DIA. 2-INCH TAPER 12—U. S. TH.

Time Min.	Pyrometer Reading	Actual Temp. Degrees F.	Deflection Readings	Actual Deflection Inches
0	2.00	100	0.26	0
1	10.60	386	0.27	0.01
2	17.50	611	0.28	0.02
3	23.80	784	0.29	0.03
4	29.00	983	0.30	0.04
5	32.70	1098	0.34	0.08
6	35.40	1180	0.43	0.17
7	37.20	1233	0.63	0.37
8	38.60	1275	0.82	0.56
8.4	38.80	1281	1.00	0.74

TEST NO. 14—1¼-INCH DIA. 2-INCH TAPER—12 U. S. TH.

Time Min.	Pyrometer Reading	Actual Temp. Degrees F.	Deflection Readings	Actual Deflection Inches
0	2.00	100	0.38	0
1	12.50	448	0.39	0.01
2	18.80	653	0.39	0.01
3	25.00	856	0.41	0.03
4	30.00	1014	0.43	0.05
5	33.70	1128	0.48	0.10
6	36.30	1208	0.63	0.25
7	38.00	1257	0.87	0.49
7.4	38.40	1268	1.00	0.62

Average Curve Readings

Temp.	Deflection
0	0
200	0.001
300	0.006
400	0.011
500	0.016
600	0.020
700	0.025
800	0.030
900	0.034
950	0.036
1000	0.044
1100	0.081
1150	0.116
1200	0.228
1250	0.454
1273	0.680

Average Time 7.9 Min.

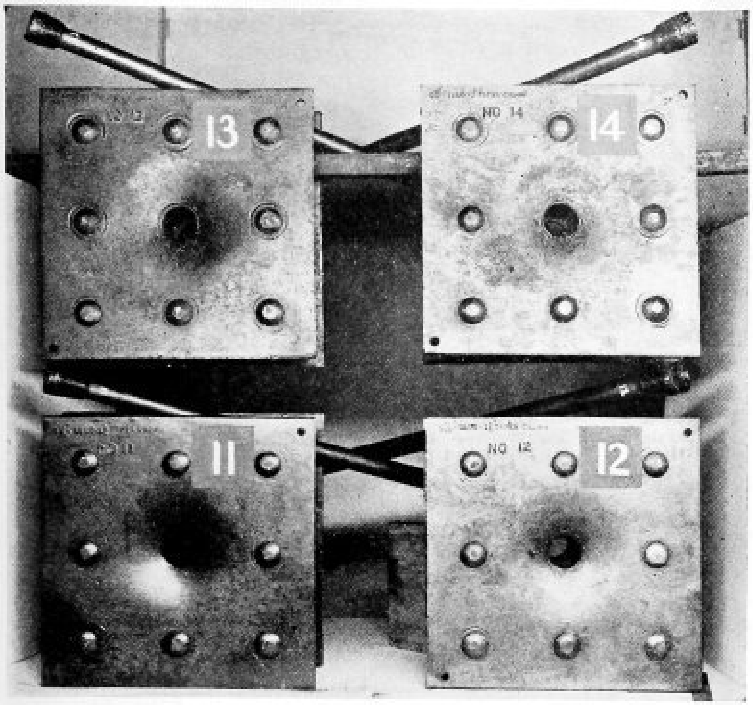

Fig. 4.—Tests on taper head crown stays—1 13/32-inch diameter at large end of taper

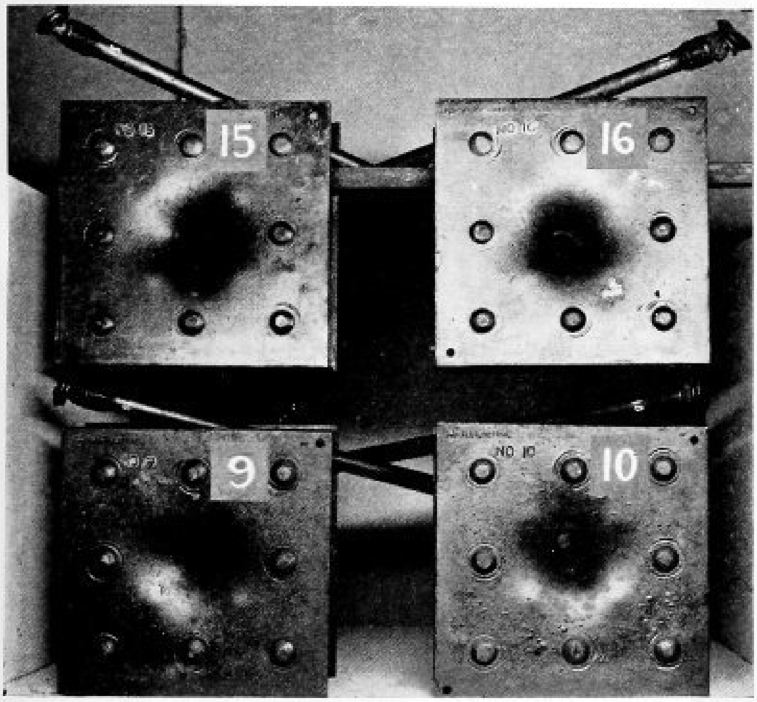

Fig. 5.—Button-head crown stay tests. Tests 9 and 10—1 7/32-inch diameter under head. Tests 15 and 16—1 11/32-inch diameter under head

from 18,000 to 20,000 pounds stress before pulling through the crown sheet. Under ordinary conditions, the bolt is called upon to stand, with 200 pounds pressure and a spacing of 4 by 4 inches, about 3200 pounds.

This committee accompanied by several Bureau of Locomotive Inspection I. C. C. inspectors and several members of our association conducted some tests to determine the strength of button-head type and hammered-head type crown bolts of various tapers under similar conditions which would exist when crown sheets are overheated, as a result of low water while under pressure. We feel that these tests demonstrate, in a very practical way, the holding power of standard types of crown bolts in a heated sheet.

Figs. 2, 3, 4 and 5 illustrate standard type crown bolts and plates which were heated and pulled while the bolts were under 3200 pounds load, which is the equivalent of load on a crown bolt with spacing 4 by 4 inches at 200 pounds working pressure.

Fig. 6 shows a drawing of the machine used to test crown bolts and the manner in which load was put on the crown bolt while under test. This exhibit also shows how the sheet was heated with a gas flame and the method used to obtain pyrometer readings and the deflection of heated sheets.

Fig. 7 is a drawing and curve reading showing eight types of crown bolts tested and the temperature of the plate and the amount of deflection in the sheet at the time each bolt failed.

The data given in Tables 1 to 8, inclusive, are the result of sixteen tests made with eight types of crown bolts. These tests show how much longer the button-head type crown bolts hold compared with the hammered-head type when the crown sheet is overheated. It also demonstrates an important and interesting feature regarding the holding power of the taper on hammered-head crown bolts. In this particular test, your attention is directed to tests Nos. 3 and 4, compared with tests Nos. 5, 6, 7 and 8. It demonstrates that the 1¼-inch diameter straight hammered-head crown bolt will support the crown sheet when overheated under approximately the same conditions as the 1⅛-inch diameter hammered-head crown bolt with 1½- or 2-inch taper in 12 inches. Generally speaking, it can be said that increased diameter of the crown bolt at the crown-sheet fit gives approximately the same holding power in a heated sheet as increased taper on the crown bolt.

It is well known that no crown bolt will hold up the crown sheet when it becomes overheated beyond certain limits. Tests, however, prove that the average size button-head type of bolt will support the crown sheet under 200 pounds working pressure until it develops a temperature of 1364 degrees F. while the average hammered-head type bolt will fail under 1237 degrees temperature. It is questionable whether we want the sheet to hold until the higher temperatures are reached or whether it should fail at a lower temperature.

This question should be debated on the convention floor because there have been some bad crown sheet failures with both types of bolts.

As shown above, a large majority of the locomotives in use today are equipped with hammered-head crown bolts.

The following reasons are generally given for favoring riveted head crown bolts with taper:

1.—They have sufficient strength.

2.—The bolt with taper can be applied more economically.

3.—They tighten in the sheet more readily and are not as easily stripped.

Table 8

TEST NO. 15—1¼-INCH DIA. ¾-INCH TAPER 1 27/32-INCH BUTTON

Time Min.	Pyrometer Reading	Actual Temp. Degrees F.	Deflection Readings	Actual Deflection Inches
0	2.00	100	0.14	0
1	11.30	409	0.14	0
2	18.80	653	0.15	0.01
3	25.50	871	0.16	0.02
4	30.70	1006	0.19	0.03
5	34.50	1152	0.25	0.11
6	37.20	1233	0.41	0.27
7	38.80	1281	0.56	0.42
8	40.20	1321	0.68	0.54
9	41.40	1355	0.77	0.63
10	42.50	1387	0.86	0.72
11	43.10	1404	0.93	0.79
12	43.40	1431	0.98	0.84
13	43.50	1415	1.03	0.89
14	43.60	1418	1.10	0.96
15	43.80	1421	1.21	1.06
15½	43.90	1427	1.32	1.18

TEST NO. 16—1¼-INCH DIA. ¾-INCH TAPER—1 15/16-INCH BUTTON

Time Min.	Pyrometer Reading	Actual Temp. Degrees F.	Deflection Readings	Actual Deflection Inches
0	2.00	100	0.10	0
1	12.60	451	0.11	0.01
2	19.80	686	0.11	0.01
3	26.00	887	0.13	0.03
4	30.60	1033	0.14	0.04
5	34.00	1138	0.20	0.10
6	36.30	1208	0.32	0.22
7	38.10	1260	0.46	0.36
8	39.10	1290	0.56	0.46
9	40.00	1316	0.64	0.54
10	40.60	1332	0.71	0.61
11	41.20	1350	0.76	0.66
12	41.70	1364	0.82	0.72
13	42.00	1373	0.86	0.76
14	42.40	1384	0.91	0.81
15	42.60	1391	0.95	0.85
16	42.70	1393	0.98	0.88
17	42.80	1396	1.02	0.92
18	42.90	1399	1.07	0.97
19	42.90	1399	1.13	1.03
20	42.60	1391	1.28	1.18
20.2	42.50	1387	1.32	1.22

Average Curve Readings Temp.	Average Curve Readings Deflection
0	0
100	0
200	0
300	0
400	0.005
500	0.009
600	0.012
700	0.016
800	0.020
900	0.024
950	0.026
1000	0.028
1050	0.039
1100	0.063
1150	0.110
1200	0.192
1250	0.320
1300	0.483
1350	0.638
1400	0.833
1408	1.198

Average Time 17.85 Min.

4.—They can be securely fitted in the crown sheet regardless of degree of radius.

5.—They give less trouble leaking than button-head bolts.

6.—Riveted-head-type bolts are not damaged from calking as easily as the button-head type.

7.—They can be manufactured at less cost than the button-head type.

8.—Savings effected by one tap and reamer accommodating several sizes of bolts reduce tool expense.

9.—Because of the taper accommodating different sizes, a smaller stock of hammered-head bolts are required to be carried.

10.—They permit a more even head condition on the fire side of the crown sheet which reduces the amount of clinker and dirt adhering to the sheet.

The data received by the committee in answer to the questionnaire disclosed the fact that the majority of the railroads in this country have changed their standard from the button-head crown bolt to the taper hammered-head crown bolt type.

This fact, in the opinion of the committee, is not conclusive evidence that the taper hammered-head crown bolt is the better type of crown bolt for locomotive boilers, for the reason that much trouble has been experienced on some of the larger railroads with taper hammered-head crown bolts leaking and failing while in service. A number of cases were reported where this type of bolt has passed out of the sheet after the bolts broke due to deterioration of the threads on the crown bolt and in the crown sheet hole. It is felt by many boilermakers that abrupt tapers are responsible for a great deal of the crown bolt trouble experienced. One of the large Eastern railroads, after having used taper hammered-head crown bolts for the past 12 years, is applying button-head crown bolts with small taper for the purpose of determining which is the better type.

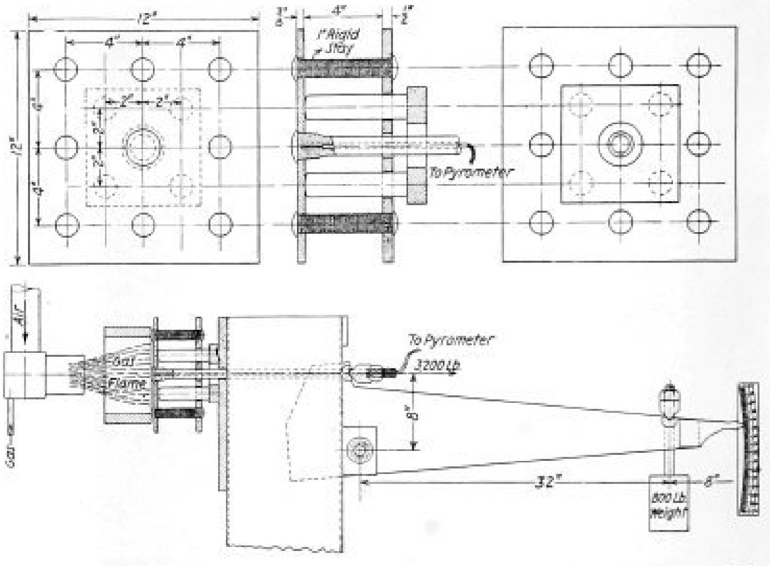

Fig. 6.—Plan view of testing machine and manner of loading the crown bolts

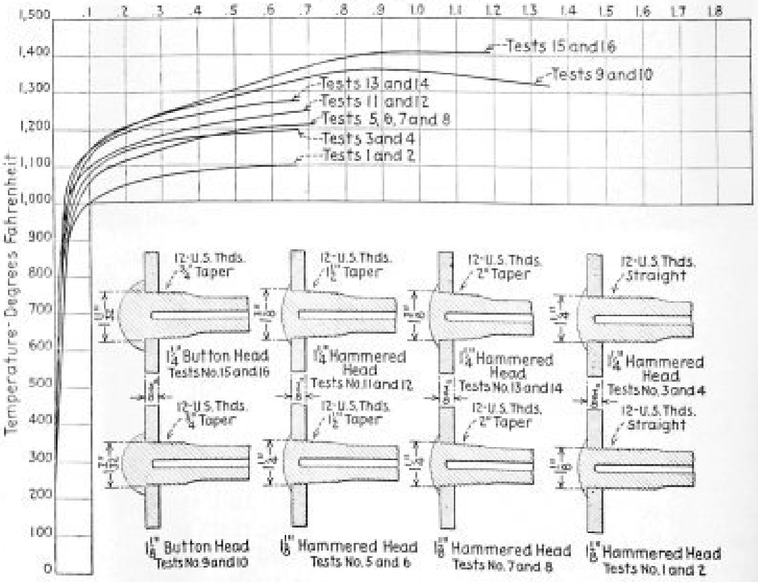

Fig. 7.—Eight types of crown bolts tested and temperature-deflection curves of plate

The committee recommends that all members of the association and others study the report of the committee carefully; compare the same with conditions found on their railroad; write comments and forward same to the secretary of the association at the earliest possible date. These comments will be sent to the chairman of the topic committee for further study and definite recommendations for the 1932 convention.

This report was prepared by a committee composed of Ira J. Pool, chairman, E. S. Fitzsimmons and R. V. Moore.

"Things Are Getting Better in This Country"

From a statement made by General Charles G. Dawes, president of the Reconstruction Finance Corporation, before the Ways and Means committee of the national House of Representatives at Washington, D. C., on April 21:

"Down here in the Reconstruction Finance Corporation we are in a position to know that things are getting better in this country—in a damn sight better position to know what is going on than are those fellows in that security peanut stand in Wall Street.

"Bank failures are falling off. The banks are beginning to make loans again. It makes no difference what Wall Street thinks down there where that peanut gambling is going on.

"You can take it from me we are approaching prosperity. The mass attitude of the people has changed from pessimism to optimism, but, of course, it will take time to realize the full results. Business is a ponderous machine and takes time to get in motion."

Weld Tests

(Continued from page 93)

A good looking weld, however, is not necessarily a good weld. A very smooth level surface, or perfectly spaced ripples, is often suspicious for the reason that, in such instances, too much time and attention has been given to obtaining a good appearing weld with the result that attention has been diverted from the important phase, that of obtaining proper penetration.

Other non-destructive tests include the X-ray and Gamma-ray tests which require more or less elaborate equipment for application. On the other hand, one of the best non-destructive methods for testing fillet welds is the stethoscope test, the object of which is to determine, by means of the tone a weld gives off when tapped with a hammer, whether the weld metal is sound or has cracks or holes. Good weld metal gives a ringing sound and poor metal a characteristic flat tone. In order to more readily identify these sounds, an ordinary physician's stethoscope is used for listening, the only extra precaution being to cap it with a soft rubber cap to prevent the application of the stethoscope to the metal from affecting the sound produced.

A great deal of progress has been made in the development of new non-destructive weld tests but many skeptics will not consider the use of welding unless non-destructive tests are developed, which, when applied to every weld, infallibly detect those which are unsatisfactory. These skeptics are apt to forget, however, that infallible non-destructive tests are almost unknown in any field of metal joining.

Hartford Steam Boiler Company Opens New Building

Hartford, the insurance city, has a new addition to the notable group of buildings housing the insurance companies. This is the beautiful new structure of The Hartford Steam Boiler Inspection and Insurance Company, which has just been completed.

In the early months of 1931 announcement was made by W. R. C. Corson, president of The Hartford Steam Boiler Inspection and Insurance Company, that the company would erect a needed new building at this time as a contribution toward helping the employment situation. Development of the plans was very rapidly pushed forward by the architect, Carl J. Malmfeldt of Hartford, and work was started in May, 1931. The building has many special features and has been designed specifically for the work of the company, including an engineering and chemical laboratory of most modern type which occupies a large part of the lowest floor.

This new building of the Hartford Steam Boiler is simple and beautiful in design and proportion.

Although the new plant is not a large one, as modern buildings go, it is an efficient structure designed along modern classical lines that suggest the nature of the company's business. Tall fluted pilasters adorn both the front and the side, with appropriately designed aluminum spandrels. In relief on these spandrels are a gear and an engine governor, symbolic of the power equipment the company insures. These spandrels were designed by Gorham, the silversmith, and serve to set off the beautiful texture of the Indiana limestone of which the building is made.

An aluminum railing leads to the front door, above which is an aluminum replica of the company seal, showing the famous old "Comet," a locomotive which ran between Hartford and New Haven in the days of 1866 when the company was founded. Throughout the decorative scheme, strips of aluminum, and window and door frames of the same material add a touch of modernism.

The building has ample space not only for the present requirements of the company but for future growth. The Hartford Steam Boiler Inspection and Insurance Company is not only the oldest company in its line in the United States, but it is the largest in the world. It is the only company which is devoted exclusively to the writing of engineering insurance and writes approximately half of all such coverage in the United States.

Cornelius M. Walsh, president and general manager of the Falls Hollow Staybolt Company, Cuyahoga Falls, Ohio, died at his home in that city on March 25 at the age of 68 years. Mr. Walsh was born at Cuyahoga Falls and spent his entire life there. He attended schools of his native town and began his business career as an employee of the lumber and planing mill establishment of Howe & Company. Mr. Walsh later bought the business and operated under the name of the Walsh Lumber Company of which he was president. He also was president of the Walsh Paper Company and Walsh Milling Company.

The Boiler Maker

VOLUME XXXII

NUMBER 5

PUBLISHED Monthly by the Simmons-Boardman Publishing Company, 34 North Crystal Street, East Stroudsburg, Pa., with Executive Offices at 30 Church Street, New York City.

Samuel O. Dunn, Chairman of Board. Henry Lee, President. C. R. Mills, Vice-President. L. B. Sherman, Vice-President. Roy V. Wright, Vice-President and Secretary. F. H. Thompson, Vice-President. E. T. Howson, Vice-President. F. C. Koch, Vice-President. John T. DeMott, Treasurer. George Slate, Vice-President and Business Manager.

Chicago: 105 W. Adams Street. Washington, D. C.: 17th and H Streets, N. W. Cleveland: Terminal Tower. San Francisco, Cal.: 215 Market Street.

Subscription price in United States and possessions, 1 year, $2.00, 2 years, $3.00; Canada, including duty, 1 year, $2.50, 2 years, $4.00; foreign countries, 1 year, $3.00, 2 years, $5.00. Single copies, 35 cents each.

Request for change of address should reach us on or before the 15th of the month, preceding the issue with which it is to go into effect. It is difficult, and often impossible, to supply back numbers to replace those undelivered through failure to send advance notice. In sending us change of address, please be sure to send us your old address as well as the new one.

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Contents

Communications

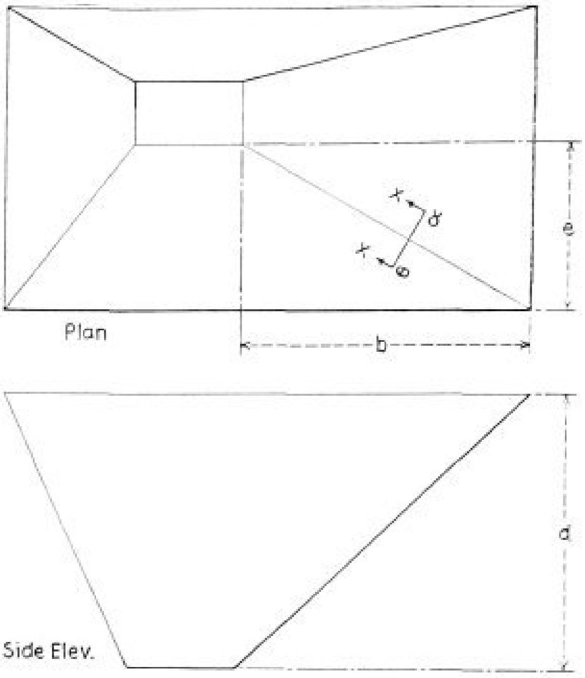
Bevel of Corner Angle

TO THE EDITOR:

In your February issue there was submitted a layout method for determining the valley angle of an irregular hopper. In view of the fact that this gives only an angle which may be protracted, it may be of interest to other readers in presenting a method whereby the angle may be calculated.

The accompanying sketch shows the plan and side elevation of an irregular hopper. To find the true angle at $x - x$, we have at the right, looking in the direction of the arrow heads, one part of the total angle which we shall designate as α. At the left we have the remaining part of the total angle or θ.

$$\text{Tangent } \theta = \frac{e\sqrt{c^2 + b^2 + d^2}}{bd}$$



Plan and side view of irregular hopper

$$\text{Tangent } \alpha = \frac{b\sqrt{c^2 + b^2 + d^2}}{cd}$$

The sum of the two parts is then equal to the total angle.

In considering the remaining angles, a similar procedure is followed.

Coatesville, Pa. AXEL F. JOHNSON.

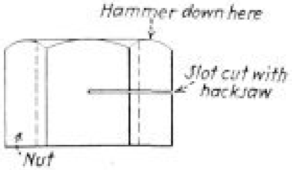
EDITOR'S NOTE.—R. A. Davis, chief draftsman, Chicago Bridge & Iron Works, Birmingham, Ala., submitted a similar solution to this problem, giving, however, a formula for the total angle. Using the foregoing sketch, this formula would be:

$$\text{Tangent total angle} = \frac{d\sqrt{c^2 + b^2 + d^2}}{cb}.$$

Fitting Nuts on Worn Bolts

TO THE EDITOR:

Here is a kink for making a nut fit a bolt with a loose or bad thread. Often the completion of a repair job has taken so long that there is insufficient time to obtain new nuts where old nuts are too loose. As a means for temporarily remedying this trouble, take a hack saw and slot halfway through the nut, as shown by the double line in the sketch. Then take a large hammer and pound down on the top of the nut, as shown by the arrow. This closes the thread and causes the nut to catch some threads at the bolt when turned on. Although the practice is not recommended it serves the purpose until new bolts may be obtained.



Nut for worn bolts

Montreal, Canada. JAMES WILSON.

Questions and Answers Pertaining to Boilers

This department is maintained for the purpose of helping those who desire assistance on practical boiler shop problems. Inquiries should bear the name and address of the writer. Anonymous communications will not be considered. The identity of the writer, however, will not be disclosed unless the editor is given special permission to do so.

By George M. Davies

Stresses in Diagonal Riveting

Q.—I am anxious to obtain some information regarding the proper determination of stresses involved in the diagonal riveting of liners or diagonal seams of boilers, as illustrated in Figs. 1 and 2.—F. C. H.

A.—The efficiency along the line *A-A* of the liner shown in Fig. 1 is obtained as follows:

To find the efficiency through the rivets *A-A*, the efficiency must first be figured as a longitudinal line not considering the angularity of the line *A-A* with a vertical center line of the shell.

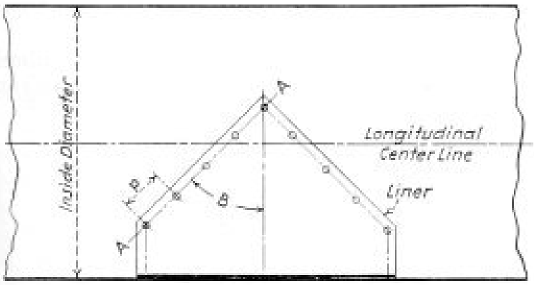


Fig. 1.—Calculating seam efficiency

$$\text{Efficiency (without considering angularity)}, E = \frac{p - d}{p}$$

where p = pitch of rivets, inches.
d = diameter of rivet hole, inches.

Providing the line of rivets *A-A* were in a horizontal line, the above formula would give the efficiency along the line *A-A*. This, however, is not the case, and the number of degrees in the angle *B* must be found. This can be determined by trigonometry or the use of a protractor.

After finding the number of degrees in angle *B*, a factor can be determined which allows additional strength due to the angularity of the line of rivets *A-A*.

The following formula gives the factor allowed due to angularity of the line of rivets *A-A*:

$$F = \frac{2}{\sqrt{3 \times \text{sine } B^2 + 1}}$$

By multiplying the efficiency obtained without considering the angularity, by the factor allowed due to the angularity, the actual efficiency along the line of rivets *A-A* is obtained.

Angular or actual efficiency = $E \times F$

In summing up, the entire formula for the efficiency along the line of rivets *A-A* can be expressed as follows:

$$\text{Angular efficiency} = \frac{2(p - d)}{p\sqrt{3 \times \text{sine } B^2 + 1}}$$

The efficiency of the diagonal seam shown in Fig. 2 is obtained in the same manner. First obtain the efficiency considering the seam as a longitudinal seam. Then determine the factor for the angularity using the same formula as for the liner in Fig. 1. The actual or angular efficiency is equal to the efficiency of the seam multiplied by the factor of the angularity.

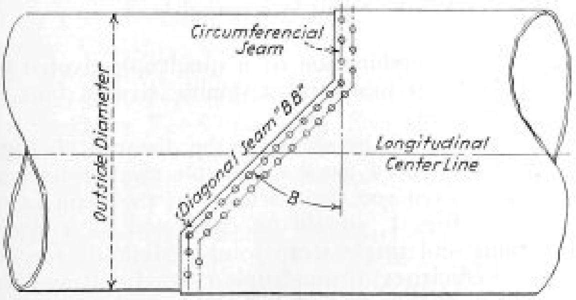


Fig. 2.—Diagonal seam

Area of Head Segment

Q.—Will you please tell me through the Questions and Answers Department of The Boiler Maker the simplest and most accurate way of calculating the area of a segment of a return tubular boiler, and the spacing of the braces, as the boiler maker in the shop would do it. How far do the flues stiffen the segment above the top row and how far does the flange stiffen the head above the flues?—H. H.

A.—The area of a segment of a head to be stayed shall be the area enclosed by lines drawn 2 inches from the tubes and at a distance *d* from the shell as shown in Fig. 1. The dimension of 2 inches is constant for all sizes of boiler heads, but the distance *d*, which represent that portion supported by the flange is dependent on the pressure and on the thickness of the head. The value of *d* may be determined from the following formula as given in the A.S.M.E. Boiler Code:

$$d = \frac{5 \times T}{\sqrt{P}}$$

where d = unstayed distance, in inches, from the shell
T = number of sixteenths of an inch in thickness of head (thus $T = 8$ for ½ inch)
P = maximum allowable working pressure pounds per square inch

Table 1.—Compiled by the Hartford Boiler Inspection and Insurance Company, gives the values for *d* in

Table 1.—Values of *d* for different plate thicknesses and pressures

Working Pressure pounds per sq. in.	Thickness of Head, inches 3/8"	7/16"	1/2"	9/16"	5/8"	11/16"	3/4"	13/16"	7/8"	15/16"	1"
50	4.24	4.95	5.66	6.36	7.07	7.78	8.49	9.19	9.90	10.61	11.31
60	3.87	4.52	5.16	5.81	6.46	7.10	7.75	8.39	9.04	9.68	10.33
70	3.59	4.18	4.78	5.38	5.98	6.57	7.17	7.77	8.37	8.96	9.56
80	3.35	3.91	4.47	5.03	5.59	6.15	6.71	7.27	7.83	8.39	8.94
90	3.16	3.69	4.22	4.74	5.27	5.80	6.33	6.85	7.38	7.91	8.43
100	3.00	3.50	4.00	4.50	5.00	5.50	6.00	6.50	7.00	7.50	8.00
110	2.86	3.33	3.81	4.29	4.77	5.24	5.72	6.20	6.67	7.15	7.63
120	2.75	3.21	3.66	4.12	4.58	5.04	5.50	5.96	6.42	6.88	7.33
125	2.68	3.13	3.58	4.03	4.47	4.92	5.37	5.81	6.26	6.71	7.16
130	2.63	3.07	3.50	3.95	4.38	4.82	5.26	5.70	6.14	6.57	7.00
140	2.53	2.95	3.38	3.80	4.22	4.64	5.07	5.49	5.91	6.33	6.76
150	2.45	2.85	3.26	3.67	4.08	4.49	4.90	5.31	5.71	6.12	6.53
160	2.37	2.77	3.16	3.56	3.95	4.35	4.74	5.14	5.53	5.93	6.32
170	2.30	2.68	3.06	3.44	3.83	4.22	4.60	4.98	5.37	5.75	6.13
180	2.23	2.60	2.98	3.35	3.72	4.10	4.47	4.84	5.21	5.59	5.96
190	2.17	2.54	2.90	3.26	3.63	3.99	4.35	4.72	5.08	5.44	5.80
200	2.12	2.47	2.83	3.18	3.54	3.89	4.24	4.59	4.95	5.30	5.66

accordance with the foregoing formula, as required for different head thickness and working pressures.

The A.S.M.E. Boiler Code also prescribes that the outside radius of the flange of the head may be used for *d*, if such radius does not exceed eight times the thickness of the head. To illustrate the meaning of the outside radius of the flange Fig. 1(b) is given. In this sketch the radius *r*, may be used for *d*, provided this distance does not exceed 8 × *t*, as shown in the figure. The greater value for *d* as found by the two rules given may be used.

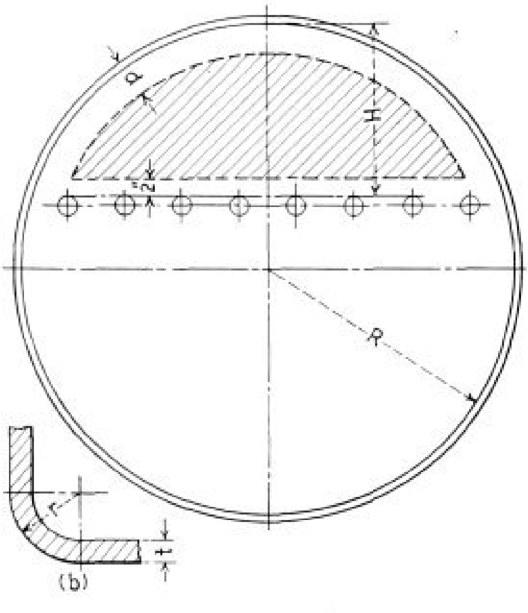


Fig. 1.—Area to be stayed

To determine the area of the segment, the following formula is used in calculating the area of the segments:

$$A = \frac{4\,(H - d - 2)^2}{3} \sqrt{\frac{2\,(R - d)}{H - d - 2} - .608}$$

where A = area to be stayed, square inch.
H = distance from tubes to shell, inches.
d = unstayed distance from shell to boundary of segment, inches.
R = radius of boiler head, inches.

Example: Given a 72-inch head, in which $H = 23\frac{1}{2}$ inches, $d = 3$ inches, and $R = 36$ inches, determine the area of the segment.

$$A = \frac{4\,(23.5 - 3 - 2)^2}{3} \sqrt{\frac{2\,(36 - 3)}{23.5 - 3 - 2} - .608} = 784 \text{ square inches.}$$

The staying of the upper segment of the head is accomplished by the use of head to head, through, diagonal, crowfoot, or gusset stays also as stipulated by the A.S.M.E. Boiler Code. The braces should be spaced as uniformly as possible and each stay computed for the load that it carries.

Efficiency of Conical Seams

Q.—Kindly advise me how to determine the efficiency of a riveted seam used in the construction of tanks or boilers having cone ends.—S. L. W.

A.—The efficiency of seams for conical-shaped tanks or boilers are computed in the same manner as similar seams for round tanks or boilers.

In the case of the conical-shaped digester submitted with the question, Fig. 1, the seam is a longitudinal seam and is a combination of a quadruple-riveted butt and double strap joint and a double riveted butt and single strap joint.

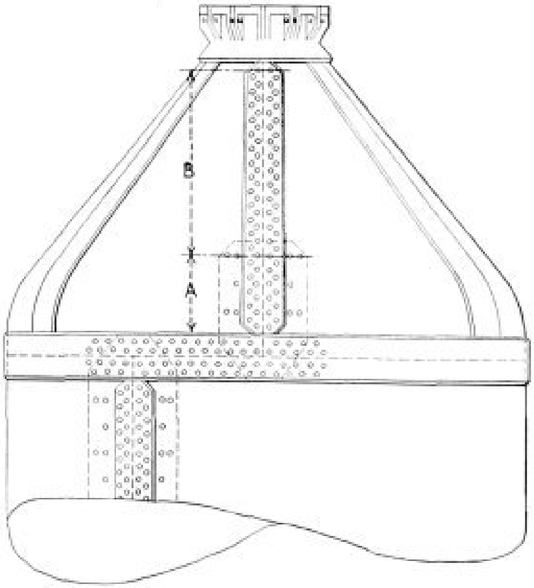


Fig. 1.—Seams in a conical head

The efficiency of the seam for the distance *A*, Fig. 1, should be computed as a quadruple-riveted butt and double strap joint and the efficiency of the seam for the distance *B*, Fig. 1, should be computed as a double-riveted butt and single strap joint. Methods for computing the efficiency of quadruple-riveted butt and strap joint and a double-riveted butt and single strap joint may be found in most structural handbooks.

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States and Cities That Have Adopted the A.S.M.E. Boiler Code

States

Arkansas
California
Delaware
Indiana
Maryland
Michigan
Minnesota
Missouri
New Jersey
New York
Ohio
Oklahoma
Oregon
Pennsylvania
Rhode Island
Utah
Washington
Wisconsin
District of Columbia
Panama Canal Zone
Territory of Hawaii

Cities

Chicago, Ill.
Detroit, Mich.
Erie, Pa.
Evanston, Ill.
Houston, Tex.
Kansas City, Mo.
Los Angeles, Cal.
St. Joseph, Mo.
St. Louis, Mo.
Scranton, Pa.
Seattle, Wash.
Memphis, Tenn.
Nashville, Tenn.
Omaha, Neb.
Parkersburg, W.Va.
Philadelphia, Pa.
Tampa, Fla.

States and Cities Accepting Stamp of the National Board of Boiler and Pressure Vessel Inspectors

States

Arkansas
California
Delaware
Indiana
Maryland
Minnesota
Missouri
New Jersey
New York
Ohio
Oklahoma
Oregon
Pennsylvania
Rhode Island
Utah
Washington
Wisconsin
Michigan

Cities

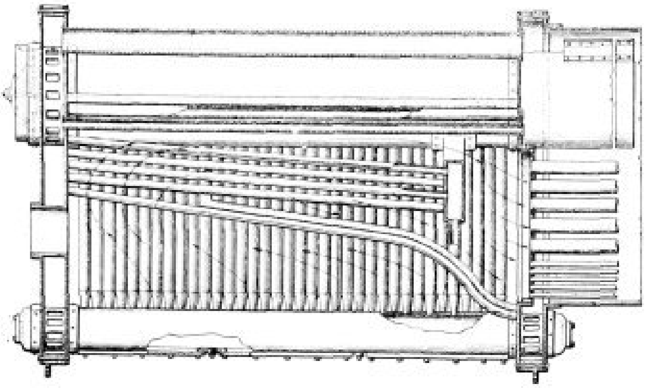
Chicago, Ill.
Detroit, Mich.
Erie, Pa.
Kansas City, Mo.
Memphis, Tenn.
Nashville, Tenn.
Omaha, Neb.
Philadelphia, Pa.
Parkersburg, W. Va.
St. Louis, Mo.
Scranton, Pa.
Seattle, Wash.
Tampa, Fla.

Selected Boiler Patents

Compiled by Dwight B. Galt, Patent Attorney, Washington Loan and Trust Building, Washington, D. C. Readers wishing copies of patents or any further information regarding any patent described, should correspond directly with Mr. Galt.

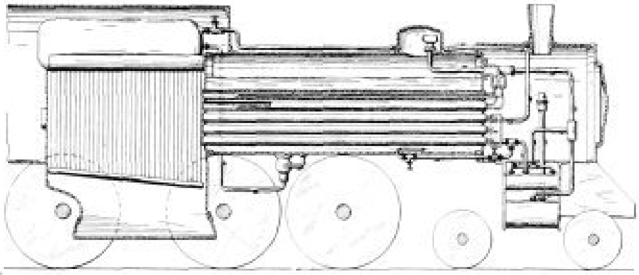
1,775,464. BOILER. JULIUS KINDERVATER, OF NEW YORK.

Claim.—In a boiler, front and rear water legs, a pair of upper drums and a pair of lower drums communicating with said water legs, vertical tubes connecting each upper drum with a lower drum, a header disposed between the water legs and communicating with both upper drums, and horizontal tubes arranged beneath the upper drums and in a plane between said upper drums and the lower drums and connecting the rear water leg with said header. Twenty-four claims.



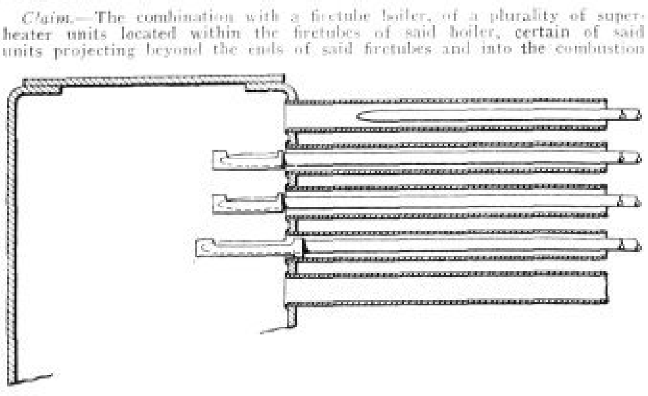
1,783,842. MULTI-PRESSURE BOILER INSTALLATION. OTTO H. HARTMANN, OF KASSEL-WILHELMSHOHE, GERMANY, ASSIGNOR TO SCHMIDT'SCHE HEISSDAMPF-GESELLSCHAFT.

Claim.—Method of initially heating up multi-pressure boiler installations having superheating devices for superheating the steam generated in the respective pressure sections of the boiler, which consists in leading steam generated in a high-pressure section of the boiler installation through a superheater heated by the same furnace as the boiler itself and thence to the water space of a low-pressure section of the boiler installation. Seven claims.



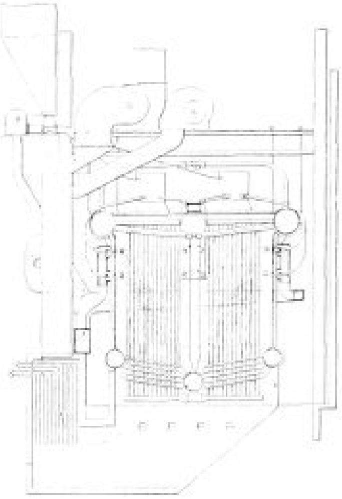
1,799,669. SUPERHEATER FOR FIRETUBE BOILERS AND PROTECTIVE MEANS THEREFOR. JOHN A. BARNES, OF CHAPPAQUA, N. Y., ASSIGNOR TO THE SUPERHEATER COMPANY.

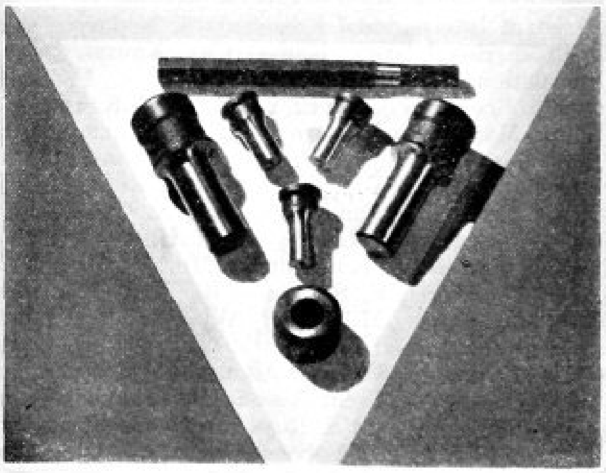
Claim.—The combination with a firetube boiler, of a plurality of superheater units located within the firetubes of said boiler, certain of said units projecting beyond the ends of said firetubes and into the combustion chamber of the boiler in order to insure a high degree of heating of said units, and a plurality of protective shields secured upon such projecting ends of said superheater units and extending lengthwise thereof, at least, into proximity to the ends of said firetubes, said shields covering the wall of said units on the side thereof facing the furnace while leaving the side of said wall facing away from the furnace uncovered. Five claims.



1,775,336. MULTIPLE-UNIT BOILER. WILFRED R. WOOD, OF LONDON, ENGLAND, ASSIGNOR TO INTERNATIONAL COMBUSTION ENGINEERING CORPORATION, OF NEW YORK.

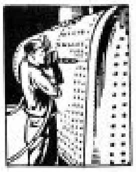
Claim.—In combination, in a boiler, two upper drums, two lower drums one under each of the upper drums, drumlike means located substantially midway between and on a different level than the two upper drums, tubes leading from the said means to each upper drum and defining the top of the combustion space for the boiler, upright substantially vertical tubes connecting each pair of upper and lower drums and defining sides of the combustion space, drumlike means located substantially midway of the lower drums and on a different level, tubes connecting the same with each of the two lower drums and defining the bottom of the combustion space, upright substantially vertical tubes defining the remaining sides of the combustion space connected for circulation, and upright substantially vertical tubes connecting the intermediate drumlike means and operating to split the combustion space into two independent combustion spaces, together with an outlet chamber common to both said combustion spaces and an offtake for the waste gases from said chamber. Two claims.





The Boiler Maker

Reg. U. S. Pat. Off.



Semi-Watertube Locomotive Firebox

The advantages to be realized by the use of watertube boilers in locomotives are being recognized today more than at any time during the past twenty years of development. The Emerson watertube firebox, which eliminates crown-sheet, radial and side waterleg staybolts, was developed and has been employed for a number of years by the Baltimore & Ohio Railroad. It is only recently, however, that engineers of this company have ingeniously developed a semi-watertube firebox which can be installed inside the conventional firebox of stayed construction, with or without a combustion chamber.

This development, which was made in connection with the program of modernization in effect by the Baltimore & Ohio for the past five years, results in a material addition to the heating surface of existing fireboxes. As fully described in this issue, one installation of the semi-watertube firebox accomplishes an increase of 32 percent in heating surface and 32 percent in evaporation, as well as an increase in the circulation in the side waterlegs of the boiler.

Coming at a time when railroads are reluctant to purchase additional motive power, this development should appeal to those railroad officials who desire increased tractive force in existing equipment. In view of the advantages of this type of boiler improvement, it is to be hoped that further opportunity may be found for its development and practical application.

The Outlook Is not Hopeless

Among the hardest hit industries in the United States, railway transportation has been one of the first to recognize the inescapable fact that changed economic conditions will require entirely new forms of service to attract and hold business in the future. Reconstruction Finance Corporation credit support has been advanced to help railroads meet their present crisis, when relief has not been available from normal sources. Individual roads, such as the New York Central and Pennsylvania, have been granted loans to complete projects which otherwise would have remained unfinished for some time.

Aside from the immediate problem, however, the railroads are planning constructively for the future to meet competition from whatever source it may develop. Pick-up and store-door delivery may prove the solution to the motor-truck menace. Equitable legislation, controlling this form of carrier, will certainly react beneficially to the railroads. Modernized methods of handling both passengers and freight, in process of development, will help the situation. Increased train speeds, elimination of inefficient practices and the provision of more comfortable and convenient equipment and facilities combined with attractive rates, such as are now in effect on many roads, will do much to promote passenger travel.

From the supply side of the industry, the actual volume of business now going into operation and maintenance of the railroads is not fully realized. Even with a large part of their equipment idle, the railroads in the first three months of the year spent over $141,000,000 for supplies and equipment, which amounts to an average of $47,000,000 a month. Expenditures for the past two months have been at about the same level.

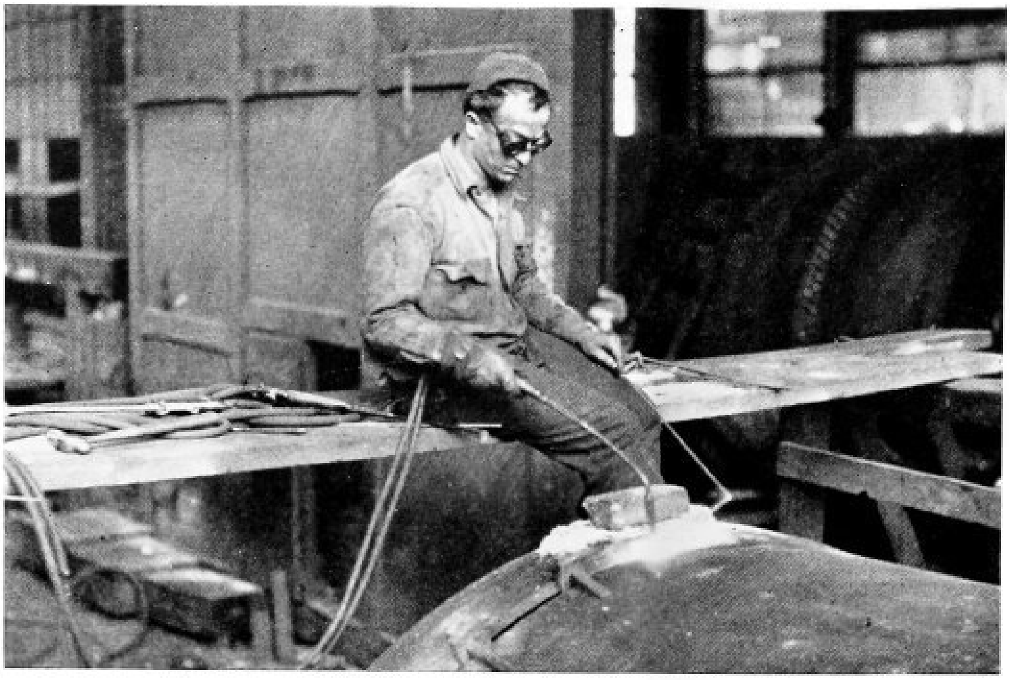
The situation is not nearly as hopeless as it appears on the surface. More constructive thought is being applied to the present problems of the railroads than ever before and from this effort will come future security.

Lincoln Arc-Welding Prize Awards

Peace-time activities of the Army and Navy as factors in industrial progress are seldom recognized as such. The award of first prize in the Second Lincoln Arc-Welding Competition to Lieutenant Commander H. N. Wallin, U. S. N., in collaboration with Lieutenant H. A. Schade, U. S. N., and second prize to Major G. M. Barnes, U. S. A., indicates that out of four hundred entries from all parts of the world their contributions to the welding art were considered most worthy.

The first prize paper described the construction of an arc-welded auxiliary naval vessel, and the second paper, the revolutionizing of ordnance equipment manufacture by welding. In all, forty-one prizes were awarded on papers covering practically the entire range of mechanical equipment and metal structures of all kinds.

Space does not permit the publication in this issue of the entire list of prize winners and details of their work. These will appear later. It is, however, opportune to point out the broad service to industry which the Lincoln Electric Company has made. Through the two competitions which this company has conducted, practical welding information in all lines has been made available for general use, which otherwise would have required years to bring out.



In the welding of pressure vessels tack-welds are supplemented by a series of wedge clamps

Controlling Expansion and Contraction in Welding*

The practical problems of compensating for the expansion of metal when it is heated by the welding blowpipe, and for the subsequent contraction when the heated metal cools have been subjects of constant study ever since the introduction of the oxy-acetylene process for joining metals. As a result, there have been developed certain precautions or methods of treatment that can be considered as fundamental. A thorough understanding of these constitutes an essential part of the fund of practical information which every welder must possess in order to work efficiently and to carry out intelligently any operation with the blowpipe.

The theory of the expansion and contraction of metals with changes in temperature is based on facts which are easily understood. Heat causes metal to expand and subsequent cooling necessarily means that the heated section will endeavor to contract into its original size and shape. Uneven heating will, therefore, cause uneven expansion, or uneven cooling will cause uneven contraction. Under such conditions stresses are set up within the metal. These forces must be alleviated, and, unless precautions are taken, warping or buckling of the metal takes place. Likewise, on cooling, if nothing is done to take up the stress set up by the contraction forces, further warping may result; or if the surrounding cool sections of the metal are too heavy to permit this change in shape, the stresses remain within the metal itself. Such stresses may cause cracking while cooling or may remain within the metal until further force is applied, as when the piece is put into use.

Due to the fact that sheet metal (1/8 inch and less in thickness) has such a large surface area per unit of weight, heat stresses tend to produce warping or buckling of the sheet. This and the contraction effect encountered on long seams are the principal points to be considered in sheet-metal welding.

The effect of welding a long seam (over 10 or 12 inches) is to draw the seam together as the weld progresses. The explanation is simple. The spot being welded is melted so rapidly that most of the expansion is taken care of in the molten metal. As the molten spot cools, it contracts and tends to pull the two edges of the seam together. The part previously welded, now cool and solid, resists this action with a consequent hinge effect that pulls the unwelded edges toward each other.

One way of overcoming this effect is to set up the pieces to be welded so the edges are nearly in contact at one end of the seam and separated at the other end a distance that varies according to the metal and its thickness. Some useful data have been compiled that indicate the average spacing allowance per foot of seam:

Steel	1/4 to 3/8 inch per foot
Brass and Bronze	3/16 inch per foot
Monel Metal	3/8 inch per foot
Aluminum	1/5 inch per foot
Lead	5/16 inch per foot
Copper	3/16 inch per foot

Sheet metal under $\frac{1}{16}$ inch in thickness is best handled by flanging the edges, tack-welding at intervals along the seam, and then welding.

The means most commonly employed to prevent buckling or warping of sheet metal during welding is to apply the principle of removing the heat from the base metal adjacent to the weld. In the case of flat seams, a heavy piece of metal such as a section of steel rail or heavy bar stock placed on either side of the seam

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* Published through the courtesy of *Oxy-Acetylene Tips.*

will effectively prevent the heat from spreading too far and will also tend to prevent movement of the parts by resisting the forces of expansion and contraction.

Exactly the same principle is applied in the construction of jigs and fixtures so universally used to hold sheet-metal parts in proper position for welding. The clamping action of the jig prevents undue movement of the parts, while the use of heavy sections in the jig at the desired points will effectively remove the heat from the base metal. In some cases, jigs are water-cooled to increase still further the ability to carry heat away rapidly.

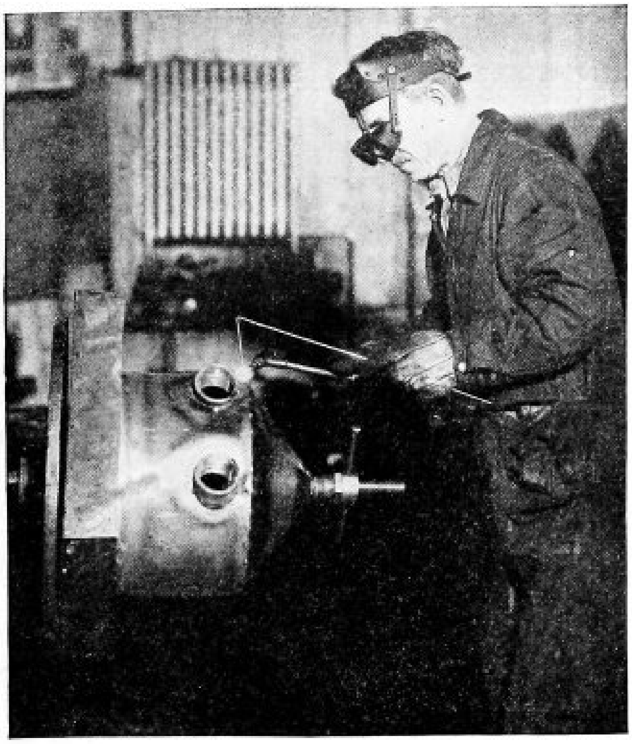
Heat may also be removed from metal adjacent to the welding zone by the use of wet asbestos cement along either side of each seam. In extreme cases a continuous flow of water can be played over the main body of the sheet.

In welding metal of plate thickness (above ⅛ inch), there is somewhat less tendency to buckle or warp, because the greater proportion of metal to surface area tends to diminish the rate of heat flow away from the welding zone. However, in welding long straight seams, the plate should be spaced about ¼ inch per foot to allow for the contraction of the seam.

For welding circumferential seams, such as are encountered in pipe, tanks and pressure vessels, it is obviously not possible to allow a progressively increased spacing around the entire seam. When pipe is lined up for welding, an even spacing of 3/32 to ¼ inch, depending on pipe size is left between pipe ends. Tack-welds are then made at specified intervals to hold the pipe in proper alinement during welding.

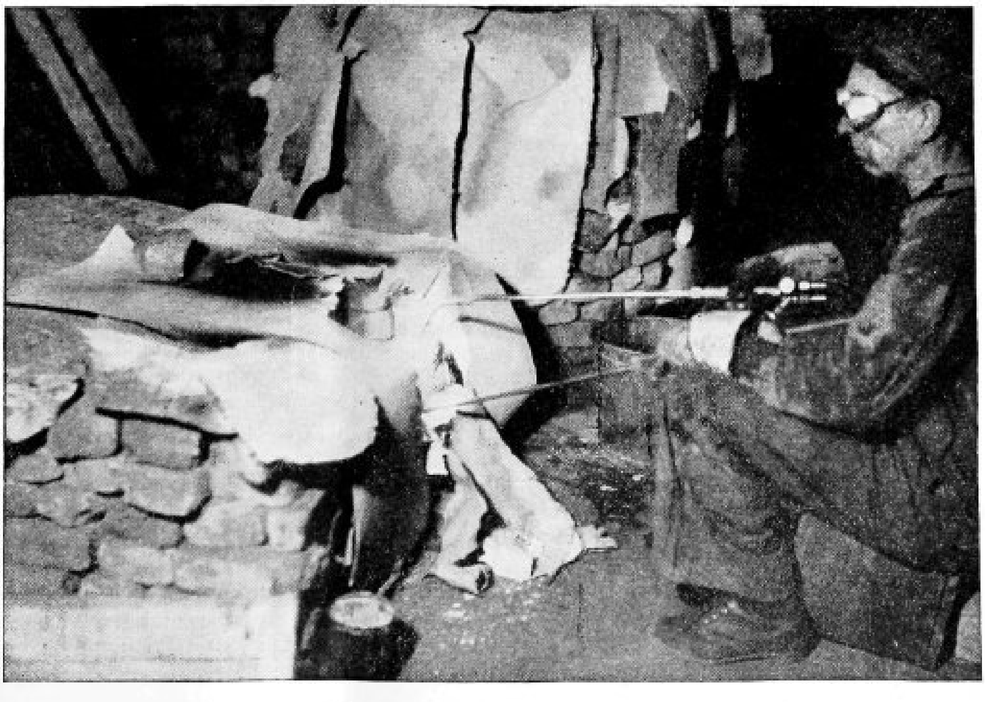
For large tanks and pressure vessels, the tack-welds are usually supplemented by a series of wedge clamps which are progressively removed just ahead of the weld and thus enable the operator to control the seam contraction. The exact procedure is determined largely by experience.

A note regarding cold-rolled shapes is of some importance. Because of the stresses that are in the metal shape from the cold working, prior to welding the worked portion of the shape should be heated to a good black heat. This will relieve this strain and there will then be no abnormal stresses to take into consideration while welding.

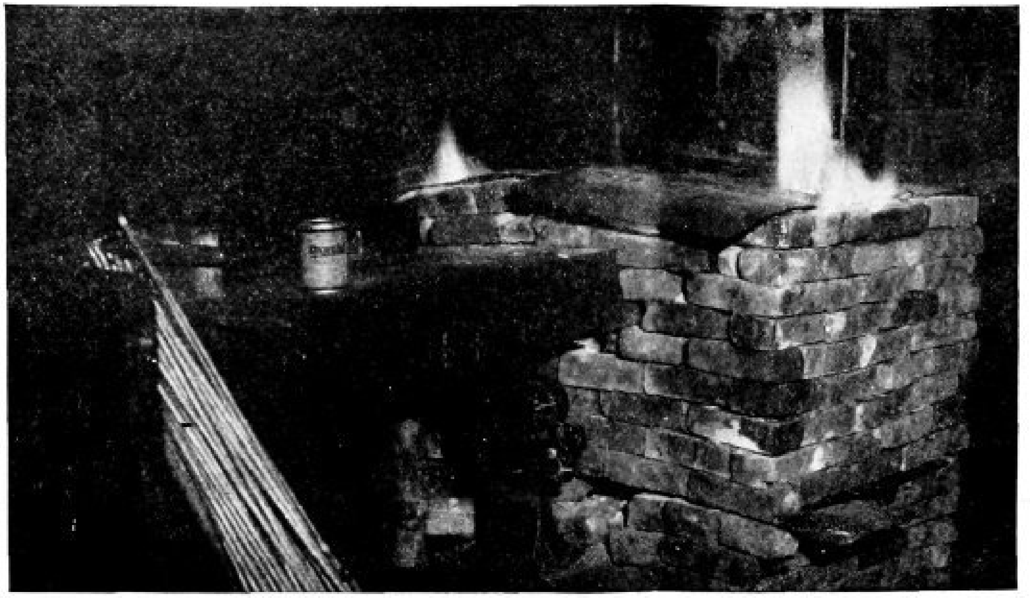


Tack welds facilitate alining sheet metal shapes

In considering heat effects while oxwelding gray iron castings with cast-iron welding rod, one point in particular should be firmly stressed: For general purposes it can be considered that iron castings are a great deal less elastic, less ductile, and, therefore, more rigid than sheet-metal or worked shapes. Furthermore, castings are generally of irregular shape, so that the heat from the blowpipe will spread unevenly through portions of the casting adjacent to the weld. Due to the greater amount of welding generally necessary on a casting,



Total preheating and careful annealing is best for cast iron welding



Partial preheating and expansion by jack will leave compression on weld

there will be more heat effect for which it will be necessary to compensate.

Careful preheating of the entire casting to a good red heat is by far the best means of assuring that expansion and contraction of the metal will be even throughout. After welding, a reheating and controlled slow cooling or annealing will assure the welder that all internal stresses are eliminated. It will also be remembered that preheating and slow cooling are essential factors in assuring proper gray iron structure in the welded casting.

Many castings are small enough so that a temporary furnace is unnecessary, and the welding blowpipe flame can be played over the whole casting to bring it to a red heat. In such cases the welding operation itself is of short enough duration to be completed before the casting loses its red heat. Here again slow cooling is essential to success for elimination of internal stress.

Again, a large casting may need only a minor repair such as building up a lug, boss, or tooth on a gear. The welding time for this type of work will be short and consequently other portions of the casting will remain cool so there is very little expansion or contraction stress to counteract. Should such work be done on a section of a casting that is irregular in shape it is well to direct the blowpipe flame toward the heavier portion so as not to overheat the lighter.

Often only local preheating is necessary or advisable. This is accomplished either in a temporary charcoal-fired furnace built only around the section to be welded, or by playing a preheating torch on the necessary part. Before using this type of stress elimination, it is well to ascertain that parts of the casting that are outside of the heated zone are such that they will not be affected by temporary distortion. That is, the outside section or sections should be of such a shape that when the expansion movement takes place this part of the casting can yield sufficiently without cracking.

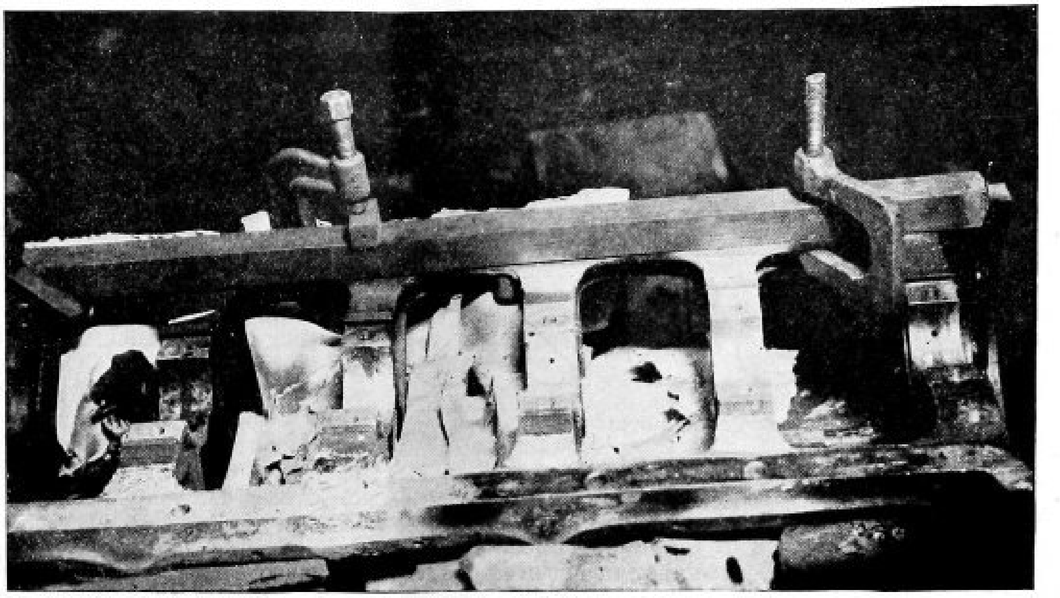
Prior to welding a fracture that extends in from the edge or from an opening of a casting, it is always advisable to drill a small hole through the casting about ½ to 1 inch beyond the visible end of the fracture. Should the crack start to run when the heat is applied, the crack will go only as far as the drill hole and stop there. This practice has often saved much work in the case of breaks in very heavy cast-iron sections where considerable heat was required.

Fractures of some internal part of a complicated casting require careful study prior to welding. Such fractures often occur on a section that can be considered as an inner rib, and that is held on either end by a heavier rim or surrounding metal that will check any expansion of the broken part. Total preheating of such a casting is, of course, the best solution to this problem, but this is often impracticable because of the part's location or size. Local preheating will then have to be done. In addition, and this is a most important point, two opposing sections of the enclosing part of the casting will also have to be preheated locally. If this were not done, when the welded section cooled and the metal tried to contract, the already cold outer sections would not give sufficiently, and the welded member would be subjected to a severe tension stress which might cause another fracture, either immediately or after the casting had been placed in service again. Occasionally it may be desirable to accomplish further expansion at necessary points by mechanical means such as wedges or jacks. In all such cases it is to be remembered that the sole object of such additional expansion of the metal is to leave the casting, when cooled, with a slight compressive strain on the welded part.

Castings that have a fracture that is entirely contained within a flat plane, such as a crack that does not extend to any edge or opening, usually must be entirely preheated, as there is no other way of relieving the stresses that would result from a localized heating by the blowpipe followed by contraction on cooling. There are, of course, special cases in some large castings where total preheating would be impractical, so that only a zone around the fracture is preheated.

Intricate casting often present special problems. Wherever previous experience is lacking, it is advisable to request the expert advice and assistance of an oxy-acetylene service operator.

Bronze-welding has been the means of eliminating a great deal of laborious and tedious work in repairing castings. Dismantling and extensive preheating of cast-iron parts is no longer necessary except in exceptional cases, because bronze-welding is done at a relatively low temperature. As expansion is controlled by the amount of heat applied, there is bound to be much less warpage from bronze-welding; and likewise on cooling, a casting does not have to contract nearly as much, thus eliminating a great deal of possible stress.



Aluminum and Monel castings should be supported during welding

In intricate shapes it may be advisable, if the work will require a large amount of heat, to preheat the casting locally. The extent of the preheating depends on the design of the casting and the nature of the break. Occasionally it may be necessary to preheat the entire casting, but usually this is unnecessary. In any case the preheating temperature is, of course, considerably lower than for cast-iron welding; just a black heat is all that is required.

Non-ferrous metals, whether in the form of sheet, plate, pipe or castings, require much the same consideration as outlined in the previous discussion, with such modifications as are made necessary by the specific characteristics of each metal.

Thus, copper conducts heat much more readily than does steel, and due allowance must be made for this in considering expansion and contraction.

Several non-ferrous metals and alloys, notably copper, aluminum, nickel and Monel metal, possess a peculiar property known as hot-shortness, which means that there is a condition of low tensile strength and absence of ductility in a temperature range below the melting point.

Monel metal, nickel and high-nickel alloys have a hot short range between the temperatures of 1450 and 1650 degrees F. Both above and below this hot short range, these metals regain their normal strength and ductility. Aluminum has a similar hot short range at a somewhat lower temperature.

Because of this property, particular attention must be given to expansion and contraction in welding these metals. Care should be taken to see that there are no undue stresses set up or existent while the weld metal and adjacent base metal are passing through the hot short range as the weld cools, in fact a very slight compression may be desirable. When preheating of these metals is necessary, as in welding castings, provisions should be made for properly supporting the part so that it will not distort or collapse in passing through the hot short range.

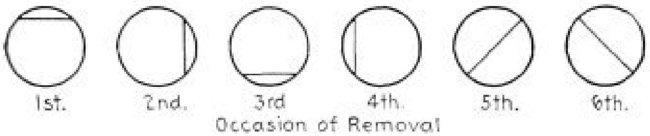
From this discussion of the proper handling of material for the elimination of the effects of expansion and contracting during welding, it becomes apparent that there are definite fundamental precautions to be taken. These are based on the fact that applied heat and subsequent cooling cause expansion and contraction stresses in metals. The general principle to be applied in each case is to so handle the work that when the weld is finished and the metal has finally cooled, the metal will have attained its original shape internally as well as externally; in other words, so that there will be no internal stresses remaining within the metal itself.

As is true of any subject of such broad scope, there are bound to be exceptions, special cases, and difficult problems which call for special treatment. When such conditions arise an oxy-acetylene service operator should be called for advice. For regular work, adherence to the suggestions outlined above and careful forethought concerning the best way to handle the expansion and contraction effect will aid materially in assuring the best possible results.

Locomotive Shop Records

Wherever a large number of engines are housed at a locomotive shop a comprehensive system of recording has to be adopted. While respecting the oft-repeated adage that "A railway cannot be run on paper," it is at least necessary to compile such records of the engine's performance as will assist in its future maintenance and working. The task of recording details of repairs, examinations, etc., usually devolves upon the foreman fitter, whose work in this direction may be materially lessened by having at his disposal a series of forms, drafted at headquarters' office, upon which specific information concerning each engine may be entered, the usual practice being to devote one form to the history of each engine rather than to enter information concerning various engines upon one form. By adopting such a method, not only can the approximate condition of any individual engine be quickly ascertained but also the task of compiling a proposal for the general shopping of the engine, for headquarters' use, is rendered easy.

The locomotive boiler, upon whose safe working much depends, naturally receives priority in the matter of detailed information. The names of important parts, such as tube sheets, crown sheets, side sheets, stays, lead plugs, smokebox sheets, and boiler barrel are tabulated on one large sheet, and a suitable abbreviation, indicative of its condition, placed against the appropriate part upon the date of examination. It is a general practice to make a monthly examination of the boiler when all components specified on the form will be examined, their condition, recorded as above, being initialed by the boiler maker making the examination. If a boiler inspector from headquarters visits the shop periodically, his report also will be noted upon the sheet and duly initialed. In addition, a tube plate diagram is kept. This is a drawing showing the disposition of the tubes in the class of engine concerned. When the latter receives a general repair at the shops it is usually fully re-tubed, and particulars of the brand and dimensions of the tubes put in are entered upon this diagram, which is forwarded to the running shed. When the boiler is subsequently cleaned for scale, some tubes are removed according to the position of the dirt, and a note is made on the diagram showing which tubes are withdrawn, by making a suitable mark on the circle representing the tube, the position of this mark differing according to whether it is the first or second, etc., occasion on which the boiler has been cleaned. The illustration shows this practice.



Method of marking boiler tubes

Thus, when determining which tubes to withdraw for cleaning on any occasion after the first, it is only necessary to glance at the diagram to make a decision. Particulars of date, brand, and mileage of tubes withdrawn, and also of those replaced, *i.e.*, whether new, second-hand, or the original ones, are also entered upon the form and initialed by the boiler maker. If any tube holes are plugged or bushed, a suitable indicating letter is written over the circle representing the tube hole concerned. A small card may also be used to record every occasion on which tubes are reported leaking and the time taken to re-expand. The number of entries on this card is a fairly reliable indication of the state of the boiler, since they will be frequent if the latter is getting dirty. A record of the dates of fitting brick arches and by whom fitted is also useful.

A practice which is sometimes instituted is that of using a duplicated form upon which necessary repairs are entered by the boiler inspector on the occasion of his visit to the locomotive shop. When the work is completed, one portion of the form is sent to headquarters as an intimation that the repairs specified thereon have been executed and to assist in the compilation of their records; the remaining portion is filed at the locomotive shop and constitutes a continuous record of boiler repairs.

In order to keep a thorough and comprehensive record of examinations carried out upon component parts and fittings of an engine, a large card, measuring perhaps 15 inches square and divided into two sections is employed. One section contains a tabulated list of all those fittings examined on a time basis; these will include boiler gage frames, trial taps, pressure gages, injectors, internal feed pipes, brake gear, sanding apparatus, carriage warming apparatus, etc., all of which are examined at either monthly or two-monthly, and, in the case of some fittings, three-monthly intervals, depending upon the mode of construction of the particular fitting, the locality in which the engine works, and the nature of the duties which it has to perform. Columns parallel and adjacent to this list are provided to record the date of examination, and the initials of the person responsible for carrying them out. The second division is similarly tabulated and contains a list of those components examined on a mileage basis such as connecting and coupling rods and brasses, pistons and valves, and motion generally, by-pass valves, lubricators, tender tanks, water pick-up gear, intermediate draw-gear, etc. Any relevant particulars concerning wear of piston valve liners, dimensions of piston rings, etc., may also be recorded here, and if at the discretion of the district locomotive superintendent, it is deemed necessary to make examination of fittings additional to those listed, space is provided for this contingency. The total mileage run by the engine from shops is also recorded on the card and subsequent mileage added in increments of 5000 miles as the engine completes them. Thus, when the foreman fitter receives intimation from the locomotive office that the engine has completed a standard mileage of, say, 5000 miles since last examination, it can be seen by a glance at the card what examination is necessary. Thus, if by-pass valves, wheels and tires, lubricators, etc., are examined at 5000 miles, connecting rods at 10,000 miles, pistons and valves at 30,000 miles, then it is apparent at once from the card what is the extent of the examination and the probable time the engine will be out of service.

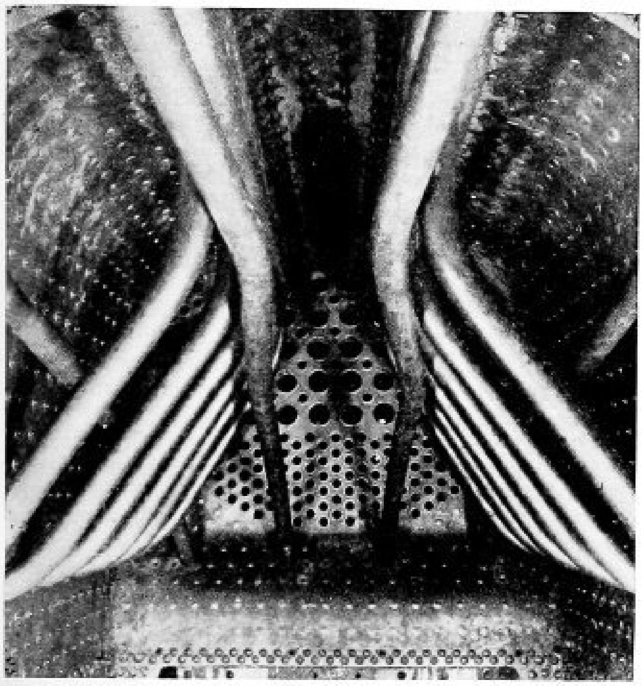
The detailed particulars of the condition of each wheel, tire, and axle of an engine and tender will occupy one side of a similar card, the reverse side of which may be used to record dates of examination of boiler, water spaces, and superheater flue tubes and elements (*i.e.*, at each washout); also of daily examinations of engines and tenders.

A useful adjunct to the cards described above is a large board, hanging in a prominent position in the foreman fitter's office and showing the position as regards repairs and examinations of every engine allocated to the shop. Small tablets, each bearing the number of an engine, are hung on suitable pegs screwed into the board which is subdivided into a number of columns. As the examinations on an engine are completed the appropriate tablet is moved to a new position on the board, indicating that fact and also the date completed. This arrangement affords an easy method of ascertaining the outstanding work at any time.

Records sent from headquarters to the locomotive shop and with which the foreman fitter may deal, include the "shopping" card on which are entered the dates of shopping of the engine, the nature of the repair, *i.e.*, whether general or "service" (a service repair being given, usually at an out-station, to enable an engine to run a further mileage before receiving general repairs), the date of fitting boiler to the engine named on the card, the date that the former is due for next internal examination, and any other relevant particulars. When a "service" repair has been performed at a station other than that at which the engine is allocated, the former will forward to the owning depot a certificate showing work done, which is invaluable in deciding what future repairs will be necessary; also, after general repairs, information on specified certificates is available for use by the foreman fitter, so that any undue wear is apparent upon examination.

Reports of engine casualties, breakdowns, etc., and records of the working of outside plant, and incidental appurtenances form veritable volumes. The work of the research office in drafting specific forms for the recording of information has, however, materially lessened the work of the foreman, who, consequently, can devote more time to the satisfactory supervision of repairs.—*The Locomotive.*

Baltimore & Ohio Develops Semi-Watertube Firebox

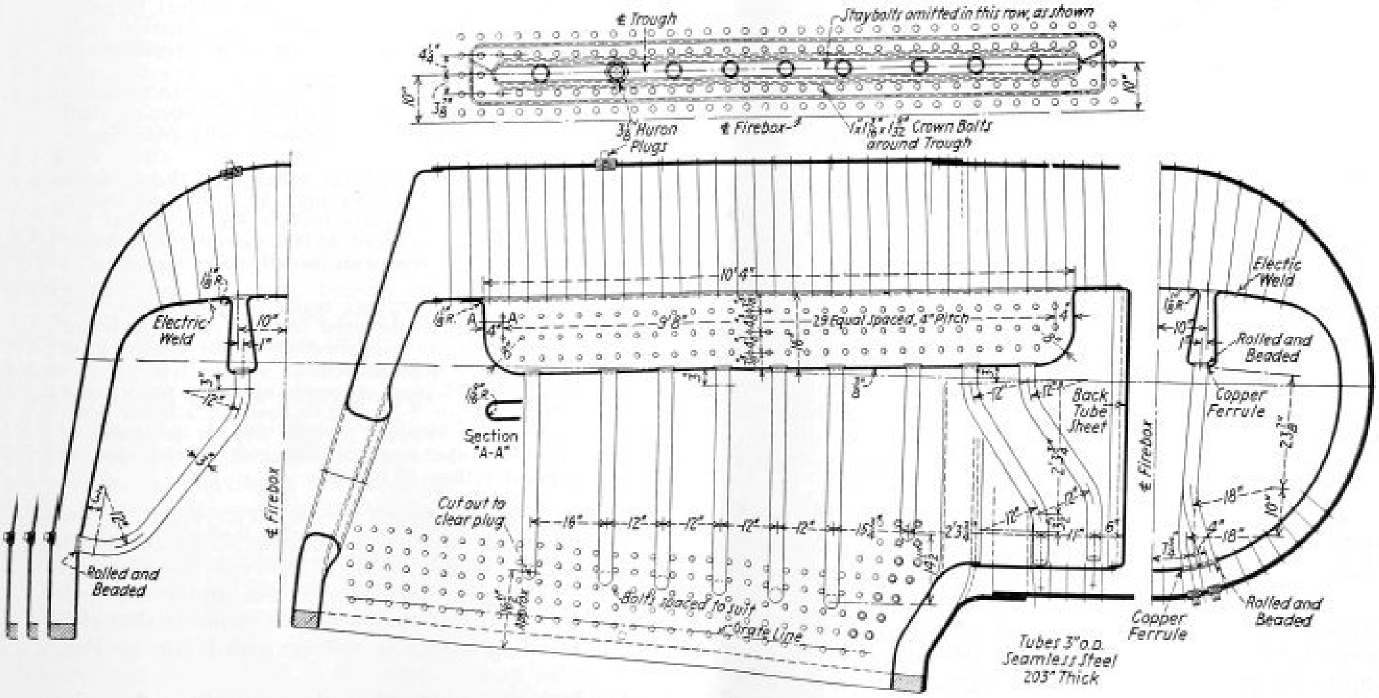


Firebox with semi-watertube installation showing the back flue sheet and combustion chamber

The Baltimore & Ohio Railroad has developed and applied to a number of its heavy 2-8-2, 4-6-2 and 4-8-2 type locomotives a semi-watertube firebox which can be installed inside the conventional firebox of stayed construction, with or without a combustion chamber. This development was in connection with a program of modernization which the B. & O. has had in effect for the past five years.

To modernize a locomotive built 10 or 12 years ago and increase the tractive force at high speeds it is necessary materially to increase the boiler horsepower. This is one reason why the B. & O. developed the Emerson watertube firebox which eliminates crown-sheet, radial and side waterleg staybolts, the only staybolts being in the throat sheet and backhead. This watertube firebox has been applied to a number of locomotives on which complete new back ends have been required as well as to two of the B. & O. test locomotives recently built by the Baldwin Locomotive Works.*

The semi-watertube firebox was developed to obtain a material addition to the heating surface from existing fireboxes. This firebox, shown in the drawing, consists of two troughs about 16 inches deep which extend longitudinally in the firebox and are welded to



Semi-watertube firebox developed by the Baltimore & Ohio and installed on a number of its locomotives

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*A description of the Emerson watertube firebox, which is referred to here, appeared in the September, 1931, issue of THE BOILER MAKER, page 230.

the crown sheet. The bottom of each trough is connected to the side waterlegs of the firebox by a number of circulating tubes. These tubes are rolled and beaded in both the side sheets and the troughs.

This semi-watertube firebox effects a marked increase in direct firebox heating surface. The heating surface of the 2-8-2 type locomotive on which the first installation was made was originally 256 square feet. With the semi-watertube firebox, an additional 74 square feet of heating surface was acquired, making a total of 330 square feet. The former firebox, by Cole's ratios, evaporated 14,056 pounds of water per hour, while the semi-watertube firebox is capable of evaporating 18,173 pounds of water per hour, or an increase of 29 percent.

Before conversion the firebox shown in the drawing had a heating surface of 327 square feet including the combustion chamber. After the semi-watertube firebox was applied, the heating surface was increased to 432 square feet. The evaporation with the original firebox, calculated according to Cole's ratios, was 18,004 pounds of water per hour and, after the semi-watertube firebox was applied, 23,793 pounds of water per hour, or an increase of 32 percent.

View inside the semi-watertube firebox looking up toward the crown sheet

Besides increasing the direct firebox heating surface, the semi-watertube firebox increases the circulation in the side waterlegs of the boiler and is expected to prolong the life of the side sheets and staybolts. The rapid circulation set up by the semi-watertube firebox will also act as a safety measure for the crown sheet. A rapid upward circulation is set up, causing an overflow of water onto the crown sheet, which will prevent the crown sheet from being pulled away from the staybolts.

The troughs are so designed that they are free to expand and move in all directions with the movement of the firebox. Opposite the ends of each circulating tube, in both the roof and side wrapper sheets, is a clean-out plug which is removed at washout periods, so that the watertubes can be thoroughly cleaned when the boiler is being washed out.

Brick arches are applied in the firebox. The arch tubes are installed in the usual manner and the arch brick is chipped out to clear the circulating tubes.

The locomotives to which these fireboxes have been applied are showing a marked increase in steaming capacity and the fireboxes have developed no trouble up to the present time. This design of semi-watertube firebox is covered by patents in this country and Canadian patents pending.

Work of the A. S. M. E. Boiler Code Committee

The A.S.M.E. Boiler Code Committee meets monthly for the purpose of considering communications relative to the Boiler Code. Any one desiring information as to the application of the code is requested to communicate with the Secretary of the Committee, 29 West 39th St., New York, N. Y.

The procedure of the committee in handling the cases is as follows: All inquiries must be in written form before they are accepted for consideration. Copies are sent by the secretary of the committee to all of the members of the committee. The interpretation, in the form of a reply, is then prepared by the committee and passed upon at a regular meeting of the committee. This interpretation is later submitted to the Council of The American Society of Mechanical Engineers for approval, after which it is issued to the inquirer.

Below are given records of the interpretations of the committee in Cases Nos. 688 (Reopened), 704 (Reopened), 710-718, inclusive, as formulated at meetings of February 18 and 26, 1932, and Cases Nos. 709 (Reopened), 719, and 720 as formulated at meeting of March 25, 1932, all having been approved by the council. In accordance with established practice, names of inquirers have been omitted.

Case No. 688 (Reopened).—*Inquiry:* When used in the construction of the boiler pressure parts, can material made in accordance with Specifications S-2 be considered as being in compliance with Pars. P-2 and P-3?

Reply: Revised A.S.T.M. Specifications A89-31T provide for firebox-quality plate. The Boiler Code Committee recommends that pending the adoption of revisions to Specifications S-2 for Steel Plates of Flange Quality for Forge Welding to make them identical with A.S.T.M. Specifications A89-31T, the use of these revised specifications be permitted as being in compliance with Pars. P-2 and P-3. In line with the above recommendation the committee has under consideration the following proposed revisions of Pars. P-2 and P-3:

> P-2*a* Steel plates for any part of a boiler subject to pressure and exposed to the fire or products of combustion shall be of firebox quality in accordance with code Specifications S-1 or S-2.
>
> *b* Steel plates for any part of a boiler subject to pressure and not exposed to the fire or products of combustion shall be of firebox or flange quality in accordance with code Specifications S-1 or S-2.
>
> *c* Minimum tensile-strength limits other than those specified for flange and firebox plates may be used provided a range of 10,000 pounds per square inch is specified, that the maximum limit does not exceed 65,000 pounds per square inch, and that the steel conforms in all other respects to Specifications S-1 and S-2.
>
> *d* Seamless steel-drum forgings made in accordance with code Specifications S-4 may be used for any part of a boiler for which firebox quality only is specified or for which either firebox or flange quality is permitted.
>
> P-3 Open-hearth steel pipe or steel tubing in accordance with Specifications S-18 or S-17 may be used for a boiler drum or other pressure part exposed to the fire or products of combustion, provided the nominal diameter of the pipe or tubing is not greater than 18 inches.

Case No. 704 (Reopened).—*Inquiry:* What is the maximum allowable unit joint working stress for Class 3 vessels, with double-welded lap-type longitudinal joints for thickness of 3/8 inch or less under the provisions of Par. U-73 of the code? It is noted that this allowable working stress is not provided for in Par. U-70.

Reply: It is the opinion of the committee that the maximum unit joint working stress for double full-fillet lap welds for longitudinal joints of plates less than ¼ inch in thickness should be 5600 pounds per square inch, and for plates ¼ inch and over in thickness, 7000 pounds per square inch of cross section. It is proposed to restrict the use of full-fillet welded tanks to certain classes of service.

Case No. 709 (Reopened).—*Inquiry:* Will it be permissible under the requirements of the Power Boiler Section of the code to use electric-resistance-welded steel pipe or tubing which meets the requirements as to chemical and physical properties of Specifications S-18 and S-17 in the construction of code boilers?

Reply: The Boiler Code Committee recommends that pending the adoption of specifications covering electric-resistance-welded steel pipe or tubing, such material be accepted on the basis of lap-welded pipe under Specifications S-18 and lap-welded tubes under Specifications S-17, provided all the requirements for lap-welded pipe or tubes in these specifications are met, and provided that electric-resistance-welded tubes under Specifications S-17 shall be normalized at a temperature above the upper critical point of the material.

Case No. 710.—*Inquiry:* Is it the intent of the provisions in Pars. P-107 and U-75 of the code that the thickness of the reinforcing pad shall not be less than ¾ inch, irrespective of the thickness of the shell or head to which it is fitted?

Reply: It is the opinion of the committee that the ¾-inch thickness of reinforcing pads given in Pars. P-107 and U-75 was intended to apply only to the thicker shells or heads, and that a thickness of 1 inch may be taken as the least thickness of shell or head to which the requirement for ¾-inch reinforcing pads applies. A revision of these paragraphs is contemplated.

Case No. 711.—*Inquiry:* Can Par. P-203*d* be applied to determine the maximum pitch between the centers of through rods, or does this section apply only to the distance between the inner surface of the shell and the centers of through rods?

Reply: Par. P-203*d* is intended to apply to the spacing in stayed surfaces between the inner surface of a shell to which the surface is attached at its boundary and the centers of the adjoining braces. It is the opinion of the committee that the spacings between the centers of through rods or other types of braces supporting a stayed surface should not exceed the limits specified in Par. P-199 or Table P-7.

Case No. 713.—*Inquiry:* Is it the intent of Par P-303, of the code to require two stop valves in the connection from a boiler to a common steam main when any one of the battery of boilers does not have a manhole opening?

Reply: It is the opinion of the committee that if any one of two or more boilers connected to a common steam main is not fitted with a manhole opening so that entry to the interior is impossible, that boiler need not be fitted with more than one stop valve. A revision of Par. P-303 to this effect is under consideration.

Case No. 714.—*Inquiry:* When outlets are formed by welding to the shell or head of a vessel a forged steel nozzle having a saddle flange and neck of seamless construction, is it not permissible to compute the reinforcing effect of the saddle flange and a portion of the neck, in place of the rules given in Par. U-75 of the code for the design of such reinforcing rings and reinforced nozzles?

Reply: It is the opinion of the committee that the construction outlined in the inquiry may be considered as meeting the intent of the code requirements provided:

1.—The net cross-sectional area of the manufactured nozzle (having a saddle flange and neck of seamless construction) on a line parallel to the axis of the shell, shall not be less than the cross-sectional area of shell plate removed on the same line. This area shall be computed by multiplying the required shell-plate thickness (calculated by the formula in Par. P-180 and/or Par. U-20, using $E = 1$) by the maximum length of the shell plate removed by the opening.

2.—The reinforcing value of such a nozzle shall consist of the cross-sectional area of the saddle flange plus the cross-sectional area of the neck to a height 3 times the thickness of the saddle flange.

3.—All welds shall be strength welds as provided for by detail 6 of Fig. U-16.

4.—In cases where the replacement value of the nozzle does not equal the above requirements, an inside reinforcing ring[1] may be used to provide the additional necessary replacement.

Case No. 716.—*Inquiry:* Can any pressure part of a boiler of circular cross section be fusion welded in accordance with the requirements of Pars. P-101 to P-111 of the code? The part is a superheater header 14 inches in diameter having a shell thickness of 1 inch.

Reply: In the opinion of the committee such a structure is properly classified as a drum or shell.

Case No. 718.—*Inquiry:* Is it necessary to post-heat the longitudinal joints of cylindrical shells of heating boilers in the same manner as preheating is required for gas welding in Par. H-80 of the code, or is such post-heating with electric welding optional? It is the impression of certain manufacturers that post-heating is not mandatory when electric welding is used.

Reply: In view of the advance in the art of welding since Par. H-80 was formulated, the committee considers the provisions for preheating or post-heating to be unnecessary.

Case No. 719.—*Inquiry:* Is it necessary, under the requirements of Par. U-53 of the code, to provide plugged threaded openings for inspection purposes in vessels approximately 10 inches in diameter by 24 inches long, built for 200 pounds working pressure, such vessels being built for the distinct purpose of containing a mixture of lubricating oil and graphite, with a mechanical stirring device, the mixture to be forced out of the tank by means of compressed air? It is pointed out that such vessels are not subject to internal corrosion because of the nature of the contents, also that Par. U-62 would permit the omission of inspection openings.

Reply: It is the opinion of the committee that where the actual service conditions of a vessel are such that the vessel is not subject to internal corrosion, the inspection openings required by Par. U-53 may be omitted. Where compressed-air tanks are used in combination with a substance which prevents corrosion on the interior, it is the intent of the code that the inspection openings required by Par. U-53 may be omitted.

Case No. 720.—*Inquiry:* As the corner radius of a flanged and dished head may, according to Pars. P-197 and U-38 of the code, thin down an amount not exceeding 10 percent of the original metal thickness, cannot this allowable decrease in thickness also be applicable over the crown of the dish, and if so, is the factor of safety used in Pars. P-195 and U-36 sufficient to provide for this thinning condition?

Reply: It is the opinion of the committee that the thickness of all portions of heads of semi-elliptical shape and the spherical portion of heads dished to a radius shall not be thinner than that required by the rules.

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[1] The outside diameter of the reinforcing ring must at least equal the outside diameter of the saddle of the nozzle.

Revisions and Addenda to the A. S. M. E. Boiler Construction Code ▲ ▲ ▲

It is the policy of the Boiler Code Committee, of the American Society of Mechanical Engineers, to receive and consider as promptly as possible any desired revision of the Rules and its Codes. Any suggestions for revisions or modifications that are approved by the committee will be recommended for addenda to the Code, to be included later on in the proper place in the Code.

The following proposed revisions have been approved for publication as addenda to the Code. They are published below with the corresponding paragraph numbers to identify their locations in the various sections of the Code, and are submitted for criticisms and approval from any one interested therein. Communications should reach the Secretary of the Boiler Code Committee, 29 West 39th Street, New York, N. Y., in order that they may be presented to the committee for consideration.

Note: In identical revisions in paragraphs of power and unfired pressure vessel codes the material appearing within brackets refers only to the text of the paragraph of the Code for Unfired Pressure Vessels.

PAR. P-23. REVISED:

P-23. *Thickness of Steam Piping.* In determining the thickness to be used for pipes at different pressures and temperatures the following formulas are to be used:

For pipes having nominal diameters of 1 in. and over:

$$P = \frac{2S}{D}(t - 0.09)$$

or

$$P = \frac{1.75S}{D}(T - 0.1)$$

where P = working pressure, lb. per sq. in.
t = minimum thickness of wall of pipe, in.
T = nominal thickness of wall of pipe including mill under tolerance of 12½ per cent, in.
D = actual outside diameter of pipe, in.
S = factor to be taken from Table P-5A.

Where the temperature differs from those in the table, the value of S may be determined by interpolation.

For pipes having nominal diameters less than 1 in. the value of the constant between the parentheses in the formulas to be subtracted from the thickness shall be as follows:

Nominal pipe size	¾ in.	½ in.	⅜ in.	¼ in.
Constant for first formula	0.08	0.07	0.065	0.06
Constant for second formula	0.09	0.08	0.07	0.065

When a steel or wrought-iron pipe is pierced with tube holes the maximum allowable stress in the ligaments in lb. per sq. in. shall not exceed the values given in Table P-5B, and the maximum allowable working pressure shall be computed in accordance with the rule in Par. P-180, provided the tube holes do not pierce the weld in the welded pipe and provided the pressure shall not be greater than that allowed for the unpierced pipe.

Table P-5A.—Values of Factor S to be Used in Formulas in Par. P-23[1]

Material	Spec. No.	For temperatures in degrees F. not to exceed 750	800	850	900	950
Seamless medium-carbon steel ..	S-18	11,500	9160	7520	5650	4000
Seamless low-carbon steel	S-18 S-17	9,000	7020	5800	4630	3440
Fusion-welded steel[2]	S-1	9,000	7200	6070	4950	3600
Fusion-welded steel, Grade B[2] ..	S-2	8,200	6600	5450	4350	3240
Fusion-welded steel, Grade A[2] ..	S-2	7,400	5900	4900	3900	2880
Lap-welded steel................	S-18	7,000	5580	4630	3690	2720
Butt-welded steel...............	S-18	5,000	3980	3310	2630	1940
Lap-welded wrought iron.......	S-19	5,300	4220	3510	2790	2060
Butt-welded wrought iron	S-19	4,500	3580	2980	2370	1750
Brass	S-24	4,500	At temperature not			
Copper	S-23	4,500	to exceed 406 F.			

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[1] The factor S given in the table is based on the allowable stress for the material used as pipe for temperatures of 750 deg. or over, and the thickness subtraction factor limits the actual working stress in the pipe due to internal pressure to a value less than S in order to allow for cross-strains and mechanical or thermal stresses that cannot be determined accurately.

[2] Fusion welded in accordance with Pars. P-101 to P-111, inclusive.

PAR. P-25. CANCEL REVISION SHOWN IN APRIL ISSUE, AND REVISE PARAGRAPH TO READ:

P-25. *Blow-Off Piping.* Blow-off piping shall be of black wrought iron (not galvanized) or black steel, and shall be as heavy as required for the feed pipe and in no case less than extra strong pipe size.

PARS. P-107 AND U-75. OMIT.

PARS. P-108 AND U-76. REVISE FOURTH SECTION TO READ:

All connections attached by fusion welding shall be stress-relieved [on vessels requiring stress relief and as required by Par. U-59*b* in all cases specified in Fig. U-16].

PAR. P-193*b*. OMIT.

PARS. P-194 AND U-33. REVISE FIRST SECTION TO READ:

The longitudinal joints of a riveted dome 24 in. or over in inside diameter shall be of butt-and-double-strap construction, or the dome may be made without a seam of one piece of steel pressed into shape. In the case of a dome less than 24 in. in diameter for which the product of the inside diameter and the maximum allowable working pressure does not exceed 4000 in. lb., the longitudinal joint may be of the lap type, provided it is computed with a factor of safety not less than 8.

OMIT THE LAST-SECTION AND REPLACE BY THE FOLLOWING:

The attachment of riveted domes and manhole frames to shells or heads of boilers [pressure vessels] shall be designed in accordance with Par. P-268*b* [U-59*b*].

PARS. P-195 AND U-36. INSERT THE FOLLOWING AFTER THE FOURTH SECTION:

All other openings which require reinforcement placed in a head dished to a segment of a sphere shall be reinforced in accordance with Par. P-268 [U-59].

INSERT THE FOLLOWING AFTER THE SEVENTH SECTION:

All other openings which require reinforcement in an elliptical head including a nozzle or ring-type manhole shall be reinforced in accordance with the requirements of Par. P-268 [U-59], and when so reinforced the thickness of an elliptical head may be the same as for a blank head in semi-elliptical form.

OMIT THE LAST SECTION.

ADD THE FOLLOWING SECTION:

Unreinforced openings in heads shall be governed by the following rules:

(1) The edge of any unreinforced opening, excluding rivet holes, shall come no closer to the line bounding the spherical portion of the head around a manhole than the distance equal to the thickness of the head, and in no case except for water-column connections shall it come within the part formed by the corner radius of a dished head.

(2) The maximum allowable diameter of any unreinforced opening in a head shall not exceed that permitted by the rules in Par. P-268*a* [U-59*a*] for a shell of the diameter, thickness, working pressure, and material, nor shall it exceed 8 in. in any case.

Table P-5B.—Maximum Allowable Stress in Ligaments Between Tube Holes in Pounds Per Square Inch

Material	Spec. No.	For temperatures in deg. F. not to exceed					
		700	750	800	850	900	950
Seamless medium-carbon steel	S-18	12,400	11,500	9160	7520	5650	4000
Seamless low-carbon steel	S-18 S-17	9,600	9,000	7020	5800	4630	3440
Fusion-welded steel [1]	S-1	11,000	10,000	8000	6750	5500	4000
Fusion-welded steel, Grade B [1]	S-2	10,000	9,110	7330	6050	4830	3600
Fusion-welded steel, Grade A [1]	S-2	9,000	8,220	6550	5440	4330	3200
Lap- or butt-welded steel	S-18	9,000	8,220	6550	5440	4330	3200
Lap- or butt-welded wrought iron	S-19	9,000	8,220	6550	5440	4330	3200

[1] Fusion-welded in accordance with Pars. P-101 to P-111, inclusive.

(3) The minimum distance between the centers of any two unreinforced openings, rivet holes excepted, shall be determined by the following formula:

$$L = \frac{A + B}{2(1 - K)}$$

Where L = distance between the centers of the two openings measured on the surface of the head in inches

A and B = diameters of the two openings in inches

K = same as defined in Par. P-268*a* [U-59*a*] for the equivalent shell described in (2)

PAR. P-259. REVISED:

P-259. A manhole reinforcing ring, when used, shall be of rolled, forged, or cast steel, and shall comply with the requirements of Par. P-268.

PAR. P-260. OMIT SECOND SENTENCE.

PARS. P-261 AND U-56. OMIT.

PAR. P-262. REVISED:

P-262. Manhole plates and cover plates shall be of rolled, forged, or cast steel, and their strength, together with that of the bolts and of the yokes, if any, shall be proportioned for the service for which they are intended.

PARS. P-268 AND U-59. REVISED.

Unreinforced Openings. *a* Plain unreinforced holes, such as threaded openings tapped directly into the shell of the boiler [pressure vessel], drilled holes for the boiler-tube type of connection and studded connections, shall not exceed the diameter given by the charts in Fig. P-27½ [U-3½],[3] nor shall they exceed a diameter of 8 in. in any case. The diameter of a threaded opening shall be taken as that at the root of the thread.

The definitions of the symbols shown in Fig. P-27½ [U-3½][3] are as follows:

d = maximum allowable diameter of openings, in.

D = outer diameter of the shell, in.

t = actual thickness of shell, in.

$K = \dfrac{PD}{2ST}$

P = working pressure, lb. per sq. in.

S = working stress, lb. per sq. in., given by Table P-6 [U-3].

When there is a series of unreinforced openings in a boiler [pressure vessel], the efficiency of the ligaments between openings shall be calculated by the rules given in Pars. P-192 and P-193 [of Section I of the Code].

Threaded Connections. All threaded connections 1-in. pipe size or over which conform to the American Pipe Thread Standard shall have not less than the number of full threads given in Table P-11 [U-6]. For smaller threaded connections there shall be at least four such threads. Other thread standards may be used provided the threaded thickness of the material conforms to Table P-11 [U-6].

If the thickness of the shell of the boiler [pressure vessel] is not sufficient to give such number of threads, a construction shall be employed which will provide at least the required number of threads.

When the maximum allowable working pressure exceeds 100 [125] lb. per sq. in., connections over 3-in. pipe size shall not be threaded into the wall of the vessel. Seal welding may be employed either on the outside or the inside of the shell.

Expanded Connections. A pipe or tube connection or forging may be attached by inserting through an opening and expanding into the shell, provided the diameter of such an opening is not greater than that permitted for unreinforced circular openings in this section. Such connections shall be expanded and flared not less than ⅛ in. over the diameter of the tube hole or they may be flared not less than ⅛ in., rolled and beaded, or flared, rolled, and welded. Such tubes shall project through the shell not less than ¼ in. nor more than ½ in. before flaring. Where such tubes enter at an angle, the maximum limit of ½ in. shall apply only at the point of least projection. The outside diameter of such a connection shall not exceed 6 in.

Studded Connections. A studded connection with a flat surface machined on the shell for a gasket may be used for attaching flanged fittings, provided the studs are engaged in the shell for a depth of at least the diameter of the stud used. The design and bolting of the flange shall be in accordance with Par. P-299 [of Section I of the Code]. Stud holes shall straddle the center line of the vessel. The equivalent diameter of the opening shall be that determined by the total length of shell removed, including stud holes, if any, on any line parallel to the longitudinal axis of the shell. The equivalent diameter shall not exceed the maximum allowable diameter of an unreinforced opening as given by the rules above, using in Fig. P-27½ [U-3½][3] the minimum thickness of the shell resulting from the machining of the flat surface.

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[3] See Figs. 1 and 2 on pages 368 and 369 of this issue.

Reinforced Openings. *b* An opening in the shell of a boiler [pressure vessel] with a diameter greater than the maximum unreinforced opening permitted by section (*a*) shall be provided with reinforcement. Such reinforcement shall consist of one or more reinforcing rings or flanges riveted, welded, or brazed to the shell and/or a tube or pipe extension or fitting welded to the shell and/or welded to or integral with the reinforcing flange. The thickness of each riveted reinforcing flange or ring shall not be less than given in Table P-11½ [U-6½].

Table P-11½ [U-6½].—Minimum Thickness Reinforcing Rings or Flanges for Riveted Construction

Thickness of shell plate, in.	Thickness of reinforcing ring or flange, in.
1/8	1/8
3/16	3/16
1/4 to 11/32	1/4
3/8 to 13/32	5/16
7/16 to 15/32	3/8
1/2 to 9/16	7/16
5/8 to 3/4	1/2
7/8	5/8
1	11/16
1 1/8 to 2	3/4
Over 2	1

The outside diameter of a riveted reinforcing ring or flange shall not be less than 1½ times the diameter of the opening in it.

The thickness of a tube or pipe extension welded to the shell and/or a reinforcing ring or flange shall not be less than that for extra strong [standard-weight] pipe of the same diameter, and shall comply with Par. P-23 [of Section I of the Code].

For nozzle fittings having a bolting flange and an integral flange for riveting or welding, the thickness of the flange attached to the boiler [pressure vessel] shall also be not less than the thickness of the neck of the fitting.

All reinforced circular or elliptical openings shall comply with the following requirements:

(1) On a line parallel to the longitudinal axis of the shell and passing approximately through the center of the opening and through the weakest section of a riveted reinforcing ring or flange, the total cross-section in the complete reinforced opening, including the shell and cross-section of fusion welds if any, and deducting for rivet holes if any, within the limits defined below shall be at least equal to the cross-section obtained by multiplying the shell thickness required by Par. P-180 [U-20], using $E = 0.90$, by twice the diameter of the opening less 2 in. The above-mentioned limits are:

A distance on each side of the center line of the opening equal to the actual inside diameter of the opening in the finished construction.

A distance on each side of the center line of the actual thickness of the shell equal to 3 times such actual thickness or 5 times the thickness of the tube or fitting, whichever is greater (see Fig. A-11 [UA-1]).

When there are two or more adjacent openings, the limits shall not be considered to overlap, and in no case shall any portion of a cross-section be considered to apply to more than one of the two adjacent openings.

(2) On either side of the line parallel to the longitudinal axis of the shell, as determined in (1), the strength of the attachment to the boiler [vessel] of each separate part of a reinforced opening shall be at least equal either to the tensile strength of the cross-sectional area determined by multiplying the shell thickness required by Par. P-180 [U-20], using $E = 0.90$, by the diameter, less 2 in., of the opening in the shell in the finished construction, or to the tensile strength of the cross-section of the reinforcement within the above limits, whichever is smaller. For riveted construction, the strength of attachment is the shearing strength of the rivets, and for fusion-welded construction it is the strength of the weld in shear or in tension, whichever is smaller.

The unit working shear stress of a weld shall not exceed 0.8 times the allowable working stress in tension [for the class of fusion-welded vessel for which the welding process is suitable].

The size of a fillet weld shall not be less than one-half the thickness of the thinner of the two parts being joined.

Riveted Connections. Materials for riveted openings shall be of rolled, forged, or cast steel.

The strength of the rivets in tension in a flange frame or ring riveted to a drum [vessel], based on the minimum tensile strength given in the specifications, shall be at least equal to that required to resist the stress due to the maximum allowable working pressure with a factor of safety of five.

The tensile stress in the rivets due to the pressure shall be computed in the following ways:

(1) For outside calking the stress shall be equal to the area bounded by the outside calking multiplied by the maximum allowable working pressure.

(2) For inside calking (and with no outside calking) the stress shall be equal to the area bounded by the inside calking multiplied by the maximum allowable working pressure.

The rivets attaching nozzles shall be so spaced as to avoid the possibility of the shell plate failing by tearing around through the rivet holes. This feature shall be checked by applying the rules given in Par. P-193*c* [of Section I of the Code], which bear on the strength where a series of holes are placed in a drum.

Seal welding may be employed in accordance with the procedure in Par. P-257*b* [U-52*b*].

Forge-Welded Connections. Forge-welded connections shall be of forged or rolled steel material, seamless tubing, or forge-welded pipe.

[Forge-welded connections shall be attached by the methods shown in Figs. U-17 and U-18.]

All forge-welded connections shall be stress-relieved in accordance with the procedure given in Par. P-108 [U-76].

Fusion-Welded Connections. Materials for fusion-welded connections shall be in accordance with Par. P-103*b* [U-71*b*].

All connections attached by fusion welding [on Class 1 vessels] shall be stress-relieved.

[All connections attached by fusion welding on Class 2 vessels requiring stress relief shall be stress-relieved.]

All connections attached by fusion welding on Class 2 vessels which do not require stress relief shall be stress-relieved except those formed by welding a plain-end pipe, welding neck or nipple directly to the vessel without the attachment of additional material, when the diameter of the opening does not exceed 10 percent of the outside diameter of the shell. Some such connections are illustrated in Figs. U-16 *A*, *C*, and *D*. Connections formed by welding to the vessel a hydraulic coupling or special forging for a threaded opening not to exceed 3 in. pipe size need not be stress-relieved.

[Connections attached by fusion welding on Class 3 vessels need not be stress-relieved.]

[All connections attached by fusion welding to forge-welded, riveted, brazed, or seamless vessels shall be stress-relieved in accordance with the requirements for connections on Class 2 vessels which do not require stress-relieving. If any such vessels are to be used for service equivalent to Class 1 fusion welding, then any fusion-welded connection must be stress-relieved.]

Fusion-welded connections [which require stress-relieving and] which are attached to vessels whose seams are of riveted construction shall be fabricated and stress-relieved prior to the making up or attachment of the courses by riveting. [If they do not require stress-relieving and are attached after riveting, the welds shall be located at a distance from the riveted seam at least equal to the diameter of the opening plus 4 times the plate thickness of the shell.]

Fig. P-6 [U-6] illustrates some types of fusion-welded connections which are acceptable.

[*Brazing.* For threaded openings in pressure vessels where brazing is permitted, if the thickness of material in the vessel is not sufficient to give the number of threads specified in section *a*, the openings may be fabricated for a threaded connection by brazing to the shell a plate or a forged boss with inside flange, or any other type of connection described in this section may be used.]

c Typical examples of the application of the above rules are presented in Pars. A-63 to A-68 [UA-10 to UA-16] of the Appendix. (Note: For the full text of these paragraphs, communicate with the Secretary of the Boiler Code Committee.)

(To be continued)

Three Explosions Caused by Lap Seam Cracks*

The most dangerous defect to which old boilers with longitudinal lap seams are subject—grooving and cracking of the seam—caused three explosions within a few weeks of each other early this year. Five men lost their lives and three were injured.

At one plant, a sawmill in a southern state, a horizontal tubular boiler 36 inches in diameter and 12 feet long failed at the longitudinal seam of the front course on the morning of March 10. This sheet tore away from the front head and second course, crashing into and demolishing a mill building. The two other courses, together with the rear head, sailed like a rocket over another building and came to rest in a copse of wood 900 feet from the starting point. With the tubes still attached to it the front head was driven against a tree with such terrific force that some of the tubes were bent almost double. Had the tree not been there, these parts of the boiler would have plunged into the residence of one of the mill workmen. One tube did break away and pass completely through the house, but fortunately it did not strike any of the several women and children who were there.

The grooving along the seam that failed was so deep that an inspector who examined the plate after the accident was inclined to believe that the explosion occurred at the ordinary working pressure of the boiler. The fireman had brought the pressure up and was in the act of rolling the engine off center, preparatory to starting the mill for the day's run, when the accident took place. Three men were killed instantly and three others were seriously injured, one critically. Destruction of the mill was complete.

A boiler of the locomotive firebox type figured in another recent explosion at a sawmill in the South. The mill had been working at a new location only two days when, while workmen were preparing to start operations on the morning of January 12, the boiler blew up and killed two men who were firing it. Failure took place along the inside lap of the course nearest the firebox, and was caused by a groove along the inside lap so deep that at some points only a thin film of sound metal remained. This explosion was extremely violent, destroying the mill building completely.

The fortunate fact that only one man was on the premises is probably the reason that no one was killed or injured when, on January 11, a 48-inch locomotive-type boiler let go at a plant in Maine. He was out of range of the force of the blast and suffered only the effects of being thrown heavily to the floor.

Previous to the accident there had been no indication that a defect existed in the boiler. Two weeks before, attendants had removed manhole covers and given the vessel a thorough washing out. The safety valve was set to blow at 95 pounds, this pressure having been reached on many occasions without signs of leakage.

The initial break occurred along the outside surface of the inside sheet forming the double riveted lap-seam of the front course. After this sheet tore loose, it sailed over several plant buildings and came down through the roof of a two-story warehouse 400 feet away. The reaction of escaping steam caused the main portion of the boiler to jump backward about 20 feet to strike the main plant building. The impact was sufficient to move the entire building 15 inches on its concrete piers. Property damage was estimated at $4000.

The plate was judged to have been 9/32 inch thick originally, and uniform corrosion with some slight pitting had reduced this to 7/32 inch in some places. There was also a well-defined crack. The edges of the crack were rough and showed signs of considerable exposure. The sound metal remaining at the crack varied from 1/16 inch to not over ⅛ inch in thickness.

This boiler was of the locomotive, wet-bottom type, 48 inches in diameter and 17 feet long. It had been bought two years before from a sawmill and was probably about 15 or 20 years old.

Gilbert E. Ryder Dies

Gilbert E. Ryder, vice-president of the Superheater Company, New York, died at his home at Larchmont, N. Y., on May 17. Mr. Ryder was born May 29, 1880, at Minneapolis, Minn. He was educated in the public schools of Sandwich, Ill., also at Tabor Academy, Marion, Mass., the University of Wisconsin and the University of Illinois. He began work as a special apprentice with the Chicago, Milwaukee & St. Paul, now the Chicago, Milwaukee, St. Paul & Pacific, and subsequently served consecutively as machinist and draftsman on that road; assistant mechanical engineer of the Briquette division, United States Geological Survey, Technical Branch; deputy smoke inspector of the City of Chicago; and associate editor of the Railway Review. In December, 1911, Mr. Ryder entered the service of the Superheater Company, New York, as a service engineer; he was subsequently appointed assistant to vice-president and in 1921 was elected vice-president in charge of sales and service. Mr. Ryder was a member of a number of technical societies, including: The American Society of Mechanical Engineers; the International Railway Supply Men's Association, with which he served as president.

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* Published through the courtesy of *The Locomotive*, of the Hartford Steam Boiler Inspection and Insurance Company.

Method of Laying Out a Branch Pipe to a Cone

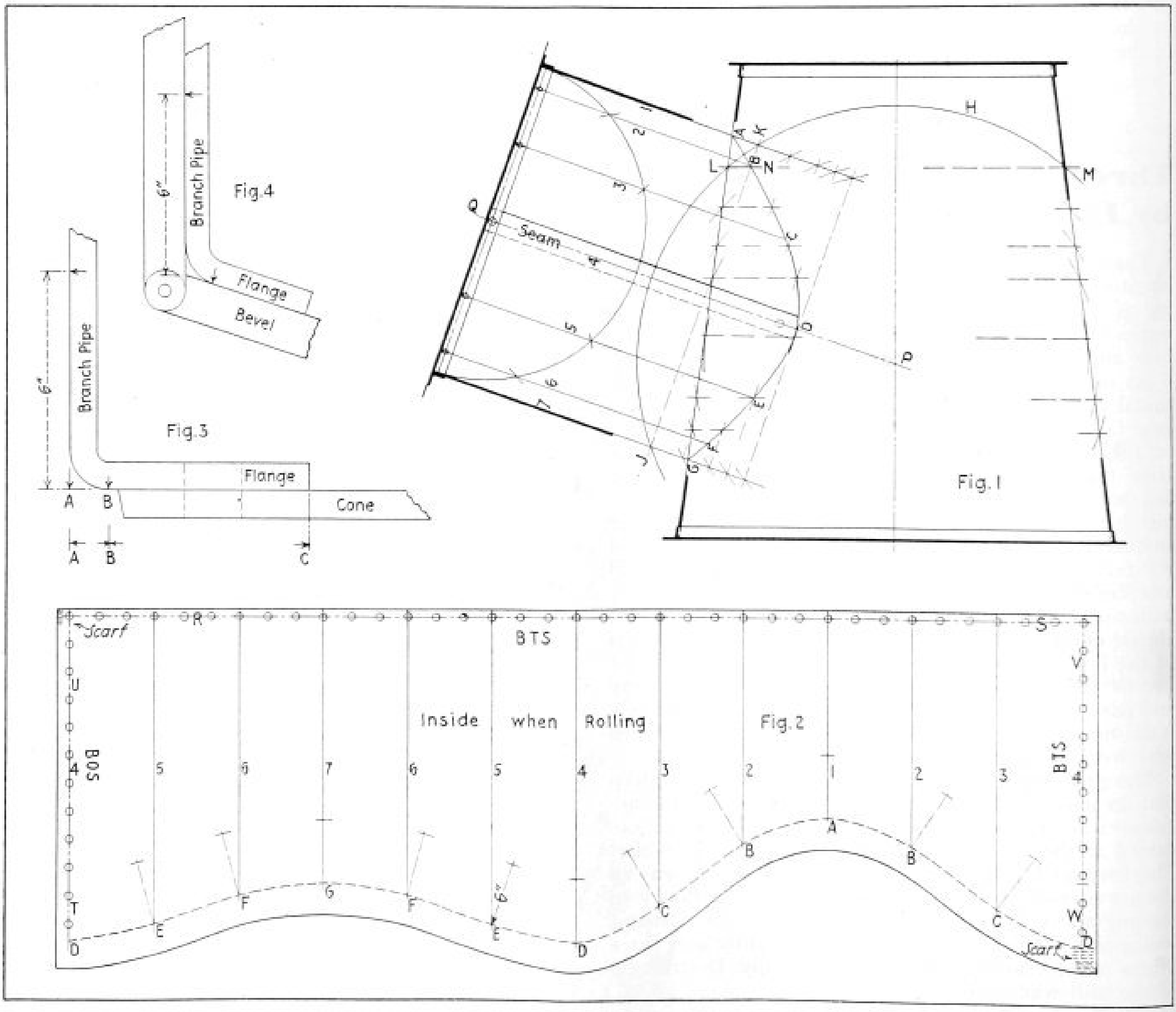
By I. J. Haddon

The layout of a branch pipe to a cone when the center lines of both the pipe and the cone meet, is comparatively simple and may be accomplished by the following method.

Lay down the elevation of the cone and pipe, showing the thickness of plates, as in Fig. 1, the inside lines of the pipe meeting the outside lines of the cone. Draw the center lines meeting at *P*. Now with *P* as center, and with various radii, draw arcs, cutting the pipe and cone. For example, take the arc *H*, it cuts the pipe in *J* and *K*, and it cuts the cone in *L* and *M*. Join *J* and *K* as well as *L* and *M*; these lines will cross at *N*; then *N* will be one point on the interpenetration line. Do the same with other arcs to obtain other points on the interpenetration line; draw a fair curve through the resulting points as shown.

With *Q* as a center and the radius of the branch pipe as a radius, describe a semicircle and divide it into any number of equal parts as shown. Through the points obtained draw lines parallel to *Q-P* meeting the interpenetration line in *B*, *C*, *E* and *F*; *A*, *D* and *G* already meet it. Number the lines *1, 2, 3, 4, 5, 6,* and *7*, as shown. As a plan of the cone in Fig. 1 is not required, we are now ready to develop the branch pipe.

Draw the lines *R-S*, *T-U* and *V-W* as shown in Fig. 2. These lines will be the centers of holes for the angle ring and seam. Make the distance between *T-U* and *V-W* equal to the circumference of the pipe measured on the *center of thickness* of the pipe, and divide *R-S*



Method of developing a branch pipe

into the same number of equal spaces, as in Fig. 1. In this case, as shown, it is 12 spaces. Let us suppose the seam is at *4;* then make the lines *4* and *4,* Fig. 2, equal to the line *4,* Fig. 1, measuring from the center of the holes in the angle ring to *D* on the interpenetration line. Do the same with all the other lines, thus obtaining the points *E, F, G, F, E, D,* etc., Fig. 2. Draw a fair curve through these points, as shown. Space the holes for the angle ring, and holes for the seam, as shown. Allow sufficient lap to take care of the size of holes in the plate. Now add, from the dotted line, sufficient material for flanging purposes. Put a light center pop at the points *E, F, G,* etc., on the dotted curve; also draw short lines normal to the curve and put a center pop on each of the lines, say 6 inches from the curve, as shown in Fig. 2.

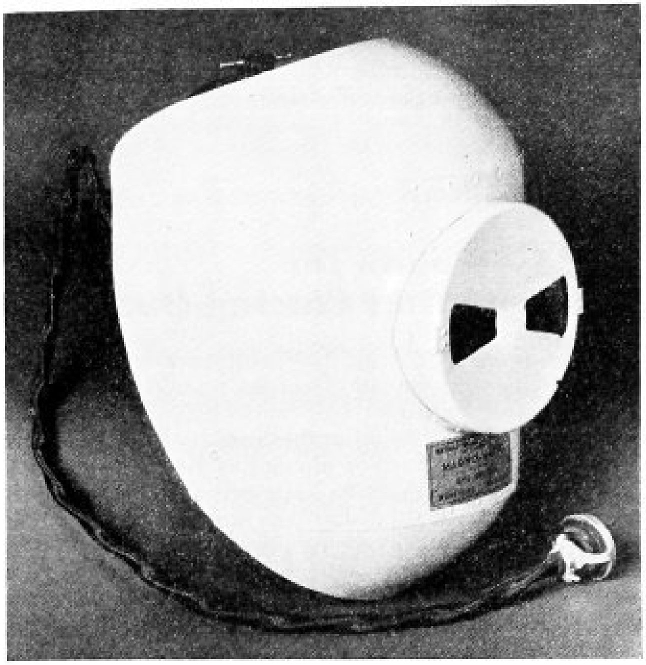
To find out how much to allow on flanging, Fig. 2; draw a full size sketch of the branch pipe and cone, Fig. 3. The center pops on the curve of Fig. 2 will, in flanging, go around from *A* to *B,* Fig. 3, so that the amount of material to allow would be from *B* to *C.* A little more than is shown should be allowed at *A,* Fig. 2, on account of the extra stretching, as may be seen by the angle it takes at *A,* Fig. 1. When flanging the pipe, after it is rolled up, seams bolted together and a rivet in the hole nearest the flange, take a bevel, as shown in Fig. 4, and put a chalk mark on it 6 inches from the inside angle. This mark should come even with the pops that were put on the pipe, Fig. 2, and it will be found that the pops at *E, F, G,* etc., Fig. 2, will be in the position as shown at *B* in Fig. 3. Scarf the plate as shown in Fig. 2 before being rolled up. The letters *BOS* and *BTS* mean "bevel other side" and "bevel this side" respectively, for calking purposes.

When the pipe is flanged, the holes may be marked on the flange, punched, then put on the cone in the correct position. The holes can be marked on the cone for drilling and the lap for the holes on the cone can be added. Burn the hole in the cone and reassemble for riveting. Of course, the cone could also be developed from the interpenetration line, as shown, but it is an out-of-date method to lay the hole out in the cone, because the cone would have to be rolled up *before* the piece was cut out to ensure its being kept at the correct curvature, etc.

I may say, the method shown of obtaining the interpenetration line Fig. 1; also applies to a pipe passing through a cone, a pipe to a pipe of different diameters, a cone to a cone, or a cone to a pipe *when their center lines meet.*

Automatic Welding Shield

The Michigan Welding Supply Corporation, Muskegon, Mich., has developed what is known as the Magno-Matic eye shield which automatically protects the welder from the injurious rays emanating from an electric arc, such as employed in welding. To this end dark protective lenses are mounted in a movable frame, the lens holder being actuated by an electro-magnet in such a manner that when the electrode touches the work the lenses instantly are projected before the eyes of the welder and remain in that position while the arc is maintained. When the arc is broken the dark lenses immediately spring back in place and the welder's view is clear; thus his vision is unimpaired at all times.



Magno-Matic eye shield with head band

The dark optical lenses conform to standard specifications for welding glass. They are held by spring clips to the lense holder so that renewal can be made readily without the use of tools. Renewal rarely is necessary, however, since these lenses with their operating mechanism are contained in a closed compartment protected by glass inside and outside. The inside glass is colored slightly to soften the effect on the eye from the instantaneous changes in light intensity. The operation of the shutting device is so fast that when the lense is open the electrode and often the work itself is at white heat, which would cause a slight strain on the eyes.

The outside glass, which offers protection against metal particles or fumes from the arc, becomes pitted and requires frequent renewals. With this in mind it is made circular in shape so that by merely revolving the glass several clean exposures may be obtained before renewal becomes necessary.

The Magno-Matic eye shield depends upon the voltage drop in the welding cable for its energy. One wire of the cord from the shield is furnished with an extension which is carried along the welding cable and connected to the cable at a point sufficiently far away from the electrode holder to obtain the required voltage drop. The shield requires a minimum of 0.7 volt at the cord connector and will operate satisfactorily up to 1.6 volts. In order to obtain this drop it may be necessary to substitute a length of smaller size welding cable but the capacity of these cables generally are ultra-conservative and this substitution is not objectionable. The normal range of 0.7 to 1.6 volts covers a corresponding range in welding current which is ample to meet most conditions. However, a wider range in welding current can be handled by the use of a duplex welding cable made up of two cables with a simple connecting device whereby one or both can be used depending upon the current handled.

Being actuated by a portion of the welding current itself the protective lenses must be inserted when the electric arc is established but the operator's view is clear until that moment. There is absolutely no excuse for him peeping around the corner of the shield to see his work before starting and thus be caught unawares by the flash. Production is increased by eliminating the time ordinarily required to remove or adjust the shield in order to see the work when not welding. Since the welder can see to place the electrode on the exact spot he wants to start the weld, he is able to obtain a better weld.

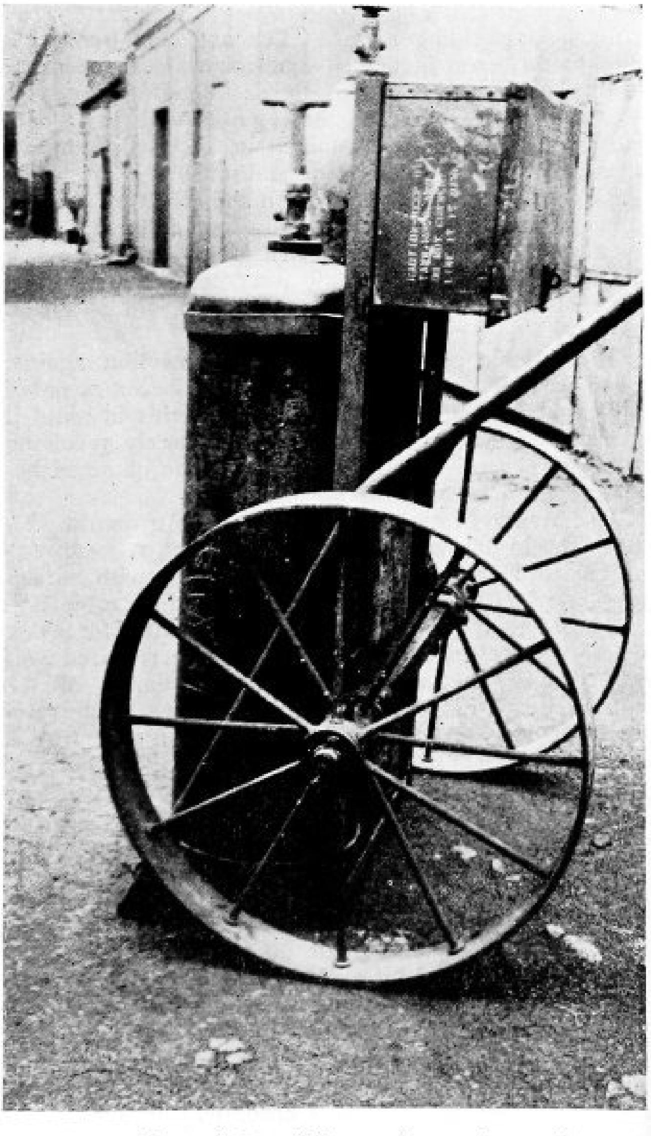
Handy Wagon for Transporting Cutting Outfits

The wagon shown in the illustration was designed for handling oxygen and acetylene tanks around the smaller repair yards.

The amount of cutting at these points is not sufficient to warrant the expense of piping gas through the yards, however, it is necessary to have cutting torch available many times during the day for removal of bolts, etc.

Wheels of 30 inches diameter are used and one man can move the wagon across a rail, if necessary, without assistance. The frame is made from 1½-inch by 1½-inch angle iron and can be riveted or electrically welded.

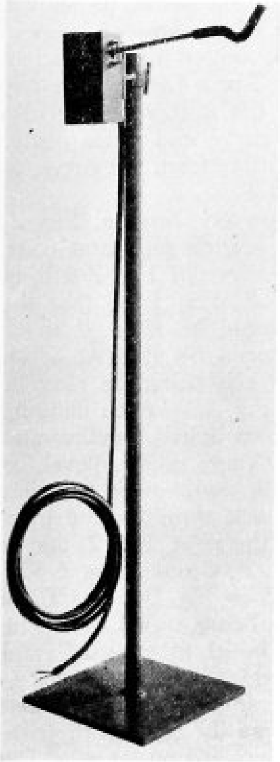
A metal box, as shown at the top of the wagon, is provided for torch tips, wrenches, goggles, etc.



A portable truck for welding equipment that can be handled by one man

Electrode Holder Stand Saves Power

A new electrode holder stand which automatically shuts down the welder one minute after the holder is placed on the stand and instantly starts the welder when the holder is removed, has been announced by The Lincoln Electric Company, Cleveland, makers of Linc-Weld motors and Stable Arc welders. The new stand will be known as the Lincoln Power Saver.



Electrode holder

It is claimed that the automatic start-and-stop feature of the new stand effects a decided saving in power consumption. Heretofore it has been common practice to let the welder run continuously during working hours even though the operator may actually be welding intermittently throughout the day.

By shutting the machine down when the operator is not actually welding, power will be saved and the overall power factor of the plant raised, since as is common with most motor-operated devices, when operating at light or no loads the power factor is reduced. Thus by shutting the machine down during idle times, it will at no time be operating at a reduced power factor.

It is claimed that the saving in power consumption will pay for the Power Saver in a short time.

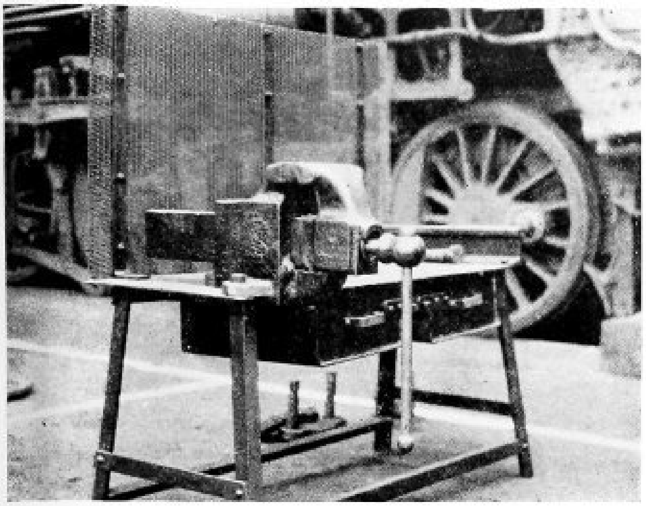
Operation of the stand is extremely simple. When the hook is depressed by the weight of the holder it activates a special timing device which, at the end of one minute, disconnects the welder from the power supply line. When the holder is removed from the hook the welder starts automatically.

The stand as shown in the illustration is of arc-welded steel construction throughout. Its broad, heavy base provides ample rigidity, yet does not impair its portability. The holder hook can be raised or lowered on the upright member to suit the convenience of the operator. Fifteen feet of rubber-covered, two-conductor cable is provided with stand for connection to welder. The Power Saver can be used whenever the welding equipment is provided with an automatic motor starter having two wire pilot control.

Screen Shield for Bench Vises

A piece of front-end netting attached to the side or end of the work bench in the shop will prove a valuable aid in preventing injuries to employees as a result of chips or flying particles from the vise striking them while they are either engaged at some job in the vicinity of the bench or are merely walking past it.

Where a great amount of chipping is done, such as



A chipping shield will prevent injuries to nearby workmen

cutting bolts or rivets out of pipe clamps or removing nuts from defective bolts and rods, it is advisable to provide a shield on the end and on one side of the bench.

The employees using the vise should be instructed to chip toward this shield, otherwise it is useless.

Special Quality Nickel Developed for Spinning

A special quality nickel is now being produced by the Huntington Works of The International Nickel Company for spinning. The improved spinning properties are due to the reduction of certain elements which affect the initial strength of nickel and the rate of work hardening. Sheets and strips made of this new material are considerably softer and may, therefore, be spun with less effort than the regular nickel.

The new material softens at a lower temperature and torch annealing can be used effectively. Most shapes can be completed without annealing, and a better finish is obtained on highly worked surfaces due to the fine grain structure of the metal. Pieces have been spun directly from flat blanks without initial draws or intermediate anneals.

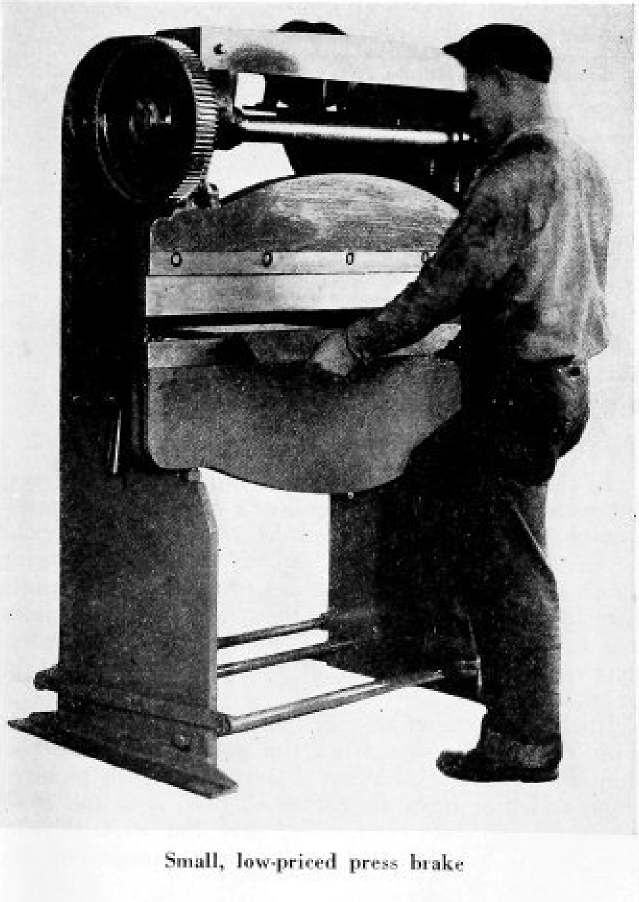
Chicago Steel Forming Press

A small powerful press brake, which will do work that heretofore required cumbersome machines which were both costly and expensive to operate, has been produced by the Dreis & Krump Manufacturing Company, Chicago. Of particular interest in these days is the fact that the price is less than half that of other machines of similar capacity which weigh from three to five times as much. Powered by a ¾-horsepower motor, the operating cost has been reduced to a minimum. Compact construction makes installation costs negligible and saves floor space. The machine can easily be moved. This press is particularly adapted to the rapid forming of sheet-metal sections, such as cabinets, fixtures and a great variety of sheet-metal work required in the average shop. Users of large press brakes have adopted this machine for smaller work, as it is faster and can be operated more efficiently. Die holders are standard and the same dies can be used as on large press brakes.

This machine is built of steel throughout, the materials and workmanship being of the same high standard as in the case of larger machines produced by this company. The bull gear and pinion are of steel. The body is constructed of two plates spaced ¾-inch apart to permit punchings to drop through to the floor. All bearings are bronze bushed except the eccentric straps, which are nickel cast. A single-plate friction clutch is built integral with the flywheel and is operated by a foot treadle. This brake works automatically with the clutch, it being possible to stop the ram at any point of the up or down stroke by releasing the treadle. The motor is mounted on adjustable hinged brackets and drives the flywheel by means of V belts. Lubrication is by the Zerk-Alemite system, a grease gun being furnished with the machine. Adjustable gibs are provided which take up wear both ways. Adjustment for various heights of dies is obtained by large set screws at each end which raise or lower the bed to the desired height. One set of two adjustable gages is standard equipment. All gears and the flywheel are completely guarded.

As die requirements vary, no dies are furnished as standard equipment. Since die holders are the same as those on larger press brakes, standard dies can be used. For this machine the company has developed multiple punching attachments, including cutting, notching and bending dies. All types of dies are available from the Dreis & Krump Manufacturing Company.

To give an idea of size, the length of the press between housings is 37 inches, the height 71 inches, and the weight 2100 pounds.



Small, low-priced press brake

The Boiler Maker

Volume XXXII Number 6

PUBLISHED Monthly by the Simmons-Boardman Publishing Company, 34 North Crystal Street, East Stroudsburg, Pa., with Executive Offices at 30 Church Street, New York City.

Chicago: 105 W. Adams Street. Washington, D. C.: 17th and H Streets, N. W. Cleveland: Terminal Tower. San Francisco, Cal.: 215 Market Street.

Subscription price in United States and possessions, 1 year, $2.00, 2 years, $3.00; Canada, including duty, 1 year, $2.50, 2 years, $4.00; foreign countries, 1 year, $3.00, 2 years, $5.00. Single copies, 35 cents each.

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Contents

Communication

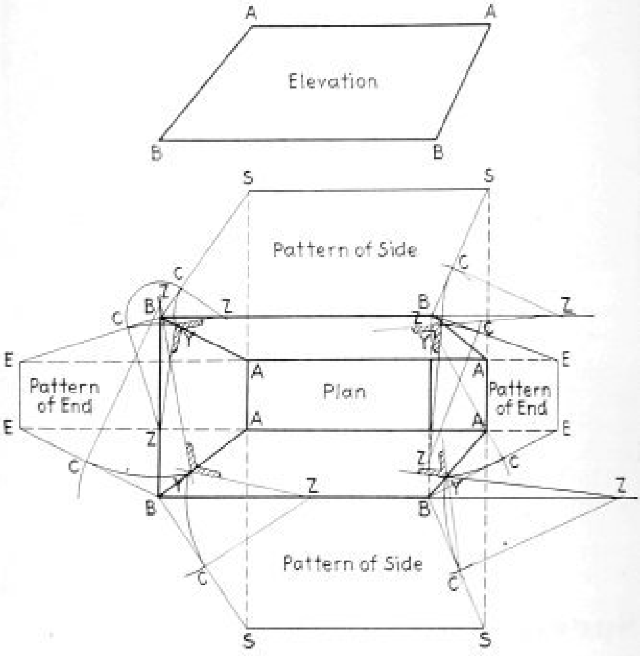
Bevel of Corner Angle

To the Editor:

In the February issue of The Boiler Maker I was interested in your correspondent's formula for determining the degree of bevel in corners of irregular hoppers. I would like to submit a sketch showing another way of arriving at the solution of this problem to obtain bevels when the plates or angle irons are used in construction of the corners; also when hoppers have to be bent at the corners on a brake or flange press.

In the illustration we have an elevation of a hopper with developed sides from the plan. The construction is as follows: Layout the elevation with required slope and base lines of plan *B-B*, *B-B*, *B-B* and *B-B;* locate the top lines *A-A*, *A-A*, *A-A* and *A-A;* develop the patterns of the sides and ends by projecting the points, *A*, *A*, *A* and *A* to the true length of same fro the base lines *B*, *B*, *B* and *B*. The development o all the corner bevels is obtained by the same principle

With the compasses set at the point *B* describe a arc cutting the lines *B-E* and *B-S* at pleasure, to establish the point *C*. From *C*, erect perpendiculars t cut the base line *B-B* at *Z*. Then with the compasse



Method of obtaining bevel angle

set at the point *Z* describe an arc from *C* to cut the corner of the hopper *B-A* at *Y*.

Connect *Y-Z* to form the angle *Z-Y-Z*. This gives the angle of the bevel of the corner, as shown. Follow the same procedure at all the corners, always noting that the angle of degree of the bevel is always toward the inside hopper.

The above will undoubtedly be of value to your readers as I have often been asked for a solution of this problem.

Schenectady, N. Y. Andrew Gray.

Trade Publication

Boilers: A new boiler catalogue of 48 fully illustrated pages is now being distributed by the Foster Wheeler Corporation, 165 Broadway, New York, N. Y. This catalogue, numbered, Bulletin B-32-3, describes the several different types of boilers manufactured in the corporation's shops and shows how these boilers are applied to complete steam generating units. The first portion of this catalogue is devoted to a description of sectional header boilers for stationary plants. These boilers are made in both cross-drum and longitudinal-drum types. The next part of the catalogue is devoted to bent-tube boilers for stationary plants. Illustrations show different arrangements of bent-tube boilers including three-drum, four-drum and six-drum types. The third section describes boilers of smaller size including the three-drum, bent-tube, low-head type and various horizontal fire-tube types. The last part of the catalogue is devoted to marine boilers of both water-tube and Scotch types.

Questions and Answers Pertaining to Boilers

By George M. Davies

This department is maintained for the purpose of helping those who desire assistance on practical boiler shop problems. Inquiries should bear the name and address of the writer. Anonymous communications will not be considered. The identity of the writer, however, will not be disclosed unless the editor is given special permission to do so.

Removable Head of Pressure Tank

Q.—Can you furnish the writer with the formula for calculating the size and thickness of a ring for a removable head on a pressure tank; also, the formula for finding the thickness of the removable head?

The writer would also like to know if there is a formula to find the size of the reinforcing angles on a vacuum tank, as mentioned in your article on "Pressure on a Vacuum Tank" in the December, 1931, issue of THE BOILER MAKER.

This information, or where to find it, will be greatly appreciated.—W. M. H.

A.—The thickness of a blank unstayed dished head with pressure on the concave side, as shown in Case I and Case II, Fig. 1, when it is a segment of a sphere shall be calculated by the formula:

$$t = \frac{8.33 \times P \times L}{2 \times TS}$$

where, t = thickness of plate, inches.

P = maximum allowable working pressure, pounds per square inch.

TS = tensile strength, pounds per square inch, originally stamped on the plate used in forming the head.

L = radius to which the head is dished, measured on the concave side of the head, inches.

The radius to which the head is dished shall not be greater than the diameter of the shell to which the head is attached.

Where two radii are used, the longer shall be taken as the value of L in the formula.

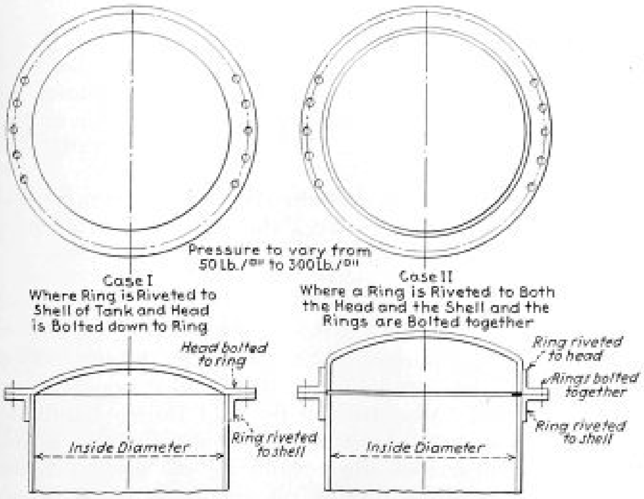


Fig. 1

The corner radius, Case II, Fig. 1, of an unstayed dished head measured on the concave side of the head shall not be less than 3 times the thickness of the material in the head; but in no case less than 6 percent of the diameter of the shell. In no case shall the thinning down at the corner radius of the knuckle of any dished head, due to the process of forming, exceed 10 percent of the thickness required by the above formula.

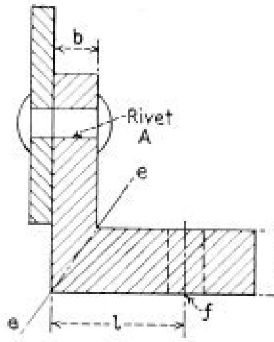


Fig. 2

The thickness of the flat portion of the head as shown in Case I, Fig. 1, should be determined as follows: The thickness required in unstayed heads, which are unpierced and are rigidly fixed and supported at their bounding edges by riveted or bolted attachments to shells or side plates, shall be calculated by the formula:

$$t = a\sqrt{\frac{0.162P}{s}}$$

where, t = thickness of plate in head, inches

a = diameter or short side of area, measured to the center of the inside row of rivets or bolts, inches

P = maximum allowable working pressure, pounds per square inch

s = allowable unit working stress, pounds per square inch, $\frac{TS}{5}$

TS = ultimate tensile strength, pounds per square inch, stamped on the plates, as provided for in the specification for the material

The size and thickness of the removable rings as shown in Case I and Case II, Fig. 1, would be determined as follows: The unit crushing stress in front of the rivet a, Fig. 2, or the shearing stress in the metal b, Fig. 2, is found by the formula:

$$W = S \times A$$

$$A = \frac{W}{S}$$

in which W = the unit stress in pounds.
S = the allowable stress for shearing or crushing, pounds per square inch = 5000 pounds per square inch
A = sectional area required to resist shearing or crushing

The application of these formulas cannot be made until the load on each inch of the circumferential length of the shell is determined.

First find the load on the head from the formula:

$$W_1 = d \times 0.7854 \times P$$

in which W_1 = total pressure on the head of the tank in pounds
d = inside diameter of shell, inches
P = working pressure, pounds per square inch

The unit stress in one inch of circumference of the shell equals

$$\frac{W}{d \times \pi}$$

where W = unit stress in pounds
W_1 = total pressure on head of tank in pounds
d = inside diameter of shell, inches

The load tending to shear or crush the ring equals

$$W = S \times A$$

$$A = \frac{W}{S}$$

Since the load is figured for a circumferential length of one inch, the thickness of the flange b, Fig. 2, would be A/1.

The number of rivets in single shear necessary to secure the ring to the shell would be:

$$N = \frac{W_1}{8800 \times a}$$

where N = Number of rivets
W_1 = total pressure on the head of the tank in pounds
a = cross-sectional area of one rivet

It will be necessary to assume a rivet diameter, determining same by securing a desirable pitch of rivets.

The next step is to determine the required number of bolts for securing the head to the tank:

$$N = \frac{W_1}{10{,}000 \times a}$$

where N = number of bolts
W_1 = total pressure on the head of the tank in pounds
a = cross-sectional area of bolt

It will be necessary to assume a bolt diameter, determining same by securing a desirable bolt pitch, so that there will be sufficient clearance between the bolts for turning the nuts in place. It may be necessary to use two bolt circles, in which case the bolts are staggered.

The next calculation is to determine the thickness of the flange d, which must be sufficient to resist bending and breaking along the plane c-c. Consider the section from the inside of the flange to the point (which is the center of the bolt circle when one bolt circle is used or midway between the bolt circles when two bolt circles are used) as a cantilever beam. The length of the beam equals l.

The bending moment equals

$$M = lW,$$

in which M = bending moment, inch pounds
l = length of cantilever (see Fig. 2)
W = unit load or stress in pounds

The required section modulus equals the bending moment divided by the allowable bending stress of the material. Allowing 10,000 pounds as a safe bending stress the section modulus equals

$$Q = \frac{M}{10{,}000}$$

The section modulus formula for a rectangular section equals

$$Q = \frac{bd^2}{6}$$

in which Q = section modulus
b = shortest side of rectangle
d = longest side of rectangle

b = unit length = one inch

$$d = \sqrt{\frac{Q \times 6}{1}}$$

where d = thickness of flange in inches
Q = section modulus

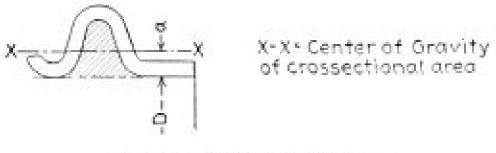
A table covering the strength of U. S. standard bolts from ¼ inch to 3 inches in diameter may be found on Page 716 of the 1924 Edition of the Marks Mechanical Engineering Handbook.

With reference to reinforcing angles on a vacuum tank, the thickness of the shell of the tank is ample for the load, therefore any reinforcing angles that may be used would be for stiffening or to protect against some strain due to the service of the tank or the manner in which the tank is supported. These stiffening angles should be of ample size and calculated for the stress placed upon them.

Mean Diameter of Morrison Furnace

Q.—A Morrison corrugated-type furnace is 48 inches in diameter where it is riveted on to the head, the thickness of the furnace material is ¾ inch and the depth of the corrugations is near 1½ inches. How do I obtain the mean diameter of this furnace? H. H.

A.—The mean diameter of a Morrison corrugated-type furnace may be computed by the following formula:



Section of Morrison furnace

$$\text{Mean diameter} = D + 2a$$

Where, D = least inside diameter in inches.
a = distance from least inside diameter to the center of gravity of the inside of the corrugation in inches, illustrated in Fig 1.

In computing the maximum allowable working pressure for corrugated furnaces the mean diameter may be taken as the least inside diameter plus 2 inches.

MODERNIZING THE OLD BOILER PLANT: Combustion Engineering Corporation, 200 Madison Avenue, New York, N. Y., has available for distribution copies of an article entitled "Modernizing the Old Boiler Plant" by Joseph Breslove, consulting engineer. In this article the author describes the revamping of several boiler plants and presents figures on operating costs before and after remodeling which show substantial economies.

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Maryland
Michigan
Minnesota
Missouri
New Jersey
New York
Ohio
Oklahoma
Oregon
Pennsylvania
Rhode Island
Utah
Washington
Wisconsin
District of Columbia
Panama Canal Zone
Territory of Hawaii

Cities

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Detroit, Mich.
Erie, Pa.
Evanston, Ill.
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St. Joseph, Mo.
St. Louis, Mo.
Scranton, Pa.
Seattle, Wash.
Memphis, Tenn.
Nashville, Tenn.
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States and Cities Accepting Stamp of the National Board of Boiler and Pressure Vessel Inspectors

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Cities

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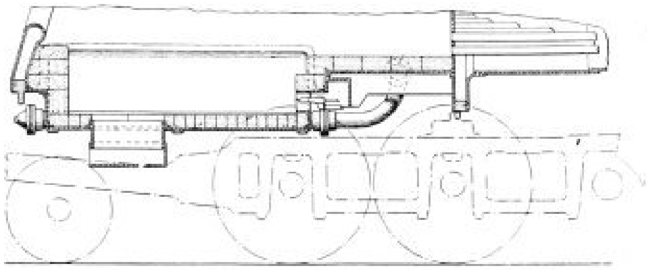
Parkersburg, W. Va.
St. Louis, Mo.
Scranton, Pa.
Seattle, Wash.
Tampa, Fla.

Selected Boiler Patents

Compiled by Dwight B. Galt, Patent Attorney, 1372-1376 National Press Building, Washington, D. C. Readers wishing copies of patents or any further information regarding any patent described, should correspond directly with Mr. Galt.

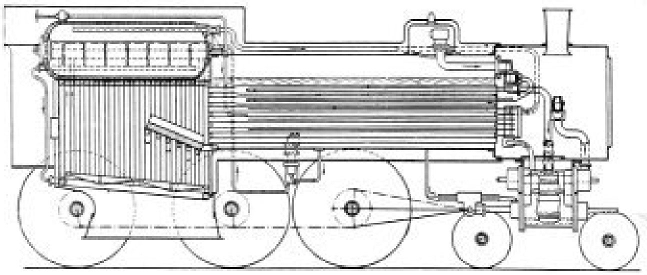
1,776,696. BOILER. JESSE C. MARTIN, JR., OF LOS ANGELES, CALIF.

Claim.—A locomotive boiler comprising, in combination, a barrel, a firebox having water legs, a water bottom for said firebox opening through one of the side sheets thereof, said water bottom curving upwardly and outwardly from an inner closed edge, and a passage for water from the barrel to the water bottom opening through one of the water legs. Eight claims.



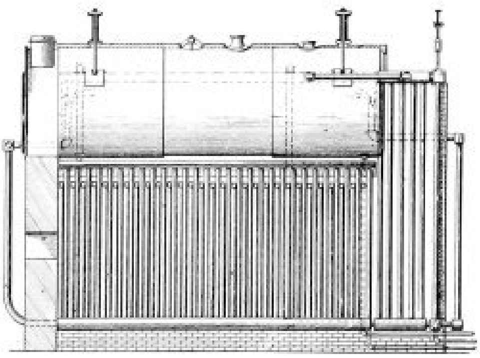
1,802,421. HIGH-PRESSURE STEAM-BOILER PLANT FOR PORTABLE ENGINES. OTTO H. HARTMANN, OF CASSEL-WILHELMSHOHE, GERMANY, ASSIGNOR TO SCHMIDT'SCHE HEISSDAMPF-GESELLSCHAFT M. B. H., OF CASSEL-WILHELMSHOHE, GERMANY, A CORPORATION OF GERMANY.

Claim.—The improvement in the generation of steam for use in multistage steam engines, which comprises heating a confined body of water by the hot combustion gases of a furnace so as to generate high-pressure steam from said water, heating a separate confined body of water by the cooler flue gases of such furnace so as to generate low-pressure steam therefrom, and conducting a portion of the high-pressure steam into said separate body of water to accelerate the generation of steam therefrom. Four claims.



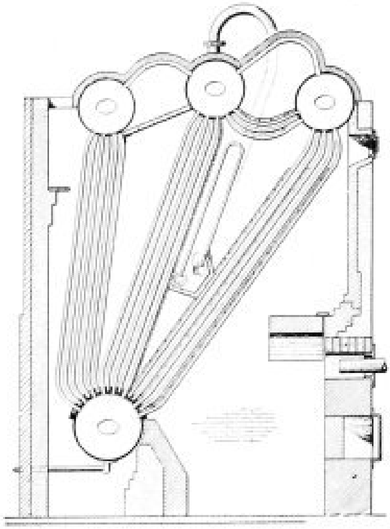
1,779,940. BOILER. PAUL G. KAISER, OF CHICAGO, ILL.

Claim.—The combination in a boiler of a boiler shell, firetubes therefor grouped to leave a vertically, longitudinally extending space for down-flowing water, a combustion chamber therebeneath, watertubes in the combustion chamber, and means affording water flow from the shell to the watertubes and communicating with the shell at a point which is above that area of the shell directly exposed to the combustion chamber, and which is in said space, thereby longitudinally to distribute the down draft of the water. Seven claims.



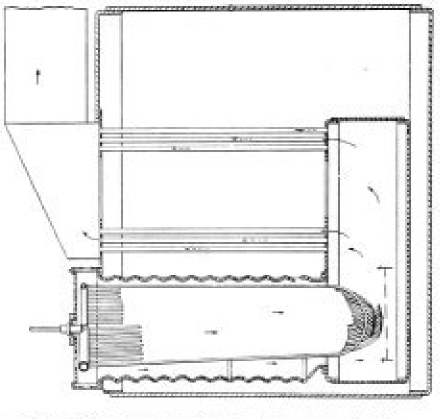
1,774,749. STEAM BOILER. DAVID S. JACOBUS, OF MONTCLAIR, N. J., ASSIGNOR TO THE BABCOCK & WILCOX COMPANY, OF BAYONNE, N. J.

Claim.—In a watertube steam boiler, upper and lower water chambers, a plurality of banks of watertubes connecting said chambers, a furnace in front of the front bank of tubes, an outlet for spent gases at the rear of the rear bank of tubes, a plurality of baffles in the front bank of tubes, with openings in successive baffles staggered relatively to each other, and a baffle on the front side of the next bank of tubes, said last-named baffle having staggered openings therethrough. Nine claims.



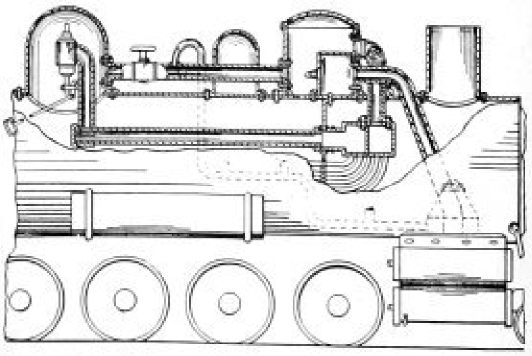
1,800,961. STEAM GENERATOR. JOHN B. SCHEEL, OF JACKSON HEIGHTS, NEW YORK, ASSIGNOR TO LA MONT CORPORATION, OF NEW YORK, A CORPORATION OF NEW YORK.

Claim.—In combination, a furnace, a set of parallel steam generating tubes extending from the front of the furnace along one wall, across the back and along an opposite wall again to the front, and a second set of similar tubes displaced in respect of the first set of tubes and crossing the first set at the rear of the furnace, said set of tubes being so disposed within the furnace that the planes containing both legs of each tube of each set are arranged at an angle to the horizontal. Fourteen claims.



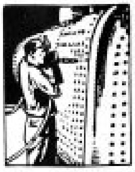
1,779,552. SYSTEM OF SUPERHEATED-STEAM DISTRIBUTION. JAMES F. MANTONYA, OF ALBUQUERQUE, N. M.

Claim.—In combination with a superheater for locomotives and the like, including a wet steam dome, a superheating unit, a distributing means from the wet steam dome to the superheating unit, a superheated steam dome connected to said superheating unit and an exterior valved connection between the superheated steam dome and the wet steam dome. Seven claims.



The Boiler Maker

Reg. U. S. Pat. Off.



Obsolete Locomotives

Of the 60,189 locomotives in the United States on December 31, 1930, the most recent complete figures available, 30,503 locomotives, or 49.3 percent were twenty years old or more and only 10,880, or less than 18 percent were under ten years of age. The advancement in locomotive engineering has been so rapid since the war, that only the power built in the last ten years may be considered modern. Numerically about 82 percent of all motive power in the United States should be regarded as obsolete.

It has been determined beyond the possibility of doubt that there is an inevitable rise in the cost of repairs with the increasing age of a locomotive. In this connection Robert S. Binkerd, vice-president of The Baldwin Locomotive Works, recently stated in the *Railway Age*: "Fleets of locomotives have had their repair costs analyzed for a series of years. Regardless of the type of locomotive, type of service or territory in which they are operated, they all show the same curve of rising costs. Good shop management may hold down the level, but is powerless to affect the rate of progression on that level. Between the first and third year the cost doubles. Between the third and twenty-fifth year it doubles again, and from the third year on there is an average annual increase of approximately 3.69 percent.

"It follows that under normal conditions a locomotive can be scrapped at the end of its twenty-first year and a new duplicate paid for out of savings in the cost of repairs that would otherwise be spent upon the supplanted locomotive during the ensuing twenty-one years. In other words, if there were no improvements in the locomotive art whatever, each 21-year-old locomotive could be supplanted by a new duplicate at no increase in cost.

"But exact duplicates of 21-year-old locomotives are practically never purchased. A new locomotive has the advantage of increased steam pressure, increased superheat, increased tractive force and increased boiler and firebox, resulting in the ability to turn out many more revenue ton-miles or revenue passenger-miles per locomotive hour and per pound of coal and water. These additional savings reduce the time required to pay for the new locomotive from twenty-one years to ten, eight, six or four years."

In contrast with the economic value of locomotives based on a replacement interval of twenty-one years, the facts show that at the end of 1931 the indicated turnover for all locomotives was once in fifty-eight years. If during the next four years new locomotives are not purchased more rapidly than during the 1920-30 decade the indicated turnover by 1935 would be once in every sixty-six years. Should the rate of purchase between 1926 and 1930 prevail the turnover by 1935 would be once in every seventy-seven years.

This condition will certainly not be allowed to continue. Modern demands for rapid railway service, and operating economy and efficiency in providing it, will inevitably require a definite and broad program of locomotive replacement in the near future.

Plate Strength

Not often is it possible to present as complete a study of the properties of steel plates and riveted joints, which enter into the construction of modern boilers and tanks, as that conducted on wide plates by the University of Illinois, the first installment of which appears in this issue. While the information available on this subject is very broad, the investigation in question develops a new range of knowledge through the equipment and facilities of the engineering experiment station at the University.

The results, obtained from the tests described, indicate that, for continuous wide plates without joints, these plates may be expected to develop about 90 percent of the unit strength of standard control specimens cut from the same plate. In some instances the plates tested developed a unit stress less than 86 percent of the strength of the control specimens.

What this means in terms of the strength and safety factors of vital structures can readily be understood. Where formerly calculations have been based on the results of standard test specimens, consideration must now be given to possible discrepancies in the strength of the full-sized structure. Further investigations on this subject of the strength of large plates should undoubtedly be undertaken and, where necessary, the rules governing the design of structures made from such plates should be revised.

Uniform Boiler Law Society

Recently the American Uniform Boiler Law Society, an organization formed to promulgate the American Society of Mechanical Engineers Boiler Code throughout the country, has been reorganized in order that it might better co-operate with related groups and serve a broader field of manufacture. To this end a new Constitution and By-Laws has been adopted. Details of the changes made will be outlined in an early issue.

Efficiency of Wide Plates*

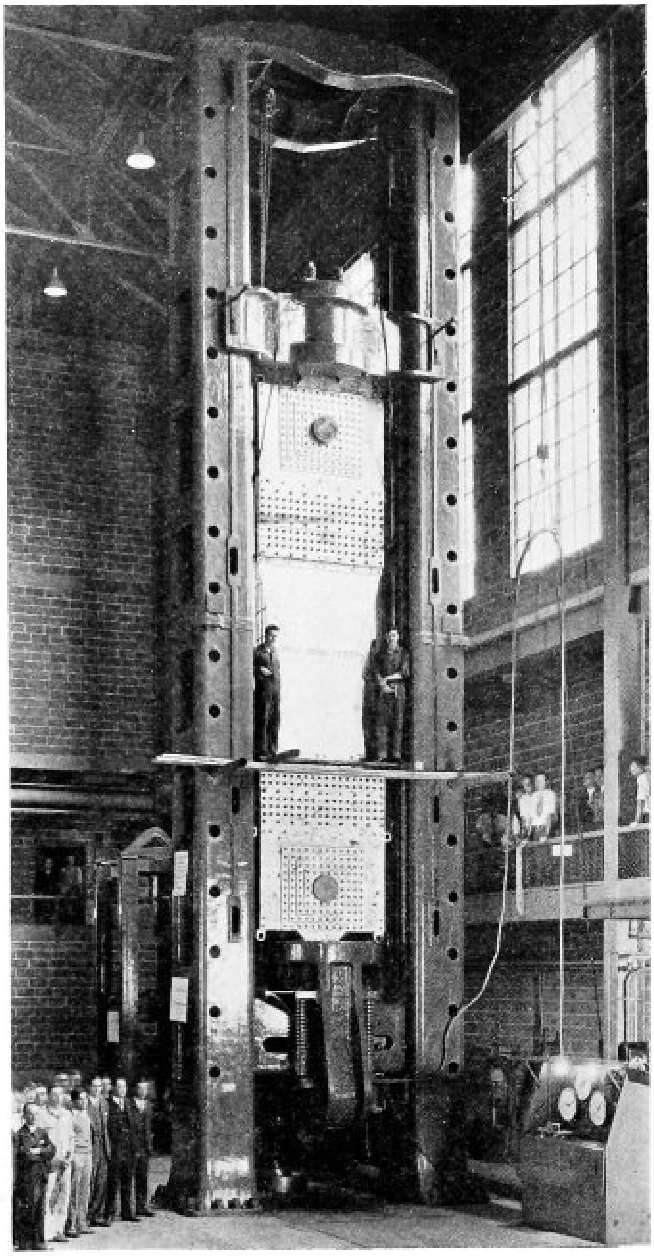


Fig. 1.—Three-million-pound testing machine with specimen bolted to special pulling heads

During the past year the Engineering Experiment Station of the University of Illinois, Urbana, Ill., has carried out investigations to determine the efficiency of various types of joints in wide plates as used in the construction of storage tanks for water and oil. These experiments, conducted by Professor Wilbur M. Wilson, James Mather and Charles O. Harris, were made possible through the co-operation of the Chicago Bridge & Iron Works, Chicago, Ill. The tests include investigations of riveted lap and butt joints; joints for which the holes were drilled and others for which they were punched; joints with rivets in single shear and others with rivets in double shear; welded joints, both lap and butt; joints having rivets supplemented with welds; and wide plates having no joints. Only the tests pertaining to wide plates having no joints are dealt with in this article, other sections of the investigations to follow at a later date.

The section of the plates varied from 40 inches by ¼ inch and 20½ inches by ½ inch to 72 inches by ⅝ inch. In addition to determining the ultimate strength of the specimens, measurements were taken to determine variations in strain, in the plate along the seam and in the case of riveted joints, to determine the slip of the rivets at various points. The physical properties of the material from which the joints were made were determined from standard tension tests of pieces cut from the plates used in making the specimen.

The tests on wide plates without joints were divided into two series. The smaller specimens of the 1928 series were tested in a 600,000-pound Riehle testing machine and the larger specimens of the 1930 series were tested in a 3,000,000-pound Southwark-Emery machine. Each machine was equipped with special pulling heads each of which consisted of a pulling bar that is gripped in the jaws of the testing machine at one end and that is pin-connected on the other end to a built-up clevis consisting of wide plates containing holes to match corresponding holes in the ends of the specimen. The specimen was fastened between the two plates forming the clevis with machine bolts. The pin was provided to insure centric loading and the wide rigid clevis distributed the load across the width of the plate, that is, along the length of the joint. The pulling heads for the 3,000,000-pound machine are attached to a specimen as shown in Fig. 1.

The strain in the plate was measured with an 8-inch Berry strain gage, measurements being taken on vertical gage lines at a number of points across the plate, on both sides and above and below the joint.

The slip, for the 1928 series, also was measured with a Berry strain gage. The holes for measuring the slip are shown in Fig. 2. Gage holes *b* and *c* are in the far plate and hole *a* is in the near plate. Changes in the distance from *c* to each of the holes *a* and *b* were measured. The difference between the change in *b*-*c* and the change in *a*-*c* is the slip between the plates. Slip readings were taken at from eight to ten points along the joint, half of the readings being taken through the large holes *D* in the near plate and the others through the hole *E* in the far plate.

The slip, for the 1930 series of plates, was measured with the instrument shown in Fig. 3. It consists of a taper pin that fits into a taper hole in the plates, and an Ames dial that indicates the distance the taper pin projects from the plate. The method of using the instrument is as follows: The taper pin is inserted and a zero reading is taken to determine the projection of the pin before the joint is loaded. The pin is again inserted and its projection measured after an increment of load is applied. If no slip has occurred the pin will project the same amount as originally, but if one plate

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* Published through the courtesy of the Engineering Experiment Station, University of Illinois, Urbana, Ill.

has slipped relative to the other the pin will project farther than before. The change in projection and the taper of the pin determine the slip that occurs. The pins had a diameter of about ¼ inch and a taper (change in diameter) of approximately 0.02 inch per inch. There was no difficulty in obtaining successive readings that checked when a tolerance of 0.0001 inch was allowed, the original set of readings being completed before the check readings were begun. The instrument was checked against a standard, consisting of a fixed pin projecting from a finished surface of a steel block, before each set of readings was taken. This instrument is both sensitive and reliable. The ease with which the specimen is prepared for the use of the instrument makes it especially satisfactory.

The bend in the plate was measured with the instrument shown in Fig. 4.

The general location of the points at which strain, slip, and bending were measured, for the 1928 series, is shown in Fig. 5. The points at which observations were made for the 1930 series were similarly located.

A typical specimen of the 1928 series for determining the strength of wide plates without joints is shown in Fig. 6. The ends were designed to slip between the two side plates forming the clevis of the pulling head, and were fastened to the clevis by means of bolts engaging the holes shown. The ends of the specimen were reinforced with side plates riveted and welded to the main plates as indicated.

The physical properties of the material were determined for each specimen from standard tension tests of control specimens cut from the parent plate. The dimensions of the individual specimens, the physical properties of the material, and the location in the parent plate of the main plates and the control specimens are given in the sections describing the tests.

Tests were made to determine the strength of a wide specimen relative to the strength of a standard tension specimen, 1½ inches wide, from the same plate. Four types of specimens were used, differing in width, thickness, and shape. Three specimens of each of three types and six specimens of one type were tested. The geometrical properties of the various types are given in Fig. 7.

The parent plate was one foot wider than the finished specimen and a strip 6 inches wide was cut from each edge to be used in determining the physical properties of the material. Two standard flat tension specimens, 1½ inches wide over the middle portion, were cut from each of these strips. The physical properties of the material obtained from the control specimens are given in Table 1, the results of the tests of the wide specimens are given in Table 2, and the manner of failure of the various specimens is given in Table 3. The unit strength of the wide plate relative to the unit strength of the material as determined from the control specimens 1½ inches wide is designated as the efficiency of the wide plate. The efficiency has been computed on two bases, on unit area and on unit width. Because the edges of the plates, where the control specimens were obtained, were thinner than the middle, the efficiency based upon the width is greater than that based upon the area. The latter, of course, is the true value. The two values of the efficiency are given in the last two columns of Table 2.

The lowest efficiency of any one plate is 85.53 percent for SDB, based upon the width. (Thicknesses were not measured for this group. The lowest efficiency for any one group is for SDA, SDB, and SDC, for which the average efficiency is 88.15 percent, based upon the width. The SDD, SDE, and SDF group is a

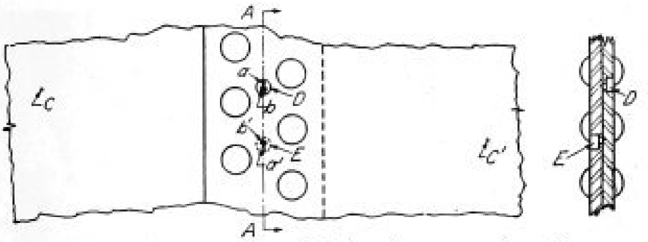


Fig. 2.—Arrangement of holes for measuring slip

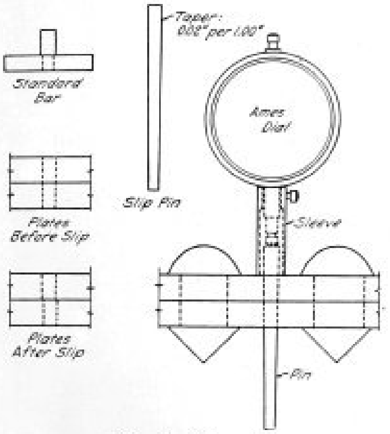


Fig. 3.—Gage for measuring slip

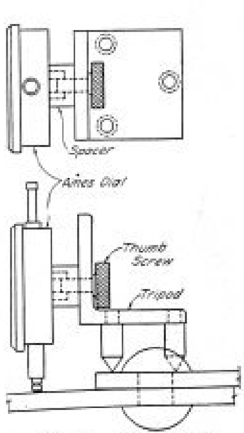


Fig. 4.—Gage for measuring bend

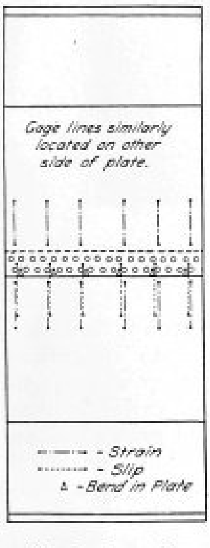


Fig. 5.—Location of strain, slip and bending points

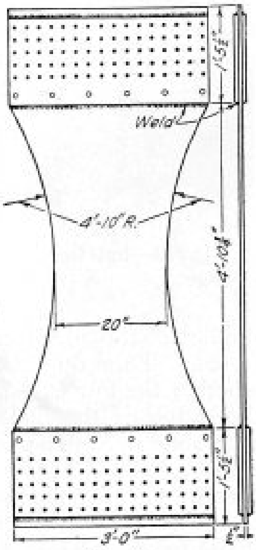
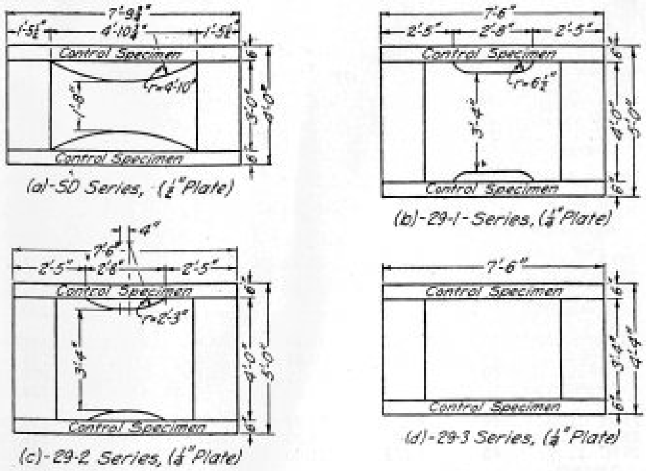


Fig. 6.—Typical specimen



(a)-50 Series, (½" Plate)

(b)-29-1-Series, (¼" Plate)

(c)-29-2 Series, (¼" Plate)

(d)-29-3 Series, (¼" Plate)

Fig. 7.—Specimens for tests of wide plates

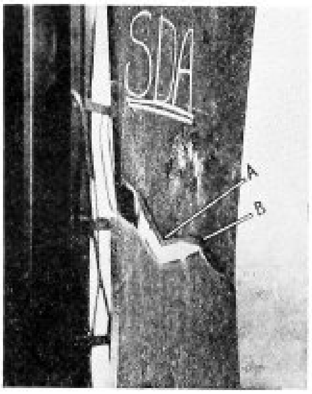


Fig. 8.—Specimen SDA

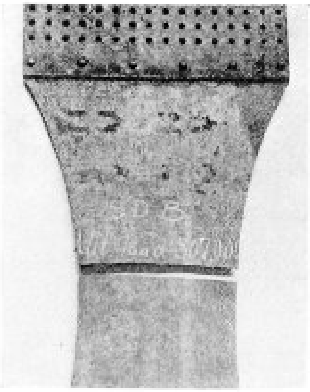


Fig. 9.—Specimen SDB

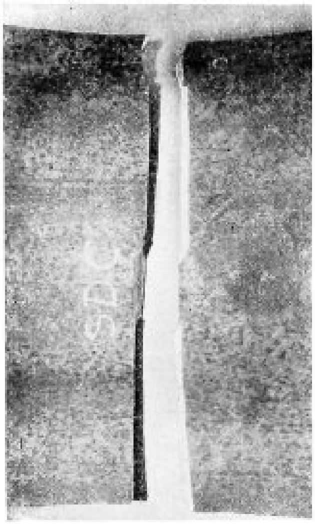


Fig. 10.—Specimen SDC

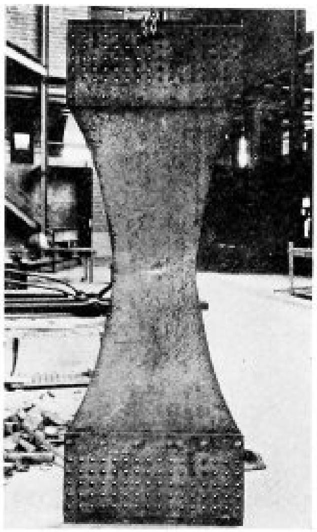


Fig. 11.—Specimen SDF

Specimens after failure

Table 1.—Physical Properties of Steel as Given by Tests of Control Specimens

Each value is the average of four tests

Specimen No.	Reduction of Area percent	Elongation in 8 inches percent	Yield Point pounds per sq. in.	Ultimate Strength	
				Pounds per in. width	Pounds per sq. in.
SDA	58.9	30.00	32,066	27,334	55,064
SDB	57.0	29.28	31,823	28,917	57,806
SDC	53.8	28.61	33,022	30,189	61,467
SDD	64.25	34.07	29,292	27,051	53,104
SDE	64.65	34.28	28,925	26,877	52,129
SDF	62.65	33.07	29,123	26,731	53,316
291A	65.68	32.63	34,336	13,303	52,579
291B	63.48	31.27	34,072	13,513	53,097
291C	61.58	31.53	34,542	14,078	53,248
292A	57.82	29.60	36,347	13,387	53,761
292B	57.39	32.47	34,128	13,135	51,817
292C	61.84	31.72	36,545	14,102	54,712
293A	62.35	32.19	33,463	13,798	52,558
293B	59.77	31.41	36,738	13,420	53,899
293C	62.47	32.25	33,640	13,551	52,991

duplicate set made after all of the other groups had been tested. This duplicate set was made to determine whether the low efficiency of the first group was due to the shape of the plates or whether the variation was an erratic one. The fact that the second group had an efficiency of 97.20 percent based upon width and 94.69 percent based upon area, indicates that the low efficiency of the first group was not necessarily due to the shape of the plate. The efficiencies based upon areas of the four last groups, the average of the tests in each group were 94.69, 93.48, 98.26, and 90.15 percent, respectively. Assuming the relation between the efficiency based upon area and that based upon width, obtained from group 2, to hold for group 1, the efficiencies, based upon area, for SDA, SDB, and SDC would be 85.34, 83.31, and 88.94 percent, respectively, with an average value of 85.86 percent. The strongest group, 292 A-B-C, had an average efficiency, based upon area, of 98.26 percent. The efficiencies of the two 64-inch by ⅝-inch plates, AW1 and AW2, were 96.4 and 99.4 percent, respectively.

The two groups of SD specimens had the same outline but the average efficiency for the two differed 9 percent. There were two differences between the two groups of specimens. The first difference was in the material, that for the first group being a little stronger and somewhat less ductile than that for the second group. The second difference between the two groups was in the manner in which the specimens were prepared. Specimens SDA, SDB, and SDC were cut to size with a flame, SDB being tested as received from the welder, whereas the edges of SDA and SDC were smoothed somewhat with a file. The edges of specimens SDD, SDE, and SDF were cut with a flame and then machined so that all of the heated material was removed and the finished edge was free from the indentations characteristic of SDA and SDC shown in Fig. 8.

The following tests were made to determine whether flame cutting injures a plate: The parent plate was cut into four strips by burning. Specimens A2 and A3 were tested as finished with the torch, except that projecting beads were knocked off. Specimens A1 and A4 were machined to the finish lines, all materials that

Table 2.—Strength of Wide Plates Without Joints, 1929-30 Series

Specimen No.	Width of Plate in.	Thickness of Plate in.	Area of Section sq. in.	Elongation in 8 in. percent	Reduction of Area percent	Ultimate Load pounds	Ultimate Strength of Plate		Ultimate Strength, Control Specimen		Efficiency of Plate based on	
							Pounds per in. width	Pounds per sq. in.	Pounds per in. width	Pounds per sq. in.	Width	Area
SDA	20½	....	....	35.1	...	491,000	23,951	......	27,334	55,064	87.62	....
SDB	20½	....	....	...	...	507,000	24,732	......	28,917	57,806	85.53	....
SDC.	20½	....	....	23.5	...	565,000	27,561	......	30,187	61,467	91.31	....
Average..						521,000	25,415	......	28,813	58,112	88.15	....
SDD	20	0.531	10.62	39.0	45.3	525,000	26,250	49,435	27,051	53,104	97.04	93.09
SDE	20	0.525	10.50	38.1	42.7	523,000	26,150	49,810	26,877	52,129	97.30	95.55
SDF	20	0.511	10.22	43.0	25.7	520,000	26,000	50,881	26,731	53,316	97.27	95.43
Average..						522,667	26,133	50,042	26,886	52,850	97.20	94.69
291A	40	0.273	10.92	25.9	33.1	531,700	13,293	48,692	13,303	52,579	99.93	92.60
291B	40	0.270	10.80	21.5	33.0	526,000	13,150	48,704	13,513	53,097	97.31	91.73
291C	40	0.275	11.00	17.5	29.5	563,000	14,075	51,182	14,078	53,248	99.98	96.12
Average..						540,233	13,506	49,525	13,631	52,975	99.07	93.48
292A	40	0.266	10.64	36.4	45.2	561,000	14,025	52,726	13,387	53,761	104.77	98.07
292B	40	0.271	10.84	37.8	...	553,000	13,825	51,015	13,135	51,817	105.25	98.45
292C	40	....	....	32.5	...	576,000	14,400	......	14,102	54,712	102.11	....
Average..						563,333	14,083	51,870	13,542	53,430	104.04	98.26
293A	40	0.273	10.92	18.9	48.2	530,500	13,263	48,581	13,798	52,558	96.12	92.43
293B	40	0.270	10.80	13.1	31.8	539,000	13,475	49,907	13,420	53,899	100.19	92.59
293C	40	0.275	11.00	11.8	19.3	498,000	12,450	45,273	13,551	52,991	91.88	85.44
Average..						522,500	13,063	47,920	13,590	53,149	96.06	90.15

Table 3.—Manner of Failure of Wide Plates

Specimen No.	Description of Failure
SDA......	Plate failed by tearing between points A and B, Fig. 8. The load was left on the specimen and four hours after initial failure the final failure occurred as shown in the figure.
SDB......	The plate failed suddenly over the entire section and without warning. The character of the fracture is shown in Fig. 9, a characteristic fracture for a brittle material.
SDC......	The plate failed suddenly across the entire section but there was a considerable elongation of the plate and the fracture, shown in Fig. 10, is a characteristic fracture for a ductile material.
SDD......	Plate failed by tearing with a large elongation and reduction of area.
SDE......	Plate failed by tearing with a large elongation and reduction of area.
SDF......	Plate failed by opening up at the center as shown in Fig. 11. The tear began at the middle of the plate and was halted by releasing the load before it had extended to the edges.
291A...... 291B...... 291C......	All plates failed by tearing at the end of the reduced section similar to the failure of 291B shown in Fig. 12.
292A...... 292B......	The plates failed by a crack opening near the center as shown in Fig. 13.
292C......	The plate failed by tearing from one edge, as shown in Fig. 14.
293A...... 293B......	Both plates tore next to the reinforcing plates at the ends where there is an abrupt change in section.
293C......	The initial tear occurred near the reduced section adjoining the end reinforcement followed by the small tear at the edge near the mid-length of the plate, as shown in Fig. 15.

was heated being removed. Specimens B1, B2, B3, and B4 were cut from a second plate in the same manner. The results of the tests are given in Table 4. These tests indicate that flame cutting increases the strength of narrow specimens slightly and reduces their ductility. The relative effect on a specimen 20 inches wide would not appear to be appreciable, and it is believed that the difference in behavior between the two groups is not due to the manner in which the edges of the plate were cut.

Diagrams showing the variation in strain across the plates indicate that the stress distribution was fairly uniform.

The efficiency of the wide plates based upon areas was 90 percent or more, except for four of the fifteen specimens tested. The ductility, although not as great as for standard tension specimens, was fair for most specimens, but on one there was practically no reduction of area or elongation.

The brittle fracture of specimen SDB is hard to explain, especially since the control tests for this specimen, reported in Table 1, showed the material to be very ductile. In order to check the physical properties of the material from this specimen, a second set of controls was cut from the main specimen after it had been tested to destruction. These controls had approximately the same properties as the original set reported in Table 1.

Summarizing the tests made on continuous wide plates without joints, these plates may be expected to develop a unit stress equal to approximately 90 percent of the unit strength of the control specimens (tension specimens, 1½ inches wide). The plates tested were 20 inches wide by ½ inch thick, 40 inches by ¼ inch, and 64 inches by ⅝ inch. Two of the fifteen plates tested developed a unit stress less than 86 percent of the strength of the control specimens.

Table 4.—Effect of Flame Cutting Upon Properties of Steel Plates

Specimen No.	How Cut	Reduction of Area percent	Elongation in 8 in. percent	Yield Point pounds per sq. in.	Ultimate Strength pounds per sq. in.
A1	Machined	60.0	27.75	37,200	62,400
A4	Machined	57.9	30.25	36,400	62,300
Average		59.0	29.00	36,800	62,350
A2	Flame-Cut	42.7	20.38	41,100	64,700
A3	Flame-Cut	49.7	21.25	39,900	63,800
Average		46.2	20.81	40,500	64,250
B1	Machined	56.4	33.13	37,100	59,100
B4	Machined	60.4	31.25	37,600	60,900
Average		58.4	32.19	37,350	60,000
B2	Flame-Cut	46.2	20.50	40,800	63,500
B3	Flame-Cut	42.4	24.38	40,500	63,200
Average		44.3	22.43	40,650	63,350

The results of tests on the main specimens which include investigations of riveted lap and butt joints will be presented in a later issue. Other tests, including those on joints for which the holes were drilled and others for which they were punched, joints with rivets in single shear and others with rivets in double shear, welded lap and butt joints and joints having rivets supplemented with welds, will follow. In the articles that will follow the strength of the plate has been taken equal to the strength of the specimen in the discussion of the efficiency of the joints, the theoretical efficiency being the ratio of the net to the gross width of the plate.

The fact, that the unit strength of the control specimens does not equal the unit stress of the wide plate, points out that plate efficiency based on tension specimens is apt to lead to erroneous results. Due allowance for this discrepancy must be taken in proper design work.

(To be continued)

Fig. 12.—Specimen 291B

Fig. 13.—Specimen 292B

Fig. 14.—Specimen 292C

Fig. 15.—Specimen 293C

Specimens after failure

The Effect of Locomotive Boiler Capacity on Railroad Revenue*

The effect of boiler capacity on the revenue-earning abilities of locomotives can be predetermined by analyzing their performance by means of pull-speed curves. The importance of boiler design and efficiency has become increasingly evident with the growing practice of American railroads to furnish high-speed, scheduled freight service to their customers. The old drag freight is seldom seen today, while freight trains running at passenger train speeds on fixed schedules are common. Some roads have even given names to these scheduled freights, as has long been the custom with crack passenger trains.

The eight-coupled freight locomotives generally used for this fast service have developed from the standard 2-8-0 freight engine of twenty years ago, through the 2-8-2, 4-8-2 and 2-8-4 types to the 4-8-4 body type of today. Wheel diameters have been steadily increased to as much as 76 inches, and the present truck arrangements have been made necessary by the constantly increased boiler capacity required. Other types of heavy duty fast freight locomotives, such as the 2-6-6-2 and the 2-8-8-2, have been and are being developed, but at present the 4-8-4 type is ably fulfilling the needs of this service on many roads.

Passenger locomotives have followed the same trend from the American (4-4-0) type, through the 4-4-2, 4-6-0, 4-6-2, 4-6-4, and 4-8-2 types to the heavy duty, high-speed 4-8-4 type, which is identical with the high-speed freight engine of today. This type is, therefore, a general service engine for all kinds of heavy tonnage, high-speed trains.

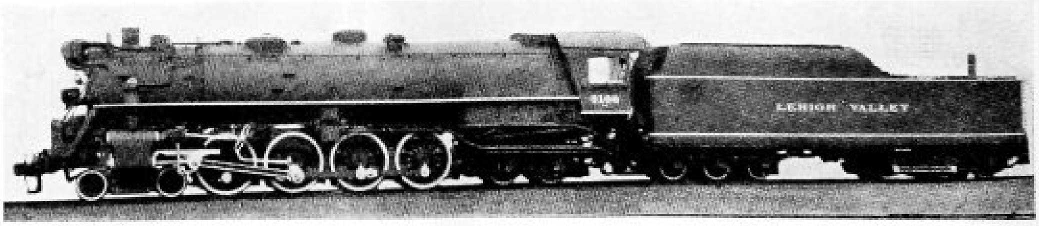
In analyzing motive power for purposes of tonnage rating or design, it is necessary to know what tonnages locomotives are capable of hauling at various speeds, and particularly at high speeds. A pull-speed curve shows these facts graphically. Such a curve can be determined actually for any locomotive with the aid of a dynamometer car which measures the actual drawbar pull at the rear of the tender, or it can be calculated from the dimensions of the locomotive.

It is necessary to have a pull-speed curve for the reason that a locomotive will develop its rated tractive force only at starting speeds. As its speed increases, the available tractive force falls off until a point is reached at which the boiler can no longer supply the steam required by the cylinders at full stroke. If greater speed is desired, it is necessary to use the steam more efficiently, and this is accomplished by shortening the cut-off, or in railroad language hooking back the reverse lever. This procedure shortens the percentage of the stroke during which steam is admitted to the cylinders, and during the balance of the stroke the expansion of the steam from boiler pressure towards atmospheric pressure does the work. As the cut-off is shortened, the available tractive force falls off more rapidly.

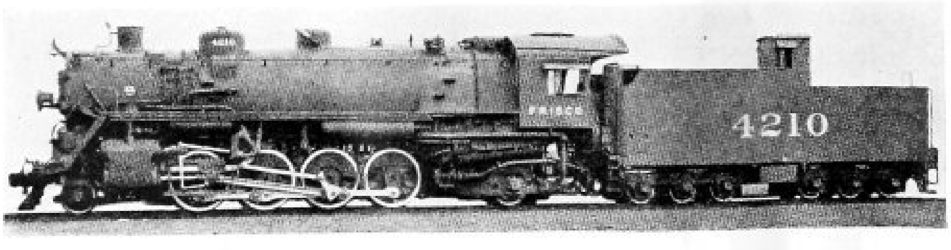
It is, therefore, evident that the rated tractive force of a locomotive, figured from its design dimensions, is of no value except at starting speeds. The steaming capacity of the boiler is the limiting factor at higher speeds. Therefore, the development of a pull-speed curve requires consideration of the boiler design and efficiency.

This article describes a method for determining pull-speed curves by calculation. The calculations are described in detail in order to serve as a guide for any

Philadelphia & Reading railroad locomotive of the 2-8-0 type



Representative type 4-8-4 locomotive of the Lehigh Valley



Frisco 2-8-2 type locomotive studied for efficiency of operation

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* From an article by Ralph P. Johnson in the July issue of *Baldwin Locomotives.*

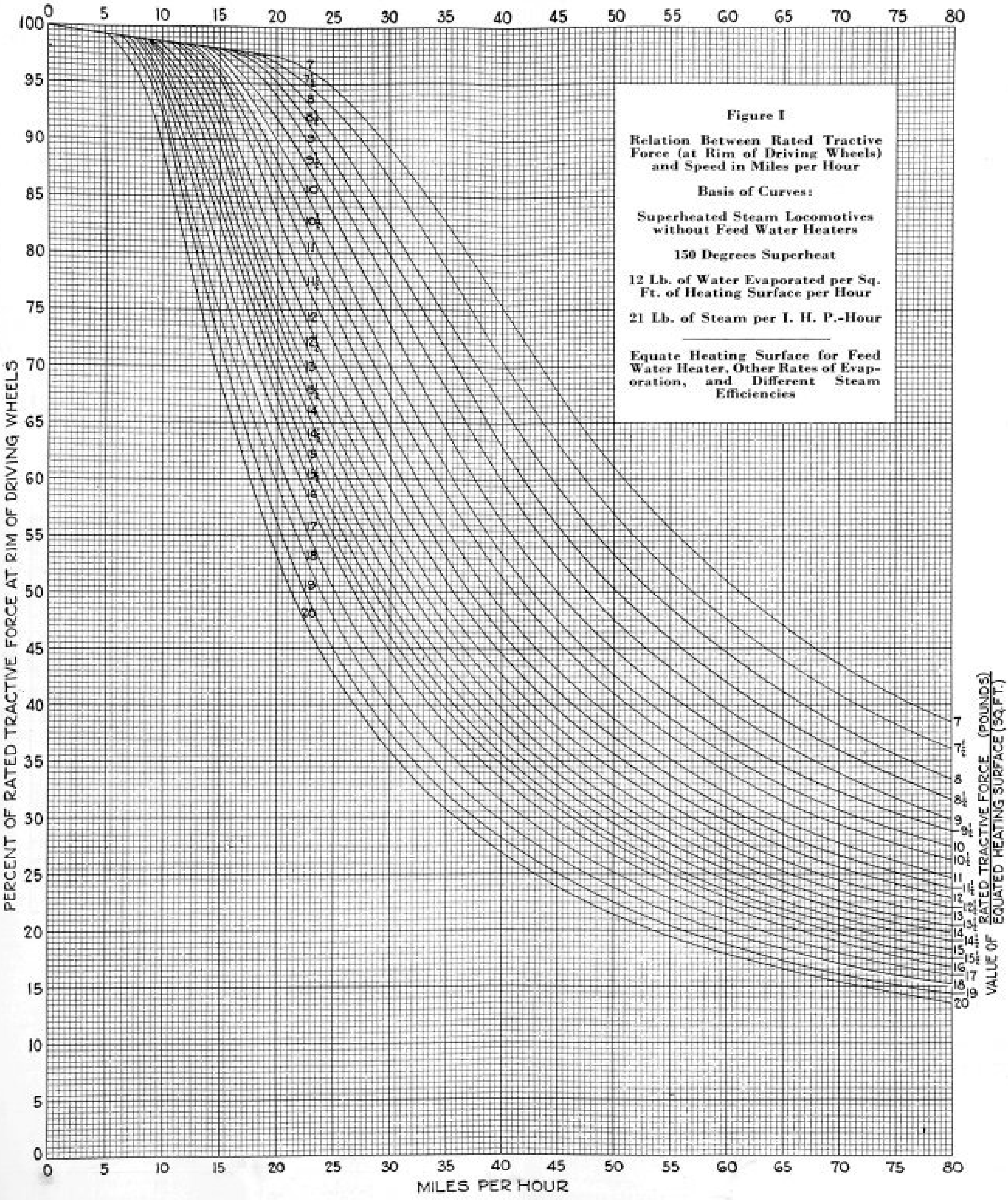


Figure I

Relation Between Rated Tractive Force (at Rim of Driving Wheels) and Speed in Miles per Hour

Basis of Curves:

Superheated Steam Locomotives without Feed Water Heaters

150 Degrees Superheat

12 Lb. of Water Evaporated per Sq. Ft. of Heating Surface per Hour

21 Lb. of Steam per I. H. P.-Hour

Equate Heating Surface for Feed Water Heater, Other Rates of Evaporation, and Different Steam Efficiencies

one desiring to analyze motive power on this basis. In order to make the method clearer, the calculations are carried through for a particular locomotive.

At starting speeds, a locomotive will develop, at the rim of the driving wheels, the rated tractive force, which is calculated from the dimensions of the engine by the formula:

$$T = \frac{0.85P \times C^2 \times S}{D}$$

Where T = Rated tractive force in pounds
P = Boiler pressure in pounds per square inch
C = Diameter of cylinders in inches
S = Stroke in inches
D = Driving wheel diameter in inches

As the speed increases, a point is reached where the boiler cannot supply steam enough for the cylinders at full stroke, and the cut-off must be reduced. It follows from this that tractive force is dependent upon the steaming capacity of the boiler as well as the dimensions of the engine. For practical purposes it is assumed that the steaming capacity is directly proportional to the heating surface, and that sufficient fuel can be fired to push the steam production to the limit set by the heating surface.

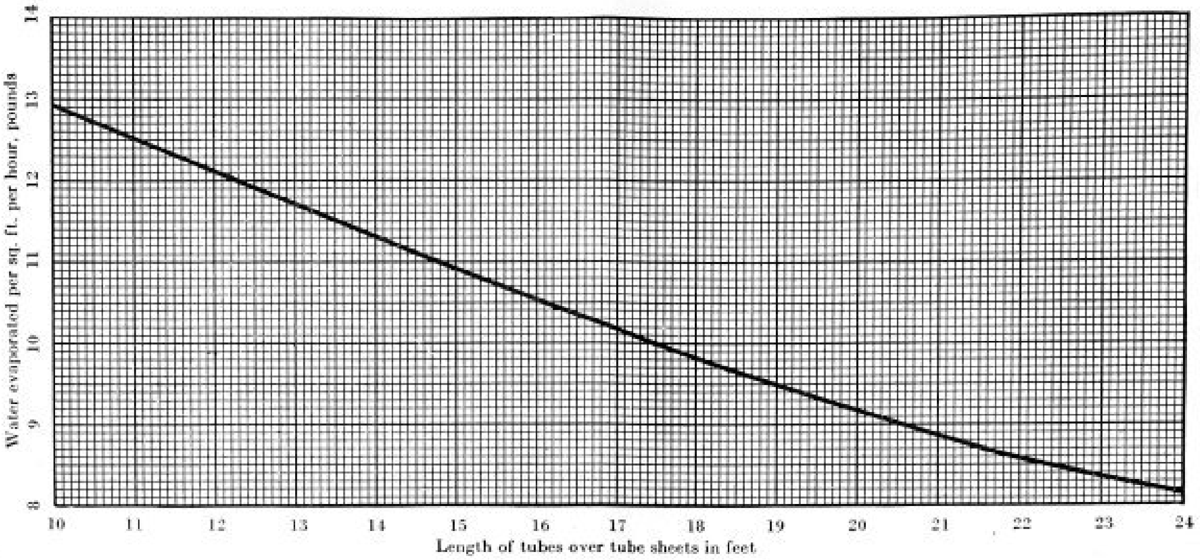


Fig. 2.—Factors of evaporation for tubes of various lengths

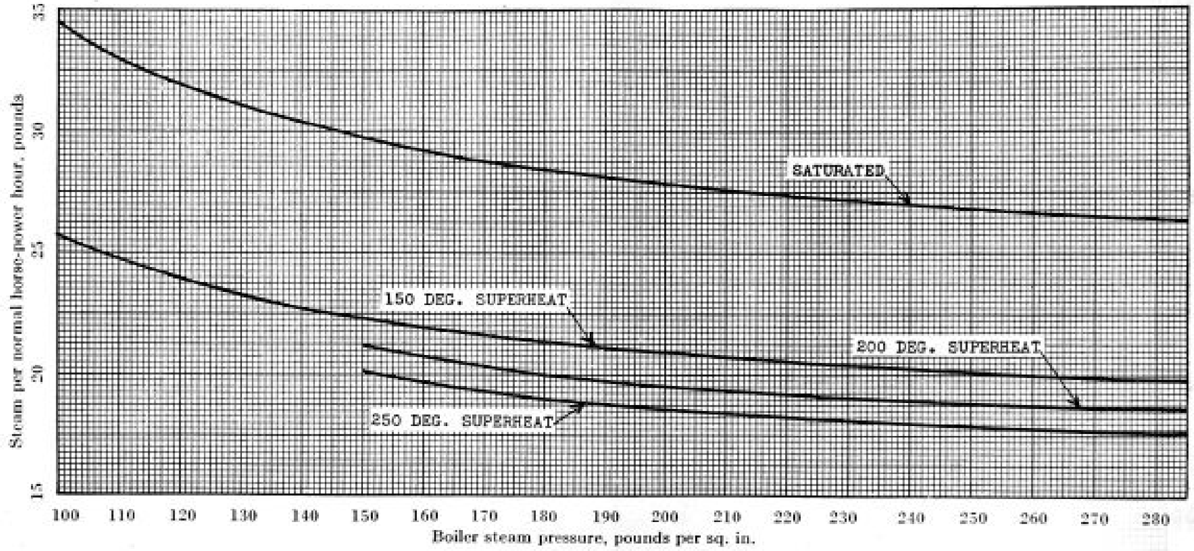


Fig. 3.—Factors for determining normal horsepower

The tractive force available at any speed will, therefore, depend upon the relation between the rated tractive force and the total heating surface. This relation can be obtained for any efficiency in the use of steam and at any rate of evaporation of water per square foot of heating surface.

The tractive forces developed at various speeds by locomotives of different proportions are shown in Fig. 1, as percentages of rated tractive force. These curves are calculated on the basis of evaporating 12 pounds of water per hour per square foot of heating surface, developing 150 degrees superheat and consuming 21 pounds of steam per indicated horsepower hour. They apply to locomotives not equipped with feed-water heaters. The lower portions of the curves are calculated, while the upper portions are made by drawing smooth curves starting at 100 percent tractive force and meeting the calculated curves at from 20 to 45 miles per hour, depending upon the value of the ratio of tractive force to heating surface. The calculated portions of the curves are based on the following formula:

$$\text{Percent of rated tractive force at given speed} =$$

$$\frac{\text{Pounds tractive force per H.P. at given speed} \times \text{H.P.}}{\text{Rated tractive force}} =$$

$$\frac{\dfrac{375}{\text{Miles per hour}} \times \dfrac{\text{Heating surface} \times 12}{21}}{\text{Rated tractive force}} =$$

$$\frac{\dfrac{375}{\text{Miles per hour}}}{\dfrac{\text{Tractive force}}{\text{Heating surface}} \times 1.75}$$

For any locomotive meeting the above specifications, the percent of rated tractive force at any speed can be read directly from the curves, using the ratio line obtained by dividing the rated tractive force in pounds by the total evaporative area in square feet.

For locomotives having different characteristics than those on which the curves are based, it is necessary to divide the rated tractive force by an equated heating surface to obtain the correct ratio line. By an equated heating surface is meant the amount of heating surface which would produce the same results if the locomotive did not have a feed-water heater, used 21 pounds of steam per indicated horsepower hour, and evaporated 12 pounds of water per square foot of heating surface.

As an example, consider an engine of the following characteristics:

Rated tractive force, pounds	66,400
Steam pressure, pounds per square inch	250
Degrees superheat	250
Feed-water heater	Yes
Direct heating surface, square feet	490
Indirect heating surface, square feet	4,932
Total heating surface, square feet	5,422
Length of tubes, feet	21.5

$$\text{Equated heating surface} = \text{Total heat. surf.} \times \frac{\text{Aver. evap. per sq. ft. of heat. surf.}}{\text{Evap. per sq. ft. of heat. surf. used for curves}} \times \frac{\text{lb. steam per I.H.P.-hr. used for curves}}{\text{lb. steam per I.H.P. for press. and deg. superheat}} \times$$

108%. (Correction for feed-water heater.)

Equated heating surface =

$$5422 \times \frac{12.9}{12} \times \frac{21}{18} \times \frac{108}{100} = 7344 \text{ square feet}$$

and the ratio for use in connection with the curves is

$$\frac{66{,}400}{7344} = 9.03$$

In the above calculations the average evaporation per square foot of heating surface is determined by dividing the rated evaporation per hour by the total evaporative surface.

The rated evaporation per hour is the sum of the evaporation from the direct and indirect heating surfaces.

The direct heating surface is supplied by the firebox, combustion chamber, syphons and arch tubes. It is assumed that this area will evaporate 55 pounds of water per square foot per hour. Therefore, the evaporation per hour from the direct heating surface will be equal to the area multiplied by the constant 55.

The direct heating surface is supplied by the firebox, tubes and flues. The average evaporation per square foot from this area varies with the length of the tubes. Fig. 2 shows the evaporation per square foot or evaporative factor, for tubes of different lengths, and the water evaporated per hour is, therefore, equal to the area of the indirect heating surface multiplied by the evaporative factor for tubes of the given length.

The equation for the rated evaporation per hour is:

> Pounds of water evaporated per hour = Direct heating surface in square feet × 55 + indirect heating surface in square feet × tube evaporative factor

The steam used per indicated horsepower hour is obtained directly from Fig. 3, for the given steam pressure and degrees of superheat.

The degrees of superheat depend upon the type of superheater. For type E, the superheat is assumed to be 250 degrees F.; for the revised type A (where the number of tubes divided by the number of superheater flues is less than six), 200 degrees F.; and for the original type A (where the number of tubes divided by the number of superheater flues is greater than 6), 150 degrees F.

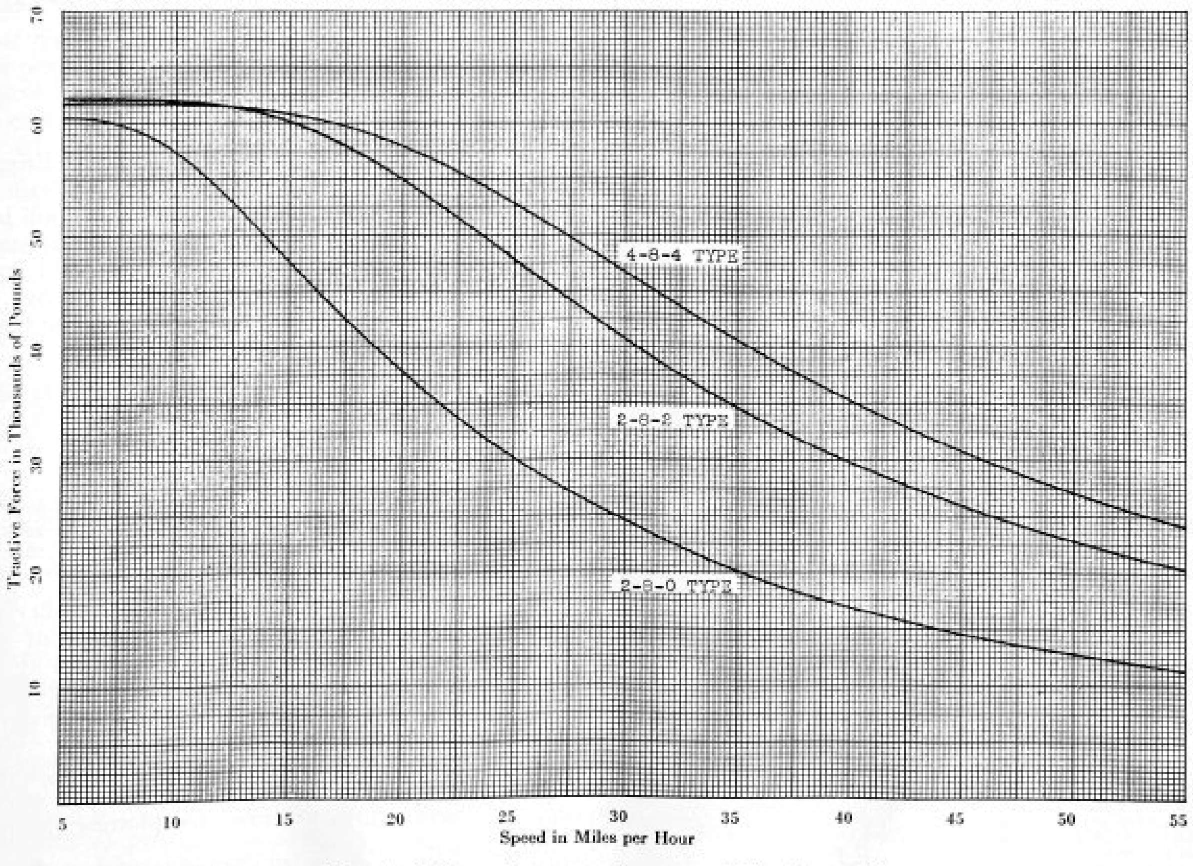


Fig. 4.—Pull-speed curves of representative locomotives

Table 1—Resistance of Freight Cars, Tenders and Locomotive Trucks on Straight Level Track in Pounds per Ton

Based on Formula $R = 1.5 + \frac{106 + 2V}{W + 1} + 0.001\ V^2$

Weight of Vehicle in Tons	RESISTANCE IN POUNDS PER TON AT SPEEDS IN MILES PER HOUR													
	5	10	15	20	25	30	35	40	45	50	55	60	65	70
15	8.8	9.5	10.3	11.0	11.9	12.8	13.7	14.7	15.8	16.9	18.0	19.2	20.5	21.8
20	7.0	7.6	8.2	8.9	9.6	10.3	11.1	12.0	12.9	13.8	14.8	15.9	17.0	18.1
25	6.0	6.5	7.0	7.5	8.1	8.8	9.5	10.3	11.1	11.9	12.8	13.8	14.8	15.9
30	5.3	5.7	6.1	6.6	7.2	7.8	8.4	9.1	9.9	10.7	11.5	12.4	13.3	14.3
35	4.7	5.1	5.5	6.0	6.5	7.0	7.6	8.3	9.0	9.7	10.5	11.4	12.3	13.2
40	4.4	4.7	5.0	5.5	6.0	6.5	7.0	7.6	8.3	9.0	9.8	10.6	11.5	12.4
45	4.0	4.3	4.7	5.1	5.5	6.0	6.6	7.2	7.8	8.5	9.2	10.0	10.9	11.8
50	3.8	4.1	4.4	4.8	5.2	5.7	6.2	6.8	7.4	8.0	8.8	9.5	10.4	11.2
60	3.4	3.7	4.0	4.3	4.7	5.1	5.6	6.2	6.7	7.4	8.1	8.8	9.6	10.4
75	3.0	3.3	3.5	3.8	4.2	4.6	5.1	5.6	6.1	6.7	7.4	8.1	8.8	9.6
85	2.9	3.1	3.3	3.6	3.9	4.3	4.8	5.3	5.8	6.4	7.0	7.7	8.5	9.3
100	2.7	2.9	3.1	3.3	3.7	4.0	4.5	4.9	5.5	6.0	6.7	7.3	8.1	8.8
115	2.5	2.7	2.9	3.2	3.5	3.8	4.2	4.7	5.2	5.8	6.4	7.0	7.8	8.5
130	2.4	2.6	2.8	3.0	3.3	3.7	4.1	4.5	5.0	5.6	6.2	6.8	7.5	8.3
145	2.3	2.5	2.7	2.9	3.2	3.5	3.9	4.4	4.9	5.4	6.0	6.7	7.3	8.1
160	2.2	2.4	2.6	2.8	3.1	3.4	3.8	4.3	4.7	5.3	5.9	6.5	7.2	7.9
175	2.2	2.3	2.5	2.7	3.0	3.3	3.7	4.2	4.6	5.2	5.8	6.4	7.1	7.8

Table 2.—Resistances and Drawbar Pulls, in Pounds, at Various Speeds

Speed, Miles per Hour	5	10	15	20	25	30	35	40	45
Machine resistance of locomotive	3375	3375	3375	3375	3375	3375	3375	3375	3375
Resistance of engine truck	157	168	182	193	209	228	245	267	301
Resistance of trailer truck	178	191	208	228	249	270	286	311	369
Resistance of tender	403	421	458	494	549	604	677	769	952
Total loco. and tender resistance	4113	4155	4223	4290	4382	4477	4583	4722	4997
Tractive force	65736	65470	65072	62416	57104	51128	45152	39508	33532
Pull back of tender = Tractive force less total resistance	61623	61315	60849	58126	52722	46651	40569	34786	28535

Table 3.—Characteristics of Locomotives

Type	2-8-0	2-8-2	4-8-4
Rated tractive force at rim of drivers, lb.	64,400	66,700	66,400
Diameter of driving wheels, inches	55½	63	70
Cylinder dimensions, inches	25 x 32	27 x 32	27 x 30
Boiler pressure, lb. per sq. in.	210	230	250
Type of superheater	A	E	E
Feed-water heater	No	Yes	Yes
Grate area, sq. ft.	94.9	80.3	88.0
Tubes, length, ft.	13½	18	21½
Heating surface, sq. ft.—Firebox	225	234	268
Heating surface, sq. ft.—Comb. chamber	71	56	79
Heating surface, sq. ft.—Arch tubes	....	14	.....
Heating surface, sq. ft.—Syphons	.....	86	143
Total direct heating surface, sq. ft.	296	390	490
Evap. from direct heat. surf., lb. per hr.	16,280	21,450	26,950
Heating surface of tubes, sq. ft.	2,369	3,994	4,932
Factor of evaporation for tubes (Fig. II)	11.5	9.8	8.7
Evap. from indirect heat. surf., lb. per hr.	27,244	39,141	42,908
Rated evaporation per hour, lb.	43,524	60,591	69,858
Evap. corrected for feed-water heater, lb. per hr.	43,524	65,438	75,447
Total evaporative heating surface, sq. ft.	2,665	4,384	5,422
Steam per I.H.P.-hr., lb. (Fig. III)	19.5	18.2	18.0
Ave. evaporation per sq. ft. heat. surf., lb. per hr.	16.3	13.8	12.9
Equated heating surface, sq. ft.	3,871	6,269	7,344
Rated tract. force ÷ equat. heat. surf.	16.6	10.63	9.03
Boiler or potential horsepower	2,232	3,595	4,191
Weight in tons—on drivers	128.3	137.4	135.0
Weight in tons—on front truck	13.4	18.5	27.5
Weight in tons—on back truck	.....	32.0	41.5
Weight in tons—tender	94.4	116.5	183.1
Weight in tons—total engine and tender	236.1	304.4	387.1
Pull back of tender at 5 m.p.h., lb.	60,400	62,200	61,800
Pull back of tender at 10 m.p.h., lb.	57,500	61,800	61,400
Pull back of tender at 15 m.p.h., lb	47,200	60,300	60,900
Pull back of tender at 20 m.p.h., lb.	37,300	54,500	58,100
Pull back of tender at 25 m.p.h., lb.	29,800	47,400	52,700
Pull back of tender at 30 m.p.h., lb.	24,100	40,600	46,700
Pull back of tender at 35 m.p.h., lb.	20,000	34,300	40,600
Pull back of tender at 40 m.p.h., lb.	17,000	29,500	34,900
Pull back of tender at 50 m.p.h., lb.	12,700	22,500	26,700

Evaporation per hour from area of firebox, combustion chamber and syphons is

490 square feet × 55 = 26,950 pounds of water

Evaporation per hour from area of tubes (21.5 feet long) is

4932 square feet × 8.7 = 42,908 pounds of water

The total evaporation per hour, from 5422 square feet of heating surface, is therefore 69,858 pounds of water, and the average evaporation per square foot of heating surface is

$$\frac{69{,}858}{5422} = 12.9 \text{ pounds of water per hour}$$

After determining the tractive forces at different speeds as just described, the next step is to obtain the pull back of the tender at those speeds. The pull back of the tender is the tractive force less the resistance of the locomotive and tender on straight level track. Using, as an example, the engine described above, we can tabulate the calculations for pull back of the tender as follows:

The mechanical resistance of the locomotive is taken as 25 pounds per ton of weight on drivers at all speeds. The distributed weights of the locomotive are as follows:

Weight on drivers	135 tons
Weight on front truck	27.5 tons
Weight on trailer truck	41.5 tons
Weight of tender, loaded	183 tons

The resistances of the engine truck, engine trailer, and tender are obtained by multiplying the weight of each in tons by the resistance per ton for that weight at each speed taken from Table 1. These resistances, together with drawbar pulls at various speeds, are given in Table 2.

The lowest line of Table 2 shows the drawbar pull available for hauling a train back of the tender at each speed, and these figures are used for plotting the pull-speed curve.

In order to show the influence of larger boilers and the latest economy devices on locomotives of approximately equal tractive forces and weights on drivers, the above method has been applied to several engines in actual service and the results tabulated and plotted. Table 3 shows the characteristics of the locomotives and Fig. 4 the pull-speed curves. The curves show clearly the greater capacity at higher speeds of the locomotives with the larger boilers. For example, following up the 40 miles-per-hour line on the diagram, we see that the 2-8-0 exerts a drawbar pull of 17,000 pounds behind the tender, the 2-8-2, of 29,500 pounds or 73½ percent more than the 2-8-0, and the 4-8-4, of 35,500 pounds, or 109 percent more than the 2-8-0, and 20.4 percent more than the 2-8-2 at this speed. Taking 70-ton cars with a resistance of 6 pounds per ton on straight level track at 40 miles per hour, these pulls are sufficient to haul 2830 tons, 4920 tons, and 5920 tons, or 40 cars, 70 cars and 85 cars respectively.

The curves in Fig. 4 indicate clearly the ability of the modern locomotive to increase earnings by reason of this increased capacity at high speeds. In other words the ton-miles per hour produced by the locomotive, which is a direct measure of the revenue-earning capacity, will be increased.

The proposition in simple terms is that for a given weight on drivers and, therefore, a given tractive force, small cylinders, small diameter wheels and a comparatively small boiler produce a correspondingly small horsepower at speeds, while big cylinders, large diameter wheels and a larger boiler will produce larger horsepowers, and therefore will enable the locomotive to run faster with more tonnage.

Welding Contractors Associate to Promote Better Welding

What does the welding industry intend to do to protect the public from irresponsible and incompetent welding concerns whose activities are a detriment, if not an actual menace to the business?

The Welding Contractors Association of New York, Inc., has been studying this problem for nearly two years. After finding other methods futile, it has decided to require the qualification by test of welders employed by its members. The welding public will also be urged to insist on the use of qualified welders for its own protection.

Because satisfactory welds depend on expert design, competent supervision, and first-class materials, in addition to workmanship, the welders are to be certified to concerns who are fully qualified to direct intelligently the welder in his work. The association is also prepared to confer with the industry on matters relating to welding. Upon request, names and addresses of recognized specialists in the various applications of welding will be furnished and everything possible will be done to give the user a yard-stick by which he can measure accurately the concern to which he entrusts his welding.

The association adopted these policies and undertook this campaign of public education in the joint interests of the responsible welding contractors and fabricators, and of the people who have need for their services.

The enlarged activities and more stringent requirements of the association may temporarily decrease the membership but the new plans have already enhanced the association's prestige. Ultimately it is expected to expand the membership until it becomes a national association embracing those welding contractors and fabricators of the country who can show a record of welding accomplishment.

Naval Officers Win Lincoln Welding Prize

Lieutenant Commander Homer N. Wallin, U. S. N., and Lieutenant Henry A. Shade, U. S. N., working in collaboration, won first prize of $7500 in the Second Lincoln Arc Welding Prize Competition sponsored by The Lincoln Electric Company, Cleveland, O. This paper, entitled "The Design and Construction of an Arc Welded Naval Auxiliary Vessel" describes the construction of an arc-welded auxiliary vessel.

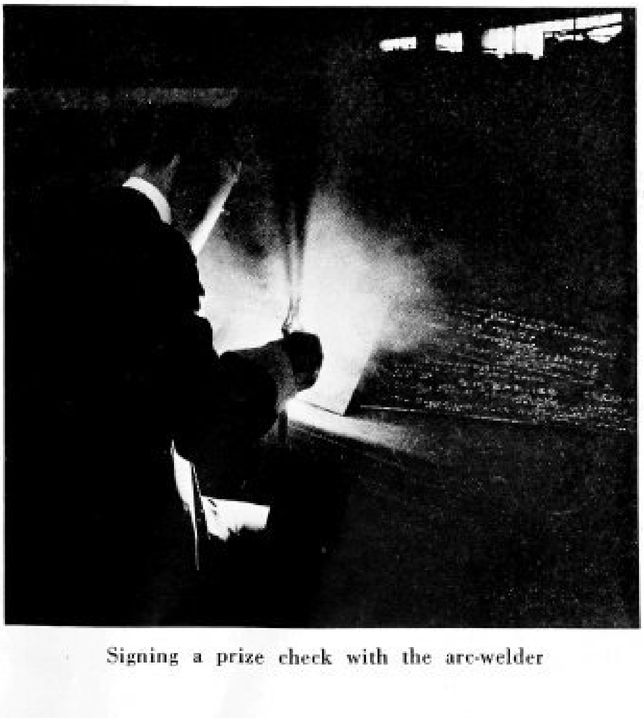
Major G. M. Barnes, U. S. A., chief of design and engineering, Ordnance Department, Watertown (Mass.) Arsenal, won second prize of $3500 for his paper "Manufacture of Ordnance at Watertown Arsenal Revolutionized Through Arc Welding."

"The Application of Arc Welding to the Design of Steel Buildings for the Resistance of Earthquake Forces" won third prize of $1500 for H. H. Tracy, structural engineer of the Southern California Edison Company, Ltd., of Los Angeles. A $750 fourth award went to Gustav F. D. Wahl and Harry E. Johns, Kiel, Germany, for a paper on a special river-sea bulk cargo ship. H. J. L. Bruff of Harrowgate, England, won fifth prize of $500 for "Strengthening Weak Iron and Steel Bridges by Arc Welding"; and a paper on arc welding of steel cars won $250 for William H. Zorn of Wyandotte, Mich. Thirty-five additional prizes of $100 were also awarded.

The checks, for $7500, $3500 and $1500 respectively, were "written" on ⅛-inch sheet steel. Each check was 24 inches long and 10 inches wide.

Welding operators in the Lincoln plant relieved the treasurer's office of the detail of filling out the checks. All information, including date, amount, payee and the name of the bank was arc welded. The checks were signed by the arc welding process.

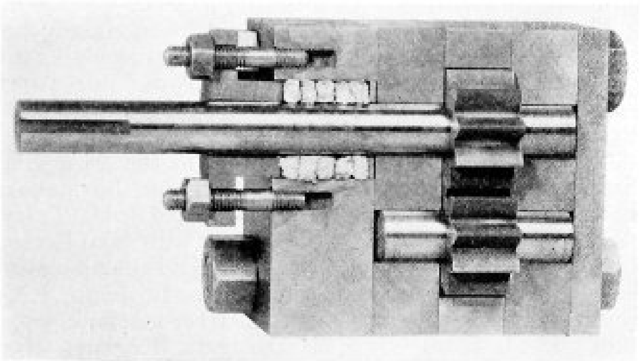
The prize winners will endorse the checks by the same



Signing a prize check with the arc-welder novel method and when the panels of steel are presented to the banks and paid, they will be returned to the bank of issue, where guards will cancel them with the aid of a sub-machine gun.

Northern Rotary Pumps of Nitralloy Steel

The Northern Pump Company, Minneapolis, Minn., has completed an entire new series of rotary pumps, revolutionary in design, in that a laminated plate construction of nitralloy steel is used throughout. The new pumps are being made in four series, XA, XD, XE and XF, ranging in capacity from 2 gallons per hour to 42 gallons per minute at 1200 revolutions per minute.



Pump with nitralloy plate bearings

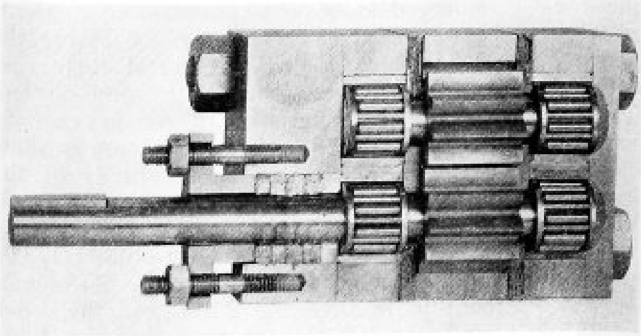
It is claimed that the extreme hardness of nitralloy steel (1000 Brinell) makes these pumps ideal for all types of oil service, including the handling of dirty crude oils and distillates at high pressures. For nonlubricating oils, the pumps would be equipped with nitralloy plate bearings such as shown in Fig. 1. Roller bearing pumps, as shown in Fig. 2, for handling clean lubricating oil would be good for pressures up to 1600 pounds per square inch on continuous service.

In each of the four series, the pump body is made of solid steel to prevent distortion from pipe strains, thus eliminating seepage, leaks and cracks. This body can be drilled and tapped for any desired purpose. In addition its mass prevents noise and vibration. In various sizes the pump parts are held together by either three or four large dowels accurately ground which serve to aline parts and to insure rigid assembly.

Nitralloy steel is used in all wearing parts of these pumps including the heat treated and ground drive shaft, and the driven shaft turning with the gears, which are also nitralloy. These gears have been designed and generated with a perfect contour for hydraulic efficiency and quiet operation. One of the principal qualities of nitralloy is that it does not distort and so gears made from it assure continuously efficient service. Helical gears are also available, if they are required for special service.

In all but the XA series, the pumps are available with either nitralloy bearings or with hardened and ground roller bearings. In all pumps nitralloy plates are assembled at the side of the gears to prevent wear and to insure long life. In the larger sizes the gears operate in a cylinder of the same material.

Because of high efficiency and wear-resisting qualities, these nitralloy pumps are especially suited for installation on hydraulic hoists, dump trucks, stokers, machine tools, and many other hydraulically operated mechanisms. The same pumps built with nitralloy bearing plates find wide application in fuel-oil burners, locomotive and industrial plants and for all types of transfer service where non-oxidizing liquids are to be handled.



Northern pump with roller bearings

British Boiler Inspectors Issue New Rules

The National Boiler and General Insurance Company, Ltd., St. Mary's Parsonage, Manchester, England, has issued a revised and enlarged edition of its handbook entitled "Steam Boiler Construction." This book covers rules of the National Boiler and General Insurance Company, Ltd., with notes on material, construction and design of steam boilers and unfired pressure vessels, and was prepared by Edward G. Hiller, chief engineer of the company. The handbook covers 245 pages and during the reprinting of this work opportunity has been taken to revise all the formulæ and constants and to rewrite the section on watertube boilers. As watertube boiler practice is in a somewhat fluctuating condition the subject is dealt with on general lines without going into so much detail as in the case of Lancashire and Cornish boilers. Illustrations of typical modern steam generating units have been included.

Alleghany Metal Welding Electrode

Joseph T. Ryerson & Son, Inc., Chicago, announces a new Alleghany metal welding electrode which is applicable to Alleghany metal and other stainless steels in the 18-8 chrome nickel group. From tests and experiments that have been made the fact has been definitely established that the rod will give uniform highly satisfactory results.

When welding with the improved electrode it is unnecessary to remove the slag before the next bead is laid down. This enables the operator to work faster and at the same time secure better penetration. Repeated tests indicate carbon increase in the bead of about 0.01 percent as compared with the original rod, thus the corrosion resistance deposit metal remains equal to that of the parent metal leaving the weld soft enough for machining.

Revisions and Addenda to the A. S. M. E. Boiler Construction Code* ▲ ▲ ▲

PAR. P-300. INSERT THE FOLLOWING AS THE FIRST THREE SECTIONS:

Piping connected to the outlet of a boiler for any purpose and which comes within the Code requirement, shall be attached (1) by screwing into a tapped opening with a screwed fitting or valve at the other end, (2) by screwing each end into tapped flanges, fittings, or valves with or without rolling or peening, (3) by joints of the Van Stone type, (4) by expanding into grooved holes, seal welding if desired. Blow-off piping of fire-tube boilers shall be attached by (1) if exposed to products of combustion, or by (1), (2), or (3) if not so exposed.

Fusion welding for sealing purposes at the juncture of bolted joints may be used.

Welding may be used to attach piping to nozzles or fittings if the rules for fusion welding and forge welding are followed.

PAR. P-310. REPLACE REVISION OF FIRST SENTENCE AS SHOWN IN APRIL ISSUE BY THE FOLLOWING:

The blow-off piping, and any piping or fittings connecting them to the boiler, shall be of black wrought iron (not galvanized) or black steel, and shall be as heavy as required for the feed pipe and in no case less than extra strong pipe size, and shall conform to the requirements of Par. P-300.

PAR. P-315. ADD THE FOLLOWING SENTENCE:

The pipes shall be attached as provided in Par. P-300.

TABLES P-11 AND U-6. CHANGE THE THIRD HEADING IN THE FIRST COLUMN TO READ:

Minimum number of threads required for connection.

CHANGE THE FOURTH HEADING IN THE FIRST COLUMN TO READ:

Minimum thickness or length required to give above number of threads, in.

FIG. P-5.

Omit illustrations *b* and *c*; transfer *a* to Par. P-104.

FIGS. P-6 AND U-16:

Fig. A. Omit reference No. 1
Fig. B. Delete
Fig. C. Eliminate reference Nos. 1 and 2. Also delete text which reads, "Limit to 8 in. maximum"
Fig. D. Eliminate reference Nos. 1 and 2
Fig. E. Delete
Fig. F. Revise to include a welded ring to show connection for a nozzle on the outside above reinforced ring
Fig. G. Include dimension "6*t*" for the overlap as shown in other illustrations.
Fig. H. Divide illustration into 2 cuts and change reference numbers
Fig. J. Divide illustration into 2 cuts and change reference numbers
Fig. K. To be revised so that joint looks like that shown in Fig. A
Fig. L No change
Illustration No. 1. To be omitted
Illustration No. 5. To be revised to show a 45-deg. angle.

It has been suggested to omit the formulas for thicknesses given below detail sketches.

Delete headings: "Unreinforced Nozzle Outlets" and "Reinforced Nozzle Outlets."

Delete text which reads: "Stress-Relieving Is Required in Fabricating Designs: B, F, G, H, J, K, and L."

Revise caption to read: "Some Types of Fusion-Welded Nozzle Construction."

FIGS. P-27 AND U-3.

Remove from body of Code and place in Appendix.

PAR. H-35. REPLACE REVISION OF LAST SENTENCE AS SHOWN IN APRIL ISSUE BY THE FOLLOWING:

When the width is 36 in. or less, the distance between the bottom of the boiler and the floor line shall not be less than 6 in., and when any part of the wet bottom is not farther from an outer edge than 12 in., it shall not be less than 4 in.

PAR. H-36. REVISED:

H-36. The minimum size of at least one fire or other access door used in a boiler or setting for boilers 30 in. and over in diameter or width shall be 12 in. by 16 in. or equivalent area, the least dimension being 11 in.

EXEMPTION CLAUSE PRECEDING PAR. U-1. REVISE ITEM (*a*) TO READ:

(*a*) to pressure vessels which are subject to Federal inspection and/or control,

PAR. U-1. REVISED:

U-1. The rules of this Section apply to unfired pressure vessels constructed of materials herein specified which are cylindrical in shape, having a combination of diameter and working pressure such that $(P - 15)(D - 4)$ is greater than 60, and to those vessels having a combination of volume and working pressure such that $(P - 15)(V - 1.5)$ is greater than 22.5,

where P = pressure, lb. per sq. in.
D = inside diameter of the vessel, in.
V = volume in cu. ft.

In the absence of definite rules in this Section on the construction of unfired pressure vessels, the specific provisions of Section I of the code may be used wherever they apply, and the vessel may then be stamped as conforming with the code.

TABLE U-1. THE FOLLOWING REVISIONS ARE PROPOSED IN THIS TABLE:

Insert the word "Guaranteed" at the beginning of the title.

Insert the following dimensions under the heading, "Size of Vessel Outlet for Safety-Valve Connections

---

* The first section of revisions to the Boiler Code appeared on page 124 of the June issue.

(Nominal pipe size and actual diameters of pipe size, in.):"

¼	⅜	½	¾	1
0.364	0.494	0.622	0.824	1.049
1¼	1½	2	2½	3
1.380	1.610	2.067	2.469	3.068

Under the heading for gage pressure, the following additions are to be noted for the corresponding diameters of valve:

Diameter of valve (in.)	Gage pressure, lb.														
	50	100	150	200	250	300	350	400	500	600	800	1000	1200	1600	2000	2400
¼	12	20	27	33	38	43	48	...	...	...	...	...	...	...	...	...
⅜	17	27	36	44	51	58	65	72	83	95	115	133	149	176	197	215
1¼	..	..	..	..	..	..	..	352	400	443	...	...	...	...	...	...
1½	..	..	..	..	..	..	..	478	528	568	...	...	...	...	...	...

PAR. U-3. ADD THE FOLLOWING:

Each safety valve ¼ in. in size and larger shall be plainly marked by the manufacturer in such a way that the marking will not be obliterated in service. The maximum allowable pressure at which the safety valve may be operated shall be cast or stamped on the body of the valve. The marking may be stamped or cast on the casing or stamped or cast on a plate or plates securely fastened to the casing and shall contain the following marking:

a Name or identifying trademark of the manufacturer.
b *Size* ....*in.*
The pipe size of valve inlet. (Where the valve inlet is not threaded the initial diameter of the inlet shall not be less than the inside diameter of a standard pipe of the same normal diameter as that of the valve.)
c *Pressure* ....*lb*
Pressure at which the valve is set to blow.
d *B.D.* ....*lb*
Blow down. (Difference between the opening and closing pressures —minimum 2 percent.)
e *A.S.M.E. Std.*

PAR. U-7. ADD THE FOLLOWING:

except that safety valves ¾ in. in size and less may have drain holes as large as possible but not less than 3/16 in. diameter.

PAR. U-12. ADD THE FOLLOWING SENTENCE:

If, the development of the art of pressure-vessel construction, other materials than those herein described become available, specifications may be submitted for consideration.

PAR. U-55. REVISED:

U-55. A manhole reinforcing ring, when used, shall be of rolled, forged, or cast steel, and shall comply with the requirements of Par. U-59.

PAR. U-57. OMIT.

PAR. U-58. REVISED:

U-58. Manhole plates and cover plates shall be of rolled, forged, or cast steel, and their strength, together with that of the bolts and of the yokes, if any, shall be proportioned for the service for which they are intended.

PAR. U-61. ADD THE FOLLOWING AS SECTION *b*;

b When plates less than ¼ in. thick are used, the manufacturer must mark each vessel in some permanent manner which will enable him to identify the heat from which the sheet in each tank has been rolled.

PAR. U-64. INSERT THE FOLLOWING AS SECOND SENTENCE:

Gas storage vessels which are too large to withstand safely the weight of the large mass of water required to fill them for hydrostatic test, may be tested by compressed air at a pressure not to exceed the maximum allowable working pressure of the vessel.

PAR. U-65. REVISE LAST SENTENCE TO READ:

A data sheet shall be filled out and signed by the manufacturer and the inspector. This data sheet, together with the stamping on the vessel shall be a guarantee by the manufacturer that he has complied with all the requirements of this section of the code.

PAR. U-66. INSERT FOLLOWING AS FOURTH SENTENCE OF FIRST SECTION:

Such separate name plates shall be used on all vessels constructed of plate less than ¼ in. thick instead of stamping the vessel itself.

INSERT THE FOLLOWING AS SECOND SECTION:

After obtaining the stamp to be used when vessels are to be constructed to conform with this section of the A.S.M.E. Boiler Code, a state inspector, or a municipal inspector of any state or municipality that has adopted the code, or an inspector employed regularly by an insurance company and who is qualified under the rules of such states or municipalities to inspect such vessels, is to be notified that an inspection is to be made; and he shall inspect such vessels during construction and after completion.

PAR. U-68*a*. ADD THE FOLLOWING SENTENCE:

When there are several vessels being welded in succession or at any one time, whose plate thicknesses fall within a range of ¼ in., only two sets of test plates for each 200 ft. of longitudinal and circumferential seams need be welded for the entire group of vessels, provided they are welded in the same way as the joints in question.

PAR. U-68*i*. REVISE FIRST SENTENCE TO READ:

i Non-Destructive Tests of Vessel. For plate thicknesses 2½ in. and less, every portion of all longitudinal welded joints of the structure, including the intersections with the girth joints, shall be radiographed by a sufficiently, powerful X-ray apparatus under a technique which will determine quantitatively the size of a defect with a thickness greater than 2 percent of the thickness of the base plate. Twenty-five percent of the length of each circumferential welded joint shall be radiographed at not less than four equally spaced intervals around the circumference. Where any one radiograph fails to comply with these requirements, all parts of all circumferential seams shall be radiographed.

PAR. U-70. REVISE SECOND SECTION TO READ:

For single-welded butt joints and for double full-fillet welds for longitudinal joints, the maximum unit joint working stress ($S \times E$) shall be as follows: For material of thickness of less than ¼ in., 5600 lb. per sq. in.; for material of thickness of ¼ in. to ⅜ in., 7000 lb. per sq. in.

Lap joints as provided for in Par. U-73*a* shall not be used in the construction of vessels for the storage of gases of any kind at pressures in excess of 100 lb. per sq. in., nor for the storage of any liquid at a temperature exceeding its boiling point at atmospheric pressure.

PAR. U-86. ADD THE FOLLOWING AS SECTION *b*:

b Resistance Welding. Where the entire area is welded simultaneously, electric-resistance butt welding, or butt welding where thermit is employed as the heating element without the introduction of extraneous metal, may be used and the ultimate strength of the joint taken as 35,000 lb. per sq. in. as in the case of forge welding. Either method may, upon the request of a manufacturer who submits proper scientific data and evidence, be given a higher rating by the Boiler Code Committee than for forge welding, provided that an authorized inspector may demand a test of any one of the welded articles he may select for the purpose, and if, after witnessing such a test, he shall doubt the advisability of using the assigned rating for the weld, the case shall be referred to the Boiler Code Committee for its decision.

PAR. U-87. INSERT THE FOLLOWING AT THE END OF THE FIRST SENTENCE OF THIS PARAGRAPH:

The structure shall be stress-relieved in accordance with the procedure given in Par. U-76.

IN THE FIRST LINE OF THIS PARAGRAPH, OMIT THE WORD "ENTIRE."

OMIT.
PARS. U-88, U-89, and U-93.

PAR. U-94. REVISED:

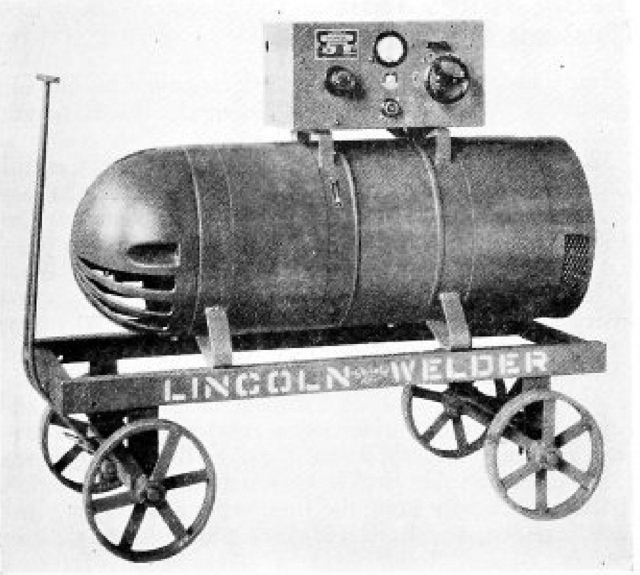
U-94. When properly brazed the strength of a joint may be calculated on a maximum unit working stress of $S \times E = 9900$ lb. per sq. in. (see Par. U-20).

FIGS. U-8 AND U-14.

Change the limit of corner radius of bend-test specimen from "0.1*t*" to "not over 0.1*t*."

FIG. U-13. Replace the dimension "about 6 in." by a reference letter "A," with the definition that "A is to be about 3 in. for ¼-in. plate and as close together as practicable for greater thicknesses."

New Shield-Arc Welder Developed



Shield-Arc welder gives uniform current

The Lincoln Shield-Arc welder whose performance is backed by its manufacturer's guarantee of more weld deposit per kilowatt hour; faster welding per kilowatt hour; and lower cost of unit welding, is announced by The Lincoln Electric Company of Cleveland, O.

A uniform welding current is one of the outstanding features claimed for the new machine. By leveling the hills and valleys or the rapid fluctuations of the usual welding current, the new welder is claimed to allow the use of a higher average current per given size rod which permits a considerable increase in the speed of welding and the amount of weld metal deposited.

The new welder is rated at 40 volts instead of the former 25, the machine having a 60 percent greater kilowatt rating than before. This rating of 40 volts is desirable for use with the shielded arc electrodes recently developed by the same company. Thus the larger capacity of the welder coupled with its higher average currents makes possible new efficiencies in welding with this type of electrode. In addition the machine offers improved operating characteristics for welding with bare or washed electrodes.

As a result of careful designing, it has been possible to eliminate the necessity for a stabilizer or kick transformer of any nature. The elimination of such losses together with the reduction of other machine losses makes possible approximately 10 percent higher efficiency at full loads and even more on overloads. There is also a smaller no load loss. This higher efficiency, the maker explains, shows remarkable results in economy of operation.

The new welder is also equipped with dual control which permits independent regulation of current and voltage which is essential for the best welding results.

Since with shielded arc welding reversed polarity is widely used, the new Shield-Arc machine is equipped with a reversible voltage control which actually reverses the direction in which the current is generated. This innovation eliminates the shifting of cable connections to change the polarity of the welding current. This reversible voltage control is operated by turning a small lever mounted on the front of the control panel. By putting this small lever in the neutral position no voltage is generated so that no current can flow through the welding leads. A center reading volt ammeter also is mounted on the control panel. This meter immediately indicates whether the machine is operating with positive or negative polarity, as well as volts and amperes.

The control panel is mounted on the machine at a handy height for ordinary control. The front is in a vertical position reducing the possibility of breakage. The control panel has a dead front with no exposed live connection in either the front or back.

As will be seen from the photograph, the upper half of the new Shield-Arc welder is entirely enclosed. This feature not only makes it drip proof, but also prevents any foreign articles from being dropped in the machine to cause injury. The machine is of arc-welded steel construction. The complete generator field is of laminated construction, being composed of electrical sheet.

The new welders are built in alternating-current and direct-current motor-driven types of 300, 400 and 600 ampere sizes. Removable running gear and draw bar simplifies the change from a portable to a stationary model.

UNSAFE FLOORS.—A heavy workman stepped on the slightly raised edge of a brick in floor, turned over and broke his ankle. Lost two months work. Does this one not emphasize the necessity for good floors?—*National Safety News.*

Irving Jones Joins Timken Organization

The Timken Steel and Tube Company, Canton, O., announces the appointment of Irving H. Jones to the position of manager of railroad sales.

Mr. Jones served in this capacity with the Central Alloy Steel Corporation for ten years, and for the last several years has been Western sales manager of The Molybdenum Corporation of America from which position he comes to The Timken Steel and Tube Company.

He has been connected with the railway supply industry and active in association work for over 20 years, playing an important part in the creation of the Allied Railway Supply Association, of which he is now president. This association is a consolidation of the Railway Equipment Manufacturer's Association, the Traveling Engineer's Association, the Air Brake Appliance Association, the Air Brake Association, the Association of Railway Supply Men, the International Railway General Foremen, the Boiler Makers Supply Men's Association, Master Boiler Makers' Association, International Railway Supply Men's Association, International Railway Fuel Association, International Railway Blacksmith's Supply Association, International Railway Master Blacksmith's Association, the Supply Men's Association, the Car Builder's and Supervisor's Association, Railway Equipment Manufacturer's Association, associated with the American Association of Railroad Superintendents.

W. E. Millhouse Elected Executive Vice-President of Burden Company

William E. Millhouse, general manager of the Burden Iron Company, Troy, N. Y., has been elected executive vice-president. The office of president, made vacant by the recent death of James A. Burden, will not be filled for at least a year. Mr. Millhouse was born in London, England. He entered the employ of the Burden Iron Company in February, 1876, starting as a feeder on a swaging machine. In 1907 he was appointed paymaster at the plant, and two years later was elected assistant secretary, becoming assistant manager in 1910. Eight years later he was promoted to general manager.

Harnischfeger Presents the P & H-Hansen Arc Welder

A bulletin under the title "The Novice Understands and the Expert Appreciates," recently published by the Harnischfeger Corporation of Milwaukee, describes in detail the construction and operating advantages of the P & H-Hansen arc welder, now manufactured exclusively by Harnischfeger.

Of particular interest to welders is the description of the so-called internal stabilizer which in this welder takes the place of the old-fashioned external stabilizer. The advantage claimed for the new type stabilizer is in the elimination of weight and extra equipment, plus the assurance of a more uniform flow of current and a consequent increase in the efficiency of the arc.

Other features of the P & H-Hansen arc welder include provision for the elimination of the complicated and trouble-making exciter, and a simple and effective single control for gradations in the welding current.

The several models, ranging from 100 to 800 amperes in capacity, are pictured and described in the new bulletin.

Boiler Manufacturers' Association Elects New Officers

At the forty-fourth annual convention of the American Boiler Manufacturers' Association, held at the Greenbrier Hotel, White Sulphur Springs, W. Va., May 23 to 25, the following new officers were elected: President, Charles E. Tudor, The Tudor Boiler Manufacturing Company, Cincinnati, O.; vice-president, E. G. Wein, E. Keeler Company, Williamsport, Pa.; and secretary-treasurer, A. C. Baker, 709 Rockefeller Building, Cleveland, O. The executive committee is composed of the following members: For a term of three years, J. G. Eury, the Henry Vogt Machine Company; M. E. Finck, Murray Iron Works Company; and A. C. Weigel, Combustion Engineering Corporation; for a term of two years, Homer Addams, Fitzgibbons Boiler Company, Inc.; George W. Bach, Union Iron Works; and G. S. Barnum, The Bigelow Company; for a term of one year, Owsley Brown, Springfield Boiler Company; F. W. Chipman, International Engineering Works; and W. C. Connelly, Foster Wheeler Corporation. H. H. Clemens, Erie City Iron Works, is *ex-officio* a member of the executive committee.

Trade Publications

Welded Construction.—Lukenweld, Inc., a division of the Lukens Steel Company, Coatesville, Pa., has issued a 24-page illustrated bulletin describing many applications of welded plate construction. This type of construction is particularly adapted to machinery and equipment parts to replace the usual casting construction for frames, foundations and other parts of machinery.

Stainless Steel Motor.—The Lincoln Electric Company, Cleveland, O., has announced the stainless steel motor. This motor, known as the Lincoln Link-Weld type E induction motor is totally enclosed and fan cooled. Being completely sealed there is no possibility of dust or dirt, moisture or fumes being present in sufficient quantities to harm windings or bearings. Of stainless steel construction the motor is arc-welded as far as possible and is available in 60 cycle ratings from ½ horsepower to 50 horsepower.

Plate Steel Boiler Nozzles.—The Worth Steel Company, Claymont, Del., has prepared a catalogue describing plate steel boiler nozzles. These nozzles known as Worth Blue Band nozzles are the logical outgrowth of the company's long experience in producing for pressure vessel fabricators, flanged and dished heads and other special forged plate work of the highest quality. This nozzle is of two-piece construction employing the Van Stone joint in the attachment of the top or bolting flange. The flange, however, is not loose as is customary in the case of Van Stone pipe joints but is firmly locked to the neck of the nozzle in the process of fabrication. Worth nozzles are available for single or double riveting for 125 pounds, 250 pounds, 400 pounds or 600 pounds working steam pressure. Tables of dimensions for the various series are given in this catalogue.

The Boiler Maker

Volume XXXII Number 7

Published Monthly by the Simmons-Boardman Publishing Company, 34 North Crystal Street, East Stroudsburg, Pa., with Executive Offices at 30 Church Street, New York City.

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Chicago: 105 W. Adams Street. Washington, D. C.: 17th and H Streets, N. W. Cleveland: Terminal Tower. San Francisco, Cal.: 215 Market Street.

Subscription price in United States and possessions, 1 year, $2.00, 2 years, $3.00; Canada, including duty, 1 year, $2.50, 2 years, $4.00; foreign countries, 1 year, $3.00, 2 years, $5.00. Single copies, 35 cents each.

Request for change of address should reach us on or before the 15th of the month, preceding the issue with which it is to go into effect. It is difficult, and often impossible, to supply back numbers to replace those undelivered through failure to send advance notice. In sending us change of address, please be sure to send us your old address as well as the new one.

The Boiler Maker is a member of the Associated Business Papers, Inc. (A. B. P.), and the Audit Bureau of Circulations, (A. B. C.)

Editorial Staff: H. H. Brown, Editor. L. S. Blodgett, Managing Editor. W. Lumbard, Associate Editor.

Contents

Communications

Sheffield Making Big Forged Steel Boiler Drums

To the Editor:

Very important work in the production of hollow forged steel drums for high-pressure boilers is being carried out in Sheffield at the Firth & Brown works, where since 1925 some 700 forged steel seamless boiler drums of various types and sizes, ranging in weight from 1008 pounds up to 70 tons each, have been manufactured. For a good part of 1931 the works maintained an output of one complete drum per day.

Included in the work now in hand is a high-pressure boiler drum for the new Ford Works at Dagenham. The steel ingot from which this drum is made weighed 175 tons as cast and the drum will be 69 tons in the finished state. Perhaps the most interesting job, however, is a reaction chamber for Trinidad Leaseholds, Ltd., which will be the biggest of its kind ever made in the United Kingdom. The ingot from which this chamber is made weighed 180 tons as cast. The finished dimensions will be inside diameter 72 inches, thickness 3¾ inches, and length 48 feet, with a weight of about 78 tons. The test pressure will be 2000 pounds per square inch.

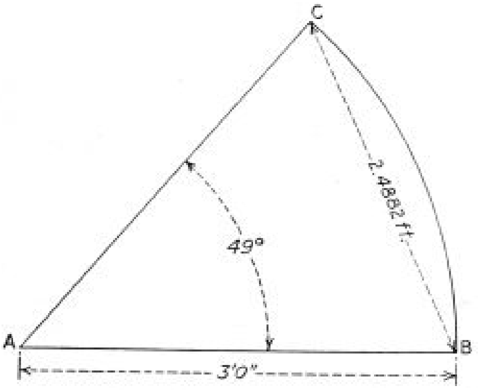
London, England. G. P. Blackall.

Method of Laying Out an Angle

To the Editor:

In the course of my experience as layerout and foreman I have noticed how difficult it is for the average man to lay out an angle with accuracy, so I have decided to pass on a speedy method which I have used successfully and with dependability.

This method makes use of a table of chords which will be found in most mechanical handbooks. The figures in the table represent the ratio of the chord to the angle with a radius of 1.0000.



Example in triangulation

As an example, it is required to strike an angle of 49 degrees at a radius of 36 inches. The number opposite 49 in the table of chords is .8294, this number multiplied by the radius 36, gives the length of the chord required, 29.85 inches or 2.4882 feet.

From a base line $A - B$ with a radius of 36 inches draw a part circle. From B, with a length equal to .8294 × 36, locate point C as shown in the accompanying diagram. A line drawn from A to C gives the required angle.

New Westminster, B. C. Wilson Blackburn.

Erosion Causes Failure of Tallow Tank

Erosion of the shell plate of a tallow tank at a packing plant in Pennsylvania caused an explosion on February 17 that injured two men and damaged property to the extent of about $2000. The tank was operated at a steam pressure of 100 pounds to the square inch. When it ruptured it wrecked the interior of the building and soared out through the roof. A part weighing 150 pounds was thrown 534 feet from the scene of the accident.

The shell of the vessel was made from quarter-inch plate. At the point of rupture erosion had thinned it until only 0.028 of an inch of metal remained.—*The Locomotive.*

Questions and Answers Pertaining to Boilers

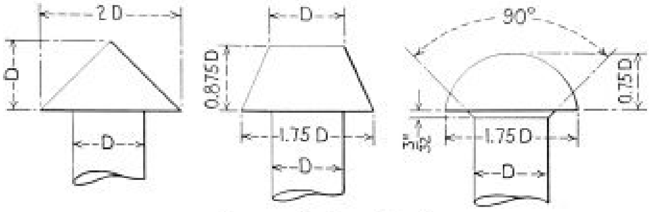
This department is maintained for the purpose of helping those who desire assistance on practical boiler shop problems. Inquiries should bear the name and address of the writer. Anonymous communications will not be considered. The identity of the writer, however, will not be disclosed unless the editor is given special permission to do so.

By George M. Davies

Dimensions of Rivet Heads

Q.—Will you tell me through the Questions and Answers Department of The Boiler Maker how I could get a fairly close to the actual diameter of a rivet? Now these rivet heads measure 2¼ inches and are the steeple-head type and they are pitched 3¼ inches. Is there any variation between the button-head and the cone head rivets? H. H.

A.—The proportions of the rivet heads to their diameter are given in the illustrations. Steeple head rivets having heads 2¼ inches in diameter would be 1⅛-inch diameter rivets.



Types of rivet heads

Layout of Transition Piece

Q.—I am submitting to your Questions and Answers Department a drawing for a transition piece which is to be layed out by triangulation, the top being elliptical in form, the base round and the taper irregular. J. L. T.

A.—The following method is convenient for laying out a tapered transition piece with a round base and an elliptical top. In Fig. 1 the elevation of the article is shown by *A-B-C-D*. In the plan view, Fig 2, *E-F-G-H* represents the projection of the elliptical top and *K-L-M-N* represents the round base. An inspection of the plan will show that the center line *F-M* divides the object into two symmetrical halves, consequently a pattern for one half will serve for the other half.

Fig. 3 is a projection of the elliptical top. A convenient way to draw an ellipse is shown in Fig. 3. First draw the diametrical lines at right angles to each other intersecting at *o*. Set out the length and breadth of the figure on these lines equally from the center *o;* set off the length *o-c*, or *o-d*. With the compasses on the longer diameter from *b* to *e*, and with *o* as a center and with the radius *o-e* describe the quadrant *e-f*. Draw the line or chord *e-f*. Set off half of it from *e* to *j* and, with *o-j* as a radius, scribe arcs on the diametrical lines as at *j-h-i-k*. Then *j* and *i* are the centers for the segmental arcs at *a* and *b*, and *h* and *k* are the centers for the lateral arcs at *c* and *d*. *P-Q-R-S* is a true outline of the elliptical top.

In working out this problem, divide *Q-P-S*, Fig. 3, into any convenient number of equal parts, as shown by the small numerals *1″*, *2″*, *3″*, *4″*, etc. In the same manner divide *K-L-M* of the base into the same number of spaces as indicated by the small numerals *1*, *2*, *3*, *4*, etc.

Next project the points *1″*, *2″*, *3″*, *4″*, etc., Fig 3, into the elevation and then to the plan, completing the plan view and locating the points *1′*, *2′*, *3′*, *4′*, etc., in the plan view. Connect the points *1*, *2*, *3*, *4*, etc., and *1′*, *2′*, *3′*, *4′*, etc., of the plan view by solid and dotted lines, as shown.

As the next step preparatory to obtaining the lines of the pattern, construct triangles whose bases are equal to the lengths of lines between points on *F-G-H* and *K-L-M*, whose altitudes are equal to the straight heights from the line *C-D* and whose hypotenuses will give the correct distance from the points on *F-G-H* to the points on *K-L-M*. The diagram of triangles represented by the solid lines is shown in Fig. 4. To obtain these triangles draw a horizontal line any convenient place, and from *T*, as shown, erect a perpendicular line and make it the same height as the vertical distance between the line *A-B* and the point *C* or *1″* in Fig. 1 represented by *1″* in Fig. 4. In the drawing, measuring from *T* on the horizontal line, set off the length of the solid line *1-1″* of the plan, Fig. 2. Connect *1′* and *1″*, Fig. 4, and the hypotenuse *1′-1″*, Fig. 4, will be the true length of the line *1-1′* of the plan view.

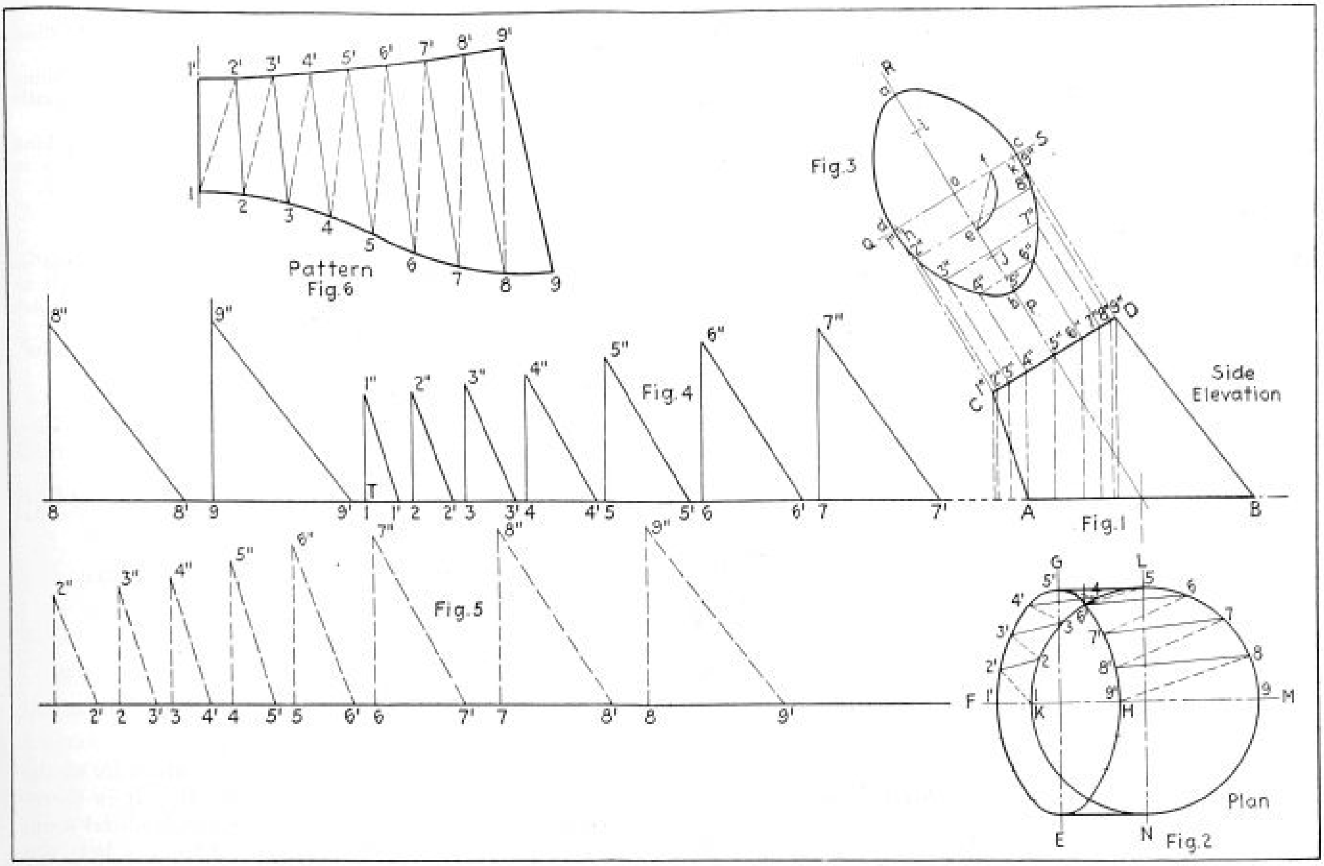
Continue in like manner, using the full lines *2-2′*, *3-3′*, *4-4′*, etc., of the plan view for the bases of the triangles and the vertical distances from the line *A-B* to the points *2″*, *3″*, *4″*, etc., as the altitudes, completing the triangles as shown in Fig. 4.

The triangles shown in Fig. 5 are constructed in the same manner, and are derived from the dotted lines *1-2′*, *2-3′*, *3-4′*, *4-5′*, etc., in the plan. These distances are used as the bases of the triangles and the vertical distances from the line *A-B* to the points *2″*, *3″*, *4″*, *5″*, etc., Fig. 1, are used as the altitudes, thus completing the triangles as shown in Fig. 5.

The hypotenuses of the various triangles in Figs. 4 and 5 are equal to the correct distances measured on the finished article between the points on *F-G-H* and K-L-M of the plant as indicated by the full and dotted lines.

In working this or any other article by triangulation, it will be found very convenient to have two pair of dividers, one pair for large spaces on *K-L-M* and the other for the smaller spaces on *Q-P-S*, thereby avoiding chances of error in resetting. Also, if two sets of trams were used, one for the solid lines and one for the dotted lines, it would save time.

For the pattern, begin by drawing a line as *1-1′*, Fig. 6. On this line set off a distance equal to *A-C* in the elevation, or *1′-1″* in the diagram of solid lines, Fig. 4. Then with the dividers set to the small spaces on *Q-P-S*,



Plan, elevation, and pattern for transition piece

and using the point *1′* as a center scribe an arc, then using the point *1* as a center and with the trams set equal to the distance *2′-2″* of Fig. 5, scribe an arc cutting the arc just drawn, locating the point *2′*. Then with the dividers set to the large spaces on *K-L-M* and with the point *1* as a center scribe an arc; then using the point *2′* as a center and with the trams set equal to the distance *2′-2″*, Fig. 4, scribe an arc cutting the arc just drawn, locating the point *2*. Proceed in like manner locating all the points of the pattern Fig. 6. Connect these points and the pattern for one-half of the development of the transmission piece is complete. A duplicate of this half will complete the development for the entire piece.

Efficiency of Welded and Riveted Patches

Q.—In order to settle an argument between several inspectors regarding a patch in the shell of a horizontal return tubular boiler in the course over the furnace I am asking your assistance. For example, take a 72-inch by 18-foot boiler with plate ½-inch thick, the patch being riveted inside of the shell as shown in the accompanying sketch.

Now one old boiler buster says that a good riveted patch does not need welding; another says it should be welded to prevent fire cracks; while one up to date cub ventured to say that a properly welded patch does not need riveting. O. B. B. says that if the job is prepared for welding it is all right, but if not prepared for welding as shown in sketch *A* it is wrong. If it is prepared as shown in sketch *B*, it is all right; but if it is seal welded, defective riveting may not show.

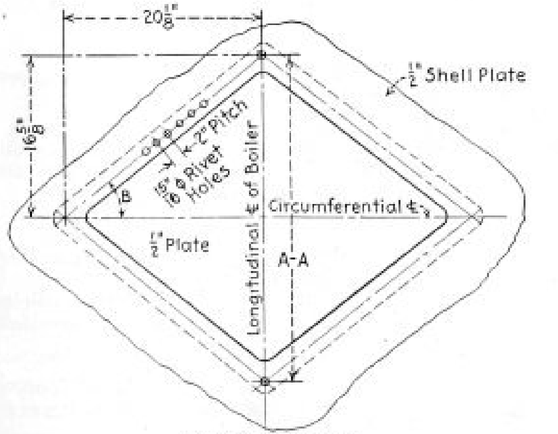
What is the efficiency of the patch? Your answer to this will be very much appreciated and will be the ruling authority. J. B. V.

A.—A good riveted patch properly designed and calked should not need to be welded for strength and tightness.

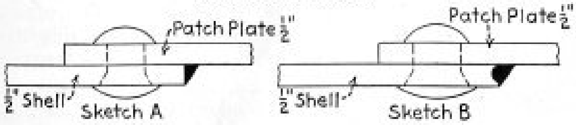
Fire cracks in firebox seams having lap joints are due to local stresses set up by the differences between the temperatures of the fire and the water sides of the joints. I do not believe that welding the calking edge of the patch would overcome this condition.

With reference to the method of preparing the weld as illustrated in sketches *A* and *B*, the ultimate strength of a calking weld is not of material importance, neither is "the design of the weld" of this kind necessary of consideration. The operator must be the judge in the number of layers needed for a tight weld, although the designer should specify a minimum amount of layers.

An all-welded patch could be applied provided the patch is applied with a double welded butt joint and welded strictly in accordance with A.S.M.E. rules for the fusion welding process. This type of patch should prove to be entirely satisfactory.



Sketch of Patch



Sketch A

Sketch B

Typical patch for calculation

The efficiency of the patch is computed as follows:

$$(1)\ \text{Efficiency} = \frac{2\,(p-d)}{p\sqrt{3} \times \text{sine}^2 B + 1}$$

where p = shortest pitch of rivets in inches
d = diameter of rivet holes, in inches
B = angle in degrees, see Fig. 1.

$$(2)\ \text{Efficiency} = \frac{\text{Net section through } A - A}{\text{Solid plate through } A - A}$$

Net section through $A - A$ is taken as the combined net section of the shell and patch. Solid plate through $A - A$ is taken for the shell only.

$$(3)\ \text{Efficiency} = \frac{N \times a \times 44{,}000 \times TS \times \text{net sectional area of shell through } A - A}{TS \times \text{solid shell plate through } A - A}$$

where N = number of rivets in single shear
TS = tensile strength of plate, 55,000 pounds per square inch
a = cross-sectional area of rivets after driving square inches

The least efficiency obtained in (1), (2), or (3) should be taken as the efficiency of the patch.

Allowable Pressure on Drum

Q.—Will you please tell me, through the Questions and Answers Department of The Boiler Maker, the formula for working the following: Calculate the allowable pressure on a drum 48 inches in diameter 10 feet long, with 3¼-inch tubes spaced equally 5¼ inches apart, with plates ½ inch in thickness.—J. J. M.

A.—The question is not clear in its description of the drum. I am assuming that the drum is drilled for tubes in a line parallel to the axis of the drum.

The A. S. M. E. Code gives the following formula for determining the maximum allowable working pressure.

P-180.—The maximum allowable working pressure on the shell of a boiler or drum for temperatures not to exceed 700 degrees F., shall be determined by the strength of the weakest course, computed from the thickness of the plate, the tensile strength stamped thereon, as provided for in the specifications for the material, the efficiency of the longitudinal joint, or of the ligament between the tube holes in shell or drum (whichever is the least), the inside diameter of the course, and the factor of safety.

$$\frac{TS \times t \times E}{FS \times R} = \text{maximum allowable working pressure, pounds per square inch}$$

where TS = ultimate tensile strength, pounds per square inch stamped on the shell plates or seamless shells as provided for in the specifications for the material
t = minimum thickness of shell plates in weakest course, inches
E = efficiency of longitudinal joint or of ligaments between openings
for riveted joints = calculated riveted efficiency
for fusion-welded joints = efficiency specified in Par. P-102
for seamless shells = 100 percent
for ligaments between openings, the efficiency shall be calculated by the Rules given in Pars. P-192 and P-193
FS = factor of safety, or the ratio of the ultimate strength of the material to the allowable stress (For new constructions FS in the above formula = 5)
R = inside radius of the weakest course of the shell or drum, inches, provided the thickness of the shell does not exceed 10 percent of the radius. If the thickness is over 10 percent of the radius, the outer radius shall be used for R.

The factor of safety used in determining the maximum allowable working pressure calculated on the conditions actually obtained in service shall not be less than 5.

For temperatures of over 700 degrees F., the working stresses given in Table P-5½ shall be used in the formula in place of $\frac{TS}{FS}$

P-192.—*Efficiency of Segment.* When a shell or drum is drilled for tubes in a line parallel to the axis of the shell or drum, the efficiency of the ligament between the tube holes shall be determined as follows:

(a) When the pitch of the tube holes on every row is equal, the formula is:

$$\frac{P - d}{p} = \text{efficiency of ligament}$$

where p = pitch of tube holes, inches
d = diameter of tube holes, inches

The pitch of tube holes shall be measured either on the flat plate before rolling, or on the median line after rolling.

Substituting the values in the question we have:
Pitch of tube holes in the drum = 5¼ inches
Diameter of tubes = 3¼ inches
Diameter of tube holes = 3 9/32 inches

$$\frac{p - d}{p} = \frac{5.25 - 3.281}{5.25} = 0.375,\ \text{efficiency of ligament}$$

The question does not give any information as to the type or efficiency of the longitudinal joint. It is therefore assumed that the efficiency of the longitudinal seam is greater than the efficiency of the ligaments between the tubes. The maximum allowable working pressure would be:

$$\text{Maximum allowable working pressure} = \frac{TS \times t \times E}{R \times FS}$$

Assuming the tensile strength of steel to be 55,000 pounds per square inch, and a factor of safety of 5, we have

$$\frac{55{,}000 \times .5 \times .375}{24 \times 5} = 85.9\ \text{pounds per square inch, maximum allowable working pressure.}$$

The heads of the drum should be dished or stayed to withstand this pressure.

Stress Relieving

Q.—What is meant by stress relieving? W. M. G.

A.—Stress relieving is the act of relieving the stress set up due to the expansion and contraction in the metal of an object while being welded.

Fusion welding of a vessel causes high temperatures to be applied locally in the vessel at the point of the weld. The molten metal of the weld is surrounded with the relatively cold mass in the walls of the vessel. With the consequent expansion and contraction there results the possibility of setting up more or less severe stresses in the weld or in the metal adjacent to it. Such stresses are referred to as "locked up stresses." There is no question whatsoever but that such stresses do exist. How severe they may be, and what their distribution is, have not been positively determined; but in many cases, and particularly in vessels of heavy plate thickness, steps must be taken to relieve such stresses if the vessel is to be safe for use.

Methods of stress relieving are given in the A.S.M.E. Boiler Contruction Code.

Associations

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Assistant Chief Inspectors—J. M. Hall, Washington, D. C.; J. A. Shirley, Washington, D. C.

Steamboat Inspection Service of the Department of Commerce

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States and Cities That Have Adopted the A.S.M.E. Boiler Code

States

Arkansas
California
Delaware
Indiana
Maryland
Michigan
Minnesota
Missouri
New Jersey
New York
Ohio
Oklahoma
Oregon
Pennsylvania
Rhode Island
Utah
Washington
Wisconsin
District of Columbia
Panama Canal Zone
Territory of Hawaii

Cities

Chicago, Ill.
Detroit, Mich.
Erie, Pa.
Evanston, Ill.
Houston, Tex.
Kansas City, Mo.
Los Angeles, Cal.
St. Joseph, Mo.
St. Louis, Mo.
Scranton, Pa.
Seattle, Wash.
Memphis, Tenn.
Nashville, Tenn.
Omaha, Neb.
Parkersburg, W.Va.
Philadelphia, Pa.
Tampa, Fla.

States and Cities Accepting Stamp of the National Board of Boiler and Pressure Vessel Inspectors

States

Arkansas
California
Delaware
Indiana
Maryland
Minnesota
Missouri
New Jersey
New York
Ohio
Oklahoma
Oregon
Pennsylvania
Rhode Island
Utah
Washington
Wisconsin
Michigan

Cities

Chicago, Ill.
Detroit, Mich.
Erie, Pa.
Kansas City, Mo.
Memphis, Tenn.
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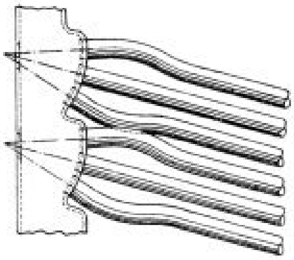
St. Louis, Mo.
Scranton, Pa.
Seattle, Wash.
Tampa, Fla.

Selected Boiler Patents

Compiled by Dwight B. Galt, Patent Attorney, 1372-1376 National Press Building, Washington, D. C. Readers wishing copies of patents or any further information regarding any patent described, should correspond directly with Mr. Galt.

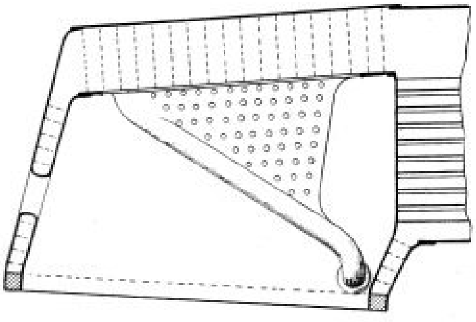
1,821,683. HEADER FOR WATERTUBE BOILERS. BENJAMIN B. WHITTAM, OF ELIZABETH, N. J., ASSIGNOR TO THE BABCOCK & WILCOX COMPANY, OF BAYONNE, N. J., A CORPORATION OF NEW JERSEY.

Claim.—A header, handholes provided in one side thereof, tubes entering the opposite side of said header spaced longitudinally along said header from each other, the ends of a plurality of longitudinally spaced tubes being bent in opposite directions and pointing towards each handhole. Six claims.



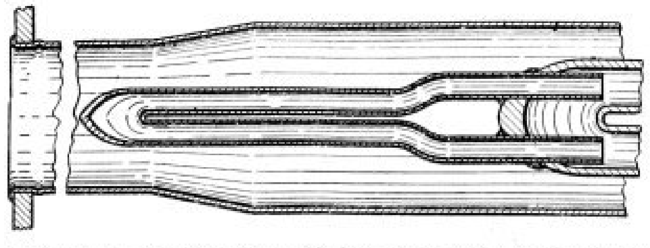
1,821,527. LOCOMOTIVE BOILER. EDWARD J. REARDON, OF CHICAGO, ILL., ASSIGNOR TO LOCOMOTIVE FIREBOX COMPANY, OF CHICAGO, ILL., A CORPORATION OF DELAWARE.

Claim.—A locomotive boiler embodying therein a firebox including side sheets, a crown sheet and front and rear sheets, a substantially rigid water circulating element disposed in the firebox and having an elongated discharge end that opens through the crown sheet and has a tubular inlet neck portion that curves downwardly and laterally to open through one of said side sheets at one of the corners of the firebox near the bottom thereof. Three claims.



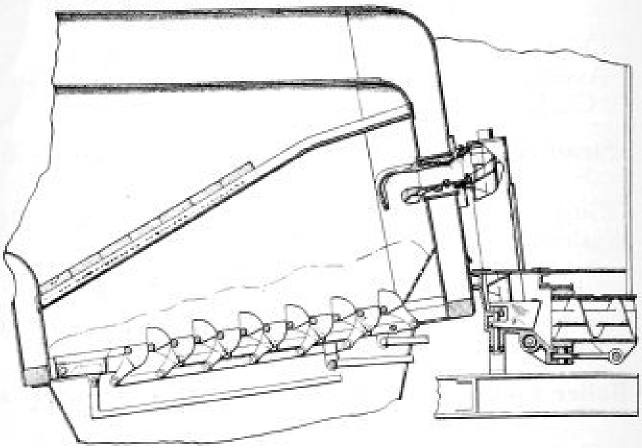
1,821,134. SUPERHEATER UNIT AND BOILER USING SAME. JOHN A. BARNES, OF CHAPPAQUA, N. Y., ASSIGNOR TO THE SUPERHEATER COMPANY, OF NEW YORK.

Claim.—A superheater unit having pipe lengths connected by a return bend and pipe lengths of smaller diameter than said first pipe lengths having mouths opening within said first lengths and extending in the same general direction as said first lengths through the wall of said bend at the outer side thereof, means forming a steam tight joint between said wall and said smaller pipe lengths, and a return bend connecting said smaller pipe lengths outside said bend. Five claims.



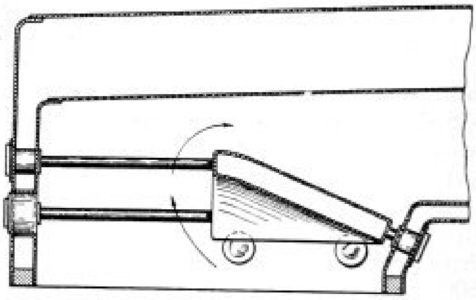
1,825,135. LOCOMOTIVE FURNACE. CHARLES J. SURDYKOWSKI, OF ERIE, PA., ASSIGNOR TO THE STANDARD STOKER COMPANY, INCORPORATED, A CORPORATION OF DELAWARE.

Claim.—In a locomotive furnace, in combination, a grate comprising rocking bars for progressively advancing fuel, a fuel delivering nozzle projecting into the furnace, and delivering upon the rearward portion of the grate only, such nozzle comprising a tube and a skirt depending from the top of the tube and spaced from its open end, and mechanical means for delivering fuel to the nozzle. Three claims.



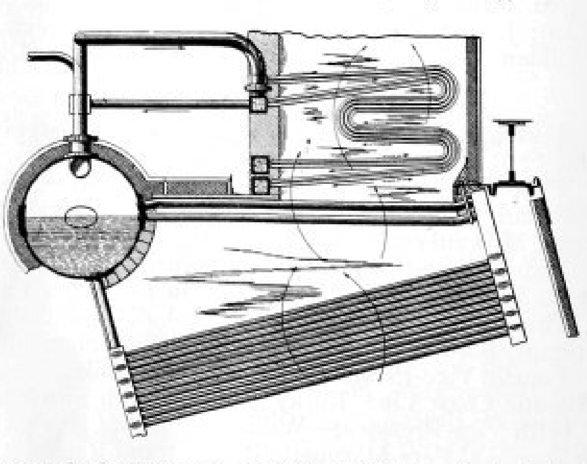
1,778,983. STEAM GENERATOR AND DRAFT CONTROL FOR LOCOMOTIVE FIREBOXES. JOHN J. REARDON AND CHARLES T. CHAKA, OF OMAHA, NEB.

Claim.—In a steam generator and draft control for a locomotive firebox having walls composed of sheets with water compartments therebetween, a water container in the firebox, conduits extending from the ends of said water container, sleeves threaded in the inner sheets of the walls extending across the space between the two sheets and extending outwardly beyond the outer sheet, said sleeves having openings therein communicating with the water spaces of the walls, a packing box held within each sleeve and slidably receiving one of the conduits, a collar received about the projecting end of the sleeve and affixed to the outer sheet, and a cap removably mounted upon said collar and acting to close the outer end of the sleeve.



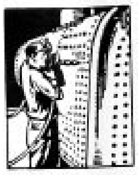
1,776,369. SUPERHEATER FOR STEAM GENERATORS. ROBERT PATON, OF CARDONALD, NEAR GLASGOW, SCOTLAND, ASSIGNOR TO THE BABCOCK & WILCOX COMPANY, OF BAYONNE, N. J.

Claim.—In a steam superheater, horizontally disposed upper headers and lower headers connected by tubes, a plurality of saturated steam inlets to said upper headers, and means for withdrawing superheated steam from said upper headers. Twelve claims.



The Boiler Maker

Reg. U. S. Pat. Off.



Repair and Maintenance Instructions

In planning future operation of shops, many of the railroads have attempted to reduce their practices to a uniform basis, both in the interest of efficiency and safety. An excellent example of a set of rules designed to accomplish this purpose, covering the inspection and care of locomotive boilers, appears in this issue. Advanced methods and adequate modern shop equipment are notable features of the railroad which developed the rules in question. In combination, these factors assure speed and accuracy in handling maintenance operations.

To meet changing conditions of types of power and service requirements, these rules are constantly undergoing a revision of detail so that at all times they constitute a tool as useful in saving time and expense as any which is included in the physical shop equipment.

The rules of the Bureau of Locomotive Inspection form the basis on which this or any set of locomotive boiler instructions must be founded, but in detail and application they can be amplified to cover all requirements of every type of power operated by a given road. While it was stated at the beginning that many roads have developed such rules, the vast majority depends on brief and too often inadequate instruction sheets to cover special types of repairs, giving no attention to including broadly the entire range of inspection and maintenance work.

There is a wide field of usefulness for rules of this kind on any railroad, and they can be developed for every branch of mechanical maintenance. Definite knowledge by those engaged in such operations will do much to eliminate haphazard and incomplete workmanship.

Railroads and Industry

The conviction is rapidly growing that the general business situation has definitely reached a level from which recovery, while slow, is nevertheless practically assured. No improvement in business can occur without affecting the railroads of the country beneficially. Added to the normal increase in traffic which must accompany any such improvement in the level of commodity production, factors within the railroad industry are working for early recovery of this basic of all national facilities. It is certain that noticeable improvement in this situation cannot long be delayed.

Definite indications have been apparent for the past few weeks that the level of commodity prices has not only flattened out but has undoubtedly made substantial gains. While small, these gains are interpreted by many economists to mean that buying of essential commodities is increasing. This in turn will influence both individuals and industrial concerns to purchase needed supplies and materials while such purchases can be made at a still low price.

This is the fundamental background necessary to any increase in railroad traffic, and for the first time in two years a definite enough trend has been established for public confidence in improvement to be justified, Within the railroad organizations this same period has been utilized for the development of constructive plans dealing with management, methods, both mechanical and administrative, and, most important of all, with the selling of railroad services to the public. Efficiency and dispatch will mark the forward movement of the railroads in the transportation scheme of the future.

Although statistics of the exact condition of rights of way, rolling stock, motive power and other physical equipment of the railroad establishment are not available, it is quite certain that a large proportion of such equipment cannot be placed in service without extensive repairs and reconditioning. The rehabilitation of rolling stock and locomotives must be undertaken within the near future if even a slight increase in traffic is to be provided for adequately. Much thought by the railroad executives, singly and as a group, is being given to this problem, and to the ways and means of accomplishing rehabilitation without involving the financial future of their roads.

A continued increase in the production of industry, even at a slow rate will automatically require the return of efficient equipment to service, and this in turn will mean work.

Boiler Orders

Orders for steel boilers in June of this year showed an increase from 265 to 328 over May, increasing also from 258,345 to 321,959 square feet. Orders for stationary boilers increased from 258 to 323, the number of square feet increasing from 248,813 to 316,662. Watertube units sold increased from 33 to 42, square footage increasing from 116,381 to 157,732.

It is evident from these statistics that the need for new power equipment is beginning to be felt. The increase may indicate that the low point in this industry has been passed.

Rules for the Inspection and Care of Locomotive Boilers

In pursuance of the policy of THE BOILER MAKER to present the locomotive maintenance practices of leading railroads of the country, the accompanying article outlines the rules and regulations governing the inspection and care of locomotive boilers of a prominent eastern system. In this issue are covered general instructions, the hydrostatic test, staybolt care and tests, special staybolt examination and similar items of maintenance. As outlined, these rules constitute modern and entirely acceptable practice, in fact they are equivalent in all respects to the requirements of the Federal rules and on the railroad in question are followed as carefully as the governmental regulations

According to instruction sheets issued to the entire mechanical staff, the rules governing the inspection and care of locomotives as outlined apply to the boilers on all steam locomotives of the railroad. The railroad in question is one of the leading systems of the United States. These boilers are in direct charge of the master mechanic in whose district they may be placed. The rules follow:

If a locomotive is out of service when any of the work or tests herein prescribed become due, the operations need not be performed until just prior to the time the locomotive is returned to service.

When the flues of a boiler are removed, the scale must be thoroughly cleaned from the inside of the boiler; all sheets, seams, rivets, braces, bolts, staybolts, etc., visible on the interior should be carefully examined; all defects found should be remedied, and a special report made of any unusual conditions found. All flues must be removed from locomotive boilers to permit this examination after 48 months' service, provided such service is performed within five consecutive years. In cases where the bottom flues and a sufficient number down the sides are removed to permit entering through the bottom of the boiler at periods in between the 48 months' service, the bottom section of the barrel sheets and flue sheets below the superheater flues should be scaled and inspected.

Wherever the jacket and lagging of a boiler are removed, the exterior of the boiler must be thoroughly inspected and defects remedied. The jacket and lagging must be removed after 60 months' service, provided such service is performed within six consecutive years, and a thorough inspection made of the entire exterior of the boiler while under hydrostatic test pressure.

In case of an accident, as a result of failure from any cause of a locomotive boiler or its appurtenances, resulting in serious injury or death to one or more persons, the boiler and all its appurtenances must be left entirely as found after such accident, until it has been examined and released by the Federal inspector, and where this is required by the state law, or by the state inspector. Where strict compliance with this would result in hindrance to or interference with traffic, the part or parts affected by such accident shall be preserved intact, so far as possible, without causing such hindrance or interference.

The hydrostatic test pressure on locomotive boilers shall be 25 percent above the maximum working pressure of the locomotive. This test must be made before a boiler is initially placed in service and after each period of 12 months' service, provided such service is performed within two consecutive years. When a locomotive has been out of service for six consecutive months or more, the hydrostatic test must be made prior to returning it to service. This test must also be made after a special staybolt examination, after removal of caps from flexible staybolts and after extensive repairs, such as application of a patch to any portion of boiler.

Wherever facilities are available for filling with hot water, the boiler must be filled with water heated as nearly as possible to the boiling point immediately before pressure is applied. Two steam gages must be provided, one connected to the standard connection in the cab and the other connected to the test cock in the roof sheet, each of these gages being connected by entirely separate lines.

When the test is being made, an authorized representative who is thoroughly familiar with boiler construction must personally witness the test, remaining outside the boiler, while another authorized representative examines the firebox from the inside. Each of the steam gages must be under constant observation during the application and continuation of the test pressure, the observation of each gage to be made by separate individuals at all times, who shall compare with each other at short intervals the pressure indicated on the gages.

After making the hydrostatic test, the dome cap and throttle standpipe must be removed and the interior surfaces and connections of the boiler examined as thoroughly as conditions will permit. After the dome cap is replaced, all the studs and nuts must be examined and the studs that do not have full threads in the nuts must be renewed.

In case the boiler can be entered and thoroughly inspected without removing the throttle standpipe, the inspection may be made by removing the dome cap only, but such variation from the rule must be noted in the record of the inspection.

When all necessary repairs have been completed, the boiler shall be fired up and the steam pressure raised to not less than the allowed working pressure, and the boiler and appurtenances carefully examined. All cocks, valves, seams, bolts and rivets must be tight under this pressure and all defects disclosed must be repaired.

The staybolts, both rigid and flexible, crown bolts, and crown bar bolts of all locomotives in service must be hammer tested once every month, preferably at boiler wash period, with the exception that flexible staybolts with caps behind grate bearers need not be hammer tested at the monthly inspection. Staybolts, both rigid and flexible, crown bolts and crown bar bolts must be hammer tested immediately after each hydrostatic test. Hollow staybolts need not be hammer tested.

An inspector specially trained for this work must tap the bolts with a hammer in order to determine whether they are broken. This test may be made either when the boiler is empty or when it has two or more gages of water under a pressure of at least 50 pounds, except that the testing of flexible bolts and crown bolts, immediately after the hydrostatic test, must be made while the boiler is under a hydrostatic pressure not less than the working pressure.

In addition to the hammer test, an inspection of all the visible tell-tale holes of the drilled bolts must be made to insure that they are open, using a drill for this purpose, if necessary. The firebox sheets must also be examined for indications of broken bolts, mud burns or bulging.

No boiler shall be permitted to remain in service when it has a broken crown bolt nor when there are one or more staybolts broken in the top row of the side, throat or fire door sheets, nor two or more adjacent broken or plugged staybolts in any part of the firebox, nor three or more broken or plugged staybolts in a circle of 4 feet in diameter, nor five or more broken in the entire boiler. Any broken or defective staybolt must be renewed as promptly as possible after discovery. Bolts with tell-tale holes leaking or plugged must be considered broken and renewed accordingly. Tell-tale holes must not be plugged except in cases of emergency. If a leak develops after an engine has been dispatched it may be plugged, if necessary, but the staybolt must be renewed upon arrival at the terminal.

The tell-tale holes of all staybolts must be kept open at all times, and must be thoroughly cleaned out with a 3/16-inch drill when locomotives are in shop for special examination of staybolts, and those visible must be cleaned out with a 3/16-inch drill prior to hydrostatic test.

Precaution must be taken to insure that all defective staybolts and crown bolts are removed. A careful examination must be made of all rigid and flexible staybolts, and crown bolts adjacent to those having defects, by cutting them out in all directions.

The caps of all expansion crown bolts in the two or more transverse rows at the front end of the roof sheet must be removed and the nuts adjusted each time a new set of flues is applied or the complete set of old flues are reset. The caps on these transverse rows of bolts must also be removed and the nuts adjusted at the first two monthly inspection periods following the application of a new firebox or the placing of a new boiler in service.

All locomotives must be given special staybolt examination, including the removal of caps from all flexible staybolts, and expansion crown bolts as required by Federal regulations, at each alternate class repairs, except that in every case when a locomotive is in shop for class repairs and the firebox or shell receives new half, three-quarters or full side sheets, it must be given special staybolt examination at that time and at the following alternate class repairs.

Locomotives that do not receive special staybolt examination within a period of two years must be removed from service and have the caps removed from all flexible staybolts and expansion crown stays as required by Federal regulation and as stated in the paragraph on flexible staybolt requirements below.

The two year period may be extended by the amount of time the locomotive has been out of service (full calendar months only to be counted), provided the total time between the removal of caps does not exceed three years. The master mechanic must arrange, if possible, to have the caps removed at the class repairs occurring in between the special staybolt examinations for those locomotives that will not, within the 24 months' period, make their mileage allotted to the alternate class repairs.

All hollow staybolts, that are behind grate bearers or stoker details and cannot be hammer tested from the inside of the firebox without removing grate bearers or stoker details, must be cut out. On locomotives equipped with duplex stokers, renew all staybolts in the back head that cannot be removed in the engine house without removal of the stoker elevators.

Hammer test all the remaining rigid staybolts in the usual manner as noted previously, and remove the defective bolts as specified.

Cut out 10 percent of all the remaining rigid bolts (not including crown bolts), selecting the majority from the zones that are known to have frequent staybolt failures, particularly those in the lower portion of the back head, the upper corners of the back head, the four corners of the side sheet, the lower corners of the throat sheet and those in other locations which in the judgment of the inspector should be given special consideration. In radial stay boilers, all staybolts that are 14 inches or more in length should be regarded as crown bolts.

Before making the special examination of rigid staybolts, the boiler inspector must obtain from the point where the permanent record is kept, a copy of the diagrams showing the results of the last special staybolt examination and all the rigid and flexible staybolts that were removed since that examination. This copy of the diagrams must be prepared from the permanent file record of the superintendent of motive power by the office which is designated by him to have charge of the same and must be forwarded promptly to the shop which is to make the special examination. The boiler inspector must determine from the diagram the relative ages of the staybolts and the localities that have had extensive staybolt failures. The rigid staybolts that were not cut out at or since the last special examination and also the rigid staybolts in the localities having had extensive failures, must be given special attention. In every case a competent boiler inspector must select the staybolts to be removed.

In cutting out the rigid bolts that are selected for removal in this special examination, the staybolts must be cut loose from the inside firebox sheet and then bent away from the inside sheet to determine the degree of fracture, if any, before cutting them loose from the outside sheet.

All flexible staybolts (including expansion crown stays) must have their caps removed and be sounded with a hammer, first from the outside and then from the inside end, and the defective bolts removed, and also the adjacent bolts in each direction until sound bolts are found. The expansion crown bolts and nuts must be examined and the loose nuts, or the bolts too short to have full thread in the nut, must be renewed. Bolts from which nuts have been removed should be checked with proper gages before new nuts are applied.

Cut out 10 percent of the flexible staybolts in the top corners of the throat sheet.

Cut out of the side sheets 10 percent of the flexible staybolts located in the top longitudinal row and 10 percent of the flexible staybolts in the outside vertical row at the front and back from the top down to the center line of boiler.

Ten percent of the flexible staybolts located in the 14 bottom longitudinal rows of the cylindrical combustion chamber from the flue sheet back to and including the row of bolts below the arch tube holes must be cut out and examined at both ends to determine the condition of the bolts in that locality.

The symbol of the shop where and the date when each class repair is made must be stamped on the wrapper sheet of the boiler in accordance with instructions entitled, "Locomotives, Marking of Fireboxes." If the wrapper sheet is renewed at other than Class 1 or Class 2 repairs, the data must be transferred to the new sheet. The letters "S. S." and "N. F. B." must follow the date of class repairs to indicate "Special Staybolt Examination" and "New Firebox," respectively.

The record of the removal of flexible staybolt and expansion crown stay caps must be shown on a form posted in the cab as outlined in Locomotive Maintenance Instructions.

All cases of leaking crown bolts, which are considered by the local inspector as evidence of low water in the boiler must be reported immediately to the superintendent of motive power, who will require a joint examination to be made by experts to determine whether there has been low water in the boiler and designate the repairs required. A written report must be made to the superintendent of motive power. No work may be done on the boiler of the locomotive until after the inspection has been made.

Crown sheets must be carefully inspected for deterioration on the water side around crown stays. The extent of deterioration of the crown sheet around crown stays must be determined at each class repairs after the firebox has had three or more years service by removing four buttonhead crown bolts in each of the outside longitudinal rows in locomotives of the 0-4-0 and 0-6-0 types, and six in the locomotives of the 0-8-0, 4-4-0, 4-4-2, 4-6-0, 2-8-0, 2-10-0, 4-6-2, 2-8-2, 4-8-2 and 2-10-2 types. The test bolts to be removed should be selected from those opposite the washout hand holes in the roof sheet. If the thickness of the crown sheet is $\frac{1}{4}$ inch or less around any of the holes, the adjacent crown bolts must be removed and holes in the crown sheet similarly inspected.

Renew the crown sheet at class repairs when the thickness of the sheet at the bottom of the deepest groove is less than $\frac{3}{16}$ inch at one or more crown bolt holes, or less than $\frac{1}{4}$ inch at 50 percent of the holes in the outside longitudinal rows.

At class repairs a crown sheet may be retained in service by applying oversize crown bolts as per conditions (a), (b) and (c) given below, provided the thickness of the crown sheet surrounding the holes is not less than $\frac{1}{4}$ inch after the sheet is threaded for new bolts.

(a) A total of five crown bolts that are over $1\frac{1}{2}$ inches, but not more than $1\frac{5}{8}$ inches in diameter, may be used in one crown sheet.

(b) Not more than three crown bolts that are $1\frac{9}{16}$ inches in diameter may be used consecutively in the same row.

(c) If either of two crown bolts is more than $1\frac{9}{16}$ inches in diameter, there must be two bolts not over $1\frac{1}{2}$ inches in diameter in between.

At the monthly boiler wash period, the crown sheet around the two outside rows of bolts must be carefully examined through the washout holes and when the dome is opened for indications of grooving on the water side. If any indications of excessive grooving are visible, test bolts shall be removed to determine the plate thickness at bolt holes. The minimum plate thickness for threading will be $\frac{3}{16}$ inch, and oversize bolts, as outlined in (a), (b) and (c) above, may be applied.

All expansion crown bolts, before leaving the manufacturing shop, shall conform to thread sizes controlled by "Go" and "No Go" snap gages, shown on standard tracings.

All new and reclaimed expansion crown bolt nuts, before leaving the manufacturing or reclamation shop, shall conform to thread sizes controlled by "Go" and "No Go" plug gages, shown on standard tracings. All expansion crown bolt nuts, before being applied to boiler, shall be checked for size, using manufacturing "Go" and "No Go" plug gages shown on standard tracings. The uniform practice for shops to follow in the gaging of nuts for size shall be to require the "Go" end of the gage to enter the nut its entire depth. Should the nut be of size permitting the "No Go" end of the gage to enter, such gage shall not enter more than full depth of the "No Go" gage.

Prior to the application of nuts on crown bolts, both the bolt and nut shall be lubricated with oil and graphite to facilitate ready removal at the subsequent period without damage to threads. When the nuts are applied to bolts, they should be run home to within two threads of a bearing, after which a careful inspection should be made to insure that nuts are a proper fit on bolts. After the nuts have been run home to a bearing, a further inspection shall be made prior to the application of caps to insure the proper adjustment of all bolts. New bolts shall be applied of a length to give not more than $\frac{1}{16}$ inch projection through the nuts. When bolts that have been adjusted several times project through the nuts sufficient to interfere with the seating of the caps, they should be faced off instead of burning them with a torch.

Not less frequently than once each two years or at time of removal of flexible staybolt caps (to comply with Federal requirements), all locomotives shall have expansion crown bolt nuts removed from all bolts and a careful inspection made of the threads on the bolts and in the nuts. The threads on bolts shall be checked for size with a condemning gage (ring type), specified for this purpose. Owing to accumulation of scale and sediment, it may be necessary to clean off the crown bolt thread (roof sheet end) before gaging. When reapplying the nuts on the crown bolts, both the bolt and nut shall be lubricated with oil and graphite to facilitate ready removal at subsequent period without damage to threads.

All boilers in locomotive service must be thoroughly washed at intervals of time, the length of which should be determined by local water conditions, but in no case must the time between wash periods exceed one month. Blowing down and refilling a boiler must not be classed as a boiler wash.

After the fire is drawn, sufficient time must be given to allow the arch bricks to cool thoroughly before draining the boiler, in order that the heat from the arch brick will not injure the side sheets, crown sheet and arch pipes. Arch bricks, when sufficiently cool to permit handling, should be removed. Water from the boiler should be blown off to the water heating system, where it is available. During this time no cold water should be admitted to the boiler to accelerate its cooling. Washing operations should immediately follow the draining of the boiler to prevent drying or hardening of the mud or scale. After the boiler is drained, remove all washout hand hole plates, plugs, caps and arch pipe plugs. Boilers must be washed with hot water where provided and at as high a pressure as the facilities provide. Special attention must be given to the bottom portion of the barrel next to the front flue sheet, the bottom portion of the combustion chamber water space, the entire length, the expansion loops, the water space above and about the fire door hole, around the stoker distributing tubes, the bottom portion of the waterlegs adjacent to the mud ring, the flues and crown sheet, for scale and mud accumulation. If an unusual condition is found in the boiler at any location, it must immediately be brought to the attention of the foreman before proceeding with the work. The nozzles used for washing boilers must be of such design as to permit the thorough washing of all parts of the boiler.

Special attention must be given arch pipes at the boiler wash period, and they must be cleaned on the inside with a turbine or other tube cleaner. They must also be cleaned on the outside by removing all clinkers adhering thereto and all roughness made smooth. Arch pipes should be gaged and a careful examination made for cracks, bulging, wear at points of contact with the arch, and pulling or leaking where fastened to the sheet. When new arch pipes are applied, the arch tube cleaner should be run through the tube and the tube washed out so as to remove any slag that may be in the tube. Special attention must be given the injector delivery pipes, both internal and external, at the hydrostatic test period, and if necessary they must be cleaned with a turbine or other tube cleaner. When local conditions warrant, the injector delivery pipes should be cleaned at more frequent intervals.

The internal delivery pipe should be tested for leaks and any leak found at the back head and at the joint in front of the crown sheet must be made tight.

After a boiler has been washed and before any plates, plugs or caps are reapplied, it must be carefully examined to see that it has been properly cleaned, that all plates, plugs, caps and their holes and fittings are in proper condition and that no foreign matter, such as waste, has been left in the boiler. The standard date tag must be applied only when the boiler is washed at a time other than that of the regular monthly inspection.

Boiler washing tools are shown on standard tracings. Existing tools that do not conform to these designs shall be continued in service provided they perform the work properly. All new tools that are made or purchased shall conform to the standard tracings.

At boiler wash periods, all flues shall be thoroughly blown out with compressed air at not less than 70 pounds pressure per square inch, applied through a ¼-inch or ⅜-inch pipe of sufficient length to extend entirely through the flue. The cleaning shall begin at the top flues and progress downward, after which a careful inspection shall be made of all flues without removing the diaphragm plates, by holding a light at one end while the flues are inspected for leaks at safe-end welds, or at the front flue sheet where grooving takes place. All flues that indicate leakage at safe-end welds shall be renewed, and flues that show evidence of leaking at the front flue sheet due to pitting or grooving shall be cut out, and if pitting is found, all small flues below the superheater and up the sides to the center shall be renewed. These instructions shall also apply in connection with renewal of a flue which has burst in service.

No tool other than a solid wrench is to be used in removing caps and plugs. Plugs that are removed from locomotive boilers shall be cleaned and reclaimed according to the following method if possible to do so within limits shown on standard tracings: "Clean by soaking for two or three hours in lye vat used for cleaning other locomotive parts and containing a 5 percent solution of caustic soda heated to 180 degrees F. Remove and wash in water. While plugs are wet, remove adhering scale with a wire brush. A careful inspection must be made of the threads, and the plugs that do not have perfect threads or correct taper shall be rethreaded on the engine lathe or by using a tapered threading die. Hand chasers shall not be used to rethread, or clean scale from plugs."

Steam gages shall be tested at least once every three months and also when any inaccuracy is reported. The siphon pipe and its connections to the boiler must be cleaned each time the gage is tested. Steam gages shall be compared with an accurate test gage or dead-weight tester, and gages found inaccurate must be corrected before being put into service.

One safety valve must be set to pop at the allowed working pressure, the second safety valve must be set to pop at 2 pounds above the working pressure, the third and fourth safety valves, if used, must be set to pop at 4 and 6 pounds, respectively, above the working pressure. The allowed working pressure is the pressure given on the latest specification card, and no deviation from this pressure will be permitted without written authority from the chief of motive power. When setting safety valves, two steam gages shall be used, one of which must be attached to a connection provided for that purpose in the roof sheet back of the dome, so that it will be in full view of the person engaged in setting the safety valves and, if the pressure indicated by the gages varies more than 3 pounds, they shall be removed from the boiler, tested and corrected before the safety valves are set. Gages shall in all cases, be tested immediately before the safety valves are set or any change made in the setting. Fill the siphon pipes to both gages with water before the pressure is turned into them. When setting safety valves, the water level in the boiler shall not be above the highest gage cock.

Safety valves shall be tested under steam at least once every three months, and also when any inaccuracy is reported. The test shall consist of raising steam to a pressure that will open both valves while a qualified inspector observes whether they open and close at the correct pressure.

The spindles of all gage cocks and water glass cocks shall be removed and the cocks thoroughly cleaned of scale and sediment at the regular monthly inspection. If necessary, the cocks themselves must be removed for this cleaning. All water glasses and water columns must be blown out and gage cocks tested before each trip. Gage cocks must be maintained in such condition that they can be easily opened and closed by hand without the aid of a wrench or other tool.

Where rings, plates and clamps, as per standard tracings have not been applied in the leg of a boiler, all 3-inch by 4-inch hand holes which are more than $\frac{1}{16}$ inch larger than standard must be reduced by adding material by the electric metallic-arc process, after which the seat for the gasket must be ground smooth and trued to conform to the surface of the boiler plate. Half the required thickness of metal added must be placed on one side of the hole and half on the opposite side, thus preventing an excessive amount from being deposited on one side.

Each hand hole plate must be ground or filed to fit accurately the contour of the boiler plate surrounding the hole to which it is to be applied. The gasket will thus be uniformly compressed and a perfect joint obtained. When hand hole joints are leaking the hand hole plate should, if necessary, be refitted to the contour of the boiler plate before a new gasket is applied.

When these hand hole plates, which are accurately fitted to the contour of the boiler plate, are removed, they must not be interchanged. Each plate must be replaced in the hole to which it was fitted, and care should be taken that it is not inverted. In order that hand hole plates can be identified, the upper half of plate shall be stamped on the outer surface with the locomotive number and location, "RF," "LF," "RB" or "LB" using ⅜-inch steel stamp.

All washout hand holes in the legs of boilers of all new locomotives, new back ends and to existing boilers at class repairs, when the firebox is renewed, should have rings, plates and clamps applied and maintained as per standard tracings. When these hand hole plates and clamps are removed at any time thereafter, they can be interchanged in the front or rear hand holes and require no marking on account of being interchangeable.

Injectors must be kept in good condition, free from scale, and must be tested before each trip. Boiler checks, delivery pipes, feed-water pipes, tank hose, tank valve and screens must be kept in good condition, free from leaks and from foreign substances that would obstruct the flow of water.

If a serious leak develops under the lagging, an examination must be made and the leak located. If the leak is found to be due to a crack in the shell or to any other defect which may reduce safety, the boiler must be taken out of service at once, thoroughly repaired, and reported to be in satisfactory condition before it is returned to service.

All steam valves, cocks and joints, studs, bolts and seams shall be kept in such repair that they will not emit steam in front of the engineman so as to obstruct his vision.

All expansion pads on boilers and all clamps and wearing plates must be inspected and cleaned and if necessary oil holes and grooves in the pad on the boiler and side clearance between the pad and clamp must be made to conform to dimensions shown on tracings entitled "Pad on Boiler." Before assembling, the rubbing surfaces of the pads, clamps and wearing plates must be thoroughly coated with engine oil and after assembling they must be oiled again through lubricating pipe.

At each monthly and annual inspection period, other than class repairs, the boiler inspector who signs the forms for boiler wash work, must examine pads and wearing plates on each side of the locomotive and where there is evidence of no free movement, they must be removed and treated the same as at class repairs.

Pulverized Coal Installation for Smith Forges

That pulverized coal is the most economical and satisfactory fuel for use in heavy forging and metallurgical furnaces has been demonstrated by the Norfolk and Western Railway, according to a statement by the railroad which sets forth the results obtained after several months use of the fuel in furnaces at its Roanoke, Va., smith shop. The coal used is the famous Pocahontas, mined along the railway in Southern West Virginia and Southwestern Virginia.

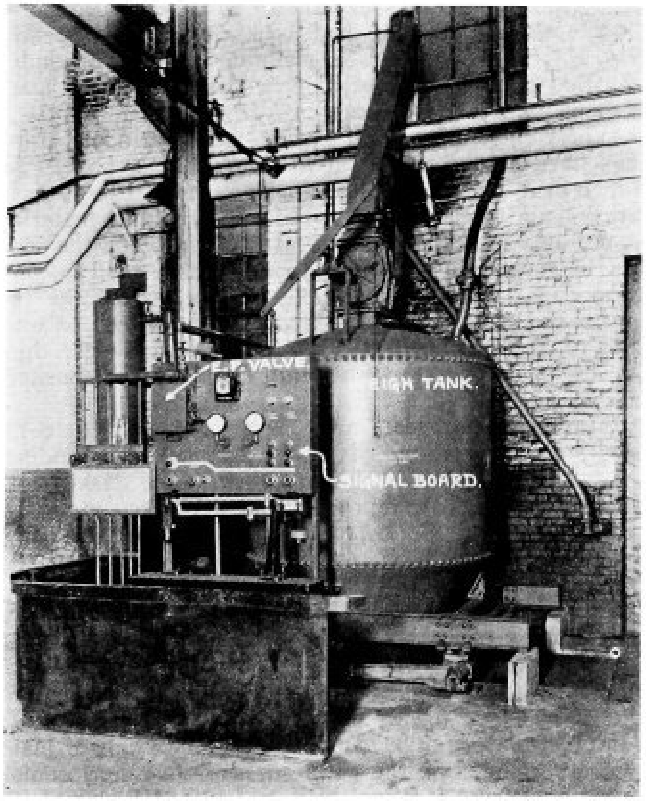


Fig. 1.—Pulverized coal weigh tank and signal board for transport system

"Following the decision to replace our hand-fired coal, heavy forging furnaces with the most modern and efficient equipment," the statement says, "the question arose as to the most economical and satisfactory type of fuel to use. Stoker fired coal, fuel oil, coal gas, and pulverized coal were investigated impartially. After a thorough analysis and investigation by our engineers, we found that pulverized coal would be the most satisfactory fuel." The announcement added that "the results obtained with pulverized coal are concrete evidence that the best type of fuel was selected."

A direct saving in fuel cost, a marked reduction in the scaling of metal because of a softer, more penetrating heat which is easily controlled, longer life for furnace linings (built of common firebrick), and cleanliness, are cited by the railroad as some of the advantages which pulverized coal has over other fuels. Records of the railroad's engineers show, according to the statement, that the railway can produce 8000 pounds of rough finished steel forgings with 3000 pounds of coal in an eight-hour working day, which is at the rate of 750 pounds of coal per ton of metal heated. This record, it is pointed out, is highly satisfactory and was made in spite of a relatively low production and intermittent operation in the smith shop since the installation of the furnaces. Under full time operation the record can be materially improved.

With the use of pulverized coal, Norfolk & Western engineers said they are able to attain a more highly reducing atmosphere in the furnaces and thereby reduce the amount of scaling or oxidation of the metal. It has also been found that pulverized coal builds up the furnace wall or lining. In other words, the heat fuses the brick surface with ash, giving the brick wall a glazed, monolithic appearance. The furnaces have been in operation six months without any brick renewals.

The fuel, which is burned with a mixture of air, is used in two heavy forging furnaces. Set between the two is a preheating furnace which utilizes waste gases from the heavy forging furnaces. The pulverized coal is fed by gravity from a storage bin in the power plant boiler room into a five-ton weigh tank, Fig. 1, where it is weighed. From the weigh tank the coal is distributed by means of a pneumatic conveying system controlled from a signal board at the weigh tank, also shown in Fig. 1. The operator at the weigh tank signals the furnace operators by an electric horn when he is ready to begin transporting coal. The furnace attendants recognize the signals by pushbuttons which show up as red lights on the main signal board. By means of other pushbuttons the weigh tank operator unlatches the transport line switching valves, which allows them to be opened by the furnace operators. The opening of an electro-pneumatic valve on the weigh tank admits compressed air (30 to 40 pounds pressure per square inch). The coal, which is aerated in the feeder, is floated through a 4-inch transport line for about 500 feet to the storage bins at the rate of about one-half ton per minute. When the first storage bin is filled the float, Fig. 2, operates to close the switching valve, thus shutting off the coal. It then flows to the second storage bin where the same operation takes place. With the closing of both switching valves an electric circuit is made which shuts off the supply from the weigh tank. Throughout the process of pulverizing, weighing, transporting and final storage at the furnace, the coal is handled as a non-explosive mixture.

An interesting feature of this equipment is a dust collector, shown in Fig. 2. It is mounted on one of the storage bins and has a vent pipe leading to the other bin. This dust collector separates the air used for transporting the coal and exhausts clean air into the shop.

Attached to the bottom of each storage bin is a motor-driven screw feeder with variable speed transmission, which feeds the coal in the desired quantity into the suction side of an exhauster, Fig. 2, where it is picked up by the air stream, thoroughly mixed and blown into the furnace through a burner, Fig. 3. Means are provided for accurately regulating the air-coal ratio for proper combustion. The pulverized coal transporting, feeding and burning equipment, was purchased from the Whiting Corporation, Harvey, Ill.

The furnaces were designed and constructed by the railroad's engineers. As stated in the foregoing these include two heavy forging furnaces and a preheating furnace. The waste gases pass from the heavy forging furnaces through ducts to the top of the center furnace, thence through an opening in the front of the arch, through the furnace, and into the stack. This furnace, the temperature of which may be regulated by opening or closing the draft doors located in the ducts, was installed primarily to preheat large section billets of high carbon content before forging. This avoids thermal ruptures and increases production of the furnace group.

Fig. 2.—Storage bin, dust collector, feeding and burning equipment for one heavy forging furnace

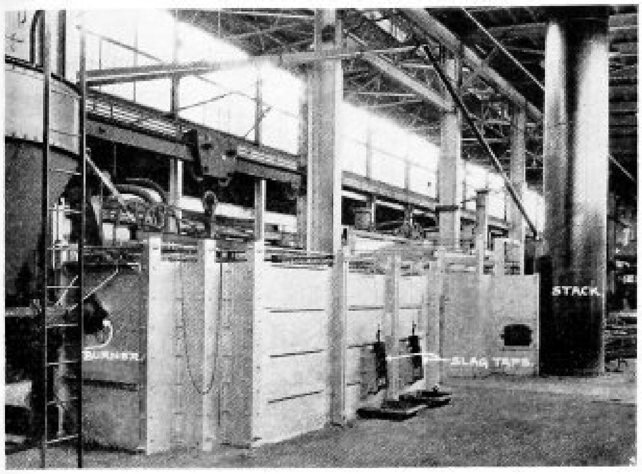


Fig. 3.—Back view of heavy forging furnace showing slag taps, draft doors and stack

The furnaces, which are constructed of ordinary firebrick laid with high-temperature cement and rigidly encased in a steel and cast-iron shell, are used in forging axles, main and side rods, piston rods, guides, and miscellaneous heavy locomotive and car forgings. They are also used in reclaiming wrought iron scrap.

The temperatures in the heavy forging furnaces range from 2200 degrees F. for steel, to 2800 degrees for reclaiming wrought-iron scrap. The temperature in the preheating furnace can be regulated from a minimum of about 800 degrees F. to a maximum of approximately 1400 degrees.

The disposal of coal ash from the furnaces has been found to be very simple. When operating at temperatures below the fusion point of the coal ash, it passes out of the stack as a light, grayish-colored dust. When operating at temperatures above the fusion point of coal ash, the ash fuses in the bottoms of the forging furnaces and runs out through slag taps into containers, shown in Fig. 3.

Seriousness of Lap Seam Crack

Lap seam cracks in the shells of horizontal tubular boilers and other types of fire-tube boilers are an old story. That, if undiscovered, they may result in disastrous explosions, also is an old story.

A boiler, located at a saw mill, was of the horizontal return tubular type, having two courses, 44 inches in diameter by 13 feet long, with double riveted lap seam longitudinal joints, and designed to carry 100 pounds pressure. An explosion occurred when the shell plate failed along the longitudinal seam of the back course, where a lap seam crack had developed. This crack had been noticed some time before, and an attempt to repair it had been made by welding along the outer surface of the shell for about 15 inches, probably all of the crack visible from the outside. An examination of the plate after the explosion, however, showed an old crack extending about 40 inches along the inner surface.

Had the serious nature of the crack been discovered and explained to the owner, and the boiler removed from service, one man's life would have been saved and seven others spared their injuries.—*The Locomotive.*

Table 1.—Physical Properties of Control Specimens, Test of Riveted Joints, 1928 Series

Specimen No.	Reduction of Area percent	Elongation in 8 inches percent	Yield Point			Ultimate Strength		
			Total pounds	Pounds per in. width	Pounds per sq. in.	Total pounds	Pounds per in. width	Pounds per sq. in.
S1A1	61.4	28.3	16,150	10,767	36,211	23,250	15,500	52,130
S1A2	65.5	37.0	16,000	10,667	34,934	22,150	14,767	48,362
S1A3	59.7	33.4	15,000	10,000	32,328	23,200	15,467	50,000
Average	62.2	32.9	15,717	10,478	34,491	22,867	15,245	50,164
S1B1	....	33.7	13,080	8,720	28,190	23,320	15,547	50,260
S1B2	....	36.2	14,175	9,450	30,480	22,685	15,123	49,170
S1B3	....	27.0	15,250	10,167	33,890	23,335	15,557	51,860
Average	....	32.3	14,168	9,446	30,853	23,113	15,409	50,430
S1C1	....	27.6	15,615	10,410	35,170	23,250	15,500	52,360
S1C2	....	34.1	13,885	9,257	30,320	22,320	14,880	48,730
S1C3	....	36.6	14,610	9,740	28,420	23,130	15,420	49,740
Average	....	32.4	14,703	9,802	31,303	22,900	15,267	50,243
S2A1	66.6	34.1	16,550	11,033	35,668	23,400	15,600	50,431
S2A2	62.3	33.8	15,100	10,342	33,481	22,350	15,308	49,559
S2A3	64.0	28.1	18,000	12,000	40,268	24,000	16,000	53,691
Average	64.3	32.0	16,550	11,125	36,472	23,250	15,633	51,227
S2B1	....	26.3	17,165	11,520	39,280	23,500	15,722	53,780
S2B2	....	33.7	14,465	9,643	31,930	22,650	15,100	50,000
S2B3	....	34.3	14,825	9,950	32,540	23,430	15,725	51,270
Average	....	31.4	15,485	10,338	34,583	23,193	15,532	51,683
S2C1	....	31.3	13,080	8,778	28,500	23,210	15,577	50,570
S2C2	....	33.8	14,135	9,487	30,800	22,850	15,336	49,780
S2C3	....	27.6	15,965	10,787	35,960	23,465	15,855	52,850
Average	....	30.9	14,393	9,684	31,740	23,175	15,589	51,067
S3A1	63.4	28.1	15,000	10,067	32,258	23,050	15,470	49,570
S3A2	68.1	38.5	14,950	10,170	32,713	22,300	15,170	48,796
S3A3	61.6	29.0	15,900	10,743	35,333	23,300	15,743	51,778
Average	64.4	31.9	15,283	10,327	33,435	22,883	15,461	50,048
S3B1	....	33.5	13,320	8,880	28,650	23,510	15,673	50,560
S3B2	....	36.0	14,305	9,537	29,630	23,170	15,447	49,610
S3B3	....	29.5	15,260	10,173	33,760	23,510	15,673	52,010
Average	....	33.0	14,295	9,530	30,680	23,397	15,598	50,393
S3C1	....	25.9	15,560	10,513	35,120	23,210	15,682	52,390
S3C2	....	36.9	14,415	9,610	31,410	22,805	15,203	49,680
S3C3	....	34.9	14,450	9,633	31,070	23,250	15,500	50,000
Average	....	32.6	14,808	9,919	32,533	23,088	15,462	50,690
S4A1	....	26.1	17,260	11,584	39,510	23,250	15,604	53,200
S4A2	....	34.7	14,920	10,013	33,160	22,380	15,020	49,730
S4A3	....	31.9	14,490	9,660	31,360	23,260	15,567	50,350
Average	....	30.9	15,557	10,419	34,673	22,963	15,397	51,093
S4B1	62.6	32.8	14,600	9,733	31,197	23,200	15,467	49,573
S4B2	68.0	34.0	15,100	10,203	32,684	22,500	15,203	48,701
S4B3	63.4	26.3	16,000	10,811	35,955	23,300	15,743	52,359
Average	64.7	31.0	15,233	10,249	33,279	23,000	15,471	50,211
S4C1	60.9	26.1	16,850	11,233	37,444	23,500	15,667	52,222
S4C2	66.2	34.5	14,100	9,527	31,126	22,050	14,899	48,675
S4C3	66.7	37.5	14,300	9,662	30,952	22,750	15,372	49,242
Average	64.6	32.7	15,083	10,141	33,174	22,767	15,313	50,046
S5A1	68.7	30.3	12,550	8,480	27,283	23,050	15,574	50,109
S5A2	65.9	34.0	14,600	9,865	31,789	22,550	15,236	49,022
S5A3	63.2	30.4	16,100	10,878	35,938	23,150	15,642	51,674
Average	65.9	31.6	14,417	9,741	31,653	22,917	15,484	50,268
S5B1	66.7	29.9	15,650	10,503	35,011	23,100	15,503	51,678
S5B2	62.4	33.5	15,550	10,367	33,952	22,550	15,033	49,236
S5B3	67.7	32.5	15,700	10,467	33,763	23,000	15,333	49,462
Average	65.6	32.0	15,633	10,446	34,242	22,883	15,289	50,125
S5C1	57.4	27.6	17,900	12,095	38,745	28,650	19,358	62,013
S5C2	53.0	30.5	17,650	11,925	38,203	28,850	19,493	62,446
S5C3	53.7	25.3	19,100	12,819	42,350	28,600	19,195	63,415
Average	54.7	27.8	18,217	12,280	39,766	28,700	19,349	62,958
S6A1	63.4	31.8	14,350	9,696	31,264	23,200	15,676	50,545
S6A2	70.0	36.0	15,600	10,612	34,211	22,500	15,306	49,342
S6A3	68.1	28.3	16,100	10,952	36,180	23,200	15,782	52,135
Average	67.2	31.9	15,350	10,420	33,855	22,967	15,588	50,674
S6B1	63.4	29.4	15,700	10,608	35,682	22,750	15,372	51,705
S6B2	66.2	36.0	16,000	10,884	36,036	22,100	15,034	49,775
S6B3	64.5	32.0	15,000	10,274	32,895	22,550	15,445	49,452
Average	64.7	32.5	15,233	10,589	34,872	22,467	15,284	50,311
S6C1	51.4	30.1	16,550	11,182	35,138	28,450	19,223	60,403
S6C2	50.6	30.9	17,750	11,913	37,606	28,400	19,060	60,169
S6C3	59.8	24.4	19,300	13,040	42,605	28,350	19,155	62,583
Average	53.9	28.5	17,867	12,045	38,450	28,400	19,146	61,052
S7A1	58.3	25.5	19,100	12,819	42,634	28,250	18,960	63,054
S7A2	57.9	27.1	17,300	11,689	37,939	28,300	19,122	62,061
S7A3	57.5	28.3	17,500	11,745	37,797	28,500	19,127	61,555
Average	57.9	27.0	17,967	12,084	39,457	28,350	19,070	62,223
S7B1	....	26.1	15,740	10,493	34,220	28,215	18,810	61,340
S7B2	....	27.0	17,110	11,407	37,200	28,720	19,147	62,430
S7B3	....	24.3	18,340	12,227	40,850	28,150	18,767	62,690
Average	....	25.8	17,063	11,376	37,423	28,362	18,908	62,153
S7C1	57.9	25.1	18,400	12,349	41,441	27,800	18,658	62,613
S7C2	53.7	27.8	18,300	12,200	40,132	28,100	18,733	61,623
S7C3	59.1	29.1	17,900	12,013	39,168	28,450	19,904	62,254
Average	56.9	27.3	18,200	12,187	40,280	28,117	18,828	62,163
S8A1	57.4	30.5	18,950	12,891	41,832	28,400	19,320	62,693
S8A2	59.8	29.8	18,050	12,196	38,985	28,350	19,155	61,231
S8A3	61.6	31.4	17,700	11,959	37,580	28,400	19,189	60,297
Average	59.6	30.6	18,233	12,349	39,466	28,383	19,221	61,407
S8B1	60.4	28.8	16,650	11,326	35,806	27,600	18,776	59,355
S8B2	58.2	30.0	17,050	11,443	36,364	27,600	18,523	58,849
S8B3	54.5	24.5	18,400	12,432	40,798	28,150	19,020	62,417
Average	57.7	27.8	17,367	11,400	37,656	27,783	18,773	60,207
S8C1	55.0	27.8	18,300	12,200	40,132	28,300	18,867	62,061
S8C2	59.7	30.3	18,950	12,633	40,578	28,200	18,800	60,385
S8C3	60.5	29.5	17,300	11,769	37,365	27,300	18,571	58,963
Average	58.4	29.2	18,183	12,201	39,358	27,900	18,746	60,470

Table 2.—Summary of Results, Tests of Riveted Joints, 1928 Series

Specimen No. (1)	Kind of Joint		Ultimate Strength pounds (4)	Manner of Failure (5)	Strength per in. width of Net Section, pounds		Strength per in. width of Control Specimens * (8)	Ratios		Actual Efficiency percent (11)
	Holes (2)	Spacing (3)			¾ inch deducted (6)	11/16 inch deducted (7)		(6)/(8) (9)	(7)/(8) (10)	
S1A.........	Drilled	2 in. chain	373,000	Plate	16,578	15,788	15,245	108.74	103.56	67.96
S1B.........	Drilled	2 in. chain	378,500	Plate	16,822	16,021	15,407	109.18	103.99	68.24
S1C.........	Drilled	2 in. chain	365,000	Plate	16,222	15,450	15,267	106.26	101.20	66.41
Average ..			372,000		16,541	15,753	15,306	108.06	102.92	67.54
S2A.........	Punched	2 in. chain	363,000	Plate	16,133	15,365	15,636	103.18	98.27	64.49
S2B.........	Punched	2 in. chain	348,000	Plate	15,467	14,730	15,461	100.03	95.27	62.52
S2C.........	Punched	2 in. chain	352,800	Plate	15,680	14,933	15,450	101.49	96.65	63.43
Average ..			354,600		15,760	15,009	15,516	101.57	96.73	63.48
S3A.........	Drilled	1¾ in. staggered	388,000	Plate	17,244	16,423	15,462	111.52	106.22	69.70
S3B.........	Drilled	1¾ in. staggered	399,600	Plate	17,760	16,914	15,598	113.86	108.44	71.16
S3C.........	Drilled	1¾ in. staggered	386,500	Plate	17,177	16,360	15,392	111.60	106.29	69.75
Average ..			391,367		17,394	16,566	15,484	112.33	106.98	70.20
S4A.........	Punched	1¾ in. staggered	386,100	Plate	17,160	16,343	15,309	112.09	106.75	70.06
S4B.........	Punched	1¾ in. staggered	394,000	Plate	17,511	16,677	15,471	113.19	107.80	70.74
S4C.........	Punched	1¾ in. staggered	367,000	Plate	16,311	15,534	15,313	106.52	101.44	66.57
Average ..			382,367		16,694	16,185	15,364	110.60	105.33	69.12
S5A.........	Drilled	2 in. staggered	376,000	Plate	16,711	15,915	15,484	107.92	102.78	67.45
S5B.........	Drilled	2 in. staggered	394,000	Plate	17,511	16,677	15,290	114.52	109.07	71.58
S5C.........	Drilled	2 in. staggered	428,000	Rivet	19,022	18,116	19,345	98.33	93.65	61.81
Average ..			399,333		17,748	16,903	16,706	106.92	101.83	66.95
S6A.........	Punched	2 in. staggered	394,000	Plate	17,511	16,677	15,588	112.34	106.99	70.21
S6B.........	Punched	2 in. staggered	388,000	Plate	17,244	16,423	15,284	112.82	107.45	70.52
S6C.........	Punched	2 in. staggered	444,000	Rivet	19,733	18,794	19,146	103.07	98.11	64.42
Average ..			408,667		18,163	17,298	16,673	109.41	104.18	68.38
S7A.........	Drilled	2¼ in. staggered	441,000	Plate	19,600	18,667	19,069	102.78	97.89	64.24
S7B.........	Drilled	2¼ in. staggered	436,000	Plate	19,111	18,202	18,908	101.07	96.27	63.17
S7C.........	Drilled	2¼ in. staggered	433,000	Rivet	19,244	18,328	18,828	102.21	97.34	63.88
Average ..			434,667		19,318	18,399	18,935	102.02	97.17	63.76
S8A.........	Punched	2¼ in. staggered	443,000	Rivet	19,689	18,751	19,221	102.43	97.56	64.02
S8B.........	Punched	2¼ in. staggered	442,000	Plate	19,644	18,709	18,773	104.64	99.66	65.40
S8C.........	Punched	2¼ in. staggered	418,000	Plate	18,578	17,693	18,779	98.93	94.22	61.83
Average ..			434,333		19,304	18,384	18,928	102.00	97.15	63.75

Theoretical Efficiencies—¾-in. holes deducted, 62.50 percent; 11/16-in. holes deducted, 65.63 percent.

* Average of three.

Determining the Strength of Net Section of Plates* ▲ ▲ ▲

Continuing a description of tests made by the Engineering Experiment Station of the University of Illinois, Urbana, Ill., the present article includes investigations of riveted lap and butt joints for which holes were drilled and others for which they were punched, and joints with rivets in single sheer and others with rivets in double sheer. This article continues a description of tests published in the July issue under the title of "Efficiency of Wide Plates" as used in the construction of storage tanks for water and oil. These experiments conducted by Professor Wilbur M. Wilson, James Mather, and Charles O. Harris, were made possible through the co-operation of the Chicago Bridge & Iron Works, Chicago, Ill. Sections of investigations covering welded joints, both lap and butt, and joints for rivets supplemented with welds will follow at a later date.

The specimens of the 1928 series, containing riveted lap joints, were used in the studies to determine the strength of the net section of plates. These tests consist of series S1 to S8, inclusive. All tests were made in triplicate. All specimens were steel plates 5/16 inch thick and 36 inches wide joined by a riveted lap joint containing two rows of ⅝-inch rivets, as shown in Fig. 1. The joints differed in the distance between rows of rivets, in the arrangement of the rivets, some joints having chain and others stagger rivets, and in the manner in which the holes were made, some specimens having drilled and others punched holes.

The two pieces that were riveted together to form a specimen were cut from a single plate. Standard tension specimens for determining the physical properties of the material were cut from the central portion. The finished specimen was approximately 7 feet 8 inches long and the joint was at the middle. The ends of each specimen were reinforced with side plates welded to the main plate to insure that the specimen would break at the joint being tested. The physical properties of the material, determined from the control specimens and given in Table 1, are typical for tank steel.

The results of the tests on the main specimens are given in Table 2. A few specimens broke by shearing the rivets, and for those the strength of the plate was at least a little greater than the strength of the specimen. Nevertheless, the strength of the plates has been taken equal to the strength of the specimen in the discussion of the efficiency of the joints. The theoretical efficiency, given at the bottom of Table 2, is the ratio of the net to the gross width of the plate. The holes, whether punched or drilled, were $\frac{11}{16}$ inch in diameter. The size of hole to be deducted in computing net width is $\frac{1}{16}$ inch greater than the nominal diameter of the rivet according to specifications for the design of tanks, and ⅛ inch greater than the nominal diameter of the rivet, according to specifications for bridges and buildings. The theoretical efficiencies, based on $\frac{11}{16}$-inch and ¾-inch holes, are 65.63 and 62.50 percent, respectively. The actual efficiencies developed by the various specimens are given in the right-hand column of Table 2.

Only two joints out of 24 failed to develop an efficiency of 62.5 percent, of these S8C, which failed by tearing the plate, had an efficiency of 61.83 percent and S5C, which failed by shearing the rivets, had an efficiency of 61.81 percent. The average efficiency of the 24 joints of these series was 66.65 percent.

The variation in strain across the plate for specimen S2B is typical of those obtained in this series of tests. In no case was there any indication of a serious lack of uniformity in the stress distribution along the joint. Fig. 2 shows specimen S2B after failure. The Leuder lines are characteristic and were easily apparent in all wide plates tested.

A discussion of tests on the 1930 series of plates as well as a summary of the results of investigations on riveted lap and butt joints will appear in the September issue, when a thorough description of the test specimens employed and an analysis of the results will be published. The 1930 series includes specimens of triple-riveted double-strap butt joints, quadruple-riveted double-strap butt joints and quadruple-riveted lap joints.

(To be continued)

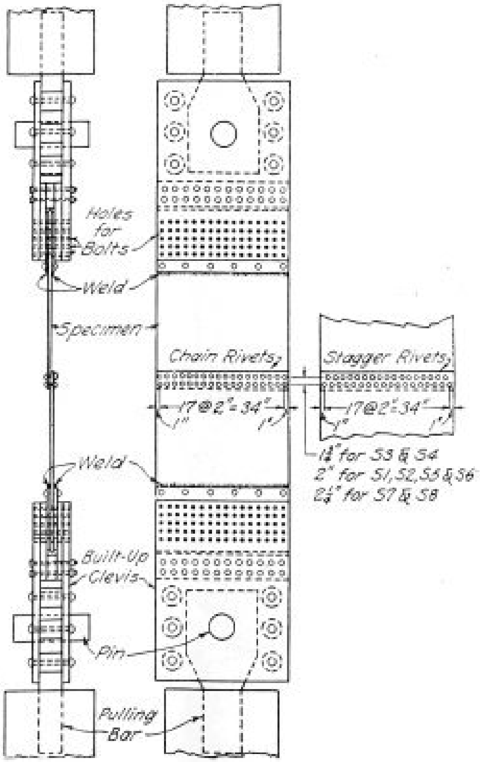


Fig. 1.—Typical lap joint of 1928 series

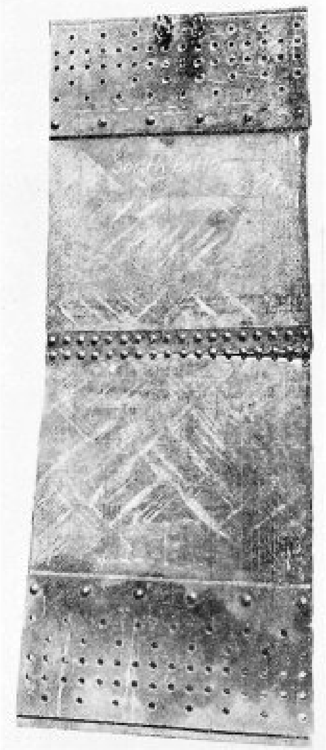


Fig. 2.—Specimen S2B after failure

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* Second of a series of articles published through the courtesy of the Engineering Experiment Station, University of Illinois, Urbana, Ill.

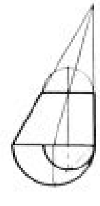
Fig.1

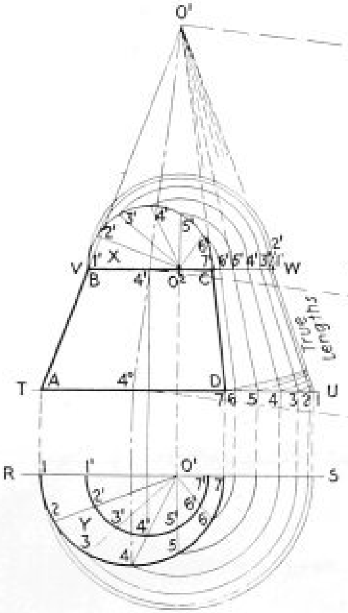

Fig.2

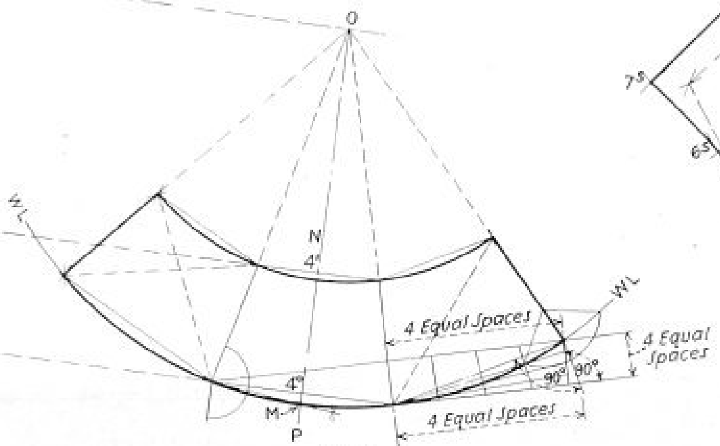

Fig.6

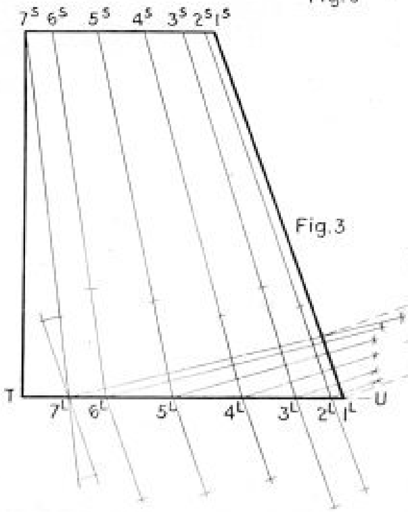

Fig.3

Fig.5

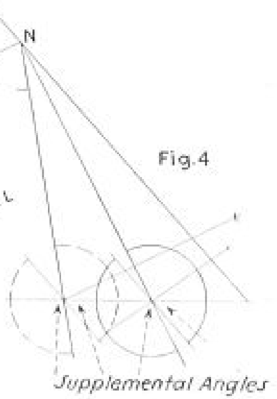

Fig.4

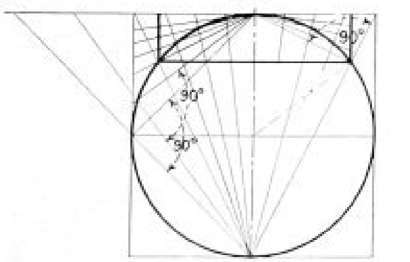

Fig.8

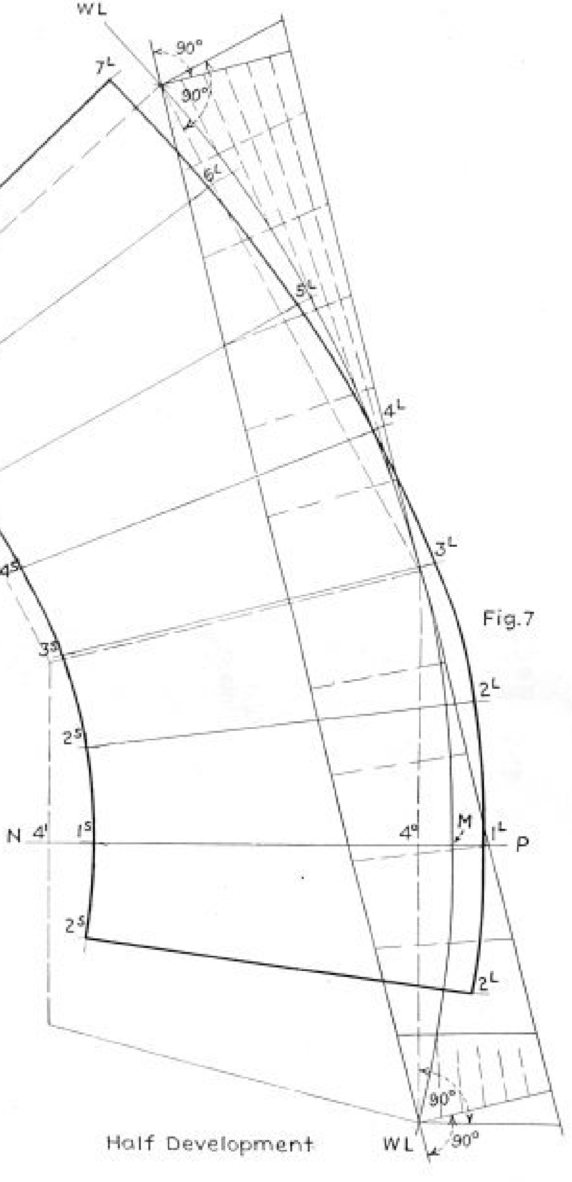

Fig.7

Half Development

Method of developing a scalene cone

Development of a Scalene Cone

By Thomas R. Gordon

The development of a conical gusset sheet, the circular bases of which are perpendicular to an element of the cone, as shown in Fig. 1, or whose axis is inclined to the plane of its base, as shown in Fig. 2, is simple when use is made of the cone's apex *0′*. When the apex is too far away, you have to resort to triangulation, a method which is not always satisfactory in its results. Since straight lines are used to develop curved surfaces, they really are chords and are shorter in length than the curves for which they are used in developing curved surfaces. Such a method often results in gusset plates and transition pieces being lopsided when they are shaped up. This is serious when applied to a gusset plate on a boiler. Very often you will find, on finishing your layout, that you have either gained on the circumference or lost on it, sometimes on one end or both. This means you have to go over the whole job again, adding a little on or taking a little off to make the circumferences right. This often throws the shape out as well.

The following is the method I use to develop a scalene cone. It can be done quickly and is very accurate:

First draw the elevation *A-B-C-D* of the cone frustum, Fig. 2. Below it draw the half plan of the large and small ends. Divide the semi-circumference into equal parts. They need not exceed 12 on any job; this one has six only, merely to show the method. Number each point *1, 2, 3, 4, 5, 6, 7, 1′, 2′, 3′, 4′, 5′, 6′* and *7′*, as shown in Fig. 2. To find the position of the apex *0′*, draw lines through the points *1-1′, 2-2′, 3-3′*, etc. Where they intersect on the line *R-S* locates the position of the apex *0′*.

To find the true lengths of the lines *1-1′, 2-2′, 3-3′*, etc., place one point of the trammels or compasses on the point *0′* in the plan, Fig. 2, set the other point to the points *1, 2, 3, 4, 5*, etc., and swing them round to the line *R-S*. From these points on *R-S* erect perpendiculars to cut the line *T-U*. Do the same with the points on the small end *1′-2′*, etc., in the elevation, Fig. 2. Swing them round to the line *V-W*. Now draw lines from the points *1′ 2′, 3′*, etc., to the points *1, 2, 3*, etc., on the line *T-U*. This will give the true lengths of the lines *1-1′, 2-2′, 3-3′*, etc.

If we had been developing the cone by radial triangulation we would have used the apex *0′* as a working point and the lengths from the apex *0′* to the line *T-U* as radii along with the equal spaces on the circumferences, *1, 2, 3, 4, 5, 6, 7*, to find the points in the development. As the apex *0′* is too far away to be used, we have to find some way to obtain the difference between each radius or element to be used in the development. It is known from geometry that if we have two converging lines which do not meet, and we wish to draw a line through a point on one of the lines, so that the line shall make equal angles with both the converging lines, the points where the line crosses the converging lines will also be equidistant from the point where the converging lines would meet if produced, Fig. 4. To do this we draw, through the point we wish to use, a line parallel to the other converging line. This makes a supplemental angle which, when bisected and the bisecting line drawn across the two converging lines, makes equal angles. The points, formed where the line crosses, are equidistant from the point of convergence *N*, Fig. 4.

We now make use of this method to determine the end of the various radii or elements *1, 2, 3, 4, 5, 6* and *7*. First draw lines parallel to the line 1^L-1^S through the points where the lines 2^L-2^S, 3^L-3^S, 4^L-4^S, etc., touch the line *T-U*, Fig. 3 which is an enlarged view of the true lengths of Fig. 2. These lines make a number of supplemental angles, which, when bisected and the bisecting lines drawn across to the line 1^L-1^S, gives us the ends of the different radii in their proper relation to each other, and is the same as if we had drawn them with the trammels, with one end in the apex *0′* and the other set to the points *1, 2, 3, 4, 5, 6* and *7* and drawn to the line *1-1′*, Fig. 2.

We now transfer these points *1, 2, 3*, etc., to a strip of wood or other suitable material, Fig. 5, for further use in the development. We now have to find the curves which would be drawn with the different radii if that were possible, or that part necessary to develop the outline of the pattern. To do this we take part of the cone's axis between the two ends marked *4°-4′*, Fig. 2, as the height of the section. Next complete the section by drawing two lines perpendicular to the line *4°-4′*, continuing into Fig. 6, and extended on both sides of the new line in Fig. 6, equal to the radius of the large end at *4°* and equal to the small radius at *4′*. Draw lines to connect the ends on both sides, giving the true shape of the section. We now draw a section on each side of this section as shown in Fig. 6 with the method shown. This gives us four points in the curve, complete the curve through these points, as shown in Fig. 6, and extend it on each side of the sections equal to at least one-fourteenth part of the diameter on the large end. Call this curve *WL-WL* or working line. This curve is the one that would be drawn by the element or radius *0′-4*, Fig. 2. All the other curves are parallel to this one and need not be drawn, as will be shown in Fig. 7.

We can now proceed with the development. At the center point *M*, Figs. 6 and 7, of the curve *WL-WL*, measure along the center line the distance *4-1* with the strip, Fig. 5. Call this point, in Fig. 7, 1^L; then take a pair of compasses set to one of the equal divisions on the large end of the cone with one point of the compasses set to the point 1^L. Describe an arc on each side of the center line; with the point *4* on the strip set to the curve *WL-WL* and placed opposite to the small arc just described, mark the point 2^L on the small arc; then set the point of the compasses to the point 2^L just found, and describe another arc. Set the point *4* of the strip on the curve *WL-WL* opposite the small arc and mark the point 3^L on the small arc. Set the compasses to the point 3^L and describe an arc cutting the large curve *WL-WL* in the point 4^L. The strip is not needed at this point as the curve is already drawn. Set the compasses to point 4^L and describe an arc. With the point *4* set to the curve *WL-WL* and opposite to the small arc, mark the point 5^L. Proceed in the same manner to find the points 6^L and 7^L. A curve drawn through these points gives the large end development.

Note: Check up the curve with the calculated length of the circumference. If the curve be long or short, open or close the compasses and step the spaces out again. The distances *4-1*, *4-2*, *4-3*, *4-5*, etc., should be drawn as small curves parallel to the large curve *WL-WL*. This will enable you to restep off the spaces much more readily should you find it necessary to do so.

Finding that our curve is now correct, we proceed to find the curve of the small end, Fig. 7. First set the compasses equal to one of the divisions on the small end of the cone. Next set the trammels to the true length of the line 1^L-1^S, Fig. 3. With one point of the trammels set to the point 1^L of the curve just found, Fig. 7, describe a small arc cutting the line *N-P* at the point 1^S. Next set the trammels to the length 2^L-2^S, Fig. 3, and placing one end on the point 2^L, Fig. 7, describe an arc with the other end, as shown. Next, set the compasses to point 1^S and describe an arc cutting the arc just found in the point 2^S. Proceed to find the other points 3^S, 4^S, etc., in the same manner by using the true length 3^L-3^S, 4^L-4^S, etc., Fig. 3. When completed, check the curve for length and if correct draw lines to connect the ends of the curves.

This completes the development of the plate up to the bend line and rivet line. Enough lap should be allowed to suit the job with the rivets placed as called for.

The method of obtaining the arc *WL-WL* is based on Fig. 8 which shows how a circle or any part of it can be found. It is known from geometry that the angle at the circumference standing on a semicircular arc is a right angle. By taking advantage of this property we are able to draw circular arcs when the chord and rise of an arc are given.

The foregoing method of developing a scalene cone has been done without the aid of mathematics. While I have gone to some length to explain the why's of the method, the method itself is quick and accurate. Further observation of the method will show that the point *M* may be placed anywhere desired on the curve *WL-WL* and the line *N-P* may be made perpendicular to it. The development then can be carried out as outlined.

Ten Rules for Getting Hurt*

1. Never wear goggles. Chips, emery dust, etc., cannot get into your eyes when you are wearing well fitting goggles.
2. Never lock the switch or valve out when working on machinery. No one will start the machine while you are in it if you do.
3. Use broken and makeshift ladders. Good ladders cost money.
4. Throw material down anywhere. If you don't stumble over it someone else may.
5. Use mushroomed tools. No use dressing up the head of the tool—the burrs will break off and fly away possibly striking you, and this saves time in the shop and gives the doctors and nurses something to do.
6. When you see broken insulation on electric wires, do nothing about it. The electricians can't be bothered to repair them, and if they did you would avoid the pleasure of a shock.
7. Walk under crane loads. If the material is properly hooked up and the chain does not break, nothing will happen. If something does happen, you know your friends enjoy music.
8. Never use a bridge. They are only ornaments and the tables are such convenient places to get hurt and get time off by that means.
9. Leave the guards off gears. If they are in place it interferes with getting your hands caught in them.
10. Always throw tools and other material from overhead. If it strikes someone it may save the material from being broken.

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* From the National Safety Council *Bulletin*.

Test Boiler for Dual Service

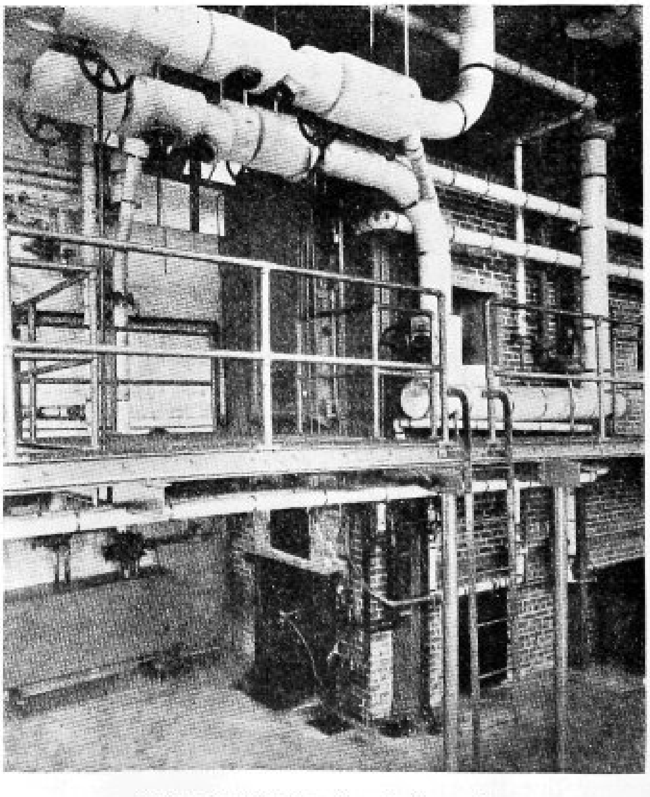
A boiler installation recently made at the plant of the Mason Regulator Company, Boston, Mass., operates at both 150 pounds and 400 pounds per square inch pressure. Normal operation is at 150 pounds; the higher pressure is carried when needed for the testing of regulators manufactured by this company.

The boiler is a 930-square foot sectional watertube oil-fired unit. To provide for sharp variations in load, the burner is automatically controlled by a diaphragm-operated mercury switch which, in turn, controls the operation of a motor on the oil feed valves.

The unit is provided with two headers, one for 400 pounds pressure and the other for 150 pounds. The former leads directly to the high-pressure test stands and is also connected through a reducing valve to the 150-pound system.

During normal operation, when no 400-pound steam is required, steam is supplied directly from the 150-pound main for general process work, oil preheating, etc. Further provision is also made for steam reduced to 30 pounds and heating pressure.

The 400-pound steam is available at controllable superheats up to 300 degrees F. so that all regulators may be tested under service conditions.



Boiler installation for dual service

Water Treatment for Locomotive Boilers*

One of the most frequently presented problems in water service is to determine whether a natural water should be used as locomotive feed water without any treatment or with merely the introduction of suitable chemicals into the water in the roadside tank or the engine tank, or whether the water should be treated in an independent wayside plant with lime and soda or whatever chemicals are necessary to soften and put it into good boiler condition. It is the opinion of the committee that few natural waters are so good that some form of treatment does not improve them and, further, that boilers are in the best condition when the water in them has a sodium alkalinity sufficient to inhibit scale and corrosion, which is never less than three grains per gallon and should always amount to 15 percent of the total dissolved solids. With good natural waters, this usually means the addition of perhaps ½ pound of soda-ash, or its equivalent, to each 1000 gallons of water put into the boiler.

Good natural waters and those intermediate waters which are hard enough to cause scale and leaking in boilers but yet do not rank with the very bad waters, are commonly handled by what is called "interior" treatment.

This does not mean that the boiler is treated in any way but merely that the water is treated with an amount and kind of chemical which will not precipitate scale matter when cold but will do so when raised to the temperature of the boiler. The chemicals for "interior" treatment cost as much as those for "complete" treatment, sometimes more, and the principal difference in expense between the two methods is the interest and depreciation on the plants.

It is the experience of the committee that complete treating plants pay a reasonable return when treating 5000 gallons per day of water having a hardness of 8 grains per gallon, but that if a smaller quantity of water is to be treated daily and the hardness is less than 8 grains per gallon, it is usually more economical to provide only interior treatment.

When, however, the 8 grains of hardness are mostly sulphate it is better to use complete treatment because the water will be likely to foam if it carries not only the 8 grains of sodium sulphate but also the total precipitate. There are also cases where railroads have wisely provided complete treatment for water containing 8 grains of hardness that is not mostly sulphate because the amount of water used per day, from one to five million gallons, is such that the advantage of water which is soft and clean as it enters the boiler is much more than the interest and depreciation on a complete treating plant.

There is a very important distinction between the possibility and the desirability of operating locomotive boilers with certain kinds of water subjected to interior treatment. It is generally accepted that for small amounts of water not exceeding a hardness of 8 grains per gallon it is satisfactory and usually cheaper to use interior treatment. In such circumstances it may be said that interior treatment is desirable. It is also beyond dispute that for waters of a hardness of 25 grains per gallon and over, the softening of the waters and the separation of the precipitate should be carried out in a complete treating plant. It is only on divisions where the average water hardness is between 8 and 25 grains per gallon that the question arises as to whether interior treatment is possible, and, if so, desirable.

By the term "average hardness" is meant not necessarily the average hardness of the waters of the division but the average hardness of the water that is used. When 75,000 gallons of water per day is taken from a low-hardness supply and 25,000 gallons per day from a high-hardness supply, the difference in method of reporting is important.

With waters having an average hardness of between 8 grains and 25 grains the question is: Below what point is interior treatment possible for tonnage trains in fast main-line service, operated by ordinary engine crews (rather than by expert western crews who have handled foaming boiler water all their lives) who are expected to handle the trains uphill or down without foaming and without the use of anti-foam compound and excessive blowing-down? We take our answer from the Wabash which uses waters that are said to average 15 grains per gallon hardness and to be almost free from alkali salts, and which has employed soda-ash, added to water in the roadside tanks, as the only boiler treatment for several years.

The average hardness of the waters handled successfully by interior treatment on the Canadian National between Sarnia, Ont., and Montreal, Que., is less than 11 grains per U. S. gallon, and the alkali salts average about 3 grains per gallon. Even with such waters the strict enforcement of a road blow-down schedule is necessary for successful operation, and experience shows that much supervision and patience are required.

It is with great hesitation that we specify limits of hardness as determining when interior treatment is possible and when it is desirable, because hardness is only one of the factors involved. We believe that the figures of 15 and 8 grains per gallon are approximately correct for waters low in alkali salts when used in an average locomotive boiler by an average crew. But a boiler with 18 inches of vertical steam space between the water surface and the steam exit will carry much worse water than can be carried by a boiler with an 11-inch steam space. Mud, oil and organic slimes in water affect its foaming qualities materially; and there are some further differences in waters which we are not yet able to explain, because on the same railroad and with other conditions approximately equal the boilers on one division always foam when the alkali concentration is 100 grains per gallon, on another division when it is 180 grains, and on still another when it is 300 grains. There are a few waters with a hardness of between 15 and 25 grains that can be handled by interior treatment when all the above conditions are in their favor, but success should not be expected by the inexperienced.

In connection with what is known as complete treatment with lime and soda-ash for very bad waters, at-

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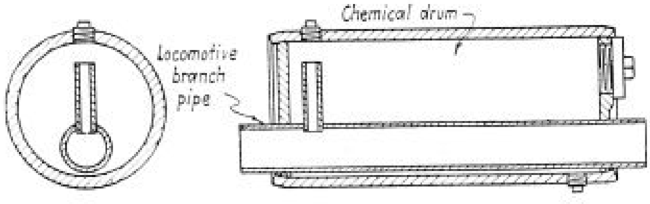
* This is an abstract of the material given in Appendix C of the report of the A. R. E. A. Committee on Water Service and Sanitation, presented at the 1932 convention. The late C. H. Koyl, who was engineer of water service of the Chicago, Milwaukee, St. Paul & Pacific, was chairman of the subcommittee that presented this report.

tention is called to the fact that some natural waters contain silica, either in a colloidal state or in solution, which is not affected by ordinary processes of water softening, so that it has often happened that water softened down to 2 grains per gallon or less, has deposited in boilers a very troublesome scale which is found to consist of about equal parts of silica and entrained calcium carbonate. At the meeting of the American Chemical Society at Indianapolis, Ind., in March, 1931, successful tests for removing silica from water by treatment with sodium aluminate were reported, the process being the formation of calcium or magnesium-aluminum-silicate which settles out.



Chemical container constructed around locomotive branch pipe

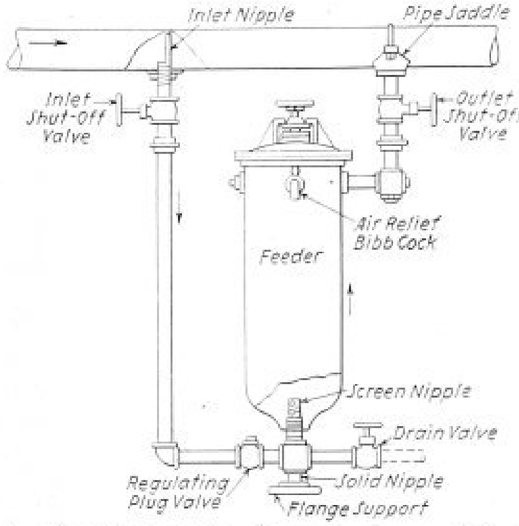
At one water station on the Chicago, Milwaukee, St. Paul & Pacific, where 30,000 gallons of Mississippi River water is treated per hour, there had always been trouble from this silica scale. However, a slight but carefully adjusted excess of sodium aluminate now decreases the silica content of the water from 0.6 grains per gallon to 0.1 grains per gallon in the treating plant, and the excess of alumina over silica in the water in the boiler effectively prevents silica scale.

Various chemicals (soda-ash, tri-sodium phosphate, sodium aluminate and the tannin compounds) are in approved use for interior treatment. Of these, soda-ash is always used in sufficient amount to antidote the sulphate hardness, the other sodium compounds being employed to supply the excess alkalinity and expedite the precipitation so as to lessen the tendency to foam. The tannin compounds are used to prevent precipitation in the injector and connecting pipes.

When chemicals are fed to the engine tank on the road, or even to the boiler while in the engine house, there are possibilities of error due not only to lack of training but sometimes to prejudice, but practical methods have been developed for making the delivery of chemical proportional to the delivery of water to either the roadside tank or the boiler. A by-pass proportioning device for adding soda-ash to the roadside tank at each water station has been in successful use on the Wabash for a number of years, while a feeder device for adding compound to the water in the roadside tank has been used quite extensively on the Erie for a long time. The latter device is connected to the water main serving the tank, and a small amount of the water flow is diverted through the feeder, dissolving the chemical balls and carrying the solution into the water tank. It is also common practice to use small pumps, which are attached to and operated by the main water pump for delivering dissolved chemicals to the roadside tank, while there are various automatic methods for delivering sodium aluminate and its mixture to tanks and water columns through the use of water motors installed on the water mains.

Another method for treating only the water used in the boiler has been used for five or six years on the Chicago & North Western. The appliance employed consists of a container constructed around the locomotive branch pipe between the injector and the boiler check, and connected with the water in the branch pipe by a single tube. The container is loaded and closed before each trip and the surging of the water in the branch pipe serves to add a small amount of compound to the water as it passes. One of the advantages of this type of feeding is that the injector is not affected by any tendency to early precipitation. There are some 400 to 500 of these appliances in successful use, and the only difficulty reported is due to leaky boiler check valves which permit hot boiler water to get into the branch pipe and clog it with precipitate.

Still another appliance for this purpose is known as the locomotive water conditioner. It is an open-type feedwater heater with provision for the storage of 700 gallons of water heated to 210 degrees F. by the recovery of heat from the exhaust steam. The storage tank provides a means for the heat acceleration of any desired chemical reaction. Means are also provided for sludging out the precipitate before the water is pumped to the boiler. This device has the advantage of returning 10 percent of distilled water to the engine tank and of removing more than 80 percent of the oxygen dissolved in the cold water.



Feeder for adding compound to water in a roadside tank as used on the Erie

The most important points to remember in connection with the use of interior boiler treatment are that the results should be subjected to frequent chemical examination and that the best means of insuring accurate treatment is to regulate the supply by one of the automatic appliances mentioned above. Nothing can take the place of careful constant supervision.

Work of the A.S.M.E. Boiler Code Committee

The Boiler Code Committee meets monthly for the purpose of considering communications relative to the Boiler Code. Any one desiring information as to the application of the code is requested to communicate with the Secretary of the Committee, 29 West 39th St., New York, N. Y.

The procedure of the committee in handling the cases is as follows: All inquiries must be in written form before they are accepted for consideration. Copies are sent by the secretary of the committee to all of the members of the committee. The interpretation, in the form of a reply, is then prepared by the committee and passed upon at a regular meeting of the committee.

This interpretation is later submitted to the council of The American Society of Mechanical Engineers for approval, after which it is issued to the inquirer and published.

Below are given records of the interpretation of the committee in Cases Nos. 588 (Reopened), 712, 715, 721, 722, 723, as formulated at meeting of April 22, 1932, all having been approved by the council. In accordance with established practice, names of inquirers have been omitted.

Case No. 588 (Reopened).—*Inquiry:* Will it be permissible in the forming of water legs in vertical tubular and firebox types of boilers, to attach the OG or flanged-in bottom edges of the plates by fusion welding? Case No. 313 permits of this construction for water legs of detached smokeless fireboxes, but the opinion of the committee is requested in regard to this construction for water legs in boilers of these particular types.

Reply: Par. P-186 of the code has been revised to provide for joints between the door-hole flanges of furnace and exterior sheets of boilers, provided these sheets are properly stayed or supported around the door-hole opening. It is the opinion of the committee that where the load due to internal pressure on the plates forming the water leg is carried by staybolting and the inside width of the water leg does not exceed 4 inches, the construction shown in Fig. 25, where both plates are flanged, will meet the requirements of the revised paragraph. The plates may be considered as fully supported if the distance from the weld to the nearest row of staybolts is not more than one-half the pitch allowed by the formula in Par. P-199.

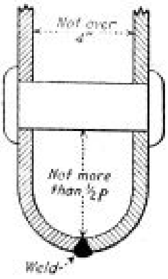


Fig. 25

Case No. 712.—*Inquiry:* Is it permissible, in the construction of welded steel-plate low-pressure heating boilers, to space staybolts a full pitch distance from a welded joint, or is this spacing limited to one-half of a full pitch?

Reply: It is the opinion of the committee that welded joints as applied in the construction of steel-plate low-pressure heating boilers are for the purpose intended equivalent to riveted joints. The allowable distance from a corner welded joint to the nearest row of staybolts may be a full pitch as provided for in the formula in Par. H-21 of the code for boilers to be operated at not to exceed 15 pounds steam and 30 pounds water. This does not apply to a welded joint in a flat surface, where the usual rules for staybolting are effective.

Case No. 715.—*Inquiry:* Will unfired pressure vessels fabricated by fusion welding under Class 2 test requirements meet the code provisions if the base metal is a high-strength copper alloy of the following chemical and physical properties?

Copper	96 percent (approx.)
Silicon	3 percent (approx.)
Manganese	1 percent (approx.)
Impurities, not over	0.5 percent
Tensile strength, pounds per square inch, min.	52,000
Yield strength, pounds per square inch, min.	18,000
Elongation in 2 inches, min.	65 percent
Reduction in area, min.	70 percent

The plate is hot rolled and annealed, and is free from injurious defects and flaws and has a workmanlike finish.

Reply: It is the opinion of the committee that annealed copper-alloy plates or sheets having the chemical composition and minimum physical properties specified in the inquiry, may be used for the construction of unfired pressure vessels by fusion welding under the general requirements of Class 2 construction, the exceptions from which are that the elongation as determined by the free bend test shall not be less than 40 percent, that the tensile strength shall not be less than that of the base metal, and that stress-relieving is not required for this kind of annealed material. The operation temperature shall not exceed 406 degrees F. The maximum allowable unit working stress shall not exceed 9000 pounds per square inch for the material and $(S \times E) =$ 7200 pounds for joints welded as above specified.

Case No. 721.—*Inquiry:* Is it necessary, under the requirements of Par. P-302 of the code, to use extra heavy valves on all steam outlets from a boiler when the pressure exceeds 125 pounds per square inch? Inquiry is also made as to the significance of the term "at least" in the first line of this paragraph.

Reply: The requirement for the extra heavy valves when the pressure exceeds 125 pounds per square inch, applies to all outlet valves in the boiler. The term "at least" is intended to indicate that the design of all outlet valves should be sufficient to withstand the working pressure, which may in some cases require heavier construction than the standard extra heavy stop valves.

Case No. 722.—*Inquiry:* Can the word "close" as used in Par. P-314, where it is stated that feed water shall not be discharged close to a riveted joint, be defined in terms of a definite distance?

Reply: It is the opinion of the committee that it is not desirable to attempt to define the word "close" in terms of a definite distance. It is the intent of the code that the feed water shall not be discharged into a boiler so that it will flow directly against or along a riveted joint. If necessary, the discharge end of a feed pipe should be fitted with a baffle to divert the flow from the riveted joint.

Case No. 723.—*Inquiry:* Is it acceptable, under the rules of the Heating Boiler Section of the code, to place wet-bottom steel-plate boilers closer to the floor line than the 12-inch clearance limit specified in Par. H-35?

Reply: It is the opinion of the committee that for certain widths of the wet-bottom portion, the clearance above the floor line may be less than the 12 inches specified in Par. H-35, and it is therefore proposed to revise this paragraph to read:

> H-35. Boilers of the wet-bottom type having an external width of over 36 inches shall have not less than 12 inches between the bottom of the boiler and the floor line, with access for inspection. When the width is 36 inches or less, the distance between the bottom of the boiler and the floor line shall not be less than 6 inches, and when any part of the wet bottom is not farther from an outer edge than 12 inches, it shall not be less than 4 inches.

Maintaining Regulator Efficiency

Regulators are a most important item of oxy-acetylene equipment. Steady, efficient operation of the blowpipe depends almost entirely upon the accuracy with which they perform their two chief functions, which are to reduce the high cylinder pressure to the relatively lower working pressure required at the blowpipe, and to maintain the working pressure constant so that there will be no variation in the amount of gas supplied to the blowpipe. Regulators for welding and cutting are of rugged construction and are designed to give a long period of satisfactory service when properly used under the operating conditions for which they are intended.

Be sure there is no oil or grease on hands or gloves. Then open the cylinder valve slightly and quickly close it. This action is generally termed "cracking," and will blow from the valve outlet any dust or dirt that otherwise might enter the regulator.

Before attaching a regulator to a cylinder make sure that the inlet connection on the regulator is also free from particles of dust or dirt which might otherwise find their way into the regulator and cause leakage or creeping.

Never force connections which do not fit. If the thread does not run easily it is an indication that the wrong regulator is in hand, or that an adaptor is needed.

After the regulator has been attached to the cylinder, and before opening the cylinder valve, make certain that the pressure-adjusting screw on the regulator is fully released by turning to the left until it runs free. This closes the valve mechanism inside the regulator so that the high pressure from the cylinder can reach only that part of the regulator which is designed to carry high pressure.

The cylinder valve can now be opened. Open the cylinder valve very slightly at first. This allows the pressure to build up slowly in the high-pressure side of the regulator and thus avoids straining the regulator or gage mechanisms. The fact that this operation has been done properly is indicated by the gage hand on the high-pressure gage moving up slowly until the cylinder pressure is registered. The cylinder valve may then be opened further, fully in the case of an oxygen cylinder but only one and one-half turns in the case of an acetylene cylinder.

When the welding or cutting is to be stopped for a short time, release the pressure-adjusting handles of the regulators by turning them to the left.

When the welding or cutting is stopped for a longer period, the following precautions should be observed: First, close cylinder valve. Then open the blowpipe valves and screw in the pressure-adjusting handle on each regulator so as to release all pressure in the regulator, as shown by the fact that both gage hands drop to zero. Close the regulator-valve mechanism by turning the pressure-adjusting screw to the left until it runs free.

This same procedure should be followed before disconnecting the regulator after use. If the regulator is provided with a dust plug, this should be placed on the regulator inlet connection. The function of these plugs is to keep dirt, grease, oil or any other foreign matter from entering the union nipple and possibly passing into the interior of the regulator.—*Oxy-Acetylene Tips.*

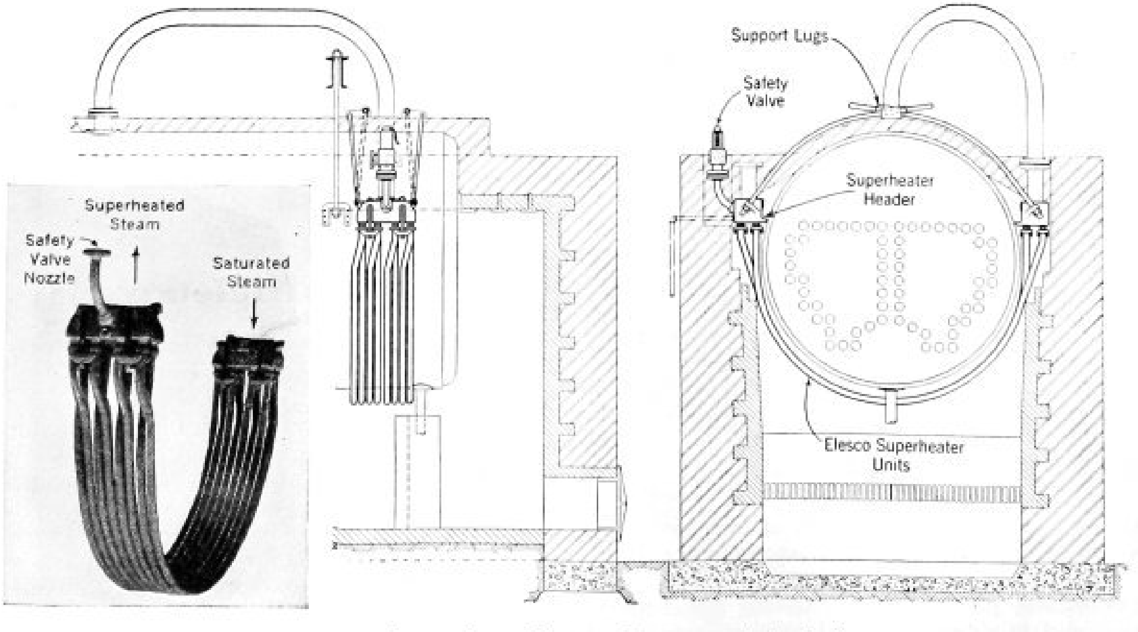
Superheater for Horizontal Return Tubular Boilers

The Superheater Company, New York, has recently developed a new and improved type of superheater, known as the Elesco superheater, designed for easy and economical installation in horizontal return tubular boilers. Unique in design, this superheater is of compact and simple construction and may be readily installed by regular plant attendants, no changes in settings or arches of the boiler or other plant equipment being required. This type of superheater is standard for all horizontal return tubular boilers.

The new superheater comprises simply two cast-steel headers suspended on each side of the boiler shell. These are connected by detachable tubular elements or units extending around the under side of the boiler. The headers are supported by adjustable clamped steel rods. By raising or lowering the headers, the units are correspondingly raised or lowered in the combustion chamber, thus regulating the superheat or final temperature of the steam.

The superheater is automatic in operation. Saturated steam is carried from the boiler nozzle through a connecting pipe to one header, known as the saturated steam header. From there it passes to the other header or superheated steam header through the units under the boiler. Through this passage it is subjected to the relatively intense heat of the gases of combustion in the combustion chamber. The steam, then superheated, passes into the main steam line for use in the plant.

By superheating steam in this manner, many economies, which hitherto were reserved to larger plants, are said to be made possible to the steam plant with horizontal return tubular boilers. Use of superheated steam in the steam power generation reduces the steam consumption of prime movers, condensation losses, and fuel and water consumption by overall plant economy.



Elesco superheater in an horizontal return tubular boiler

The Boiler Maker

VOLUME XXXII NUMBER 8

PUBLISHED Monthly by the Simmons-Boardman Publishing Company, 34 North Crystal Street, East Stroudsburg, Pa., with Executive Offices at 30 Church Street, New York City.

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Chicago: 105 W. Adams Street. Washington, D. C.: 17th and H Streets, N. W. Cleveland: Terminal Tower. San Francisco, Cal.: 215 Market Street.

Subscription price in United States and possessions, 1 year, $2.00, 2 years, $3.00; Canada, including duty, 1 year, $2.50, 2 years, $4.00; foreign countries, 1 year, $3.00, 2 years, $5.00. Single copies, 35 cents each.

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THE BOILER MAKER is a member of the Associated Business Papers, Inc. (A. B. P.), and the Audit Bureau of Circulations, (A. B. C.)

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Contents

GENERAL:

COMMUNICATION:

QUESTIONS AND ANSWERS:

Communication

Horizontal Tubular Boiler Supporting Lugs

TO THE EDITOR:

A brick mason, experienced in steam boiler setting, recently asked the writer for information as to the method of locating supporting lugs upon a boiler shell. He also asked if there was any authentic rule for locating and alining such lugs. The brick mason further stated that, judging by the different angles at which lugs sometimes stood on the same boiler, such lugs seemed to have been thrown at the boiler and stuck where they hit.

The drawing from which a boiler is to be built or set should show the location of the supporting lugs; the dimensions should give the distance of each lug from the ends of the boiler, and the distance from center to center of the lugs. An end view of the shell should also show the vertical location of the supporting lugs, together with the dimensioned distance above the horizontal center of the shell. It is evident that the lugs must be placed high enough so that they will not project into the fire space where the brickwork closes in against the shell of the boiler. This vertical position must be fixed before the pattern for the supporting lugs has been made; and all lugs made from that pattern should always be placed at the same vertical distance above the center line of the boiler. In locating the lugs, it does not matter if the castings be placed an inch higher or lower upon the shell, but the lugs should be so placed that the surface which is to rest upon the brickwork should always lie parallel to the horizontal diameter of the boiler.

One effective, but not very scientific, way of bringing the bearing surfaces of the four lugs into proper horizontal position is as follows: Determine the vertical height of the lugs above the inside bottom of the top row of tubes. Then place a piece of pipe or scantling in the two outside tubes of the top row, the four pieces being all of the same thickness and made to lie horizontally without tipping downwards as they project from the tubes at each end of the boiler. Procure two pieces of board, each with a length at least two feet greater than the diameter of the boiler. The width of these pieces of board shall be such that, when placed on edge upon the pipes or rods projecting from the outermost upper boiler tubes, the top edges of the boards shall be at the height it is desired to place the supporting brackets. The brackets may then be placed against the boiler shell at the required distance from the ends of the shell, the brackets being moved up or down until the bearing surface of each alines with the top edges of the two boards. While held in that position, the rivet holes for the lugs may be marked upon the boiler shell.

In case the bearing surface of a bracket will not lie level and in alinement with the top edges of the boards, slide the bracket or lug up or down until its bearing surface lies parallel with the edges of the boards, but slightly above or below their level. It makes very little difference if a bracket be a little too high or too low; the brickwork will take care of that. However, it sometimes does make a whole lot of difference if a bracket or lug pitches, particularly in an upward direction. The result, in the case of the pair of lugs which rest upon the expansion rollers, is to crowd the brickwork outward and to crack or ruin the setting at those points.

When two-piece lugs are used, one precaution always should be taken. Let each lug be so marked that the pieces belonging together will be used together when the lugs are re-assembled in bricking the boiler into position. Failure to place the proper pieces together is very likely to cause some of the lugs to be "cock-billed" and, consequently crowd away the brickwork. If such sectional lug pieces are not marked, then great care in assembling them must be taken to bring the bearing surfaces level, when placed in position.

As a matter of precaution when marking rivet holes for lugs, it is almost a necessity that one hole be drilled and a temporary bolt inserted before marking the remaining rivet holes. Lugs are such unwieldly things that it is best to fasten them in exact position by a bolt and make sure that the lug has been correctly placed before marking the remaining rivet holes. The extra time required is small in comparison with accuracy secured.

Indianapolis, Ind. JAMES F. HOBART.

Questions and Answers Pertaining to Boilers

This department is maintained for the purpose of helping those who desire assistance on practical boiler shop problems. Inquiries should bear the name and address of the writer. Anonymous communications will not be considered. The identity of the writer, however, will not be disclosed unless the editor is given special permission to do so.

By George M. Davies

Scarfing

Q.—What is scarfing and what parts require scarfing?—G. M.

A.—Scarfing is the drawing or thinning down of a part of a sheet, generally at a corner, which is made to a taper.

Parts requiring scarfing are corners of back heads, throat sheets, firebox door sheets, cylindrical corners of boilers, tanks, stacks and ends of butt straps that slip under adjoining courses.

Efficiency of Diamond Patch

Q.—Your articles in THE BOILER MAKER are very interesting and I always read them. In the April issue, page 90, you discuss the efficiency of circular patches. A formula is given, in the last part of this article, for the ligament efficiency for a diamond patch, and I notice that this does not agree with that published in the A. S. M. E. Code chart, P-13. I am very interested in this subject, and I would appreciate any information that you can give me on this equation. T. C. S.

A.—The formula referred to in the question, as published in the April issue, is as follows:

$$\text{Efficiency} = \frac{2\,(p - d)}{p\sqrt{3 \times \text{sine}^2 B + 1}}$$

where p = shortest pitch of rivets, in inches
d = diameter of rivet holes, in inches
B = angle in degrees, see Fig. 1.

This formula is composed of two parts; namely, the efficiency of the seam and the ratio of the strength of the seam in a diagonal direction to that of an equivalent seam in a longitudinal direction. Thus the formula could read:

$$\text{Efficiency} = \frac{p - d}{p} \times \frac{2}{\sqrt{3 \times \text{sine}^2 B + 1}}$$

The first part of the formula, $\frac{p - d}{p}$, represents the efficiency of the joint in a longitudinal direction, the second part of the formula, $\frac{2}{\sqrt{3 \times \text{sine}^2 B + 1}}$, being the ratio of the strength of a diagonal joint to that of an equivalent longitudinal joint. This part of the formula is derived as follows:

When a boiler is under pressure, the strain on the boiler is different in different directions, being greatest along a circumferential direction and least along the direction of the boiler's length. In fact, the strain on a boiler girthwise, is precisely twice as great as the strain in a lengthwise direction.

Referring to Fig. 2, let P be the pull exerted circumferentially upon a section of the shell one inch long. Then $\frac{1}{2} P$ will be the pull exerted upon an equal length of the girth joints. The total strain on the joint Z is compounded of the total horizontal pull on the length X, and the total vertical pull on the length Y. The stresses acting on Z may therefore be summed up as follows:

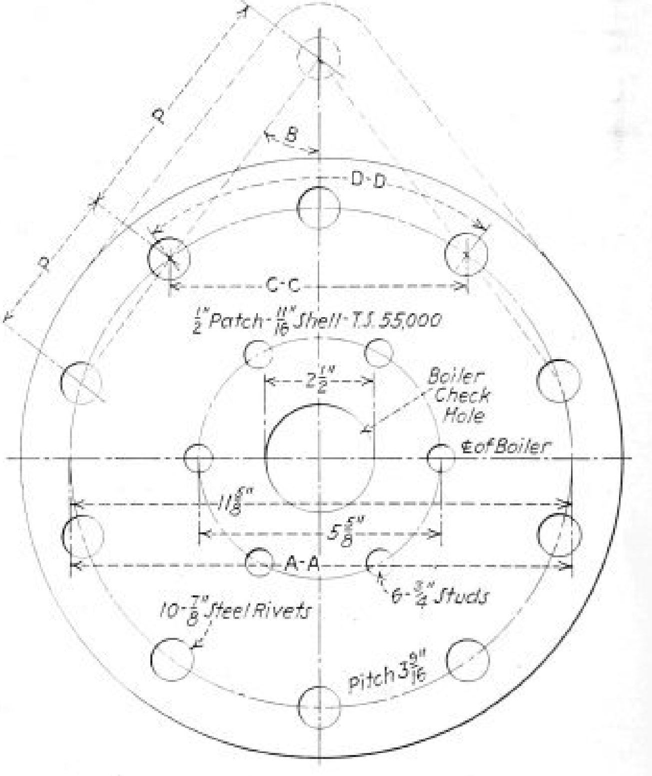


Fig. 1

(1)—A horizontal pull equal to $\frac{1}{2} PX$,
(2)—A vertical pull equal to PY.

These stresses act along the whole length of Z, so to find the stress per unit length of the diagonal joint, we have to divide them both by Z. Hence the horizontal and vertical stresses, on each unit length of Z are $\frac{PX}{2Z}$ and $\frac{PY}{Z}$ respectively. Now we see from the geometry of the figure that $\frac{X}{Z}$ = cosine B, and $\frac{Y}{Z}$ = sine B; and if these substitutions are made, we find that each unit length of the diagonal joint is subjected to the following forces:

(1)—A horizontal stress of $\frac{1}{2} P$ cosine B.
(2)—A vertical stress of P sine B.

This condition is indicated in the diagram given in Fig. 3.

The total stress X is found by adding the squares of the two forces indicated by the dotted lines in Fig. 3,

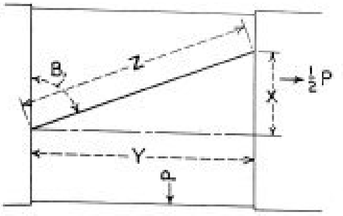


Fig 2

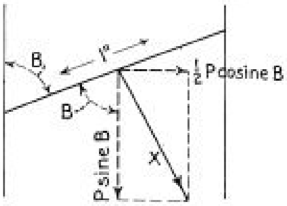


Fig. 3

then extracting the square root of the sum. This gives us $X = \sqrt{\frac{1}{4} P^2 \text{ cosine}^2 B + P^2 \text{ sine}^2 B} =$

$$\tfrac{1}{2} P \sqrt{\text{cosine}^2 B + 4 \text{ sine}^2 B}$$

This may be simplified by noting that $\text{cosine}^2 B = 1 - \text{sine}^2 B$, substituting this in the foregoing equation we have:

$$X = \tfrac{1}{2} P \sqrt{1 + 3 \text{ sine}^2 B}$$

which is the desired expression for the total stress acting upon each unit length of the joint.

Total stress on diagonal joint $= \frac{1}{2} P \sqrt{1 + 3 \text{ sine}^2 B}$
Total stress on longitudinal joint $= P$.

The ratio of the strength of the diagonal seam to the equivalent longitudinal seam is as

$$= \frac{P}{\frac{1}{2} P \sqrt{1 + 3 \text{ sine}^2 B}}$$

$$= \frac{2P}{P \sqrt{1 + 3 \text{ sine}^2 B}}$$

$$= \frac{2}{\sqrt{1 + 3 \text{ sine}^2 B}}$$

This is the ratio of the strength of a diagonal joint to that of an equivalent longitudinal joint.

Strength of Staybolts

Q.—Would it reduce the tensile strength and efficiency of a staybolt, 1⅛ inches in diameter, to countersink it with a 7/16-inch drill, 5/16 inch deep, on the outside of a locomotive boiler carrying 200 pounds working pressure? The bolts are applied from the outside of the boiler, cut with an oxy-acetylene torch, driven with ball die, then countersunk. Will you give the proper requirements for countersinking staybolts?—J. T.

A.—The strength of a staybolt is determined by dividing the load on the staybolt by the least cross-sectional area of the bolt, giving the load in pounds per square inch on the bolt in tension. In the case of the staybolt given in the question, this would be the load on the staybolt divided by the cross-sectional area of the bolt at the root of the threads minus the cross-sectional area of the 7/16-inch diameter drilled hole.

Theoretically, however, the above is not exactly true should the countersink in the end of the staybolt not extend into the bolt the full thickness of the outside plate, in which case the threads on the staybolt from the inside edge of the outside plate, even with the bottom of the countersunk hole, could be taken in shear, as helping to support the load on the staybolt. The above, however, is exceptional to the rule and question should be raised with the authorities under whose jurisdiction the boiler comes before using this type of connection.

The A. S. M. E. Code Par. 220 (c) and (d) provides as follows:

(c) The maximum allowable stress per square inch at point of least net cross-sectional area of staybolts and stays or braces shall be as given in Table P-7. In determining the net cross-sectional area of drilled or hollow staybolts, the cross-sectional area of the hole shall be deducted.

(d) The length of the stay between supports shall be measured from the inner faces of the stayed plates. The stresses are based on tension only. For computing stresses in diagonal stays, see Pars. P-221 and P-222.

It will be noted here that the stresses are to be based on tension only.

The Interstate Commerce Commission Rule 26 provides:

(26) *Telltale Holes.*—All staybolts shorter than 8 inches applied after July 1, 1911, except flexible bolts, shall have telltale holes 3/16 inch in diameter and not less than 1¼ inches deep in the outer end. These holes must be kept open at all times.

Dome Cap of Locomotive Boiler

Q.—Kindly advise how the dome cap of a locomotive boiler, as shown in the attached sketch, should be calculated. A. B.

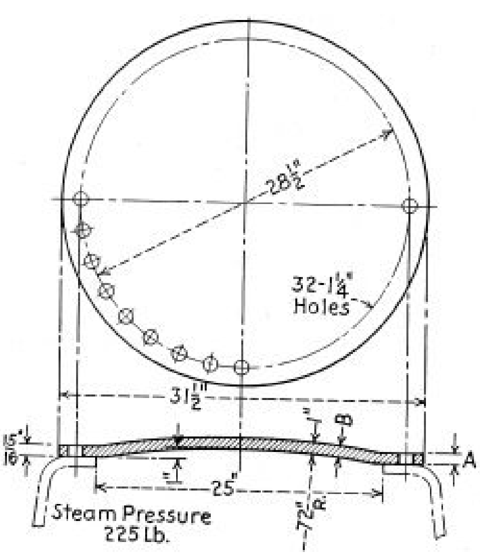
A.—The A.S.M.E. Code gives the following formulas for determining the thickness of heads:

(1)—The thickness of a blank unstayed dished head with the pressure on the concave side, when it is a segment of a sphere, is as follows:

$$t = \frac{8.33 \times P \times L}{2 \times TS}$$

where t = thickness of plate, inches
P = maximum allowable working pressure, pounds per square inch
TS = tensile strength, pounds per square inch originally stamped on the plate used in forming the head
L = radius to which the head is dished, measured on the concave side of the head, inches.

(2)—The thickness required in an unstayed flat head, which is unpierced and is rigidly fixed and supported at the bounding edges by riveted or bolted attachments to the shell or side plate, is as follows:



Locomotive boiler dome cap

$$t = a \sqrt{\frac{0.162\ P}{S}}$$

where t = thickness of plate in head, inches

a = diameter, or short side of area measured to the center of the inside row of rivets or bolts, inches
P = maximum allowable working pressure, pounds per square inch
S = allowable unit working stress, pounds per square inch
TS = ultimate tensile strength, pounds per square inch stamped on the plate as provided for in the specifications for the material.

For locomotive boilers other than those built under the A.S.M.E. Code, the second formula should be used modified as follows:

a = diameter or short side of area measured to the center of the dome cap gasket
S = allowable unit working stress, pounds per square inch or $\dfrac{TS}{\text{Factor of safety}}$

Assuming a locomotive boiler having a dome cap as illustrated in the question:

Given: Steam pressure = 225 pounds per square inch
Diameter of gasket = 25½ inches
Factor of safety, I.C.C. = 4
Tensile strength of steel = 55,000 pounds per square inch.

Substituting in formula (2):

$$S = \frac{55{,}000}{4} \text{ and } t = a\sqrt{\frac{0.162\ P}{S}}$$

$$S = 13{,}750,\ t = 25.5\sqrt{\frac{0.162 \times 225}{13{,}750}}$$

$$t = 25.5 \times \sqrt{0.00625}$$
$$t = 25.5 \times 0.051$$
$$t = 1.30 \text{ inches.}$$

The thickness of the dome cap at A should be $1\frac{5}{16}$ inches.

Dome caps of this type are as a rule finished where the caps are secured to the dome. This finish would require the plate from which the dome cap was made to be at least $\frac{1}{16}$ inch thicker than the thickness of the finished portion of the cap, thus making the thickness of the unfinished dished portions of the dome cap B 1⅜ inches thick.

It will be found by checking in formula (1) that this thickness for the dished portion of the cap is ample.

The number of studs required is determined by dividing the total load on the head by the allowable working load for each stud.

The allowable working load on studs can be found in any engineering handbook.

Welding Boiler Shells

Q.—When welding the seam of a boiler, does the shell metal melt or become plastic to any degree when the electrode melts? Can you explain how the molten weld metal blends with the base metal if it does not melt or soften in any degree? W. M. G.

A.—Fusion welding is the process of securing two pieces of metal by the application of concentrated heat of high temperature to the edges of the pieces to be joined, usually with the addition of some filler metal at the joint. The edges of the metal at the joint are brought to a molted state by the high temperature of an electric arc or the flame of a gas torch and the additional metal required is provided in the form of a rod or wire which is likewise melted into the joint.

Welding can only take place when molten metal from the electrode falls on molten metal of the pieces to be welded.

Circumference of Inside and Outside Angle

Q.—I would like to know how you figure the circumference of an inside and outside angle.—P. J. T.

A.—There are two good working rules for determining the exact length of an angle iron, where the angle iron is formed into a circle to fit either inside or outside a circular tank or pipe. These rules apply whether the angle iron is bent hot or cold.

For an outside angle, with the heel of the angle toward the center of the circle, the diameter used in computing the length of the angle iron may be obtained as follows: Taking the symbols indicated in Fig. 1 and calling the inside diameter of the ring D, then the desired diameter is $D + \frac{1}{3}W + T$. The length of the angle iron is found by multiplying this diameter by 3.1416.

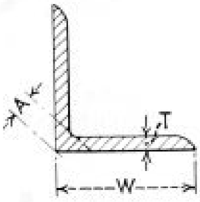


Fig. 1

For an inside angle, if D is the outside diameter of the ring, the diameter to be used when computing the length of the angle iron is $D - (\frac{1}{3}W + T)$; and the length is 3.1416 times the amount obtained from this formula.

Another rule is as follows: For outside angles, the diameter used in computing the length is taken as $D + 2A$, where D is the inside diameter of the ring and A is the thickness of the root of the angle measured diagonally as indicated in Fig. 1. For inside angles, if D is the outside diameter of the ring, then the diameter to be used in computing the length should be $D - 2A$.

Some small allowances are frequently made, due to the stretch caused by punching the holes in the angle iron, but this is best determined by observation, since no definite allowance can be stated. By inserting in the holes the small pieces which have been punched out, the angle irons may be bent after the holes have been punched, to comparatively short radius without tearing the metal from the rivet holes to the edge of the angle iron or destroying the shape of the holes.

Using Steam at Temperature of 860 Degrees

Two points of distinction are claimed for a new boiler and turbo-generator recently placed in operation at the Burlington station of the Public Service Corporation of New Jersey. The arrangement is said to be the first in this country to employ steam at the high pressure of 730 pounds per square inch and temperature of 860 degrees F.; the 18,000 kilowatt turbine is the first of its size in the country to operate at 3600 revolutions per minute, most large turbines having a rotor speed of 1,800 revolutions per minute.

The new boiler has a greater capacity than the combined output of ten older boilers in the plant. Those boilers are now held in reserve while steam from the new boiler is used at reduced pressure to operate the turbines they once served.—*The Locomotive.*

Ralph P. La Porte, formerly a member of the executive staff of the Edward G. Budd Manufacturing Company, Philadelphia, Pa., has been appointed manager of manufacture of the International Heater Company, Utica, N. Y.

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States

Arkansas
California
Delaware
Indiana
Maryland
Michigan
Minnesota
Missouri
New Jersey
New York
Ohio
Oklahoma
Oregon
Pennsylvania
Rhode Island
Utah
Washington
Wisconsin
District of Columbia
Panama Canal Zone
Territory of Hawaii

Cities

Chicago, Ill.
Detroit, Mich.
Erie, Pa.
Evanston, Ill.
Houston, Tex.
Kansas City, Mo.
Los Angeles, Cal.
St. Joseph, Mo.
St. Louis, Mo.
Scranton, Pa.
Seattle, Wash.
Memphis, Tenn.
Nashville, Tenn.
Omaha, Neb.
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Philadelphia, Pa.
Tampa, Fla.

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Missouri
New Jersey
New York
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Oklahoma
Oregon
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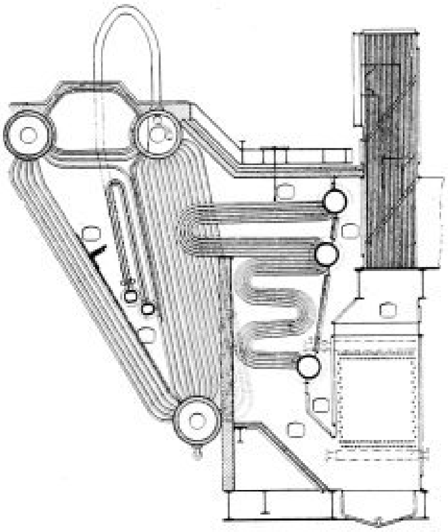
Seattle, Wash.
Tampa, Fla.

Selected Boiler Patents

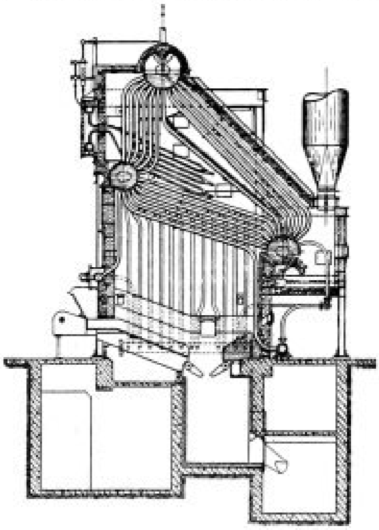
Compiled by Dwight B. Galt, Patent Attorney, 1372-1376 National Press Building, Washington, D. C. Readers wishing copies of patents or any further information regarding any patent described, should correspond directly with Mr. Galt.

1,823,701. STEAM BOILER. ARTHUR D. PRATT, OF SHORT HILLS, N. J., ASSIGNOR TO THE BABCOCK & WILCOX COMPANY, OF BAYONNE, N. J., A CORPORATION OF NEW JERSEY.



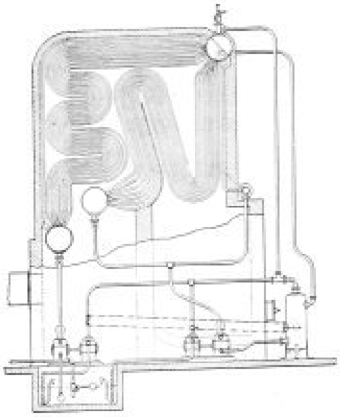
Claim.—In combination, a steam boiler and its setting comprising two side by side flues, the gases flowing upwardly in a first flue and downwardly in a second flue, vertically extending boiler tubes located in said first flue, and reheater tubes having portions overlapping with said boiler tubes in said first flue and having other portions extending across said second flue. Ten claims.

1,827,865. BOILER. PAUL WRIGHT, OF BIRMINGHAM, ALA.



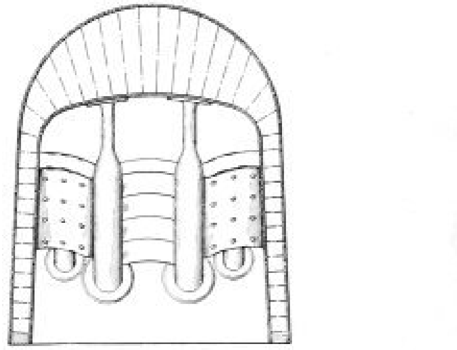
Claim.—In a steam boiler, the combination of mud, water and steam drums, banks of tubes connecting said drums in a closed circuit, a plurality of headers exterior to the furnace, tubing connecting the mud drum with each header, and a bank of tubes extending within the furnace from each header directly to the water drum. Eleven claims.

1,828,870. DOUBLE-CIRCUIT FORCED-CIRCULATION WATER-TUBE BOILER. CHARLES E. LUCKE, OF NEW YORK, ASSIGNOR TO THE BABCOCK & WILCOX COMPANY, OF BAYONNE, N. J., A CORPORATION OF NEW JERSEY.



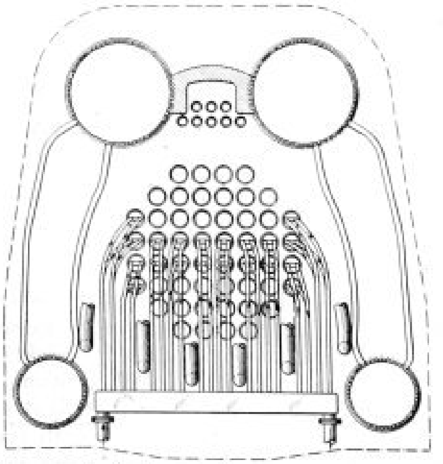
Claim.—A steam generating system having in combination a boiler furnace, steam generating wall tubes spirally arranged within said furnace so as to enclose the combustion chamber thereof, steam generating wall tubes spirally arranged so as to enclose the outlet flue, a circulating pump for forcing water into each group of said tubes, a separate connection from said pump to each group of said tubes, an economizer section, a steam and water collector for receiving steam and water from said second group and water from said economizer, a feed pump for forcing water through said economizer into said collector, a steam and water separating chamber connected to said first group and to said collector, and a connection for supplying said circulating pump with water from said separating chamber. Seven claims.

1,822,937. LOCOMOTIVE BOILER FIREBOX. EMMETT FRANK SMITH, OF ST. LOUIS, MO., ASSIGNOR TO LOCOMOTIVE FIREBOX COMPANY, OF CHICAGO, ILL., A CORPORATION OF DELAWARE.



Claim.—In combination with a locomotive firebox including side sheets and a front sheet, a water circulating element connected at one end to said front sheet and extending upwardly and rearwardly therefrom, a hollow water arch member disposed between one of said side sheets and said water circulating element, said arch element being open at one side and being fixed in said side sheet and closed at its other side which is disposed adjacent to but out of engagement with said water circulating element and means connecting the front end of said arch element with and opening through said front sheet. Two claims.

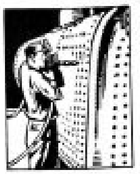
1,784,378. LOCOMOTIVE SUPERHEATER. JOHN E. MUHLFELD, OF SCARSDALE, N. Y., AND JOHN C. CHAPPLE, OF BROOKLYN, N. Y.



Claim.—In combination with a locomotive firetube boiler and firebox and a vertically positioned protection wall at the front end of the firebox in spaced relation to the firetube flue sheet, a superheater header mounted between said protection wall and the flue sheet below the open ends of the firetubes, and superheater elements connected with said header and having parts positioned contiguous to said protection wall and extending forwardly into the firetubes. Three claims.

The Boiler Maker

Reg. U. S. Pat. Off.



S. S. Observation Boiler Explosion

Negligence in its most criminal form cost the lives of seventy-two persons and injured about fifty more when the boiler aboard the *S. S. Observation* exploded in the East River, New York, on September 9. While numerous inquiries by civic and governmental officials are under way at the present time to determine the cause of this disaster, no evidence has been developed to indicate the exact condition of the boiler.

According to records, the *Observation* was built at Brooklyn, N. Y., in 1888 and had a length of 92.4 feet, a beam of 17.8 feet, a depth of 9.2 feet and a gross tonnage of 122. She had two engines and one return tubular boiler 6 feet 3 inches in diameter and 11 feet 6 inches long. This boiler had been installed in 1909.

While most of the evidence submitted at the inquiries is of a more or less indefinite nature, it appears that nearly all the rules for good boiler operation had been broken. It was testified that salt water had been used in the *Observation*. While no Federal law prohibits such a practice, it is well known that continued use of salt water causes caustic embrittlement and its long continued use might be interpreted as negligence.

Former employees of the owners of the boat told of the refusal of Captain George Forsythe, one of the owners, to make repairs deemed necessary to put the boiler in proper condition. One of the former firemen stated that on June 1 last a serious leak developed in the port waterleg. It was welded on June 11 but by June 26, the leak had returned with such seriousness that the boiler had to be filled three or four times a day.

It was further testified that the boiler had been mended several times with cement in violation of Federal regulations, but the water and steam continued to leak. It also developed that the Sound Welding Company, which repaired the ship's boiler, did so without official authorization and filed no report of the work with the government. The welder who did the job had no experience in marine welding.

Irreplaceable damage has been caused by an owner who desired to save a few pennies by using salt instead of fresh water, by neglecting necessary repairs and by employing unauthorized and unskilled welders to patch the boiler, instead of thoroughly overhauling the boiler at the proper time. Such practices as these are only manifest when a major disaster of this sort occurs.

This disaster should serve as a drastic warning for the owners of hundreds of old excursion boats now operating on the coastal and inland waters of the United States. It is natural in these times to attempt to reduce operating and maintenance expenses to a minimum; but there is no excuse for a shipowner risking the lives of hundreds of passengers by refusing to maintain his boilers in proper condition.

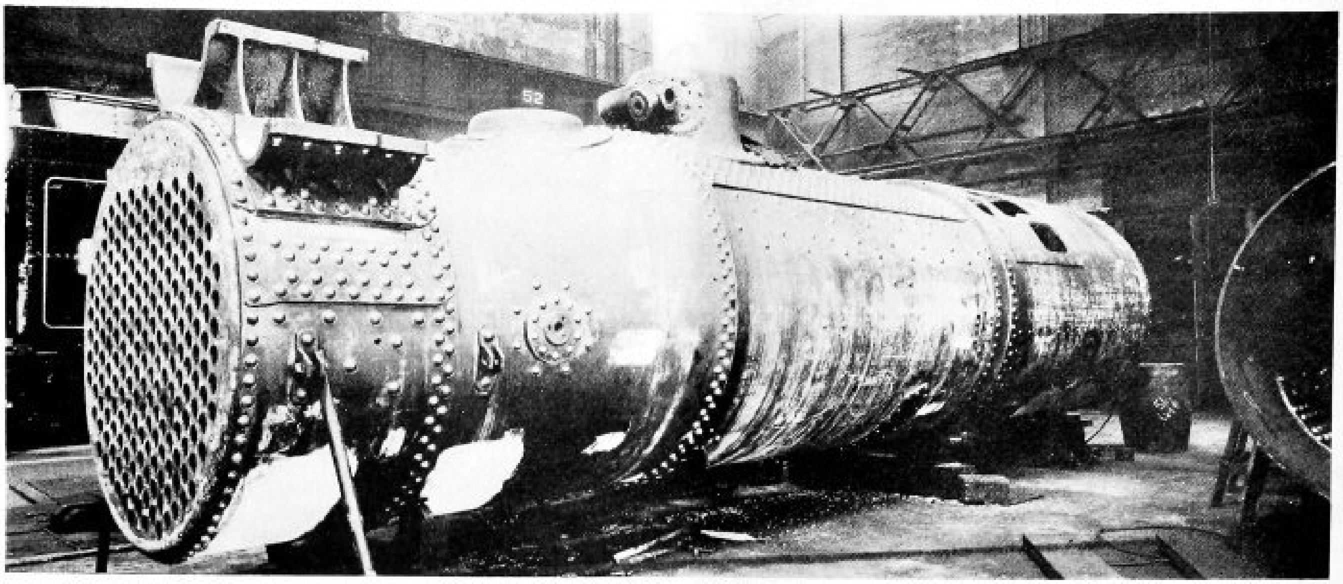
Locomotive Design

So little information has been available covering the construction and subsequent performance of the first multi-pressure locomotives to be built in this country and in Canada, that the papers presented on this subject at the Spring meeting of the American Society of Mechanical Engineers are of particular interest to the entire railway industry. An abstract of the paper presented by H. B. Bowen, chief of motive power and rolling stock of the Canadian Pacific, describing the C.P.R. locomotive No. 8000, appears in this issue. Further abstracts by J. B. Ennis, vice-president of the American Locomotive Company and F. A. Schaff, president of the Superheater Company, will be published later.

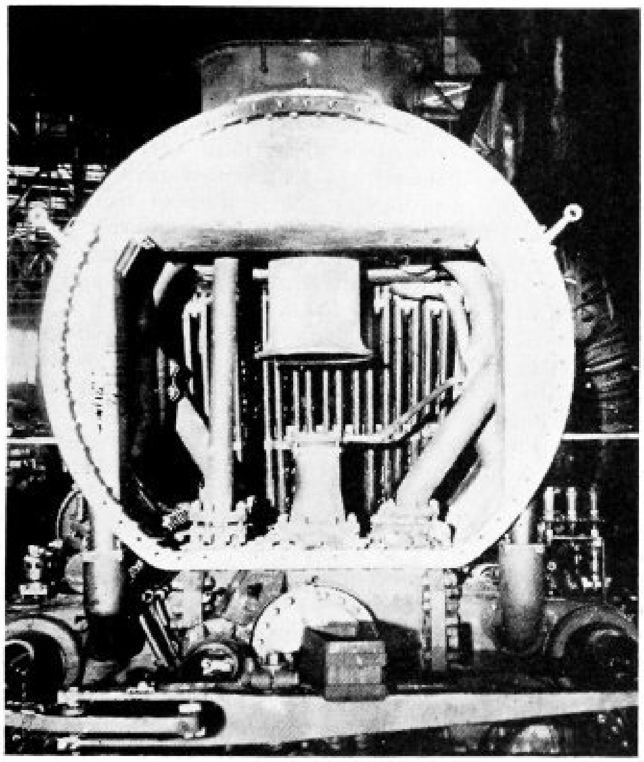
It has long been recognized that locomotives of even a decade ago cannot perform the service demands of heavy tonnage, high speeds and long runs that must be met by the railroads today. Therefore, in order to maintain the motive power of the country in a condition efficiently to meet these demands, more or less drastic changes in design were to be expected. In the conventional type boiler, the possible pressure and total temperature increases are limited by dimensions and by the materials used in construction.

By turning to the combined watertube and firetube locomotive, several of which have been in operation for some years, the designers accomplished an increase in pressures and temperatures with consequently greater capacities. In the process of evolving the locomotive of the future, the present development through the use of multi-pressure boilers offers excellent possibilities. Any future construction program of motive power undoubtedly will take these developments into account.

For this reason every man from the mechanical engineers and superintendents down to the men in the shops, who will have the actual maintenance of these locomotives in charge, should become familiar with their construction and operation. It is certain that with the return of anything like normal tonnage demands the railroads of the country will commence the much-needed replacement of obsolete motive power. Any such program of replacement will undoubtedly take advantage of the multi-pressure design which represents the greatest advance from conventional locomotive construction yet undertaken.



Right side of the low-pressure boiler



Smokebox of C. P. R. locomotive No. 8000

Developing the Multi-Pressure Locomotive

Three papers discussing various phases of the design and operating performance of the Canadian Pacific multi-pressure locomotive No. 8000 were presented at the spring meeting of the American Society of Mechanical Engineers which was held at Bigwin, Ont., June 27 to July 1, 1932, inclusive. These papers composed a symposium on the development of high steam pressures for locomotives which was sponsored by the Railroad Division. They were presented by H. B. Bowen, chief of motive power and rolling stock, Canadian Pacific; J. B. Ennis, vice-president, American Locomotive Company, and F. A. Schaff, president, the Superheater Company. C. E. Brooks, chief of motive power and car equipment, Canadian National, presided at the meeting.

At the present time there are five locomotives of multi-pressure design which have been built for service in Europe and North America. The two locomotives on this continent are the No. 8000, 2-10-4 type on the Canadian Pacific, and the New York Central No. 800, 4-8-4 type. Both of these locomotives are equipped with what is known as Elesco boilers.

The C. P. R. locomotive was placed in service the latter part of May of last year. The second locomotive to be equipped with the Elesco boiler was delivered to the New York Central from the Schenectady, N. Y., plant of the American Locomotive Works the latter part of 1931. Both locomotives were designed through the co-operative efforts of the American Locomotive Company, the Superheater Company, and the mechanical departments of the respective railroads.

As was pointed out by various speakers at the Bigwin meeting, both locomotives follow the same general scheme of design with respect to the construction of the boilers and fireboxes. The essential differences pertain largely to the frames and running gear because of the differences in wheel arrangement. The New York Central locomotive, No. 800, which has been designated Class H8-1a, burns soft coal, while the Canadian Pacific, assigned Class T4a, burns oil.

The C. P. R. No. 8000, Class T4a, was built for comparison with the road's Class T1a locomotives, which have the same wheel arrangement. A report of some of the results of the comparative tests are included in Mr. Bowen's paper. The following is an abstract of the paper presented at the Bigwin meeting by Mr. Schaff, which describes the steam generating system of the Canadian Pacific locomotive.

The development of steam generation has been characterized by a steady trend towards higher steam pressures. At the beginning of the century, locomotive boiler pressures had risen to 180 pounds or 190 pounds from 150 pounds or 160 pounds, which had been considered high pressures during the preceding decade. By 1910, 200 pounds was generally regarded as the limit for simple-expansion locomotives, although during the compound era 215 and 225 pounds had been used to some extent.

With the introduction of superheated steam there was a lull in the advance of pressures as the advantages gained by the use of superheat were adequate to meet the then requirements of added capacity and fuel economy. In more recent years the insistent demands for greater economy and higher sustained locomotive capacity with heavier loads and faster schedules caused designers again to increase the working steam pressure and also to provide for higher steam temperatures.

The locomotive boiler in all its major features has changed but little, save in size. It is true that proportions have been changed and many devices have been added which have materially increased the efficiency, but in the main the boiler is as it was a century ago. Brotan in Europe and others, following in his wake, built watertube fireboxes to better provide for higher working pressures. This trend evidenced itself in America through the introduction of the Jacobs-Schupert firebox on the Atchison, Topeka & Santa Fe, the Baldwin Locomotive No. 60,000, the McClellan firebox on the New York, New Haven & Hartford, and more recently, the watertube fireboxes on the Delaware & Hudson. The Gresley-Yarrow watertube boiler on the London & North Eastern in England, and the Winterthur locomotive in Switzerland, represent constructions which aimed at suitability for higher working steam pressures.

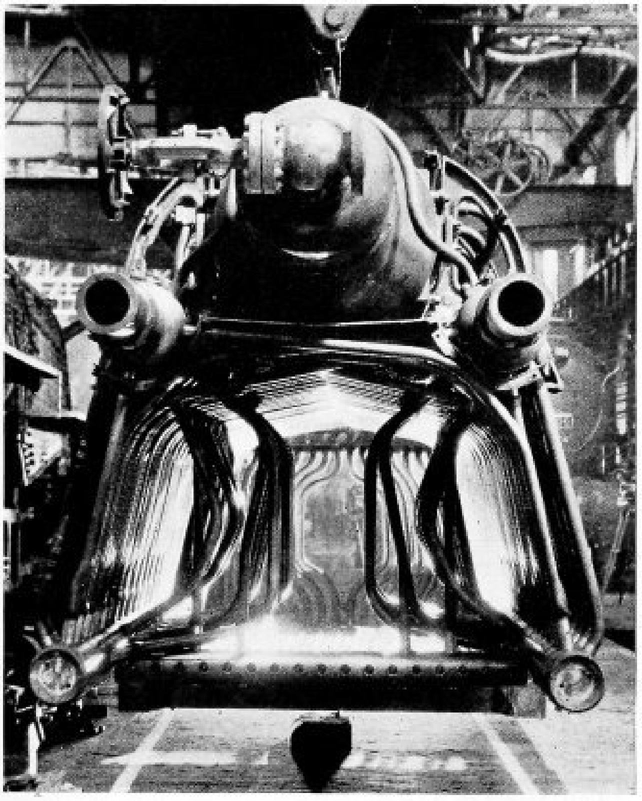
With the present state of the art it is generally considered that the conventional type of locomotive boiler with stayed surfaces is not suitable for pressures above approximately 300 pounds per square inch. It is essential therefore to utilize some sort of watertube construction for pressures materially higher than this figure.

The Elesco multi-pressure system as used on the Canadian Pacific No. 8000 seems to meet adequately not only the requirements for substantially high pressure, but offers corollary advantages contributing to ease of maintenance.

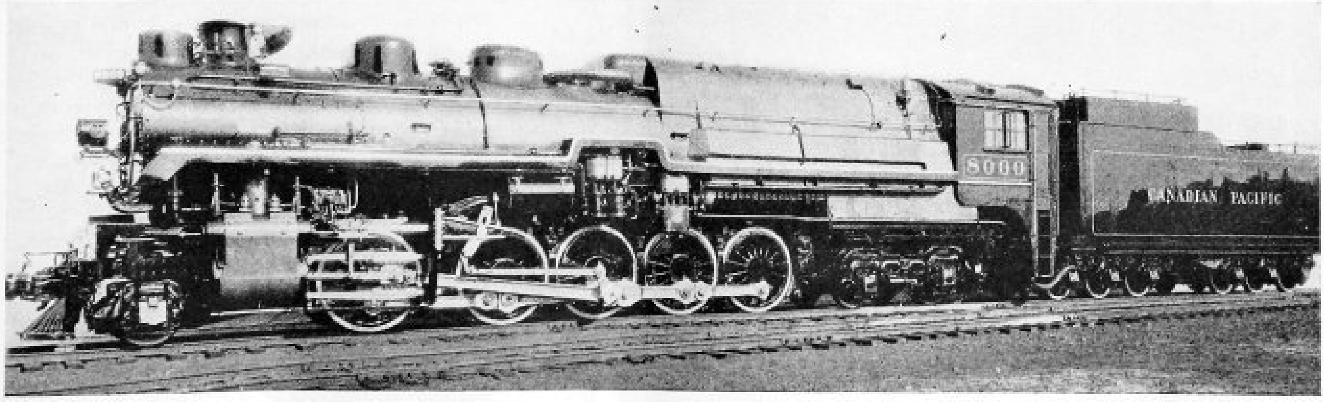
The system consists of three separate units: The closed circuit, the high-pressure boiler, and the low-pressure boiler.

The Closed Circuit: Referring to the illustration below, showing the construction of the firebox and closed circuits, the bifurcated seamless steel tubes are 2 inches and 2½ inches outside diameter. The front view shows that the furnace walls are formed by one straight riser tube and one bent cross-over tube changing alternately.

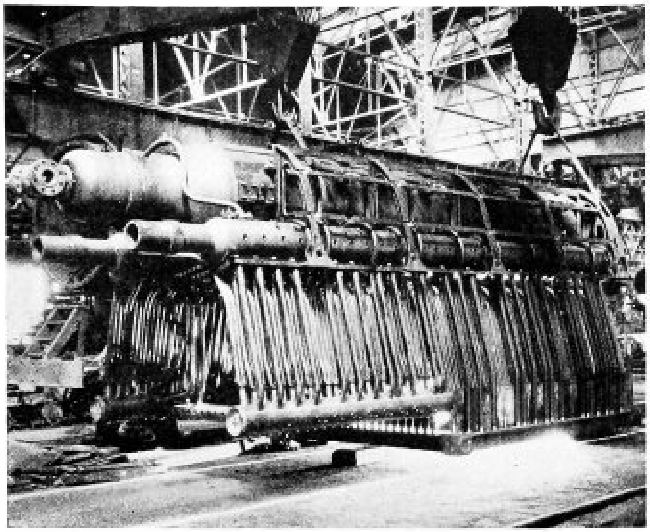
The tube ends are rolled into the firebox ring at the bottom and at the top into the steam separator drums. A special feature of the boiler is that the downcomer tubes are not located within the gas path but outside of the wall of riser tubes, thus insuring a positive water circulation without the dangerous reversals of flow encountered on some designs of watertube boilers. The



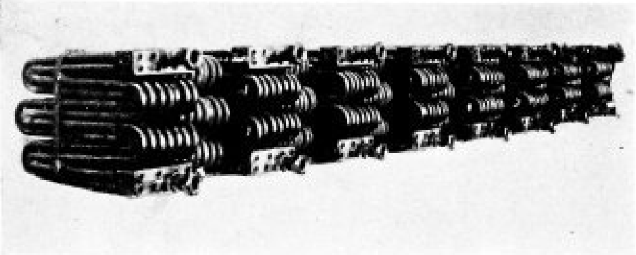
Closed circuit and high-pressure boiler looking toward the rear



Canadian Pacific locomotive No. 8000 equipped with the Elesco boiler



Front view of the left side of the closed circuit and high-pressure boiler



Heat-transfer elements as enclosed in the high-pressure drum

tube system is filled approximately to the center of the steam separator drums with distilled water, which serves as a heat carrier.

The steam generated is taken from the steam space of the separator drums and carried by short riser pipes to the heat-transfer elements located in the main steam drum on top of the firebox. During its flow through these elements, the steam, by losing its latent heat to the lower-temperature feed water of the high-pressure boiler condenses, thus generating indirectly the high-pressure live steam used in the high-pressure cylinder. The condensate flows out of the lower part of the heat transfer elements through the downcomers or condensate tubes to the water collectors at the bottom of the firebox and combustion chamber. There it again enters the riser tubes, thus completing the circuit. As distilled water is used and the system is sealed, no scale can be formed in the watertubes which are in contact with the fire.

This is probably the outstanding feature of this design as one of the principal difficulties with watertube construction in locomotive boilers has been the necessity for careful and frequent cleaning of the tubes to avoid accumulation of scale. This cleaning is a slow and expensive operation requiring the locomotive to be out of service a considerable portion of the time. With the indirect system as used on the No. 8000, the watertubes will never require cleaning. Thus, aside from the time and labor saved, there is the further advantage of avoiding the breaking and remaking of numerous high-pressure joints at the washout plugs, etc.

The flow of water and steam in the system is a natural circulation depending upon the hydraulic head of the liquid and unassisted by mechanical means. There is no fixed steam pressure in the closed circuit but it depends on the steam output of the locomotive. Under normal conditions the boiler carries about 1350 pounds per square inch, and under peak loads it may rise to 1600 pounds.

The pressure in the closed circuit is such that the required temperature differential is obtained and the heat flow from the closed-circuit steam to the high-pressure boiler water produces and maintains the 850-pound pressure in the high-pressure boiler. The two safety valves of the closed circuit are set at a pressure of 1700 pounds per square inch, in order that there shall

Comparison of Multi-Pressure Locomotives

Road	C.P.R.	N.Y.C.	P.L.M.	L.M. & S.	G.-St.
Boiler pressures:					
Closed circuit	1300-1700 lb.	1300-1700 lb.	1300-1560 lb.	1300-1700 lb.	1300-1700 lb.
High-pressure boiler	850 lb.	850 lb.	850 lb.	900 lb.	850 lb.
Low-pressure boiler	250 lb.	250 lb.	200 lb.	250 lb.	200 lb.
Firebox:					
Grate area	77 sq. ft.	65 sq. ft.	41.8 sq. ft.	28 sq. ft.	26.6 sq. ft.
Volume	353 cu. ft.	318 cu. ft.	236 cu. ft.	175 cu. ft.	212 cu. ft.
Fuel	Oil	Bit.	Bit.	Coal	Bit.
Heating surfaces:					
Closed circuit	520 sq. ft.	430 sq. ft.	322 sq. ft.	200 sq. ft.	218 sq. ft.
Low-pressure boiler	3,746 sq. ft.	3,229 sq. ft.	1,830 sq. ft.	1,335 sq. ft.	1,370 sq. ft.
Total evaporating	4,266 sq. ft.	3,659 sq. ft.	2,152 sq. ft.	1,535 sq. ft.	1,588 sq. ft.
High-pressure superheater	941 sq. ft.	835 sq. ft.	400 sq. ft.	285 sq. ft.	323 sq. ft.
Low-pressure superheater	1,102 sq. ft.	1,070 sq. ft.	415 sq. ft.	385 sq. ft.	320 sq. ft.
Total superheater	2,043 sq. ft.	1,905 sq. ft.	815 sq. ft.	670 sq. ft.	643 sq. ft.
Total combined	6,309 sq. ft.	5,564 sq. ft.	2,967 sq. ft.	2,205 sq. ft.	2,231 sq. ft.
Heat-transfer coils, inside	750 sq. ft.	660 sq. ft.	460 sq. ft.	295 sq. ft.	315 sq. ft.
Boiler dimensions:					
Low-pressure boiler, inside dia.	82 in.	80 in.	63¾ in.	66⅛ in.	61¾ in.
Low-pressure boiler, length over tube sheets	19 ft. 1⅜ in.	18 ft. 2⅛ in.	16 ft. 3 in.	13 ft. 2½ in.	13 ft. 10¾ in.
Low-pressure boiler, no. of flues	214	194	132	130	116
Low-pressure boiler, diam. of flues	3½ in.	3½ in.	3¼ in.	3 in.	3¼ in.
High-pressure boiler, inside diam.	39 in.	39 in.	37⅜ in.	36 in.	36 in.
High-pressure boiler, length overall	25 ft. 2 in.	23 ft. 9½ in.	20 ft. 4⅜ in.	14 ft. 11 in.	16 ft. 11 in.
Steam separator drums, inside diam.	12 in.	12 in.	10¼ in.	10¼ in.	11 in.
Combustion chamber drums, inside diam.	7¼ in.	7¼ in.	3 in.	3¼ in.	3⅛ in.
Closed circuit tubes, outside diam.	2 and 2½ in.	2 and 2½ in.	2 in.	2 in.	3 in.
No. of superheater units—H.P.	49	44	32	24	30
L.P.	61	55	66	32	56
Superheater tubes, outside diam.	1 3/16 in.	1 3/16 in.	15/16 in.	1 in.	15/16 in.
Water content:					
Closed circuit	2,660 lb.	2,360 lb.	....	1,250 lb.	1,360 lb.
High-pressure boiler	5,180 lb.	4,665 lb.	4,000 lb.	2,500 lb.	3,150 lb.
Low-pressure boiler	16,500 lb.	16,750 lb.	8,900 lb.	6,775 lb.	7,500 lb.
Ratios:					
Closed circuit heat. sur. ÷ grate area	6.75	6.62	7.7	7.15	8.2
Total evap. sur. ÷ grate area	55.4	56.2	51.4	54.8	59.7
Total heating sur. ÷ grate area	82.0	85.7	70.9	78.8	83.8
Closed circuit heat. sur. ÷ total evap. heat. sur.	12.2	11.3	15.0	13.0	13.7
Heat. trans. coil heat. sur. ÷ closed cir. heat. sur.	1.44	1.61	1.43	1.48	1.45
High-press. sup'h'r. sur. ÷ closed cir. heat. sur.	1.81	2.03	1.24	1.43	1.48
Low-press. sup'h'r. heat. sur. ÷ low-press. heat. sur.	0.295	0.33	0.227	0.288	0.233

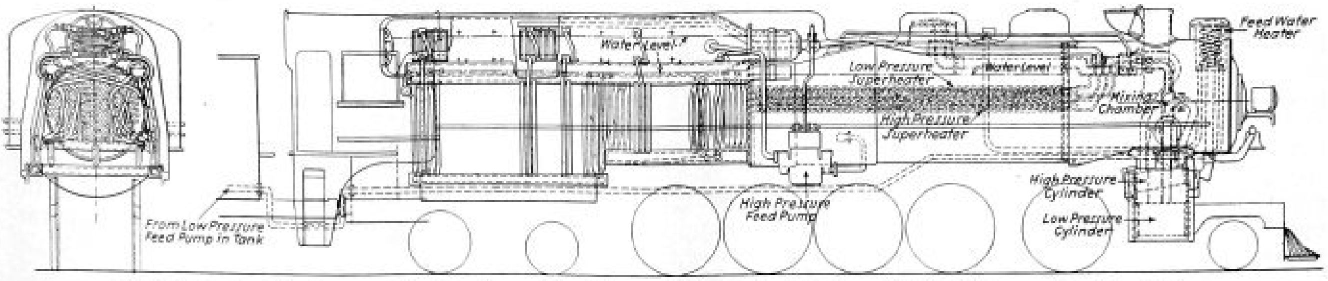


Chart showing the steam and water flow of the Elesco boiler as installed on C. P. R. locomotive No. 8000

be an adequate range which will preclude the loss of water.

The illustrations will make clear the arrangement of the high-pressure boiler. The high-pressure steam of 850 pounds per square inch is generated in a seamless forged nickel-steel drum 39 inches in inside diameter, which is located above the closed circuit and protected from contact with flames and firebox temperatures by the cross-over tubes of the closed circuit, by a plate of heat-resisting steel and by lagging. The high-pressure steam is generated indirectly by means of the heat-transfer elements in the boiler drum. The average water level in the drum is about 7 inches above the center line, so that from 80 to 90 percent of the elements are covered by water. The steam flows through two dry pipes located at the highest part of the boiler and perforated at their upper circumference. They emerge as one pipe at the front end of the drum, lead to a shut-off valve and from there to the high-pressure superheater header in the smokebox of the locomotive.

One of the illustrations on page 182, shows the third part of the system, the low-pressure boiler which carries 250 pounds per square inch pressure. It is similar to the barrel part of an ordinary boiler with a circular rear flue sheet riveted to the boiler shell instead of to the firebox. The boiler is fitted with 3½-inch outside diameter flues containing a 1$\frac{3}{16}$-inch outside-diameter superheater unit of the Type E design. The units are divided into two groups near the vertical center line, those on the right-hand side comprising the high-pressure superheater and those on the left-hand side forming the low-pressure superheater. Both superheaters are made of the same material and corresponding units have like dimensions. Maximum interchangeability is therefore provided.

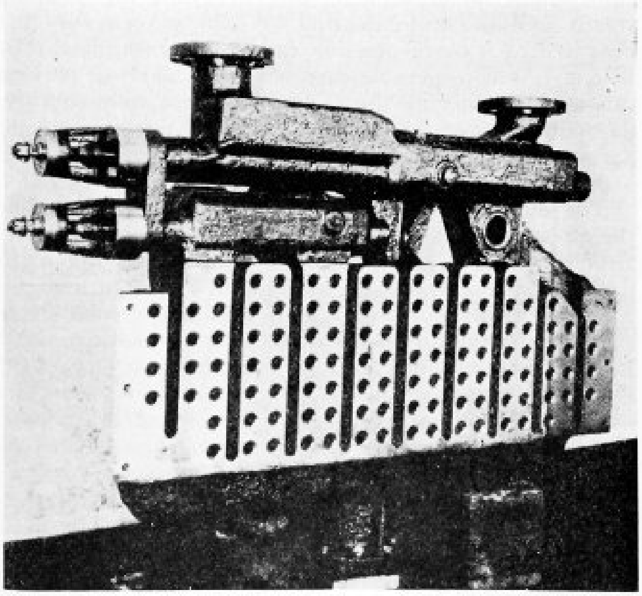
Ordinary feed water is drawn from the tender tank by a standard feed pump, and forced through the exhaust-steam feed-water heater in front of the stack, to the low-pressure boiler which it enters at a temperature of from 200 to 20 degrees F. The low-pressure boiler serves partly as an evaporator for the low-pressure steam and partly as an economizer for the high-pressure boiler.

The high-pressure feed water is drawn by the high-pressure feed pump from the low-pressure boiler at approximately 250 pounds pressure and 400 degrees F. temperature, and delivered to the high-pressure boiler drum. Besides using the less expensive and lighter heating surfaces of the low-pressure boiler for preheating the high-pressure feed water, this procedure has the advantage that most of the scaling matter remains in the low-pressure boiler where it can easily be cleaned out. Only a small amount of foreign matter is carried over into the high-pressure drum. Inasmuch as the high-pressure drum is not exposed to flame, any foreign matter which may be deposited in the drums or on the heat-transfer elements is of a soft nature and can be readily washed off through the clean-out holes which are provided for this purpose along the top of the drum.

The steam generated in the high-pressure boiler at 850 pounds pressure, flows through the outside dry pipe to the superheater header in the smokebox, then through the units of the Type E superheater and through the multiple throttle to the high-pressure cylinder located between the frames underneath the smokebox. Steam from the low-pressure boiler at 250 pounds pressure passes through a tangential dryer and a conventional dry pipe to a low-pressure superheater header in the smokebox. From there it passes through the low-pressure Type E superheater to the multiple-valve throttle, then to two mixing chambers located in the smokebox.

These mixing chambers are formed by increasing the diameter of each low-pressure steam pipe. The low-pressure live steam is mixed with the high-pressure exhaust steam by means of a perforated nozzle located in this chamber and piped to the high-pressure steam chest. The mixture flows to the two outside low-pressure cylinders. By this means the high-pressure exhaust steam is reheated in a simple manner and the difficult problem of oil separation, so vital for all types of separate reheaters, is avoided. The exhaust from the low-pressure cylinders is utilized in the usual way for drafting the engine and preheating the low-pressure feed water.

The high-pressure and low-pressure superheater headers are of the Type E through-bolt design. They are separate, but cast integral. Each header is fitted with a multiple-valve throttle, the high-pressure and the low-pressure differing only insofar as the high-



Bottom view of the combined high- and low-pressure superheater header pressure throttle has valves of considerably smaller diameter on account of the increased pressure and greater density of the high-pressure steam. The two throttle cam shafts are linked together in such a manner that each low-pressure valve opens ahead of the corresponding high-pressure valve. The cam shafts are operated by one throttle lever in the cab and the throttle operation is exactly the same as with a normal locomotive.

The boiler heating surfaces and the cylinder sizes and cut-offs are so calculated that full pressure is maintained in both the high-pressure and low-pressure boilers during operation. To prevent opening of the high-pressure safety valves when the throttle is closed, and for accelerating the steaming of the low-pressure boiler at firing up, a cross-over line is arranged which allows superheated high-pressure steam to be passed into the low-pressure boiler. Flow of steam through this line is controlled by a cone-seated valve.

There are three water gages on the boiler. The one for the closed circuit is placed outside of the cab, to be observed only at the beginning and end of each run to ascertain that no water has been lost in service. During the run the system is controlled indirecty by two thermo-couple pyrometers, which show the temperature of the steam and give an indication of the performance of the closed circuit.

Inasmuch as the low-pressure boiler tubes are only in contact with gases of relatively low temperature together with the absence of the crown sheet of the normal type of boiler, the safety of the design is apparent.

The water in the tubes forming the firebox and combustion chamber is scale free. Overheating of tubes due to scale formation, a quite common failure on ordinary high-pressure boilers, is, therefore, entirely eliminated.

An additional safety factor is the relatively low energy stored in the multi-pressure boiler, in spite of the higher steam pressures. All three units together contain only approximately 70 percent of the British thermal units found in a normal-pressure locomotive of equal capacity. The separation of the boiler into three units increases the safety still further, as all three units are not likely to be damaged at the same time. In case of a failure of a watertube in the closed circuit due to faulty material or leakage, the energy of the escaping steam will be only about 15 to 20 percent of that liberated by a corresponding failure on a standard type boiler. As considerable care has been taken to prevent steam from entering the cab in case of a tube rupture, the damage, therefore, will certainly be negligible, compared with a crown-sheet or arch-tube failure, in an ordinary locomotive.

The engine is expected to be more economical than the present type of locomotive. The savings are derived from the higher working capacity of the high-pressure steam which can be generated even at a slightly lower expense of fuel than normal-pressure steam since for 1 pound of steam at 250 pounds pressure, 1201 British thermal units are required, against only 1195 British thermal units at 850 pounds pressure. Furthermore, the boiler is designed for steam temperatures at the superheater headers of about 750 degrees F. which also contributes to the increased economy.

The table shows some of the characteristic dimensions of five locomotives of this type. Three of these are for fast passenger service. The New York Central locomotive is designed for fast freight operation and the Canadian Pacific engine is specifically intended for heavy grade work.

The Canadian Pacific locomotive No. 8000 is the largest of the five locomotives which have thus far been constructed, and in which the multi-pressure indirect steam-generating system has been employed.

Abstracts of the papers presented at the Bigwin meeting by Mr. Ennis and Mr. Bowen will appear in an early issue of THE BOILER MAKER. These abstracts will cover additional data on features of the construction of the New York Central locomotive of this type and operating details.

Oxweld Extensometer

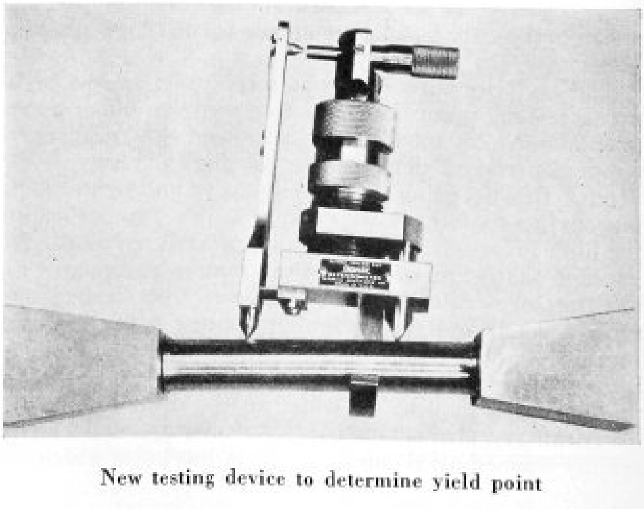
The Oxweld extensometer, a new testing device which makes it possible to determine the yield point of specimens being tested in an Oxweld portable tensile testing machine, has been announced by The Linde Air Products Company, 30 East 42nd Street, New York.

The sensitivity of the extensometer is such that it will indicate the elongation in a standard A. S. T. M. test specimen occasioned by the heat of the human hand. Although highly sensitive, this precision instrument is simple, rugged, self-contained and self-compensating.

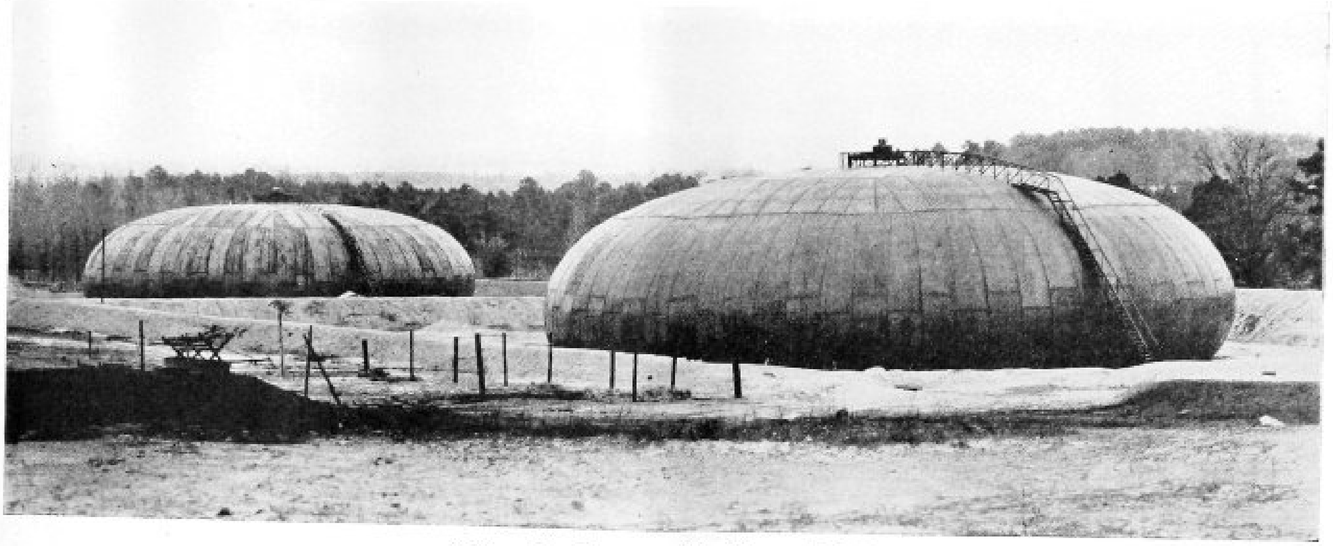
The two points of the extensometer are clamped on the specimen to be tested in such a manner that the elongation between the two points during tension is multiplied in a ratio of 5-to-1 by a lever bar making electrical contact with the micrometer head. The moment of contact is indicated by the flash of a small electric light in the extensometer head. This exclusive feature makes it possible for a single operator using the portable tensile testing machine in combination with the extensometer to determine both the ultimate strength and the yield point of a specimen.

It is recognized by the American Society for Testing Materials that an elongation of 0.01 inch in a 2-inch gage length indicates the yield point of many materials including structural steel and malleable iron. To determine the yield point with the extensometer it is only necessary to back off the micrometer head two complete turns (0.050 inch, which allows for the 5-to-1 ratio in the lever bar) from the zero reading (the point at which electrical contact is broken) for the specimen and apply the load, increasing the tension until the light flashes. The flash indicates that the yield point has been reached, and this may be read from the load indicating device on the tensile testing machine.

The extensometer may also be used as a strain gage to determine the elongation of a specimen under a specific load.



New testing device to determine yield point



Hortonspheroids erected in Texas oil field

Spheroidal Oil Storage Tanks Welded by Shielded-Arc Process

*By C. M. Taylor**

Everyone has seen drops of water on a hot stove, or drops of mercury on a flat surface. Each drop is flattened at the bottom and has curved, bulging sides. Since the surface tension in such a drop is everywhere the same, the shape of the drop must be such as to produce equal tension in all directions in all planes tangent to the surface. This is the principle upon which the Chicago Bridge and Iron Works builds patented spheroidal tanks, known as Hortonspheroids, for oil storage.

The construction of these tanks requires less material than cylindrical tanks, and, because the stresses in a spheroid are less than in a cylinder, lighter plate may be used. A further advantage of this type of construction is the decrease in evaporation losses. An ideal Hortonspheroid would be shaped exactly as the drop of water mentioned above. Actually the shape is slightly different, since the structure must withstand not only the stresses induced by full liquid and gas loadings, but also those induced by partial loadings of liquid and gas.

In the closing months of 1931, there were completed at Longview, Texas, two 80,000 barrel Hortonspheroids for the East Texas Refining Company. They are designed for 1 pound per square inch gas pressure and are 136 feet in diameter and 40 feet high. The bottoms are flat. Spheroids of such proportions are not self-supporting under partial loadings. Therefore 36 vertical trusses in radial planes are provided in the overhanging portion, and a rafter, girder, and column system under the central portion.

Even with this framing the spheroids are unable to withstand a gas pressure of 1 pound per square inch unless some liquid is in the overhanging part. For this reason provision is made for venting the gas when the liquid level falls below 10 feet from the bottom. A combination liquid pressure and vacuum vent is provided at the top, the vacuum element opening inward at approximately ½ ounce per square inch vacuum, and the pressure element opening outward at 1 pound per square inch gage pressure. This vent is covered by United States patent.

The plates for these structures were shipped directly from mill to site as were all of the straight shapes for the framing. Only the curved shapes, the valves, stairways and other accessories were shop fabricated. The spheroids are entirely welded by the shielded-arc process using "Fleetweld" electrodes and welders manufactured by The Lincoln Electric Company, Cleveland. There were no fitting-up holes or bolts. The ¼-inch bottom rests directly on the ground, but was assembled, welded and water tested on wooden supports. It was then lowered to its final position in the same manner that the bottoms of ordinary storage tanks are lowered. The bottom plates are lapped, tacked underneath, and continuously welded above. The plates shipped for the bottom were all rectangular and of the same size. Those forming the curved periphery were burned to form in place.

On the circle joining the bottom and the $\frac{3}{16}$-inch curved side sheets is a girder with a vertical web connected to the posts which form parts of the radial

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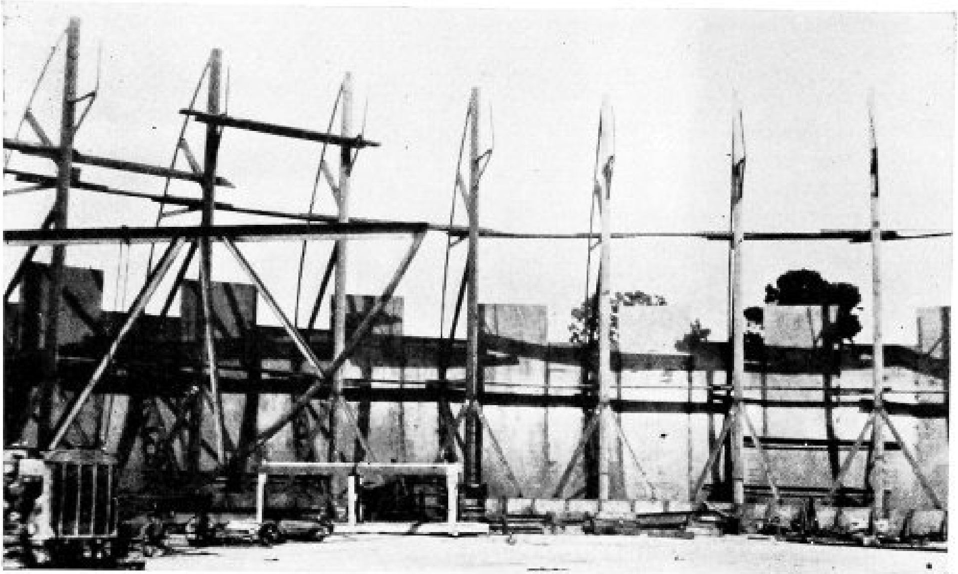
* Vice-president, The Lincoln Electric Company, Cleveland.



Interior view of outside ring of structural all-welded framing in one of the 80,000 barrel Hortonspheroids at Longview

trusses in the overhanging part of the spheroid. This girder distributes the post loads over a ring about 4 feet 6 inches wide, running entirely around the bottom. The $\frac{3}{16}$-inch side sheet courses are placed lengthwise in meridian planes. The lower plates of alternate courses were welded to trusses before the trusses were raised. Each truss was assembled and welded in a jig. The side sheets are rectangular plates lapped, and continuously welded on both sides.

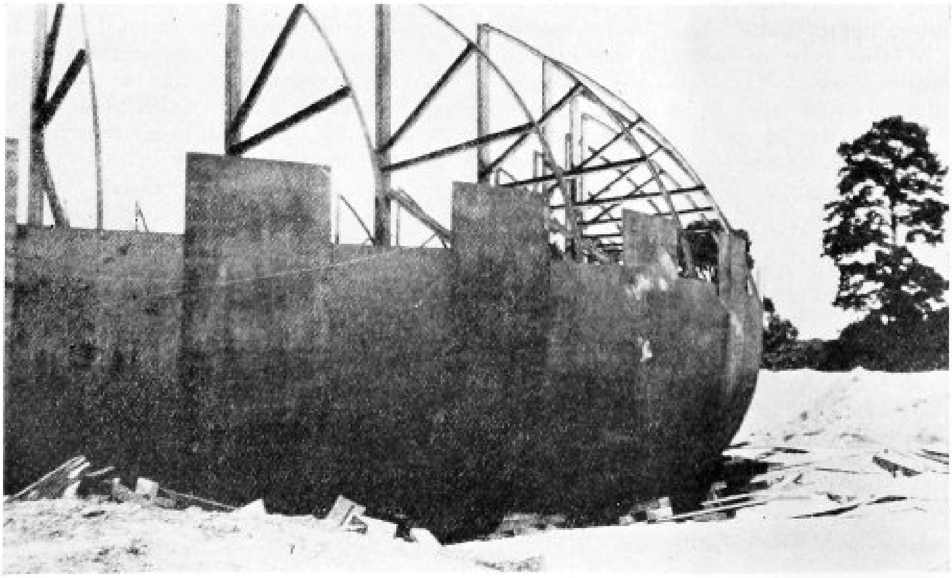
Directly above the lower girder and connected to the tops of the radial trusses is another circumferential girder into which the radial roof rafters are framed. The framing system in the central portion is similar to that for an ordinary 80,000-barrel storage tank. It consists of one center column, an inner circle of 5 columns with their tops connected by girders, an outer circle of 10 columns with their tops connected by girders, and a system of radial rafters laid on the center column and the girders, the outer ends of the outer circle of rafters being welded to the upper circumferential girder as previously stated. For emphasis it is repeated that no fitting-up holes or bolts were used either in shapes or plates.

The roof consists of $\frac{3}{16}$-inch plates with lapping edges welded continuously on the top side. The plates shipped for the roof were all rectangular and those connecting to the curved side sheets were burned to shape in the field.

The spheroids were tested by filling them with water and then emptying them. Very few leaks were discovered. They have functioned properly since they were put in service, although they should still be considered as experimental structures.

Each of the Hortonspheroids weighs approximately 500,000 pounds, about 60,000 pounds less than a standard 80,000-barrel oil-storage tank. It is interesting to note that if the volume of these tanks was increased to 1,000,000 or even 10,000,000 barrels, the height would remain constant. Only the diameter would be increased.

American Welding Society.—The fall meeting of the American Welding Society will be held at the Hotel Statler, Buffalo, N. Y., October 3-7, inclusive. The tentative program for this meeting includes technical sessions on the welding of non-ferrous metals, fundamental research in welding, and merchandising. An exposition of equipment will be held during the meeting.



Construction view of spheroidal oil storage tank, showing a close-up of the first ring of the shell plating from outside

Suggested Rules for Care and Operation of Power Boilers

At the annual meeting of the American Boiler Manufacturers' Association at White Sulphur Springs, W. Va., a committee composed of W. C. Connelly, E. R. Fish, Dr. Henry Kreitzinger and C. O. St. Clair offered the following preliminary report covering rules for the care and operation of power boilers:

The suggestions given below are for the guidance of those interested in the operation of steam power boilers, and cover only the chief details. For more complete information we refer those interested to the booklet prepared by the American Society of Mechanical Engineers, entitled "Suggested Rules for Care and Operation of Power Boilers."

1. *Preparing Boiler for Service.* If the boiler is new or has been out of service for an extended period, or has been opened for cleaning and repairs, examine the inside of the boiler and make sure that it is free from tools and foreign matter, also that there is no one in the boiler; then close the manhole openings. In the case of a new boiler, boil out with a mixture of soda ash and caustic soda. Fill the boiler with water to about the middle line of the water glass and add 1 pound of soda ash and 1 pound of caustic soda for each 1000 pounds of water held within the boiler. Dissolve the chemicals thoroughly before introducing into the water. Close the boiler and start a light fire sufficient to carry about 5 pounds pressure. Continue the boiling for two or three days, then empty the boiler and wash it thoroughly with fresh water.

See that the feed valves and the blow-off valves are in good working order and that same are properly closed when starting the boiler. Operate all dampers and make sure that they are free and in good condition. See that both inlet and outlet dampers are open.

2. *Filling the Boiler.* When a boiler has been emptied of water, do not fill it again until it has become cool. Use hot water if available. Vent the boiler in some way to permit the escape of air. This should be done before the boiler is filled. When filling the boiler, bring the water level to a point where—when the boiler is generating steam—the water level will be at its normal position when operating at full working pressure and the flow of steam has been reached.

3. *Lighting the Fire.* If the boiler is operated with powdered coal, gas or oil as fuel, ventilate the setting thoroughly before lighting the fire. Never turn on the fuel supply when starting up, nor after the snapping out of a burner without first placing in the furnace a lighted torch or a piece of burning waste to ignite the fuel instantly.

4. *Cutting-in Boiler.* In firing up a boiler that is connected with others that are already in service, keep its stop valve closed until the pressure within the boiler has become exactly equal to that at the steam main. Then open the stop valve a bare crack and slowly increase the opening until the valve is wide open. The complete operation of opening the valve should occupy at least two minutes. Close the valve at once if there is the slightest evidence of any unusual jar or disturbance about the boiler. See that the steam main to which the boiler is to be connected is thoroughly drained before this valve is opened.

5. *Water Level.* Whenever going on duty, determine immediately the level of the water in the boiler. Make sure that the glass gage and gage cocks and all the connections to the boiler are free and in working order. Do not rely upon the glass gage altogether but use the gage cock several times daily to check the water level as shown in the glass gage. If the boiler is equipped with two sets of water gages the water levels should correspond closely. If an unusual difference is noted, determine and remove the cause even if necessary to cool the boiler.

6. *Low Water.* In case of low water, immediately cover the fuel bed with ash, fresh fine coal, or earth. Close the ashpit doors and leave the fire door open. If powdered coal, gas or oil is used as fuel, shut off the supply from the burner. Do not turn on the feed water under any circumstances and do not open the safety valve nor tamper with it in any way. Let the steam outlets remain as they are. Get your boiler cool before you do anything else.

7. *Safety Valve.* Each safety valve should operate automatically and with sufficient frequency to determine its actual opening pressure. Each safety valve should have a hand lever with which it may be opened by hand cautiously every day taking care to avoid being scalded by the escaping steam. If the actual operating pressure as shown by the steam gage exceeds the pressure at which the safety valve is supposed to blow, inform your employer immediately.

8. *Blowing Down.* If the feed water is impure and the boiler is not provided with a continuous blow down, the blow-off valve should be opened at the beginning of each shift to remove the sediment which accumulates at the low point of the boiler.

If there is a blow-off valve and a blow-off cock in the blow-off line, open the cock and then open the valve slowly. After blowing down one gage of water, the valve should be closed slowly before the cock is closed to avoid water hammer action.

The surface blow-off connection, if one is provided, should be opened at intervals of one to three hours, depending upon the quality of the feed water. The amount of water discharged should not exceed 1 inch as indicated by the gage glass.

9. *Foaming.* In case of foaming, check the draft and cover the fires with fresh coal, or in case of oil or gas burning, shut off all burners immediately. Shut the stop valve long enough to find the true level of the water. If this is sufficiently high, blow down some of the water in the boiler and feed in some fresh water. Repeat this several times if necessary. If the foaming does not stop, cool off the boiler, empty it, and find out the cause of the trouble.

10. *Priming.* When wet steam affects the operation of the prime mover, determine if the water level is unusually high or if the load upon the boiler is above normal. A reduction to standard water and load condition should eliminate the priming unless the water baffles or steam baffles are in need of attention.

In case of excessive wet steam due to severe priming and foaming, it is best to consult an experienced water purification engineer about the remedy to be applied.

11. *Banking the Fire.* Clean the fire before banking. Push the live coal back against the bridge wall and cover with fresh coal. Clean the balance of the grate, close the ashpit door and fire door. Adjust the damper

but do not close it tightly. In fact the damper should be arranged so it cannot be closed completely. Before breaking the banked fire, be sure the damper is wide open and that the furnace and breeching leading to the stack have been ventilated.

12. *Cooling the Boiler.* Both the boiler and the brick settings should be cooled before draining the water out of the boiler. In the case of coal firing, first dump the fire and then close the ashpit door, open the fire door and the damper. At the same time be sure to close the stop valve on the boiler if it is one of a battery of boilers. When the pressure in the boiler has been reduced to about 10 pounds and the brickwork has become cool (which requires at least two or three hours) after the fire has been dumped or the oil or gas burners shut off, then the blow-off valve may be opened to drain the boiler, provision being made for admitting air to the boiler. The manholes and handholes may then be opened.

13. *Entering Boiler.* Before entering the drums of a boiler, see that the blow-off valve, the main stop valve, the feed-water valves, and any other valves are all closed tight. Use some reliable method of safeguarding any one entering a boiler drum. When electric extension cords are used for entering the boiler, the cord should not only be well insulated but also designed to withstand mechanical injury, and must be maintained in good condition. The lamps should be provided with suitable guards.

14. *Cleaning Boiler.* Boiler cleaners when entering the setting must guard against injury and burns from accumulation of flue dust. Poking the nozzle of a water hose beneath the surface of a pile of flue dust may cause a dangerous explosion. Do not step into flue dust until it is positively known to be cooled beneath the surface. Before any person enters the boiler drums, make sure that these parts of the boiler have been properly ventilated and that there is no inflammable gas present.

In cleaning boilers, use hose with sufficient water pressure or hand tools if necessary, to remove scale. See that the water does not come in contact with the brickwork of the combustion chamber. The condition of the feed water will determine how often a boiler should be washed out. Care must be taken that all tubes are clean. With fairly good feed water it may suffice to turbine the three rows of tubes nearest the fire at least once a month. No set rule can be given as to how often a boiler should be turbined. In operating a mechanical tube turbine cleaner, the pressure used to operate same should be as low as will suffice to do the work. Do not allow the cleaner to operate for more than a few seconds upon any one spot, and be certain that the turbine cleaner goes through the entire length of the tube.

15. *Oil in Boilers.* Cylinder oil or other high test oils act as a non-conductor of heat and if not kept out of a boiler are likely to cause overheating as well as leakage at the seams and tube ends. Where there is a possibility of oil entering the feed water through the use of condensed steam, provision in the way of suitable filters or chemical treatments should be made for its removal. Additional precautions should be taken in this removal through the location of suitable separators, traps, and drips on the exhaust steam lines. Oily deposits may be removed by scrubbing and scraping. Kerosene if applied locally in sufficient quantity will loosen or dissolve the coating. An open light must not be used in and around the boiler being treated with kerosene. Soda ash, if used in reasonable amounts will aid in the elimination of the oil.

16. *Fusible Plug.* The high temperature fusible plug, if used, should be filled with pure Banca tin and stamped as Mass. Std. or A. S. M. E. Std. by its maker. The fusible plugs should be cleaned at each boiler washing and should be replaced with a new plug annually. Do not attempt to refill the old brass casing with new fusible material.

17. *Leaks.* Establish a routine of opening each door about the setting and looking carefully for leaks. If leaks are discovered at any time they should be located and repaired; but cool the boiler first.

18. *Tube Failure.* In case of a tube failure keep the outlet damper open, check the fire, and close off any oil or gas burners and shut the boiler stop valve. Maintain the feed water supply if possible and then shut off the feed water supply when the brickwork has cooled sufficiently to prevent any injury to the boiler by overheating from the hot brickwork.

19. *Repairs Under Pressure.* No repairs of any kind should be made, either to boiler or to piping, while the part upon which the work is to be done is under pressure. This applies to calking, to tightening up bolts and to repairs to every kind of pressure part.

20. *Sediment.* To remove sediment from the boiler, open the blow-off valve preferably in the morning, or before circulation has started up. The valve should be opened wide for a few moments. They should, however, be opened and closed slowly so as to avoid shocks from water hammer action.

21. *Feed-Water Treatment.* The treatment of boiler feed water is a specialized branch of chemical engineering. Consequently, in all cases where feed-water treatment is necessary, boiler plant operators should place their problems in the hands of competent technical specialists, abiding by their advice.

22. *Gasket Surfaces.* In cleaning gaskets from finished surfaces, care shall be taken not to score or cut the surface.

23. *Pumps and Injectors.* Feed pumps and injectors shall be maintained in good condition. All feed-water tanks shall be maintained free of oil and shall be cleaned as often as necessary to prevent an objectionable accumulation of sediment or other undesirable foreign matter.

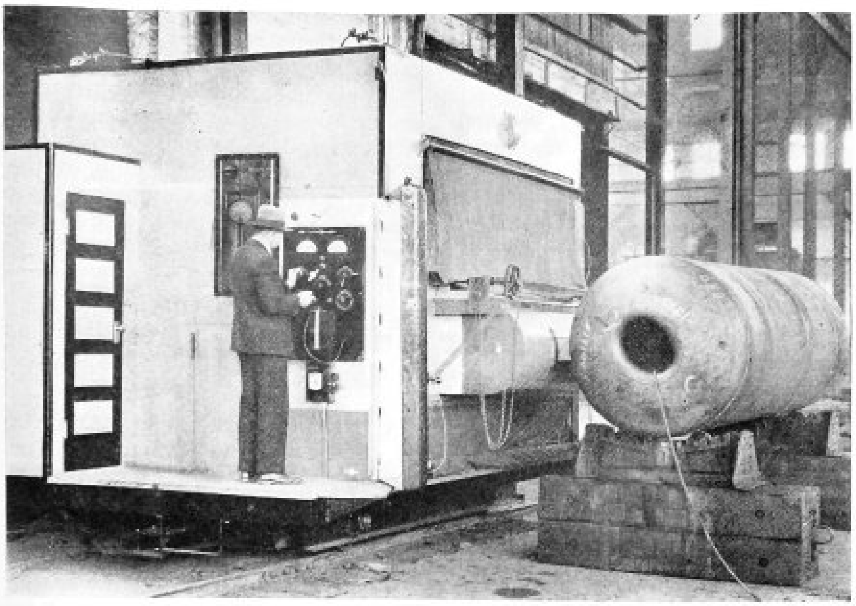
24. *Laying Up Boilers.* If a boiler is to be out of service for a month or more, it should be emptied, opened and cleaned internally and externally. After washing the interior of the boiler, drain it well and then dry it out by building a small fire of newspapers or other light material in the furnace. Do not let the boiler get too warm to be comfortable to the hand. See that no moisture can collect within or about the boiler. If the boiler is to be idle for a considerable length of time, paint all external parts with red lead and linseed oil.

If desired, the manhole and handhole plates may be placed in position without tightening them. Remove and store the steam pressure gage and its siphon pipe.

Clean the setting inside and outside. An idle boiler which remains filled with water and covered with soot deteriorates rapidly.

The Greenfield Tap and Die Corporation, Greenfield, Mass., is distributing its new small tool catalogue No. 32. The entire line of taps, dies, screw plates, twist drills, reamers, gages, pipe tools and miscellaneous items manufactured by this company are completely listed and described.

The Lukens Steel Company, Coatesville, Pa., has prepared an interesting booklet entitled "What Lukens Nickel-Clad Steel is Doing for Industry," in which is described the many applications and methods of fabricating nickel-clad steel.



Equipment set up to X-ray a pressure vessel. The power plant is located in a cab mounted on motor-driven trucks of standard gage. The cab can be picked up by crane and moved to any part of the shop. Notice the door at the operator's right which shields him from any scattered X-rays which might escape into the room. To his left is a dark room for loading film holders and storing films

Equipment for the X-Ray Inspection of Welded Pressure Vessels*

For the inspection of fusion-welded boiler drums and pressure vessels as required by the A.S.M.E. Boiler Code, the Henry Vogt Machine Co., Louisville, Ky., has installed a 345,000-volt, cascade-type X-ray machine with many distinctive features. The machine is designed to give maximum penetration, complete protection from the X-ray beam and from the high voltage, flexibility of operation, and maximum protection to its own mechanism.

The high-voltage generating apparatus is housed in a cab of channel and angle iron framework with welded sheet metal sides. The floor is of oak, which acts as a shock absorber. The cab is mounted on motor-driven trucks of standard railway gage. The top framework is braced and equipped with cables, so that the entire machine may be lifted by an overhead crane and swung to any part of the shop.

The two X-ray tubes are mounted inside a lead drum which is X-ray proof. The advantage of using two tubes with one high-voltage generating apparatus lies in the fact that they cut down the time of inspection of longitudinal seams 50 percent. The drum is adjustable vertically, laterally and horizontally.

The electrical control for the X-ray machine proper is of the shock-proof type, panel mounted. The operator, standing on the platform which is an integral part of the cab, can observe the high-voltage apparatus through a large window and also see the boiler drum or vessel which he is photographing through a lead glass window in a lead-lined door.

To the left of the operator on the control platform is a dark room equipped with safety cabinet red lamp and all conveniences for loading film holders and storing used and unexposed films. Should the operator have to journey back and forth from the stationary dark room to the machine, he would lose considerable time unless a great number of film holders were available. With the dark room so close at hand, the operator can load his film holder while the machine is in operation. He can also X-ray an entire drum before taking any of the exposed films to the dark room.

The high-voltage generating equipment constitutes a

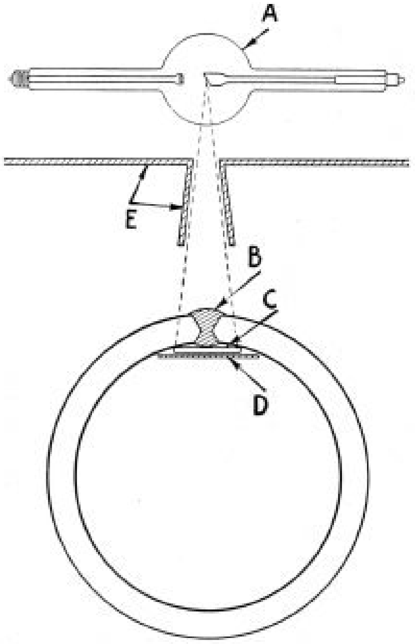


Diagram showing how the X-ray pictures are taken. The X-rays, emitted from tube A, pass through the weld, B, and produce a shadow picture on the film C, which is backed by a sheet of lead foil at D

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* Reprinted through the courtesy of *The Iron Age*.

radical departure from conventional design. It is so constructed that instead of the step-up transformer and rectifying units being built in one case and having a definite capacity limit, they are built in several sections. Each section is a separate unit capable of furnishing 115,000 volts. Three of these blocks are essentially connected in series to give a total output of 345,000 volts. This arrangement not only permits conservation of space for a given voltage, but with the cascade principle the power of the equipment may be increased at any time by adding another block. In this manner the power of the apparatus may be increased at will to a million volts.

The rectifying tubes, which are an integral part of each transformer block, are of the heavy-duty type, especially suited to industrial work. They are provided with radiator fins which dissipate the heat and permit continuous operation without being subjected to strain. They can be operated in any position. As an added precaution against physical damage, they are mounted on spring suspensions which act as shock absorbers, the flexible cable leads preventing excessive vibration.

No wood or paper insulation is used anywhere in the unit. Both inside and outside the transformer, Bakelite and electrose are the insulating media. To guard against a corona, nickel-plated tubing is used for high-voltage electrical connections. Corona shields in the form of balls or spherical caps are provided at every point where the discharge might occur.

An oil-immersed voltage stabilizer insures constancy of operation and protection for the machine and X-ray tubes. By employing the stabilizer, the operator can set the machine and leave it, knowing that the current characteristics will automatically be maintained constant.

The electrical circuit of the machine is made through an oil-immersed magnet switch which makes possible the use of remote control apparatus and relays. An automatic timing switch is included, so that the operator may turn the machine on, set the timing switch and attend to other duties during the exposure time. This permits closer duplication of exposure time and assures more uniform results through accurate control.

Differing from an ordinary circuit breaker, the overload relay carries a low current and is capable of fine adjustment. It breaks the circuit through a magnet switch, shutting the machine off if a sudden surge of current from any source places an undue strain on the X-ray tube. Likewise, it protects the operator as it "kicks out" immediately if a person accidentally comes in contact with the high-voltage equipment.

Accurate determination of voltage is desirable, since an overload on the tubes greatly lessens their life. This is taken care of by meters on the control board and by a sphere gap actuated by a small crank located at the control panel for accurate measuring of the voltage in the high potential circuit.

It is desirable to have the path of the X-rays completely inclosed from the tube to the steel drum, since a phenomenon of scattered and secondary radiation is observed when X-rays hit a heavy object. An adjustable lead-lined portal is provided for this purpose, insuring better definition.

To accommodate vessels of different diameters the entire drum can be raised or lowered by means of two synchronously-driven screws, located at either end of the drum, operated by an electric motor.

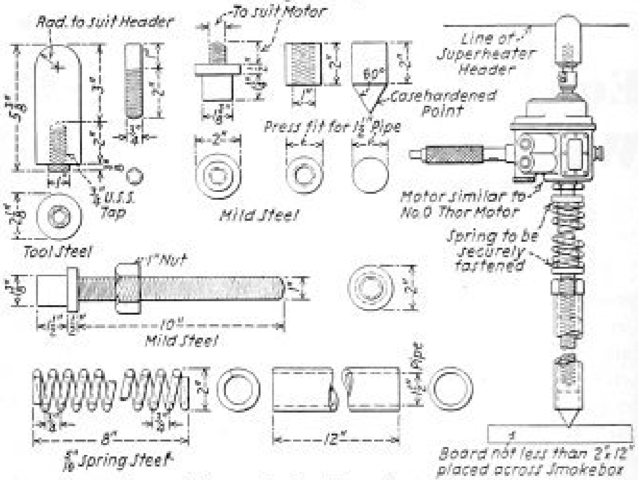
In order that the sharpest possible shadow may be cast, it is desirable to hold the film as close to the weld as possible. This is done in either of two ways. A magnetic type of holder, permitting a universal adjustment of the X-ray plate, or a mechanical type is used, depending upon the immediate problem at hand.

Films are dried in a special film dryer equipped with a heater which lessens the time required for finishing the film. The dryer also acts as a ventilator, since its exhaust is discharged outside through a light-proof baffle. The room also is equipped with dark room lamps with different types of filters and X-ray film illuminators.

The entire X-ray unit was designed and built by the Kelley-Koett Manufacturing Company, Covington, Ky., to conform to the particular requirements of the Henry Vogt Machine Company.

Device for Grinding Superheater Headers

A simple device for grinding the header seats for superheater pipes is shown in the drawing. The case-hardened point and sleeve, which is tapped for 1-in. standard threads, are pressed into the ends of a piece of 1½-in. extra-heavy pipe, and the adjusting screw is screwed into the sleeve. The motor adjusting screw is replaced by the spring seat and the spring is held in place between the collars on the spring seat and the head of the adjusting screw. The grinding head is threaded to receive a stud by means of which the head is held in a standard drill chuck.



Assembly and details of the device for grinding superheater headers

A 2-in. by 12-in. plank secured to the smoke-box front-end studs serves as a backing for the device. Adjustment to suit the length of smoke-box is made by means of the screw and nut. The flexibility of the spring permits of a slight horizontal and vertical movement of the motor while grinding, thus insuring a good joint.

Work of the A.S.M.E. Boiler Code Committee

The Boiler Code Committee meets monthly for the purpose of considering communications relative to the Boiler Code. Any one desiring information as to the application of the Code is requested to communicate with the Secretary of the Committee, 29 West 39th St., New York, N. Y.

The procedure of the Committee in handling the cases is as follows: All inquiries must be in written form before they are accepted for consideration. Copies are sent by the Secretary of the Committee to all of the members of the Committee. The interpretation, in the form of a reply, is then prepared by the Committee and passed upon at a regular meeting of the Committee. This interpretation is later submitted to the Council of The American Society of Mechanical Engineers for approval, after which it is issued to the inquirer and published.

Below are given records of the interpretations of the Committee in Cases Nos. 725 and 727, as formulated at the meeting of May 20, 1923, and Cases Nos. 724, 726, 728, 729, and 730, as formulated at the meeting of June 23, 1932, all having been approved by the Council. In accordance with established practice, names of inquirers have been omitted.

CASE No. 724.—*Inquiry:* Is it acceptable, under the rules of the Heating Boiler Section of the Code, for boilers to be operated at not to exceed 15 pounds steam or 30 pounds water, having a volume as determined from the overall dimensions not to exceed 50 cubic feet to attach stays, staybolts, and staybolt parts, to plates requiring support, by fillet welds instead of inserting through chamfered holes and welding as required by Par. H-83 provided said stays are fitted snugly to the plate for the full cross section of the stays before welding and provided the fillet welds are made to conform with Class 2 test requirements of Section VIII of the Code (Par. U-69)? Destructive tests of the construction proposed as shown in Fig. 28, the results of which have been submitted, have shown these forms of attachment to be safe.

Reply: Based on the test results submitted, it is the opinion of the Committee that if the stays described in the inquiry and as shown in Fig. 28 are welded in the way proposed the construction shown may be considered the equivalent of the form of staybolt attachment specified in Par. H-83 for use in boilers under the conditions specified.

CASE No. 726.—*Inquiry:* An interpretation is requested of the requirement in Par. P-111 of the Code limiting the circularity of a cylindrical shell to 1 percent. Does this limit apply to the total circumference or does it signify 1 percent plus or minus the specified diameter?

Reply: It is the opinion of the Committee that the maximum variation in circularity of a cylindrical shell specified in Par. P-111 shall be based on the difference between the maximum and minimum diameters at any section which shall not exceed 1 percent of the nominal diameter.

CASE No. 727.—*Inquiry:* In the application of a fusion-welded nozzle which requires stress relieving, to a Class 2 vessel which is not required to be stress relieved, is it permissible to locally heat the nozzle and an annular ring of the shell around the nozzle?

Reply: It is the opinion of the Committee that where nozzles are required to be locally stress relieved, Pars. P-108 and U-76 of the Code require the simultaneous stress relieving of the nozzle and a band of the width specified in Figs. P-6 and U-16, extending entirely around the circumference of the vessel.

CASE No. 728.—*Inquiry:* Is it not desirable that the edges of the hole in the die block shown in Fig. 1 of Specifications S-17 for making flange tests of boiler tubes be rounded off, preferably to a radius of 1/32 inch, in order to avoid a sharp corner? Attention is called to the fact that Pars. P-249 and L-51 of the Code call for the removal of the sharp edges of the tube holes where tubes are fitted into the tube sheets of boilers.

Reply: It is the opinion of the Committee that the provision of a sharp corner in the die block for making flange tests will cause unnecessarily severe treatment of the tube and the Committee therefore believes that it will be desirable to revise this illustration to show a rounding of the corner to a radius of 1/32 inch.

CASE No. 730.—*Inquiry:* Is the limitation in the fourth line of Par. L-42 of the present edition of the Code for Boilers of Locomotives correct where it states "P shall not exceed $2T$"? Attention is called to the fact that the term P as used in the formula refers to the maximum allowable working pressure and this would obviously permit only an extremely low working pressure.

Reply: The use of the term P in the fourth line of Par. L-42 (present edition of the Code for Boilers of Locomotives) is a typographical error and the term p should be used referring to the pitch in inches.

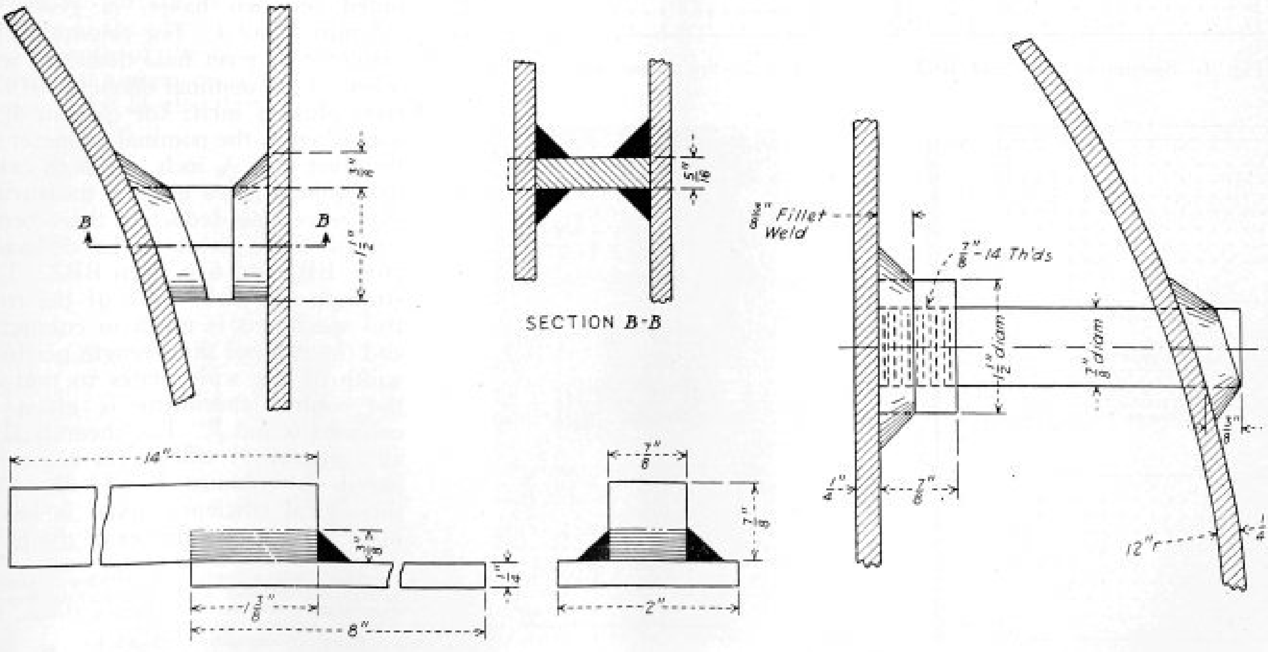


Fig. 28.—Details of staybolt welding for testing boilers

Determining the Strength of Net Section of Plates* ▲ ▲ ▲

The Engineering Experiment Station of the University of Illinois, Urbana, Ill., recently completed tests on joints in wide plates. This article, the third in a series, describes the test specimens and the results of tests for determining the efficiency of net section of the 1930 series of plates. The first article in the series was published in July under the title "Efficiency of Wide Plates"; and the second, entitled "Determining the Strength of Net Section of Plates," appeared in August. These experiments, conducted by Professor Wilbur M. Wilson, James Mather and Charles O. Harris, were made possible through the co-operation of the Chicago Bridge & Iron Works, Chicago, Ill.

The 1930 series includes two specimens each of two types of joints and four specimens of one type. There were: (1) two triple-riveted double-strap butt joints, the specimens being designated as BB1 and BB2; (2) four quadruple-riveted double-strap butt joints, the specimens being designated as AB1, AB2, AB3, and AB4; and (3) two quadruple-riveted lap joints, the specimens being designated as AL1 and AL2. All joints were designed so that, as nearly as practicable, the strength of the net section of the plate would equal the shearing strength of the rivets. The size of butt straps and the rivet spacing for these specimens are given in Figs. 1, 2, and 3. Fig. 4, a reproduction of a photograph of AB2 after failure, shows the reinforcing plates at the ends of the specimen and also the holes for bolting the specimen to the clevis of the pulling head, which are similar for all specimens of these three types.

The triple-riveted butt joints and two of the four quadruple-riveted butt joints failed by tearing the plates. The quadruple-riveted lap joints and two of the four quadruple-riveted butt joints failed by shearing the rivets, but the plates in these joints were near failure when the rivets sheared. Tests of the three types of joints are discussed in the following paragraphs.

The specimens BB1 and BB2 are triple-riveted double-strap butt joints in plates 72 inches by $\frac{7}{16}$ inch, the rivets having a nominal diameter of ¾ inch. The detail dimensions of the specimen are given in Fig. 1. The physical properties, determined from the control specimens and reported in Table 1, indicate that the material was both ductile and strong. The ultimate strengths of the wide specimens, given in Table 2, were 1,650,000 and 1,695,000 pounds for BB1 and BB2, respectively. The strength per inch width of net section, computed on two bases, is given in columns 3 and 4. For column 3, the diameter of rivet hole deducted was taken as the nominal diameter of the rivet plus ⅛ inch; for column 4, it was taken as the nominal diameter of the rivet plus $\frac{1}{16}$ inch. In both cases the ¼-inch holes used in measuring slip, were also deducted, there being ten of these holes to be deducted from BB1 and five from BB2. The strength per inch width of the control specimens is given in column 5 and the ratio of the strength per inch width of the wide plates to that of the control specimens is given in columns 6 and 7. The theoretical** and measured† efficiencies are compared in columns 8 and 9. The theoretical efficiency given is based upon the actual diameter of the hole,

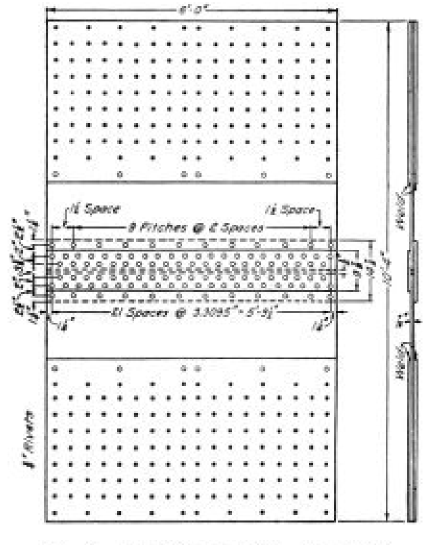


Fig. 1.—Specimens BB1 and BB2

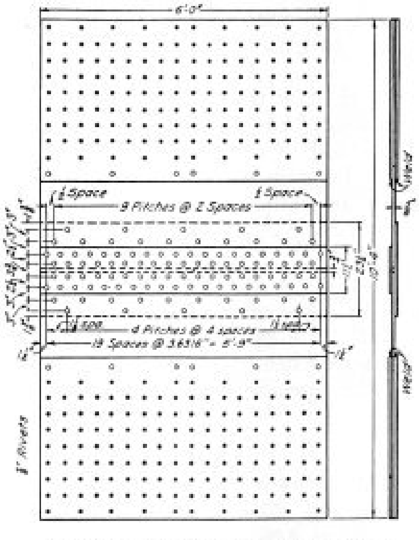


Fig. 2.—Specimens AB1 and AB2

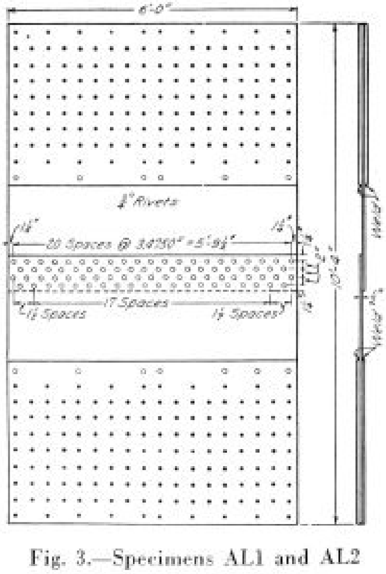


Fig. 3.—Specimens AL1 and AL2

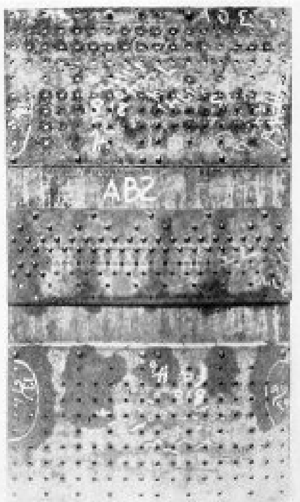


Fig. 4.—Specimen AB2 after failure

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* Third of a series of articles published through the courtesy of the Engineering Experiment Station, University of Illinois, Urbana, Ill.

** At weakest section.

† Ratio of strength developed by wide plate to product of gross width of plate and strength per inch width developed by control specimens.

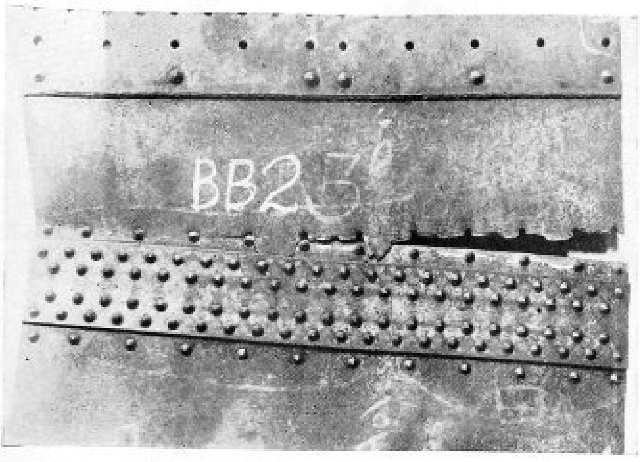


Fig. 5.—Specimen BB2 after failure

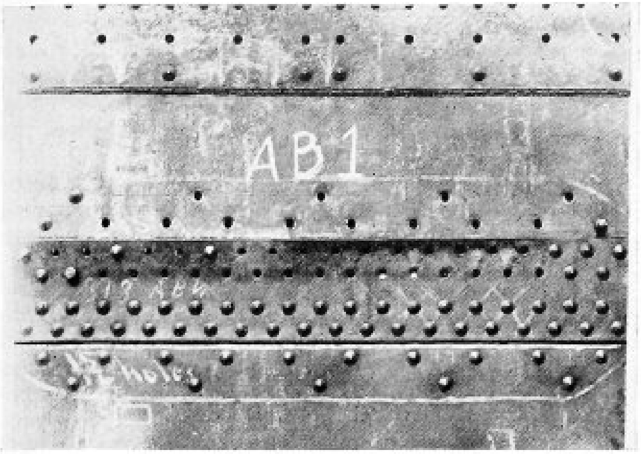


Fig. 6.—Specimen AB1 after failure

nominal diameter of the rivet plus $\frac{1}{16}$ inch. These two specimens developed almost their theoretical efficiency. Plate BB1 tore on a transverse section through the outer row of rivets, but the tear in BB2 extended into the second row of rivets at two points, as shown in Fig. 5.

The specimens AB1, AB2, AB3, and AB4 are quadruple-riveted double-strap butt joints in plates 72 inches by ⅝ inch, the rivets having a nominal diameter of ⅞ inch. The detailed dimensions of the specimens are given in Fig. 2. The physical properties, determined from the control specimens and reported in Table 1, indicate that the material was both ductile and strong. The ultimate strength was 2,310,000, 2,305,000, 2,800,000 and 2,270,000 pounds for AB1, AB2, AB3, and AB4, respectively. The theoretical and measured efficiencies are compared in Table 2. The strength developed by AB1 and AB2 was less than 90 percent of the theoretical strength of the net section of the plates. The fact should be noted, however, that failure was due to shearing of the rivets and not to tearing of the plate. An inspection of Fig. 6, indicates that the plate had elongated greatly at a section through the outer row of rivets and it is likely that it would not have carried much more load than that to which it was subjected when the rivets failed. The strength developed by AB3 and AB4 was about 95 percent of the computed strength of the net section of the plates through the outer row of rivets.

For both of the latter specimens failure was due to tearing the plate on a section that passed through the outer rivet in the third row and zigzagged back and

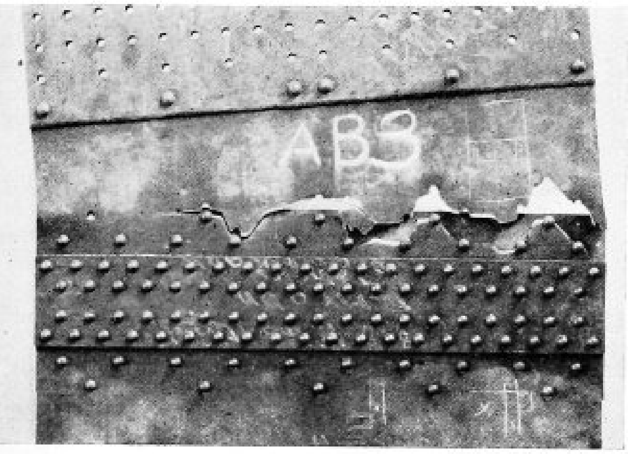


Fig. 7.—Specimen AB3 after failure

Table 1.—Physical Properties of Steel as Given by Tests of Control Specimens, 1930 Series

Specimen No.	Reduction of Area percent	Elongation in 8 inches percent	Yield Point pounds per sq. in.	Ultimate Strength	
				Pounds per in. width	Pounds per sq. in.
AB1	54.3	28.17	39,950	39,650	63,475
AB2	54.2	27.78	39,940	40,730	64,880
AB3	61.3	30.5	34,700	36,600	57,200
AB4	63.1	30.9	34,100	36,200	57,300
BB1	59.8	26.35	31,610	28,354	64,480
BB2	59.5	27.78	32,290	28.090	64,580
AL1	61.5	28.54	30,140	29,520	58,850
AL2	63.8	28.41	28,760	28,350	57,740
AW1	67.8	32.79	27,760	33,266	53,290
AW2	66.8	32.72	27,730	33,030	53,580

Table 2.—Efficiency of Net Section, 1930 Series

Specimen No.	Ultimate Strength, Pounds	Strength per in. width of Net Section. Hole Deduction Equal to		Strength per in. width of Control Specimen, Pounds	Ratio		Efficiency	
		Nominal Diameter of Rivet Plus ⅛ inch, Pounds	Nominal Diameter of Rivet Plus $\frac{1}{16}$ inch, Pounds		(3)/(5)	(4)/(5)	Theoretical*	Measured
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)
AB1	2,310,000	35,550 [1]	35,350 [1]	39,650	0.8966	0.8916	90.9	80.8†
AB2	2,305,000	35,050 [2]	34,800 [2]	40,730	0.8605	0.8540	91.9	78.5†
AB2	2,280,000	34,700	34,500 [2]	36,600	0.9481	0.9426	91.9	86.6
AB4	2,270,000	34,500	34,300 [2]	36,200	0.9530	0.9475	91.9	87.1
BB1	1,650,000	27,950 [3]	27,600 [3]	28,354	0.9858	0.9734	83.0	80.8
BB2	1,695,000	28,150 [4]	27,800 [4]	28,090	1.0021	0.9897	84.7	83.8
AL1	1,610,000	30,700 [4]	30,000 [4]	29,520	1.0400	1.0163	74.6	76.0
AL2	1,642,000	31,400 [4]	30,600 [4]	28,350	1.1076	1.0794	74.6	80.5
AW1	2,050,000	.....	.....	33,266	.....	.....	100.0	96.4
AW2	2,100,000	.....	.....	33,030	.....	.....	100.0	99.4

---

[1] Eight ¼-inch holes for measuring slip were also deducted (see Fig. 9a).

[2] Five ¼-inch holes for measuring slip were also deducted (see Fig. 11).

[3] Ten ¼-inch holes for measuring slip were also deducted (see Fig. 7b).

[4] Five ¼-inch holes for measuring slip were also deducted (see Figs. 8, 12a, and 13a).

* Based on actual diameter of hole, nominal diameter of rivet plus $\frac{1}{16}$ inch.

† These specimens failed by shearing the rivets and the plates might have developed an efficiency higher than those given, but Fig. 9a shows that the plate was about to fail and probably would not have carried a load much greater than the one to which it was subjected when the rivets failed.

forth between the rivets of the outer two rows as shown in Fig 7. A part of the 5 percent difference between the computed strength on a transverse section through the outer row of rivets and the strength actually developed by the specimens is due to the fact that the zigzag section of failure is weaker, theoretically, than the transverse section.

Specimens AL1 and AL2 were quadruple-riveted lap joints in plates 72 inches by ½ inch, the rivets having a nominal diameter of ¾ inch. The detailed dimensions of the specimens are given in Fig 3. The physical properties, determined from the control specimens and reported in Table 1, indicate that the material was both ductile and strong. The ultimate strengths were 1,610,000 and 1,642,000 pounds for AL1 and AL2, respectively. The theoretical and measured efficiencies are compared in Table 2. The strength developed by these specimens is somewhat greater than the theoretical strength of the net section of the plates and the failure was due to the failure of the rivets. The plates had elongated greatly at a section through the outer row of rivets.

There were many features about the tests of riveted joints in wide plates that are of particular interest. One of these is the uniformity with which the stress normal to the joint is distributed across the plate. Strain readings were taken with an 8-inch strain gage on five gage lines on each side of each of the two plates constituting a joint, for all specimens.

(To be continued)

Outside Blow-Off for Locomotives

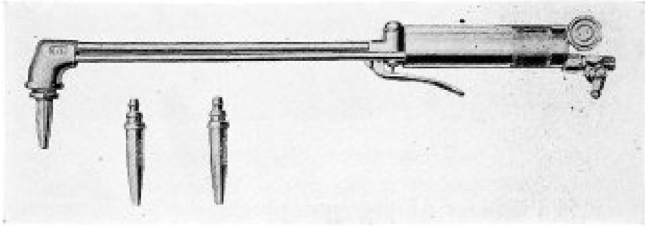
The outside blow-off piping and sump arrangement shown in the two drawings can be adapted to practically any engine house. Three 2-inch Barco joints are provided in each of the two boiler connections which are located on both sides of the track. When not in service, the flexible boiler connections are laid on a frame, 15 inches high, the posts of which are made of 4-inch by 4-inch yellow pine and the cross-piece of 2-inch by 4-inch yellow pine material. These frames are set firmly in the floor, 9 feet 6 inches from the center of the track.

The sump can be located in any suitable spot outside the building and discharged into the sewer, or on the ground. All of the pipe used in the blow-off installation should be extra-heavy wrought iron. All fittings 2½ inches and over should be of extra heavy semi-steel and flanged. All screw fittings 2 inches and under should be extra heavy cast iron.

Cutting and Welding Torch

Tips, Inc., 515 Cathedral Street, Baltimore, Md., international distributor of standardized cutting and welding tips, apparatus and accessories, announces a new cutting and welding torch.

The type NVM cutting and welding torch is highly improved in design and will cut or weld any thickness of metal within range of the process. Welding is accomplished by a mere change of tips.



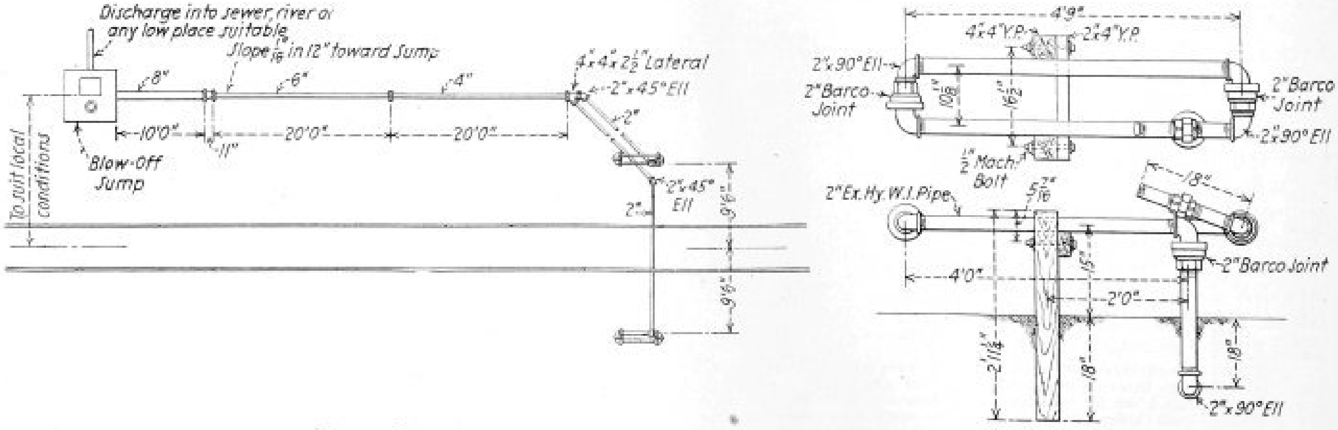
NVM cutting torch with welding tips

The torch is strongly constructed from the best materials obtainable. The valves and base are of high-grade forged bronze and the tubes are special weight, triangularly arranged to give greatest transverse strength. The high-pressure valve is operated with an improved lever underneath the torch, which has been designed for simplicity of operation. All replacements can be made without disassembling the torch, as the important parts are accessible from the outside.

The 90-degree head of the Type NVM torch is designed to take standard Type NV conical seated tips which have the same standard as those used in Airco Davis-Bournonville cutting torches.

Champion Welding Rods

After 40 years of making rivets, The Champion Rivet Company, Cleveland, O., is announcing a new line of general purpose welding rods. While the Champion Rivet Company will still continue to market the rivets for which it has become famous over the past 40 years, the sales organization will have with it a line of welding rods to offer its customers. Several years of experimenting are behind this line of welding rods and the company is now satisfied that the new rod will gain as dominant a position in this field as the rivets have held.



Pipe and sump arrangement for an outside blow-off for locomotives

The Boiler Maker

VOLUME XXXII NUMBER 9

PUBLISHED Monthly by the Simmons-Boardman Publishing Company, 34 North Crystal Street, East Stroudsburg, Pa., with Executive Offices at 30 Church Street, New York City.

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Subscription price in United States and possessions, 1 year, $2.00, 2 years, $3.00; Canada, including duty, 1 year, $2.50, 2 years, $4.00; foreign countries, 1 year, $3.00, 2 years, $5.00. Single copies, 35 cents each.

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THE BOILER MAKER is a member of the Associated Business Papers, Inc. (A. B. P.), and the Audit Bureau of Circulations, (A. B. C.)

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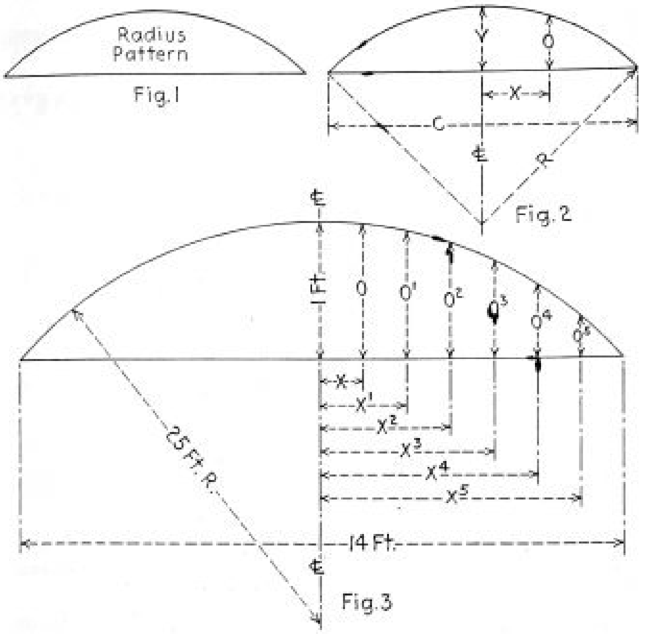
Contents

Communication

Computing the Radius

TO THE EDITOR:

The drawing of a radius of large dimension in the shop sometimes places the layerout in an awkward position. For example, the location of the center point for the trammels to draw a large radius may be in some cases a certain number of feet away from the plate, as in the case of drawing a 15-foot radius on a plate 24 by 24 inches. This means that the layerout must erect some sort of a platform the required distance away and on the same level with the plate he is laying out, so that he can locate the center point for his trammels. This is a loss of time because every time the radius is larger than the plate the foregoing condition will prevail.



Drawing a large dimension radius

We know from experience that the rigidity of a wooden beam used with trammel points for large work is hard to find. The flexibility of these wooden beams as they increase in length is something the layerout must not overlook, because the least pressure on these wooden beams in drawing a large radius will cause them to bend and affect the accuracy of the radius.

The layerout does not always have on hand these wooden beams for large work. Also, due to the different temperatures in the shop, after a certain length of time in use, these wooden beams will distort and will have to be replaced with new ones. So from the above examples it seems that it would be a good practice for the layerout to know the formula to compute the radius and to have on hand an assortment of these circular patterns of different sizes of radius made from sheet iron, as shown in Fig. 1.

The versine or height of these patterns will, of course, depend upon the length of the chord or *vice versa*. To compute the radius, chord, and versine, as shown in Fig. 2:

Where O = the unknown height.
R = the radius.
V = versine or height.
C = the chord.
X = any distance from the center line on the chord.

Then the formulae below are used:

Hence, $O = \sqrt{R^2 - X^2} - (R - V)$.

$$R = \frac{c^2 + 4 \times v^2}{8 \times v}$$

$$V = R - \sqrt{R^2 - \frac{c^2}{4}}$$

$$C = 2 \times \sqrt{v \times (2 \times R - v)}$$

Examples:

If R = 25 feet and v = 1 foot, how long is C?

$C = 2 \times \sqrt{1 \times (2 \times 25 - 1)} = 2 \times \sqrt{49} = 2 \times 7 = 14$ feet.

(Continued on page 200)

Questions and Answers Pertaining to Boilers

This department is maintained for the purpose of helping those who desire assistance on practical boiler shop problems. Inquiries should bear the name and address of the writer. Anonymous communications will not be considered. The identity of the writer, however, will not be disclosed unless the editor is given special permission to do so.

By George M. Davies

Layout of Tapered Pipe Connection

Q.—Please show me how to lay out the tapered pipe connection shown in the accompanying illustration.—P. J. T.

A.—Figs. 1, 2 and 3 illustrate the elevation, side and plan views of the tapered pipe connection submitted with the question.

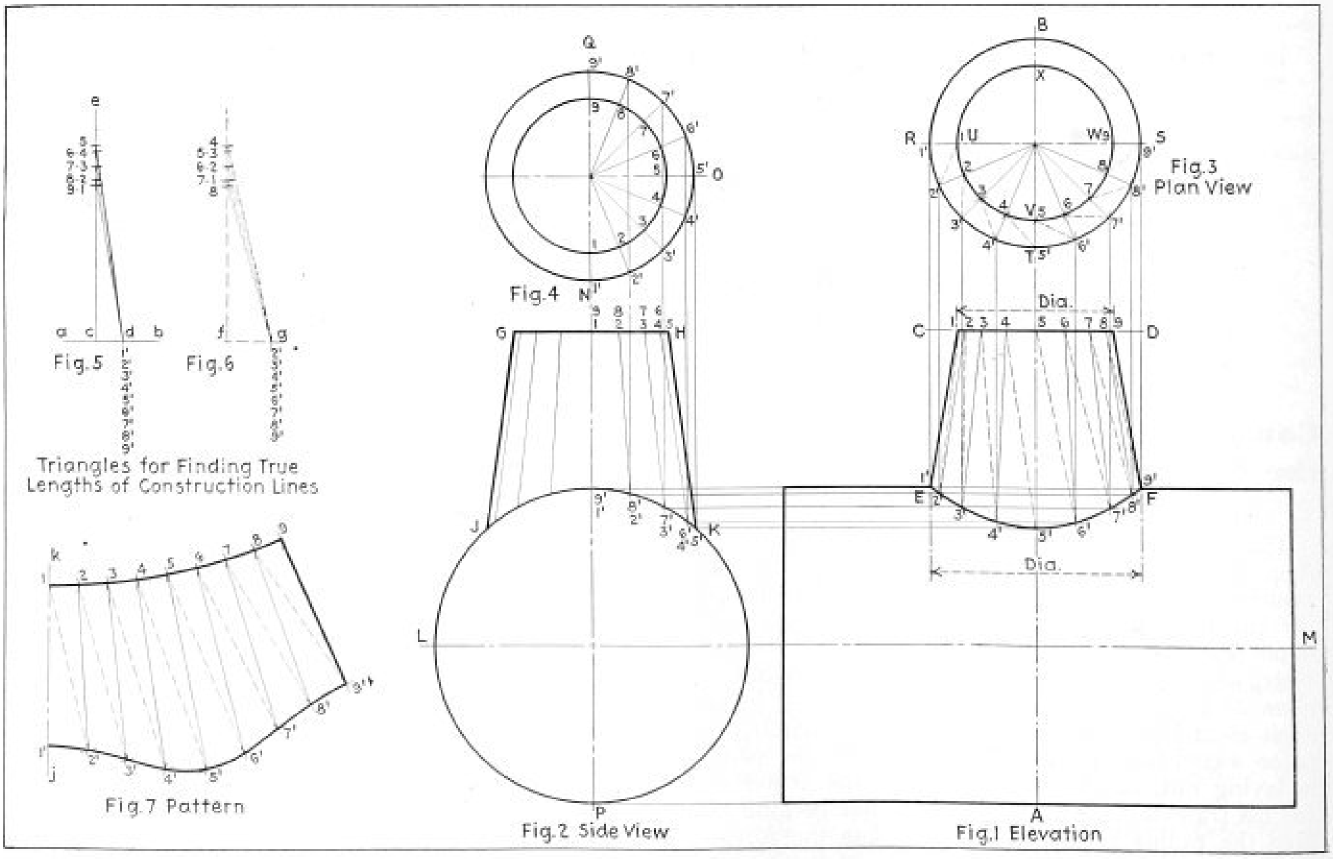
In the elevation, the tapered pipe connection is shown by *C-D-E-F*. In the plan view, the top of the pipe connection is shown by *U-V-W-X* and the bottom of the pipe connection by *R-T-S-B*. An inspection of the plan will show that the center line *R-S* divides the object into two symmetrical halves, consequently a pattern for one half will serve for the other half.

In laying out the elevation, Fig. 1, the line *E-F* which is the intersection of the tapered pipe connection and the cylinder must be determined from the plan and side views as follows.

Divide the semicircle *R-T-S* in the plan view into any number of equal parts, the greater the number of parts taken the more accurate the development. In this case eight divisions are taken; number the divisions *1'*, *2'*, *3'*, *4'*, to *9'* as indicated, then, parallel to the center line *A-B*, draw lines through the points *1'*, *2'*, *3'* to *9'*, extending these lines down into the elevation.

Then draw Fig. 4, which is a plan view drawn directly over the end view. Divide the semicircle *N-O-Q* into the same number of equal parts as previously taken, numbering these divisions *1'*, *2'*, *3'*, *4'*, to *9'*, the points being numbered to correspond to the same points in the plan view Fig. 3.

Through these points and parallel to the line *P-Q*, draw lines extending down into the side view Fig. 2, cutting the cylinder and locating the points *1'*, *2'*, *3'*, *4'* to *9'*, Fig. 2. Then parallel to the center line *L-M*, draw lines through the points *1'* to *9'*, Fig. 2, extending these into the elevation and cutting the corresponding lines extending down from the plan view, Fig. 3, locating the points *1'*, *2'*, *3'* to *9'* in the elevation. A line drawn through these points will give the line *E-F*, the intersection of the tapered pipe and the cylinder.



Elevation, side and plan views of tapered pipe connection

To make a development of the tapered connection, divide the semicircle *U-V-W* of the plan view, Fig. 2, into the same number of equal parts as the semi-circle *R-T-S* was divided. Connect the points *1-1′*, *2-2′*, *3-3′*, *4-4′*, *5-5′*, *6-6′*, *7-7′*, *8-8′* and *9-9′* with solid lines and the points *1-2′*, *2-3′*, *3-4′*, *4-5′*, *5-6′*, *6-7′*, *7-8′*, and *8-9′* with dotted lines as illustrated. Then parallel to the center line *A-B* draw lines through the points *1*, *2*, *3*, *4*, *5*, to *9*, Fig. 3, extending these into the elevation, Fig. 1. Where these lines cut the line *C-D*, number the points *1*, *2*, *3*, to *9* as shown. Connect the points *1*, *2*, *3*, to *9* and *1′*, *2′*, *3′*, to *9′* in the elevation with solid and dotted lines in the same manner as described for the plan view, Fig. 3.

The next step is to determine the true lengths of these construction lines, as illustrated in the diagram of triangles Figs. 5 and 6.

Fig. 5 shows a series of triangles for determining the true length of the solid surface lines of the elevation, these triangles are determined as follows: Draw the line *a-b* and at *c* lay off the distance *c-d* equal to *R-U* of the plan view, Fig. 3. From an inspection of the plan view it will be noted that the bases of the triangles will all be the same, as the semi-circles *U-V-W* and *R-T-S* are concentric. At *c*, and perpendicular to *a-b*, erect the line *c-e*. From *c*, and on the line *c-e*, step off the distance *c-1* equal to the vertical distance between the point *1′* and the line *C-D* in the elevation, Fig. 1. Connect *1* and *d*, forming the hypotenuse, which is the true length of the surface line *1-1′* in the elevation. In like manner, from *c* and on the line *c-e* step off the distance *c-2* equal to the vertical distance between the point *2′* and the line *C-D* in the elevation, Fig. 1. Connect *2* and *d*, forming the hypotenuse, which is the true length of the surface line *2-2′* in the elevation. Proceed in like manner using the vertical distances between the points *3′*, *4′*, *5′* to *9′* and the line *C-D* in the elevation, Fig. 1, as the altitudes of the triangles obtaining the true lengths of the solid surface lines.

The true lengths of the dotted surface lines are obtained in exactly the same manner and are illustrated in Fig. 6. The line *f-g* is taken equal to *U-2′*, Fig. 3 and is the same for all triangles. The altitudes for the triangles are obtained as explained in Fig. 5, the hypotenuses being the true lengths of the dotted surface lines.

The pattern is developed as follows: On the line *j-k* Fig. 7, step off the distance *1-1′* equal to the line *C-E* of the elevation, Fig. 1. Then with the dividers set equal to the distance *1′-2′*, Fig. 3, which is one of the equal parts into which the semicircle *R-T-S* was divided, and with the point *1′* as a center, scribe an arc. Then with the trams set equal to the distance *1-2′*, Fig. 6, scribe an arc cutting the arc just drawn locating *2′*.

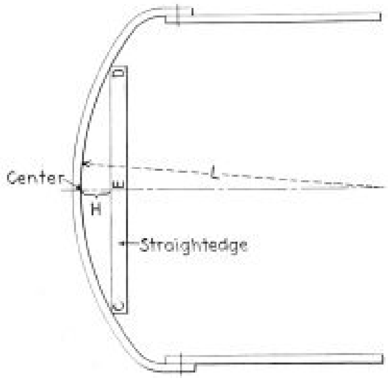
Then with the point *1*, as a center and with the dividers set equal to the distance *1-2*, Fig. 3, which is one of the equal parts into which the semicircle *U-V-W* was divided, scribe an arc. Then with point *2′* as a center and the trams set equal to *2-2′* Fig. 5 scribe an arc cutting the arc just drawn locating the point *2*.

Proceed in the same manner making the distances *2-3*, *3-4*, etc., equal to the equal spaces *2-3*, *3-4*, etc., Fig. 3, the distances *2′-3′*, *3′-4′*, etc., equal to the equal spaces *2′-3′*, *3′-4′*, etc., Fig. 3, and making the length of the solid and dotted lines equal to their corresponding lines in Figs. 5 and 6, locating all the points of the pattern Fig. 7. Connect these points and the pattern for one-half of the development of the tapered connection is complete. A duplicate of this half will complete the development for the entire piece.

Radius of Boiler Head

Q.—Will you tell me how to determine the radius of a boiler head as shown in Fig. 1?—H. H.

A.—The radius of a blank unstayed dished head with the pressure on the concave side, when it is a segment of a sphere, as illustrated in Fig. 1, shall be calculated by the following formula:



Determining radius of dished head

$$L = \frac{2 \times TS \times t}{8.33 \times P}$$

where

t = thickness of plate, inches.
P = maximum allowable working pressure, pounds per square inch.
TS = tensile strength, pounds per square inch originally stamped on the plate used in forming the head.
L = radius to which head is dished measured on the concave side of the head, inches.

Where two radii are used, the longer shall be taken as the value of *L* in the formula.

The radius to which the head is dished shall not be greater than the diameter of the shell to which the head is attached.

To find the radius of an actual boiler head as illustrated in the question, first locate the center of the head by scribing from the perimeter of the head. Then place a straight-edge of given length against the head as shown in Fig. 1, *C-D* being equivalent to the length of the straight-edge. Divide the length of the straight-edge into two equal parts as *C-E* and *E-D*.

Set the straight-edge around the head so that the point *E* on the straight-edge is directly over the center of the head; measure the distance *H* as shown in Fig. 1.

The distance *E-D* and *H*, being known, calculate the radius from the following formula:

$$\text{Radius} = \frac{ED^2 + H^2}{2\,H}$$

Ratio of Rated Horsepower to Furnace Volume

Q.—Will you kindly publish in the Questions and Answers Department of THE BOILER MAKER the proper ratio of rated horsepower to the cubic capacity of furnace volume of a portable locomotive-type boiler where natural gas is the fuel to be used?—T. C. E.

A.—The proper ratio of rated horsepower to the cubic capacity of furnace volume is dependent upon the kind of gas used.

Blast Furnace Gas.—In burning blast furnace gas the essential feature is ample combustion space, in which the combustion of gases may be practically completed before striking the heating surfaces. The gases have the power of burning out completely after striking the heating surfaces, provided the initial temperature is sufficiently high; but when the combustion is completed before such time, the results are more satisfactory. A furnace volume of approximately 1 to 1.5 cubic feet per rated boiler horsepower will give a combustion space that is ample.

Natural Gas and By-Product Coke Oven Gas.—For these gases a large combustion space is essential and a volume of 0.75 cubic feet per rated horsepower will be found to give good results.

Repairs to Welded Seam

Q.—If a bad spot shows up in a welded seam of a boiler under X-ray, how do they cut it out to reweld it?— W. M. G.

A.—Pin holes, cracks or other defects shall be repaired only by chipping, machining or burning out the defect and rewelding. For gas welding, the metal around the defects shall be preheated to a dull red for a distance of at least 4 inches all around. Any preheating means may be used, such as a flange fire, gas or oil burner, so the heat will get well back into the plate and expand it thoroughly. After welding, the drum should be reheated in the vicinity of such weld until the heat has equalized in the dull red spot, and then slowly cooled. For metallic arc welding, preheating or reheating is not required.

Drums shall be stress-relieved after any welding repairs have been made. After repairs have been made, the drums shall again be tested in the regular way.

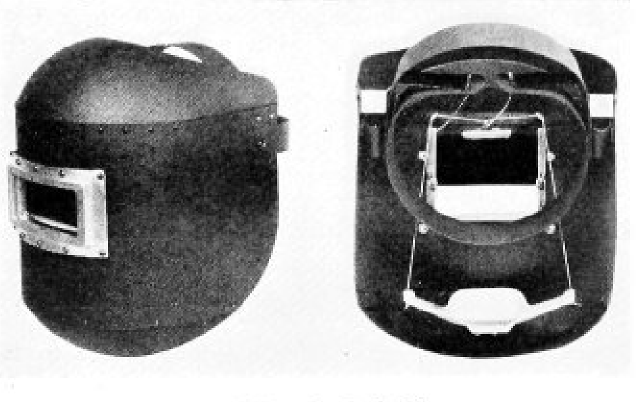
Lincoln Head Shield

A welding operator's head shield, designed to increase production by raising the operator's efficiency, has been developed by The Lincoln Electric Company, Cleveland, O. This new type shield, known as the Weld-Fast head shield, embodies a movable protective lens which allows the operator clear vision without raising the shield. Thus both hands are free at all times and welding can progress with few interruptions.

This shield is an innovation in that the protective lens is held in place in a vertical slide by a spring; slight pressure with the chin on an aluminum rest raises the lens and allows the operator to view the work or change electrodes. When pressure is removed, the lens automatically falls back into place. A stationary cover glass guards the inner lens against spatter and also protects the operator from flying sparks in case the movable lens is accidentally raised.

In order to keep the weight of the shield at a minimum, aluminum is used for the case enclosing the glasses and sturdy featherweight fiber is used for all other parts.

The sides of the shield are brought well back and close to the head, preventing flashes from other operators. The front curves inward to assist in protecting the chin. An adjustable head-band assures the operator of complete comfort while wearing the shield. There is sufficient room in the shield to allow the operator to wear eye glasses or spectacles without interference.



Weld-Fast head shield

Computing the Radius

(Continued from page 197)

If $R = 25$ feet and $C = 14$ feet how high is v?

$$V = 25 - \sqrt{25^2 - \frac{14^2}{4}} = 1 \text{ foot.}$$

If $C = 14$ feet and $v = 1$ foot how long is R?

$$R = \frac{14^2 + 4 \times 1}{8 \times 1} = 25 \text{ feet.}$$

So from the above solutions we know how long a chord is whose versine = 1 foot for a 25-foot radius or *vice versa.* We will now proceed to compute a 25-foot radius pattern whose versine = 1 foot, whose chord is 14 feet and whose distances X are 1 foot, 2 feet, 3 feet, etc., from the center line on the chord, as shown in Fig. 3.

Hence, $O = \sqrt{25^2 - 1^2} - (25 - 1) = 11\ 49/64$ inches for X.

$O^1 = \sqrt{25^2 - 2^2} - (25 - 1) = 11\ 1/32$ inches for X^1.

$O^2 = \sqrt{25^2 - 3^2} - (25 - 1) = 9\ 53/64$ inches for X^2.

$O^3 = \sqrt{25^2 - 4^2} - (25 - 1) = 8\tfrac{1}{8}$ inches for X^3.

$O^4 = \sqrt{25^2 - 5^2} - (25 - 1) = 5\ 15/16$ inches for X^4.

$O^5 = \sqrt{25^2 - 6^2} - (25 - 1) = 3\ 15/64$ inches for X^5.

A line carefully drawn through these points will be the required radius.

Explanation: In the above example, in making the distances X equal to 1 foot, 2 feet, etc., from the center line on the chord, we do away with elaborate calculations to explain the article. Computing the radius is looked upon as developing a radius by mathematical calculations; and in all branches of development, the closer the spaces are together the more accurate the results will be. So the layerout must remember that the higher the degree of accuracy he tries to secure to compute a radius, the closer these distances X must be together.

Schenectady, N. Y. WILLIAM MORRISON.

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States and Cities That Have Adopted the A.S.M.E. Boiler Code

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Arkansas
California
Delaware
Indiana
Maryland
Michigan
Minnesota
Missouri
New Jersey
New York
Ohio
Oklahoma
Oregon
Pennsylvania
Rhode Island
Utah
Washington
Wisconsin
District of Columbia
Panama Canal Zone
Territory of Hawaii

Cities

Chicago, Ill.
Detroit, Mich.
Erie, Pa.
Evanston, Ill.
Houston, Tex.
Kansas City, Mo.
Los Angeles, Cal.
St. Joseph, Mo.
St. Louis, Mo.
Scranton, Pa.
Seattle, Wash.
Memphis, Tenn.
Nashville, Tenn.
Omaha, Neb.
Parkersburg, W.Va.
Philadelphia, Pa.
Tampa, Fla.

States and Cities Accepting Stamp of the National Board of Boiler and Pressure Vessel Inspectors

States

Arkansas
California
Delaware
Indiana
Maryland
Minnesota
Missouri
New Jersey
New York
Ohio
Oklahoma
Oregon
Pennsylvania
Rhode Island
Utah
Washington
Wisconsin
Michigan

Cities

Chicago, Ill.
Detroit, Mich.
Erie, Pa.
Kansas City, Mo.
Memphis, Tenn.
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Omaha, Neb.
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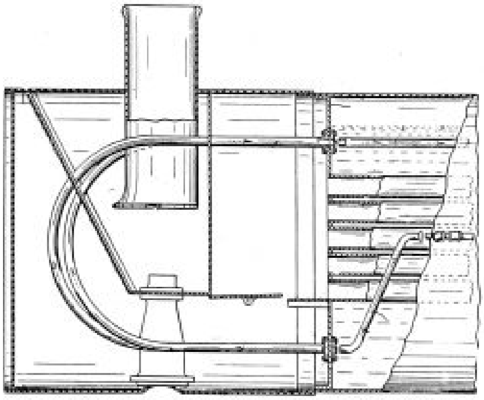
St. Louis, Mo.
Scranton, Pa.
Seattle, Wash.
Tampa, Fla.

Selected Boiler Patents

Compiled by Dwight B. Galt, Patent Attorney, 1372-1376 National Press Building, Washington, D. C. Readers wishing copies of patents or any further information regarding any patent described, should correspond directly with Mr. Galt.

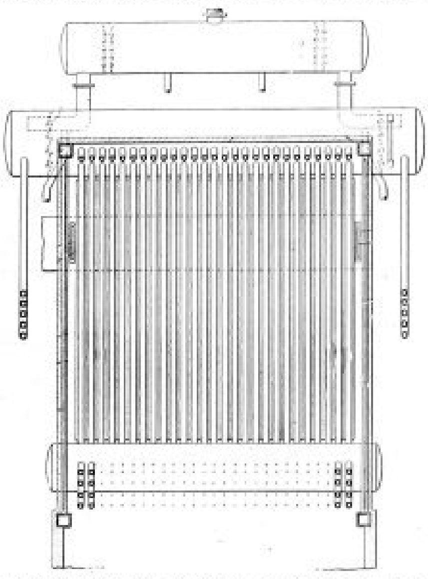
1,809,888. POSITIVE WATER CIRCULATOR. CLIFFORD M. CRANE, OF ST. PAUL, MINN., ASSIGNOR TO LUCIUS WINCHESTER, OF CHICAGO, ILL.

Claim.—A water circulating means for a boiler, having a front flue sheet and a hot gas chamber at the front of the same, comprising one or more open-ended conduits extending from the lower front portion of the boiler and flue sheet forwardly into said chamber then upwardly and rearwardly to the flue sheet and boiler adjacent the water level thereof, and a conduit in the lower portion of said boiler for each of said first mentioned conduits, each of said last mentioned conduits having its front end spaced a short distance from and substantially alined with the lower end of its respective first mentioned conduit and means for directing water through said last mentioned conduits to assist the heat in said gas chamber in causing a circulation of water from the lower portion of the boiler to the top thereof. Eight claims.



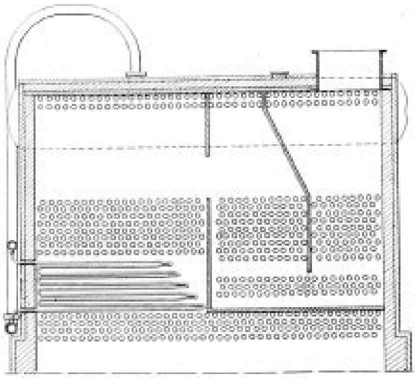
1,809,097. STEAM GENERATOR. WILFRED R. WOOD, OF LONDON, ENGLAND, ASSIGNOR TO INTERNATIONAL COMBUSTION ENGINEERING CORPORATION, OF NEW YORK, N. Y., A CORPORATION OF DELAWARE.

Claim.—In combination a steam and water drum, louvre-like means for dividing said drum into communicating compartments, a plurality of substantially upright radiant heat evaporating tubes discharging into at least second drum and at least one compartment of the first drum which does one compartment of said drum, a second drum, louvre-like means for dividing said second drum into communicating compartments, means for establishing communication between at least one compartment of the not directly receive the discharge of said tubes, and a steam outlet for at least one compartment of the second drum which is not directly in communication with the first drum. Three claims.



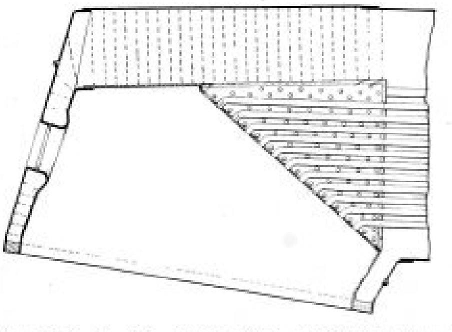
1,808,951. BOILER WITH SUPERHEATER. CHARLES W. GORDON, OF PLEASANTVILLE, N. Y., ASSIGNOR TO THE SUPERHEATER COMPANY, OF NEW YORK, N. Y., A CORPORATION OF DELAWARE.

Claim.—In a horizontal watertube boiler, a plurality of watertubes extending longitudinally of the boiler in inclined relation to the top thereof, a plurality of baffles extending longitudinally of said tubes to provide a plurality of communicating gas passes through which the furnace gases flow in a tortuous passage from the furnace to the stack, said baffle including a vertically extending pair which define with a side wall of the boiler a first gas pass, the members of said pair being vertically spaced from each other above the uppermost tubes to provide an opening between said first gas pass and the second gas pass, said opening being of the same width throughout its entire length, and a transversely inclined baffle located in proximity to said opening for deflecting gases passing therethrough downwardly into said second gas pass. Four claims.



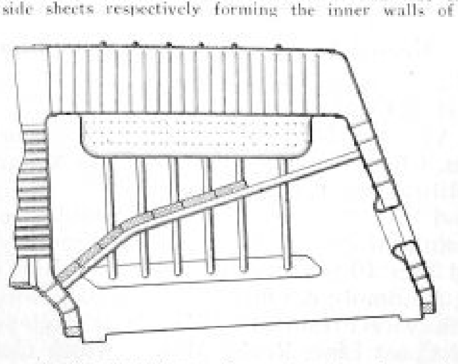
1,803,685. LOCOMOTIVE BOILER. TORRIS H. ALFREDS, OF PARK RIDGE, ILL., ASSIGNOR TO LOCOMOTIVE FIREBOX COMPANY, OF CHICAGO, ILL., A CORPORATION OF DELAWARE.

Claim.—A locomotive boiler embodying therein a firebox having a crown sheet and a front sheet and flues leading from said front sheet, a flat hollow element fixed to said crown sheet and front sheet respectively to open therethrough, and including a wall inclined upwardly and rearwardly from said front sheet, certain of said flues extending through said element to be fixed in and to open through said inclined wall thereof. Five claims.



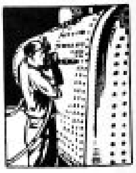
1,807,473. SIPHON GENERATOR AND CIRCULATOR FOR LOCOMOTIVE BOILERS. GEORGE H. EMERSON AND WILLIAM B. WHITSITT, OF BALTIMORE, MD.

Claim.—The combination with a locomotive boiler having a firebox including a flue sheet, a door sheet, a crown sheet and side sheets, said crown and side sheets respectively forming the inner walls of an upper water space and side water legs, of a water steaming and circulating device comprising a trough supported by the crown sheet and extending longitudinally of the firebox from a point adjacent the flue sheet to a point adjacent the door sheet and opening at its top through the crown sheet into the said upper water space, and tubes extending from the bottom of the trough to a side sheet and connecting said trough with the associated side water leg. Ten claims.



The Boiler Maker

Reg. U. S. Pat. Off.



Modern and Obsolete Motive Power

When a complete analysis of the effect of age on locomotive maintenance costs indicates that repair expenditures per unit of potential transportation capacity double in twenty-one years, it is time to consider the replacement of obsolete motive power with modern locomotives. Thomas R. Cook, in an article in this issue, shows that, with the savings due to repairs only, new locomotives can replace motive power twenty years of age and over and amortize the investment with a five percent interest charge in seventeen years.

Because of the advancement in locomotive and boiler design appreciable savings in maintenance costs over old motive power may be realized in view of the relatively low cost of repairs of new locomotives. It is desirable from the point of view of all-around efficiency to operate modern railroads with modern power. Where savings in repairs alone will pay for the cost of modern equipment, the acquisition of new motive power should be seriously considered.

One Hundred Years of Progress

One hundred years ago, or three years after the advent of the first locomotive, the *American Railroad Journal*, the oldest trade publication in the United States, came into existence. This month the *Railway Mechanical Engineer*, a direct descendant, without a break in the line, of this century-old "advocate of internal improvements," and one of a group of Simmons-Boardman publications of which The Boiler Maker is a member is celebrating its centennial.

While the *Railway Mechanical Engineer* followed the trend in railroad transportation and was established to disseminate the various developments in locomotive construction, the railroad industry, during the past century, has grown in such proportions that it is made up of numerous trades, each specializing in one particular phase of railroading. With the establishment of the Master Steam Boiler Makers' Association in 1902, *Motive Power*, the predecessor of The Boiler Maker, was founded to deal with the art of scientific boiler making, sheet metal work and the generation and use of motive power.

The industry of boiler making, as applied to railroading, has not stood still during the past century; for upon the ability of boiler designers and boiler makers depended the production of steam generators capable of producing the tractive force required by the railroad mechanical departments. The original low-pressure boilers used a century ago bear little resemblance to the high-pressure boilers of the present day.

While the development of the locomotive boiler during the past century has followed, with few exceptions, along conventional lines, limitations in early construction were caused by the inability of rolling mills to produce large-sized plates. When it is considered that practically no machinery for handling heavy material was available during the first part of the 19th century, it may be realized that the development in design was limited by the ability of manufacturers and their machinery to fabricate boilers. At the present time, however, with the production of high-grade steel, modern machinery and welding methods, a number of limitations imposed upon early designs have been removed. The use of the all-welded firebox has been proven in service, while other developments in boiler staying and construction render the modern locomotive boiler quite different from that produced one hundred years ago.

Inspection of Watertube Boilers

Since the watertube boiler in marine installations is a much more delicate type of steam generator than the ordinary Scotch or return-tube type of boiler, it demands not only greater care in its actual operation and maintenance, but it requires more frequent internal and external inspection to discover and remedy incipient faults. In England a number of analyses have been carried out which show that, on land, watertube boilers are liable to fracture in the vicinity of the longitudinal seams, particularly where lap joints instead of butt joints are employed.

It is apparent that such defects as these are slow in development, for it is only on boilers which have been in service for ten years or more that such faults have been detected. Their timely discovery, however, has no doubt prevented explosion.

It is also indicated from survey that fractures are apt to occur at the boundaries of the flats or steps pressed in the drums on some designs of boilers in order to receive the tubes, at the seams connecting cross boxes with the underside of steam and water drums and at the edges of the feed inlet openings in dished end plates. Generally, a preliminary warning of the existence of a fracture is manifest by leakage. For this reason any leakage, however slight, should be subjected to thorough inspection to prevent serious explosions and to remedy such faults as do exist.

Effect of Age on

Locomotive Maintenance Costs*

By Thomas R. Cook

In a study of the maintenance of several thousand locomotives made for The Baldwin Locomotive Works, the author shows that repair expenditures per unit of potential transportation capacity double in twenty-one years. He also indicates that in view of the relatively low cost of repairs, modern locomotives can replace present power of twenty-one years of age and amortize the investment with five percent interest in the same length of time.

Repair costs vary with use, size and type of the locomotives, and customary units of "Cost per locomotive year" or "Cost per locomotive-mile" do not permit of a proper comparison as between different railroads or over a period of years for the same line. The first ignores the effect of use, while the second gives no consideration to the size and character of the engine. It has therefore been found necessary to establish a unit which gives due weight to the ability of the modern locomotive to produce high drawbar pull at speed. Extended studies have pointed to the unit of "horsepower miles" as the proper divisor to use in calculating costs of locomotive repairs. Consideration has been given to the use of "tractive force miles" for this purpose, but there are many locomotives with practically identical tractive force that vary greatly in power at certain speeds and necessarily differ in cost of repairs for equal use. The use of horsepower miles has been tested on a large number of locomotives and the results have been found to be uniformly consistent.

Horsepower, as used for this purpose, must take into account the ability of the plant to make steam and the efficiency of transforming the potential energy in the steam into useful work; further, the calculations should be easily and readily made.

The ability of the plant to make steam (that is, to evaporate water from the temperature of the feedwater into steam at the working pressure of the locomotive) is dependent in general on four factors:

(a) The amount of direct heating surface contained in the firebox arch tubes, thermic syphons and combustion chambers.

(b) The amount of indirect heating surface contained in the flues or tubes.

(c) The length of flues or tubes.

(d) Whether or not the plant is equipped with a feedwater heater, exhaust-steam injector or other means of using available heat in increasing the feedwater temperature.

The efficiency of the transformation of the potential energy of the steam into useful work (tractive force at the rim of the drivers at various speeds), is dependent in the main on two factors: (a) working pressure; (b) degree of superheat.

Using only the principal factors involved in the problem as outlined above, we have developed a formula for the determination of horsepower as designated in the unit for measuring repair costs. As the formula does not take distance into consideration, the result of its application is not horsepower in the ordinary acceptance of the word and we have, therefore, referred to it as boiler or potential horsepower.

In application, the boiler or potential horsepower of the locomotive is determined by dividing the total or adjusted pounds of water evaporated per hour by a factor which represents the normal weight of steam required per indicated horsepower-hour.

In determining the evaporation of water per hour, the following factors are used for direct and indirect heating surface.

1—Fifty-five pounds per square foot is the basis for the direct heating surface, consisting of firebox, combustion chamber, thermic syphons and arch tubes.

2—The evaporation per square foot of indirect heating surface varies with the length of tube, as shown in Table I.

3—The total evaporation of the boiler is the aggregate of that of the direct and indirect heating surfaces. When a feedwater heater is used, this sum is adjusted by an increase of eight percent.

The factors used as divisors in figuring boiler horsepower vary with the steam pressure, with saturated and superheated steam, and with the degree of temperature of the latter as shown in Table II, or in graphic form in Fig. 1. The factors for superheated steam locomotives are shown for three degrees of superheat, to be used for locomotives equipped as follows:

Type A, original (small)	150 deg.
Type A, improved (large)	200 deg.
Type E	250 deg.

The empirical values used in the preparation of the formula reflect the actual facts. The factors for evaporation are standard, while those covering the use of steam (transformation of energy) are based on various test data and start with the well known figure of 21 lb. of steam per i. hp. hr. for a superheater locomotive at 200 lb. pressure and 150 deg. of superheat (Type A superheater), and with 28-lb. of steam per i.hp. hr. for a saturated locomotive at 200-lb. pressure.

The determination of potential horsepower for the various classes of locomotives on any given railroad is comparatively simple. In addition to the constants given in the formula, all the information necessary can be found in the locomotive registers of the railroad companies. The actual computations for the determination of the horsepower of 12 locomotives of varying capacities are illustrated in Table III.

The average cost of repairs per horsepower-mile is approximately one-tenth mill. In order to have a repair cost unit that can be easily used and compared, we have adopted 10,000 horsepower-miles, called the "horsepower unit." The average cost of repairs per horsepower unit is, therefore, approximately $1.00.

If a locomotive of 3,000 hp. makes 30,000 miles per year, its performance is as follows:

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* Abstract of an article which appeared in *Baldwin Locomotives* for April, 1932.

$$\frac{3{,}000 \times 30{,}000}{10{,}000} = 9{,}000 \text{ horsepower units}$$

If the total cost of repairs of the above locomotive is $10,000, the unit cost will be $10,000 divided by 9,000, or $1.11. The cost per locomotive mile can be derived from the above figures as follows:

$$\frac{\$1.11 \times 3{,}000}{10{,}000} = 33\ 1/3 \text{ cents}$$

An analysis of the cost of locomotive repairs on a number of roads has been made, using the longest period wherein such costs represent present day practice and a fairly constant value. In general, it is found that the railroads have made substantial reductions in equipment maintenance since 1920, reaching an approximately even trend in 1927.

The analyses, as a rule, have covered the years 1927, 1928 and 1929. In making this study the repair cost of each class of power for each year of age was determined. For instance, a group of locomotives purchased in 1917 was ten years old in 1927, eleven years old in 1928 and twelve years old in 1929. On another group of the same class, purchased in 1920, we would have repair costs for seven, eight and nine years of age. Thus we would obtain costs of maintenance per horsepower unit on this particular class from the ages of seven to twelve years. In this study this range in age varied generally from three to sixteen years.

These compilations indicate a decided upward trend in the cost of repairs with increasing age. This rise is abrupt during the early years with constant but less rapid increase thereafter.

The various classes of locomotives on any one road have shown a uniform repair cost per horsepower unit at the same year of age regardless of the size of the machine, varying from articulated locomotives of 140,000 lb. tractive force to the 2-6-0 type of less than 30,000 lb. tractive force. The trend of these costs varies for individual roads, due to some extent to special operating conditions, but largely to differences in maintenance policy.

Fig. 2 shows the total cost per horsepower unit of 3,370 locomotives covered in this study, comprising 8,317 locomotive repair years. Curve A is the cost per horsepower unit of all locomotives at the various indicated ages. Curve B is a five-year moving average of the points shown on line A. Curve C is the least square graduation from the fourth to the twenty-eighth year. A smooth curve was extended from the first year, meeting the straight line trend beginning at the fourth year, and the straight line trend was extended beyond the twenty-eighth year.

Curve C reflects the normal cost of maintenance of the locomotives included in our study. It will be noted that the actual costs of repairs to engines over 25 years

Table I—Factors of Evaporation for Tubes of Varying Lengths

Tube length, ft.	Evaporation, lb. per hr.	Tube length, ft.	Evaporation, lb. per hr.
10.0	13.00	17.5	10.00
10.5	12.75	18.0	9.85
11.0	12.55	18.5	9.70
11.5	12.30	19.0	9.50
12.0	12.10	19.5	9.35
12.5	11.80	20.0	9.20
13.0	11.65	20.5	9.05
13.5	11.45	21.0	8.95
14.0	11.25	21.5	8.80
14.5	11.10	22.0	8.65
15.0	10.90	22.5	8.55
15.5	10.70	23.0	8.40
16.0	10.50	23.5	8.30
16.5	10.35	24.0	8.20
17.0	10.20		

Table II—Factors for Determining Normal Horsepower

Steam pressure, lb.	Lb. saturated steam	Steam per normal hp.-hr., with superheat of		
		150 deg.	200 deg.	250 deg.
170	29.00	21.75	20.60	19.50
175	28.75	21.65	20.40	19.30
180	28.60	21.45	20.25	19.20
185	28.40	21.30	20.10	19.05
190	28.30	21.25	19.90	18.90
195	28.10	21.10	19.80	18.80
200	28.00	21.00	19.70	18.70
205	27.80	20.90	19.60	18.60
210	27.75	20.80	19.50	18.50
215	27.60	20.70	19.40	18.40
220	27.50	20.65	19.30	18.30
225	27.40	20.55	19.25	18.25
230	27.30	20.50	19.10	18.20
235	27.20	20.45	19.05	18.15
240	27.10	20,35	19.00	18.10
245	27.00	20.25	18.90	18.00
250	26.90	20.20	18.85	17.95
255	26.85	20.15	18.80	17.90
260	26.80	20.10	18.75	17.85
265	26.75	20.05	18.75	17.80
270	26.65	20.00	18.75	17.80
275	26.60	20.00	18.75	17.75
280	26.55	19.95	18.70	17.75
285	26.50	19.90	18.70	17.75

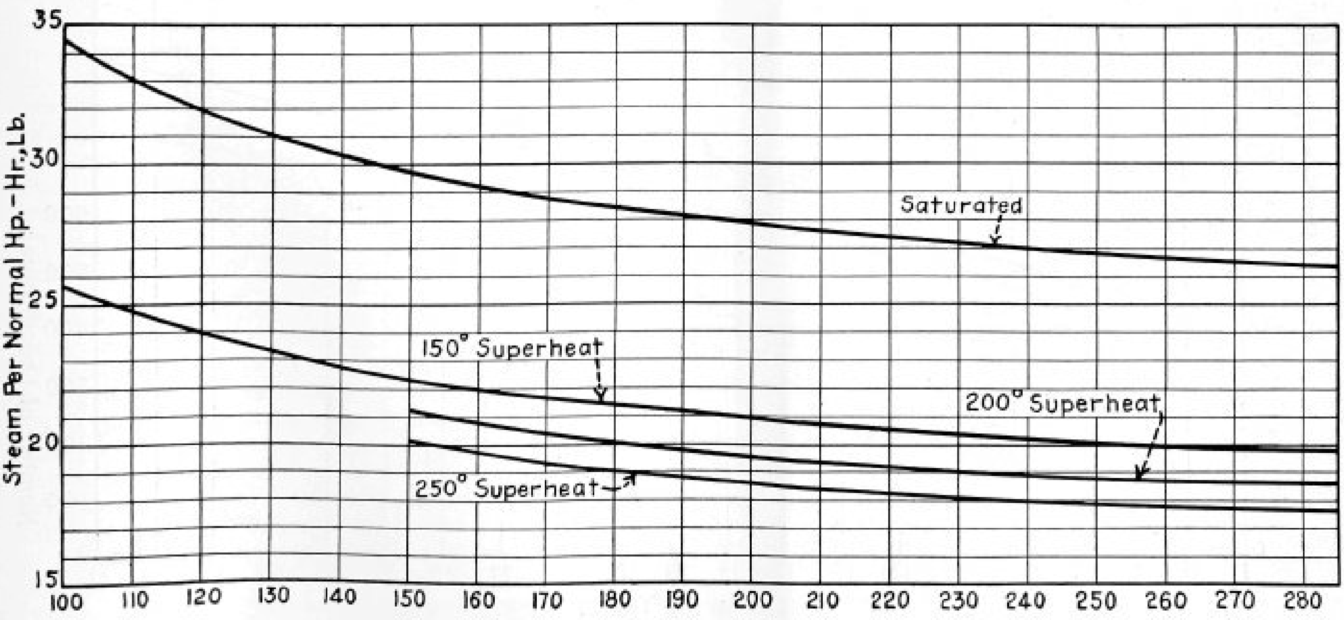

Fig. 1.—Factors for determining normal horsepower

Table III—Potential Horsepower Calculations

A Engine No.	B* Direct heating surface, sq. ft.	C Evap., direct heating surface, lb. (B x 55)	D* Indirect heating surface, sq. ft.	E* Length of tubes ft. in.	F† Tube factor	G Evap., indirect heating surface, lb. (D x F)	H Total evap. per hour lb. (C + G)	J* Degrees super-heat	K* Steam pressure, lb.	L‡ Steam factor	M Hp. (H ÷ L)	N* Feed water heater	O Hp. corrected for F.W.H. (M × 1.08)§
1	199.7	10,984	2,086.8	14-2	11.25	23,477	34,461	...	190	28.30	1,218	No	1,218
2	175.7	9,664	1,769.8	16-0	10.50	18,583	28,247	150	200	21.00	1,345	No	1,345
3	232.4	12,782	3,446.0	21-0	8.95	30,842	43,624	150	200	21.00	2,077	No	2,077
4	259.6	14,278	3,894.3	21-0	8.95	34,854	49,132	200	185	20.10	2,444	No	2,444
5	325.0	17,875	3,084.0	18-6	9.70	29,915	47,790	200	200	19.70	2,426	Yes	2,620
6	288.0	15,840	4,369.0	20-0	9.20	40,195	56,035	150	200	21.00	2,668	No	2,668
7	313.0	17,215	3,978.0	19.0	9.50	37,791	55,006	200	200	19.70	2,792	Yes	3,015
8	433.0	23,815	4,878.0	21-0	9.95	43,658	67,473	250	250	17.95	3,759	Yes	4,060
9	558.0	30,690	4,656.0	21-0	8.95	41,671	72,361	250	275	17.75	4,077	Yes	4,403
10	598.0	32,890	5,516.0	21-0	8.95	49,368	82,258	250	300	17.55	4,687	Yes	5,062
11	680.0	37,400	6,420.0	24-0	8.20	52,644	90,044	250	250	17.95	5,016	Yes	5,425
12	866.0	47,630	6,800.0	22-0	8.65	58,820	106,450	250	250	17.95	5,930	Yes	6,404

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* From the locomotive specifications.
† See Table I.
‡ See Table II.
§ Applied only when feedwater heater is used.

of age are, with few exceptions, above the trend line C. As a rule, these older locomotives are assigned to branch and minor runs where it is impossible to make large mileage, which fact contributes to a great extent to the erratic costs. It is believed that the trend line C indicates the cost of repairs that would have obtained if the locomotives in question had been kept in regular service.

The trend line C rises rapidly from the first to the third year. The straight line trend from the third year on indicates an increase in cost per horsepower unit of $.036 per year. On the basis of an average annual cost of $1.07 for the first twenty years of life, this amounts to an increase of 3.4 percent. Assuming an annual cost of locomotive repairs at $10,000,000 this yearly increment amounts to $340,000.

With the method used in making up the cost curves it follows that the costs of repairs for, say, the twentieth year are based upon locomotives purchased twenty years ago. It is not probable that the maintenance of the modern locomotives will reach as high a cost as those of older design, and, therefore, any savings in repairs based on this trend will be understated as the savings indicated would be increased to the extent that the cost of repairs of the locomotives built today would fall below the trend on present equipment.

Curve D on this same sheet is the record of the railroad having the lowest repair costs of all the roads

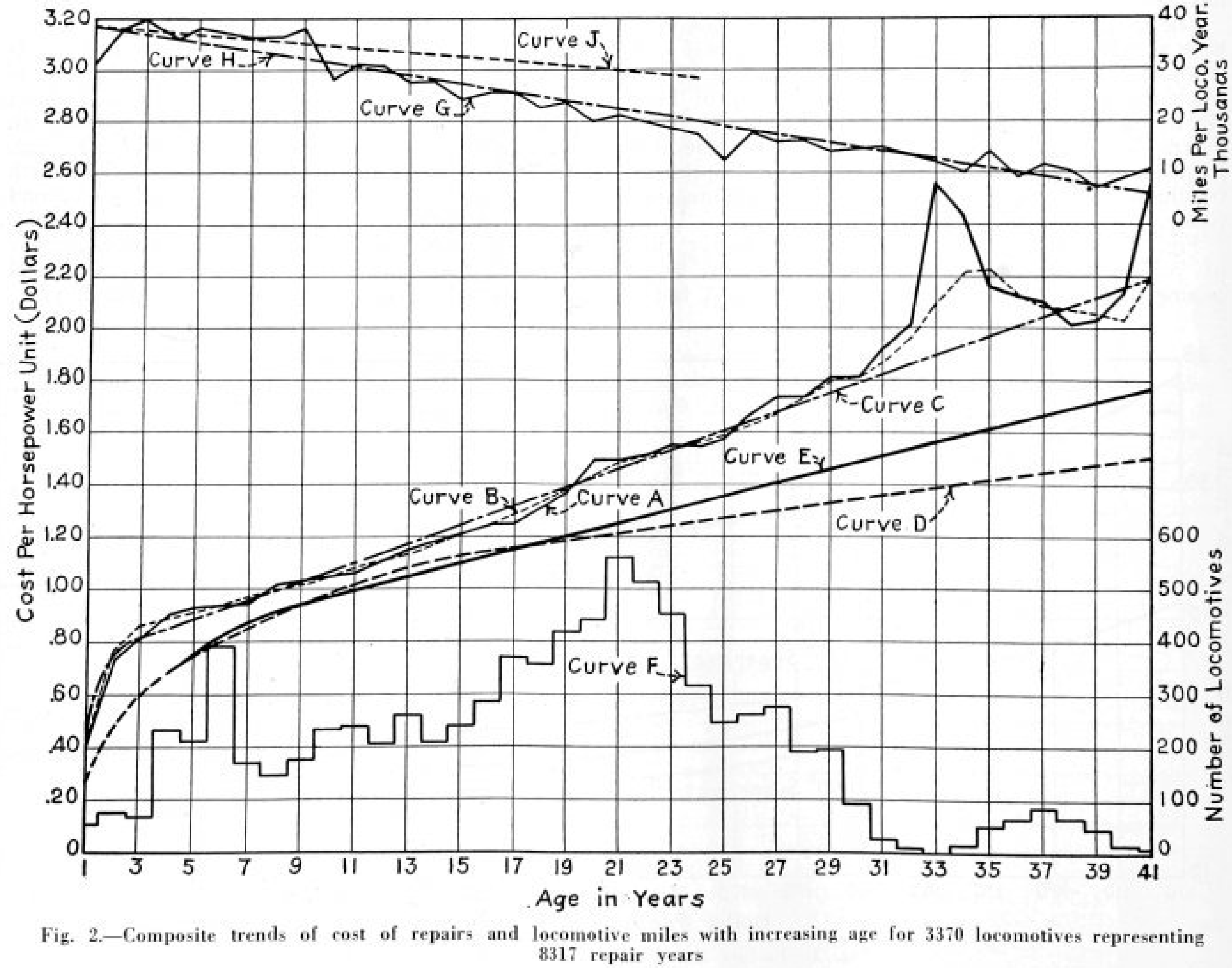


Fig. 2.—Composite trends of cost of repairs and locomotive miles with increasing age for 3370 locomotives representing 8317 repair years

studied. This curve was used in calculations of savings on specific engine runs on the road in question. On this particular railroad, locomotives are retired after about the twenty-sixth year which results in a downward trend during this period, due to utilizing the full mileage possible before retirement. To compensate for this and indicate the lowest cost, curve D has been revised by taking a trend determined by the method of least squares from the eighth to the twenty-second year. The resulting trend line is shown by curve E. Curve C then represents the average of the roads studied and curve E the least costs indicated as a result of such a survey.

At the bottom of the graph curve F indicates the number of locomotives making the average cost at the year indicated, and it will be seen that for all but the first three years approximately 292 locomotives were involved in the compilation of this research up to the twenty-ninth year.

At the top of the graph are curves showing the mileage made by engines at various ages. Curve G shows the actual average mileage of the locomotives shown in curves F and C. Curve H is the trend of this line made by the method of least squares. It is of interest to know that the yearly mileage of the road with the low cost represented by cost curve D so closely coincides with curve H, that the slight difference made it impracticable to show this by a separate line.

The decrease in annual mileage with increasing age is due to a large extent to the transfer of power to less and less important runs. With the modern locomotive, and especially the large power now being purchased for the more important runs with heavy traffic, a reduction in mileage as great as that shown in trend H is not anticipated, as heavy modern power will not be used on branch line service even when old. Exactly what the trend of the new power will be is purely conjectural. For the basis of our calculations we have assumed curve J showing a decrease from 39,000 miles per year in the first year to 30,000 in the twentieth year.

With an assumed annual mileage and knowing the cost of repairs at various years of age, the savings to be obtained from repairs alone and the results of application of these savings to the amortization of the investment can be determined on the assumption that a known amount of freight can be handled over a division or engine run at a given speed with the same expenditure of horsepower units whether or not the size or character of the power is changed. This assumption is practically correct; traffic conditions and comparative suitability of the power will cause slight variations in detail application. On the above basis and using the cost per horsepower of $27, we have made a number of determinations which are summarized in the following paragraphs.

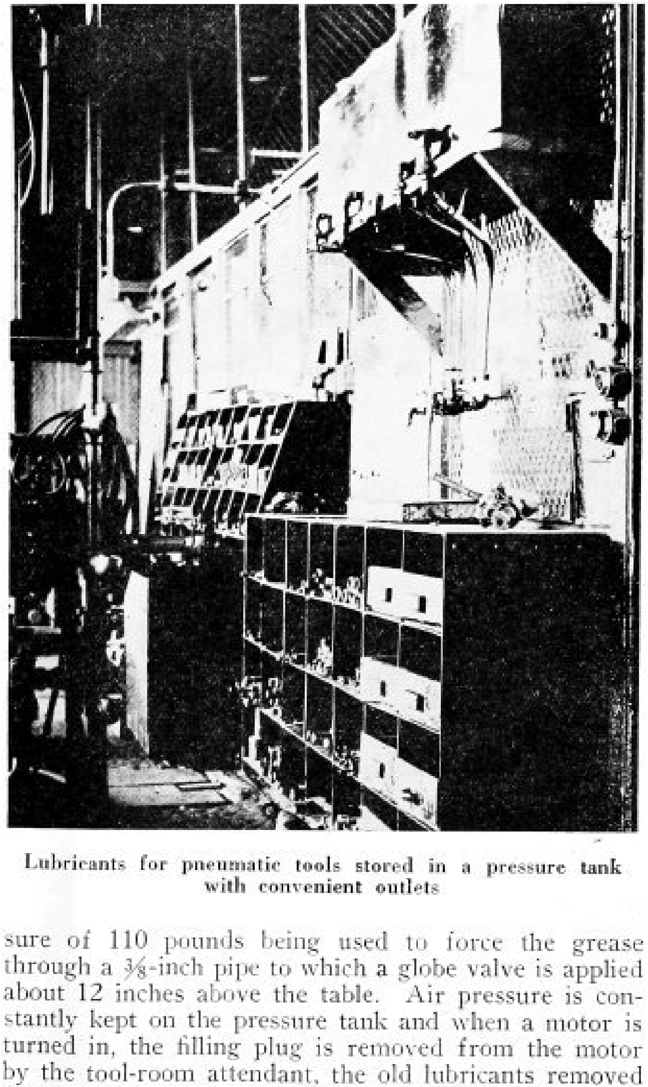
On the basis of mileage curve H and cost curve C, representing the average of all roads studied, we find that with the savings due to repairs only, new locomotives can replace those twenty years old and amortize the investment with a five percent interest charge in seventeen years. With mileage curve J, but using curve E (the lowest to be anticipated from these studies), new locomotives can replace those twenty-two years of age, amortizing the investment in twenty-one years and can replace the locomotives twenty years of age, amortizing the investment in twenty-four years.

This study indicates that with the relatively low cost of repairs shown in Curve E, modern locomotives can replace present power of twenty-one years of age and amortize the investment with five per cent interest in the same length of time.

Lubricating Pneumatic Tools

Such pneumatic tools as hammers and motors should be returned to the tool room each day for adjustment and lubricating. The various sizes of hammers and motors require different grades of lubricating oils and greases and the work should be performed by a workman who is thoroughly familiar with these requirements.

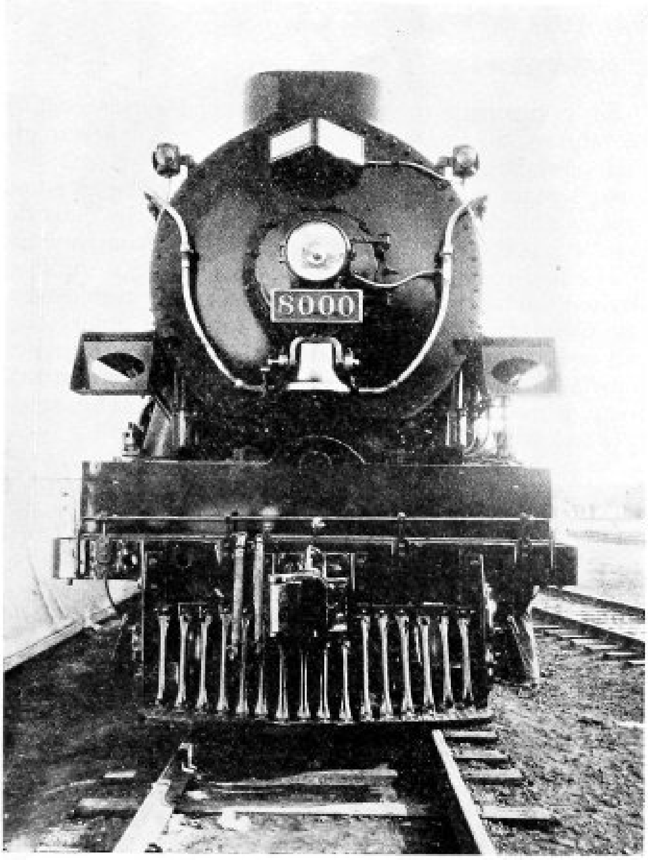
The tool-room foreman employed at an eastern shop devised the lubricating system shown in the accompanying photograph. Four different kinds of lubricating oils and greases are placed in separate compartments of a metal tank which is located over the table in the tool room where the machines are lubricated. The metal pressure tank is hydrostatically tested to permit a pressure of 110 pounds being used to force the grease through a ⅜-inch pipe to which a globe valve is applied about 12 inches above the table. Air pressure is constantly kept on the pressure tank and when a motor is turned in, the filling plug is removed from the motor by the tool-room attendant, the old lubricants removed if they indicate foam or deterioration and new lubrication forced into the motor by placing it under the proper spigot.



Lubricants for pneumatic tools stored in a pressure tank with convenient outlets

It is recommended that a storage tank be provided and the air screened before it is permitted to enter the pressure tank and come in contact with the lubricants as it sometimes contains grit and other foreign particles that are likely to cause damage when carried into the air tools. By systematically maintaining and adjusting pneumatic tools and using the lubricant best suited for each particular tool, many years of satisfactory tool service can be obtained at low cost.

Developing the Multi-Pressure Locomotive[*]



The C. P. R. multi-pressure locomotive

A symposium of three papers discussing the design and performance of the Canadian Pacific multi-pressure locomotive No. 8000 was presented at the spring meeting of the American Society of Mechanical Engineers, held at Bigwin, Ont., June 27 to July 1, 1932. An abstract of the first of these papers, by F. A. Schaff, president of the Superheater Company, appeared in the September issue of THE BOILER MAKER. This article includes the abstract of the paper by J. B. Ennis, vice-president, American Locomotive Company. The paper by H. B. Bowen, chief of motive power and rolling stock, Canadian Pacific, will appear next month.

In the development of the multi-pressure locomotive it was the intention from the start to deviate as little as possible from the conventional lines of locomotive design and to use as many parts of the Canadian Pacific Class T-1 locomotive as practicable. This locomotive, built by the Montreal Locomotive Works in 1929, is the most powerful freight locomotive of the Canadian Pacific. Its total weight is 370,000 pounds, its weight on drivers is 312,800 pounds, and its tractive force 77,200 pounds without the booster. The multi-pressure locomotive was supposed to be not less, and if possible more powerful than the Class T-1, without increasing the number of wheels. Therefore, the requirement to adhere, in general, to the design of the Class T-1 locomotive was natural.

In two major respects, however, novel features had to be introduced: in the locomotive engine and in the structural connection of the boiler and locomotive chassis.

Regarding the locomotive engine, it was estimated that 63,000 pounds of steam would be furnished, of which approximately 56 to 47.6 percent would be high-pressure steam of 850 pounds pressure, and the remainder of 250 pounds pressure. Both high- and low-pressure steam were supposed to be superheated. It was expected that the total temperature of the first would be between 600 degrees and 800 degrees F., while the total low-pressure steam temperature would fluctuate between 550 degrees and 725 degrees F., depending on the rate of firing.

To utilize the heat contained in the steam in the most efficient manner, it is necessary to use double-expansion for the high-pressure steam. The most practical combination of the two-stage expansion with one-stage expansion of the low-pressure steam generated separately, is the one applied in the Schmidt-Henschel Locomotive of the German State Railways, namely, the mixing of the exhaust from the first stage of the high-pressure expansion with the low-pressure live and superheated steam, using this mixture for the second stage of the expansion. By doing so, the necessity of interstage heating for the high-pressure is eliminated, resulting in simplification of the locomotive.

It is obvious that more than two cylinders are necessary for this scheme. The cross-compound system is not possible in view of the excessive size of the single low-pressure cylinder for a locomotive with 320,000 pounds weight on the drivers. Either three or four cylinders are necessary, and then several cylinder arrangements could be considered:

1—A tandem compound arrangement, having the advantage of the outside location of all four cylinders without the disadvantage of a crank axle. The drawback, however, being heavy reciprocating parts, resulting in poor balancing, the inaccessibility of piston-rod packings between the cylinders, and the undesirable lengthening of the front part of the locomotive, with ensuing increase in weight.

2—A four-cylinder compound arrangement as used in the high-pressure locomotive of the French Paris-Lyons-Mediterranean Railway. This would require a two-crank axle, which, in view of the power of the locomotive, it would be impossible to construct in the limited space between the frames.

3—A three-cylinder arrangement with a single crank axle as used in the Schmidt-Henschel locomotive, with a middle high-pressure and two outside low-pressure cylinders.

4—A three-cylinder arrangement differing from the foregoing by making one inside low-pressure cylinder between the frames and two outside high-pressure cylinders. This would also require too large a cylinder, which it would be impossible to place inside the frames.

Thus, of all the four arrangements, the most practical seemed to be the third. Although this necessitated the use of a crank axle, it was thought that as the power on the middle pin is less than in the three-cylinder

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[*] This is the second of three articles on the development of the multi-pressure locomotive. The first appeared in the September issue.

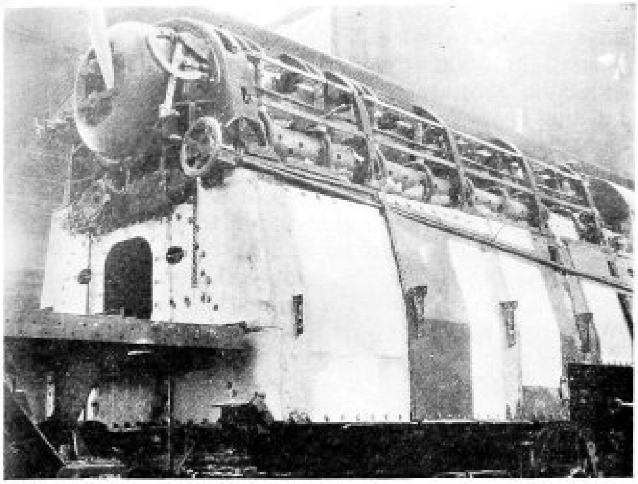
4-12-2 type Union Pacific locomotive,* a satisfactory three-cylinder arrangement could be worked out.

Having decided on this, the next question was the distribution of power. The most natural thing to do would be to divide the power equally between the cylinders. It was not so much the division of tractive force at starting, as the proper division of power under running conditions, as the overall efficiency would be more affected by the latter than by the former. On the other hand, it was rather desired to get slightly less tractive force from the middle cylinder so as to protect the middle main pin.

The three-cylinder arrangement with two working pressures offers a very convenient means for equalizing power without exceeding stresses in the middle pin, by introducing a longer cutoff in the middle cylinder. By doing so, the size of the cylinder can be made smaller and thus the piston thrust in the middle cylinder can be kept within limits. However, this would mean either introducing a complication in the valve gear, or using an independent valve motion for the middle cylinder, whereas from the start it was agreed to use the Gresley valve motion, as in the majority of three-cylinder locomotives built in this country. In view of this, it was decided to make the cutoffs in all three cylinders equal by using the Gresley gear, and to compromise between division of power at starting and under running conditions in such a way as to keep the diameter of the middle cylinder as small as possible.

A study of indicator cards for different cutoffs and various cylinder sizes was made, in connection with service conditions and boiler capacities, and it was found that the ratio of volume of the two low-pressure cylinders to that of the high-pressure cylinder should be 5.4. This determined the following dimensions for the cylinders: 15½-inch diameter and 28-inch stroke for the middle high-pressure cylinder, and 24-inch diameter and 30-inch stroke for the two outside low-pressure cylinders.

On the basis of 850 and 250 pounds pressure in the two boilers, the diameter of driving wheels was made 63 inches, 85 percent coefficient in the tractive force formula, and making allowance for the areas of the high-pressure piston rod and piston-rod extension, the tractive force was estimated to be 83,300 pounds, corresponding to 3.84 factor of adhesion. A study of tangential forces in this three-cylinder locomotive proved that this factor should not cause slipping of the locomotive.

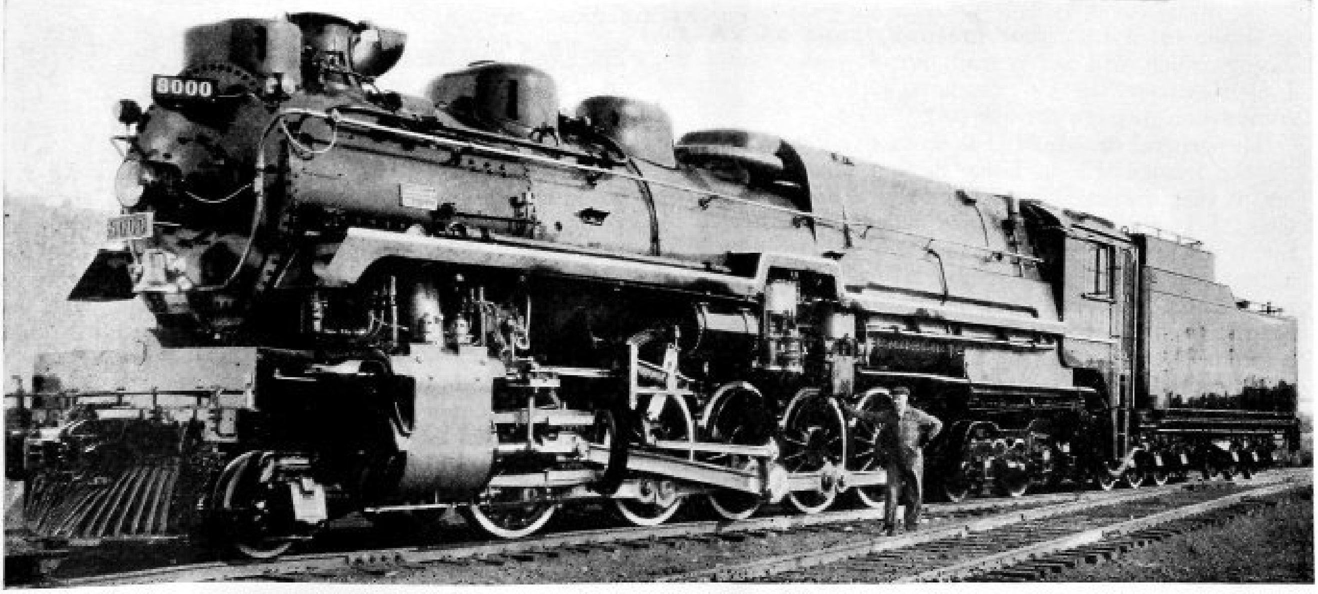


Special framing and air-tight casing used in the construction of the firebox

In view of some uncertainty in distribution of power under actual conditions which this experimental locomotive will have to meet, provision was made for the possibility of a slight change in the sizes of cylinders by making bushings in the cylinders of sufficient thickness.

The middle cylinder is of the same type as used on ordinary three-cylinder locomotives, with the cylinder barrel, steam chest and boiler saddle cast in one piece. In this case, the front bumper and bumper bracket are cast integral with the cylinder. Special precaution has been taken to make the bolting of the cylinder as rigid as possible.

The cylinder is made of cast steel to a special specification, with a silicon content of 0.40 percent and manganese 0.82 percent. The cylinder barrel is inclined at



Canadian Pacific double-pressure locomotive built at its Angus shops with the co-operation of the American Locomotive Company and the Superheater Company

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* See the May, 1926, issue of THE BOILER MAKER, page 123.

an angle of 8 degrees 31 minutes to the horizontal to permit the middle main rod to clear the front driving axle, the second being the crank axle. The cylinder bushing is made of high grade cast iron of a composition insuring good wearing qualities. The steam chest is horizontal, as the case usually is in three-cylinder locomotives with Gresley valve motion, and has two high-grade cast-iron bushings for the high-pressure 6-inch piston valve. Steam is admitted between the piston-valve heads (inside steam admission) through a 4½-inch steam pipe. The exhaust is directed through two 5-inch openings, one on each side of the cylinder, into the two mixing chambers, where it meets the low-pressure superheated steam and whence it is further directed into the two low-pressure cylinders.

The exhausts from the outside cylinders have to pass through the middle cylinder into the exhaust pipe and nozzle. A rectangular 5½-inch by 10-inch opening is provided on each side of the middle cylinder in the fitting surface. To insure tightness between the middle and outside cylinders, special copper wire interwoven asbestos gaskets are used. Provision is made for passing the booster exhaust through a specially made passage in the saddle into the exhaust pipe, in case of future application of a booster. The openings for the booster pipe connections are, for the present, blocked off by welded-in plates.

The valve gear for the outside cylinders is of the Walschaert type and gives 7½-inch valve travel. The valve motion for the inside cylinder is derived from the two outside gears by the combined Gresley valve motion frequently used on three-cylinder locomotives in America. The Gresley levers have ball bearings in the main fulcrum and in the connection between the two levers. As stated above, the travel and cutoffs are the same in all cylinders. The reverse is of the screw type, which is standard on the Canadian Pacific. No power reverse is used.

In the multi-pressure locomotive the low-pressure boiler is in principle of the same design as the conventional boiler. The high-pressure boiler in itself, however, does not offer sufficient rigidity to the locomotive structure. It was, therefore, necessary to enclose the firebox and the combustion chamber in a special framing which would form a rigid connection between the boiler and locomotive frame.

Moreover, a watertube firebox requires an air-tight casing which will not permit outside air to enter the firebox, except the air which is admitted for proper combustion through the ash pan.

The general design of the locomotive chassis, including the frames, wheels, brake rigging, etc., is very similar to that of the Canadian Pacific 2-10-4 locomotive, Class T-1, except of the parts which were affected by the difference in the type of boiler and machinery. As a matter of fact, a great number of parts and the tender as a whole are practically identical. The locomotive frame, however, for this experimental engine was not a locomotive bed as in the T-1 class, but of the built-up type, consisting of two frames (right and left) 6½-inches thick, made of nickel cast steel.

The locomotive is designed to negotiate 18-degree curves. For this purpose the front axle has the Alco-lateral motion device, with a play of 1 inch on each side. The front engine truck is of the Commonwealth two-wheel, cast-steel, outside-bearing type with 7¼-inch lateral movement, while the trailing truck is of the four-wheel, cast-steel, Commonwealth type, with 10¼-inch lateral displacement necessary for 18-degree curves.

Following the example of the German State Railway locomotive, the cranks were spaced at approximately 120 degrees for the sake of torque uniformity. As it was expected, this crank setting resulted in an uneven exhaust sound. The engine being of the compound type and having only four exhausts per revolution, sounds "lame" at low speeds, but at speeds above 15 miles per hour, the non-uniformity of the exhaust vanishes.

The crank setting is, of course, corrected for the inclination of the middle cylinder and for the difference in strokes of the high and low-pressure cylinders. For the middle crank it is 129 degrees 12 minutes with the left-hand crank, and 110 degrees 48 minutes with the right-hand crank, the right being the leading. The angle between the two outside cranks is exactly 120 degrees.

The counterbalances for the reciprocating weights are distributed evenly between all driving wheels except the main. The total percentage of counterbalancing of the locomotive, figured by the static method, is 34.75 percent.

All driving and trailer-truck wheels of the locomotive have brakes operated by a Westinghouse automatic brake equipment, with 8½-inch cross-compound air compressor. The driving wheel brakes are divided in two sections, the front section operating the three front axles from two 14-inch by 10-inch cylinders, while the back portion operates the two rear pair of drivers from two 12-inch by 10-inch cylinders. The trailer truck has a separate brake applied to all wheels, using two 8-inch by 12-inch cylinders. Three main reservoirs are used on the locomotive, with a total capacity of approximately 80,000 cubic inches.

All steam auxiliaries are operated by low pressure, except the two high-pressure pumps which are operated by high-pressure steam. There are two cab turrets, one for superheated and one for saturated steam. The first has connections to the high- and low-pressure feed pump, the air compressor, the blower, the oil-burner operating valves, the mechanical-lubricator heating coil, the headlight dynamo and the whistle. The second turret has valves to the steam-heat line and to the injector.

Abstract of the paper by Mr. Bowen will appear in the next issue of THE BOILER MAKER. Operating details and the advantages of high-pressure steam will be discussed.

Self-Consumer Automatic Welder

The Champion Rivet Company of Cleveland, Ohio, for the past forty years one of the outstanding rivet manufacturers in the country, announces an automatic device that should add much to the art of welding and incidentally reduce costs to a minimum. While it is true that automatic welding has been in existence ten years or more, costs of original equipment have been high and many users have hesitated making this investment.

The device is of simple structure consisting of arms extended from a bus bar so arranged that as electric current passes through these arms, the electrode which has been attached to the work gradually consumes itself without any effort on the part of an operator. This feature has prompted the term "self-consumer" when reference is made to this machine.

It offers real possibilities in the welding of pressure vessels where longitudinal seams are prominent. Length of seam plays no part as with the proper arrangement of contacts any length rod will consume itself.

Tests of Riveted Joints in Wide Plates*

The Engineering Experiment Station of the University of Illinois, Urbana, Ill., recently completed tests on joints in wide plates. This article, the fourth in a series, describes the test specimens and results of tests for determining the efficiency of net section of the 1928 series of plates. The first article in the series was published in July under the title "Efficiency of Wide Plates"; and the second and third, entitled "Determining the Strength of Net Section of Plates," appeared in August and September. Conducted by Professor Wilbur M. Wilson, James Mather and Charles O. Harris, these experiments were made possible through the cooperation of the Chicago Bridge & Iron Works, Chicago, Ill.

Of the lap joints of the 1928 series that failed by tearing of the plate, 20 in all, all but one developed an efficiency equal to the theoretical efficiency based on a diameter of deducted hole equal to the nominal diameter of the rivet plus 1/8 inch, and 13 developed the theoretical efficiency based on a diameter of deducted hole 1/8 inch greater than the nominal diameter of the rivet. Both specimens of the 1930 series containing a lap joint developed the full theoretical efficiency when determined on either basis. Apparently the eccentricity of the lap joint does not affect the ultimate strength of the plate. This is consistent with the results of tests on narrow strips. The bending of the plate due to the eccentricity of the joint was measured for specimens AL1 and AL2. For these specimens the relation between the unit tension on the net section and the bend in the plate at the top edge of the joints and at the bottom edge is given by the diagrams of Fig. 1. Each point on each curve represents the average of the bending measured at five points along the joint. These diagrams indicate that for these specimens the bending varied approximately with the stress on the net section up to 5000 pounds per square inch, and beyond this a slight increase in stress caused a more rapid increase in the bending.

The lap joints AL1 and AL2, typical for tank construction, were calked according to standard practice for a water or oil tank. The calking of the joint separated the two plates between the outside row or rivets and the calked edge, otherwise the joint would have been opened when bending in the plate began. But with contact at the edge only and with an elastic flexure in the plate from the outside row of rivets to the edge, the calked edge was able to follow up and maintain contact until considerable bending had occurred. The calked edges were coated with whitewash and observations were made at five points along each calked edge to determine when the whitewash cracked, indicating that the calked joint had opened. For specimen AL1 the whitewash first cracked at a load of 300,000 pounds. At this load it cracked at three points on one calked joint and at two points on the other. A minute crack was noted at one end of one calked joint of AL2 at a load of 200,000 pounds, but there was no further opening at this point nor any cracking at any other point along this joint at 300,000 pounds, indicating that the crack detected at 200,000 pounds was probably not due to the opening of the joint.

For the other calked joint of this specimen the whitewash cracked at three points along the edge at a load of 300,000 pounds. It would appear, therefore, that at least for these joints, a leak might be expected at a load of 300,000 pounds, corresponding to a stress on the net section of 10,920 pounds per square inch.

If rivets are staggered and the net section on a zigzag diagonal line equals the net section on a straight transverse line, the plate should, theoretically, fail along the zigzag diagonal. The net distance between rivets on the outside row for specimens BB1 and BB2, based on $\frac{13}{16}$-inch holes, is 6.29 inches on a zigzag line and 5.81 inches on a straight transverse line. The load on BB1 was released as soon as the plate started to tear, but the failure, as far as it developed, was along a transverse line. The failure of BB2 was partly along a transverse and partly along a zigzag line (see Fig 5, page 195, September issue) indicating that the stagger was such that the plate was about equally likely to fail along either of the two lines, yet the net section was 7 percent greater along the zigzag than along the transverse line.

The specimens of the 1928 series were planned to determine the relative merits of chain rivets and stagger rivets. The specimens of the S1 and S2 series are the same as those of the S5 and S6 series except that the former are chain riveted and the latter stagger riveted. The efficiency of the specimens having chain rivets, the average of six tests, was 65.51 percent; and the efficiency of those having stagger rivets, the average of four tests, was 69.94 percent. The specimens of the S3 and S4 series and of the S7 and the S8 series have stagger rivets and are the same in every

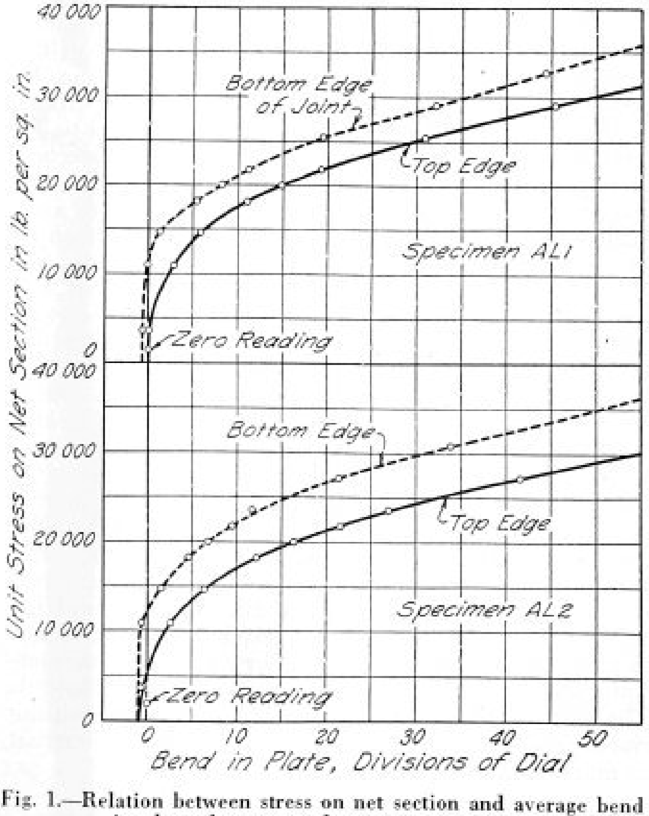

Fig. 1.—Relation between stress on net section and average bend in plate along seam, lap joint, 1930 series

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* Fourth of a series of articles published through the courtesy of the Engineering Experiment Station, University of Illinois, Urbana, Ill.

Table 1.—Comparison of Theoretical and Actual Efficiencies for Various Types of Riveted Joints

Specimens	Description of Joint	Theoretical Efficiency* Hole Deducted Nominal Diameter of Rivet Plus 1/8-inch percent	Theoretical Efficiency* Hole Deducted Nominal Diameter of Rivet Plus 1/16-inch percent	Actual Efficiency percent
1928 Series, 24 tests	Riveted lap joints. Two rows of ⅝-in. rivets. Spacing same for both rows	62.50	65.63	66.65
AL specimens of 1930 Series, 2 tests	Riveted lap joints. Four rows of ¾-in. rivets. Spacing same for all rows	72.7	74.6	78.25
AB specimens of 1930 Series, 2 tests in which specimens failed by shearing rivets*	Quadruple-riveted double-strap butt joint. ⅞-in. rivets. Spacing for outer row four times as great and for second row two times as great as for inner rows......	90.8	91.4	79.65
AB specimens of 1930 Series, 2 tests in which specimens failed by tearing plate	Quadruple-riveted double-strap butt joint. ⅞-in. rivets. Spacing for outer row four times as great and for second row two times as great as for inner rows......	91.3	91.9	86.7
BB specimens of 1930 Series, 2 tests	Triple-riveted double-strap butt joint. ¾-in. rivets. Spacing for outer row two times as great as for inner rows.........	82.8	83.85	82.30

* Holes for measuring slip deducted in computing theoretical efficiency for 1930 series. See note at bottom of Table 8. Holes for measuring slip did not affect net section for 1928 series.

way as the S5 and S6 except that the distance between the rows of rivets is 1¾ inches for S3 and S4 and 2¼ inches for S7 and S8, whereas this distance for the S5 and S6 series is 2 inches, a difference that seems to have but little effect upon the strength of the joint. The efficiency of the S3 and S4 series, the average of six tests, was 69.66 percent; the efficiency of the S7 and S8 series, the average of four* tests, was 63.66 percent. Comparing the average efficiency for the 14* specimens having stagger rivets with that for the six specimens having chain rivets, it was found that the average for the former was 68.03 percent and for the latter 65.51 percent. Thus it is seen that the efficiency of the joint was slightly greater for the stagger rivets than for the chain rivets, the difference being about 2.5 percent.

The specimens of the 1928 series were planned to determine the effect upon the efficiency of the net section of punched and drilled holes. For each series there is one group of three specimens for which the holes were drilled and a similar group of three specimens for which the holes were punched. The average efficiency for the specimens having drilled holes was 67.97 percent and for those having punched holes, 66.58 percent, the average of ten* tests for each type of specimen, a difference of 1.39 percent in favor of the drilled holes.

The practice is often followed in designing a riveted joint for wide plates having the rivet spacing greater for the outer than for the inner row of rivets, thereby increasing the theoretical efficiency of the joint. Of the joints tested in the investigation reported in this section, all of the 1928 series and AL1 and AL2 of the 1930 series had the same rivet spacing in all rows; for specimens BB1 and BB2, the spacing in the outer row was twice as great as for the inner rows; and for AB1, AB2, AB3, and AB4 the spacing was four times as great in the outer as in the inner rows. The theoretical and actual efficiencies are compared in Table 1. The data in this table apparently indicate that omitting rivets from the outer row does not increase the actual as much as it does the theoretical efficiency.

All of the specimens of the 1928 series and the AL specimens of the 1930 series were lap joints. Only four of the 24 specimens of the 1928 series failed by shearing the rivets. For these the shear, the average for the four specimens, was 40,690 pounds per square inch based on a ⅝-inch diameter of rivet, and 33,615 pounds per square inch based on a 11/16-inch diameter. For the two AL specimens the average unit shearing strength was 44,880 pounds per square inch based on a ¾-inch diameter and 38,240 pounds per square inch based on a 13/16-inch diameter. Ten rivets from these specimens were measured after the plates had been tested and the average diameter of the ten was found to be 0.794 inch. The strength developed in shear, based on this diameter, is 40,050 pounds per square inch. Twelve undriven rivets taken from the same lot as those used for the AL specimens, tested as individual rivets in single shear, developed a strength, the average for the twelve specimens, of 42,000 pounds per square inch.

The specimens AB1 and AB2 of the 1930 series were quadruple-riveted double-strap butt joints that failed by shearing the rivets. There were fifteen ⅞-inch rivets in single shear and 39 in double shear for each joint. The shearing stress developed by these rivets was 41,360 pounds per square inch based upon a diameter of ⅞ inch, and 35,940 pounds per square inch based upon a diameter of 15/16 inch. A number of rivets were measured after the plates had been tested and the diameter, the average for ten rivets, was found to be 0.930 inch. The strength developed in shear, based on this diameter, was 36,610 pounds per square inch. Twenty-four undriven rivets taken from the same lot as those used for the AB specimens were tested as individual rivets, twelve being tested in single shear and twelve in double shear. The ultimate strength in single and double shear was 41,250 and 37,750 pounds per square inch, respectively.

The width of the joint parallel to the line of stress is rather large for the AB specimens, and the rivets in the outer row might be expected to be subjected to more than their share of the stress due to the deformation of the plate. For one specimen the rivets in the outer row failed before those in the other rows, but the study of rivet slip failed to establish any consistent differences in the slip on the various rows of rivets.

These tests apparently indicate that the ultimate unit strength developed by the rivets in the large groups of these specimens, when based on the nominal diameter of the rivet, exceeds slightly the strength developed by undriven rivets tested as individuals; but when the unit strength developed by the rivets of the specimens is based upon a diameter equal to the diameter of the rivet hole, it is somewhat less than the stress developed by undriven rivets tested as individuals. The rivets of the lap joints were as strong, relative to the corresponding undriven rivets, as those in the butt joints.

Observations were made to determine the slip of the riveted joints at various loads. For specimens AL1 and AL2 the slip was measured along three transverse sections, one through each of the outer rows of rivets and one on the center of the joint. The slip is greater at each of the outer rows of rivets than it is along the middle of the joint. Slip was first detected in AL1 and AL2 at loads of 400,000 and 300,000 pounds, respectively. For the former there was a slip of 0.0004 inch at one point and for the latter there was a slip of 0.0003 inch at two points, at these loads. Slip became general along the seam for the two specimens at loads of 500,000 and 400,000 pounds, respectively. The average slip on the two outside rows of rivets was 0.0002 and 0.0005 inch for AL1 and AL2 at a load of 434,600 pounds,

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* The specimens that failed by shearing the rivets were not included in this average.

corresponding to a unit shear on ¾-inch rivets of 12,000 pounds per square inch.

For specimens AB1 and AB2 the slip was measured along four transverse sections, one through each of the outer rows of rivets and one through each of the two rows adjacent to the middle of the specimen. At the latter, the slip between the main plate and each of the butt plates was measured separately. The slip was not consistently greater on the outer than on the inner row of rivets. Slip was first detected at a load of 400,000 pounds for both AB1 and AB2.

The average slip on the two outside rows of rivets was 0.0021 and 0.0004 inch for AB1 and AB2 at a load of 671,460 pounds, corresponding to a unit shear on ⅞-inch rivets of 12,000 pounds per square inch.

For specimens BB1 and BB2 the slip was measured along four transverse sections, one through each of the outer rows of rivets and one through each of the two rows adjacent to the middle of the specimen. At the latter, the slip between the main plate and each of the butt straps was measured separately. The slip was not consistently greater on the outer than on the inner row of rivets. Slip was first detected at a load of 500,000 pounds for both BB1 and BB2.

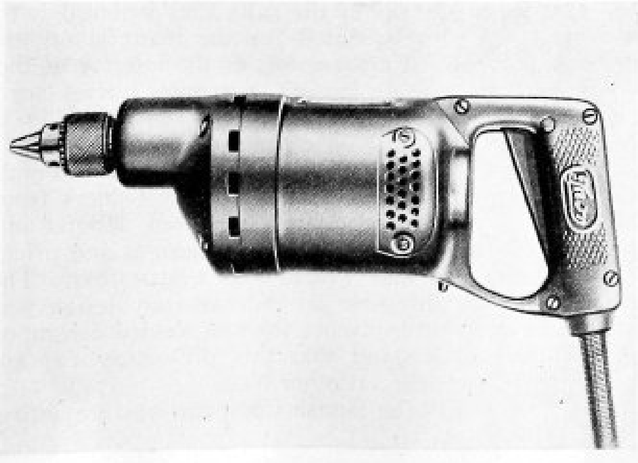
The average slip on the two outside rows of rivets was 0.0002 and 0.0002 inch for BB1 and BB2 at a load of 519,000 pounds corresponding to a unit shear on ¾-inch rivets of 12,000 pounds per square inch.

These tests apparently indicate that a measurable slip occurred at a stress considerably below that for which rivets in tanks are designed for the specimens of the 1928 series, but that for those of the 1930 series appreciable slip began at a stress about equal to the design stress. The slip for the outer row of rivets was not consistently greater than for the inner row. For all specimens of the 1930 series, the slip was smaller at the ends than at the middle of the seam although the strain in the plate was uniform across the specimen. For the 1928 series the slip was observed to be quite uniform along the seam.

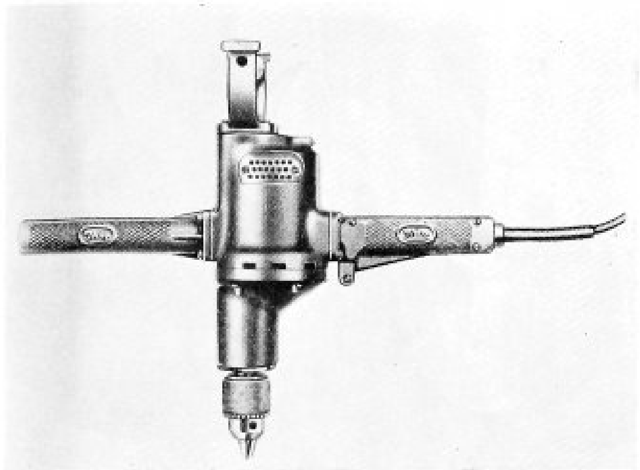
(To be continued)

Two Universal Electric Drills

The Independent Pneumatic Tool Company, 600 W. Jackson Blvd., Chicago, is now manufacturing two new electric drills—a ¼-inch size, with grip switch, known as "USA"; and a ½-inch size, with side switch, designated as "URA."



Thor USA ¼-inch capacity universal electric drill



Thor URA ½-inch capacity universal electric drill

The armature shaft of the motor is made from alloy steel with pinions milled directly on the shaft. The armature core construction is of the open straight slot type which permits the use of separate form-wound coils to be laid into the slots without first being flattened. The commutator is made in one unit, independent from the shaft, and is balanced and tested at 1100 volts before being assembled with the shaft. A brass sleeve is employed which expands and contracts in equal proportion with the copper bars, preventing an uneven surface and excessive sparking which cause melting of the solder on the lead wires.

Precision ball bearings of the closed back type are used throughout. This type of bearing prevents grease from entering into the motor because of suction of the fan and at the same time is subject to lubrication from the gear case because the open end of the bearing faces the gear case. The switch is of the double-pole type and brakes both sides of the line at the same time, thereby disconnecting the current from the line to the motor. The switch on the "USA" type is operated by a long lever and is so arranged that it can be changed to either safety or positive switch by merely pressing the lock button on the side of the handle. The field case and handle are made of aluminum in one piece, with heavy rib work on the inside, which makes the outside of the case and handle free from bolt or stud lugs. The field case, gear case and centerplate have hardened and ground steel bushings, into which the outer race of the bearings fit, preventing loose bearings. Both ends of the spindle are provided with oversize ball bearings, one fitted into the centerplate bearing, thus forming a rigid and compact assembly, which prevents vibration and insures a true running chuck.

The USA has a capacity of ¼ inch. The speed is 2500 revolutions per minute. The weight is 8 pounds. The URA has a capacity of ½ inch. The speed is 500 revolutions per minute and the weight is 14½ pounds.

Pro-Tek.—The DeVilbiss Company has developed a hand protector known as Pro-Tek, a white cream which, when rubbed into the skin before working, forms a protective film, soluble only in water. Sticky substances and soiling materials are powerless to penetrate. It is described as the invisible glove. When Pro-Tek is used, lacquer, paint, grease, metallic dusts and similar materials can be quickly and completely removed by washing the skin under running water.

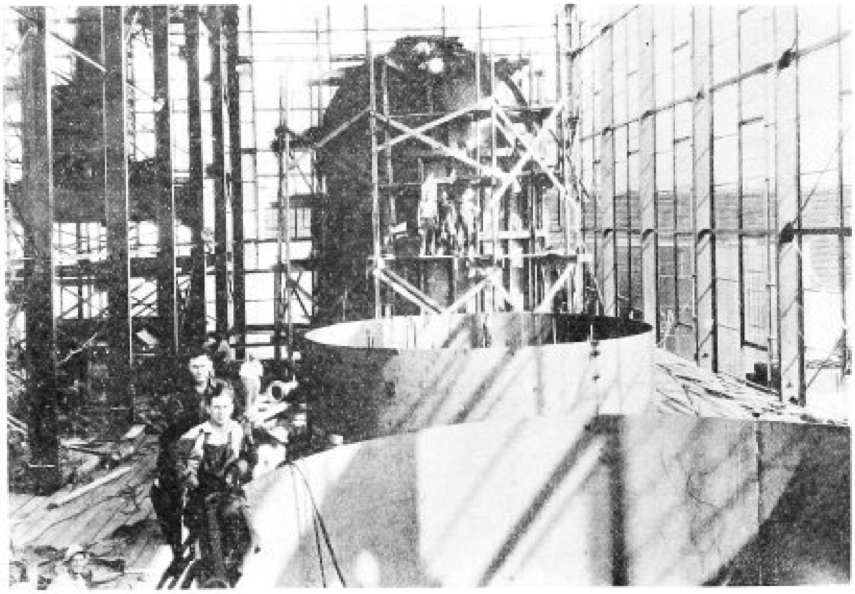


Fig. 1.—View showing third evaporator assembled and being welded. The lower sheets are set on the first and second evaporators

Welded Evaporators Designed for Unusual Service[*]

By W. H. McClara[†]

Three large welded evaporators have recently been designed, erected and tested for a potash company in Southern California. The first and second effect evaporators are each 22 feet in diameter by 55 feet high, and the third effect evaporator is 22 feet in diameter by 49 feet high. The evaporators were designed for an internal steam pressure of 10 pounds per square inch, and, in addition, for a 75-foot head of brine weighing 87 pounds per cubic foot or an equivalent pressure of 45 pounds per square inch at the bottom of the cone.

All three evaporators were subjected to a hydrostatic test of 52 pounds per square inch internal pressure at the bottom of the cone. In addition, each evaporator was subjected to a vacuum test of 28 inches of mercury, which equals 13.7 pounds per square inch in absolute external pressure, and this pressure was required to be maintained with only a drop of 1 inch of mercury per hour. This specification requirement was made without difficulty, as no measurable drop in vacuum occurred in 12 hours. The test of 28 inches vacuum was made when the barometer reading was 28.3 inches. No leaks occurred through the welds.

In the past, evaporators of this capacity have been made of cast iron, in sections of a size permitted by foundry practice, the joints being machined and bolted together with gaskets or other joint material.

These cast-iron evaporators, designed for low pressures and high vacuum, have developed defects that shorten their useful life as units, which by virtue of the durable material used, would be expected to have long life. Some of these defects are quite serious and require an evaporator to be shut down for repairs. It is reported that the Potash Company loses nearly a thousand dollars in potash production each day that one of these units is disabled. For instance, an operator's mistake subjected one of these cast-iron evaporators to about 30 pounds per square inch internal pressure, which caused it to leak in perhaps a hundred places. After this treatment it is difficult to maintain a vacuum. The remedy is to close down the unit, clean it out, cool it, strip the heavy insulation off the outside, repair it, test it and replace the insulation.

At other times, large pieces of salt falling to the bottom of the cone from the sides of the unit, where it collects, crack the castings forming the cone. This also stops production, as the unit must be immediately shut down and castings replaced or repaired. At this point it might be stated that this condition has been remedied by installing welded steel cones inside the cast-iron cones. These welded steel cones are made of ¾-inch flange steel and stand up under the impact of the falling boulders of salt.

As the Potash Company had found that steel was not corroded by the action of the salts they believed it to be a material entirely suited for the manufacture of these evaporators. Furthermore, as the interior of the vessel must be entirely smooth, not even a rivet head being permitted inside, welding was desirable as a method of joining all the parts into one unit.

The Potash Company, therefore, secured bids on both the cast iron and welded designs. Many bidders from all parts of the United States submitted designs and prices. The writer's firm submitted designs and prices on both cast-iron and welded-steel evaporators. The price put in by this firm on the cast-iron design was slightly over 10 percent more than on welded design, or $6300. We understand that this difference was approximately the same on other bids.

The first cost to the Potash Company was not entire-

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[*] Abstract of a paper submitted for the Lincoln Second Arc Welding Prize Competition sponsored by the Lincoln Electric Company, Cleveland, O.

[†] Designer, the Consolidated Steel Corporation, Ltd., Los Angeles, Cal.

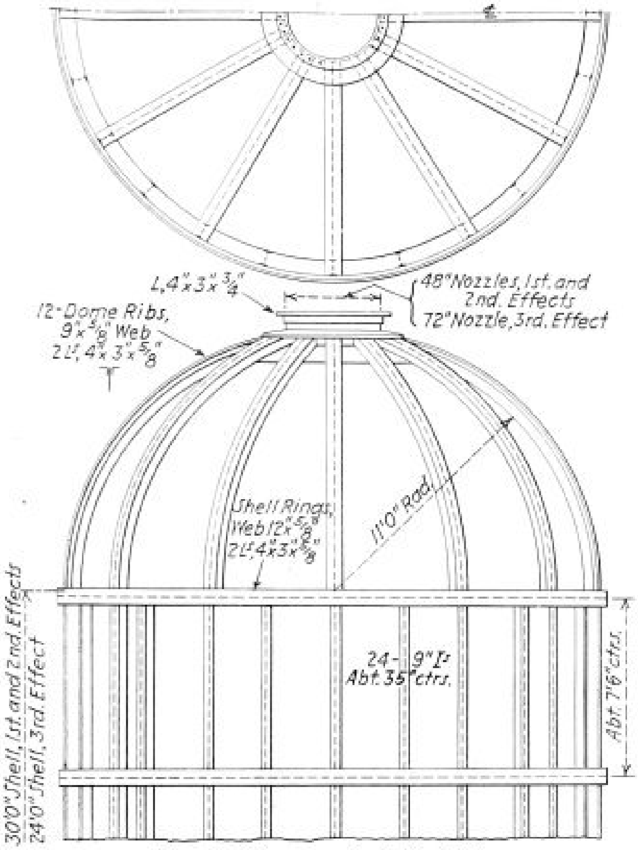


Fig. 2.—Dome and shell detail

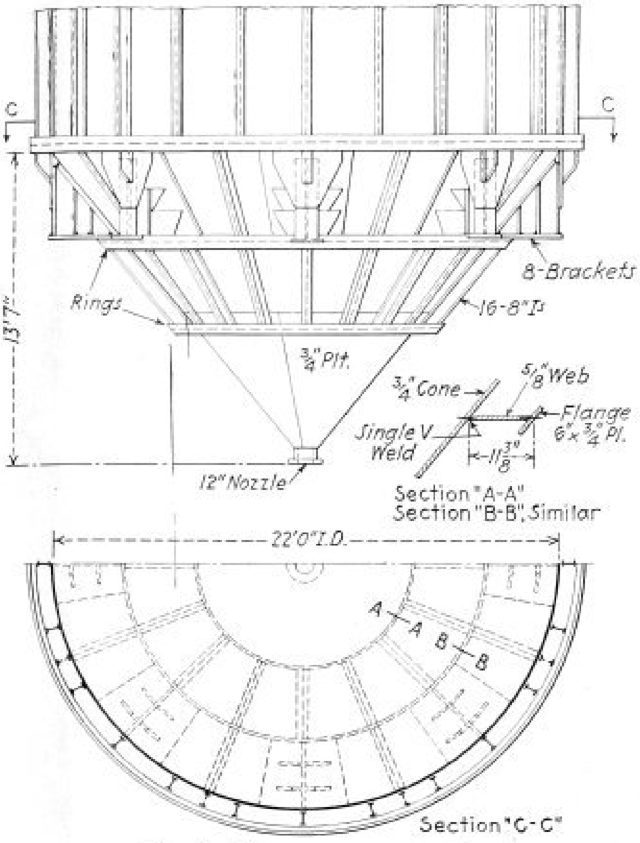
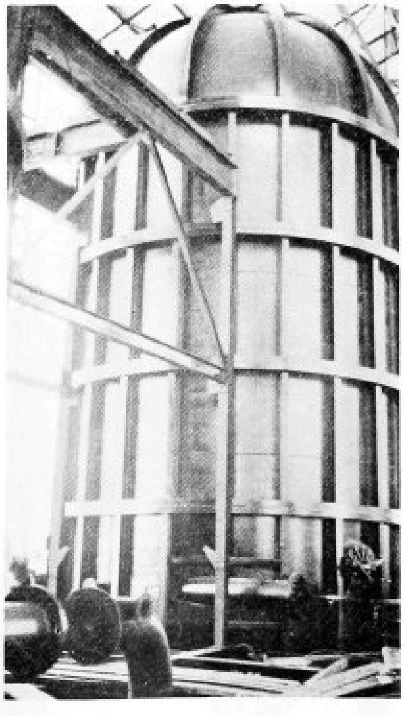


Fig. 3.—Bottom cone construction

ly the primary consideration. The main feature of the welded evaporators is the economy when measured by daily service; that is, producing tons of potash every day in the year, year after year, without a shut-down. The Potash Company will save the expensive repairs which have been necessary on the cast-iron evaporators. They will also avoid the losses in production by cast-iron evaporators being out of commission.

Welded steel design, rather than riveted steel design, was used because of the desirability of not having any lap seams, butt straps or rivet heads projecting into the shell, and also because of the necessity of transmitting heat quickly through the shell and into the supporting ribs and ring girders when the evaporators were quickly cooled. The specifications required that the evaporators be held at a temperature of 238 degrees F. for 24



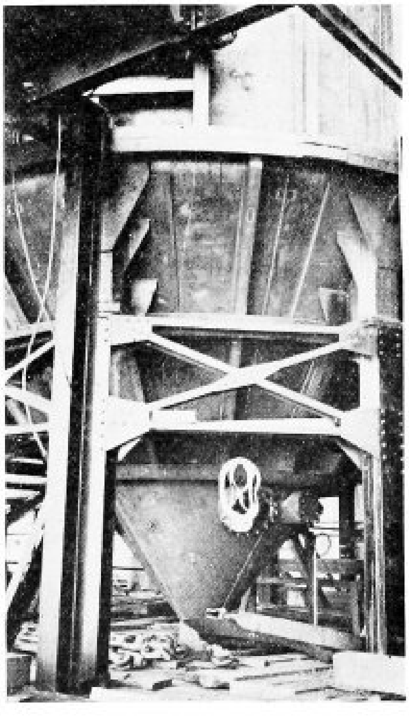


Fig. 4, at left, shows the assembled dome and shell, while Fig. 5, at right, is a view of the bottom cone construction hours and then suddenly subjected to a change in temperature when they are washed out with wash water at 140 degrees F. Considerable stress is occasioned by this temperature change and it is necessary to have the supporting members become as nearly an integral part of the shell itself as possible, so that heat may be as quickly as possible transmitted through the whole. This could better be accomplished by welding than by any other known means. It was felt that the transmission of heat through parts welded in this manner would be much quicker than through parts which might be riveted together. The particular parts affected are the ring girders which, being the supporting members with the greatest section, would naturally be affected most by this heat change.

Due to the vacuum loads to which these evaporators are subjected, it is of utmost importance that they be kept round, and during erection a tolerance of ½-inch variation in radius over an arc of 8 feet was allowed. The results, as stated, show that no two diameters could differ more than two inches. Great care was necessary to accomplish this result and it is felt that it was much more easily done with welded construction than could have been accomplished by any other type of construction.

Another item in the welded evaporator's favor is the ability to change location of sight glass flanges, manholes, pipe nozzles, etc., almost up to the date of completion of erection. On cast-iron evaporators these fittings must be located before the patterns are made and can be changed only at the expense of new patterns and castings.

The following is a brief outline of the specifications as applied to the first effect evaporator. The specifications for the second and third are similar.

Inside diameter	22 feet
Height of cone	13 feet
Height of shell	30 feet
Height of dome	11 feet
Total volume	15,846 cubic feet
Steam pressure	10 pounds per square inch
Vacuum, 28 inches of Mercury	13.7 pounds per square inch, external
Maximum temperature	238 degrees F.
Weight of brine	87 pounds per cubic foot
Weight of wash water	69 pounds per cubic foot
Temperature of wash water	140 degrees F.
Steel	Flange quality
Maximum weight of brine	15,846 × 87 pounds = 1,378,600 pounds

The maximum internal ring stress at top of cone was computed as follows:

$$\frac{RP}{t} = \frac{132 \times 36}{.75} = \frac{4752}{.75} = 6336 \text{ pounds per square inch}$$

where, t = ¾-inch plate or .75 inch
P = Pressure, pounds per square inch
R = Radius = 11 feet or 132 inches.

Fig. 2 shows a dome and shell detail. The external atmospheric pressure of about 2000 pounds per square foot is carried by the structural ribs and rings on the outside of the ¾-inch plate shell.

The main shell rings are designed to safely carry their load when elliptical to the extent that the major axis exceeds the minor axis by 6 inches. The contract specified that this difference be limited to 2 inches. When erected this difference was less than 1 inch. Each 9-inch beam carries the external load over the area of 35 inches by 90 inches, should a flat spot be left in erection. The 9-inch beams also stiffen the shell in a vertical direction when subjected to the balanced atmospheric pressure on dome and cone.

The shell is three courses high. Each course is composed of three plates 120 inches wide, ¾ inch thick and 23 feet long. After planing the bevel edges on the plates, they were accurately rolled and shipped standing on their long edge to prevent damage in handling.

The main rings were each shipped in three pieces and field-welded to the shell plates and to each other.

The weld joining the ring girder to the shell is adapted to fast work, as all welding is done from above. It is also well adapted to the transmission of heat from shell to ring when the shell is the hotter. Heat is readily transmitted from ring to shell when the ring is the hotter, a condition which occurs when relatively cool wash water is suddenly pumped into the evaporator.

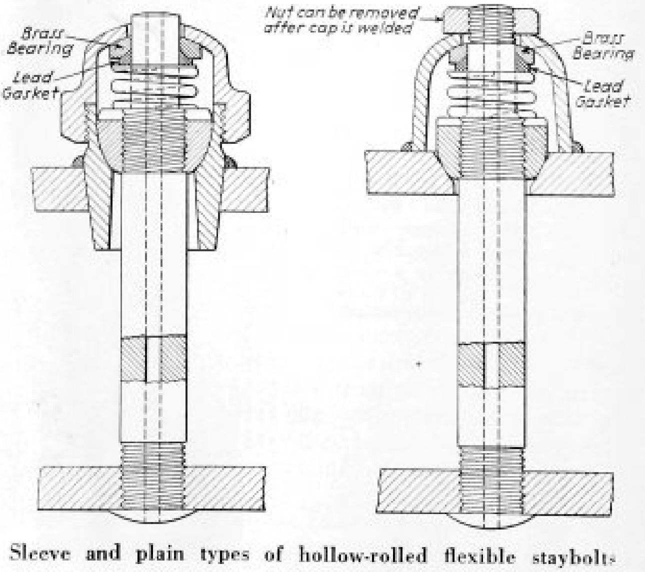
This temperature difference was considerably affected by the type of insulation used on the outside. Many experiments were made to check the theoretical calculations and to determine what difference existed across the girder and its weld to shell.

Fig. 3 shows the bottom cone construction. The cone was shop-assembled completely upside down where most of the welding could be done from above. It was then match marked and shipped in the largest sections possible. Eight brackets were on each cone. It will be noted that while the normal load carried is 54 tons, each bracket could be subjected to a 100-ton load by filling the evaporator entirely full of brine.

Hollow-Rolled Flexible Staybolts

About two years ago the Old Dominion Iron & Steel Works, Inc., Belle Isle, Richmond, Va., developed and patented a new type of hollow-rolled flexible staybolt. This device, worked out at the suggestion of the chief mechanical officer of one of the principal railroads, has been used in a number of successful service tests, resulting in several separate installations.

As will be seen from the illustrations, these flexible staybolts are constructed with a telltale hole extending through the entire length of the bolt. Patents cover



Sleeve and plain types of hollow-rolled flexible staybolts both plain and sleeve types, as shown in the illustration, and the bolts are adaptable where other styles of flexible bolts are in use. It will be seen that the hollow staybolt material is merely threaded at both ends with the bearing pieces, springs and caps attached at the outer end.

The principal advantages claimed by the Old Dominion Iron & Steel Works, Inc., for these staybolts, when made from hollow-rolled staybolt bars manufactured by that company, are:

1. The staybolt material is purchased to rigid specifications both chemically and physically. If, therefore, the staybolt bars are subjected to drilling and heating to forge the heads, the material is changed from the original specifications, and the value of the specifications is lost. When hollow-rolled bars are used, however, and the heads of the bolts are formed by threading, there is no change in the material after inspection has been made to the original specifications.

2. As the hole extends through the entire length of the bolt, inspection may be made from the outside, in conformity with Interstate Commerce Commission Rules, thus eliminating the expense of removing caps and the use of electrical devices for inspecting telltale holes.

According to the Old Dominion Iron & Steel Works, these flexible staybolts can be furnished by any of the established manufacturers of flexible staybolts. When the Old Dominion hollow-rolled staybolt bars in wrought iron or electric steel are specified, no charge is made for the patent rights.

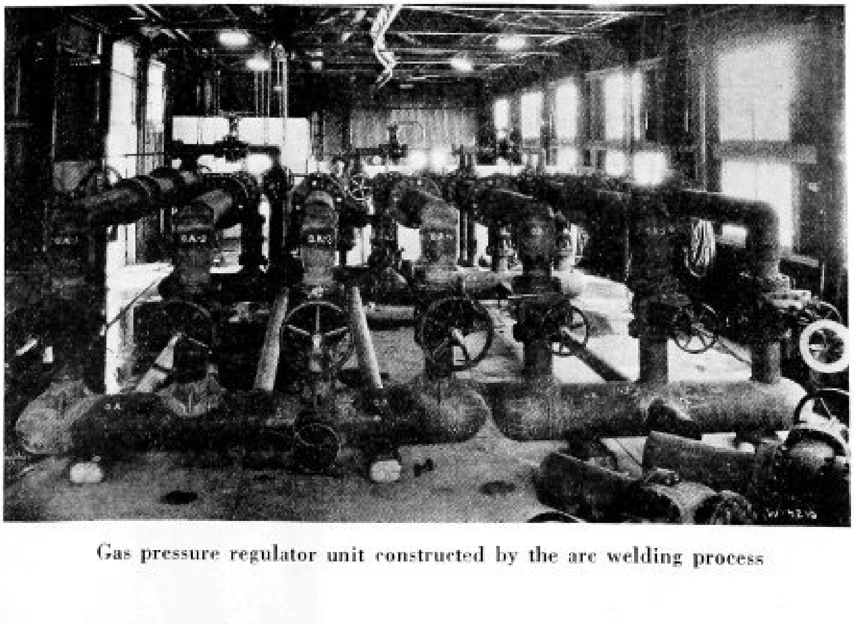
Arc Welding Gas Pressure Regulators

A very considerable reduction in construction and assembly costs of a gas pressure regulator unit has been made possible by the utilization of the shielded-arc process of arc welding which has made such rapid strides recently.

The unit in question is installed in the Muncie, Indiana, station of the Ohio Fuel Gas Company to reduce the pressure on trunk lines for city use. It consists of six high-pressure regulators and 12 gate valves; in the meter end there are seven merco stops and seven gate valves as well as seven 10-inch all-welded orifice flanges. On the whole installation, shown in the illustration, are 120 all-welded steel flanges. It weighs 40,000 pounds.

Adaptability of the shielded arc technique is shown by the variety of pipe sizes. The regulator consists of 45 feet of 16-inch pipe; 22 feet of 12-inch pipe; 159 feet of 10-inch pipe; 40 feet of 8-inch pipe; 80 feet of 6-inch pipe; 71 feet of 3-inch pipe; 20 feet of 2½-inch pipe.

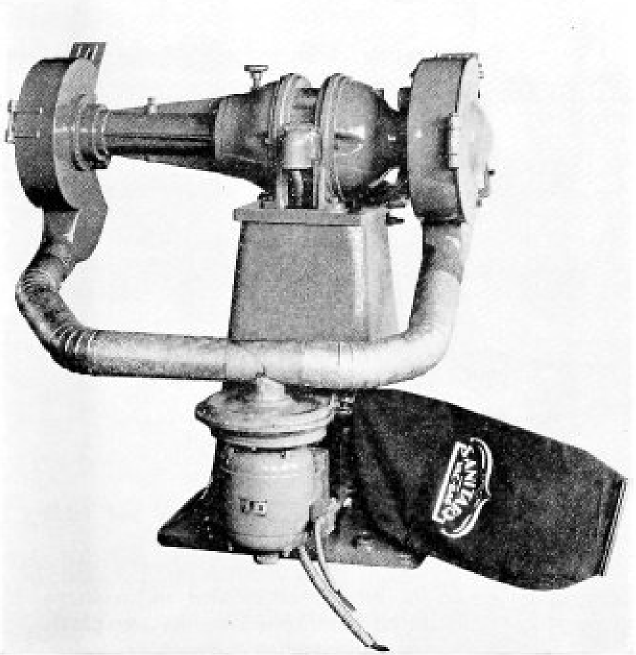
The Ohio Fuel Gas Company is an extensive user of welding and is equipped with standardized arc-welded jigs and fixtures for production of fittings. Flanges are cut by gas on an arc-welded automatic set-up and held in position on a special fixture. All pressure welding is done with the shielded-arc Fleetweld electrode and Stable Arc welders of The Lincoln Electric Company, Cleveland.



Gas pressure regulator unit constructed by the arc welding process

Combination Grinding and Buffing Machine

The Standard Electrical Tool Company, Cincinnati, Ohio, has announced a new motor-driven exhaust unit which may be mounted on its heavy-duty grinders and buffers. The illustration shows the combination grinding and buffing machine equipped with a 3-horsepower General Electric motor with automatic starter and push-button station. Both the grinder and buffer sides of the machine are fitted with enclosed hinged door guards.



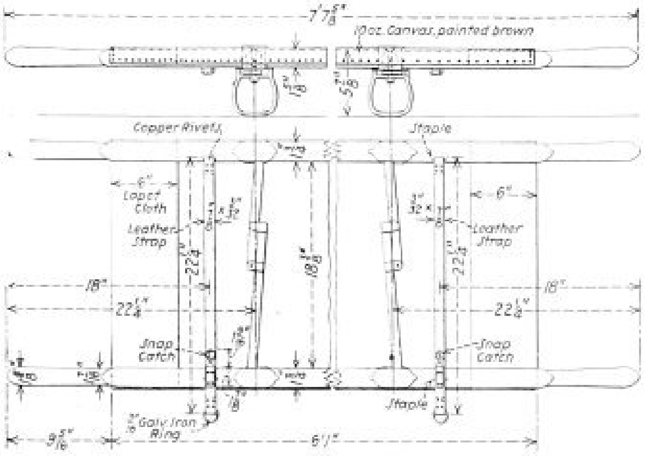
Exhaust unit mounted on Standard electric grinder

On the back of the pedestal is mounted a ¾-horsepower, 3600-revolution per minute, motor-driven exhaust unit fitted with a dust collecting bag. The net weight of this machine is 846 pounds.

This attachment when mounted on grinding and buffing machines prevents the accumulation of dust and small fragments of metal which collect around the usual grinder or buffer.

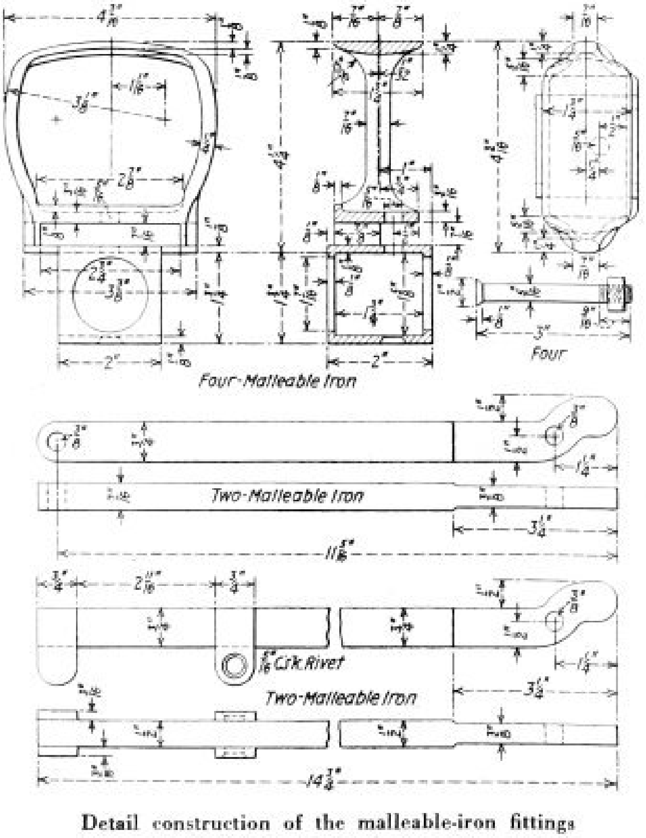
Folding Stretcher for First-Aid Departments

Accidents will happen in the best regulated shops. For that reason the first-aid or safety departments of a well equipped shop usually have folding stretchers located at strategic points around a shop or repair yard for the handling of those so injured that they are unable to walk. The folding stretcher shown in the two drawings can be made in the carpenter and smith shops, if forgings are substituted for the malleable-iron parts.



Assembly drawing showing the construction and folding arrangement of the stretcher

The wood side rails and handles are of seasoned white ash or hickory. Ten-ounce canvas duck, painted brown, is tacked to the side rails with upholstery nails. Leather shoulder straps, 3/32-inch by ⅞-inch, are provided at both ends, as shown. The stretcher is 6 feet 1 inch long by 18¾ inches wide. The length overall is 7 feet 7⅝ inches. The height from the floor is 5⅞ inches.

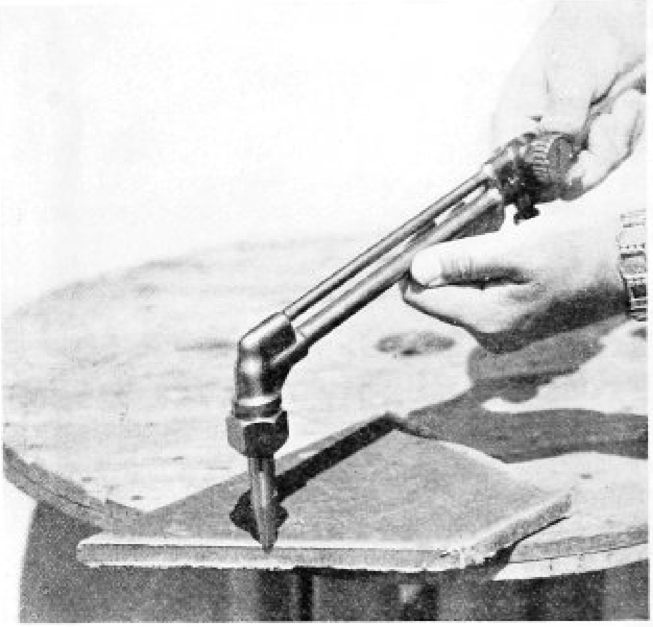


Detail construction of the malleable-iron fittings

Cutting Attachment for Welding Torch

The Smith Welding Equipment Corporation, 2619-33 Fourth Street, S. E., Minneapolis, Minn., has placed upon the market what is known as the "Money-Saver" cutting assembly, or attachment, for a welding torch. To maintain the efficiency of the cutting assembly and yet to manufacture it at a considerably less cost was the problem involved.

In designing this cutting assembly the engineering department eliminated every customary feature, usually considered necessary in the manufacture of such apparatus, such as levers for the cutting oxygen, adjustable unions to change the position of the assembly in the handle, etc., and substituted for these features valves of the greatest simplicity.



Money Saver cutting assembly

The cutting oxygen valve is a triple-thread, screw valve with knurled thumb wheel as shown, so arranged that it can be opened quickly and closed with the thumb. The valve forging has been reduced to the simplest possible design, the cutting tip automatically evens up the flames. Double seats and all other parts, not absolutely essential to operation have been eliminated, resulting in simplicity of construction and manufacturing costs far less than the conventional type of cutting assembly. The new assembly has been approved by the Underwriters' Laboratories.

Addenda to Boiler Code.—The American Society of Mechanical Engineers has issued a number of addendas to the A.S.M.E. Boiler Construction Code as approved August 17, 1932. These addendas refer to sections on miniature boilers, locomotive boilers, care of power boilers, heating boilers, material specifications, power boilers and unfired pressure vessels. In all, 65 pages have been issued at this time.

The Boiler Maker

Volume XXXII Number 10

Published Monthly by the Simmons-Boardman Publishing Company, 34 North Crystal Street, East Stroudsburg, Pa., with Executive Offices at 30 Church Street, New York City.

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Contents

Communications

Layout of Tapered Pipe Connection

To the Editor:

In the Questions and Answers Department of the September issue of The Boiler Maker, P. J. T. requested the layout of a tapered pipe connection. After reading the explanation of the development, I think that the pattern, Fig. 7, has not been properly developed.

The distances between the points *1′, 2′, 3′* to *9′*, Fig. 7, were made equal to one of the equal parts into which the semicircle *R-T-S*, Fig. 3, was divided. In other words, the length of the edge of the pattern measured along the miter line was made equal to the circumference of the circle *B-R-T-S*. This should not have been done for the reason that the miter line follows the curvature of the horizontal cylinder and the distance between the points *1′, 2′, 3′* to *9′*, Fig. 7, are not equal.

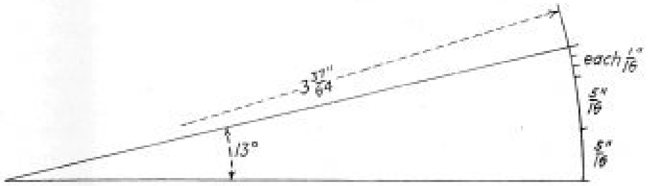
Before attempting to lay out the pattern, the hole in the cylinder would have to be developed in order to obtain the distances between the points *1′, 2′, 3′* to *9′* and consequently the length of the miter line.

Muskegon, Mich. Sidney Cheasley.

Simple Measurement of an Angle

To the Editor:

It often happens in the boiler shop that it is necessary to measure an angle without the aid of a protractor, or it is necessary to layout an angle with no other implements than the scale and the compasses. A simple means of measuring an angle is to draw an arc similar to that shown in the illustration with a radius of 3 37/64 inches. With the arc so drawn, measure the length of the arc between the two lengths of the angle. Each 64th of an inch will then equal one degree.



Simple method of measuring an angle

In measuring the arc, the distance between the points of the compasses should not be greater than five or six-sixteenths of an inch. This precaution is necessary to prevent great errors due to the fact that the chord subtending the arc is less in length than the arc itself.

Taking the radius of 3 37/64 inches, the length of a semicircle is found by calculation to be 11.241 inches. Multiplying 180 by 1/16 inch gives 11.250 inches. The difference is 0.009 inch, which can be neglected. The thickness of a line drawn by a pencil would be greater than this error.

For greater accuracy in measuring an angle, take a radius of 7 5/32 inches, which is double the radius of 3 37/64 inches. In this case 1/16 of an inch as measured on the arc would equal half a degree.

Graz, Austria. Johann Jaschke.

Coal Dust and Oil as Ship Fuel

A combination of fuel oil and coal dust for firing boilers is being tried out on the Cunard liner *Scythia*. With the crude oil is used coal pulverized so fine that the mixture runs freely through the jets under the boilers.

The present experiment with colloidal or "coal-loaded" fuels calls to mind similar tests made in 1919 by the late L. W. Bates, a consulting engineer. It was reported at that time that the test showed that an efficient and economical substitute had been found for oil and coal as steamship fuel, but shortly thereafter fuel oil became so cheap that no general use of "coal-loaded" oil was undertaken.—*The Locomotive.*

Questions and Answers Pertaining to Boilers

By George M. Davies

This department is maintained for the purpose of helping those who desire assistance on practical boiler shop problems. Inquiries should bear the name and address of the writer. Anonymous communications will not be considered. The identity of the writer, however, will not be disclosed unless the editor is given special permission to do so.

Pitch and Size of Rivets

Q.—Will you show how to solve for the proper pitch and size of rivets on a single, double and triple-riveted lap seam, and on a double, triple and quadruple butt seam?—W. M. G.

A.—In determining the proper pitch and size of rivets for a single, double or triple-riveted seam, the first thing to consider is the rivet diameter. Rivets may fail in either of two ways, namely, by shearing or crushing. For each thickness of plate there is a critical diameter of rivet equally strong for shearing and crushing. Rivets smaller than the critical diameter fail by shearing and those larger by crushing.

Table 1 gives the critical rivet diameters for various thicknesses of plate.

Table 1.—Critical Rivet Diameters

Plate Thickness Inches	Steel Rivet Diameters Single Shear Inches	Steel Rivet Diameters Double Shear Inches	Iron Rivet Diameters Single Shear Inches	Iron Rivet Diameters Double Shear Inches
1/4	0.679	0.340	0.804	0.402
5/16	0.849	0.424	1.005	0.503
3/8	1.019	0.509	1.206	0.603
7/16	1.188	0.594	1.407	0.704
1/2	1.358	0.679	1.608	0.804
9/16	1.528	0.764	1.809	0.905
5/8	1.698	0.849	2.010	1.005
11/16	1.867	0.934	2.211	1.106
3/4	2.037	1.019	2.412	1.206
13/16	2.207	1.104	2.613	1.307
7/8	2.377	1.188	2.815	1.407
15/16	2.547	1.274	3.016	1.508
1	2.716	1.358	3.217	1.608

The single riveted lap seam as shown in Fig. 1 is the simplest type of seam. A unit section of the seam would be between the lines a and b.

With adequate laps the tearing of the plate and the failure of the rivets need only be considered in this type of joint. The plates joined are supposed to be of the same thickness. Whenever one plate is thinner than the other, the joint should be designed as if for the thinner plate.

The resistance to failure for section a-b would be as follows:

(1) Resistance to tearing plate between holes = $(p-d)\ t \times TS$

(2) Resistance to shearing one rivet = $\dfrac{\pi d^2}{4} \times S$

(3) Resistance to crushing one rivet = dtc

where p = pitch of rivets, inches
d = diameter of rivets, inches
t = thickness of plate, inches
TS = tensile strength of plate, pounds per square inch
s = shearing strength of rivet in single shear, pounds per square inch
c = crushing strength of rivet, pounds per square inch.

These three equations comprise the only ways in which this joint can fail. To make a perfectly designed seam the resistances to failure should all be equal.

The seam would then have an exact balance of strength in all its features. Therefore the above three equations should be solved simultaneously for values of p and d.

Equating (2) and (3)

$$\frac{\pi d^2}{4} \times S = dtc$$

$$d = \frac{4tc}{\pi S}$$

This is the critical diameter of the rivet equally liable to fail by crushing or shearing; and Table 1 gives a series of numerical values for this expression. Commercial rivet diameters vary by sixteenths of an inch, hence none of those shown in Table 1 are readily procurable.

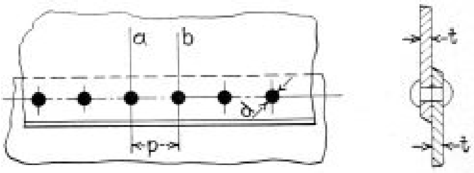


Fig. 1.—Single-riveted lap seam

If the above theoretical rivet diameter could be used, (1) equated to (2) or (3) would yield a value for the pitch, and the latter would be the same from whichever equation it was derived. Imagine such a rivet to be procurable and to be used in the seam at hand. Equating (1) and (2),

$$(p\text{-}d)t \times TS = \frac{\pi d^2 \times s}{4}$$

$$p = d + \frac{\pi d^2 \times s}{4t \times TS}$$

and equating (1) and (3),

$$(p\text{-}d)t \times TS = dtc$$

$$p = d\left(1 + \frac{c}{TS}\right)$$

A numerical problem will best illustrate the above fact.

Problem: Find the pitch value for a single-riveted lap joint using steel rivets of the critical diameter in 3/8-inch plate,

when TS = 55,000 pounds per square inch
c = 96,000 pounds per square inch
s = 45,000 pounds per square inch.

From Table 1 the critical diameter of rivet for 3/8-inch plate is 1.019 inches. Substituting in the above equations we have,

$$p = 1.019 + \frac{3.1416 \times 1.019 \times 45{,}000}{4 \times 0.375 \times 55{,}000}$$

$$p = 2.798 \text{ inches}$$

$$\text{and } p = 1.019\left(1 + \frac{96{,}000}{55{,}000}\right)$$

$$p = 2.798 \text{ inches}$$

Thus $p = 2.798$ inches in both equations.

Substituting these values in the original equations (1) (2) and (3) we have

(1) Tearing $= (p\text{-}d)\ t \times TS$
$= (2.798 - 1.019)\ 0.375 \times 55{,}000 = 36{,}690$ pounds

(2) $\text{Shearing} = \dfrac{\pi\ d^2}{4} \times S$

$$= \frac{3.1416 \times 1.019^2}{4} \times 45{,}000 = 36{,}690 \text{ pounds}$$

(3) Crushing $= dtc$
$1.019 \times 3.75 \times 96{,}000 = 36{,}690$ pounds

It is evident that a joint so designed is perfectly balanced in all its proportions, has no weak spots, and is as liable to failure in one way as another. Evidently the above procedure is purely theoretical. With commercial rivet sizes it is impossible to balance all the resistance to failure even in a joint as simple as the one under consideration. With a rivet other than the critical size, the joint will no longer be equally strong in all ways. Thus, suppose a rivet 1 inch in diameter had been chosen. By reference to Table 1 it is seen that this rivet would fail by shearing.

Equating (1) and (3) as before,

$$(p\text{-}d)t \times TS = dtc$$

$$p = d\left(1 + \frac{c}{TS}\right)$$

Substituting the values,

$p = 1$ inch
$c = 96{,}000$ pounds per square inch
$TS = 55{,}000$ pounds per square inch
$S = 45{,}000$ pounds per square inch

$$\text{we have, } p = 1\left(1 + \frac{96{,}000}{55{,}000}\right)$$

$$p = 2.763 \text{ inches.}$$

The three resistances to failure will then yield numerical values as follows:

(1) Tearing $= (p\text{-}d)t \times TS = 36{,}000$ pounds

(2) $\text{Shearing} = \dfrac{\pi\ d^2}{4} \times S = 35{,}340$ pounds

(3) Crushing $= dtc = 36{,}000$ pounds.

The resistance against tearing and crushing are balanced since they were equated in determining the pitch, while that of shearing shows a lack of strength due to the small rivet chosen. Had a 1⅛-inch rivet been chosen the maner of rivet failure would have been by crushing.

The simple principle illustrated above of throwing the two features of joint strength into balance, while all others represent excess or lack of strength, forms the basis of the design of riveted joints.

In summing up the foregoing, it may be said that the general procedure in all joints is as follows:

(1) Write all possible methods by which the joint may fail.
(2) Select a commercial size of rivet, based on Table 1.
(3) By reference to Table 1 determine the probable method of failure of the rivet chosen.
(4) Equate the rivet failure to the plate failure and solve for the pitch.

Removable Head of Pressure Tank

In the June issue under the question "Removable Head of Pressure Tank," formulas were given for determining the size and thickness of a ring for a removable head on a pressure vessel. It has been brought to my attention that all the possible ways in which the ring may fail are not covered in this answer.

The ring could also fail in the manner illustrated in Fig. 1. In this mode of failure, which is a common one, cracks start at the free edge of the hub of the shell ring and progress into the metal as indicated at *ccc*, sometimes passing into the rivet holes and sometimes avoiding them. This is due to a circumferential tension in the hub of the shell ring, which is greatest at the free edge of the hub; and it is this circumferential tension which gives rise to the cracks illustrated in Fig. 1.

For the sake of simplicity in making references, consider that the shell has been turned up on end, so that the end to which the ring under discussion is attached is uppermost. Adapt the terminology shown in Fig. 2 for naming the parts of the ring; that part that is riveted directly to the shell being called the "hub" while the projecting flange will be called the "leaf" of the ring.

The internal stresses, within the material of the ring are due, of course, to the external forces to which the ring is subjected. The steam pressure, acting upon the interior of the vessel after the manner indicated in Fig. 3, produces a longitudinal stress s in the shell, and this tension is thrown upon the ring, through the rivets by which the shell is attached to the ring. The total pull of the shell upon the ring (which will be denoted by the letter L in the formula) is found by multiplying the area of the circle, whose radius is R in Fig. 3, by the internal pressure, as expressed in pounds per square inch. The pressure radially upon the upper part

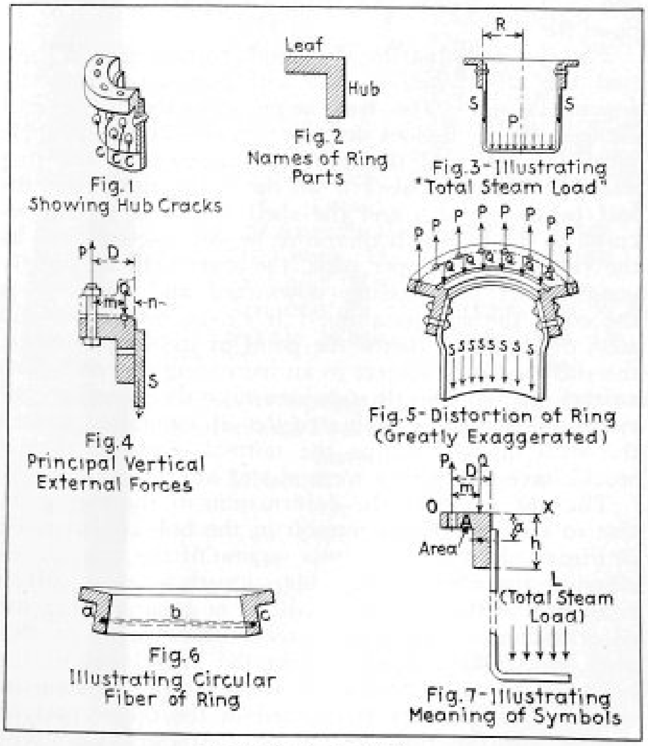


Fig.1 Showing Hub Cracks

Fig.2 Names of Ring Parts

Fig.3-Illustrating "Total Steam Load"

Fig.4 Principal Vertical External Forces

Fig.5-Distortion of Ring (Greatly Exaggerated)

Fig.6 Illustrating Circular Fiber of Ring

Fig.7-Illustrating Meaning of Symbols

Methods of ring failure

of the ring, when the ring is not covered by the shell, undoubtedly gives rise to a corresponding stress in the ring; but the stress due to this cause is small and does not need to be taken into account.

The radial pressure of the steam against that part of the shell which overlaps the ring is probably carried, for the most part at least, by the shell itself, without material direct influence upon the ring. The pressure of the steam upon the circular space on the top face of the ring, which lies just within the packing circle, likewise gives rise to a stress within the ring; but this also is small and need not be taken into account.

In addition to the direct steam pressure just considered, the ring is subject to two other sources of stress, which are indicated in Fig. 4 by the forces P and Q, P being the total tension upon one cover-bolt, and Q being the downward pressure of the cover upon as much of the packing circle (measured around the circumference of this circle) as corresponds to one of the cover bolts.

Under the influence of these various forces, the ring becomes deformed in the manner indicated on a grossly exaggerated scale in Fig. 5. The actual deformity of the ring is so small that it is practically impossible to demonstrate any deformation by direct observation of the ring itself; yet, small as it is, it is undoubtedly as real as though the ring were composed of rubber. In plastically elastic materials such as rubber, a large deformation is accomplished by the development of only a comparatively small amount of internal stress, but in a material such as iron or steel, the development of an exceedingly small deformation corresponds to the existence of internal stresses that are enormous when compared with those in rubber. Hence the deformation of the ring we are considering is of the greatest importance, even though it is exceedingly small.

The deformation of the ring as illustrated in Fig. 5 may be described as follows. The cover bolt tension P, acting upon the leaf of the ring near its outer edge, tends to raise this outer edge; and the downward pressure on the packing circle tends to depress the inner edge. This is also true of the longitudinal tension s upon the shell.

The result is that the ring tends to take such a form that the upper face of the leaf assumes a conically concave shape. The ring being supposed to be stiff enough so that it does not bend materially at the angle when the leaf and the hub join, the hub of the ring must be thrown outward at the same time that the leaf bends upward, and the shell, which is rigidly secured to the hub, will therefore be deformed as well as the ring. At its upper edge, the sheet may be slightly compressed; but passing downward and away from the edge, the compression, if it exists, diminishes and soon disappears. Below the point of its disappearance, the shell becomes subject to an increasing stretch, which reaches its maximum somewhere near the lower row of rivets. After passing this region of maximum stretch the shell quickly attains the normal diameter that it would have if the ring were absent altogether.

The way in which the deformation of the ring gives rise to circumferential tension in the hub is illustrated in Fig. 6 which shows a cross section of the ring. Considering any fiber of the material such as a-b-c, which runs around the ring circularly, it is apparent that the deformation of the ring increases the length of this fiber, if the fiber is situated in the lower part of the hub, and so must necessarily throw it into a state of tension. If the fiber is situated in the upper part of the leaf of the ring, it will be shortened instead of stretched and it will therefore be subjected to compressive stress. The entire ring (except along a single imaginary horizontal surface which separates the parts in tension from those in compression and which therefore corresponds to the "neutral surface" or "neutral axis" of a beam) will be in a state of circumferential tension or compression, the lower part of the ring being in tension and the upper part in compression.

The formula for finding the maximum circumferential tension in the ring (this maximum tension coming at the bottom of the hub) is:

$$F = \frac{(mNE + LD)\ (h\text{-}a)}{6.2832\ (I\text{-}a^2 \times A)}$$

The significance of most of the letters will be understood from Fig. 7.

A = Area of cross section of the ring in square inches.

a = Distance in inches, from the face of the leaf to the center of gravity of the cross section.

I = Moment of inertia of the cross section with respect to the line $O = X$.

h = Total length of ring, in inches, from face of the leaf to the extremity of the hub.

D = Distance, in inches, from the inner face of the ring to the cover bolt circle.

L = Total steam load on the head, in pounds.

N = Number of cover bolts.

m = Distance in inches, from the middle of the packing to the cover bolt circle.

E = Excess of actual tension on each cover bolt, over and above that which would be just sufficient to sustain the steam load on the head.

A, a and I are to be computed for the smallest cross section.

The quantity E in the formula is unfortunately quite uncertain in magnitude. The workman, if he chooses, can easily put an enormous tension upon each cover bolt, so as to stress both the bolt and the mouthpiece ring, to which it is connected, far more severely than is necessary. The ideal way to set up the nuts on the cover bolts is to screw up each nut carefully, with an amount of force just sufficient to prevent leakage when the vessel is under pressure. There is no use trying to compute what the actual tension on the bolt may be with careless handling of the wrench. Generally speaking, we may assume that in good practice the stress applied by the wrench to each bolt will not exceed more than 1000 pounds, the tension that would be required in order to keep the joint tight. Hence it will be fair, perhaps, to take 1000 or (at most) 1200 pounds as the value of E in the formula.

Trade Publications

Numerous applications of inorganic flux-coated welding electrodes are described in a new booklet just published by the Metal & Thermit Corporation, 120 Broadway, New York, N. Y., which earlier this year introduced Murex heavy mineral-coated electrodes. The new booklet describes the welding of mild steel and boiler plate, stainless-steel alloys, manganese steel, high-carbon steel and stainless iron.

FEED-WATER TREATMENT.—"No Scale, No Sludge, No Mud, The Application of Zeolite Water Softeners to the Treatment of Boiler Feed Water" is the title of a 36-page booklet just published by The Permutit Company, 440 Fourth Avenue, New York.

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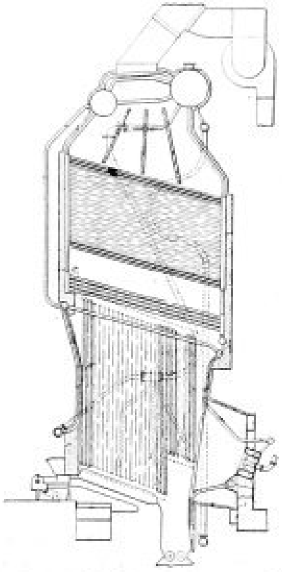
Selected Boiler Patents

Compiled by Dwight B. Galt, Patent Attorney, 1372-1376 National Press Building, Washington, D. C. Readers wishing copies of patents or any further information regarding any patent described, should correspond directly with Mr. Galt.

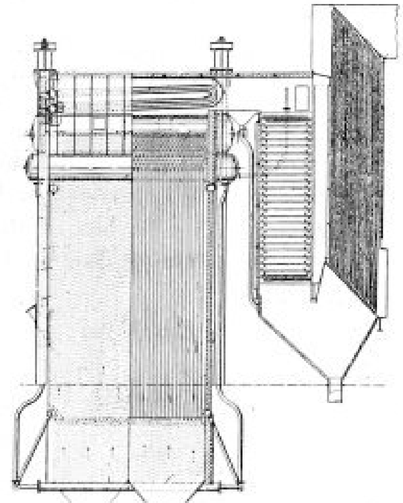
1,807,318. STOKER-FIRED BOILER INSTALLATION. EDWIN LUNDGREN, OF FREDERICK, MD., ASSIGNOR TO INTERNATIONAL COMBUSTION ENGINEERING CORPORATION, OF NEW YORK, N. Y., A CORPORATION OF DELAWARE.

Claim.—A boiler installation including a stoker-fired combustion chamber having an extension chamber, spaced watertubes in said extension chamber so located as to provide air chambers in said extension chamber, means for admitting air to said air chambers, and burner means located in said extension chamber for introducing finely-divided fuel to be burned in space, the air from said air chambers, passing through the spaces between said watertubes for admixture with the finely divided fuel. Five claims.



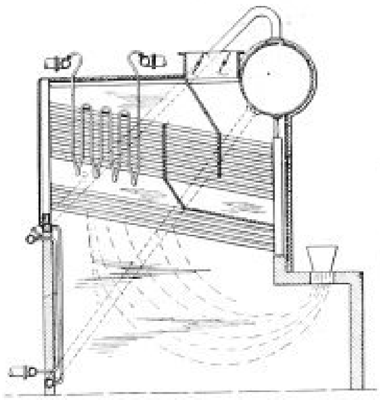
1,804,907. BOILER. BENJAMIN B. WHITTAM, OF ELIZABETH, N. J., ASSIGNOR TO THE BABCOCK & WILCOX COMPANY, OF BAYONNE, N. J., A CORPORATION OF NEW JERSEY.

Claim.—In combination, a furnace having a vertically disposed wall with a plurality of pulverized coal burners therein adapted to project burning fuel into the furnace and lengthwise thereof, a steam and water drum extending lengthwise of the furnace, a plurality of vertically disposed rows of watertubes extending lengthwise of the furnace between the burners, and means connecting each of said rows to said drum. Six claims.



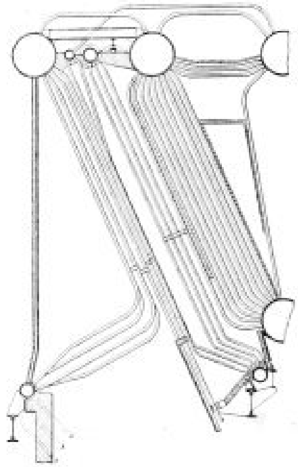
1,807,238. BOILER. CHARLES W. GORDON, OF PLEASANTVILLE, N. Y., ASSIGNOR TO THE SUPERHEATER COMPANY, OF NEW YORK, N. Y.

Claim.—A boiler comprising inlet and discharge headers, a bank of steam generating tubes connecting said headers, said tubes being arranged at one end in spaced rows generally parallel to the direction of gas flow and at the other end in staggered relation with respect to the direction of gas flow, and baffling to direct combustion gases across said tubes in a plurality of transverse passes. Eleven claims.



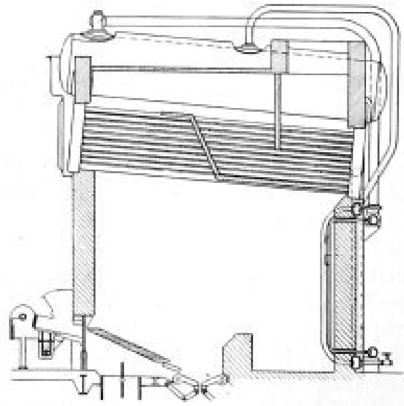
1,827,529. STEAM BOILER AND SUPERHEATER. IVAN L. LANGVAND, OF BARBERTON, O., ASSIGNOR TO THE BABCOCK & WILCOX COMPANY, OF BAYONNE, N. J., A CORPORATION OF NEW JERSEY.

Claim.—In a boiler of the Stirling type, upper and lower drums, banks of tubes connecting said drums, a superheater behind the first bank of boiler tubes, and means to cause gases from the furnace of said boiler to cross some of the tubes of said first bank twice before crossing said superheater tubes. Seven claims.



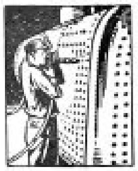
Re. 18,082. RADIANT SUPERHEATER. BENJAMIN BROIDO, DECEASED, LATE OF NEW YORK, N. Y., BY THE SUPERHEATER COMPANY, OF NEW YORK, N. Y., ASSIGNEE.

Claim.—In an apparatus of the class described, the combination of a furnace, a radiant type superheater comprising parallel vertical alined superheater tubes adjacent to a wall of said furnace and watertubes parallel to the superheater tubes and in a plane parallel to and in advance of the plane of the superheater tubes, said superheater tubes being so located relatively to the watertubes and to the furnace that they are exposed to direct radiant heat from said furnace. Five claims.



The Boiler Maker

Reg. U. S. Pat. Off.



Modern Steam Generators

There can be little question but that evolution is going ahead rapidly in the development of the steam boiler, or, as it has come to be called in recent times, the steam generator. Evidence of this trend in the locomotive field is noted in the construction of various types of boilers for high pressures that are wide departures from the conventional. The most recent, of course, is the multi-pressure type for the Canadian Pacific Railroad and for the New York Central, which has been described recently in THE BOILER MAKER.

In the stationary field the most novel probably of the steam generators developed in this country is the mercury boiler of the General Electric Company. Of the more or less conventional types of stationary boilers, the watertube boiler has developed along the lines of higher pressures, as is true also of boilers for marine installation. Few radically different types have been adopted in these fields, however, but interest in the development of such generators in Europe is gradually concentrating the attention of engineers on the problem of reducing the size of the boiler plant, while at the same time increasing the capacity and improving the efficiencies obtainable.

European departures in practical boiler design are noted in the Löeffler high-pressure generator, in the new Brown-Boveri unit, known as the Velox boiler for stationary or marine installation and the most recent, the Sulzer continuous tube steam generator developed at Winterthur, Switzerland. The latter is described in this issue.

The boilers cited are all in more or less a state of development and probably no one offers anything like a complete solution of the many problems that modern practice has erected. They do, however, indicate a trend and each one when it has been studied in operation over a period of time will unquestionably advance the general knowledge in this field of design and carry boiler progress towards another stage of service to the community.

The point these developments emphasize is that this country is somewhat behind in its contribution to the science of boiler design. It is almost inevitable that the boiler of the future will be a wide departure from present types and in its development the designing engineers of this country should play a prominent role. Slack periods in industry make available an opportunity for original investigation in this as in other directions. It would be not only of tremendous scientific interest but would give a considerable economic advantage if a purely American steam generator of high pressure, high capacity, high efficiency and compact design could be developed. Certainly the engineers of no nation are better fitted, nor have they better facilities for doing this job. Nowhere would such generators have as wide application.

Hydraulic Riveting

In spite of the ever-increasing value of fusion welding as a major process in the construction of metal structures of all kinds, riveting still performs the principal function of connecting the strength joints of a boiler. Developments in the technique of riveting are by no means at an end as evidenced by the studies involved for determining the correct holding time of hydraulic riveting as outlined in this issue.

Over a long period of time engineers of the Baldwin Locomotive Works have studied this matter of holding time under pressure to accomplish maximum results in this direction. The experiments involved the testing of sample joints made up of a wide variety of plate thicknesses and different diameters of rivets. Pressure was held on the rivets for various periods of time and subsequently the samples were cut through the axes of the rivets and examined to determine the effect of the holding times employed. Experimental boilers were constructed to test the value of the information thus obtained from the studies on joints.

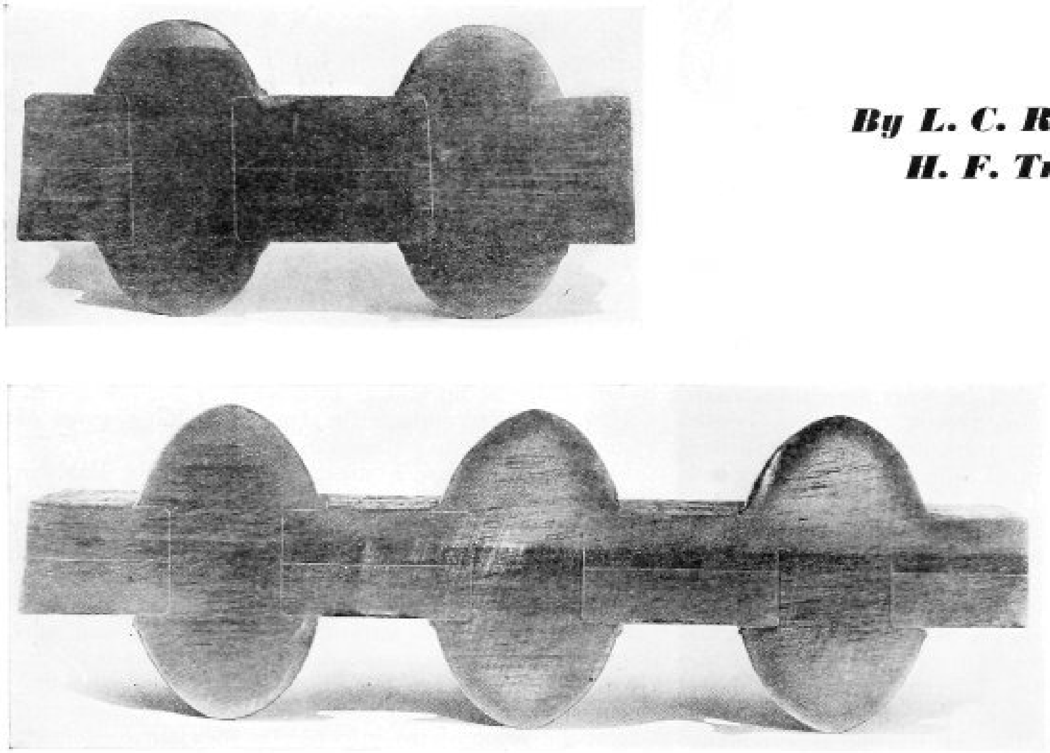
The results on the actual boilers were carefully checked and a standard holding time for each diameter of rivet was determined. Charts were then prepared for the shop staff, covering the requirements for holding time. In order to avoid dependence on the judgment of the operators, the development was carried one step further and an electrical timing device was perfected to control the hydraulic riveting machines automatically. Six machines have been equipped with the device in the boiler shop at Eddystone and over a considerable period of use they have proven completely successful.

On the quality of uniform tightness of all riveted joints in a boiler depends its value in service and its future maintenance requirements. It would seem that studies of this kind adapted to practical construction needs would go far in assuring the continued use of riveting as a major process for joining the strength members of vessels operating under high pressures.

Plant Improvements

The work of the National Committee on Industrial Rehabilitation, of which A. W. Robertson, chairman of the Westinghouse Electric & Manufacturing Company, is the chairman, organized late in August, has as its object the renewal and improvement of industrial plant equipment. Capital expenditures by industrial concerns to this end not only will bring returns in improved efficiency and lowered production costs but will aid in creating jobs in the equipment industries. Employment given in this way will inevitably react to the benefit of industry as it imposes no burden on any group.

Automatic Control of Holding Time in Hydraulic Riveting*



By L. C. Ruber and
H. F. Transue

Figs. 1 and 2.—Test Specimens of boiler seams, cut through the axes of the rivets, to determine the effects of different holding times

The joining of plates by means of rivets is a most important operation in steam boiler construction. In the early days of the art, the hot rivets were driven by hand, the operator using a hammer of weight proportioned to the size of the rivet. This work required great skill and endurance and was comparatively slow and costly. A great advance came with the development of the hydraulic riveter, which exerted enormous pressure on the rivets and formed the heads precisely and quickly. It also permitted the use of much larger rivets than had been possible previously.

With the advent of the hydraulic riveter came a new problem—that of determining just how long a rivet should be held under pressure to give the maximum result. If the pressure were released too quickly the rivet would still be expanded and ductile from the heat, the plate would spring and the rivet would stretch. When finally cooled, the rivet would be loose, and leakage might occur around the rivet and in the seam.

To determine definitely the proper length of time for holding the pressure onto rivets, The Baldwin Locomotive Works carried on a wide series of experiments in making sample joints, using different thicknesses of plates and different diameters of rivets. Pressure was held on the rivets for various periods of time, and the samples were then cut through the axes of the rivets and carefully examined to ascertain the effects of the various holding times. Such test specimens are shown in Figs. 1 and 2.

As a result of these tests the holding time was considered to be tentatively determined, but as a further experiment several boilers were constructed, using the holding time which showed the best results on the test specimens. The results on the actual boilers were carefully checked, and a standard holding time was determined for each diameter of rivet. Tests were then carried out on each of the hydraulic riveters in the shop, and to make sure that the proper holding times were maintained, each rivet driven was timed with a stop watch.

As a result of these experiments a chart was prepared to show the holding time required for each diameter of rivet.

Next came the problem of controlling the work to insure precise conformance to the charted times in actual practice. Any method which depended on human ability to observe the passage of time and to transmit and act upon signals exactly to the second was found to have a degree of error. To eliminate this error, and to insure the riveting pressure being held automatically and exactly for a predetermined length of time, an electrical timing device was designed, built and patented by The Baldwin Locomotive Works. The first device was installed on one of the hydraulic riveters in the Philadelphia boiler shop in 1924. Since that time additional devices have been added and six hydraulic riveters in the boiler shop at Eddystone are now equipped with electric automatic timing devices, and all boiler shell riveting is done under their control. From time to time improvements have been made in the controlling device, and today it is a

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* Published through the courtesy of *Baldwin Locomotives.*

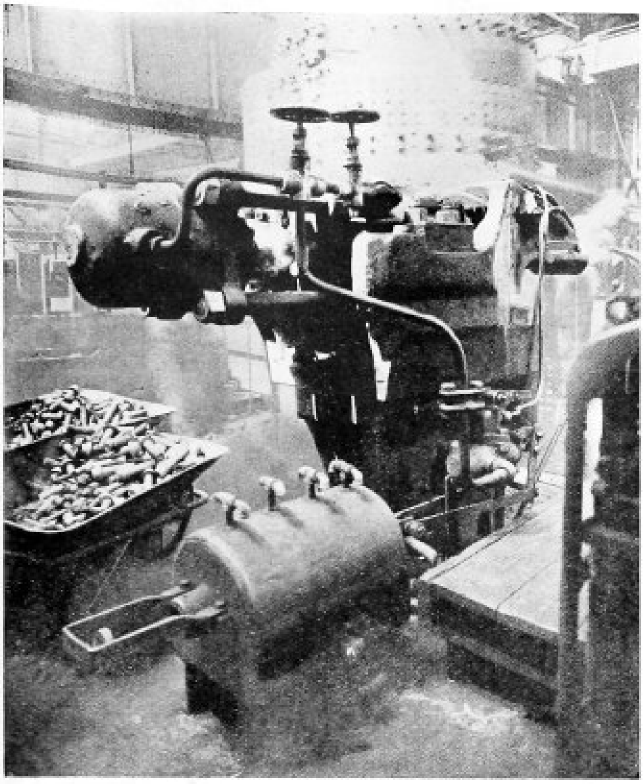


Fig. 3.—The electromagnet

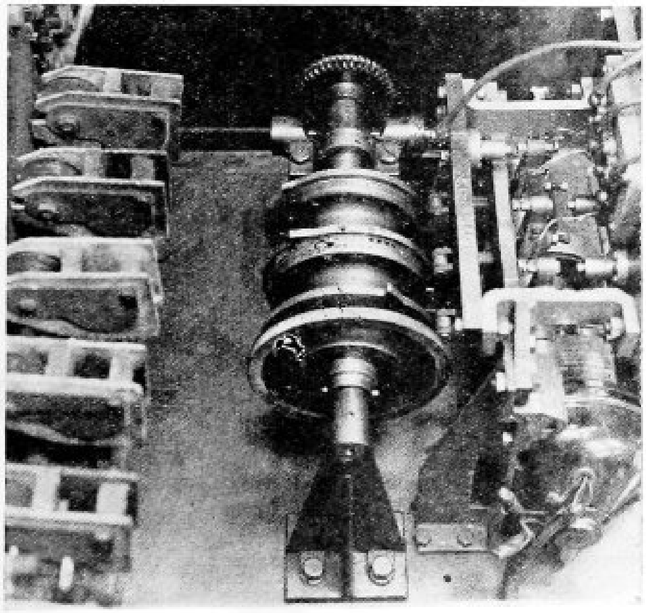


Fig. 4.—The timing drum

simple, sturdy and reliable mechanism which contributes great assistance to satisfactory boiler making.

The timing is accomplished by means of an electric timing drum. This drum energizes an electromagnet to operate the three-way hydraulic valve, which actuates the hydraulic plunger. The device consists essentially of the following parts:

An electromagnet.

A timing drum with motor drive.

An electric circuit, with pushbutton, contactors and wiring.

The function of the electromagnet, shown in Fig. 3, is to operate the three-way hydraulic valve. It is a double-coil direct-current magnet with a single steel plunger enclosed in a cast-iron split housing. One coil is excited to operate the valve for forward motion of the hydraulic plunger and the other coil for reverse motion. Bushings in the ends of the cast-iron housings serve as guides for the brass extensions to the magnet plunger. The extension on one end is provided with a clevis for connecting the plunger to the handle of the hydraulic valve. The other extension is provided with a ring which acts as a stop. The strokes are cushioned by means of rubber washers.

The timing drum, shown in Fig. 4, provides a means for setting a definite "hold-on" period and for operating switches to actuate and control the hydraulic riveter to the period for which it is set. Boiler shop practice requires "hold-on" periods varying from four to twelve seconds, and the drum is designed to provide all divisions of this range. It is driven at one revolution per minute through a worm-gear reduction from a fractional horsepower motor which rotates at 1725 revolutions per minute. It consists of three steel disks with projections on the circumference which, when revolving, trip small switches in the electric current. These projections are spaced 120 degrees apart on each disk, thus allowing three cycles of operation per revolution. When put into motion the middle disk closes an electric circuit through the magnet for forward motion of the hydraulic plunger; the third disk closes an electric circuit for reverse motion of the hydraulic plunger, and the first disk opens a holding circuit to the motor contactor, thus stopping the motor.

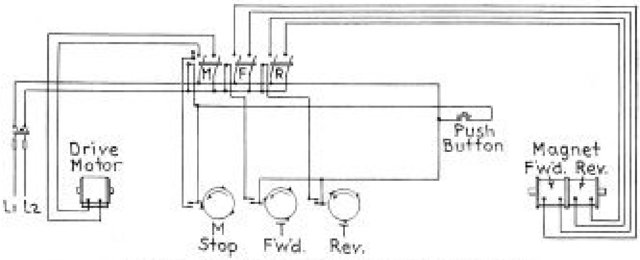


Fig. 5.—Wiring diagram for automatic timer

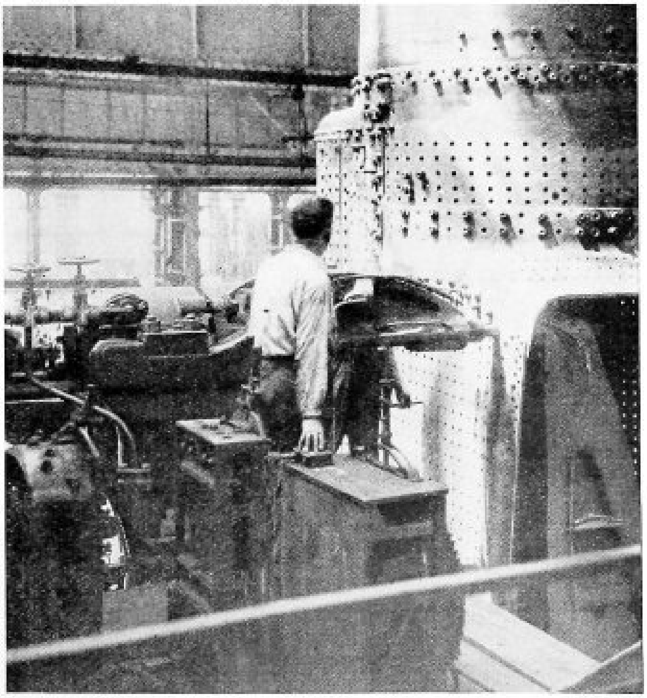


Fig. 6.—Operating a hydraulic riveter equipped with the electrically-controlled timer

The timing is accomplished by changing the angularity of the projections on the middle and third disks by setting the third with respect to the middle. A hollow hand wheel with knurled edges is provided for this purpose. The operator, grasping the hand wheel, may turn the third disk to the desired position, and when he releases it a compression spring inside the wheel forces teeth in the middle and third disks to mesh, thus holding them in position. Numbers stenciled on the third disk correspond to the number of seconds hold-on time desired, and any number placed opposite the arrowhead on the middle disk indicates the number of seconds of hold-on that will occur in operation. The drum, with its motor and gearing, is installed in a steel cabinet for protection.

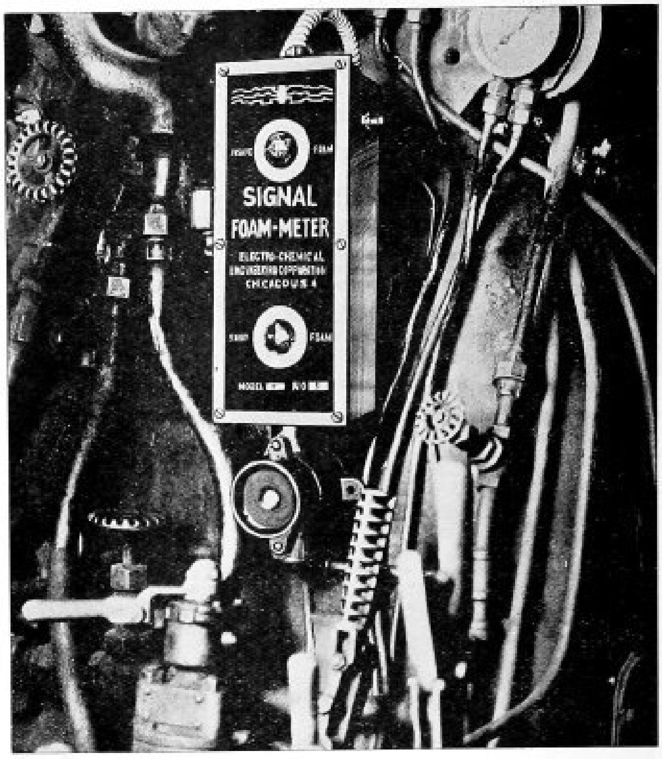
In the electric circuit, and convenient to the operator's hand, is placed a pushbutton which controls the entire operation. Magnetic contactors are used for carrying the main current to the motor and to the electromagnet to relieve the pushbutton and trip switches from possible arcing. These contactors, as well as trip switches, are installed in the steel cabinet with the timing drum. The necessary wiring to complete the electric circuit is installed in steel conduit. Fig. 5 shows the wiring diagram.

When riveting is to be done, the leader of the rivet gang determines from the chart the hold-on time required for the size of rivet to be driven, and sets the timing drum accordingly. When the heated rivet is in position and ready for the pressure, he signals the operator, who presses the pushbutton. From that instant the operation of the hydraulic apparatus no longer depends on human action, but is controlled entirely by the timing device. The motor starts and turns the drum, which is kept revolving by means of a holding circuit in the contactor until the end of the cycle. As the drum begins to revolve, the middle disk closes a circuit through the electromagnet and the hydraulic plunger moves forward. After the hydraulic plunger has formed the rivet and has held the pressure for the number of seconds for which the device is set, the third disk causes the plunger to back away from the rivet. The drum continues to revolve until the end of the cycle, when the first disk stops the motor. The device is then ready to control the forming of another rivet as soon as the latter is inserted and placed in front of the hydraulic plunger. Fig. 6 shows the device in actual operation.

Foam-Meter for Locomotives

An electric foam-meter which indicates the foam conditions in a boiler and which automatically opens and closes a blow-off valve when that is necessary, is now being marketed by the Electro-Chemical Engineering Corporation, a subsidiary of the Pyle-National Company, Chicago, Ill. The condition of foam in the boiler is detected by two pairs of electrodes set into the top of the boiler above the forward end of the firebox. One pair of electrodes is longer than the other. In each case, one electrode of each pair is insulated with Bakelite and the other is in contact with the boiler shell. The balance of the equipment consists of a signal foam-meter with yellow and red light indications placed on the boiler head, an automatic ¾-in. air-operated blow-off valve and several relays.

When the foam rises high enough to reach the long pair of electrodes they are short circuited by the foam. This completes a circuit which includes a sensitive relay that receives its current from the headlight generator or a battery. The relay in turn causes the ¾-in. blow-off valve to open and lights the lower or yellow signal in the foam-meter. After the boiler has been blown down sufficiently, the foam subsides below the end of the electrode, thus breaking the electrical circuit through the relay and simultaneously the secondary circuit operating the Electromatic blow-off valve, thus closing the valve.



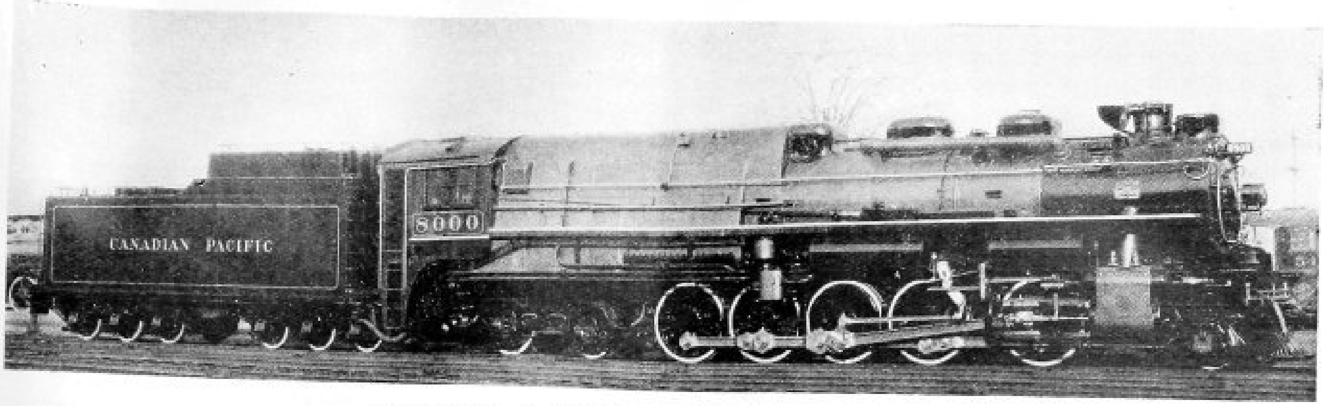
Signal Foam-Meter applied to the boiler head of a locomotive

The valve operates only after the foam has contacted the long pair of electrodes for at least 15 seconds. This is accomplished by a thermostatic time delay placed in series with the electric circuit operating the Electromatic blow-off. This time delay prevents undesirable frequent operation of the blow-off valve due to momentary contact of foam or water with the electrodes. Such momentary contact may be caused by surging or splashing of the water in the boiler.

Should the foam rise high enough to reach the short pair of electrodes, the red light in the foam-meter indicates an unsafe foaming condition, which calls for the manual operation of a main blow-off cock to supplement the Electromatic blow-off.

TRIBUTE TO THE NORTHERN PACIFIC.—Out in Starbuck, Minn., they still appreciate the Northern Pacific. Plans are being made to observe the 50th anniversary of the establishment of railroad service in that city. Starbuck is on the Little Falls-Morris branch of the railway, which was built by the Little Falls & Dakota back in 1882. During November of that year, the first trains over the line were operated by the Northern Pacific.

W. A. Taylor has been appointed division manager of A. M. Byers Company, Pittsburgh, Pa., in charge of the Chicago office of the company.



Canadian Pacific Multi-Pressure Locomotive No. 8000

Operation and Maintenance of the C. P. R. Multi-Pressure Locomotive

By H. B. Bowen[*]

Increasing steam pressures and temperatures have to a great extent been the measure of the progress made in the development of greater power at reduced cost. While locomotive development has made marked advances, it has nevertheless lagged behind stationary practice in utilizing the manifest advantages of higher pressures. Conventional locomotive design, with a large diameter firetube boiler construction, definitely restricts the maximum pressure that can be used, and the Canadian Pacific has gone nearly as far as practical in the increasing of steam pressures with the conventional form of locomotive boiler construction.

All obtainable information concerning new types of locomotives has been studied and such designs as had been transformed into reality have been investigated thoroughly. In 1929 the writer, during a trip to Europe, investigated carefully all of the available examples of new and improved types of locomotives. Conferences with the Superheater Company, Ltd., and its associate organizations, both in the United States and abroad, resulted in further investigations being made of locomotives in England and on the Continent, which utilized the multi-pressure system of steam generation.

Conferences with officers of the German State Railways, the London, Midland & Scottish in England, and the Paris, Lyons & Mediterranean Lines in France, had served to focus attention upon this type of steam-generating apparatus. Subsequently, further conferences with the American Locomotive Company and the Superheater Company resulted in the decision by the railroad to construct a heavy freight locomotive suitable for service on its severe mountain divisions, and its 2-10-4 type locomotive No. 8000, is the result. This locomotive is termed the T4-a class and is, so far as its running gear is concerned, a close duplicate of the T1 class locomotives which are now the standard locomotive for the service referred to. While the T4-a class is slightly heavier as to total weights, it is, nevertheless, very closely the same as the T1 class in so far as weight on the drivers is concerned.

The three-cylinder arrangement has permitted satisfactory operation with a somewhat lower factor of adhesion and, consequently, has a somewhat larger tractive force. High pressures and temperatures are not only

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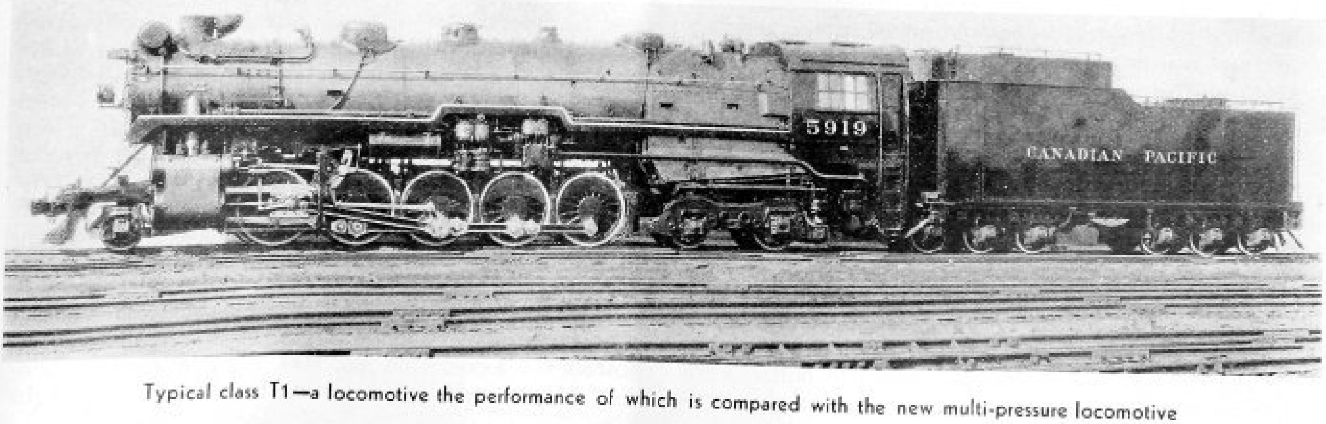
[*] Chief of Motive Power and Rolling Stock, Canadian Pacific.



Typical class T1—a locomotive the performance of which is compared with the new multi-pressure locomotive

PERFORMANCE RECORD T4 AND T1 CLASS LOCOMOTIVES

Class	T4-a	T4-a	T4-a	T4-a avg.	Improvement T4-a over T1, percent	T1-a avg.	T1-a	T1-a	T1-a
Locomotive No.	8000	8000	8000	......	......	......	5907	5905	5905
BEAVERMOUTH TO GLACIER (WESTBOUND) 22.5 MILES									
Date	4-23-32	4-15-32	6-22-32	......	......	......	4-26-32	4-28-32	......
No. of cars, total	37	44	16	......	......	......	20	17	......
Weight of train, equiv. gross tons	1,029	1,060	1,103	1,064	......	1,069.5	1,046	1,093	......
Equiv. gross ton-miles	23,152	23,850	24,818	23,940	......	24,063	23,535	24,592	......
Lb. of oil, total	3,374.8	4,399.8	4,689.9	4,154.7	......	4,878.4	5,144.4	4,612.5	......
Lb. of oil used per 1,000 equiv. g.t.m.	146.0	184.8	188.9	173.2	14.9	203.5	218.8	188.2	......
Lb. of oil used per locomotive-mile	149.9	195.5	208.4	184.6	......	216.8	228.6	205.0	......
Lb. of water, total	47,850	47,130	49,150	48,040	......	49,885	47,840	51,930	......
Lb. of water used per lb. of oil	14.2	10.7	10.47	10.79	5.3	10.25	9.3	11.2	......
Total time on road	1 hr. 16 min.	1 hr. 45 min.	1 hr. 47 min.	1 hr. 36 min.	......	1 hr. 24 min.	1 hr. 18 min.	1 hr. 31 min.	.....
No. of stops	.....	.....	.....	......	......	......	......	......	.....
Total time delayed	.....	.....	.....	......	......	......	......	......	.....
Ruling grade and curve	2.2 percent, 10 deg.	2.2 percent, 10 deg.	2.2 percent, 10 deg.	......	......	......	2.2 percent, 10 deg.	2.2 percent, 10 deg.	......
ALBERT CANYON TO GLACIER (EASTBOUND) 19.4 MILES									
Date	4-30-32	6-15-32	6-23-32	......	......	......	4-25-32	4-27-32	4-29-32
No. of cars, total	49	37	41	......	......	......	44	45	44
Weight of train, equiv. gross tons	1,088	1,076	1,108	1,090.6	......	1,053	1,058	1,051	1,052
Equiv. gross ton-miles	21,107	20,874	21,495	21,158.6	......	20,441	20,525	20,389	20,408
Lb. of oil, total	4,583.5	4,960.7	4,951	4,831.7	......	5,467	5,424.8	5,753.6	5,192.7
Lb. of oil used per 1,000 equiv. g.t.m.	217.1	237.6	230.3	228.3	14.7	267	264.6	283.4	254
Lb. of oil used per locomotive-mile	236.2	255.7	255.2	249	......	281.2	279.5	296.5	267.6
Lb. of water, total	54,460	49,150	54,070	52,560	......	54,040	50,570	53,270	58,270
Lb. of water used per lb. of oil	11.88	9.9	10.92	10.9	10.1	9.9	9.3	9.2	11.2
Total time on road	1 hr. 52 min.	1 hr. 19 min.	1 hr. 48 min.	1 hr. 40 min.	......	1 hr. 33 min.	1 hr. 23 min.	1 hr. 51 min.	1 hr. 26 min.
No. of stops	2	.....	1	1	......	2	2	1	3
Total time delayed	29 min.	.....	20 min.	16 min.	......	2 min.	2 min.	1 min.	2 min.
Ruling grade and curve	2.2 percent, 10 deg.	2.2 percent, 10 deg.	2.2 percent, 10 deg.	......	......	......	2.2 percent, 10 deg.	2.2 percent, 10 deg.	2.2 percent, 10 deg.

attractive from the standpoint of improved thermodynamic performance, but in themselves bring about desirable changes in construction and performance. For example, higher pressures necessitate multi-cylinder construction to fully utilize the steam expansively. This immediately gives a more uniform torque and permits lowering the adhesion factor between the driving wheels and the rails, which in turn provides for the better utilization of weight. Higher pressures give a better steam flow and reduce proportionate pressure drops. It is customary to allow a 15 percent drop in boiler pressure in figuring the maximum tractive force. This loss, however, is reduced as the pressure rises, giving greater proportionate tractive forces. Higher pressure means smaller cylinders, removing some of the restrictions due to large cylinders as determined by customary boiler pressures. The higher density also provides for the use of smaller pipes for feeding both cylinders and auxiliaries, which is a most grateful relief to those who have struggled in trying to fit in the maze of complicated piping required by even the comparatively simpler conventional type of locomotive.

Multi-pressure locomotive No. 8000 is the joint production of the Superheater Company, the American Locomotive Company and the Canadian Pacific, who are, respectively, responsible for the high-pressure steam generating system, the three-cylinder arrangement, valve motion and the general locomotive proportions, and the design and construction, which was done at our Angus shops, Montreal, Que. It was turned out for operation in July, 1931, and during the ensuing few months the engine was used in freight service between Montreal and Smith Falls, Ont., to determine its operating characteristics and, particularly, to develop the arrangement of oil burners which it was anticipated would have to be somewhat different from the standard burner arrangement used on our western lines.

Construction was commenced in November, 1930. The machinery, together with the low-pressure boiler and firebox framework, was erected in a complete unit, while the closed system and high-pressure boiler were assembled on a jig especially constructed for this purpose in the boiler shop. Work on these separate units progressed simultaneously and on April 10, 1931, the closed system and high-pressure unit being completed, were subjected to a hydrostatic test, each being found tight under the required test pressure. On April 13 the closed circuit and high-pressure unit were released from the jig and transferred to the machinery and low-pressure boiler unit. With the addition of cab, fixtures, fittings and piping, the engine was ready for steam test.

The steam test involving the closed circuit, high-pressure and low-pressure boilers, was carried out in two stages; i. e., it was thought desirable to work gradually to the maximum. Accordingly, the first test was discontinued when a pressure of 500 pounds per square inch was reached in the high-pressure boiler. Full pressure of 850 pounds per square inch was reached on final test.

Various features show excellent examples of how metallurgical developments have permitted constructions that would not have been possible with the materials of construction available only a few years ago. Stainless steels for valves and valve seats have surmounted the limitations of bronze and the corrosion defects of ordinary steels. Stainless-steel plates have provided a direct baffle for the oil flame to prevent the flame coming in direct contact with the drums. These plates, while resistant to high temperatures, have given some trouble on account of expansion, which necessitated some changes in clearance. Low-carbon nickel steel for seamless drum construction has permitted high factors of safety with reduced weight. Nickel-steel boiler plate in the low-pressure boiler has also given the requisite strength with approximately 30 percent reduction in weight. Nickel-steel forgings have provided toughness and resistance to abuse and impact with minimum weight in various driving and motion parts.

It is customary on the Canadian Pacific to operate locomotives by the pool system. Naturally, it is desirable to make the operation of all locomotives as simple and as uniform as possible. This was kept in mind in all controls for locomotive No. 8000, and in spite of there being two superheaters and two throttles for the high-and low-pressure cylinders, the throttle mechanism was worked out so that only one throttle lever is used for opening both throttles.

The operation is identical with that of an ordinary locomotive. The low-pressure boiler is fed with a standard Elesco CF-1 feed pump and is provided with a Hancock inspirator as an auxiliary. Incidentally, the CF-1 pump for the low-pressure boiler is located within a compartment provided in the tender tank; a location that we had previously tried out with considerable success and advantage. The high-pressure boiler is pumped with a specially designed CF-1 boiler feed pump adapted to high-pressure service. A duplicate of this pump is also provided as a standby and auxiliary in connection with feeding the high-pressure boiler. The high-pressure water pumps are located one on either side of the boiler.

The cut-off control in the valve motion is a straight duplicate of that of an ordinary locomotive. The oil-burner control is also identical and the only deviation is the two varying water levels for the high- and low-pressure boilers, respectively, and the two boiler-feed pumps for feeding these two boilers. The cross-over valve is an addition and there are other features that require periodic checks by the engineman, but these have all been so simplified that it has not constituted any objection from the standpoint of simplicity of operation. In fact, after the initial period the locomotive was turned into the regular pool and is now handled by any engineman who may be assigned to it on any individual run. There has been keen interest on the part of the enginemen and a desire to be assigned to locomotive No. 8000.

The most difficult operating problem on the Canadian Pacific is the movement of both freight and passenger traffic over the Mountain sub-division in British Columbia. Numerous heavy grades are encountered, the worst of which is 22.5 continuous miles of almost uniform 2.2 percent grade.

In 1929, 20 new locomotives were put in service on this sub-division, of the 2-10-4 wheel arrangement, using 275 pounds boiler pressure and two simple cylinders. These locomotives immediately introduced new standards of economy and performance, and by no means the easiest problem was selected when the Canadian Pacific decided to construct for direct comparison a multi-pressure locomotive of substantially the same weight, wheel arrangement and proportions as the T-1 class engines now handling traffic on this sub-division.

Locomotive No. 8000 burns oil as fuel and the proportions of the flash pan, size, number and location of oil burners constituted a real problem, as there was practically no precedent on which to base the design. When the locomotive was originally turned out, several experiments were made with location of a single burner and a double burner, and after a considerable amount of experimenting, proper locations for two burners were decided upon, one of which amply provides for ordinary demands with the second one being cut in when required by abnormal conditions.

It will be appreciated just how difficult this problem was when it is understood that three separate boilers—the closed system, the high-pressure boiler and the low-pressure boiler—are all supported by the one firebox and combination of burners, and that a balance in the heat distribution must be maintained between these three heat-absorbing units through a wide range in steam demands of the locomotive.

This has been partially offset by introducing a by-pass valve from the high-pressure to the low-pressure boiler so that excess steam can be by-passed without losing it through the safety valves. Even as yet this by-pass arrangement is used to some extent, but, of course, the ideal solution to be accomplished is the final adjustment and balance of component features of operation so as to obviate entirely the necessity of any by-passing.

This can, of course, only be accomplished by close observation and adjustment of the locomotive in operation until the final proportions and adjustments are definitely determined. Originally, the cross-over valve gave trouble in that it caused severe disturbances of the water due to by-passing the high-pressure steam, which was overcome by a change in design of the valve outlet into the low-pressure boiler.

Proper draft adjustments and boiler conditions are somewhat more difficult to secure with locomotive No. 8000 as the exhausts are secured at uneven intervals. The uneven effects of the exhaust on the draft conditions of the engine are more noticeable at low speeds. This has been greatly improved with better combustion and increased superheat temperatures by installing baffle plates in the smokebox with apertures to more evenly diffuse the draft, a change in stack diameter and changes in the exhaust nozzle. While great improvements have been effected, the problem is so different from an ordinary locomotive that still further improvements can be expected.

In operation the locomotive has thoroughly lived up to expectations in the matter of uniform torque, marked advantages in hauling heavy loads at low speeds due to more even torque, smooth running development of maximum tractive force and slipping far less than the T-1 engines under adverse rail conditions.

At the present time there is no doubt as to the increased maintenance cost of the multi-pressure engine over that of the conventional simple locomotives operating over the same division. This is generally true of locomotives radically different in design and construction, as the shop men lack experience on this type of locomotive and must become acquainted with the various details which are different from those on the locomotives they have been maintaining. Many experimental changes are also continually in progress at present which are apt to be mistaken for regular maintenance. Boiler men, not acquainted with the new boilers, require more time to wash out the two systems than they will when thoroughly acquainted with this work.

The safety valves presented a difficult problem in that their dimensions were strictly limited and that they must operate under conditions of vibration and exposure not met with in stationary practice. The valves originally applied were not altogether successful. The valves on the closed circuit system gave trouble largely on account of the actual details of construction, which has been overcome by rebuilding the valves in the railway shops by the use of a new form of seat and valve. The high-pressure safety valves also required a change in form of construction, but at present it appears that these difficulties are largely eliminated.

The check valves in the water-delivery line to the high-pressure boiler gave trouble on account of the very high concentrated load in the valve being almost impossible to maintain it in a tight condition for any length of time due to the pounding it received. The solution was a twin check valve with the proper capacity provided by two valves side by side in the same body, so that the total load on each valve of reduced diameter was little greater than the load on the single valve used in conventional practice.

Although over one year has elapsed since the loco-

(*Continued on page 237*)

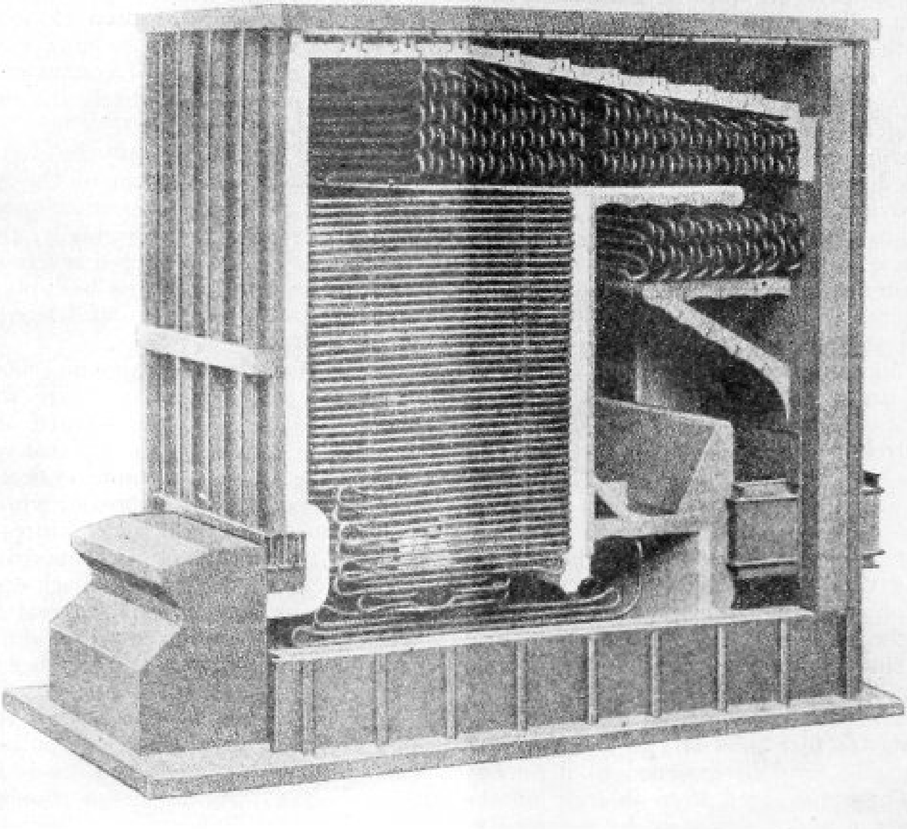


Fig. 1.—Model of single-tube steam boiler

Novel Swiss Steam Generator Employs Continuous Tube Heating Surface*

It is seldom advisable to offer descriptions of new inventions and designs before they have been actually tested on a reasonably large scale, but a justifiable exception is presented by a type of steam generator which Messrs. Sulzer Brothers, Winterthur, Switzerland, is now constructing after two years' tests with an experimental plant. When we say that such a generator, with, for example, an output of 20,000 pounds of steam per hour, will have a continuous steel tube about 2 inches in diameter and no less than 1½ miles long, we shall excite the interest of our readers.

A steam generator of this type at present under erection in a textile works will have a normal output of 18,000 pounds per hour at a pressure of 1400 pounds per square inch, and will supply steam to a high-pressure turbine, the exhaust from which is employed for heating the buildings and for process work (dyeing, bleaching, etc.). A larger generator, about 40,000 pounds per hour evaporation, is also being made for a district heating plant in Zürich. The high-pressure steam from this generator will also go to steam turbines, and the exhaust from these turbines will be employed for heating the numerous buildings of the Federal Technical University, a hospital, and neighboring quarters.

A model of the generator is shown in Fig. 1, and the tube arrangement in Fig. 3. In Fig. 1 the mechanical stoker is indicated by the block on the left side. The rectangular coil immediately following it is the combustion chamber, F in Fig. 3. This coil is only one tube thick and forms the superheater, the steam being delivered at its summit, as shown at B, Fig. 3. Rising from the combustion chamber the hot gases pass successively through groups E D C, and pass to the uptake through an outlet on the right, depositing soot, etc., in the trough seen in section in the model.

In order to prevent the tubes sagging together, they are separated by distance pieces welded on at intervals. The feed, already heated, enters the tube at A in contra flow to the hot gases. Messrs. Sulzers are satisfied that in a short time it is converted into a froth which gradually becomes dry superheated steam, and in that form issues at a high velocity at B. In the actual plant it will directly enter the steam main of the turbine. No separator is used.

Since it is impossible to produce a steel tube, 2 inches in diameter and 1½ miles long, the separate lengths have to be welded together in the same manner as in the manufacture of ordinary superheater tubes. The process, which has been carefully worked out, leaves a smooth internal surface.

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* Reprinted from *The Engineer*, October 14 issue.

The question as to "What is to be done in the case of local failure of the tube?" is certain to arise. The answer is that a length can be cut out and a new length welded in at any part. It will be noticed that all the bends are accessible, as in the ordinary type of superheater, and Messrs. Sulzer are so satisfied that repairs to any part of the boiler can be quickly effected that they are ready to guarantee the replacement with a reasonable time limit of any part of the tube.

Owing to the fact that there is no reserve of water in generators of this general type, it is usual to control the feed supply by appropriate apparatus. When the fuel is oil, the burners are also controlled. That cannot be so readily done in the present instance, since the boilers in question will be coal fired, but the control apparatus will regulate the speed of the automatic stokers and operate dampers. Furthermore, in the case of a sudden cessation of demand for steam, the tube will be at once flooded with water to prevent overheating.

The feed control apparatus is shown diagrammatically by Fig. 2. Here the upper part of the dark coil A represents the economizer and the generator proper, and the lower part the superheater. The looped extension to the right of the superheater is a thermostat to which we shall return. The main feed enters the top of the coil direct from a reciprocating feed pump B. If this pump alone were used, then the temperature of the steam would follow the amount of feed fairly accurately, as long as the changes in the demand for steam were slow. If they were rapid, however, undesirable changes in the steam temperature might take place, owing to the great length of the tube. To prevent such changes, a small quantity of feed is delivered into the superheater through the valve H. Both supplies are regulated by the temperature control valve D, while the pressure of the steam is regulated by a delivery valve.

The thermostat operates the temperature-regulating valve D, which operates the valve L, which, in its turn, controls the pressure in the branched pipe Y. This pressure controls two servo-motors, one of which governs the speed of the feed pump B, while the other opens, more or less, the valve H in the secondary feed pipe. In consequence of the pressure of the springs in the control valves of the servo-motors being greater when the servo-motor piston is in the top position than when it is in the lower position, *i.e.*, when raising steam at a high rate, the temperature would become higher than when raising steam at a low rate. To get over this difficulty, a third servo-motor E is employed. This servo-motor is controlled from the temperature regulator D, and, as shown in the diagram, it increases or diminishes the load on the spring K. Now, the control valve of the servo-motor E is in its central position only when the steam is at normal temperature. At any other temperature it moves, causing the piston of the servo-motor to move and to increase or diminish the pressure on the spring K. This alters the position of the valve L until the supply of primary and secondary feed is just sufficient to keep the temperature of the steam at the normal.

Tests carried out during two years with an experimental generator capable of raising 8000 pounds of steam per hour showed that this system of feed and temperature regulation worked excellently.

We may fitly conclude this brief description of a generator which is certain to arouse much interest among engineers with Messrs. Sulzer Brothers' own statement of the considerations which influenced its design:

(1) In order to avoid steam bubbles clinging to the tube walls, the speed at which the working medium flows through the generator is increased. To be certain of obtaining sufficient speed at every point, the generator is constructed as a single tube of great length, the feed being pumped into one end and the steam issuing in a

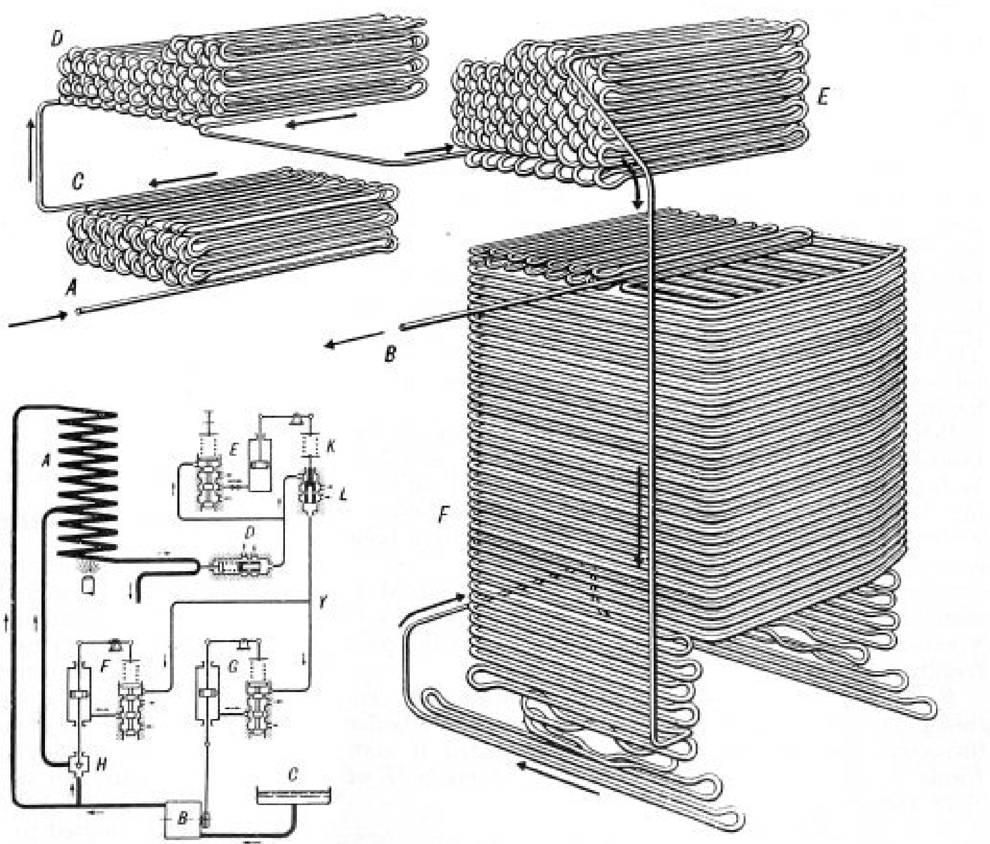


Fig. 2.—(Lower left) Feed control apparatus shown diagrammatically. Fig. 3.—(Right) Tube arrangement of new single-tube steam generator

superheated state from the other. On account of the positive flow, it is absolutely impossible for the steam to remain stationary anywhere; the mixture of water and steam hurries as a froth through the tube in the evaporating zone. Also, in consequence of the positive flow, it is unnecessary to take any of the precautions previously required for ensuring circulation in the tube, which is led in many coils through the hot flue gases. Consequently it is possible to meet the special requirements of any particular case.

(2) In principle, condensate is used for feeding the generator, since all experience made with the former high-pressure boilers has shown that the use of condensate is the most certain way to avoid deposits forming on the tubes.

(3) Since the generator is constructed as a single continuous tube, expensive boiler drums are eliminated. The storage capacity is, however, low, and measures had therefore to be taken so that the rate of evaporation should be rapid enough to suit fluctuations in service conditions. This is effected by adopting completely automatic regulating devices, which keep the steam pressure and temperature always approximately constant, and regulate the quantity of feed to comply with the heat of the furnace. According to circumstances, the firing is also automatically regulated; for example, in accordance with the pressure in the steam pipes, or in plants with low-pressure steam storage in accordance with the pressure at which the steam is stored. Through the regulating apparatus, which is connected with automatic safety devices, attendance on the generator is greatly simplified.

Work of the A. S. M. E. Boiler Code Committee

The Boiler Code Committee meets monthly for the purpose of considering communications relative to the Boiler Code. Any one desiring information as to the application of the code is requested to communicate with the Secretary of the Committee, 29 West 39th St., New York, N. Y.

The procedure of the committee in handling the cases is as follows: All inquiries must be in written form before they are accepted for consideration. Copies are sent by the secretary of the committee to all of the members of the committee. The interpretation, in the form of a reply, is then prepared by the committee and passed upon at a regular meeting of the committee. This interpretation is later submitted to the council of the society for approval, after which it is issued to the inquirer and published.

Below are given records of the interpretations of the Committee in Cases Nos. 706, 714, 725, 729 and 731, as formulated at the meeting on September 16, all having been approved by the council. In accordance with established practice, names of inquirers have been omitted.

Case No. 706.—*Inquiry:* Would a boiler drum fabricated of steel tubing purchased abroad which meets Code Specifications S-17 or S-18 be acceptable under the code requirements?

Reply: It is the opinion of the committee that steel tubing purchased abroad should be acceptable by boiler inspectors for the purpose specified provided it conforms to all the requirements of Specifications S-17 or S-18 and the provisions of Par. P-3.

Case No. 714.—(Annulled).

Case No. 725.—*Inquiry:* Is not the use of a diameter of 48 inches in the second section of Par. U-31 inconsistent with the present reading of the first section of this paragraph, and should not the limit of ½-inch plate be substituted for the 48 inches diameter?

Reply: It is the opinion of the committee that it was the intention to eliminate limiting diameters in Pars. U-30 and U-31 when the change was made from previous editions of the code and that the omission to change the "48 inches in diameter" in the second section of Par. U-31 was an oversight and there should be substituted therefore the words "½ inch in thickness."

Case No. 729.—*Inquiry:* What is the definition of the term "lethal" as referred to in the Code for Unfired Pressure Vessels?

Reply: It is the opinion of the committee that in this section of the code the word "lethal" is to be interpreted as applying to poisonous gases or liquids of such a nature that a very small amount of the gas or vapor of the liquid mixed or unmixed with air when breathed is dangerous to life. For purposes of this code, this class includes substances of this nature which are stored under pressure or may generate a pressure if stored in a closed vessel. Some such substances are hydrocyanic acid, carbonyl chloride, cyanogen, mustard gas, and xylyl bromide. For the purposes of this code ammonia, chlorine, natural or manufactured gas, propane or butane are not considered as lethal substances, but it was the intention of the committee that their storage should not be permitted in Class 3 pressure vessels.

Case No. 731.—*Inquiry:* Is it permissible to attach by electric resistance butt welding, projecting metal elements to the shell or waterbearing portions of a power boiler for the purpose of forming extended heating surface provided extensive tests have demonstrated that the effect of such resistance welds is scarcely noticeable as far as the physical properties of the shell are concerned? It is recognized that Par. P-186 imposes sharp limits on the area of welding that can be applied to a boiler shell, but attention is called to the fact that the electric resistance welding used in this case involves a minimum of heating of the plate.

Reply: It is the opinion of the committee that the limitations imposed on the welding of non-pressure parts to power boiler shells or drums applies particularly to fusion welding and was not intended to apply to resistance welding where the entire area is welded simultaneously. The attachment of metal elements for extended heating surface by electric resistance butt welding does not conflict with the requirements of the code and it is the opinion of the committee that the electric resistance method of attaching projecting metal elements may be used under the conditions specified without reducing the maximum allowable working pressure.

Course in Designing for Welded Construction

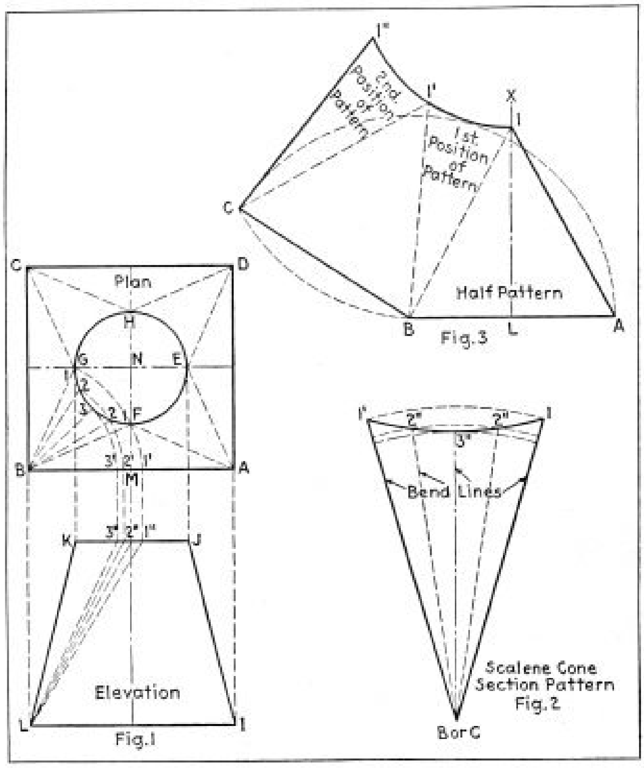
Success of the combined practical and theoretical one-week welding course offered by John Huntington Polytechnic Institute, Cleveland, in co-operation with The Lincoln Electric Company, has led to the decision to repeat the course several times during the winter season.

The course consists of a week's intensive work, six days being spent in the operator's training school of The Lincoln Electric Company. Each evening from Monday to Friday a lecture on designing for arc-welded construction is given at John Huntington Institute.

Enrollment is limited to 30 men and applicants must be college graduates in engineering or the equivalent.

Rapidity in Laying Out

By William Morrison



Transition piece development

The importance of accurate methods of laying out patterns in the least possible time, which is an important factor in these days of sharp competition, has induced the writer to lay before the readers of THE BOILER MAKER a rapid method which may be employed in developing the pattern for an irregular solid whose base is square and whose top is round. This method is applicable whether the top opening of the solid is placed exactly in the center of the base or at one side or corner.

With very little additional attention the student will be able to produce these patterns with the greatest accuracy without the aid of triangulation usually recommended as the correct method for producing these patterns. The principles of the method shown here are more or less closely related to the principles of triangulation, but the student will be able to grasp the reason for and the use of each and every line more readily, due to the simplicity of the method.

The following is a rapid method for developing an irregular solid from square to round with the center of the top, in this example, directly over the center of the base. First, draw the elevation *I-L-K-J*, Fig. 1. Above the elevation draw the plan *A-B-C-D*. Locate the center of the plan *N* so that the line *A-B* of the plan is about 10 inches above the line *K-J* of the elevation.

In all examples of pattern development, where some portion of the object is represented by a circle, the circle must be divided into parts. As the circle, in this case, is in the center of the square, making the four corners symmetrical, it is necessary only to divide the one-quarter circle into a number of equal parts, as shown by the points *1, 2, 3, 2* and *1*. The writer has found this number of equal parts to be quite effective to explain the method, although a still greater number will more closely approximate the circle. After spacing the one-quarter circle into the number of equal parts, the dividers should be carefully laid aside so that their setting will not be altered, as they are to be used later on.

From points *1, 2, 3, 2* and *1* in the plan, draw lines to the apex *B*. The corners in the plan shown as *F-B-G, G-C-H, H-D-E* and *E-A-F* should be considered as sections of scalene cones. Now, using *B* as a center and radii equal to *B-1, B-2* and *B-3* in the plan, describe arcs intersecting *A-B* at *1', 2'* and *3'*, as shown. From these points drop vertical lines down to the elevation, intersecting the line *J-K* at *1", 2"* and *3"*. From these points draw lines to *L*. Then *L-1", L-2"* and *L-3"* will be the true lengths of the lines shown in plan by *B-1, B-2* and *B-3*, respectively.

To create a high degree of rapidity in laying out these patterns, it is essential to make a pattern on light sheet iron of the scalene cone section, as shown by an enlarged view in Fig. 2. To develop this pattern proceed as follows: With radii equal to *L-1", L-2"* and *L-3"* in the elevation and with the dividers used to space off the one-quarter circle in the plan, start at *1* in Fig. 2 and step off arcs having similar numbers as *1-2", 2"-3", 3"-2"* and *2"-1'*. From these points draw lines to apex *B*. If upon these lines small holes are punched at random, this pattern can also be used to locate the bending lines.

To lay out a half pattern proceed as follows: In Fig. 3, draw the horizontal line at *A-B* equal in length to *A-B* in plan; next, bisect *A-B* and locate point *L*. From this point erect a perpendicular line *L-X*, place the scalene cone pattern in the first position as shown, then draw the lines *B-1-1'-B*. Then *1-L* should equal *L-K* in the elevation, Fig. 1, which represents the true length through *F-M* in the plan.

Now, using *1-B*, Fig. 3, as a radius and *1'* as a center, describe the arc *B-C* and intersect it by an arc struck from *B* as a center and with *B-A* as a radius, locating the point *C*. Next, place the scalene cone pattern in the second position, as shown; then draw the lines from *C-1'-1"-C*. Next, connect the two intersections by drawing the lines *1-A* and *B-C*, which completes the half pattern.

In drawing the lines around the scalene cone pattern in these correct positions, the layerout should see that they do not move. In drawing the center of the plan *N* so that there is 10 inches between *A-B* and *J-K*, this is done so that, with the aid of a square, the points *1', 2'* and *3'* on *A-B* can be drawn to the line *J-K* much more rapidly and accurately than if the plan were drawn below the elevation and perpendicular lines erected up to line *J-K*.

Various problems of this nature will arise in the shop, and if the principles of the foregoing method are thoroughly understood these will be easily mastered. On some problems it may be found necessary to make two or three instead of one of these patterns.

Applying job analysis to the above example, without the aid of the pattern shown in Fig. 2, where a half pattern is required as shown in Fig. 3, the scalene cone section would have to be developed twice and on a one-piece pattern four times. For the foregoing example, with the aid of the pattern, it is only necessary to develop the scalene cone section once. Also, the drawing number can be marked on these patterns for duplicate orders so that no plan and elevation views are required for such duplicates.

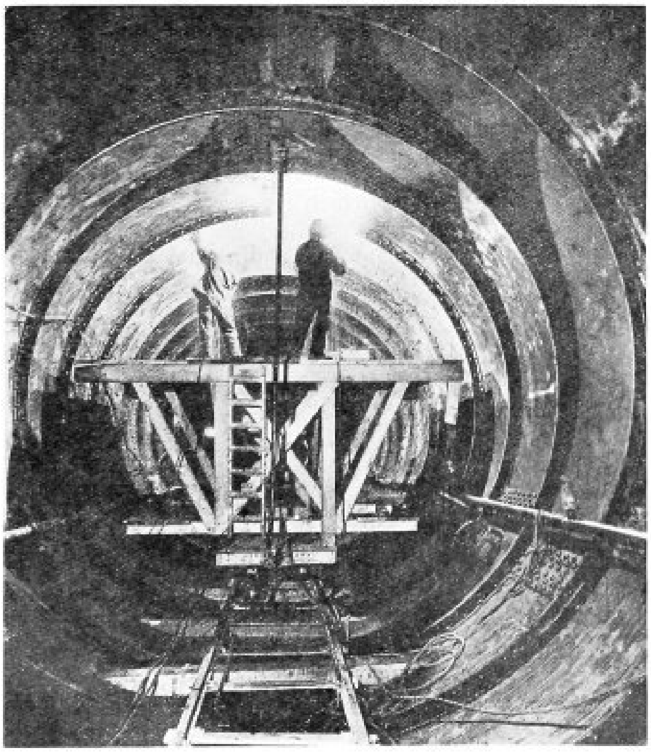
For the benefit of the student, the writer has gone to a considerable length to explain the whys of the method. Familiarity with the method itself will create a degree of rapidity that a pattern for an irregular solid, as shown in Fig. 3, can be laid out in the same amount of time that is required to read this article.

Arc Welding of Tunnel Liner Plates Insures Watertightness

As construction progresses at record speed on the 21-mile-long New York City tunnel No. 2, engineers are watching with interest the various methods employed. Arc welding of the steel liner plates to insure watertightness was one of the time saving devices utilized.

This $43,000,000 tunnel extends from the Hillview Reservoir in Yonkers to Brooklyn, and is part of the Board of Water Supply's system for delivering water to the Boroughs of Bronx, Queens and Brooklyn. Specifications called for special steel liner construction with arc-welded calked joints near and under the East River where seepage might be encountered.

The steel liner is made up of 1-inch plate fabricated in sections above the ground. Three sections form a 20-foot circle. A total of 13,920 feet of 3/16-inch fillet welds were laid above ground along 4-inch by 4-inch by ½-inch angles which were riveted to the liner segments.



Liner plates for tunnel being arc welded

The illustration shows welding operators at work laying the calking welds after the sections had been assembled in the tunnel. Each ring required 102 feet of continuous welding where two angles butt together. Operators averaged 31 lineal feet of welding per man per hour using the shielded-arc method of welding and equipment manufactured by The Lincoln Electric Company, Cleveland. Patrick McGovern, Inc., has the contract for all work on the tunnel.

Shop Safety Important During Slack Periods

During a period of slack employment in an industrial plant special effort should be taken to guard the safety of the workers, in the opinion of Dr. James H. Greene, manager of the Co-operation Department of Industrial Relations of the Studebaker Corporation, who recently addressed the Automotive and Machine Shop Section of the Annual Safety Congress, held at Washington, D. C., October 3 to 7.

Dr. Greene outlined a specific program to be followed, based on the favorable safety results which have been secured in the Studebaker plant. "While we do not claim to have a perfect set-up in our plant," he stated, "the improvement in our accident frequency and severity rates, as well as in our compensation experience, indicates, beyond the shadow of a doubt, that our safety work has paid for itself, and more."

Dr. Greene contended in his talk that the safety work of a plant should be considered as one part of a strong, centralized, personnel organization which during times of depression could more easily adjust itself to the needs of an economy program.

This involves of course trained workers to participate in all phases of personnel work, an objective not easy to attain. "On the other hand, such workers are much more valuable as safety men if they know something of the other allied phases of the personnel program. While I have no facts and figures available, it is my earnest belief that safety work has suffered less in plants and organizations where there was a well rounded personnel than in plants where safety was perhaps the only personnel activity of any importance which was being pushed.

When work is slack there are certain specific things which can be done to further a safety program which are not possible during peak operations. These may be listed briefly as follows:

1. Machinery should be checked over for hazards which can easily be overlooked under maximum production conditions.
2. Personnel workers who are handy with tools can make necessary repairs to guards, thus saving maintenance costs.
3. All safety equipment which normally is in constant use can be checked over and inspected during slack periods.
4. It is possible to watch operations more in detail than is possible under normal operating conditions and hence provide safeguards against hazards which might easily be overlooked.
5. There is more time for safety training.
6. Systematic inspection of elevator cables, chain hoists, and similar equipment can be conducted during such a period.
7. This is a good time to inaugurate a program of in-

spection of drawings of all new dies and tools, including safety device features.

During a period of slack work employees find difficulty in maintaining the rhythm necessary for safety and satisfactory production. If a man has been off for a day or two, a certain period of readjustment is necessary when he returns to work. This is the dangerous time as far as accidents are concerned. Constant care and watchfulness must be exercised to see that he does not lapse into careless habits of work. During such a period it follows necessarily that the safety division will have to continue its work with a reduced personnel. If other personnel workers can make it their business to take up this slack, the safety program can be maintained and the safety record of the plant will not be endangered.

The employment division will co-operate by emphasizing the importance of safety at the time of employment, transfer, and other contacts with employees. The medical division will assist in the analysis of accident causes, preparation of literature, talks before employee groups, and by personal contact. The training division will emphasize safe practices in every training program. The employee service division will assist in the educational program through the use of posters and publicity in the employees' publication, safety playlets on recreational programs, etc. Safety is such an important part of the industrial relations program that I believe the thread of safety and safe practices should run through all the contacts of personnel workers with employees.

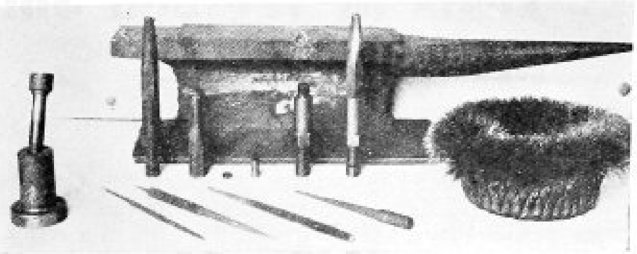
Welding and Cutting Tips Repaired

Oxy-acetylene welding "tips," as they are called in railway-shop parlance, become dirty, filled with carbon, or have enlarged orifices, after a certain period of service, and must be cleaned and repaired or replaced. Evidence of the necessity for this work is usually afforded by backfiring, shortened flame or generally inefficient torch operation.

The following method of conditioning torches for further service is followed at one midwestern railroad shop, typical repaired parts being shown in the illustration. First, the mixing head, with enclosed injector and retaining cap screw, and the copper tip, are separated from the torch proper and from each other, being immersed in a boiling Oakite solution for 12 to 18 hours to dissolve or loosen all deposits of carbon. The head is cleaned externally and the cap and injector removed. The loosened carbon is blown out as thoroughly as possible with compressed air and any remaining carbon removed with one of the tools shown in the foreground of the illustration.

The copper tip is cleaned inside and out and a special hand reamer used to take off any carbon that sticks to the walls of the mixing chamber. Care is exercised not to remove any metal from the interior walls of the tip. The tip is then lightly swaged on a tapered recess in the anvil shown in the background, until the hole in the extreme end is smaller than when new. This hole is then reamed out to the original size, assuring a round hole of the correct diameter to give the best results with the particular size of welding head and tip being repaired.

The injector is cleaned with a No. 60 wire brush mounted on a motor-driven emery wheel and the carbon removed from the interior with a pointed tool. Care is taken not to enlarge the hole beyond the original diameter and to keep the same injector with the head in which it was originally used. With all parts thoroughly cleaned, the head and tip are reassembled, ready for further service. They are polished with the wire brush. The bottom of the tip is then faced off slightly with a file and the tip is ready for service.



Typical oxy-acetylene cutting and welding tips and the tools used in cleaning and repairing them

Oxy-acetylene cutting tips, comprising, in the type illustrated, an external shell and an internal nozzle, are also cleaned and repaired in a similar manner. These tips start backfiring and will not maintain a constant flame, once they become dirty. The external shell is swaged in a special die, as shown at the left in the illustration, which slightly and uniformly closes the hole in the end. The internal nozzle holes are cleared out with hand drills to the original size and the tips reassembled ready for service.

In one month at the shop mentioned, the following parts were cleaned and repaired: Five No. 8 welding heads and tips, five No. 10 welding heads and tips, nine No. 12 welding heads and tips, 27 No. 15 welding heads and tips, 29 No. 2 external cutting tips, 25 No. 2 internal cutting tips and four No. 3 external cutting tips. The total cost of this work was about $73, which may be compared with a cost for the same parts, new, of approximately $500. It will not be maintained, of course, that the repaired tips are the equivalent of new tips, from the point of view of potential service life, but the work done has placed them in condition to give effective service for another more or less extensive period.

Operation and Maintenance of C. P. R. Locomotive No. 8000

(*Continued from page 231*)

motive was first put into service, the breaking in period on eastern lines, its transfer in knocked-down condition and re-assembly on the western lines and the various adjustments required while the locomotive was in regular operation on its assigned location on the mountain sub-division has not made it possible to give a complete summary of the operating performance of the locomotive over the entire period since the locomotive was built. The engine performance has been closely followed by observers and results show that a fuel saving of 14.8 percent has been effected on this division under regular service conditions at slow speeds on 2.2 percent grades between Albert Canyon, B. C., and Glacier, B. C., eastbound and Beavermouth, B. C., and Glacier westbound. Tests conducted on the eastern lines over the level Winchester sub-division between Montreal and Smith Falls showed fuel economies of 25 percent under higher speed and heavy tonnage.

Two of the tables show the performance of locomotive No. 8000 as compared with locomotives of the No. 5900 (T-1 class) on the heavy grades of the Mountain division, both east and west bound.

Tests of Welded Joints in Wide Plates* ▲ ▲ ▲

The Engineering Experiment Station of the University of Illinois, Urbana, Ill., recently completed tests on joints in wide plates. This article, the fifth in a series, describes the test specimens and results of tests for determining the strength of welds and the strength of diagonal joints. Conducted by Professor Wilbur N. Wilson, James Mather and Charles O. Harris, these experiments were made possible through the co-operation of the Chicago Bridge & Iron Works, Chicago, Ill.

Two specimens having welded lap joints and two combination welded-and-riveted lap joints were tested in the 1928 series. The specimens have the same general dimensions as those shown in Fig. 6 on page 139 of the July issue. The section at the middle is 20½ inches by ½ inch and the rivets have a nominal diameter of ¾ inch. Figs. 1 and 2 show specimens after failure. The results of the tests are given in Table 1. The failure of the rivets to add naturally to the strength of the joint consisting of both welds and rivets is attributed to the fact that the weld is much more rigid than the rivets, so that the two did not act together.

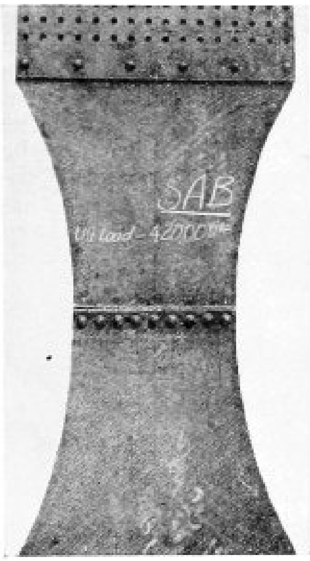


Fig. 1.—Combination welded and riveted lap joint

Fig. 2.—Welded lap joint of 1928 series after failure

Fig. 3.—Welded butt joint of 1930 series after failure

Two wide specimens of the 1930 series contained welded butt joints. The plates were 64 inches by ⅝ inch. Fig. 3 shows a specimen after failure. In preparing the plate for welding, the ends were machined to a bevel of 60 degrees and a backing strip was used. After the bead had been run, the backing strip and the bead were chipped flush with the plate so that the section through the weld was the same as through the original section of the plate. The weld was made with a hand-operated electric arc, and a coated rod was used. The ultimate strength for the two specimens was 2,050,000 and 2,100,000 pounds, respectively. Specimen AW1 broke outside of the weld and AW2 in the weld, although the latter developed a higher stress than the former. The efficiencies for the two specimens were 96.4 and 99.4 percent, values that compared favorably with the efficiencies of wide plates of from 85 to 98 percent, given in Table 2 on page 140 of the July issue. The width of the specimens decreased from 64 inches to 60½ inches for AW1 and to 61½ inches for AW2. The curvature of the edge, originally straight, is apparent in Fig. 3.

Specimens containing welded lap joints in plates 11½ inches by ⅜ inch were included in the 1930 series. These plates were welded in the vertical position to simulate the field conditions encountered in assembling a tank. Fig. 4 shows the dimensions of the specimens and also the parent plate for these specimens. The plates A1 and A2 are full-width control specimens. Control specimens 1½ inches wide were cut, one each, from CA1, CA2, CB1, CB2, CC1, and CC2. The results of the tests are given in Table 2. The specimens having a transverse weld developed 92.74 percent of the strength of the wide plate without joint, whereas the specimens having a diagonal weld developed 99.90 percent of the strength of the continuous plate. Fig. 5 shows the specimens after failure.

The difference between the efficiencies of the joints reported in Tables 1 and 2 is attributed to the development in the art of welding during the period between the dates when the two series of tests were made, particularly to the development in the shop where these specimens were made. There was some development in apparatus and in welding rods but of considerably more importance was the development of experience and skill on the part of the welder. The quality of hand welding must of necessity depend primarily upon the skill and integrity of the workman. The quality of weld made by one workman is no indication of the quality of weld that will be made by another workman using the same rod and the same apparatus.

Two types of specimens contained diagonal joints; one type was a welded lap joint and the other a riveted lap joint. The strength of the welded joint is given in Table 2. It developed an efficiency, based on the strength of a plate similar to the one welded, of 99.90 percent.

The riveted joint was 11½ inches by ⅜ inch and had two diagonal rows of ¾-inch rivets as shown in Fig. 6. The rivets were so located that there were not more than two on any transverse section. Besides the riveted joints there were two control plates 11½ inches wide without joints and three standard tension control specimens, 1½ inches wide. The location of the various specimens in the parent plate is shown in Fig. 6.

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* Fifth of a series of articles published through the courtesy of the Engineering Experiment Station, University of Illinois, Urbana, Ill.

Table 1.—Strength of Welded Lap Joints, 1928 Series

Specimen Number	Kind of Joint	Ultimate Strength, pounds	Ultimate Load in pounds per inch length of bead	Strength of Control Specimens, pounds per square inch	Efficiency* of Joint, percent	Manner of Failure
SAA.....	Rivet and Weld	451,600	22,030	52,192	84.42	Plate tore outside of joint
SAB.....	Rivet and Weld	420,000	20,490	51,761	79.16	Weld failed
SAC.....	Rivet and Weld	454,000	22,150	51,879	85.38	Weld failed, followed by plate
Average		441,867	21,550	51,944	82.99	
SCA.....	Weld	428,000	20,880	57,277	72.85	Weld tore
SCB.....	Weld	401,000	19,560	50,908	76.85	Weld tore
SCC.....	Weld	426,000	20,780	54,393	76.41	Weld sheared from plate
Average		418,333	20,450	54,193	75.37	

---

* Efficiency of joint is ratio of actual strength to product of area of cross section and unit strength of control specimens.

Table 3.—Strength of Riveted Diagonal Joints, 1930 Series

Specimen Number	Kind of Specimen	Ultimate Strength, pounds per square inch	Remarks
1, 2, 3.....	Standard tension specimens 1½-inch wide	56,300	Reduction of area, 66.1 percent; elongation in 8-inch, 31.8 percent
P1-A	11½-inch plate, no joint	54,000	
P1-B	11½-inch plate, no joint	53,750	Reduction of area, 52.40 percent
Average		53,875	
P2-A	Riveted diagonal joint	48,400*	Diagonal tear along rivet row
P2-B	Riveted diagonal joint	48,450*	Diagonal tear along rivet row
Average		48,425	

---

* Based on transverse section with two 13/16-inch holes out.

Table 2.—Strength of Welded Lap Joints, 1930 Series

Specimen Number	Kind of Joint	Strength of 1½-inch Control Specimens, pounds per square inch	Ultimate Load		Efficiency of Weld Based on A1 and A2 as 100 percent	Reduction of Area, percent	Type of Failure
			Total pounds	Pounds per square inch			
A1	No Joint	55,100	227,100	53,500	....	35.30	
A2	No Joint	52,390	219,200	52,500	....	39.50	
Average..............		53,745	......	53,000	....	37.40	
B1	Transverse Lap Weld	55,220	218,000	51,300	96.90	21.60	Transverse failure at edge of weld
B2	Transverse Lap Weld	55,590	199,500	47,000	88.68	20.90	Transverse failure at edge of weld
Average..............		55,405	......	49,150	92.74	21.25	
C1	Diagonal Lap Weld	55,460	222,000	51,500	97.17	35.00	Transverse failure beginning at end of weld
C2	Diagonal Lap Weld	55,080	227,700	54,400	102.64	34.20	Transverse failure beginning at end of weld
Average..............		55,270	......	52,950	99.90	34.60	

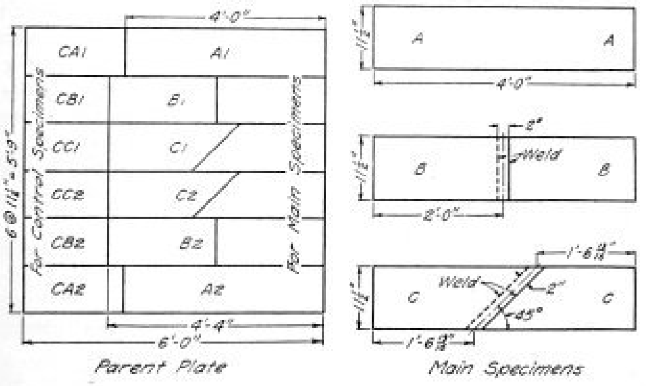


Fig. 4.—Parent plate and welded lap joints for 11½-inch specimens

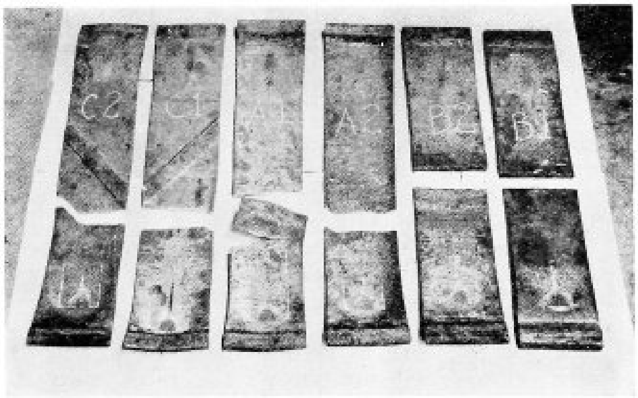


Fig. 5.—Welded lap joints after failure

The results of the tests are given in Table 3. The efficiency of the joints was 89.88 percent based on the 11½-inch control specimens and 86 percent based on the standard tension control system. This latter is comparable with an efficiency of about 76 percent which is what can reasonably be expected from a transverse lap joint in a ⅜-inch plate using ¾-inch rivets. The diagonal seam is longer than the transverse seam, making a longer joint to calk, but no more rivets or lap of plate is required, since the transverse joint has three rows of rivets whereas the diagonal one has only two rows.

A discussion of the effect of arc welding upon the strength of plates as well as the strength of field welds will appear in the December issue, which will conclude this series of articles on the tests of joints in wide plates. The results of tests reported in these articles will be summarized to justify a number of interesting conclusions.

(To be continued)

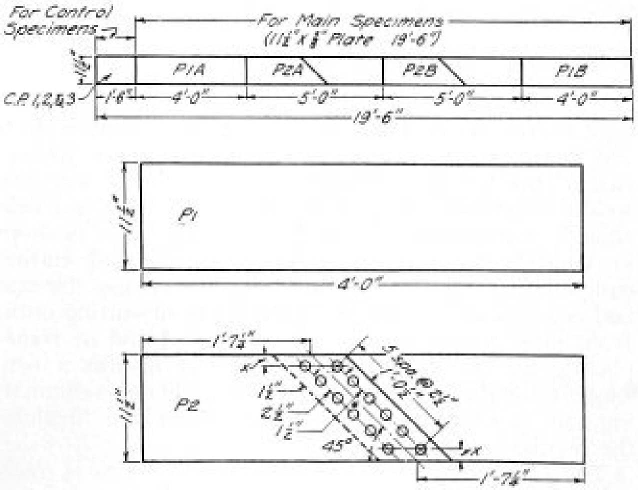
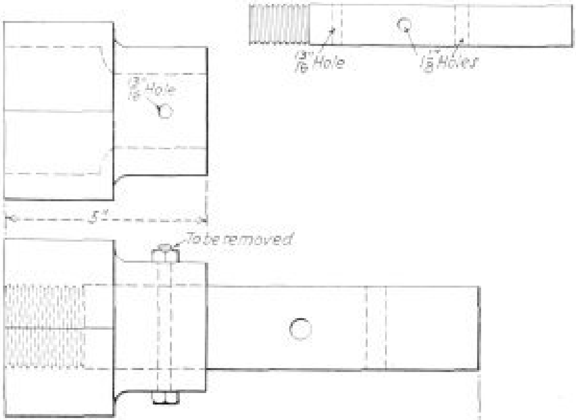


Fig. 6.—Riveted diagonal joint

Wrench for Removing and Applying Blow-Off Valves

Shown in the sketch is a wrench for removing or applying blow-off valves. The socket is made of soft steel 5 in. long and turned on a lathe to the design shown. It is bored to a depth of 2 in. and made hexagon in the smith shop to suit the size of the valve. Care should be taken to have sharp corners in the hexagon socket as it is sometimes necessary to use a hammer on the wrench to loosen the valve so that it will turn.

A 2-ft. length of 2-in. pipe, threaded at one end and drilled for one 13/16-in. hole and two 1⅛-in. holes as shown, is required. The socket is placed over the hex portion of the blow-off valve and the 2-in. pipe is screwed into the valve connection with the hands and turned so that the 13/16-in. hole in the pipe coincides with the hole in the body of the socket. A ¾-in. bolt is then inserted which secures the pipe to the socket so that the two will turn together. A 1-in. iron buggy bar is then inserted through one of the two 1⅛-in. holes for turning the wrench.

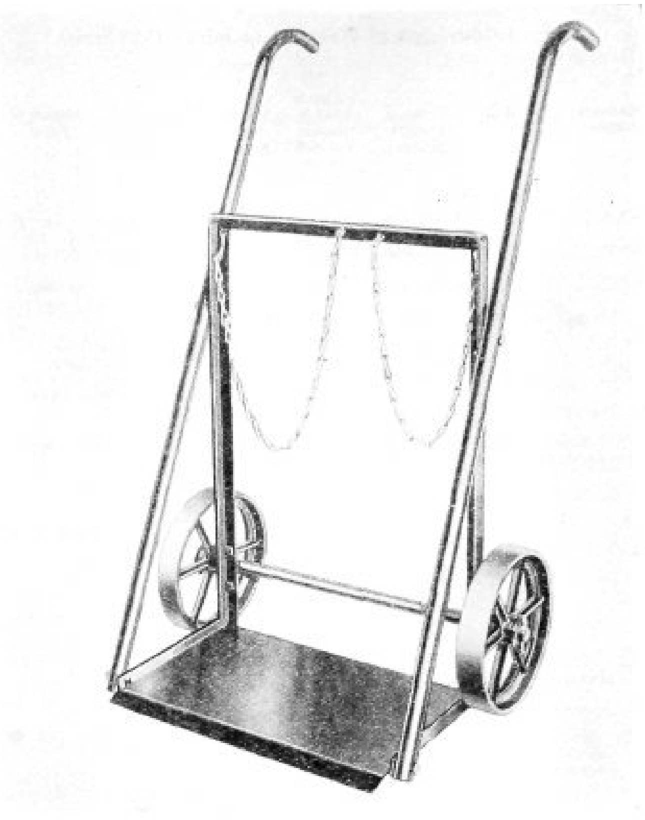


Blow-off valve wrench—The socket is made hexagon to suit the valve

This wrench, which is used in the shops of a southeastern railroad, will not slip off the valve under any condition of service or application.

Truck for Gas Welding and Cutting Equipment

The Linde Air Products Company, 30 East 42nd Street, New York, has introduced a new cylinder truck for distribution by Prest-O-Weld and Purox jobbers east of the Rocky Mountains. This new oxy-acetylene welding accessory, known as the No. 5 cylinder truck, should be of considerable interest to welders and to shops in which portable oxy-acetylene welding and cutting equipment is used. This truck greatly increases the ease and convenience of moving the welding or cutting outfit from place to place in the shop or yard and of transporting the outfit to outside jobs. It insures a firm support for the oxygen and acetylene cylinders, eliminating any possibility of overturning them and breaking the regulators or gages.



No. 5 cylinder truck for gas welding equipment

The frame of the new No. 5 Cylinder Truck is made of 1¼-inch angle iron welded into a permanently rigid unit. A beveled steel plate welded to the frame forms the cylinder platform. The handles are made of 1¼-inch pipe curved to provide a firm, easy grip. The handles are firmly bolted to the cylinder platform and to the upper part of the frame. Being bolted, they may be removed very easily, which permits the truck to be partly dismantled for shipping or storing in a small space.

The cylinders are held firmly in position on the truck by means of chains which are adjustable to accommodate cylinders of different sizes.

The truck has 12-inch cast-iron wheels with 2-inch tires. These facilitate easy handling of the truck especially when swinging it from the upright to the rolling position. The axle is bolted to the frame, permitting rapid dismantling for shipment or storage.

The overall size of the No. 5 cylinder truck is 48 inches high by 30 inches wide. Its weight is only 80 pounds. It is attractively finished with a durable, rust-preventing black enamel which will not readily chip.

Heat Vaporizes Heavy Oil, Explodes Tank

Three men, two of them city firemen, were recently killed at Knoxville, Tenn., by the explosion of oil vapor in a large storage tank. The accident occurred under unusual circumstances. Workmen who had been ordered to empty the tank found that the cold weather had thickened the oil so much that it would not flow, so they built a fire under the tank. This fire got beyond their control, forcing them to call on the city fire department for assistance. The firemen had just arrived when oil vapor inside the tank became ignited, hurling the 16,000-gallon receptacle endwise from its concrete saddle and sending the head crashing through the side of a nearby sheet-metal building. The victims were struck by the head.—*The Locomotive.*

The Boiler Maker

VOLUME XXXII NUMBER 11

PUBLISHED Monthly by the Simmons-Boardman Publishing Company, 1309 Noble Street, Philadelphia, Pa., with Executive Offices at 30 Church Street, New York City.

Samuel O. Dunn, Chairman of Board, Henry Lee, President. C. R. Mills, Vice-President. L. B. Sherman, Vice-President. Roy V. Wright, Vice-President and Secretary. F. H. Thompson, Vice-President. E. T. Howson, Vice-President. F. C. Koch, Vice-President. John T. DeMott, Treasurer. George Slate, Vice-President and Business Manager.

Chicago: 105 W. Adams Street. Washington, D. C.: 17th and H Streets, N. W. Cleveland: Terminal Tower. San Francisco, Cal.: 215 Market Street.

Subscription price in United States and possessions, 1 year, $2.00, 2 years, $3.00; Canada, including duty, 1 year, $2.50, 2 years, $4.00; foreign countries, 1 year, $3.00, 2 years, $5.00. Single copies, 35 cents each.

Request for change of address should reach us on or before the 15th of the month, preceding the issue with which it is to go into effect. It is difficult and often impossible, to supply back numbers to replace those undelivered through failure to send advance notice. In sending us change of address, please be sure to send us your old address as well as the new one.

THE BOILER MAKER is a member of the Associated Business Papers, Inc. (A. B. P.), and the Audit Bureau of Circulations, (A. B. C.).

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Contents

Communications

Container for Staybolt Taps

To the Editor:

It is a pity, indeed, to have nice staybolt taps and reamers knocked around in a boiler maker's tool box, i. e., a nail keg, to be dulled by rough contact with other tools. The average staybolt reamer is from 18 to 25 inches in length, and the sharp, projecting threads are easily and quickly damaged by rough handling. In some shops these taps and reamers have little compartments provided for them in special boxes, but it would require the services of half the Volstead enforcement squads to keep the tools in their boxes while not in use.

The best way of keeping these taps and reamers in good shape—or, at least, the best way the writer has seen—is to procure a length of brass tubing long enough to barely contain the longest tool in the set. The diameter of the tube should be at least 4 inches so that the hand can be easily inserted to pull out the tool wanted.

Close the lower end of the tube with a screwed cap to which has been attached a flange or plate large enough to allow the tube to stand upright without being knocked over by a light touch. Fit a pipe cap to the upper end of the tube and make the threads in such manner that the cap will be oiltight when screwed on by hand. Fill the tube one-third to one-half full of whatever kind of oil is used in the shop for thread-tapping. Slide the taps and reamers carefully into the tube, put on the screwed cap, and the affair is ready for a job or for the tool room. A tap or a reamer, taken from the tube, is ready for instant use without the necessity of lubrication before putting the tool to work.

Indianapolis, Ind. James F. Hobart.

Replacing Broken Rivets Below a Ship's Waterline

To the Editor:

The boiler maker is often asked to fix leaks in the hull of a ship as a result of loose rivets or faulty calking. These leaks often occur in the frames; and it is customary to drill a hole, tap it, apply the putty gun and inject red lead putty. Where loose rivets occur, this injecting is useless; the rivets must come out and be replaced.

Here is a method of replacing loose and broken rivets below the waterline of a ship which has saved the cost and time of docking the ship as well as saving the cargo. Take the number of countersunk bolts which you may require and screw the nuts well down so that they will run easily with your fingers. Now get a lath and cut or pare it lengthways so that it will be small enough to pass through the hole where the broken or loose rivet is. Tie a long, strong string to one end of the lath and get the bolt and washers ready; now cut the head off the bad rivet and back it out of the hole, making sure that the hole is drifted large enough to allow the bolt to enter easily. One has to be lively at this job if he can't swim. Now have your helper go on deck and go, in a small boat, above the position where you knocked the rivet out. You now force the lath out through the hole and it will float up to the surface of the water, carrying the string with it. Your helper will catch the string and tie the bolt at the end (not the head). You will then pull in the string and fish the bolt into the rivet hole. Apply a good red lead grommet and some putty on the bolt; now put on one or more washers and screw the bolt up good and tight. Replace any other rivets in the same manner.

A good method of keeping the water out after pushing the lath through the hole is to fill up the hole with a stick of hard tallow, which breaks the force of the water until you get the bolt fished into the hole. This repair is only temporary, as the rivets must be replaced when the ship is dry docked. If you get a chance to try this repair, you will find it exciting.

Montreal, Canada. James Wilson.

Questions and Answers Pertaining to Boilers

This department is maintained for the purpose of helping those who desire assistance on practical boiler shop problems. Inquiries should bear the name and address of the writer. Anonymous communications will not be considered. The identity of the writer, however, will not be disclosed unless the editor is given special permission to do so.

By George M. Davies

Pitch and Size of Rivets

Q.—Kindly explain in the simplest way how to find the pitch and size of rivets on boilers and tank work.—G. F.

A.—The answer to this question will be found in the October issue under the heading "Pitch and Size of Rivets."

Correcting Fire Cracks

Q.—What is the best method of correcting fire cracks on the round seam of an 84-inch diameter boiler, the seam being directly over the fire? The cracks are from the calking edge to the rivet holes for a distance of about 60 inches. Would chipping them out and welding, then countersink the holes and redriving the rivets be a good method? If a patch would be the right method, what type of patch would it take? Should the patch be on the inside or outside of the boiler, the working pressure of the boiler being 150 pounds? The type of joint on the round seam is according to sketch herewith. What causes fire cracks? Do you think a different style rivet head on the outside of the boiler would prevent fire cracks?—G. W. M.

A.—Fire cracks on girth seams, from the calking edge to the rivet holes may be welded as illustrated in Fig. 1. Procedure:

(1)—Remove rivet and "V" out crack.

(2)—Butt weld as shown.

(3)—Ream out hole for reapplying rivet after welding.

Due to the fact that welding is not generally accepted by the various states it is advisable to have the inspector, under whose jurisdiction the boiler comes, inspect the condition and obtain his permission to weld the cracks before proceeding.

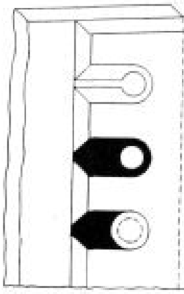


Fig. 1.—(Left) Fire cracks on girth seams may be welded in this manner. Fig. 2.—(Below) Details for designing diagonal patches on return tubular boilers

Should a patch be required, it can be designed from instructions given in Fig. 2. The patch should be applied on the inside wherever possible, thereby eliminating mud pockets.

Fire cracks in seams having cap joints are due to local stresses set up by the differences between the temperatures of the fire and the water sides of the joints. I do not believe that a different style rivet head on the outside of the boiler would prevent fire cracks, however counter-sunk head rivets would be the type of head to use.

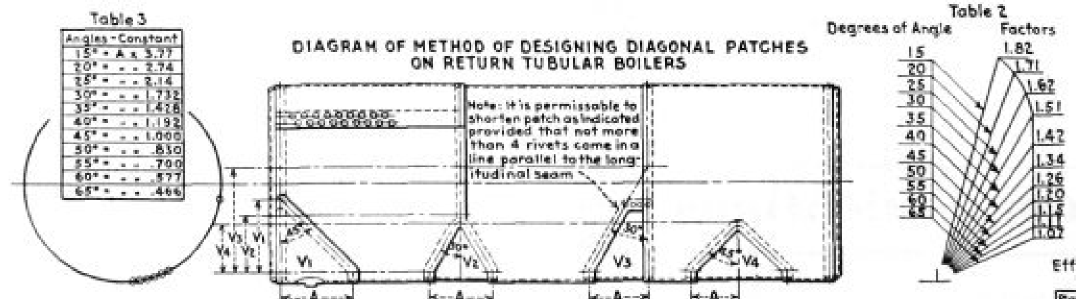


Table 3

Angles - Constant
15° = A x 3.77
20° = " " 2.74
25° = " " 2.14
30° = " " 1.732
35° = " " 1.428
40° = " " 1.192
45° = " " 1.000
50° = " " .850
55° = " " .700
60° = " " .577
65° = " " .466

Table 2

Degrees of Angle	Factors
15	1.82
20	1.71
25	1.62
30	1.51
35	1.42
40	1.34
45	1.26
50	1.20
55	1.15
60	[illegible]
65	1.07

To determine Roundabout length of Patch:

First-determine length A. Then multiply A by the constant in Table 3 corresponding to the angle obtained from Table 2. This gives the vertical height of the patch as shown at V, V1, etc. which should be marked-on the boiler shell. It must be noted that this height is measured from a point level with the ℄ of the highest rivet in the short roundabout seam

Example: Patch at roundabout seam on 54" diameter boiler. Shell and patch 5/16" plate. 55,000 TS Long seam, double riveted lap, 3/4" rivet hole. 2 7/8" pitch 73.9%. Select pitch of rivets to be used in patch from Table I say 1 7/8 and 15/16" hole = 56%. A = 24". Then 73.9 ÷ 56 = 1.32 nearly. The next higher factor in Table 2 is 1.34 which corresponds to 40°. By Table 3, an angle of 40° gives a costant 1.192 A x 1.192 = 24 x 1.192 = 28 5/8 inches = height V

The effective % of the patch seam is 56 x 1.34 = 75.24 which is stronger than the original longitudinal seam.

Draw the diagonal lines which locate the ℄ of rivet in the patch. Lay out all laps at 1½ times the diameter of rivet hole used.

Caution:

When patch rivet holes are countersunk, proper allowance must be made for the metal that is removed by increasing the pitch as shown below

Plate	For 45° Countersunk	
1/2"	Add to Pitch	1/4"
7/16"	" " "	3/16"
3/8"	" " "	3/16"
5/16"	" " "	1/8"
1/4"	" " "	1/8"

Countersink must be not deeper than 3/4 thickness of plate.

To obtain the Patch Angle required:

Divide the efficiency of the longitudinal seam found on the certificate of inspection by the efficiency of the patch seam selected from Table I. This gives the minimum factor required to maintain the strength of the shell. Take from Table 2, the angle corresponding to this factor or next higher factor which gives the required angle of the patch.

Note: Firebox steel must be used to replace firebox steel in repairing boilers.

The use of tank steel is strictly prohibited.

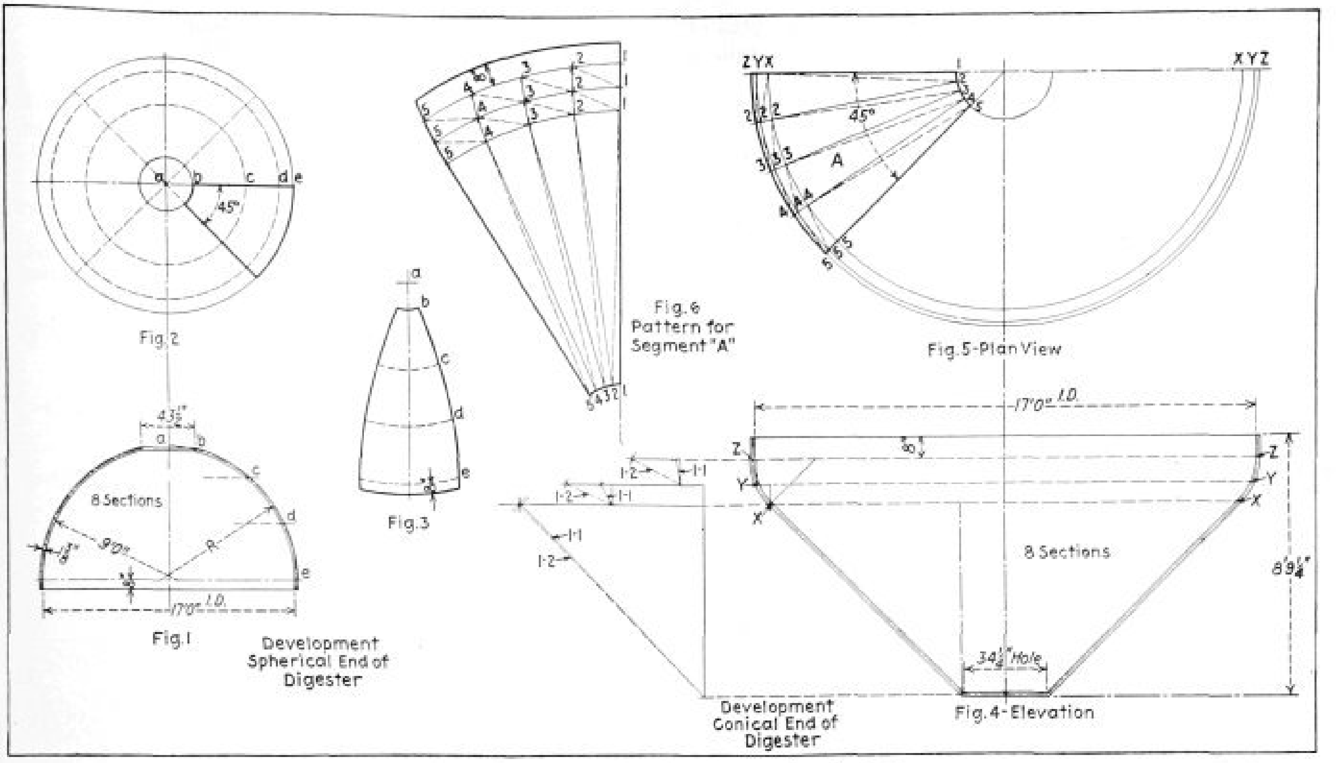
Rivet holes for patches shall be drilled full size with patch in position or may be punched, not to exceed 1/4" less than full size for plates over 5/16" and 1/8" less than full size for plates 5/16" or less in thickness and then reamed to full size with patch in place.

Table I
Efficiences of Single Riveted Seams

Plate	1/2	7/16	3/8	5/16	1/4	Net Sec.
Hole	1 3/4 Pitch					
15/16						46
13/16						53
11/16			45	54		60
	1 13/16 Pitch					
15/16						48
13/16						55
11/16			43	52		62
	1 7/8 Pitch					
15/16						50
13/16	44	50				56
11/16			42	50		63
	2 Pitch					
15/16						53
13/16	41	47	55			59
11/16			38	47	59	65
	2 1/8 Pitch					
15/16	52					55
13/16	39	44	52			61
11/16			37	44	55	67
	2 1/4 Pitch					
15/16	49	56				58
13/16		42	49	53		63
11/16			35	42	52	69

The welding of patches on shells or drums of boilers is strictly prohibited

Tensile Strength 55,000 lb.
Rivet Shear 44,000 lb.



Method of developing spherical and conical ends of digesters

Development of Ends of a Digester

Q.—Will you publish through the Questions and Answers Department of THE BOILER MAKER an accurate method and simple way to develop the spherical and conical ends of a digester as per the dimensions on the enclosed sketch.—G. E. L.

A.—To develop the spherical and conical ends of the digester submitted with the question, first take the spherical end as shown in Fig. 1. To develop this end, draw an arc through the center of the thickness of the plate, producing this arc to intersect the center line at the point *a*, Fig. 1. Mark off any convenient number of points on this arc as *b*, *c*, *d* and *e*, it is not necessary to have points equally spaced. Draw circles equal to the diameters at each of the chosen points, as in Fig. 2, and as the end of the digester is to be made with eight plates in the circumference, draw lines 45 degrees apart to indicate the circumferential width of the plates.

Next, with *a-b*, *a-c*, *a-d* and *a-e*, as measured along the arc *a-e*, Fig. 1, as radii, draw circles as in Fig. 3. Step off on each of these last arcs, Fig. 3, the length of the respective circles between the 45-degree lines in Fig. 2 and join the points so fixed by a curve.

The resultant figure will represent a development of the spherical end of the digester. To this add the 8-inch straight portion as shown in Fig. 3, completing the development.

The proof of the accuracy of this construction is that the distance from *a* in Fig. 1 to points *b*, *c*, *d* and *e*, is obviously equal all the way around the end of the digester, therefore on the flat plate these distances will be circular with radii equal to the lengths *a-b*, *a-c*, *a-d* and *a-e*, as in Fig. 3.

The width of the plate measured along the arcs in Fig. 3 is, of course, the length betwen the 45-degree lines measured along the circle, equal to the diameter of the end of the digester at the point in question.

If lap seams are used, an allowance should be made by providing a lap on the sides and bottom of the pattern. The plate thickness also must be duly considered.

To develop the conical end of the digester, the first step in the layout is to draw the elevation and plan of the conical end of the digester as shown in Figs. 4 and 5 respectively. The plan view is a projection of the neutral axis of the plate.

As the end of the digester is to be made with eight plates in the circumference, draw lines 45 degrees apart to indicate the circumferential width of the plates. Segment *A*, Fig. 5, represents one plate.

For a development of the pattern for segment *A*, the triangulation method may be employed, but, to apply it in this case, a number of planes must be passed through the object, as shown in the elevation, as at *x-x* and *y-y*. The sections through these planes are circular in form and should be shown as such in the plan view, Fig. 5. By this arrangement the segment *A* can be divided into triangles and their true lengths found with which the patterns may be laid off.

For example, segment *A* is prepared, the arcs along the bases *z-z*, *y-y*, and *x-x* being divided into four equal parts. Lines are drawn in from *1* to *1*, *2* to *2*, *3* to *3*, etc., and *1* to *2*, *2* to *3*, etc., their vertical positions being indicated in the elevation. To find their true lengths, draw right-angle triangles, using for the bases the distances between points 1-2, 2-2, etc., of the plan. The corresponding heights are shown projected from the elevation to the vertical legs of the triangles. With the hypotenuses and spaces from the plan and profile of the upper base complete, the pattern for the segment, Fig. 6, may be drawn.

This problem is somewhat out of the ordinary in that the piece is of a conical form, having a warped surface at the base in a part of it. This complicates the layout somewhat, but in all cases of this kind a very close approximation of the required patterns may be secured by the triangulation system of development.

To further increase as nearly as possible accurate work with the triangulation method, it is always advisable to divide views of any form, to be so developed, into a great number of triangular sections, as the greater the number the nearer will be the form of the resulting patterns to their actual given dimensions.

If lap seams are used, allowance should be made for by providing a lap on the sides and top of the pattern. The plate thickness also must be duly considered.

Circular Patch Efficiency

Q.—Will you please figure out in detail:

$$(1)\ \text{Efficiency} = \frac{S}{A}$$

$$(2)\ \text{Efficiency} = \frac{\text{Net section through } A\text{-}A}{\text{Solid plate through } A\text{-}A}$$

$$(3)\ \text{Efficiency} = \frac{N \times a \times 44{,}000 + TS \times \text{net sectional area of shell through } A\text{-}A}{TS \times \text{solid shell plate through } A\text{-}A}$$

After finding this efficiency would the working pressure for the patch be figured by the formula

$$\frac{\text{Strength of material} \times \text{thickness of plate} \times \text{efficiency of patch}}{\text{Radius of boiler} \times \text{factor of safety}} =$$

working pressure?

If so, should the thickness of plate be ½ inch for patch or 11/16 inch for shell?—R. H. L.

A.—Fig. 1 illustrates the circular patch in question.

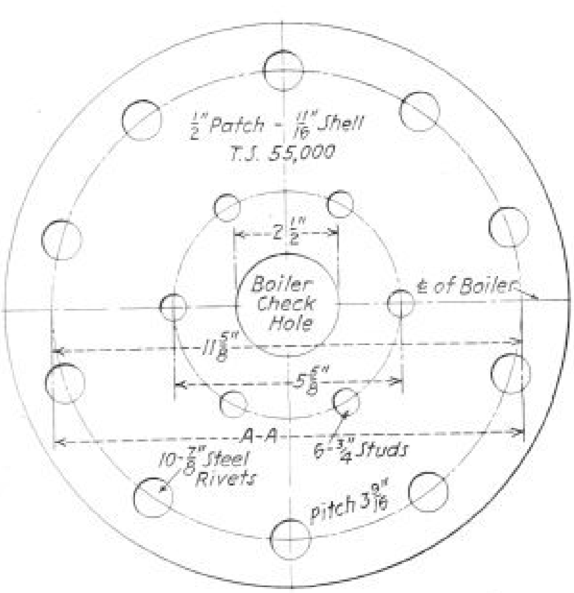


Fig. 1.—Circular patch problem

The efficiency is computed as follows:

$$(1)\text{—Efficiency} = \frac{S}{A},$$

where S = net plate measured on arch subtended by cord A, Fig. 2.

A = a cord equal in length to the radius of the rivet circle (Fig. 2) for circles of 12 inches or under.

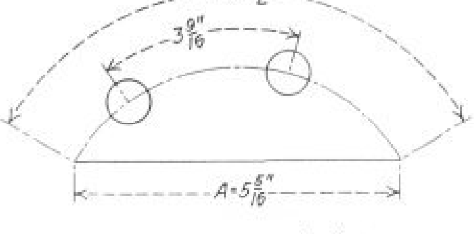


Fig. 2.—Computing patch efficiency

Thus, in Fig. 2, $A = 5\ 5/16$ inches and $S = 5½$ inches.

Substituting in the formula we have:

$$\text{Efficiency} = \frac{5.5 - (2 \times .9375)}{5.3125} = .682 = 68.2 \text{ percent.}$$

$$(2)\text{—Efficiency} = \frac{\text{net section through } A\text{-}A}{\text{solid plate through } A\text{-}A}$$

Net section through A-A is taken as the combined section of the shell and patch.

Solid plate through A-A is taken for the shell only.

Substituting, we have:

$$\text{Efficiency} = \frac{11.625 - (2.5 + 2 \times .75) \times 1.1875}{11.625 \times .6875}$$

$$\text{Efficiency} = \frac{9.0546}{7.992} = 1.13 = 113 \text{ percent.}$$

$$(3)\text{—Efficiency} = \frac{N \times a \times 44{,}000 + TS \times \text{net sectional area of shell through } A\text{-}A,}{TS \times \text{solid plate through } A\text{-}A}$$

where, N = number of rivets in single shear (5).

TS = tensile strength of plate, 55,000 pounds.

a = cross-sectional area of rivet after driving, square inches.

Substituting, we have:

$$\text{Efficiency} = \frac{5 \times .69029 \times 44{,}000 + 55{,}000 \times 11.625 - (2.5 + 2 \times .75) \times .6875}{55{,}000 \times 11.625 \times .6875}$$

$$= \frac{440{,}183}{439{,}570} = 1.00 = 100 \text{ percent.}$$

The least efficiency obtained in (1), (2) or (3) should be taken as the efficiency of the patch:

(1) = 68.2 percent.
(2) = 113 percent.
(3) = 100 percent.

The efficiency of this patch would be 68.2 percent.

The formula for the working pressure of the boiler as stated in the question is correct.

The thickness of the plate to be used in this formula is the thickness of the shell plate (11/16 inch in this case).

Steam Pressure of Locomotive Boiler

Q.—How do you find the total steam pressure on a locomotive boiler?—G. F.

A.—The total steam pressure on a locomotive boiler (allowable working pressure) is obtained by computing the allowable working pressure based on the strength of the various parts of the boiler, such as:

(1)—Shell
(2)—Braced and stayed surfaces
(3)—Stays and staybolts
(4)—Wrapper sheet
(5)—Dome.

Formulas for determining the allowable working pressure based on the strength of any of the above may be found in the A.S.M.E. Code, Section III Locomotive Boilers.

The least allowable working pressure obtained from these calculations would be the total steam pressure allowed on the boiler.

Associations

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Assistant Chief Inspectors—J. M. Hall, Washington, D. C.; J. A. Shirley, Washington, D. C.

Bureau of Navigation and Steamboat Inspection of the Department of Commerce

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Assistant International President—J. N. Davis, Suite 522, Brotherhood Block, Kansas City, Kansas.

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States and Cities That Have Adopted the A.S.M.E. Boiler Code

States

Arkansas
California
Delaware
Indiana
Maryland
Michigan
Minnesota
Missouri
New Jersey
New York
Ohio
Oklahoma
Oregon
Pennsylvania
Rhode Island
Utah
Washington
Wisconsin
District of Columbia
Panama Canal Zone
Territory of Hawaii

Cities

Chicago, Ill.
Detroit, Mich.
Erie, Pa.
Evanston, Ill.
Houston, Tex.
Kansas City, Mo.
Los Angeles, Cal.
St. Joseph, Mo.
St. Louis, Mo.
Scranton, Pa.
Seattle, Wash.
Memphis, Tenn.
Nashville, Tenn.
Omaha, Neb.
Parkersburg, W. Va.
Philadelphia, Pa.
Tampa, Fla.

States and Cities Accepting Stamp of the National Board of Boiler and Pressure Vessel Inspectors

States

Arkansas
California
Delaware
Indiana
Maryland
Minnesota
Missouri
New Jersey
New York
Ohio
Oklahoma
Oregon
Pennsylvania
Rhode Island
Utah
Washington
Wisconsin
Michigan

Cities

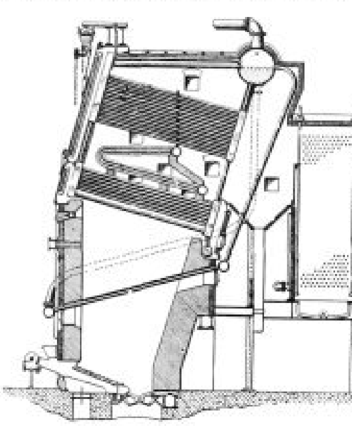
Chicago, Ill.
Detroit, Mich.
Erie, Pa.
Kansas City, Mo.
Memphis, Tenn.
Nashville, Tenn.
Omaha, Neb.
Philadelphia, Pa.
Parkersburg, W. Va.
St. Louis, Mo.
Scranton, Pa.
Seattle, Wash.
Tampa, Fla.

Selected Boiler Patents

Compiled by Dwight B. Galt, Patent Attorney, 1372-1376 National Press Building, Washington, D. C. Readers wishing copies of patents or any further information regarding any patent described, should correspond directly with Mr. Galt.

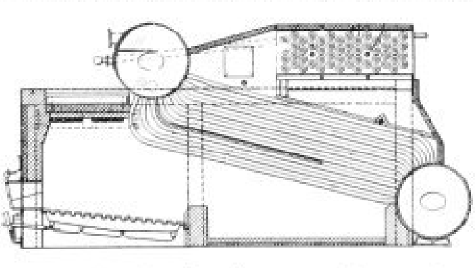
1,790,750. STEAM BOILER. DAVID S. JACOBUS, OF JERSEY CITY, N. J., ASSIGNOR, BY MESNE ASSIGNMENTS, TO FULLER LEHIGH COMPANY, A CORPORATION OF DELAWARE.

Claim.—A watertube boiler having a bank of watertubes spaced across the bank, a drum and connections between the ends of the tubes and the drum to form a circulatory system through said watertubes, a furnace to supply hot gases to said watertubes, the distance between the source of heated furnace gases and the entrance to the bank of tubes being sufficient to permit substantially complete combustion before the gases enter said bank, rows of slag screen tubes extending across the furnace between said source and said bank of tubes, and over which the gases flow before reaching said bank, means associated with said slag screen tubes constructed and arranged to cause a change in the direction of the gases and adapted to receive deposit therefrom, the gas flow spaces between the tubes in said rows of slag screen tubes being greater than the gas flow spaces between the tubes in said bank, and connections between the ends of the tubes in said rows and the drum to form a circulatory system through the tubes of said rows, independent of the circulation through said bank, substantially all of the combustion air being admitted below said rows of slag screen tubes. Eighteen claims.



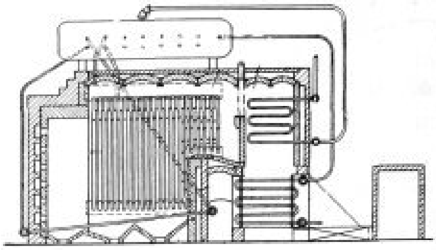
1,790,593. STEAM GENERATOR. JOSEPH JOHN NELIS, OF BROOKLYN, N. Y., ASSIGNOR TO FOSTER WHEELER CORPORATION, OF NEW YORK, N. Y., A CORPORATION OF NEW YORK.

Claim.—A steam generator of a type having a low center of gravity and requiring only limited head room, and comprising an upper transverse steam and water drum, a lower transverse water drum a substantial distance rearwardly thereof and connected thereto by a plurality of horizontally inclined watertubes, a combustion chamber projecting in front of the steam and water drum and extending rearwardly therefrom to a point adjacent said lower water drum, an inclined baffle extending rearwardly between the tubes from the steam and water drum, a second inclined baffle extending forwardly immediately above the uppermost watertubes, said baffles overlapping each other and providing a lower rearwardly and downwardly extending gas pass and an upper forwardly and upwardly extending gas pass, an economizer comprising a plurality of transversely extending watertubes positioned above said second baffle and arranged to receive gases passing out of the forward end of said upper gas pass, the overall height of the generator at said economizer being substantially the same as at said steam and water drum, a baffle extending horizontally in said steam and water drum above the points of connection of said watertubes to said drum, feed water conduits connecting the forward tubes of said economizer to said steam and water drum below said drum baffle, and a steam outlet connection from said steam and water drum above said drum baffle. Two claims.



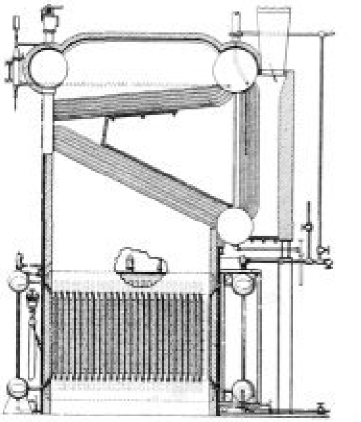
1,790,396. BOILER IN WHICH THE STEAM IS GENERATED INDIRECTLY. FRIEDRICH WEMPE AND HEINRICH PEPERKORN, OF CASSEL-WILHELMSHOHE, GERMANY, ASSIGNORS TO SCHMIDT'SCHE HEISSDAMPF-G. M. B. H., OF CASSEL-WILHELMSHOHE, GERMANY, A CORPORATION OF GERMANY.

Claim.—In a steam generating plant, a furnace construction comprising a combustion chamber, a vertical flue having at least two superposed heat receiving units disposed therein, a horizontal flue connecting the combustion chamber with the vertical flue, a plurality of heat receiving tubes within the combustion chamber, said tubes being arranged to form the side walls, rear wall, and roof of the combustion chamber and thereby receiving heat mainly by radiation, a plurality of tubes arranged in said horizontal flue and thereby receiving heat mainly by contact with the hot gases of combustion, and a substantially vertical damper member disposed between the horizontal and vertical flues, said member being vertically adjustable to cause the hot gases to pass either over or under said member and being arranged for movement at a height substantially corresponding to that of the upper heat receiving unit. Four claims.



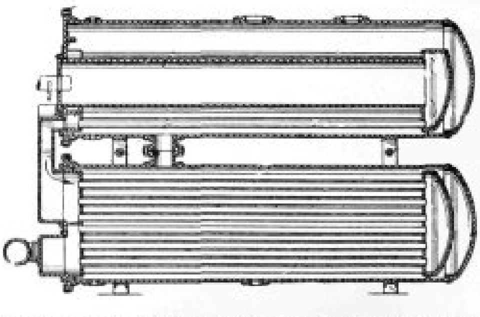
1,790,199. FURNACE ECONOMIZER, AUXILIARY BOILER, AND WATER SCREEN COMBINED. GLENVILLE A. COLLINS, OF LOS ANGELES, CAL., ASSIGNOR TO COLLINS WESTERN CORPORATION, OF LOS ANGELES, CAL., A CORPORATION OF CALIFORNIA.

Claim.—In a device of the class described, a fire chamber including a side wall, a boiler including water circulating tubes passing through independent apertures in said wall, said boiler being movably mounted whereby it may be bodily moved from the fire chamber. Four claims.

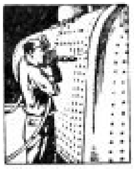


1,793,178. HIGH-PRESSURE BOILER. WILLIAM A. J. KREAGER, OF DENVER, COL., ASSIGNOR, BY MESNE ASSIGNMENTS, TO BARQUE ROYALTY, INC., OF DENVER, COL., A CORPORATION OF COLORADO.

Claim.—A high-pressure steam generator comprising two cylindrical members located above each other, means for connecting the interiors of the members, a cylindrical combustion chamber located in the upper member, the inner end of the combustion chamber terminating in a header, a header enclosing the front end of the combustion chamber, flues connecting the headers, a header near each end of the lower cylindrical member, a plurality of flues connecting the interiors of the last named headers, the front header in the lower cylindrical member having two compartments, means for connecting one of said compartments with the interior of the header enclosing the front end of the combustion chamber and means for connecting the other compartment with the atmosphere. Six claims.



The Boiler Maker



Annual Index

The annual index of The Boiler Maker for the year 1932 will be mailed without cost to each subscriber whose request for it is received at our New York office on or before January 15, 1933.

Locomotives in Service

A study of the motive power situation of the country very clearly indicates an increasing difficulty in meeting a gradually increasing traffic demand with locomotives in serviceable condition. Inevitably, and, possibly, not so far in the future, a concentrated effort to rehabilitate motive power will be undertaken.

On Class 1 roads as of November 1, the last figures available from the A. R. A., Car Service Division, there were 51,729 locomotives on the line as contrasted with 53,647 on January 1. In serviceable condition there were 42,539 locomotives on November 1, of which 9008 were stored. The number of serviceable locomotives decreased from the first of the year almost 4000 and the stored serviceable almost 2000.

Again, on January 1, 13 percent of the locomotives on the line were under or waiting class repairs, while on November 1 this percentage had increased to 17.8 percent requiring major maintenance operations. Throughout the year this percentage of locomotives waiting class repairs has steadily increased. Any additional gain in traffic, which gives evidence of coming as noted in the gradual but steady rise in car loadings over recent weeks, will be reflected in greater shop activity.

The longer the improvement in locomotive conditioning is delayed, so much more work will be essential to put motive power in order. Every department of the railroads' activities is equally affected.

Considering locomotive maintenance, and, as an integral part, boiler work, two things are evident. Any increase in shop activity will mean more employment, and from the supply standpoint will mean more buying of equipment, tools and materials.

In this connection A. G. Pack, chief inspector of the Bureau of Locomotive Inspection, in his annual report to the Interstate Commerce Commission states that during the past year 8 percent of the steam locomotives inspected by his department were found with defects or errors that should have been corrected before the locomotives were placed in service.

In the pursuit of economies, there is danger that this percentage might increase rapidly. If it is to be held down or reduced it means that the locomotive maintenance departments must increasingly give attention to essential class repairs. This can only result in more work for the shop forces and more buying of absolutely needed supplies.

High-Pressure Locomotives

While high-pressure steam for locomotives, within reasonable limits, offers a number of distinct advantages, it is brought out in a recent report of the committee on locomotive construction, presented before the American Railway Association, that such advantages may be attained only by designing boiler and machinery parts capable of meeting higher pressures without undue cost of maintenance. The use of the watertube firebox with the elimination of side and crown sheets and the elimination of side- and crown-sheet staybolts has resulted, in a number of cases, in more economical boiler maintenance. This saving is due to the elimination of maintenance cost due to broken staybolts and to plate cracking around staybolt holes. On one railroad alone, the cost of stripping boilers for the purpose of testing and renewing broken and leaky staybolt and radial stays amounted to $720,000 in one year. Such an experience indicates the necessity for proper boiler design to eliminate firebox troubles and excessive locomotive maintenance.

While greater power and greater capacity engines may be obtained by the use of high-pressure steam, it is indicated that a number of difficulties with cylinder and valve lubrication have been experienced. Such difficulties have caused increased maintenance and have necessitated stronger globe valves, pipe and cab fittings. To overcome these difficulties it may be desirable and probably necessary to discard the piston valve in favor of some form of valve, such as the poppet valve, which does not require lubrication.

The advantages of high-pressure steam cannot be disregarded; and disadvantages found in early experiences should not condemn its future use. It may be necessary to substitute tubes for conventional design, or bring about an improvement in material for plates, bolts, stays and braces, an improvement in their arrangement or both to reduce to an equable basis the cost of maintenance.

Advantages may be attained only by progress in the art of using high-pressure steam, with designers developing boiler and machinery parts capable of meeting higher pressure without undue cost to maintenance.

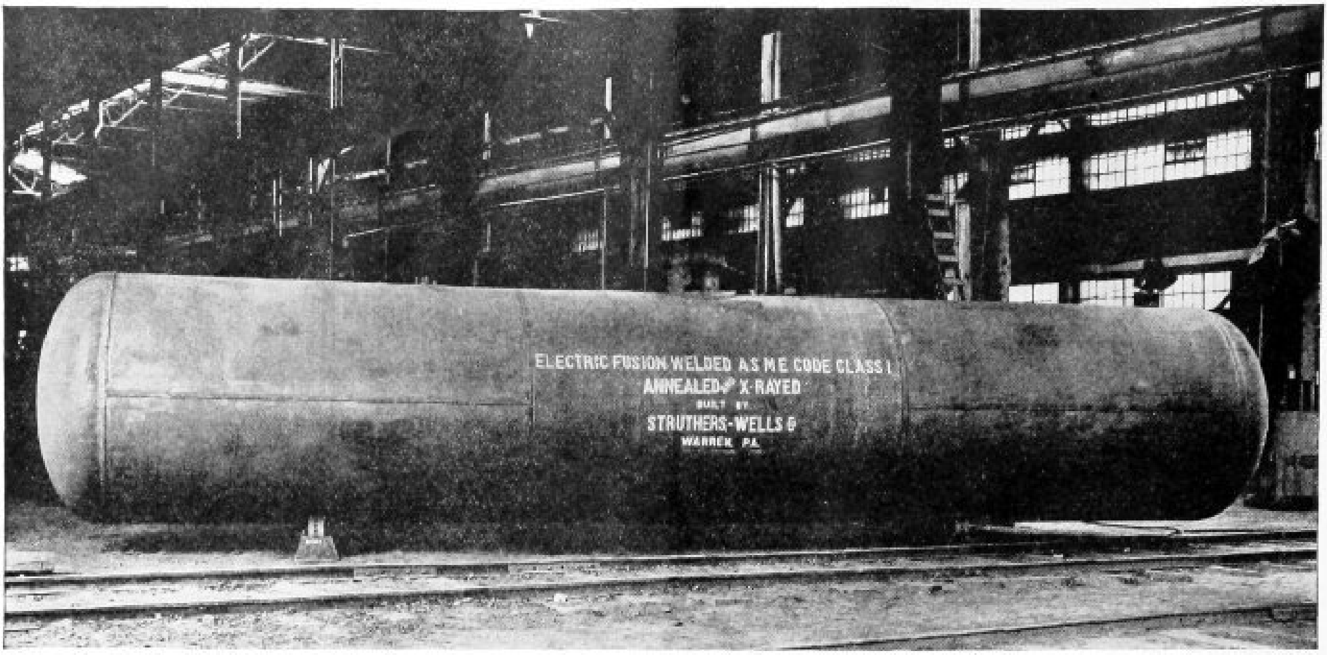


Fig. 1.—Type of welded vessel for hazardous service

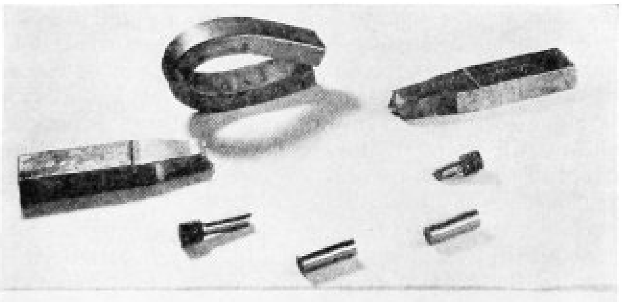
Application of Welding to Pressure Vessels*

By P. R. Hawthorne†

Developments in electric-arc welding during the past few years have resulted in recognition of the art by users of welded products, as well as the regulatory bodies, as the most efficient method of joining heavy metals to form vessels for high pressures and hazardous service. While formerly its use was limited to repair work and the fabrication of light shells for low pressures, it is now used successfully to join the thinnest of metals as well as metals of any practical thickness, even six inches or more, for vessels subjected to enormous pressures and for most severe and hazardous services.

For certain classes of vessels, electric-arc welding has entirely replaced riveting as well as all other methods of welding, and new applications and uses are continually being developed.

Electric-arc welding does not now, as formerly, compete with gas welding to an equal extent. Each method has, in general, found its particular economical and practical uses. For certain purposes the electric arc cannot compete with the gas torch. The improvements made in gas welding and cutting during the last decade are a credit to the manufacturers and engineers responsible therefor. New uses are being continually found and greater economies being effected in the application of the gas torch to cutting and welding. The cutting and shaping of metals by electric arc will probably never be developed to compete with gas, economy and quality of work considered. The gas torch has proved to be a most valuable adjunct to electric welding, particularly in heavy plate, for cutting and otherwise removing metals prior to and during the electric welding operation. It is true, beyond question, that without the gas torch, the preparation of plates over two inches



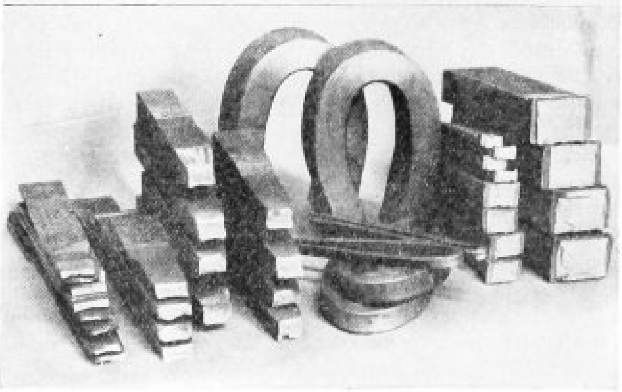


Fig. 2.—(Above) Complete set of test samples required for Class 1 vessels. Fig. 3.—(Below) Complete set of test samples required for Class 2 vessels

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* Paper presented at the Fall meeting of the American Welding Society, held in Buffalo, October 6. Cuts published through the courtesy of the *Journal of the American Welding Society.*

† Connected with Struthers-Wells Company.

thick would be very expensive and the preparation of many irregular pieces of welded vessels would be practically impossible.

The examples of welded products to be herein illustrated have, for the most part, been built in accordance with the A. S. M. E. Code for Classes 1 and 2. Class 1 requires physical tests to be made on sample pieces attached to and welded with shell, X-ray examination of welded seams and stress-relieving by heat treating the completely finished vessel in one unit.

Class 2 requires qualification of welders by making welded test plates of different thicknesses which, when tested, must meet physical standards as set up in the Code.

Fig. 2 shows a complete set of physical test samples as required for each Class 1 vessel. In this group is one all-weld metal tension specimen, threaded both ends, the tensile strength of which must equal the minimum range of plate used in vessel and the elongation in two inches must be at least twenty percent. The long straight piece is a flat tension specimen of the original plate thickness, reduced at its sides to make the entire load come on the weld cross-sectional area. The strength of this piece must also equal that of the minimum range of vessel shell plates. The third is termed a free bend specimen, machined four sides, left as near as possible to original plate thickness, and when bent under press or in testing machine the elongation in the deposited weld metal section, on the outer fibers must be at least 30 percent. The two small cylindrical pieces are specific gravity specimens. Each of these, when weighed, must equal 7.80 times their own volume of water. In the following table are given average results on ten sets of specimens representing ten actual vessels, illustrations and descriptions of which will be given later.

	Results	Required
All Weld Tension	60,200	55,000
All Weld Elong.	33.3%	20%
Red. Sec. Tension	57,200	55,000
Free Bend Elong.	41%	30%
Specific Gravity	7.88	7.80

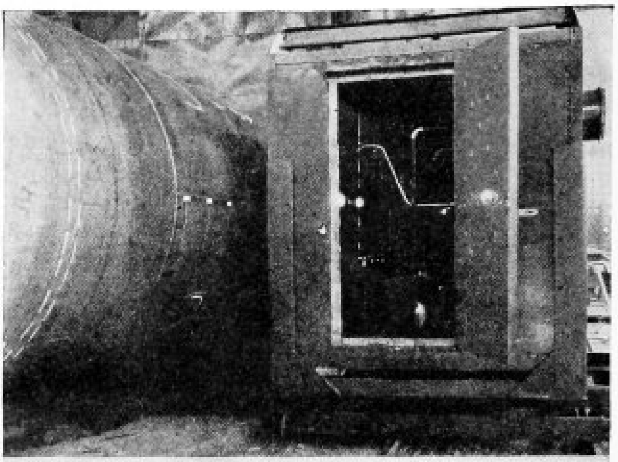
NOTE: No single test fell below average. All reduced section tests broke outside of weld.

Fig. 3 shows a complete set of samples as required for qualification of each individual welder for Class 2 welding. Welders must requalify every six months. These pieces are cut from plates of thicknesses of ¼, ⅝ and 1½ inches and the results of the tests must meet standards set up in the Code.

Undoubtedly one of the highest compliments yet paid to electric-arc welding was in the fabrication of vessels shown in Fig. 1. These are for handling liquids which remain in liquid state solely by being confined under their own vapor pressure. The service is extremely hazardous. Any leakage or failure might result in enormous property damage and loss of life through fire or explosion.

The initial order was recently finished and the vessels are now in service. They are about 7 feet inside diameter, 39 feet long, shell and heads being of plate one inch thick.

The joint efficiency for Class 1 Code vessels, under which these vessels were made, is 90 percent. The physical tests made on specimens attached to and welded with shell show these joints to be 100 percent efficient, in that the strength of the welds exceed, by far, the strength of the shell plates. For welding methods used heretofore in the fabrication of vessels for this service, the allowable working stress in the steel has been 8000 pounds per square inch, as given in the 1931 edition of A. S. M. E. Code, while the Code allows for these particular electric welded vessels, a working stress of 9900 pounds per square inch. This figure is based on the allowed joint efficiency of 90 percent steel with a minimum tensile strength of 55,000 pounds per square inch and a working pressure with factor of safety of five.



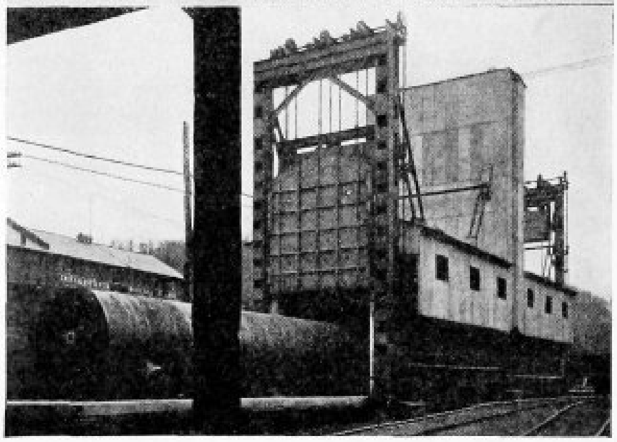
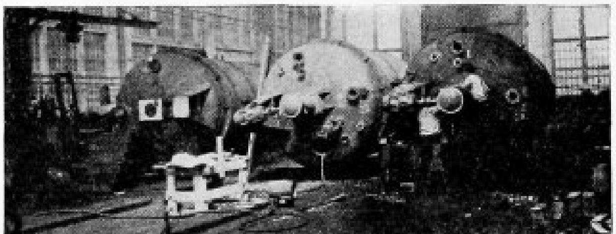


Fig. 4.—(Above) X-ray equipment for testing vessels. Fig. 5.—(Below) Furnace for stress-relieving welded vessels

This results in an allowable working stress of 23.7 percent greater than for vessels of this type heretofore permitted.

The specifications included most rigid tests, in addition to Code requirements.



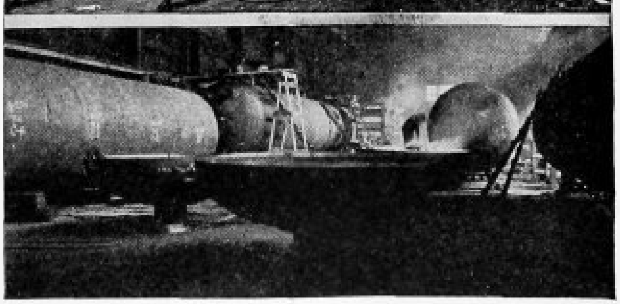


Fig. 6.—(Above) Welding department. Fig. 7.—(Below) Three vessels of complex design

The hydrostatic test consisted in each vessel being filled with water, pressure raised to 350 pounds per square inch, each welded seam hammered at 6-inch intervals with full swing blows of 10-pound sledge with vessel under 350 pounds pressure, then raising pressure to 475 pounds for period of thirty minutes while vessel was examined for leaks and sweats. Each vessel successfully passed these tests with no one showing any indication of leak. Very accurate work was required in forming the heads and shells and in assembling the tanks. The specifications allowed no offset more than 1/16 inch between abutting plates. The rings, of which there were three per vessel, had to be rolled much more truly circular than usually required in order that no point at end of any ring would be offset more than 1/16 inch from adjoining ring or head.

The process used was Struthers-Wells Company's "tempered arc," coated electrode; welding being done in a gaseous sheath to prevent contamination of metal while in vapor or highly liquid state, and through a combination of chemical reactions and metallurgical changes produces welds with properties which far exceed code requirements.

After welding was done, the seams were explored by X-ray. This included every foot of longitudinal and girth seams. These films, placed end to end, constitute a panoramic view of each seam and are a permanent record of the quality of the welds.

Complete X-ray equipment of high capacity is required for this work. A general view of this apparatus, with vessel in position, is shown in Fig. 4. This equipment is built especially for the examination of welds in heavy plate and is of much greater capacity than the ordinary commercial X-ray apparatus. The one shown is capable of making satisfactory films of welds in plate 3¼-inch thick, and on longitudinal seams, two exposures, each 17 inches long, are made at each setting.

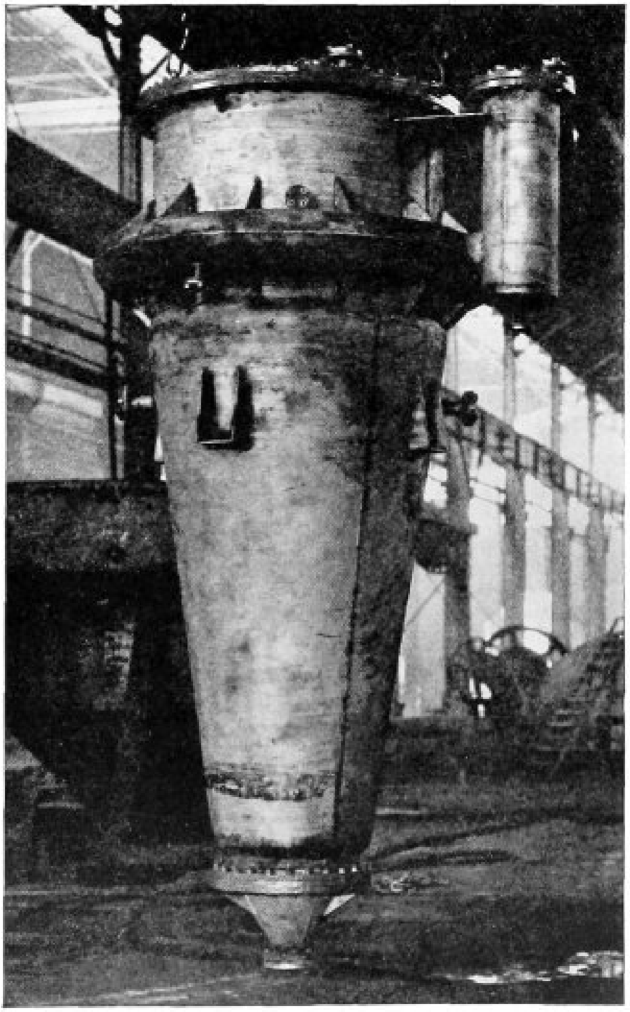


Fig. 8.—Jacketed vessel of chrome-nickel steel

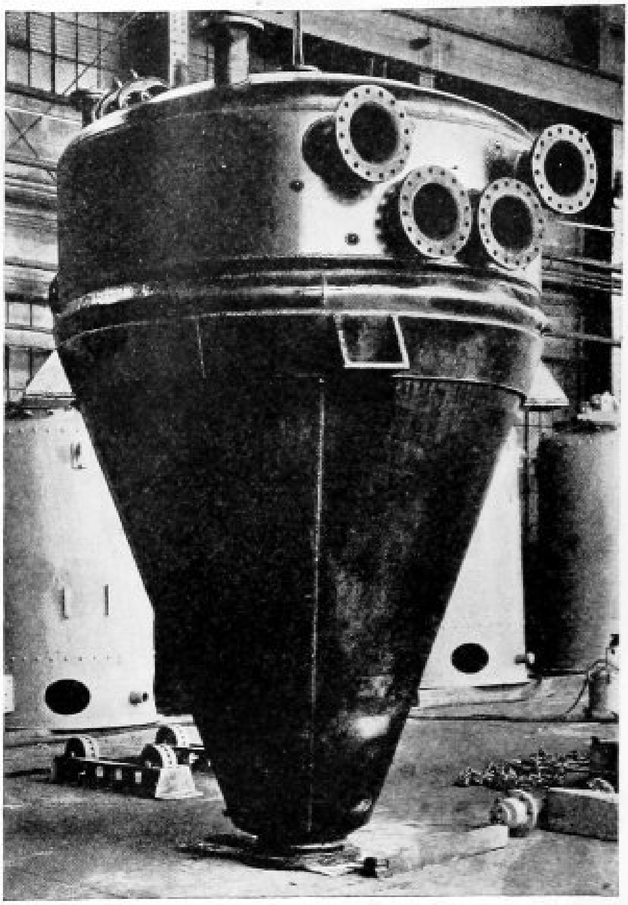


Fig. 9.—A welded vessel of complex design

After completion, and before the hydrostatic test was applied, each vessel was placed in furnace, the temperature brought up slowly to 1200 degrees F., held at 1200 degrees for 1¼ hours, then allowed to cool slowly in furnace to about room temperature. Fig. 5 shows this furnace, which is of the car type, the dimensions being 13 feet 3 inches wide by 15 feet 3 inches high by 82 feet 4 inches long, all inside dimensions. A bulkhead is built into the furnace at about midlength which may be lowered when required for short vessels. The lining and refractories used in the construction are capable of withstanding temperatures as high as 2200 degrees F. for heat treatment of alloy vessels and products.

The temperature at forty different points in the furnace is automatically controlled within very close limits. Eight charts record the temperature and time throughout the annealing operation and become a permanent record.

Fig. 6 shows one bay of the welding department with vessels being welded. These vessels are 8 feet inside diameter, about 45 feet long and of plate over 1 inch thick. They are Code, Class 1, and are being built for oil refining service. The oil industry was the pioneer user, through necessity, of electric welded vessels, by reason of failure of vessels made by other, and now practically obsolete, methods. It is largely due to the success of arc-welded vessels in the service of oil re-

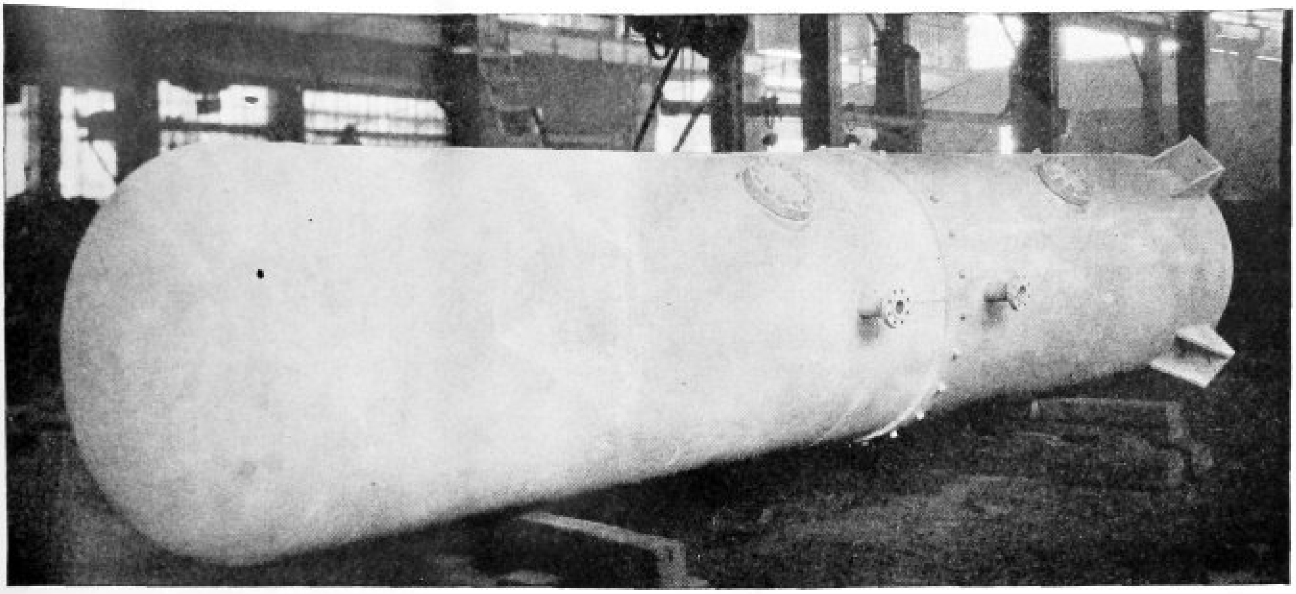


Fig. 10.—Welded vessel 5 feet in diameter with walls 3 inches thick

fineries that other industries became interested and at the present time practically all major industries use welded products of some nature.

Fig. 7 shows three vessels of a very complex design. These are about 10 feet diameter and about 20 feet long. Each has a steam jacket over the entire surface except for the top head, jacket being welded up, placed over and welded to tank and made an integral part of the completed structure. Note the large number of openings in the heads and the different angles of the axes of these. There are additional openings in opposite end, both through jacket shell into jacket space, and through both jacket and inner shell. Extreme accuracy is demanded in the alinement of machined flanges and other component parts of vessels of this type.

Inside the vessels are usually provided agitating or stirring devices and scrapers which must run very close to, but not touch, the shell. While these vessels were made of flange steel, many complex designs have been made of alloy steel. Fig. 8 shows a jacketed vessel made of chrome-nickel steel. A pressure vessel, as indicated by fastenings of openings, is provided with agitators and scrapers driven through gearing mounted on top of vessel.

Fig. 9 shows another complex design of vessel. The upper cylindrical portion is provided with many openings set at various angles. The lower conical portion is provided with jacket. Both jacket and vessel proper operate under pressure. In the background are a number of vessels, the design of which indicates they were built for low-pressure operation and the stitch riveting along seams indicates Class 3 welding.

Fig. 10 shows a vessel 5 feet inside diameter with walls 3 inches thick. The test pressure applied was 2000 pounds per square inch and it is now in operation under varying intermittent pressures up to 900 pounds, the rise and fall of pressure occurring almost instantaneously.

The building of Class 1 electric-welded vessels requires a vast amount of equipment, considerable of which does not apply to product made by any other method, particularly in case it is intended to equip a plant for the manufacture of vessels of the largest practical sizes in order to be able to serve all industries. To cover the entire field will require, in addition to the usual plate fabrication plant equipment, such items as large plate furnace, heavier rolls, presses and planers, boring mills, additional gas-cutting equipment, heavier cranes for handling completely assembled vessels, specially built devices for economical handling of vessels in process of fabrication, larger capacity testing equipment consisting of pumps, gages, fittings, etc., X-ray apparatus with all the accessories, annealing furnace at least 70 feet long inside, complete physical testing laboratory equipment and considerable chemical laboratory apparatus. In addition will be required the development of a process and the training of the shop organization in the application of same, which is radically different from the application of the ordinary commercial methods.

Further requirements are changes in shop methods and the specific duties of shop personnel, training the men to operate new shop equipment and laboratory apparatus as well as to train these men to think in terms of Classes 1 and 2 welding. This applies equally as seriously to other departments—estimating, engineering and sales, as well as executive—all of which must change their long cherished theories and fancies to terms of new type of product made by new and recently developed methods.

For building Class 2 vessels we may omit the X-ray, as the Code, at this particular moment, does not require X-ray. However, it is a valuable accessory to the development of the welding process, and most certainly of value as convincing proof of quality to prospective customers and their inspectors. Exacting purchasers of Class 2 vessels would be more inclined to place orders for Class 2 vessels with fabricators equipped to do Class 1 work than with those who did not have all the means to do the higher grade work. In fact, it is doubtful, as indicated by the attitude of the principal users of welded vessels in regard to the technical features of the processes and product of their sources of supply, if any manufacturer could secure sufficient volume of Class 2 work with which to maintain an organization, without purchasing and installing complete equipment for Class 1 welding.

The annealing furnace may be also theoretically omitted; however, there are certain designs of Class 2 vessels which require annealing to conform to Code, and these are not unusual. Then, as most manufacturers know, it is both economy and good practice, even if not required by Code, to subject Class 2 vessels, or parts thereof, to heat treatment at some stage during or after completion.

The building of Class 2 vessels does not mean, as the term might indicate, that a lower standard of welding will suffice, but there is required a quality of deposit equal to Class 1. Some engineers believe, and some experiences with customers' inspectors bear out the contention, that for Class 2 welding, the Code requirements are more rigid than for Class 1. So the use of Class 1 welding is required to do Class 2 work, and the cost per foot of welding Class 2 work is by no means less than for Class 1 welding.

While there has been required the development of processes, the design, purchase and installation of new, very expensive equipment, the addition of new methods and radical changes in existing departments, teaching men new trades, and, no doubt, in some cases, almost complete reorganization and an enormous amount of time and effort to convince the buying public the merits of the product, all at enormous cost, the result has been almost universal adoption by industry of electric-welded vessels for pressures higher and in units greater than heretofore possible to fabricate, more efficient and economical in operation, lighter in weight, at prices not out of line, nor in the end expensive (maintenance considered). This means of fabrication, with all the safeguards required by codes in their manufacture, and the manufacturer's efforts to establish a reputation for quality work, has been a worthy and commendable factor toward actual insurance against property damage.

Class 1 and Class 2 electric-welded vessels, when made as required by Code and used for service and under conditions intended, cannot be other than 100 percent efficient and 100 percent safe.

Short Course in Welding

Indicative of the trend in industry to use periods of lessened activity to study ways and means of improving production and maintenance methods, was a course in welding design held at the plant of the American Rolling Mills, Middletown, O., early in November.

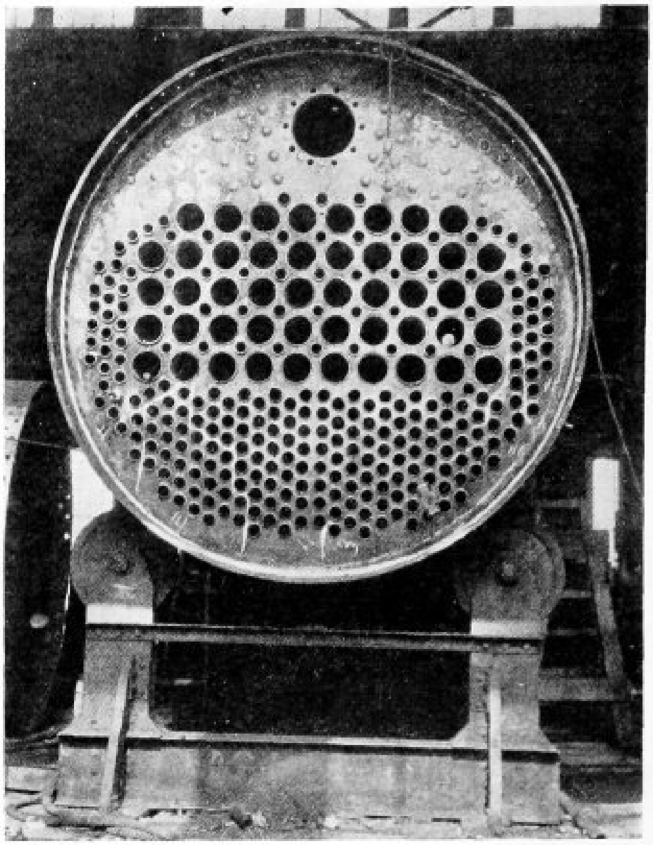
A three-day course in the arc-welding process and the theory of design for welded steel construction was held under the direction of E. W. P. Smith, consulting engineer for The Lincoln Electric Company, Cleveland. An average of 35 men attended the classes.

Mr. Smith's lectures consisted of discussions of arc welding generators, and the various electrodes, calculation of stresses in welded joints, substitution of welded steel for castings and the theory of redesign for welded construction. Lectures were supplemented by demonstrations in the shop.

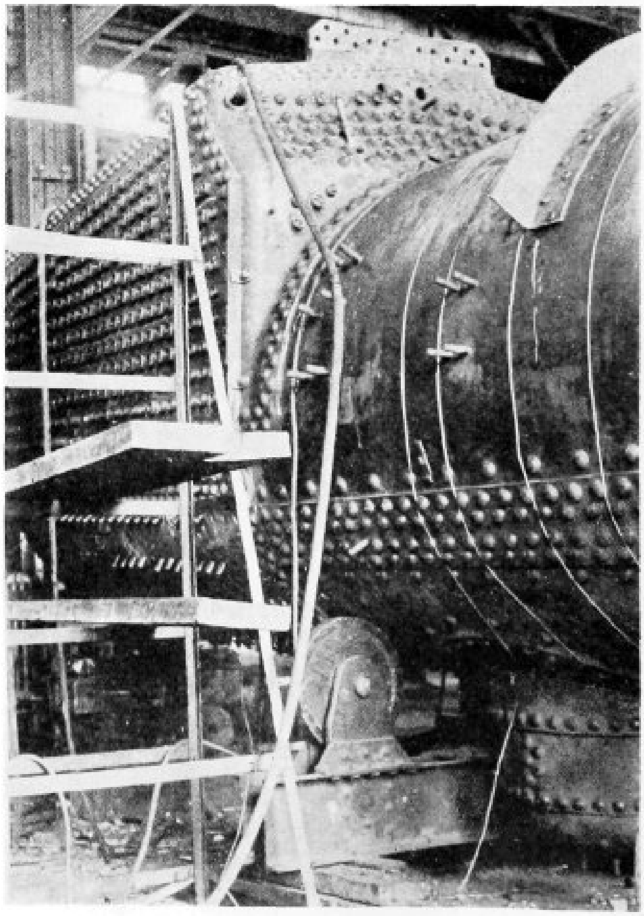
A Handy Horse for the Boiler Shop

Most locomotive boiler repair shops are provided with some sort of device for rotating boiler shells to convenient position for riveting, welding, etc. The horse shown in the two drawings and in the illustrations on this page was designed in the mechanical engineer's office and is used in the boiler shop of an eastern railroad.

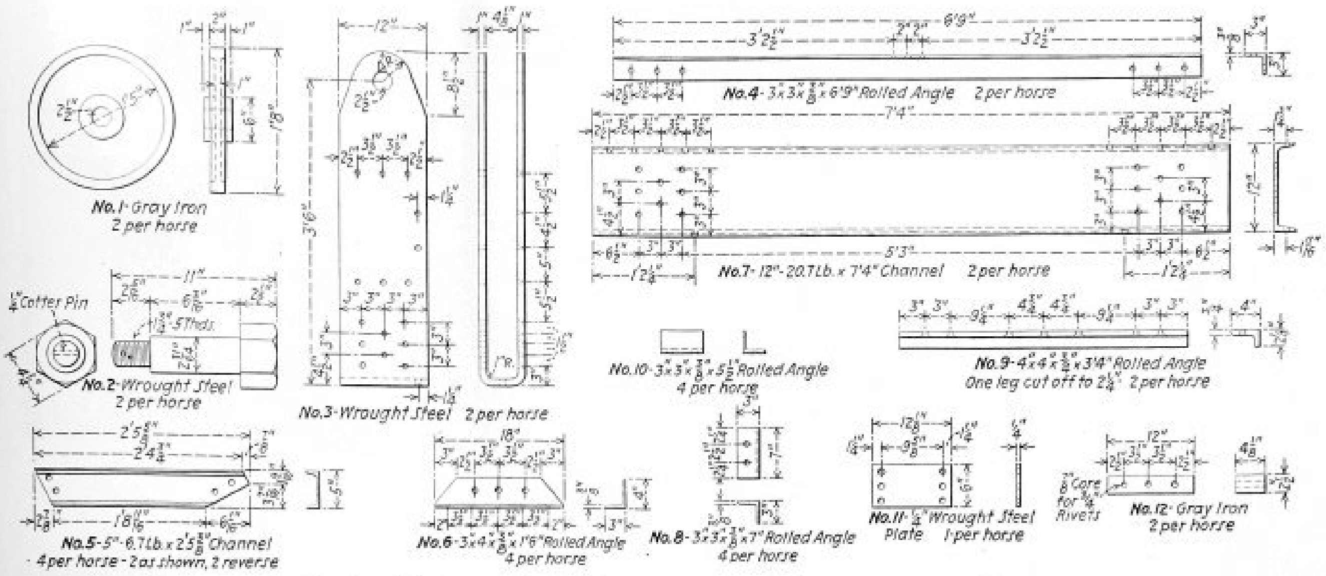
The rollers *1* are grey iron castings, 20 in. in diameter. They are mounted on U-shaped trunnions *3* of 1-in. by 12-in. wrought steel. The trunnions are secured



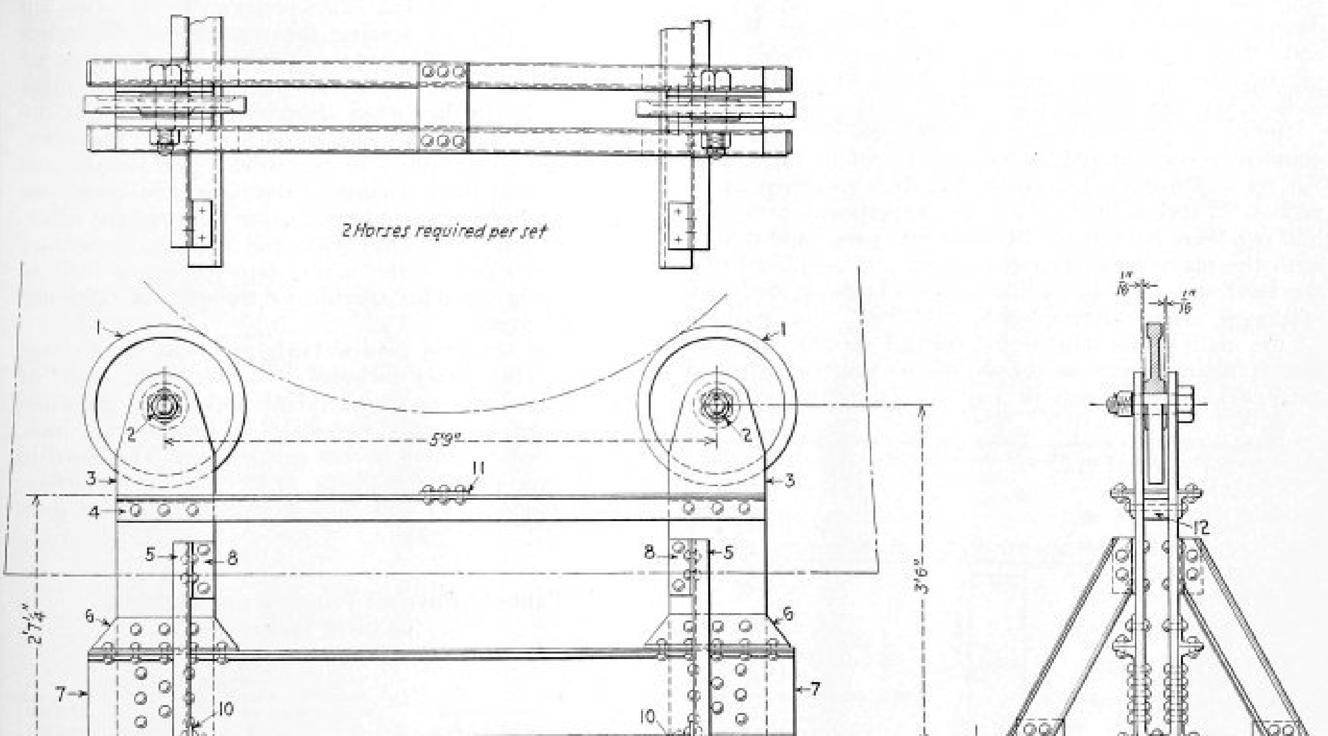
Side view of new boiler shop horse



Roller arrangement facilitates turning boilers



Details of the horse designed for rotating boilers when riveting or welding



Assembly drawing of the improved boiler-shop horse

to the foundation cross-channels *7* by 3-in. by 4-in. by 3/8-in. angles *6,* 1 ft. 6 in. long. They are formed to the shape of a U, as shown, and extend down between the backs of the cross-channels *7* to the floor plates. The cross-channels *7* are 12-in., 20.7-lb. section, 7 ft. 4 in. long. The trunnion pieces are braced with 5-in., 6.7-lb. channels *5,* 2 ft. 5 3/8 in. long, as shown. A cross brace *4* of 3-in. by 3-in. by 3/8-in. angle, 6 ft. 9 in. long, is applied 2 ft. 7 1/4 in. from the floor. The rollers *1* are spaced 5 ft. 9 in. apart, center to center, and this spacing is not adjustable. However, the spacing between the rollers is suitable for the majority of boilers in service on this railroad. All holes are 13/16 in. for 3/4-in. rivets. The roller pin *2* is made of a 2-31/64-in. diameter machine bolt, the end of which is cut and threaded for a 1 3/4-in. bolt, secured by a 1/4-in. cotter pin.

This horse supersedes a horse of shop construction. The rollers of the old-type horse were mounted directly on the cross channel which, because of the low height, had to be set on blocks to permit the bottom of the fire-box to clear the floor when the shell was being rotated. The height of 3 ft. 6 in. from the floor to the roller axle pin *2* of the new design is sufficient to provide clearance for the wide fireboxes used on the large freight locomotives operated by this road.

Strength of Field Welds*

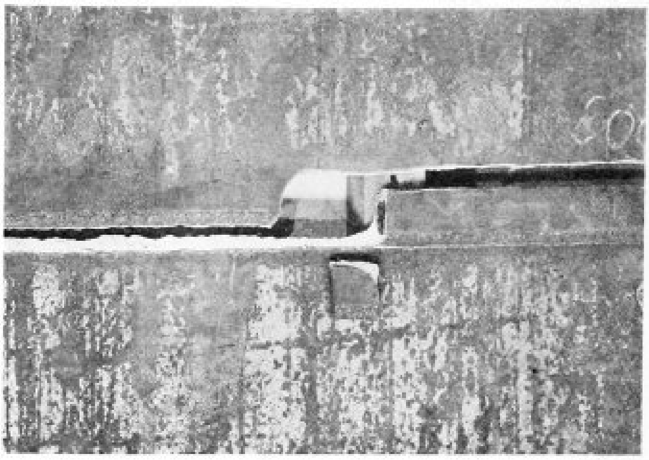


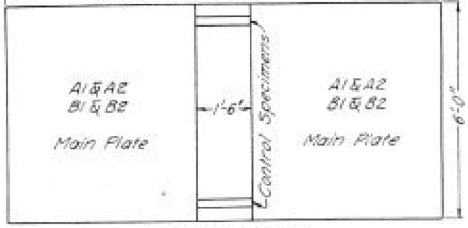
Fig. 2.—Plate after test

The Engineering Experiment Station of the University of Illinois, Urbana, Ill., recently completed tests on joints in wide plates. This article, the sixth and last in a series, describes the test specimens and results of tests for determining the strength of field welds. Conducted by Professor Wilbur N. Wilson, James Mather and Charles O. Harris, these experiments were made possible through the co-operation of the Chicago Bridge & Iron Works, Chicago, Ill.

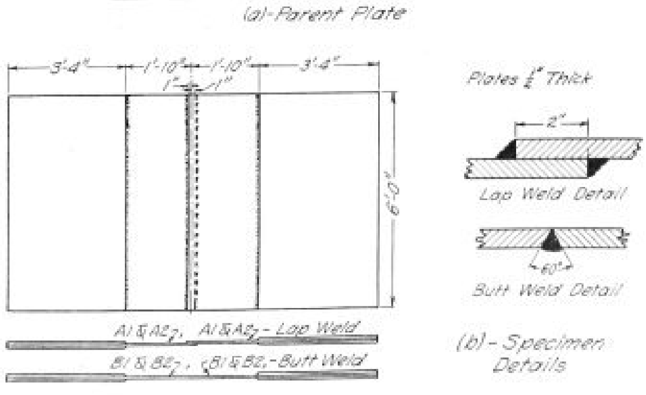
Joints of four wide plates were welded under field conditions encountered in the erection of storage tanks for oil and water. The plates for each specimen had a section 72 inches by ½ inch. Two specimens were butt and two were lap joints. All specimens were hand welded with the plates in a vertical position and so placed that the bead was vertical. A different workman welded each specimen, using an uncoated weld rod. The position of the main plates and of the control specimens in the parent plate as well as the details of the two types of field welds, are shown in Fig. 1. The physical properties of the material as obtained from control specimen are given in Table 1.

The results of the tests are given in Table 2. Both butt welds failed in the weld, the efficiency of B1 being 80.31 percent and of B2, 72.78 percent. Of the two lap welds, A1 failed by tearing the plate about 20 inches from the weld. This wide plate had an efficiency of 86.08 percent, based upon the strength of the control specimens 1½ inches wide. Specimen A2 failed at the weld and for the remaining portion it followed the other, rather than to a failure in the weld. The failure was unusual in that for a portion of the joint it followed one weld and for the remaining portion it followed the other. A piece about 2½ inches wide and 6 inches long tore free and fell out of the main plate leaving a hole as shown in Fig. 2. This specimen developed an efficiency of 94.11 percent.

A second series of field-welded specimens were tested in 1932. This series included two specimens each, of four types. Two specimens, MN1 and MN2, consisted of a lap joint, connecting two plates 64 inches by ½ inch, the joint being similar to that in specimen A1 shown in Fig. 1, except that the plates were lapped 2½ inches; two specimens, XY1 and XY2, consisted of a butt joint



(a)-Parent Plate



(b)-Specimen Details

Fig. 1.—Parent plate and details of specimens for field weld tests

Table 1.–Physical Properties of Material for Field Welds

Specimen Number	Control Specimen	Yield Point pounds per square inch	Ultimate Strength pounds per square inch	Elongation in 8 inches, percent	Reduction of Area percent
A 1	1	35,640	65,250	28.13	51.72
	2	34,460	64,000	26.75	52.01
	Average	35,050	64,310	27.44	51.87
A 2	1	31,880	58,450	29.63	54.20
	2	31,040	58,330	30.00	55.36
	Average	31,460	58,390	29.82	54.78
B 1	1	35,920	66,900	25.37	51.84
	2	35,530	66,770	22.75	53.35
	Average	35,730	66,890	24.06	52.60
B 2	1	34,800	65,710	26.63	53.45
	2	34,870	65,800	25.25	50.53
	Average	34,840	65,760	25.94	51.99
AW 1 1932	Average of 4.	36,100	59,900	31.8	64.7
AW 2 1932	Average of 4.	36,100	59,500	32.4	63.3
XY 1	Average of 4.	30,100	56,800	32.3	63.5
XY 2	Average of 4.	30,000	56,900	31.7	60.9
MN 1	Average of 4.	29,900	56,900	31.3	61.8
MN 2	Average of 4.	30,600	57,100	30.5	61.4
OP 1	Average of 4.	29,900	56,900	32.2	61.4
OP 2	Average of 4.	30,400	57,100	31.7	60.2

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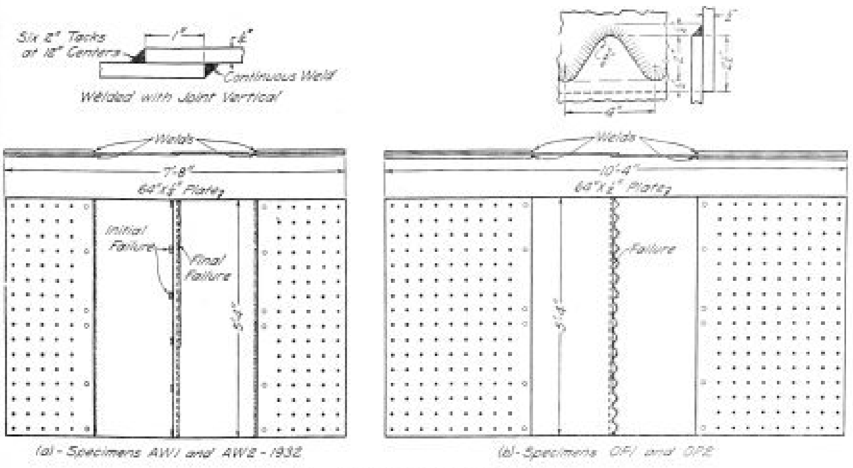
* Sixth and last of a series of articles published through the courtesy of the Engineering Experiment Station, University of Illinois, Urbana, Ill.



(a) - Specimens AW1 and AW2 - 1932 (b) - Specimens OP1 and OP2

Fig. 3.—Special Lap Joints

like B1 and B2, Fig. 1, connecting two plates 64 inches by ½ inch. All specimens except OP1 and OP2 were welded with the plates in a vertical position and so placed that the joint was vertical. The bead, however, was laid back and forth across the joint so that the pull on the specimen was parallel to the line of the bead. All joints were hand welded with a coated rod. The position of the main plates and of the control specimens in the parent plate are similar to those for the 1930 series. The physical properties of the material as obtained from control specimens are given in Table 1.

The results of the tests are given in Table 2. Both butt welds, XY1 and XY2, failed by tearing the plate near the junction with the side reinforcing plates where the specimen is connected to the pulling clevis. The failure is similar to the failure of 293C. These specimens developed efficiencies of 89.0 and 92.4 percent, comparable with the efficiencies of wide plates without joints.

Both plain lap welds having a continuous bead along each edge, MN1 and MN2, failed by tearing the plate. For MN1 the tear followed the weld for nearly the full length of the joint, for MN2 the tear followed the weld for about one third of the width of specimen and then extended into the plate away from the weld. These specimens developed efficiencies of 91.3 and 95.7 percent, respectively.

The two specimens, AW1—1932 and AW2—1932, containing lap welds having a continuous bead on one side and stitch-beads on the other, failed in the same manner. The plate first tore around the stitch-weld and then along the continuous bead as indicated in Fig. 3a. These specimens developed efficiencies of 87.4 and 79.6 percent, respectively.

The two lap welds for which one edge was scalloped and the other straight, and which were welded on the

Table 2.–Strength of Field Welds

Specimen Number	Section, inches	Type of Joint	Ultimate Strength Total pounds	Ultimate Strength Pounds per square inch	Strength of Control Specimens, pounds per square inch	Efficiency of Joint, percent	Elongation in 8 inches, percent	Reduction of Area, percent	Remarks
A 1	72 x 0.498	Lap Weld	1,985,000	55,360	64,310	86.08	9.84*	7.21	Plate tore 20 inches from weld.
A 2	72 x 0.503	Lap Weld	1,990,000	54,950	58,390	94.11	5.54*	4.06	Plate tore at weld.
B 1	71.9 x 0.51	Butt Weld	1,970,000	53,720	66,890	80.31	4.34*	7.28	Weld failed.
B 2	71.9 x 0.50	Butt Weld	1,720,000	47,860	65,760	72.78	2.21*	1.20	Weld failed.
MN 1	63.81 x 0.504†	Lap Weld	1,665,000	51,900	56,900	91.3		1.67¶	Plate tore at top of weld starting at north edge.
MN 2	63.81 x 0.503†	Lap Weld	1,722,000	54,600	57,100	95.7		1.67¶	Plate tore, the tear beginning at top of weld at north edge and extending up into plate.
OP 1	64.12 x 0.504†	Lap Weld‡	1,495,000	46,300	56,900	81.5		0.97¶	Plate tore on a transverse section tangent to weld, the scallops being torn from the plate of which they originally formed a part. See Fig. 3b.
OP 2	63.87 x 0.503†	Lap Weld‡	1,533,000	47,800	57,100	83.6		1.57¶	Same as OP 1.
AW 1 1932	63.87 x 0.242†	Lap Weld‡	806,000	52,300	59,900	87.4		1.82¶	First tore at inside tacks, then tore at bottom of continuous bead starting at north edge. See Fig. 3b.
AW 2 1932	63.87 x 0.243†	Lap Weld‡	734,000	47,300	59,500	79.6		1.08¶	Same as AW 1.
XY 1	63.87 x 0.503†	Butt Weld	1,655,000	51,500	56,800	89.0		2.74¶	Plate tore at end near junction with side plate. Similar to failure of 293C.
XY 2	63.87 x 0.504†	Butt Weld	1,638,000	52,500	56,900	92.4		4.90¶	Same as XY 1.

* Failure did not occur in the length in which the elongation was measured.
† Thickness is average thickness of four control specimens.
‡ See Fig. 3.
¶ Reduction in width.

scalloped edge only, specimens OP1 and OP2, both failed by tearing the plate on a transverse section tangent to the waved bead as shown in Fig. 3b. These plates developed efficiencies of 81.5 and 83.6 percent, respectively.

Although all specimens failed by tearing the plates, the efficiency of the joints was less than 100 percent. The plain lap-welded joints, MN1 and MN2, and the butt-welded joints, XY1 and XY2, developed efficiencies nearly equal to the efficiency of wide plates without joints. The lap-welded joints were slightly stronger than the butt-welded ones. The restraint to lateral flow offered by the joint probably caused the plates to fail at a load somewhat lower than the load which would have been carried by a similar wide plate without joint.

The greater strength developed by the field welds of the 1932 series over that developed by the similar field welds of the 1931 series is attributed to the difference in the weld rods used in the two series, the 1931 series having been welded with an uncoated rod and the 1932 series with a coated rod, both rods being manufactured by the same company.

The tests reported in this series of articles apparently justify the following conclusions:

(1) Continuous wide plates without joints (those tested were 20 inches by ½ inch, 40 inches by ¼ inch, and 64 inches by ⅝ inch) may be expected to develop a unit stress equal to approximately 90 percent of the unit strength of control specimens (tension specimens, 1½ inches wide). Two of the 15 plates tested developed a unit stress less than 86 percent of the strength of the control specimens.

(2) Double-riveted and quadruple-riveted lap joints having the same rivet spacing in all rows developed a unit stress on the net section equal to or slightly greater than the unit strength of the control specimens. Triple-riveted double-strap butt joints developed a unit stress on the net section just slightly less, about 2 percent, than the unit strength of the control specimens. Quadruple-riveted double-strap butt joints developed about 95 percent of the strength of the control specimens on the net section.

(3) Omitting rivets from the outside rows of a joint does not increase the actual efficiency as much as the theoretical.

(4) The ultimate unit strength in shear developed by the rivets in the large groups of the specimens of the 1930 series, when based on the nominal diameter of the rivets, exceeds slightly the strength developed by undriven rivets tested separately; but when the unit strength developed by the rivets of the specimens is based on a diameter equal to the diameter of the rivet hole, it is somewhat less than the strength developed by undriven rivets. The rivets in the lap joints were as strong, relative to the corresponding undriven rivets, as those in the butt joints.

(5) The bend in lap joints begins to increase rapidly at a unit stress in the plate considerably below the usual value of the design stress. For calked joints, this bending of the plate caused the calked joint to open at a unit stress on the net section considerably smaller than the usual design stress. The unit ultimate strength of the net section of plates in lap joints was equal to the strength of the 1½-inch control specimens.

(6) The two welded butt joints in plates 64 inches by ⅝ inch developed as great a strength as similar continuous plates without joints.

(7) Welding side plates onto the main plate of the specimens did not weaken the main plate at the section where the weld occurred.

(8) Of the welded lap joints in plates 11½ inches by ⅜ inch, welded with the plates in a vertical position, the two with transverse welds developed 90 percent and the two with diagonal welds developed 100 percent of the strength of 11½ inches by ⅜ inch continuous plates without joints cut from the same parent plate as the pieces used for the welded specimens.

(The End)

Recent Explosions of Air Tanks

Overpressure was not a factor in the recent explosion of an air tank in a Milwaukee laundry. The tank was used in conjunction with a motor-driven air compressor and had as protective devices not only a safety valve set to relieve at 80 pounds but also a motor cut-out adjusted to function automatically at a pressure of 5 pounds under that limit. As both devices were in good order, it seemed that the explosion must have occurred at a pressure not greater than that for which the vessel was intended to operate.

The tank was 36 inches in diameter, 6 feet long, and was made of ¼-inch plate with double riveted lap joints. Its lower head, however, had at some time been replaced with a shallow, skirtless head fastened in place by welding. The explosion occurred when the weld material sheared and permitted the saucer-like disk to tear loose around its entire circumference. Fig. 1 shows the flatness of the head and the manner in which failure took place.

An air tank of about the same length as the one just described, but somewhat smaller in diameter, exploded recently in the top story of a garage in New Orleans. Its lower head broke several heavy joists while passing through the floor below and damaged an automobile there. The tank itself crashed up through the roof of the garage.

Air was pumped into this tank by a motor-driven compressor. The owner was of the opinion that the safety valve stuck, thus permitting the compressor to build up an excess pressure. However, from an examination of the welded joint that failed, it appeared more probable that defective welding was the cause, for around at least a third of its circumference the head had scarcely been joined to the shell plate.

Some people apparently do not consider hazardous the

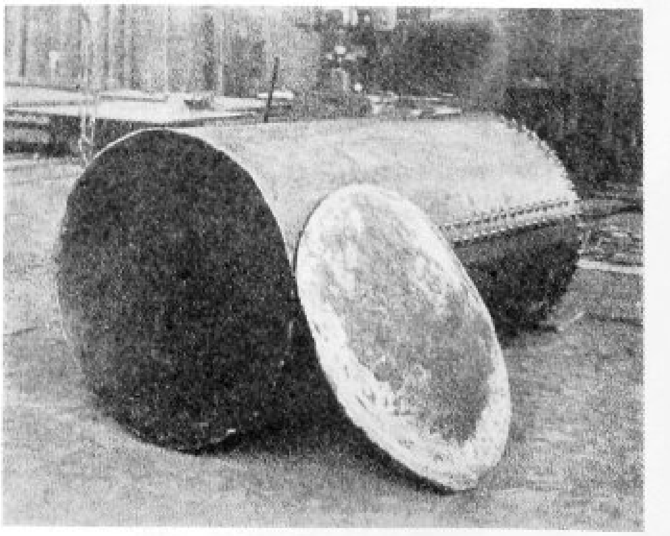


Fig. 1.—Appearance of tank and head after failure.

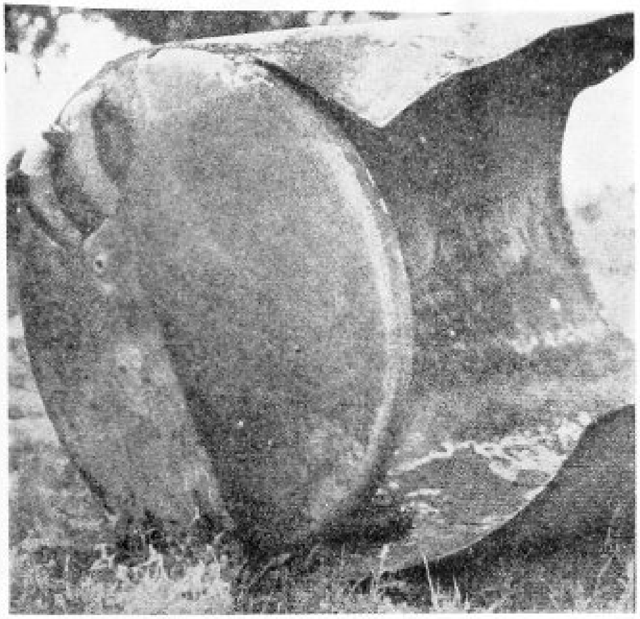


Fig. 2.—Head partially attached to shell

tank used in the hydro-pneumatic system of water supply. They overlook the fact that, although the tank is partly filled with cold water, the compressed air in the remaining space renders the container as dangerous as any air receiver.

At a greenhouse in Maumee, Ohio, the water supply system was of the kind in which pressure is created by trapping air in a tank and pumping water in with it. The tank was 14 inches in diameter and 24 feet long, and was made up of three courses of ¼-inch plate with butt welded longitudinal and girth seams. A gasoline engine drove the water pump and there was neither a relief valve nor any automatic arrangement for shutting down the engine when the required pressure had been reached. Prevention of over pressure depended on the operator stopping the engine when sufficient pressure showed on the gage.

As could so easily happen with this arrangement, there came a day a few weeks ago when the pressure became excessive. The resulting explosion tore the tank apart at each of the two girth seams, at the longitudinal seam of each course, and at both head seams. In Fig. 2 may be seen one course to which the head is still partly attached. No one was injured as a result of this explosion.—*The Locomotive.*

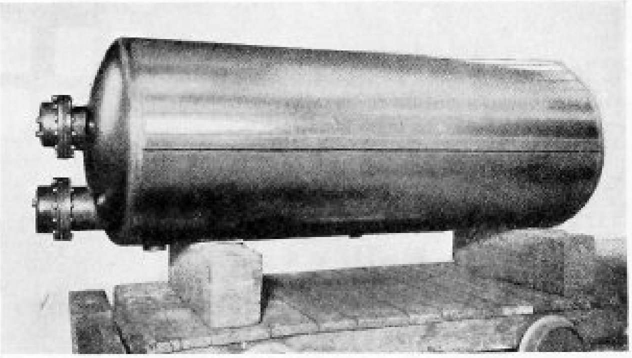
Nickel-Clad Water Storage Heater

The first nickel-clad steel water storage heater installed on this continent was placed in service in February of this year at the Jolicoeur Laundry, Ltd., Montreal, Can. The heater is of 1000-gallon capacity, tested to pressure of 135 pounds per square inch, and operating at working pressure of 90 pounds per square inch. The shell was formed from one nickel-clad plate, 149¾ inches by 119 inches by $\frac{5}{16}$ inches, with 10 percent cladding. The two heads were formed from $\frac{7}{16}$ inch nickel-clad steel, also 10 percent clad. The heater is of welded construction, with the continuity of the pure nickel interior preserved by welding with nickel to close the break in the clad surface at the joints. Outside welds are ordinary steel welds. The heater was fabricated by the Montreal Locomotive Works, Ltd.

Nickel-clad steel is a joint development of The International Nickel Company, New York, and Lukens Steel Company, Coatesville, Pa. The material is a hot-rolled bi-metal, composed of a light layer of pure, solid nickel permanently and inseparably bonded to a heavy layer of flange quality steel. Nickel-clad steel is rolled in thicknesses of $\frac{3}{16}$ inch and heavier, with the pure, solid nickel layer usually 10 percent of the total plate thickness, although heavier layers can be furnished if desired. Large sizes of nickel-clad plates also can be furnished.

The surface clad with pure, solid nickel possesses the same chemical and physical properties as hot-rolled or hot-forged nickel in other forms, and provides all of the advantages and characteristics of pure, solid nickel construction, including its immunity to many corrosive elements, freedom from iron-contamination and prevention of discoloration of the contained product.

In contrast to metals with light gage linings, the nickel layer and the steel layer in nickel-clad steel are literally welded together over the entire area of the plate. The bond of the two metals is complete, and there can be no separation, no deterioration or destruction of the bond, under normal conditions of temperature change, vacuum, pressure or mechanical shock. Buckling, peeling or cracking cannot occur.



Nickel-clad tank of 1000 gallons capacity

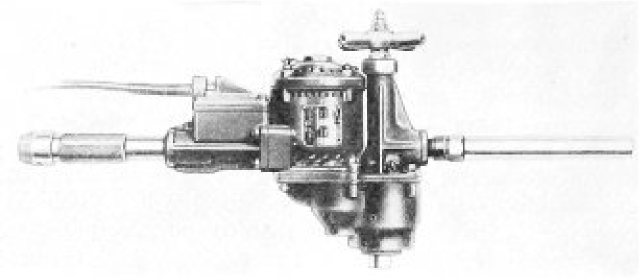
H. Jolicoeur, secretary and treasurer of The Jolicoeur Laundry, states that the nickel-clad storage heater has entirely eliminated rust in the laundry's hot-water supply. The nickel-clad heater has proved satisfactory in every way, and the laundry is now contemplating installation of a second nickel-clad heater to replace its other steel storage heater.

Drill and Reamer with Offset Motor

The Independent Pneumatic Tool Company, 600 W. Jackson Blvd., Chicago, has designed a new type of drill and reamer, which is a radical departure from the usual straight design of machine.

As can be noted in the illustration, the offset motor is separate from the frame and feed screw post. This design makes a short, rugged machine and permits the use of a feed of greater range. These advantages cannot be incorporated in the customary straight type electric drill and reamer. This design also permits the use of a short spindle, so necessary when operating in close places.

Another unusual feature of this new machine is the



New type Independent drill and reamer

switch handle, which is of the safety roll type and is incorporated exclusively in Thor high-frequency tools. This switch cannot be operated except by a turn of the operator's hand, and automatically closes when the hand is removed from the grip. This prevents accidents so commonly resulting in shipyards, railroad shops, etc., from electric tools that have the customary lever-type switch. The new Thor switch handle is similar in appearance to the throttle used on pneumatic drills. It has no exposed slots and openings, and there is no possibility of dust or dirt entering through the handle.

This new machine is compact in design, perfectly balanced, and is very easily handled. Grip handle can be had in place of feed screw, if desired.

This new drill and reamer is made in five sizes, ranging in speed from 215 to 525 revolutions per minute, in drilling capacity from ⅞ inch to 1¾ inches, and in reaming capacity from 11/16 inch to 1⅛ inches.

New Editions of A. S. M. E. Boiler Code

The Boiler Code Committee of the American Society of Mechanical Engineers, 29 West 39th Street, New York, has issued the 1932 editions of the A. S. M. E. Code for Low-Pressure Heating Boilers and the A. S. M. E. Code of Unfired Pressure Vessels. These sections, IV and VIII, of the A. S. M. E. Boiler Construction Code have incorporated therein the revisions and addenda published in August of this year.

Keeping Boiler Auxiliaries Repaired

A few weeks ago some very extensive damage occurred in a plant containing ten large watertube boilers. As a result of low water in eight of the ten boilers, several tubes ruptured, a few pulled out of the drums, and many others were so badly distorted that it was necessary to replace them. As the other two boilers were not in service at the time, the mishap shut the plant down. Fortunately, it did not cause any damage to persons, or property other than to the boilers themselves.

The apparent cause for the low water was the accumulation of sediment which, coming over with wet steam, clogged up strainers in the steam pipes and very seriously reduced the flow of steam to the feed pumps.

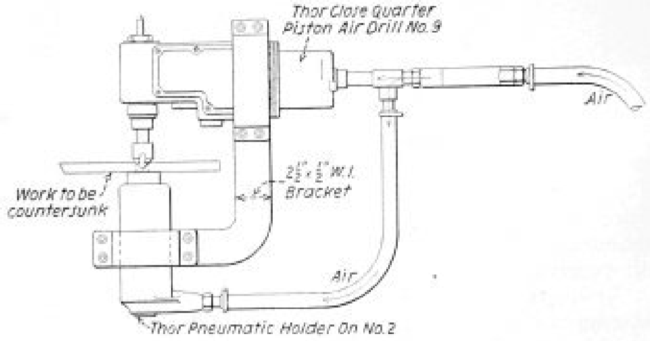
The plant had three turbine-driven centrifugal feed pumps, two of which were in constant service. When attendants noticed the lowering water line and slowing of the pumps, they made strenuous efforts to get more water into the boilers, but even after they started the third pump the water level continued to fall. The damage occurred before other protective steps could be taken.

Low water is a rather frequent source of trouble, but to have so many boilers involved at one time in exactly the same way is quite unusual. This case may well serve as a text for calling the attention of plant engineers to the possibility of the failure of the feed supply, and the need for seeing that pumps are always kept in prime condition and fully serviceable. It also serves to emphasize the necessity of giving careful attention to all other equipment and auxiliaries on which the proper and safe operation of a boiler plant depends.—*The Locomotive.*

Countersinking Holes in Flue-Sheet Flanges

A portable device for countersinking holes in the flange of flue sheets can readily be made from a Thor close-quarter piston air drill and a Thor pneumatic holder-on as shown in the drawing.

The drill and holder-on are securely fastened together by means of a bracket made of ½-in. by 2½-in. wrought iron or steel. A short nipple and tee are placed between the drill and handle. The side outlet of the tee is connected by means of a short hose to the holder-on. Thus, a turn of the throttle handle starts the countersinking tool and admits air to the holder-on simultaneously.



Combination set-up of Thor No. 9 air drill and No. 2 holder-on for countersinking the holes in flue-sheet flanges

Locomotive Shipments

October shipments of railroad locomotives from principal manufacturing plants, as reported to the Bureau of the Census, United States Department of Commerce, totaled 12 locomotives as compared with 13 in September, 12 in October, 1931, and 51 in October, 1930. Unfilled orders at the end of October totaled 87 locomotives as compared with 99 at the close of the previous month, 147 at the end of October, 1931, and 157 at the end of October, 1930.

These statistics do not include data on locomotives built by railroads in their own shops.

A. M. Snodgrass has been appointed production manager of the Pittsfield, Mass., works of the General Electric Company. In the employ of the company 20 years, Mr. Snodgrass was superintendent of the meter department at Fort Wayne, Ind., for several years prior to 1930. Since June 1 of that year he had been production manager of the West Lynn, Mass., works of the company.

The Boiler Maker

VOLUME XXXII NUMBER 12

PUBLISHED Monthly by the Simmons-Boardman Publishing Company, 1309 Noble Street, Philadelphia, Pa., with Executive Offices at 30 Church Street, New York City.

Samuel O. Dunn, Chairman of Board. Henry Lee, President. C. R. Mills, Vice-President. L. B. Sherman, Vice-President. Roy V. Wright, Vice-President and Secretary. F. H. Thompson, Vice-President. E. T. Howson, Vice-President. F. C. Koch, Vice-President. John T. DeMott, Treasurer. George Slate, Vice-President and Business Manager.

Chicago: 105 W. Adams Street. Washington, D. C.: 17th and H Streets, N. W. Cleveland: Terminal Tower. San Francisco, Cal.: 215 Market Street.

Subscription price in United States and possessions, 1 year, $2.00, 2 years, $3.00; Canada, including duty, 1 year, $2.50, 2 years, $4.00; foreign countries, 1 year, $3.00, 2 years, $5.00. Single copies, 35 cents each.

Request for change of address should reach us on or before the 15th of the month, preceding the issue with which it is to go into effect. It is difficult and often impossible, to supply back numbers to replace those undelivered through failure to send advance notice. In sending us change of address, please be sure to send us your old address as well as the new one.

THE BOILER MAKER is a member of the Associated Business Papers, Inc. (A. B. P.), and the Audit Bureau of Circulations, (A. B. C.).

EDITORIAL STAFF: H. H. Brown, Editor. L. S. Blodgett, Managing Editor. Warner Lumbard, Associate Editor.

Contents

GENERAL:

COMMUNICATIONS:

QUESTIONS AND ANSWERS:

Communications

Replacing Rivets below the Waterline

To the Editor:

The writer was much interested in a communication from Mr. James Wilson, on page 241 of the November issue of The Boiler Maker, in which the gentleman described a method of replacing broken rivets in the seams of a steel ship having countersunk head bolts. Mr. Wilson tells how a string attached to a float was passed outward through a hole from which a rivet had been removed. This string was picked up by men in a boat alongside the ship and a countersunk head bolt was attached to the string, lowered into the water and then drawn into place in the rivet hole by means of the string.

Although the rivet holes were countersunk on the outside of the ship, it is not clear to the writer how a bolt could be pulled into a rivet hole with a string wrapped and tied around the end of the bolt unless the body of the bolt was very much smaller than the rivet hole?

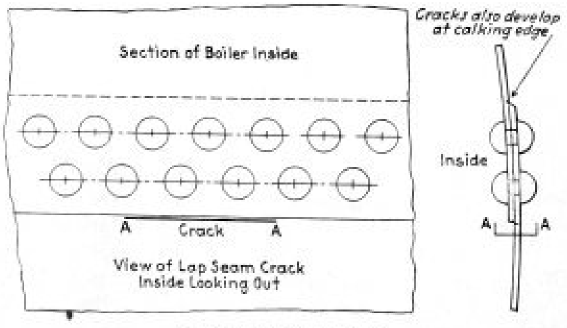
Furthermore, by what feat of juggling was it made possible, even if the hole was countersunk, to make the bolt stand perpendicular to the ship's side, in order that the bolt could be drawn into the rivet hole? Would it not be a good scheme to drill and tap a small axial hole in each end of the bolt and screw a small screw-eye into each hole? Then the string could be fastened to the screw-eye in the threaded end of the bolt. The bolt could then be lowered by means of the second string and also juggled into a horizontal position so as to be more easily pulled into the rivet hole by means of the strings attached to the threaded end of the bolt.

Indianapolis, Ind. James F. Hobart.

Welding Lap Seam Cracks

To the Editor:

The accompanying sketch shows a lap seam crack as found by the writer on a boiler operated in an ice plant. The crack was approximately 8 inches long and a sharp knife could be pushed in the crack about $\frac{3}{16}$ of an inch, so that from preliminary examination it appeared that it was nearly through the plate. However, after the brickwork was removed and the boiler cleaned on the outside, there appeared no irregularity or indication of a leak along the line of the internal crack on the outer surface of the shell.



Lap seam crack repair

I am writing this letter for the information of the younger men who are coming up in the boiler shop, and have incorporated electric and acetylene welding along with general boiler work. Do not attempt to weld or build up any defect along the longitudinal seam in any part of the boiler, steam dome, or mud drum; in fact, on no pressure vessel or anywhere on a boiler or pressure vessel where shell plates have wasted away. Cut out the defective section and apply a new plate.

However, before such repairs are made, the boiler owner should first call in a boiler inspector. I am writing in a non-Code state, but most boilers are covered by insurance and an inspector is always available.

Jacksonville, Fla. C. S. Handlee.

Questions and Answers Pertaining to Boilers

This department is maintained for the purpose of helping those who desire assistance on practical boiler shop problems. Inquiries should bear the name and address of the writer. Anonymous communications will not be considered. The identity of the writer, however, will not be disclosed unless the editor is given special permission to do so.

By George M. Davies

Strength of Boiler

Q.—If examining a boiler, what would you take as a guide for strength? —C. C.

A.—A guide for strength is the efficiency of the longitudinal riveted joint of the boiler examined, the strength of which is always less than that of the solid sheet.

Working Pressure of Boilers

Q.—Two boilers have the same diameter and are constructed of the same material; on what is their working pressure based?—C. C.

A.—The working pressure of two boilers having the same diameter and made of the same material would be based on the efficiency of the longitudinal seam or ligaments between openings, provided that the same factor of safety was allowed in each case.

Heat

Q.—What is heat?—C. C.

A.—Heat is the energy of molecular motion, or that condition in which the particles of matter are in a state of active motion. It consists in the undulations, rotations, or vibrations of the minute particles or molecules of which bodies are composed. A hot body is in a state of violent internal agitation, or molecular vibration, and the hottest bodies are those in which the vibrations have the greatest velocity and greatest amplitude. Hence, heat is motion, and it obeys the general laws of motion.

The different states in which heat is found to exist are only different states of motion. Heat is transmitted to motion through the agency of steam, gas or air.

Heat has no weight. It can only be measured by temperature and quantity.

Grooving and Pitting

Q.—What is grooving? What is pitting and how is it prevented?—C. C.

A.—Grooving may be described, when referring to steam boilers, as surface cracking of iron caused by its expansion and contraction under the influence of differing temperatures. It is attributable generally to the too great rigidity of the parts of the boiler affected.

Pitting, in a steam boiler, is a form of corrosion resulting in rows of minute holes or pits like pock marks. Pitting is most capricious in the location of its attack; it may be in a series of holes often running into each other in lines and patches, eaten into the surface of the iron to a depth, sometimes, of ¼ inch. Pitting is a dangerous form of corrosion and the dangers are increased when its existence is hidden beneath a coating of scale.

Zinc suspended in the water is an effective protector from corrosion in boilers. The action of zinc in preventing corrosion is due to the fact that, when two metals of dissimilar character are immersed in a liquid capable of chemically acting on both of them, and are connected or in metallic contact, the metal which is most affected or acted on by the exciting medium becomes the positive or corroded element, and the other becomes the negative or inactive element, and thus escapes corrosion as long as the metals are in contact. Zinc being the most readily acted on, it becomes the corroded element, and concentrates and absorbs corrosion which would attack the metal of the boiler if the zinc were not present and in metallic contact.

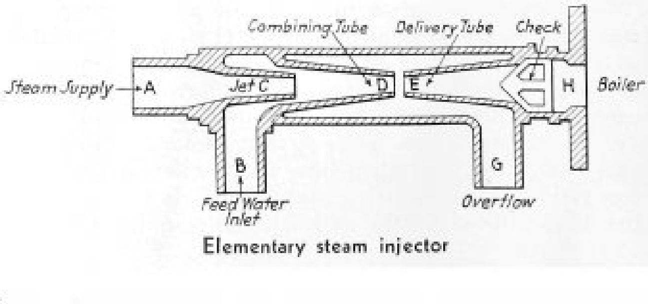
When feed waters contain acids which cause corrosion, their effect may generally be neutralized by treatment with alkalies, for which purpose soda-ash is frequently employed.

The Gunderson process is for use in the prevention of corrosion, such as pitting. The corrosion or pitting of steel and other metals takes place only under water or at least in the presence of moisture, and proceeds only by the solution of atoms of iron in the water. Iron can dissolve in water only when an equivalent number of hydrogen ions can be released from the water at some other point on the metal and therefore to prevent corrosion or pitting, it is necessary to prevent the escape of hydrogen ions from the water in the boiler. The Gunderson process accomplishes this by keeping the metal coated with metallic arsenic, which prevents the access of hydrogen ions to their only door or escape, the cathodic iron surface. This method can be used, it is believed, equally well on soft and hard waters.

Action of Injector

Q.—An injector is feeding a boiler at 100 pounds steam pressure. Describe its action fully.—C. C.

A.—An injector forces water into the boiler because the kinetic energy of a jet of steam is much greater



Elementary steam injector

than that of a jet of water escaping under the same conditions.

A simple illustration of an elementary steam injector is shown in the drawing on page 260.

Steam at 100 pounds per square inch pressure is admitted through the steam supply *A*, flows through the steam jet *C* and combining tube *D* and passes out through overflow *G*. The action of the steam jet through *C* creates a vacuum in the feed-water inlet *B*. The vacuum created in the feed-water inlet *B* draws in the water which, meeting the steam in the combining tube *D*, condenses the jet of steam. The steam issues from the overflow *G* as a jet of water, rapidly increasing in velocity, which builds up pressure against the boiler check *H*. The continued increase in velocity of the jet causes pressure against the boiler check to exceed the boiler pressure of 100 pounds per square inch and the latter begins to open, part of the water entering the boiler and part flowing out through the overflow *G* until the velocity in the combining tube *D* has become so great that all resistance is overcome and the check valve *H* is forced wide open, all the water entering the boiler. The action of the jet passing from the combining tube *D* to the delivery tube *E* now creates a vacuum in the overflow which causes the overflow to cease, closing the overflow valve, thus shutting out the air which would otherwise be forced into the boiler.

Brazing

Q.—Why is a brazed joint of an unfired pressure vessel allowed more strength in figuring the safe working pressure than any other form, or than a welded joint? What is the difference between them?—W. M. G.

A.—The A. S. M. E. rules for the construction of unfired pressure vessels, paragraphs U-68, U-69, and U-70 provide the following allowable strength for the fusion process of welding:

Par. U-68.—Class 1 Vessels.—The joint efficiency for seams welded in accordance with instruction for welding seams of class 1 vessels, to be used in applying the rules in paragraph U-20 (maximum allowable working pressure) shall be taken as 90 percent.

On the basis of 55,000 pounds per square inch tensile strength of plate with a factor of safety of 5, the allowable strength of the weld would be:

$$\frac{55{,}000}{5} \times .90 = 9900 \text{ pounds per square inch.}$$

The maximum unit joint working stress would be 9900 pounds per square inch.

Par. U-69.—Class 2 Vessels.—The joint efficiency, for seams welded in accordance with instructions for welding seams of class 2 vessels, to be used in applying the rules in paragraph U-20 (maximum allowable working pressure) shall be taken as 80 percent.

On the basis of 55,000 pounds per square inch tensile strength of plate with a factor of safety of 5, the allowable strength of the weld would be:

$$\frac{55{,}000}{5} \times .80 = 8800 \text{ pounds per square inch.}$$

The maximum unit joint working stress would be 8800 pounds per square inch.

Par. U-70.—Class 3 Vessels.—The maximum unit joint working stress for seams welded in accordance with instructions for welding seams of class 3 vessels is as follows:

Double-welded butt joints for all joints = 8000 pounds per square inch.

Single-welded butt joints for girth or head joints = 6500 pounds per square inch.

Double full-fillet lap welds for girth joints only = 7000 pounds per square inch.

Plug or intermittent welds for girth or head joints = 5600 pounds per square inch.

The rules for brazing (paragraph U-94) provide that, for joints properly brazed, the strength of the joint may be calculated on a maximum unit working stress of 8550 pounds per square inch.

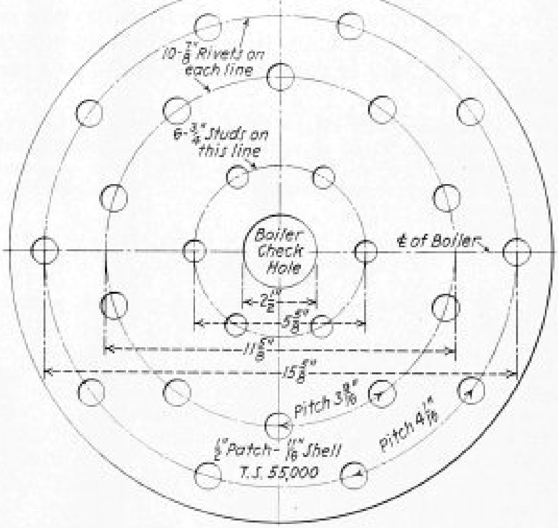
The strength allowed for welded joints, namely 9900, 8800, 8000, 6500, 7000 and 5600 for the various conditions of fusion welded joints and 8550 for brazed joints are allowed because of evidence submitted to the boiler code committee showing that these working stresses are safe and have proven satisfactory over a period of years.

Brazing is generally understood to mean the joining of metals by a film of brass, this being the chief difference between brazing and other forms of welding.

Efficiency of Circular Patch

Q.—Please give the correct formulae for calculating the efficiency of the enclosed patch.—F. J.

A.—The patch enclosed with the question is illustrated in Fig. 1. I would consider the patch to be a boiler check hole reinforcing plate, the rivets in the outer row being spaced too far apart for calking as required for a steamtight patch.



Circular Patch Problem

The thickness of the reinforcing liners must be at least 75 percent of the thickness of the plate to which they are attached. The rivets must have a shearing strength of at least 82 percent of the tensile strength of the metal removed.

Spacing of Crown Bars

Q.—Regarding the firebox of a boiler having crown supported by crown bars, Par. P-230 *b*, A. S. M. E. Code; will the bars have to be spaced longitudinally not to exceed the staybolt pitch or does the pitch required apply only to the rivets attaching the bar to the plate? How far will the bars have to extend on the sides towards the bottom? An interpretation of the foregoing would be appreciated.—J. A. S.

A.—For the benefit of those who do not have a copy of 1932 A. S. M. E. Boiler Construction Code, Par. P-230 *b* is as follows:

Par. P-230 *b*: In a form of reinforcement for crown sheets where the top sheet of the firebox is a semicircle and the top part of the circle not exceeding 120 degrees in arc is reinforced by arch bars extending over the top and down below the top row of staybolts at the sides of the furnace beneath the semicircular crown sheet, these arch bars being riveted to the water side through thimbles, the maximum allowable working pressure should be determined by adding to the maximum allowable working pressure for a plain circular furnace of the same thickness, diameter, and length determined by the formula in Pars. P-239 and P-240, the pressure P_1 determined from the following formula which is a modification of that in Par. P-241 *a*:

$$P_1 = 10{,}000{,}000 \quad \frac{b \times d^3}{D_1 \times D^3},$$

provided that the maximum allowable working pressure must not exceed that determined by the formula for furnaces of the Adamson type in Par. P-242 when L is made equal to D_1, and also provided that the diameter of the holes for the staybolts in the crown bars does not exceed 1/3 b, and the cross-sectional area of the crown bars is not less than 4 square inches. Par. P-199 would govern the spacing of the staybolts, rivets, or bolts attaching the sheet to the bars, and Par. P-212 d the size of the staybolts, rivets or bolts,

where, b = net width of crown bar, inches

d = depth of crown, bar, inches.

D_1 = longitudinal spacing of crown bar which shall not exceed twice the maximum allowable staybolt pitch, inches.

D = two times radius of crown sheet.

For constructions in which the crown sheet is not semicircular, or in which other features differ from those specified above, a test shall be made in accordance with Par. P-247, or in such other manner as the committee may prescribe, and the working pressure shall be based thereon.

My interpretation would be that the crown bars should be spaced longitudinally not to exceed the staybolt pitch, as the spacing of the staybolts, rivets or bolts attaching the sheet to the bars is governed by Par. P-199 which is as follows:

P-199: The maximum allowable working pressure for various thicknesses of braced and stayed flat plates and those which by rules require staying as flat surfaces with braces or staybolts of uniform diameter symmetrically spaced, shall be calculated by the formula:

$$P = C \times \frac{T^2}{p^2},$$

where, P = maximum allowable working pressure, pounds per square inch

T = thickness of plates in sixteenths of an inch

p = maximum pitch measured between straight lines passing through the centers of the staybolts in the different rows, which lines may be horizontal, vertical or inclined, inches

C = 112 for stays screwed through plates not over 7/16 inch thick with ends riveted over

C = 120 for stays screwed through plates over 7/16 inch thick with ends riveted over

C = 135 for stays screwed through plates and fitted with single nuts outside of plate or with inside and outside nuts omitting washers (see Par. P-203)

C = 150 for stays with heads not less than 1.3 times the diameter of the stays, screwed through plates or made a taper fit and having the heads formed on the stays before installing them and not riveted over, said heads being made to have a true bearing on the plate

C = 175 for stays fitted with inside and outside nuts and outside washers where the diameter of washers is not less than $0.4p$ and thickness not less than T.

It is noted in the above that the pitch (p) may be horizontal, vertical or inclined.

However, in Par. P-230 *b*, the value of D_1 does imply that the longitudinal spacing of crown bar may be spaced not to exceed twice the maximum allowable staybolt pitch. I believe this to be an allowable value for D_1 in the formula.

Before proceeding to space the crown bars longitudinally more than the staybolt pitch provided in Par. P-199, I would advise that a ruling be obtained from the A. S. M. E. Boiler Code Committee.

The second part of the question relative to extending the crown bars on the sides towards the bottom, is covered in Case No. 593 Interpretations of the A. S. M. E. Boiler Code which is as follows:

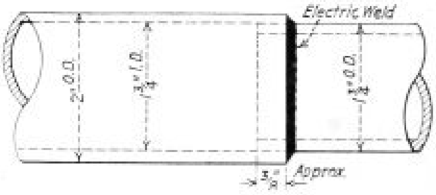
Case No. 593—Inquiry: Is it the intent of Par. P-230 *b* of the code that in applying the arch bar type of reinforcement to crown sheets of circular form, the crown sheet may be of the shape of a semicircle, or is it essential that the part of the circle constituting the crown sheet must not exceed 120 degrees in arc, so that the edges of the crown sheet will form a corner with the side sheets.

Reply: It was not the intent of the rule Par. P-230 *b* that the crown sheet to be reinforced should be so limited in shape so as to form a corner with the side sheet. It is the opinion of the committee that under this rule it is permissible for the crown sheet to be formed as a semicircle, but the portion that may be supported by the arch bar form of reinforcement is limited to an arc of 120 degrees. Attention is called to the fact that Par. P-230 *b* specifies that the arch bars should extend over the top and down below the top row of staybolts at the sides. By top row of staybolts is meant the top row of those staybolts at the side which run between the parallel sheets of the water leg.

Safe-Ending Boiler Flues

Q.—The accompanying sketch shows a method of safe-ending locomotive flues with a smaller diameter flue, the small end to go in the firebox tube sheet. This will eliminate swedging the tube and will also get more water to the firebox tube sheet. I would like to have your comments on this method.—M. McN.

A.—In this design, the chief difference from the conventional manner of safe ending flues, is that one tube is inserted inside of the other and secured with electric weld on the outside, while the conventional manner is to butt weld the safe end to the tube in an electric flue welder. Of the two methods I believe that butt welding the safe ends in an electric flue welding machine would be the most economical even when taking the swedging operation into account.



Safe-Ended Flue

In the proposed method, the variations in the diameters of the tubes would have to be taken into account if a snug fit is to be obtained.

The chief objection would be that the safety of the structure is dependent upon the strength of the weld and before making any application of safe ends in accordance with the method outlined, I would suggest the sketch be submitted to the inspector, under whose jurisdiction the flues come, for approval.

Trade Publication

Flue Tools—Gustav Wiedeke Company, Dayton, O., has issued a 95-page catalogue covering Wiedeke Ideal flue tools for power plants, locomotives and refinery stills. This work describes such tools as roller expanders, roller and flaring expanders, sectional expanders, tube setting expanders, tube cutters, beading and flaring tools, air hammers and motors for use on locomotive boilers; watertube, marine and fire return boilers; condensers, heaters, burners and similar units; and for oil refineries.

Associations

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Assistant Chief Inspectors—J. M. Hall, Washington, D. C.; J. A. Shirley, Washington, D. C.

Bureau of Navigation and Steamboat Inspection of the Department of Commerce

Assistant Director—D. N. Hoover, Jr., Washington, D. C.

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Vice-Chairman—D. S. Jacobus, New York.

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States and Cities That Have Adopted the A.S.M.E. Boiler Code

States

Arkansas
California
Delaware
Indiana
Maryland
Michigan
Minnesota
Missouri
New Jersey
New York
Ohio
Oklahoma
Oregon
Pennsylvania
Rhode Island
Utah
Washington
Wisconsin
District of Columbia
Panama Canal Zone
Territory of Hawaii

Cities

Chicago, Ill.
Detroit, Mich.
Erie, Pa.
Evanston, Ill.
Houston, Tex.
Kansas City, Mo.
Los Angeles, Cal.
St. Joseph, Mo.
St. Louis, Mo.
Scranton, Pa.
Seattle, Wash.
Memphis, Tenn.
Nashville, Tenn.
Omaha, Neb.
Parkersburg, W. Va.
Philadelphia, Pa.
Tampa, Fla.

States and Cities Accepting Stamp of the National Board of Boiler and Pressure Vessel Inspectors

States

Arkansas
California
Delaware
Indiana
Maryland
Minnesota
Missouri
New Jersey
New York
Ohio
Oklahoma
Oregon
Pennsylvania
Rhode Island
Utah
Washington
Wisconsin
Michigan

Cities

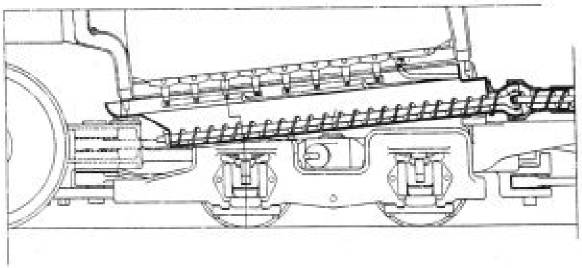
Chicago, Ill.
Detroit, Mich.
Erie, Pa.
Kansas City, Mo.
Memphis, Tenn.
Nashville, Tenn.
Omaha, Neb.
Philadelphia, Pa.
Parkersburg, W. Va.
St. Louis, Mo.
Scranton, Pa.
Seattle, Wash.
Tampa, Fla.

Selected Boiler Patents

Compiled by Dwight B. Galt, Patent Attorney, 1372-1376 National Press Building, Washington, D. C. Readers wishing copies of patents or any further information regarding any patent described, should correspond directly with Mr. Galt.

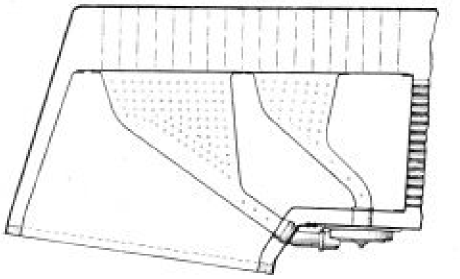
1,790,768. LOCOMOTIVE ASH CONVEYER AND COLLECTOR. MENAHEM RIVKIN, OF PHILADELPHIA, PA.

Claim.—The combination in a locomotive, of a frame; a boiler having a firebox mounted on the frame; an ash pan located under the firebox end of the boiler; a tender; an ash receptacle on the tender; and means for conveying ashes from the ash pan to the receptacle on the tender. Eight claims.



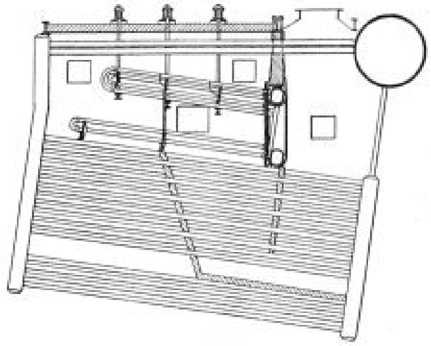
1,803,714. LOCOMOTIVE BOILER. BERT E. LARSON, OF PARK RIDGE, ILL., ASSIGNOR TO LOCOMOTIVE FIREBOX COMPANY, OF CHICAGO, ILL., A CORPORATION OF DELAWARE.

Claim.—A locomotive boiler having a firebox including a fire and a combustion chamber, a water circulating and steaming member in each chamber and each including an inlet end, with one of said filler ends disposed at substantially a right angle to the other, means providing a well below the boiler and in communication therewith, means providing a slip joint connection for said inlet end of the member in the combustion chamber with said well and means providing a ball and socket joint connection for the inlet end of said member in the firebox with said well. Four claims.



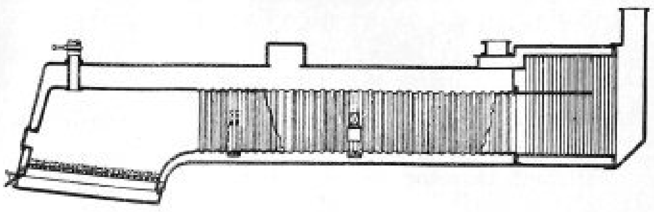
1,827,892. SUPERHEATER. WALTER F. KEENAN, JR., OF PELHAM, N. Y., ASSIGNOR TO FOSTER WHEELER CORPORATION, OF NEW YORK, A CORPORATION OF NEW YORK.

Claim.—In a superheater boiler, a superheater comprising two banks of superheater elements separated by a space sufficient to permit the entrance of a workman between said banks for inspection and repair purposes and adapted to receive elements from either bank without interference with elements in the other bank, and soot blower tubes transversely mounted in said space and supported from one of said banks and adapted to discharge cleaning fluid jets against said elements. Seven claims.



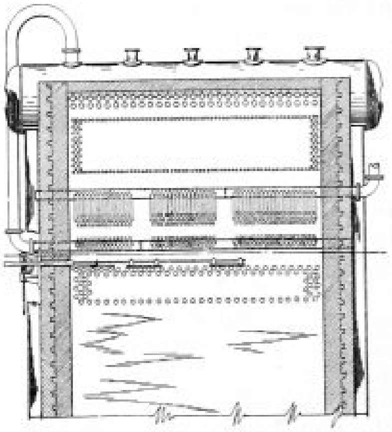
1,791,099. FUEL-BURNING DEVICE. HENRY G. LYKKEN, OF MINNEAPOLIS, MINN.

Claim.—A locomotive fuel-burning device comprising a combustion chamber at one end and a stack at the other end, said combustion chamber having a fuel introducing opening in the rear wall thereof, means for normally maintaining a fuel bed in the bottom of said chamber, an elongated tubular extension of approximately uniform diameter leading from the front end of said combustion chamber, and above the normal level of the fuel bed approximately to said stack, and means for forcibly projecting a mixture of powdered fuel and air into said combustion chamber above said fuel bed so that said mixture will be initially ignited by the fuel bed and its combustion will be completed within said tubular extension. Five claims.



1,789,076. SUPERHEATER BOILER. HOWARD J. KERR, OF WESTFIELD, N. J., ASSIGNOR TO THE BABCOCK & WILCOX COMPANY, OF BAYONNE, N. J., A CORPORATION OF NEW JERSEY.

Claim.—In a steam boiler, two spaced banks of horizontally inclined boiler tubes, a bank of superheater tubes located in the space between said banks of boiler tubes, said bank of superheater tubes being divided across the flue into at least two groups of tubes with a free unoccupied space between the groups, and means movable across said space for regulating the flow of the gases over said superheater tubes by varying the amount of flow through said space. Eleven claims.



1,809,133. WATERTUBE BOILER CONSTRUCTION. KINGSLEY L. MARTIN OF MONTCLAIR, N. Y.

Claim.—In a boiler of the watertube type having a furnace at the front end and comprising a bank of tubes inclined from front to rear, upright headers at the ends of the tubes and a steam and water drum above the headers with which they communicate, a combustion chamber below the forward end of the boiler with a bridge wall at its rear end, a baffle around the lower row of tubes extending rearward from the front header, a part of the length of the tubes and terminating directly above the bridge wall, a cross baffle extending upward and to the rear from the end of the baffle at an obtuse angle to such baffle, a third baffle on one of the upper rows of tubes extending from the rear header forwardly a part of the length of the tubes, and across baffle extending downward from the end of said last baffle at an obtuse angle to such baffle.