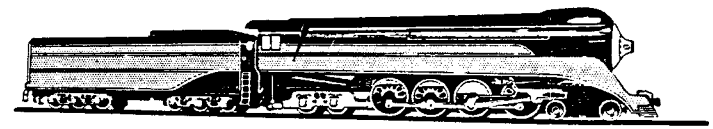
TABLE OF CONTENTS

STAYBOLTS

Published by the
FLANNERY BOLT COMPANY
Manufacturers of
All Types of Flexible and Rigid Staying Equipment for Boilers.
General Offices and Works, BRIDGEVILLE, PA.



VOL. 9 OCTOBER, 1940 No. 1

THE AMOUNT AND DISTRIBUTION OF FIREBOX AND TUBE EXPANSION IN A LOCOMOTIVE BOILER

By G. R. Greenslade

IN DEALING with the problems arising from unequal expansion and contraction of the different parts which make up a locomotive boiler structure, it is well to make a study of the factors which determine the magnitudes of the expansion in these various parts. Among these factors are the calorific value of the fuel, the rate of burning, the amount of air used in combustion, the temperature of the flame, the direction and distribution of heat flow, and the thermal conductivities of the materials involved in the heat transfer.

In the discussion which is to follow these topics will be treated in the order in which they appear as factors involved in the various phases of the expansion problem.

For convenience of handling we will divide the discussion into four general subjects, as follows:

FIRST: Determining the temperature difference between the inner and outer boiler sheets of a locomotive in service.

SECOND: Deformations caused by the difference in temperature between the inner and outer boiler sheets of a locomotive in service.

THIRD: The linear thermal expansion of the flues and tubes of a locomotive boiler.

FOURTH: The distribution and mechanical effects of firebox and tube expansion in a locomotive boiler.

We will now proceed with the development of the first of these subjects.

DIAGRAM SHOWING THE TEMPERATURE DECREASE ALONG THE PATH OF HEAT FLOW OUTWARD FROM THE FIREBOX

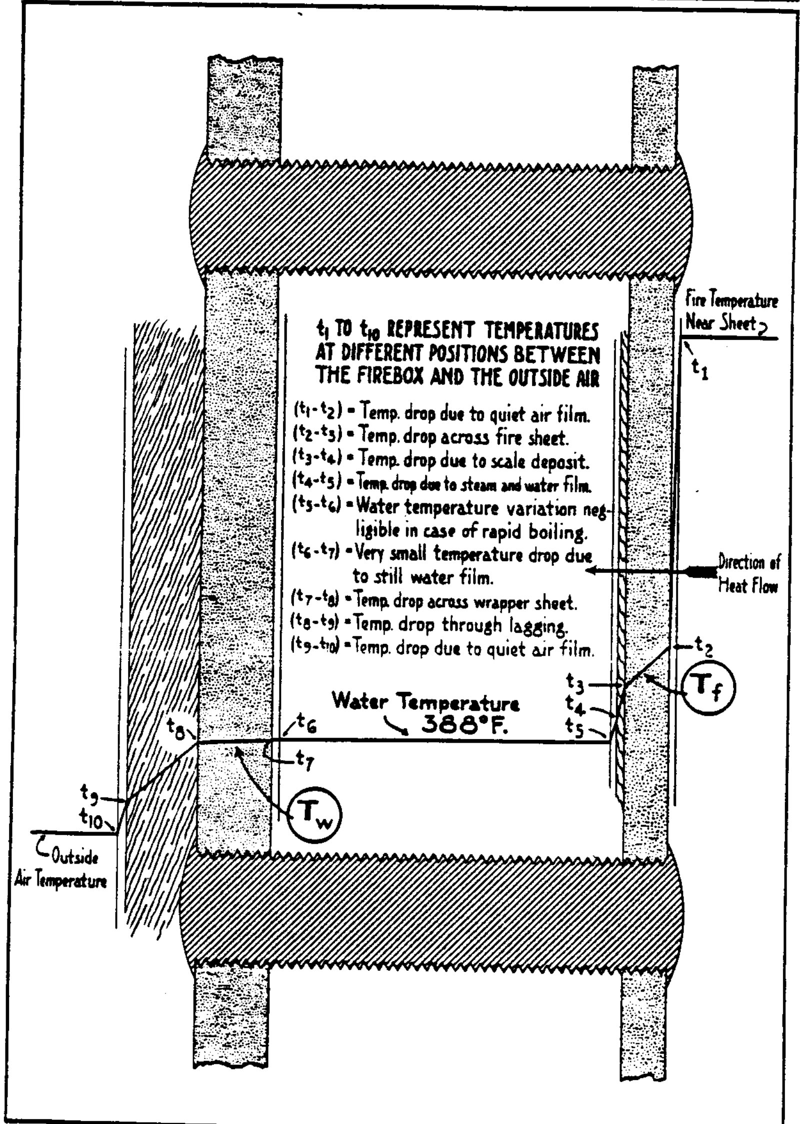


FIGURE 1

Determining the Temperature Difference Between the Inner and Outer Boiler Sheets of a Locomotive in Service

DUE TO contact with flames and hot gases and by the absorption of radiated heat, the firebox of a locomotive in service becomes hotter, and consequently expands more than does the wrapper sheet. This condition causes more or less warping and deforming of the boiler and in some zones severe straining of the staybolts.

In discussing the problems concerning various boiler movements and stresses, and in showing something of their importance, we will begin by determining how large the temperature difference between the inner and outer boiler sheets may become. It is well to mention at the outset that since boiler conditions are so extremely irregular, and since there are so many factors involved in the computations, it is only possible to get an approximate solution of what we might call a typical case. For this typical case we will consider a certain 4–8–2 locomotive capable of evaporating 8,000 gallons of water per hour at 200 pounds gage pressure when fired at its maximum rate.

Most of the damage done to a locomotive boiler results from the extreme conditions which arise, so we wish to take the case of the maximum temperature difference which may be developed between the inner and outer sheets. Some engineers have maintained in the past that the greatest temperature difference occurs at the time of firing up the engine, before the boiler water and wrapper sheet have had time to get hot. This is not necessarily so, however, for theory as well as actual measurements show that the temperature difference between the two sheets is greatest when the amount of heat passing from the firebox into the boiler water is greatest, that is, when the fire is most intense. This condition is not usually obtained until after full steam pressure has been developed in the boiler.

The computation of the temperature difference which we wish to know cannot be accomplished by a single operation because the heat, in traveling outward from the firebox, passes through a number of regions which are distinctly different in regard to their thermal conductivities. An examination of the diagram shown in Figure 1 will make this clear. The heat flows in the general direction shown by the heavy arrow. In doing this it first encounters a thin layer or "film" of mixed gases and air against the fire sheet. It is always found that when a gas or liquid is confined by the surface of a solid the very thin layer in actual contact with the surface remains almost still, even though the fluid is in rapid motion a little farther away. A "film" such as this offers high resistance to the flow of heat and consequently causes a large fall in temperature. We have indicated this in the diagram by the drop from temperature t_1 to t_2, (See Figure 1). The heat next passes through the fire sheet which, though much thicker than the gas and air film, causes only the relatively small temperature drop t_2 to t_3. This is on account of the high thermal conductivity of steel.

On the water side of the firebox there is usually a deposit of boiler scale which is so poor a conductor of heat that even if it is but a few thousandths of an inch in thickness it brings about a marked decrease in temperature. In our diagram, the temperature difference between the two sides of this deposit is expressed as t_3 minus t_4, (See Figure 1).

At the surface of contact between the boiler water and the deposit of scale there is another insulating "film" of relatively quiet fluid. Instead of being a gas as in the former case, this film is of water, or rather a mixture of water and steam, and offers a lower resistance to the flow of heat. Nevertheless, it causes a temperature decrease which is so large that we must consider it in our computations.

In following the irregular temperature line of Figure 1 from t_1 to t_{10} the amount of its drop toward the bottom of the page shows proportionally the fall of temperature which takes place in the various regions as the heat travels away from the fire in the direction indicated by the arrow.

The successive temperature drops which we have enumerated so far have brought us to t_5, the temperature of the boiler water. This is 388 degrees Fahrenheit for 200 pounds of steam which is the normal working pressure of the type of boiler under consideration. When a locomotive is steaming rapidly, there is practically no drop in temperature through the water space. Before boiling begins this temperature variation may amount to a great many degrees, but when steam is up and the locomotive is on heavy road duty, the entire mass of water is practically at the boiling point. Measurements made, at the instigation of the Flannery Bolt Company, show that at times, due to the feed water's circulating back along the mud ring the water temperature in this region is lowered, but aside from this and the occasional formation of temperature "pockets" here and there the water is not far below the boiling point which it should have at the pressure indicated by the steam gage. This equalizing of temperature is the result of a number of factors, the most important of which are the following: turbulence caused by the rapid formation of steam, thermal circulation of the mass of boiler water, and surging resulting from the motion of the locomotive.

For the reasons mentioned above it is safe to assume that t_5 and t_6 of our diagram are practically the same. The remaining temperatures and the corresponding temperature drops from t_6 on to t_{10} are self explanatory in view of their similarity to those already considered. You will observe that no factor has been included to take account of scale on the surface of the wrapper sheet. This is because the deposit here is of very uncertain thickness and, what is more important; there is not, relatively speaking, a very large quantity of heat lost through the wrapper sheet and since the temperature drop is proportional to the heat flow, the effect due to scale on the outer sheet may be neglected. As a matter of fact, on account of its insulating nature, it causes a reduction in the loss of heat to the outside air.

THE COMPUTATION OF THE TEMPERATURE DIFFERENCE BETWEEN THE FIRESHEET AND THE WRAPPER SHEET

In all cases when we speak of the temperature of a sheet it should be understood to mean the average temperature across the section, or what is the same thing, the temperature at the middle of the thickness. As indicated by the arrows in the diagram, the two sheet temperatures are as follows:

Temperature of fire sheet $= T_f =$ average between t_2 and t_3,
Temperature of wrapper sheet $= T_w =$ average between t_7 and t_8.

The following computations take up the various temperature drops separately and in the order in which they may be determined most readily. The results obtained are over-all averages and do not apply to the fluctuations which are continually taking place in small local regions. A locomotive is often operated at a very high rate of heat transfer as compared with the ordinary stationary engine, consequently we may expect to get relatively greater temperature differences along the line of heat flow outward from the firebox.

THE TEMPERATURE DROP THROUGH THE FIRESHEET (t_2 - t_3)

The total evaporation surface of the boiler under consideration is 4,430 square feet. This includes the combined surfaces of the firebox, combustion chamber, flues, and arch tubes. Tests have shown that the evaporation in this boiler when operated at highest firing rate is 66,491 pounds of water per hour. Therefore, the evaporation per square foot of heating surface for forced firing would be 66,491 ÷ 4,430 or 15 pounds. Evidently the rate of evaporation from the forward portions of the tubes is far less than this, while that from the firebox and combustion chamber is far greater. Some tests made on a locomotive boiler equipped with vertical steam-tight bulkheads which divided the length of the water-space into a number of distinct sections, gave valuable information concerning the distribution of the evaporation and we find that for a consumption of 90.8 pounds of bituminous coal per square foot of grate surface per hour, which is the maximum firing rate of this particular 4–8–2 locomotive, the part of the total boiler evaporation taking place in the firebox and combustion chamber is approximately 45.1%. This value is obtained from the curve shown in Figure 2.

Then the firebox and combustion chamber evaporation (maximum) = 66,491 × 0.451 = 30,000 lbs. per hour.

The area of firebox and combustion chamber = 292 square feet.

Then E, the evaporation per square foot of firebox and combustion chamber surface is given by:

$$E = \frac{30.000}{292} = 102.7 \text{ lbs. per hour.}$$

CURVE SHOWING THE PORTION OF THE TOTAL BOILER EVAPORATION PRODUCED BY THE FIREBOX SURFACE

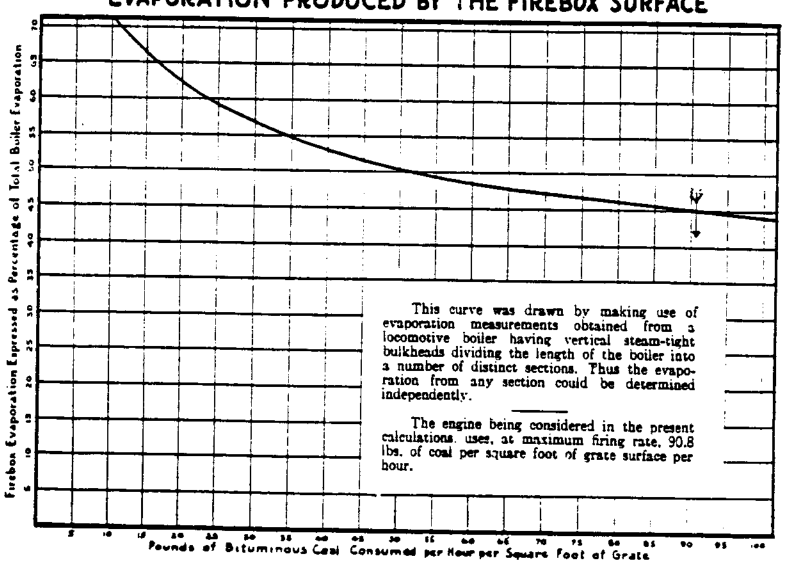


FIGURE 2

In order to find the difference in temperature between the water and fire sides of the firebox sheet we will use the conductivity equation:

$$Q = \left(\frac{K}{b}\right) A\, z\, (t_2 - t_3) \qquad \text{in which:}$$

K = Thermal conductivity, (of boiler plate in this case).

Q = Total heat in B.T.U. passing through the plate in z hours.

A = Area of the plate in square feet.

b = Thickness of the plate in feet.

t_2 and t_3 are the temperatures (°F) on the fire and water sides of the sheet, respectively.

By transposing the conductivity equation we have:

$$(t_2 - t_3) = \frac{Q}{Az} \times \frac{b}{K}$$

We see from the above notation that the quantity $\frac{Q}{Az}$ equals the heat passing through a square foot of plate in an hour.

In this case, the heat transmitted is taken up by the boiler water, so we have;

$$\frac{Q}{Az} = E \times (\text{B.T.U. absorbed per pound of water}).$$

The quantity in parenthesis equals the latent heat of vaporation plus the heat required to raise the feed water to the boiling point. At a normal working pressure of 200 lbs. the boiling point is 388° F. and we find from the tables that the latent heat is 838 B.T.U. per pound and the heat required to raise the feed water from 50° F. to 388° F. = 343 B.T.U. per pound, (average specific heat for this temperature range = 1.016).

Then: $$\frac{Q}{Az} = E \times (838 + 343)$$

Substituting for E its value 102.7 lbs. and adding the numbers in parenthesis gives:

$$\frac{Q}{Az} = 102.7 \times 1{,}181 = 121.289 \text{ B.T.U. per square foot}$$

per hour transmitted through the firebox surface.

Now K, for Boiler plate = 25.4 in B.T.U. per square foot of area per hour per °F. for a plate 1 foot thick; $b = \frac{3}{8}'' \times \frac{1}{12} = \frac{1}{32}$ foot.

Then the temperature drop across the firesheet is;

$$(t_2 - t_3) = \frac{121.289}{25.4 \times 32} = \frac{121,289}{812.8} = 149° \text{ F.}$$

THE TEMPERATURE DROP CAUSED BY SCALE ON THE WATER SIDE OF THE FIRESHEET

It is impossible to make an accurate computation of this factor because the thermal conductivity of scale varies over a wide range, probably from 0.2 to 2.0 B.T.U. per square foor per hour per °F. for a thickness of 1 foot, the particular value for each case depending upon the hardness, fineness of structure, and composition. A value which would probably approximate the average condition would be K = 1.0

Then our equation for temperature difference taken from the previous problem becomes:

$$(t_3 - t_4) = \frac{Q}{Az} \times \frac{b}{K}$$

In which t_3 and t_4 are the boundary temperatures of the scale deposit on the firesheet side and on the waterside, and

b = the thickness of the scale in feet.

If we substitute the value of $\frac{Q}{Az}$ as already determined and put in the numerical value of K, we have;

$$(t_3 - t_4) = 121.289 \text{ b}$$

Since the foot is a large unit for indicating scale thickness, we will express b in thousandths of an inch and call it b′. Then since there are 12 inches in a foot;

$$b' = \frac{b}{12,000}$$ and we have:

$$(t_3 - t_4) = \frac{121.289}{12.000} b' = 10.1 b' \text{ in degrees F.}$$

For a concrete example, we will assume (as a working average for a boiler in a fairly good water district) that the scale is 0.012″ thick, that is, b′ = 12

Then: $(t_3 - t_4) = 10.1 \times 12 = 121°$ F.

This temperature drop would almost vanish if the sheet were thoroughly cleaned of scale. On the other hand, in bad water districts boiler scale is often found a sixteenth of an inch or more in thickness. In such cases $(t_3 - t_4)$ would have much larger values even though the conductivity (K) should approach the maximum value of 2.0 B.T.U. per square foot per hour per °F. for a foot thickness.

In order to emphasize the importance of keeping the firebox sheets clean by the use of proper feed water treatment we point out that each 0.001″ of boiler scale may cause a rise of 10°F. in the operating temperature of the sheets.

THE TEMPERATURE DROP THROUGH THE FILM ON THE WATERSIDE OF THE FIREBOX SHEET

The film of steam and quiet water close to the water side of the firebox sheet retards the flow of heat as already mentioned and makes a considerable temperature difference between the water and the surface of the boiler scale.

The equation for this difference of temperature is:

$$(t_4 - t_5) = \frac{Q}{Azk}$$

in which the notation is the same as used before, except that k is the conductance of the film, not the conductivity (conductivity is a specific term which must be divided by the thickness to give the conductance of a body). The advantage of using conductance in this case is that it enables us to find the temperature drop across the film without knowing its thickness.

Now; $\frac{Q}{Az} = 121,289$ B.T.U. per square foot per hour

as before, and k, as obtained by using a curve (See Figure 3) made from some experimental data of Clement and Garland, = 960 B.T.U. per square foot per hour per °F. for the above rate of heat transfer to water at an average temperature of 110° F., assuming also a convection current in the boiler of 1.2 feet per second.

EXPERIMENTAL DETERMINATIONS OF HEAT CONDUCTANCE OF THE QUIET LAYER OF BOILER WATER IN CONTACT WITH THE FIRE SHEET

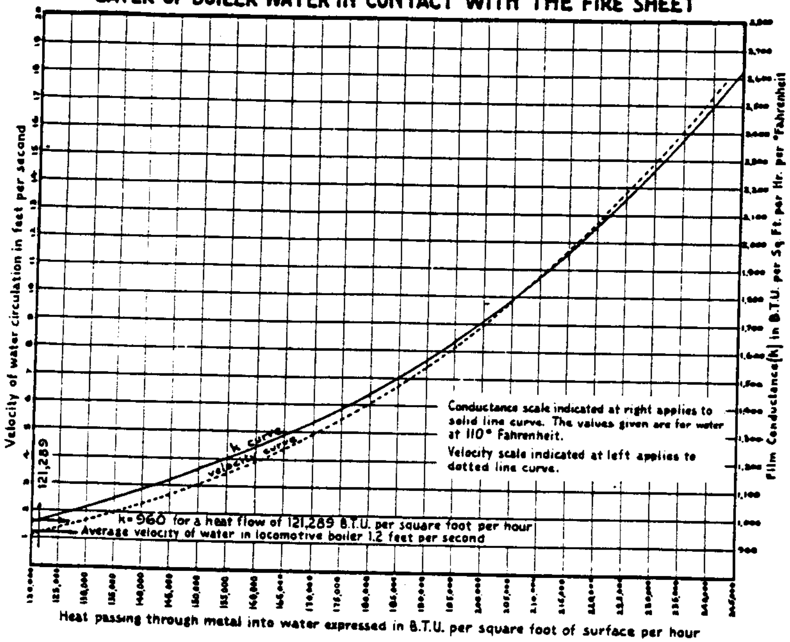


FIGURE 3

We believe this to be a very suitable value for the velocity of flow of the mass of water because it is the average of a large number of measurements made at various locations in the waterspace of a typical radially stayed boiler.

This value of k is not the one to be used in the present problem however, as we must make two corrections in order to reduce the data of Clement and Garland to the physical conditions which exist in a locomotive boiler. The first of these corrections is necessary because the water in the locomotive boiler is at a much higher temperature than that in the experimental apparatus. The second correction is of the nature of a shape factor, since a locomotive boiler is radically different in its dimensions from those of the simple cylinders used in securing the ordinary tabulated data on heat flow. This last correction is, of necessity, more or less of an approximation.

FIRST CORRECTION

The film conductance (k) varies with the fluidity (J) of the water as follows:

k is proportional to $J^{0.8}$

and J varies with the temperature of the water in accordance with the temperature fluidity curve shown in Figure 4.

From this curve we see that J = 1.6 at 110° F. the average temperature of the water in the experimental apparatus, and J = 7.05 at 388° F., the temperature of the water in the locomotive boiler.

Therefore, if we let k′ equal our corrected value of k, we have:

$$k' = k \frac{7.05^{0.8}}{1.6^{0.8}}$$

Substituting for k the value 960 obtained from the curve (Figure 3) gives;

$$k' = 960 \frac{7.05^{0.8}}{1.6^{0.8}} = 960\ (4.4)^{0.8} = 960 \times 3.27 \text{ or,}$$

$$k' = 3{,}139 \text{ B.T.U. per square foot per hour per } {}^\circ\text{F.}$$

CURVE SHOWING THE VARIATION OF THE FLUIDITY OF WATER WITH TEMPERATURE

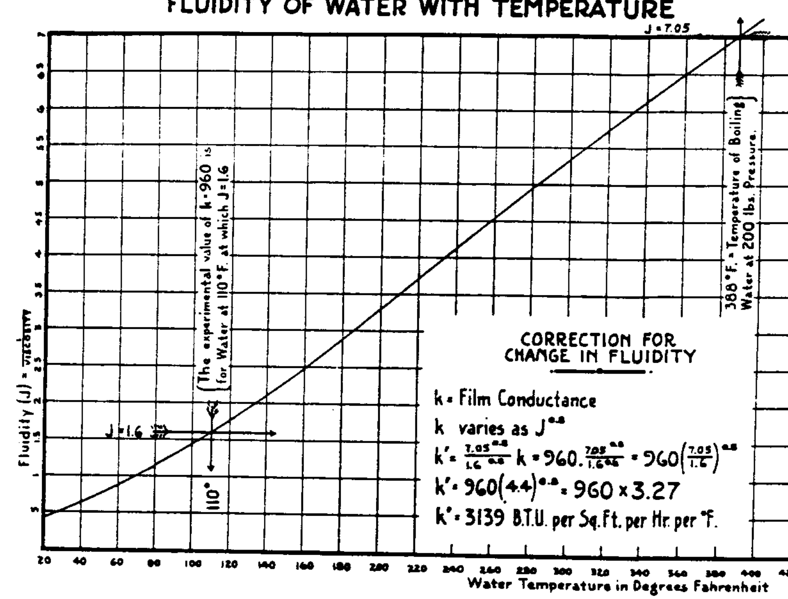


FIGURE 4

SECOND CORRECTION

It has been found experimentally that the film conductance (k) is also a function of the shape of the apparatus through which the heat flow takes place, thus k is proportional to $S^{0.2}$, where S is the shape factor, a quantity which varies as the quotient of the area of the metal surface through which the heat flow takes place divided by the volume of the water space. This makes S inversely proportional to the hydraulic radius and also inversely proportional to what we will call an Effective Diameter (D) of the metal boundary of the water space.

So we have;

$$k \text{ is proportional to } \left(\frac{1}{D}\right)^{0.2}$$

Now the diameter in the case of the experimental apparatus used by Clement and Garland was 1 inch, while the Effective Diameter of the locomotive firebox as figured on the basis of the surface and volume is, as near as we can approximate it, 27.8 inches. Therefore if we designate the final corrected value of k by the symbol k″, and let D′ and D″ equal the Effective Diameters of the experimental apparatus and the firebox respectively we may write;

$$k'' = \left(\frac{D'}{D''}\right)^{0.2} \times k' \qquad \text{then,}$$

$$k'' = \left(\frac{1}{27.8}\right)^{0.2} \times k' = \frac{k'}{1.945} = 0.515\ k'$$

and since k′ equals 3,139 we have;

$$k'' = 0.515 \times 3{,}139 = 1{,}615 \text{ B.T.U. per sq. foot per hour per °F.}$$

In arriving at the value of this quantity k″, it has been necessary to extrapolate somewhat and to use approximations for some of the terms. Therefore, it is to be expected that considerable error may be present in the result. On this account we have computed k″ by two other methods in order to check the results. By using McAdams and Frost's Empirical formula, we obtained k″ = 1,080 B.T.U. per square foot per hour per °F. and by means of another formula we got k″ = 1,460 B.T.U. per square foot per hour per °F.

These results indicate that our value of k″ = 1,615 is at least of the right order of magnitude and since it is probably more nearly applicable to the present case, we will use it in computing the temperature drop across the steam and water film $(t_4 - t_5)$.

By the substitution of k″ for k, the temperature difference equation which we have already mentioned as being applicable to this case becomes,

$$(t_4 - t_5) = \frac{Q}{Az} \times \frac{1}{k''}$$

and since $\frac{Q}{Az}$ equals 121,289 and k″ equals 1,615,

$$(t_4 - t_5) = \frac{121{,}289}{1{,}615} = 75° \text{ F.}$$

An examination of the works of John Blizard and H. Kreisinger shows that these men obtained by thermocouple measurements on stationary boilers, film temperature drops of from 20°F. to 30°F. As we would expect considerably greater differences in cases of locomotives in heavy duty the value which we have just obtained is probably not very far amiss.

THE TEMPERATURE DROP ACROSS THE WRAPPER SHEET $(t_7 - t_8)$

If no heat were lost through the lagging and exposed outer surface of the boiler, there would be no difference in temperature across the wrapper sheet, but since there are radiation and convection losses, a fall of temperature results. In order to determine the value of this temperature drop $(t_7 - t_8)$ we must find the rate of heat loss.

The total outside boiler surface of the locomotive which we are considering = 817 square feet.

The total quantity of water evaporated by the boiler per hour at the maximum firing rate = 66,491 pounds.

The number of B.T.U. required to heat and evaporate 1 pound of water = 1,181 as obtained before, so the total quantity of heat taken up by the water = 66,491 × 1,181 = 78,525,871 B.T.U. per hour.

Measurements of radiation and outside convection losses taken with engines at rest run from 0.6% to 1.3%, according to experiments conducted by Blizard, Kreisinger, and others.

Taking as an average value 1%, we will apply a correction for a locomotive velocity of say 30 miles per hour. This we find from published data to be approximately 300% increase in heat loss, making a total of 4% at this velocity.

Then if H = the total radiation and convection loss:

$$H = \frac{4 \times 78{,}525{,}871}{100} = 3{,}141{,}035 \text{ B.T.U. per hour.}$$

Our quantity $\frac{Q}{Az}$ = the rate of heat transfer through unit area.

Therefore, $$\frac{Q}{Az} = \frac{H}{A}$$

where A is the surface area, in this case equal to 817 square feet.

Then:

$$\frac{Q}{Az} = \frac{3{,}141{,}035}{817} = 3{,}850 \text{ B.T.U. per square foot per hour.}$$

Now we can apply our original temperature difference equation and determine $(t_7 - t_8)$ as follows:

$$(t_7 - t_8) = \frac{Q}{Az} \times \frac{b}{K}$$

The conductivity, K = 25.4 B.T.U. per square foot per °F. per hour for a plate one foot thick as before.

The thickness of the wrapper sheet is ⅝″, so, $b = \frac{5}{8} \times \frac{1}{12} = \frac{5}{96}$ feet.

Then the temperature drop across the wrapper sheet is:

$$(t_7 - t_8) = \frac{3{,}850}{25.4} \times \frac{5}{96} = 7.9° \text{ F.}$$

Here we have lumped the radiating surface all together without dividing it into two parts, i.e.: lagged and unlagged. So before going farther, we will check this result by a more analytical method in which we will consider each kind of surface separately.

We find by looking through the publications, that for boilers fired at a high rate, the radiation and convection loss is about 6 B.T.U. per square foot per °F. from the lagged surface and about 8 B.T.U. per square foot per °F. from the unlagged portion. These values must be multiplied by the differences in temperature between the outer surface and the atmosphere, and must be corrected for velocity.

Second Method of Computation

Assuming that the temperature of the outdoor air is 50° F. we can proceed as follows:

The loss through the unlagged surface = 8 × (367° — 50°) = 8 × 317 = 2,536 B.T.U. per square foot per hour.

The quantity 367° being approximately the outside temperature of the unlagged sheet.

Increasing this value by 300% in order to correct for the velocity of the locomotive, and making use of our conventional notation for the rate of passage of heat through unit surface area, we have:

$$\left(\frac{Q}{Az}\right)_{\text{unlagged}} = 4 \times 2{,}536 = 10{,}144 \text{ B.T.U. per square foot per hour.}$$

Since the conductivity is 25.4 and the thickness of the plate is 5/96 of a foot as before, the temperature drop across the unlagged portion of the wrapper sheet may be obtained from the conductivity equation,

$$(t_7 - t_8)_{\text{unlagged}} = \frac{Q}{Az} \times \frac{b}{K}$$

by substituting the numerical values for the different symbols. Thus,

$$(t_7 - t_8)_{\text{unlagged}} = \frac{10{,}144 \times 5}{25.4 \times 96} = 29.0° \text{ F.}$$

For the lagged portion of the wrapper sheet we have:
The loss through the lagged surface = 6 (150° — 50°) = 600 B.T.U. per square foot per hour. 150° F. being the approximate value of the surface temperature of the jacket which covers the lagging.

Correcting as before for velocity gives us:

$$\left(\frac{Q}{Az}\right)_{\text{lagged}} = 4 \times 600 = 2{,}400 \text{ B.T.U. per square foot per hour.}$$

Then the temperature drop across the lagged portion of the wrapper sheet is:

$$(t_7 - t_8)_{\text{lagged}} = \frac{2{,}400 \times 5}{25.4 \times 96} = 4.9° \text{ F.}$$

We will now find the average value of this temperature difference for the whole boiler. Approximately ¼ of the outer surface of the boiler is unlagged and ¾ lagged: so we have

$$(t_7 - t_8)_{\text{average}} = \frac{1}{4} \times 29.0 + \frac{3}{4} \times 4.9 = 10.93° \text{ F.}$$

This is the average value of the fall of temperature across the wrapper sheet as obtained by the second method of computation; whereas the first method gave 7.9° F. If we use a figure half way between the two, i.e.,

$$(t_7 - t_8) = 9.5° \text{ F.}$$

we will not be very far wrong for the given conditions. If the outdoor air were much warmer, say 90° F. and the engine velocity were very slow $(t_7 - t_8)$ would be almost negligible.

The Temperature Drop Through the Still Water Film at the Wrapper Sheet $(t_6 - t_7)$

This quantity is practically negligible as the heat flow through the wrapper sheet is relatively small, but since we have already determined the latter, we can easily compute $(t_6 - t_7)$ by the same method that we used in determining the temperature drop across the steam and water film $(t_4 - t_5)$. On working this out we find that $(t_6 - t_7)$ is approximately 3° F. Here also as in the previous case the drop in temperature would practically vanish in very hot weather if the engine were moving slowly.

The Difference in Temperature Between the Firebox and Wrapper Sheets $(T_f - T_w)$

We are now ready to sum up the various temperature drops between the two sheets. As already mentioned, we will consider that the temperature at the central position in the cross section of each sheet represents the sheet temperature, for the relative expansion depends not on the surface temperatures of the metals but on their average temperatures.

T_w = The average temperature of the wrapper sheet.

T_f = The average temperature of the firesheet (See Figure 1).

Then the difference in temperature between the two sheets $(T_f - T_w)$ equals the sum of the following:

One-half of the temperature drop across firesheet, $[\frac{1}{2}(t_2 - t_3)]$	75° F.
The temperature drop across the scale deposit, $(t_3 - t_4)$	121° F.
The temperature drop across the steam and water film, $(t_4 - t_5)$	75° F.
The temperature drop across the still water film at the outside sheet, $(t_6 - t_7)$	3° F.
One-half of the temperature drop across the wrapper sheet, $[\frac{1}{2}(t_7 - t_8)]$	5° F.
Therefore, $(T_f - T_w)$ is equal to	279° F.

It is perhaps worth while mentioning here that there are two additional minor corrections which we might easily apply, but which have been omitted because they are opposite in effect and consequently largely neutralize each other. One of these factors is that a locomotive which is evaporating 66,491 pounds of water per hour and at the same time losing through the wrapper sheet some 3% or 4% of its energy, must have a correspondingly larger amount of heat passing from the firebox into the water. This would tend to increase the temperature differences along the path of heat flow. The second factor, which is opposite in effect, is that with one inch diameter staybolts spaced in the firebox four inches apart each way, 5% of the water side of the firesheet consists of staybolts projecting into the water. These iron projections form easier paths of flow for the heat than would otherwise exist; consequently the radial flow of heat from the firebox is disturbed in the regions near the staybolts; some of the heat being shunted into and along the bolts. The effect of this is to decrease the values of $(t_3 - t_4)$ and $(t_4 - t_5)$ slightly. But since, as already stated, these factors largely correct each other, we will pass them by.

Since the temperature of the boiler water, at 200 pounds gage pressure, is 388° F., it is apparent that the wrapper sheet temperature,

$$T_w = 388° - (3 + 5)° = 380° \text{ F.}$$

and the firesheet temperature,

$$T_f = 388 + (75 + 121 + 75)° = 659° \text{ F.}$$

If there were no scale on the water side of the firesheet then the temperature at the mid-section of the sheet, T_f would be lowered by 121° F., so that for the case of a perfectly clean sheet T_f would be only 538° F. at the maximum firing rate. However, this ideal condition is never quite attained, and unfortunately, there have been cases in which the scale on the sheets has been allowed to accumulate to such an extent that blistering at the fire side of the sheet and even yielding of the sheet between staybolts has resulted. Let us analyze this to find out what thickness of scale would be sufficient to bring about this condition.

Suppose we were to operate our boiler in a scaly condition such as would cause the firesheets, or portions of them, to reach temperatures around 900° F., then the surfaces next to the fire would reach 975° F., a dull red heat. At this temperature scaling would be accelerated and, in time, blistering might even occur in places. Furthermore, when the temperature of ordinary firebox steel is raised from 538° F. to 900° F., the tensile strength is reduced to approximately one-half of the value which is characteristic of the material at the lower temperature.

We would then have, in addition to the accelerated oxidation of the sheets, a gradual yielding which would, sooner or later, cause such troubles as leakage around the staybolts, cracking and bulging of the sheets, and various other complications, to be discussed later on, which result from excessive expansion of the firesheets.

This certainly pictures a decidedly undesirable condition in the operation of a boiler. Yet just such a condition may arise from a mere scale deposit, hardly over 1/32" in thickness, adhering to the water side of the firesheet.

The scale has caused the sheet temperature to rise from 538° F. to 900° F., or 362° F. in excess of the ideal condition. As already shown, each 0.001" of scale may increase the temperature of the sheet 10° F. Then it would require a layer of scale only 0.036" in thickness to bring about the conditions described. This again emphasizes the importance of keeping the sheets reasonably free from scale. As previously mentioned, scale deposits vary widely in their thermal conductivities. One type of scale of relatively high conductivity might have to be much thicker than 0.036" to produce the conditions described, while another of lower conductivity than the average might be just as troublesome, though not as thick.

Very little has been done in the way of actual pyrometric determinations of the temperature differences which have been computed above. Various investigators have made a number of measurements of some of them, while others have not been experimentally determined at all. However, such data as is available bearing on these problems serves to substantiate the results of the computations to such an extent that it is reasonably certain that in fireboxes similar to the one considered the temperature difference between the two sheets frequently reaches and, as just illustrated, may at times materially exceed the value of 279° F., which we have determined. Consequently this value will be used in computing boiler deformations and staybolt stresses in the paragraphs which follow.

Deformations Caused by the Difference in Temperature Between the Inner and Outer Boiler Sheets of a Locomotive in Service

When iron or steel is heated it expands, and when a portion of a fabricated steel structure is heated to a higher temperature than the remainder, a certain amount of warping and bending is brought about in order to take care of the excessive expansion of the hotter portion. The amount of the warping depends upon the amount of the temperature inequality, and the magnitude of the disruptive stresses which are brought into play depends upon the rigidity of the structure.

In designing steel equipment, which is subjected to repeated heatings and coolings of an irregular nature, it is often difficult to secure sufficient flexibility to insure against cracking and failure due to thermal deformations.

The locomotive boiler is a typical example in regard to this difficulty. The greater temperature of the firebox sheet relative to the wrapper sheet brought about by direct radiation from and contact with the flames, introduces stresses which cause considerable anxiety among the mechanical officers of the railroads. We have shown, by means of experimental investigations and computations, the magnitude of the temperature differences which might exist between the inner and outer sheets of a 4–8–2 locomotive at times of maximum firing rate and therefore maximum rate of heat transfer from the burning fuel to the water in the boiler.

In order that the reader may easily visualize the changes in temperature as they occur along the line of heat flow or transfer from the burning gases outward through the boiler, the diagram shown in Figure 1, has been prepared by making use of a theoretical average flame temperature together with the temperature differences which occur as the heat travels through the different zones or layers, as previously determined.

AVERAGE THEORETICAL TEMPERATURE OF THE BURNING GASES IN A LOCOMOTIVE FIREBOX

An approximate value of the flame temperature in the firebox, combustion chamber, and flues of a locomotive may be obtained as follows. The increase in temperature, in degrees Fahrenheit, which takes place in a quantity of material W, is equal to the amount of heat gained by the material (H, expressed in B.T.U.), divided by the product of its weight in pounds W, and its specific heat S. The actual resulting or final temperature will therefore be the sum of this quantity and the original temperature T_o, that is:

$$T = \frac{H}{W\,S} + T_o$$

Now, in the case under consideration, the heat gained is equal to the total heat produced by the combustion of the fuel minus the amount which is lost by direct radiation to the relatively cool surfaces within which the burning gases are confined. The weight of the material heated is equal to the weight of the fuel burned plus the weight of the quantity of air required for the combustion. If for simplicity we neglect the weight of ash and cinder residue this quantity may be considered equal to the weight of the gases of combustion, and since the original temperature is the temperature of the outside air and the fuel in the tender we may write our equation in the following form:

$$T = \frac{H_F - H_R}{W_G \times S_G} + T_A$$

in which,

T = The flame temperature average for the entire combustion space.

T_A = The temperature of the outside air. This is assumed to be 50° F., the same value that has been used in previous computations.

H_F = The heat of combustion of the fuel in B.T.U. per pound.

H_R = The heat absorbed by the firebox and tubes by direct radiation from the fuel bed and flames, i.e. in B.T.U. per pound of coal.

W_G = The weight of the combustion gases per pound of coal burned.

and S_G = 0.238 B.T.U. per pound per degree F. This is an approximate value of the specific heat of the fuel, air, and combustion gases averaged over the entire heating and burning action.

A fair value of the heat of combustion of soft coal is 13,720 B.T.U. per pound. This figure was obtained by taking the average B.T.U. value of 178 different kinds of Bituminous Coal. It has been found that in locomotives used today approximately 52% of the heat of combustion is absorbed by direct radiation to the inner surfaces of the boiler. This includes radiation to the surfaces of the firebox, arch tubes, combustion chamber, and to parts of the flue and tube surfaces adjacent to the firebox since part of the heat energy of the flame shines directly or obliquely into the ends of the flues and tubes. Then, too, the radiating flames travel along for some distance into the tubes. When these factors are considered this value of 52% direct radiation would seem to be in very close agreement with the percentage of total evaporation which takes place at the firebox, as determined on Page 6.

Then: $H_R = \frac{52 \times 13{,}720}{100} = 7{,}134$ B.T.U. per pound of coal.

There is enough oxygen in about 10.2 pounds of air to burn a pound of average bituminous coal if all of the oxygen is used. Experience has proven however, that unless an amount in excess of this is used the combustion is incomplete, a certain amount of the carbon of the coal passes off as carbon monoxide, and a part of the oxygen remains uncombined. The amount of excess air used varies from 50 to 100% according to conditions. With good firing it should be possible in locomotive operation to keep this down to 70%. In this case the total air used would be 10.2 x 1.7 or 17.34 pounds per pound of coal burned. Then W_G the weight of the combustion gases is equal to 17.34 plus 1, or 18.34 pounds per pound of coal, assuming that practically all of the coal is consumed and is therefore rendered gaseous through combination with the oxygen of the air.

The approximate temperature of the hot gases may now be obtained by substituting in the equation given above and solving as follows:

$$T = \left(\frac{13{,}720 - 7{,}134}{18.34 \times 0.238}\right) + 50$$

$$T = \frac{6{,}586}{4.365} + 50 = 1{,}559° \text{ F.}$$

If this same process of combustion were to take place in a chamber having walls made of highly insulating materials, the temperature would rise to approximately 3,000° F.

Temperature Diagram of Locomotive Boiler

Having the flame temperature and the temperature of the boiler water, which is 388 degrees F. for a gauge pressure of 200 pounds per square inch, a complete boiler temperature diagram may be made by employing, along with these values, the various temperature losses brought about by the different layers encountered by the heat as it flows outward from the firebox. Such a diagram is shown in Figure 5. This supplements the diagram shown in Figure 1, for we now have

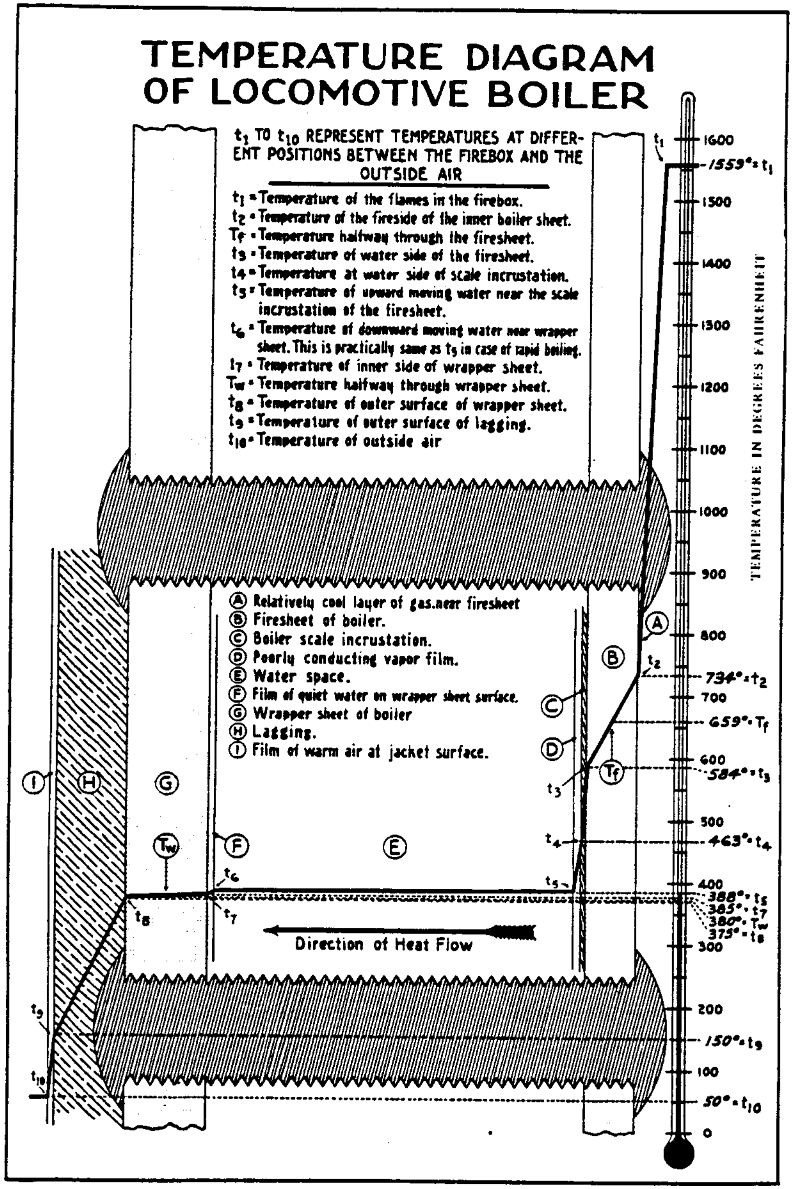


FIGURE 5

actual temperatures at the various locations instead of merely temperature differences across the various adjacent layers. It may be noted that the greater drop in temperature occurs where the flames come in contact with the fire sheet (See t_1 and t_2 of the Figure). This drop from 1,559 degrees F. to 734 degrees F., or a decline of 825 degrees F., is brought about by a film of relatively quiet air only a small fraction of an inch in thickness. Were it not that 50 percent of the heat of the flames is taken up by the fire sheet through direct radiation, this sudden drop in temperature would be much greater. The more thorough the scouring of the fire sheet, by rapidly moving flames, the less will be this temperature difference and its accompanying loss, but even under the best conditions it retains a high value. There are two other temperature drops due to quiet insulating films. One of these is from t_4 to t_5 of the Figure. This film is of steam and moisture in contact with the sheet. Any means of increasing the circulation in this vicinity would reduce the temperature drop and increase the boiler efficiency. The third quiet insulating film which appears in the diagram is the relatively slow moving layer of air in contact with the jacket (See t_9 and t_{10} of the Figure). This film, however, makes for efficiency since it retards the escape of heat to the outside air. High train velocities decrease this insulating film by the scouring effect of the air. The result is that the heat loss is increased and the boiler efficiency reduced. The surface temperature drop indicated in the diagram was computed for a train velocity of 30 miles per hour. The loss of heat from the boiler by exposure to the outdoor air is about 4 percent at this velocity.

The remaining regions of temperature decline shown in the diagram are in the boiler sheets themselves, "B" and "G" of Figure 5, in the scale deposit on the fire sheet "C", and finally in the lagging "H". These values have been computed on the basis of the conductivities of the materials and the rate of heat flow through them when the locomotive is being fired at its maximum rate. Under average conditions the temperature differences would be much less, but the values given are used because it is essential to compute the maximum amount of uncompensated expansion of the fire sheet and the maximum bending of the sheet and staybolts which results. It is true that there will not ordinarily be a temperature drop from 463 to 388 degrees F., on account of the insulating layer of boiler scale "C", on the water side of the fire sheet as indicated in the diagram. If the sheet were perfectly free from scale this quantity would be zero. However, with bad water, a condition such as that shown may easily exist, and it is essential that the fire sheets and staybolts be designed to withstand the most severe conditions that may be encountered in service.

THE UNCOMPENSATED EXPANSION OF THE FIREBOX SHEET

Now that we have the maximum difference of temperature between the two sheets,

$$(T_f - T_w) = 659 - 380 = 279 \text{ degrees F.},$$

we can compute their differential expansion. The amount of expansion which takes place due to a like rise of temperature in both sheets does not enter into the computation of staybolt stresses, for, if the temperature rises uniformly throughout the locomotive, it "grows" uniformly. Therefore in our computations we use only temperature differences.

Consider the *horizontal row of staybolts along the side just below the radials*. In the same 4-8-2 locomotive this row is approximately ten feet in length and the fire sheet "grows" more than does the wrapper sheet, due to its higher temperature. It expands from the middle in both directions. So in considering the movement of the firebox sheet with reference to the wrapper sheet at the position of the *front staybolt* in this row, there is an unbalanced *longitudinal* expansion taking place over a length of five feet or 60 inches of sheet.

This relative movement may be computed as follows:

If L = the length of the expanding sheet in inches,

l = the amount of the uncompensated expansion in inches, and

a = the thermal coefficient of expansion (.000007 for mild steel),

Then we have from our definition of coefficient of expansion:

$$l = aL\,(T_f - T_w),$$

in which T_f and T_w are the sheet temperatures as before.

Substituting in the equation the values given above we have:

$$l = .000007 \times 60 \times 279$$
$$= 0.117''$$

There is a vertical component of movement also which would be measured from the mud ring in case the staybolt were located above it. The mud ring forms a natural base for vertical movements because both the inner and outer sheets are rigidly fastened to this member.

The vertical and horizontal movements of the fire sheet at the position of any staybolt combine so as to give movements in the directions indicated approximately by the lines and arrows shown in Figure 6.

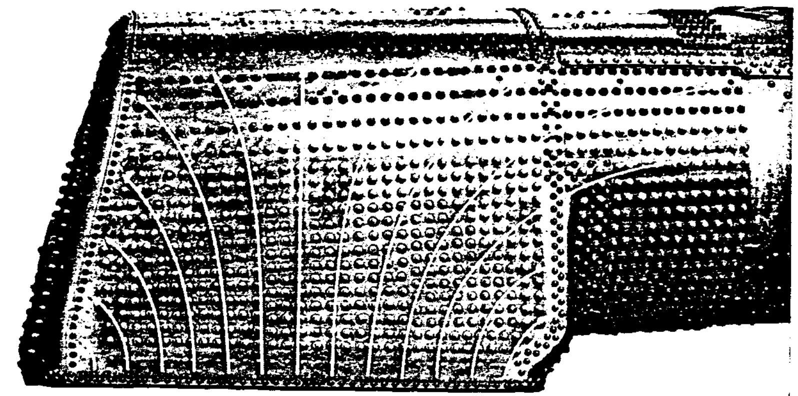


FIGURE 6—FIREBOX EXPANSION DIAGRAM
The arrows indicate the direction of movement of the firesheet, with reference to the wrapper sheet.

The flow of heat through the firesheet is much more rapid in some locations than in others. For instance, the portion of the sheet which is shielded by the brick arch transmits heat relatively slowly as compared with the adjacent portions which are in direct contact with the flames. For this reason the temperature elevation and the resulting expansion vary from place to place over the firesheet surface. Therefore, we cannot assume that the expansion diagram, or, even the equations, represent, in minute detail, the conditions at a small localized area. They represent rather the over-all effect which we wish to study.

The approximate value of the relative vertical and horizontal movements may be computed as above by using the transverse middle section for the longitudinal base, and the mud ring for the vertical base. The combined effect of the vertical and longitudinal movements of the fire sheet can be obtained approximately, when the position under consideration is not too close to the mud ring, by squaring each component and extracting the square root of their sum. This will be a diagonal movement of the firebox sheet with reference to the wrapper sheet.

For the particular staybolt under consideration the diagonal movement obtained by combining the longitudinal and vertical components is equal to 0.18″. As this staybolt is at a point considerably removed from the middle of the mud ring the relative movement of the sheets is rather large. At less remote points the movement would be proportionally less.

Some years ago, at the instigation of the Flannery Bolt Company, a series of investigations were conducted regarding the deflections of rigid and flexible staybolts and the movements of the firebox sheets in locomotives while they were being fired up or cooled down. In measuring the relative movements of the inner and outer sheets of locomotive boilers it was found that there occurred, for staybolt positions similar to that which we have been considering above, movements as great as 0.043″. In the experimental work on the cylindrical type boilers the locomotives were not fired at maximum capacity inasmuch as the tests were made in the round house instead of during a service run. We would, therefore, expect to get considerably larger values when the locomotives are in operation pulling heavy trains.

Another phase of the investigation throws a little light on the expansion problem. That is, there nearly always occurred somewhat greater movements in the case of the flexibly stayed boilers than in similar ones rigidly stayed. This points out to us that a considerable portion of the expansion of the firesheet is taken up by a bending or lateral warping of the firesheet itself. The amount of this will be computed later. It is somewhat more prominent in the rigidly stayed boilers than it is in those flexibly stayed.

THE RELATIVE BENDING OF FLEXIBLE STAYBOLTS AND FIRE SHEET DUE TO THE EXPANSION OF THE SHEET

When the fire sheet, due to its higher temperature, and therefore greater expansion, moves along with reference to the wrapper sheet as computed above, the staybolts which connect the two sheets must be deflected. If they are rigid staybolts they are bent oppositely at the ends forming a curve similar to the center portion of the letter S. If they are flexible staybolts the bend takes place mostly at the firebox end. The stresses brought into play by this flexing of the bolt where it is rigidly attached to the fire sheet introduce bending moments which cause the sheet to warp or wave between rows of staybolts. This bending of the fire sheet relieves the strains in the staybolts by amounts varying from a few per cent up to as much as 40 per cent depending upon the lengths and diameters of the bolts and upon the thicknesses of the sheets. The following computations will give the relation between the amount of bending which takes place in the staybolts as compared with that which takes place in the fire sheet. The deflections of a flexible staybolt, free to pivot at one end, is given by the equation,

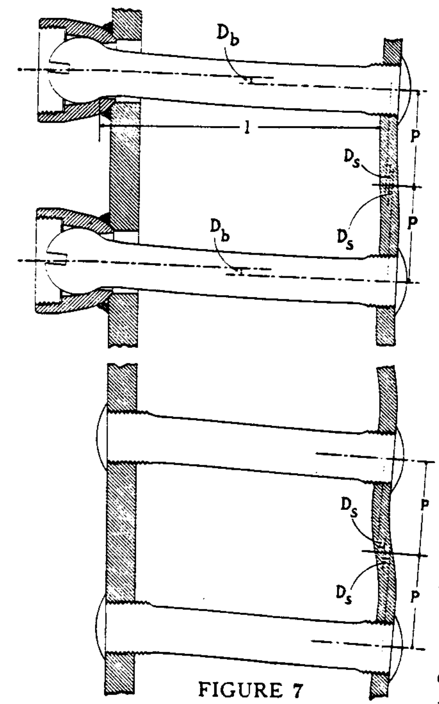


FIGURE 7

$$D_b = \frac{F\,l^3}{3E_b\,I_b} \quad \text{(See Figure 7)}$$

in which,

D_b = The total amount of bending or deflection of the bolt.

F = The side thrust or bending force on the headed end of the bolt due to the movement of the fire sheet.

l = The length of the bolt from the inner side of the head to the water side of the fire sheet.

E_b = The elasticity of staybolt iron, i.e., 26,000,000 pounds per square inch.

I_b = The moment of inertia of the bolt section.

This is $\frac{\pi r^4}{4}$

Now, in the case of a flexible staybolt, the bending moment $M_b = F \times l$, that is, the product of the bending force and the length of the bolt, so we have:

$$D_b = \frac{M_b l^2}{3 E_b I_b}$$

We will solve this for the bending moment in order to be able to compare it with the bending moment applied to the fire sheet. Transposing we have:

$$M_b = \frac{3 D_b E_b I_b}{l^2}$$

A similar equation applies to the warping or bending of the firebox sheet by the forces.

$$M_s = \frac{3 D_s E_s I_s}{p^2} \quad \text{(See Figure 7)}$$

in which,

M_s = The bending moment applied to the fire sheet by a staybolt which is being deflected.

D_s = ½ of the amount of the sheet deflection between two rows of staybolts, or the deflection which takes place in a length of sheet equal to p of Figure 7.

E_s = The elasticity of boiler plate which is approximately 28,000,000 pounds per square inch.

I_s = The moment of inertia of the rectangular sheet section of width equal to the distance between staybolts.

p = The lever arm of the force tending to bend the sheet, i.e., the distance from the center line of a row of staybolts to the neutral position or inflection point situated half way between two adjacent rows of staybolts.

Now,

$$I_s = \frac{\text{Staybolt pitch} \times (\text{sheet thickness})^3}{12}$$

$$= \frac{4 \times (\text{Sheet Thickness})^3}{12}$$

From the mechanical relations which exist between the bolt and the fire sheet, as shown in Figure 7, it may be seen that the bolt is being flexed by a force the moment of which, at the fire sheet, is M_b. In order that equilibrium may exist, the sheet must exert upon the bolt an equal and opposite moment. The latter consists of two moments, M_s below the bolt and M_s above the bolt, according to the diagram. Therefore,

$$M_b = 2M_s,$$

and,

$$\frac{3 D_b E_b I_b}{l^2} = 2 \times \frac{3 D_b E_b I_b}{p^2}$$

Eliminating I_b and I_s by making use of the values of these quantities as given above we have,

$$\frac{3 D_b E_b \pi r^4}{l^2 \times 4} = 2 \frac{3 D_s E_s 4 (\text{Sheet Thickness})^3}{p^2 \times 12}$$

Let us assume that we have a fire sheet ⅜″ (0.375″) thick and a flexible staybolt 4″ long by 1″ in diameter, with its threaded end upset to give a diameter of 1″ at the root of the thread. Then the effective radius would be 0.5″. Substituting the numerical values in the last equation, we now have,

$$\frac{3 \times D_b \times 26{,}000{,}000 \times 3.1416 \times \overline{0.5}^4}{\overline{4}^2 \times 4}$$

$$= 2 \frac{3 \times D_s \times 28{,}000{,}000 \times 4 \times \overline{0.375}^3}{\overline{2}^2 \times 12}$$

By transposing and solving we obtain,

$$D_s = 0.324\ D_b$$

That is, under these conditions, the portion of the sheet represented by the distance from the center line of a row of staybolts to the neutral position half way to the next row bends 0.324 times as much as the bolt bends over its entire length. It may be seen in Figure 7 that the actual bending or deflection of the sheet over the four inches between rows of staybolts is equal to 2 D_s, or twice the value computed above, or

$$\text{Total } D_s = 0.648\ D_b$$

A table of values of D_b and Total D_s, computed by means of the formula, for different lengths of bolts may be found on Page 33. The table also contains, for comparison purposes, similar values of D_b and Total D_s computed for the case in which rigid bolts of the same length are employed. We will show mathematically how the latter values may be obtained.

COMPARISON OF FIBRE STRESSES IN RIGID AND FLEXIBLE STAYBOLTS

As a step in the derivation just mentioned we will first prove that the fibre stresses at the firebox end of a rigid staybolt, introduced by the bending of the bolt, are twice as great as they are for a flexible staybolt which can move freely in its seat in the sleeve.

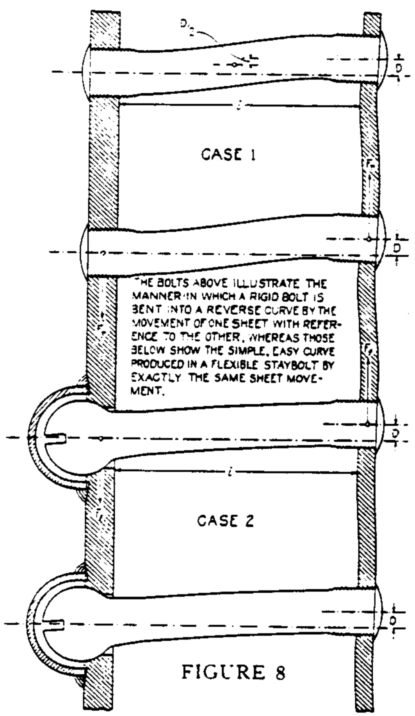


FIGURE 8

Let us consider, first CASE–1 of Figure 8. This illustrates a bolt or beam rigidly attached at both ends and having one supporting plate displaced or moved along in a direction parallel to its length.

We will let F_r = the bending force on the rigid bolt due to the sheet displacement, l = the free length of the bolt, that is the length between sheets, D = the deflection of the staybolts due to the movement of the sheet, E = the elasticity of the staybolt iron, I = the moment of inertia of the bolt section, and, M_r = the bending moment as applied to the rigid bolt at the position where it is a maximum, i.e., at the cross-section nearest to one of the sheets. From the theory of mechanics we obtain the following formula for the maximum bending moment of a beam supported and bent in this manner:

$$M_r = \frac{F_r\,l}{2}$$

As shown in the sketch, a rigid bolt is bent into a reverse curve with a neutral point in the middle where no bending takes place. When the sheet is displaced the neutral point moves along one-half of this displacement or an amount equal to D/2. The following mechanical relation exists between this shift in the neutral point and the other quantities mentioned above:

$$\frac{D}{2} = \frac{F_r\,(l/2)^3}{3\,E\,I}$$

$$D = \frac{F_r\,l^3}{12\,E\,I}$$

If we solve the first and third of the above equations for F_r, and then eliminate F_r by equating the right hand members we have, after transposing and simplifying:

$$D = \frac{M_r\,l^2}{6\,E\,I} \quad \text{(Deflection of rigid bolt, CASE 1)}$$

The important thing about this equation is that it involves the bending moment and the sheet movement.

We will now obtain a similar equation for CASE–2 of Figure 8, (See preceding page), in order that we will be able to compare the bending moments that the two types of bolts are subjected to. CASE–2 deals with the flexible staybolt and is like the problem of a beam or bolt rigidly attached at one end, flexibly attached at the other, and with one supporting plate displaced or moved along in a direction parallel to its length.

The symbols will have the same significance as in CASE–1 except in two instances. First, the magnitude of the bending force which will be applied to the flexible bolt will be different from F_r even though the displacement of the sheet is the same. It will obviously be less due to the simpler type of flexure which occurs in this type of bolt. We will call this force F_f. The values of the Forces F_r and F_f are not required, as these quantities cancel out in the transformation of the equations. Second, there will be a new value for the maximum bending moment which will be applicable to the case of the flexible bolt. We will call this M_f. The position at which the bending moment will have this maximum value will be at the cross-section of the bolt just in contact with the fire sheet. Its magnitude may be represented by the following equation;

$$M_f = F_f\,l$$

From the theory of mechanics, we have for this case:

$$D = \frac{F_f\ l^3}{3\ E\ I}$$

Solving each equation for F_f and equating the right hand members, as before, gives the following:

$$D = \frac{M_f\ l^2}{3\ E\ I} \quad \text{(Deflection of flexible bolt, CASE 2)}$$

Now let us compare a rigid and a flexible staybolt of practically the same free length and similarly located in the firebox. Then l will be the same in both cases and the deflection of both bolts will be the same.

Therefore we may equate the expressions for deflections derived in CASES 1 and 2 thus;

$$\frac{M_r\ l^2}{6\ E\ I} = \frac{M_f\ l^2}{3\ E\ I}$$

Now since the bolts of the two types which are being compared will of course be made of the same material and will be of the same diameter, E and I will be the same in the two cases and, therefore, the above equation will reduce to the following:

$$M_r = 2\ M_f$$

That is to say, the maximum bending moment at the fire sheet end of a rigid staybolt is equal to two times what it is at the fire sheet end of a flexible staybolt. Since that part of the fibre stress, caused by bending at the fire sheet end of a bolt, is proportional to the maximum bending moment, the same relation applies to this quantity also. We are not considering in this connection the portion of the fibre stress due to steam pressure as this is the same for either type of bolt. The total fibre stress at any point in a staybolt is of course equal to the sum of the stresses due to tension and bending. We might add that the fibre stresses at the firebox end of a staybolt are partially relieved by the bending or warping of the sheet, but as this is the case for the flexible as well as for the rigid staybolt we do not need to consider it where both flexible and rigid bolts are used in the same boiler. If we are comparing the fibre stresses in the bolts of a complete rigid installation with those of a similar complete flexible installation we cannot assume that the relative sheet movements will be the same, and, consequently, the proportionality represented by the equation will be modified. This will be discussed in more detail later.

THE RELATIVE BENDING OF RIGID STAYBOLTS AND FIRE SHEET DUE TO EXPANSION OF THE SHEET

Let us return to the equation which we have just derived, namely,

$$M_r = 2\ M_f$$

If we consider this in relation to the diagram of Figure 7, it is evident that for the same travel or movement of the sheet the rigid bolt subjects the sheet to a bending moment twice as great as does the flexible staybolt. Since the equation dealing with the warping or bending of the fire sheet (D_s) as previously stated on Page 28, is,

$$M_s = \frac{3\ D_s\ E_s\ I_s}{p^2},$$

we observe that the deflection is directly proportional to the bending moment to which the sheet is subjected. Therefore, in obtaining our values for the relative bending of fire sheet and staybolt where rigid bolts are used, it is merely necessary to double the results of the computations made for the various lengths of flexible bolts.

The table containing both series of values follows.

TABLE SHOWING THE RELATIVE BENDING OF THE STAYBOLTS AND FIRE SHEET COMPUTED FOR 1" DIAMETER STAYBOLTS HAVING 1⅛" DIAMETER UPSET ENDS THREADED AND SCREWED INTO A FIRE SHEET ⅜" THICK

Length of Staybolt Between Supports	Relative Bending of Fire Sheet and Staybolt	
	For Flexible Installations	For Rigid Installations
4 inches	$D_s = 0.648\ D_b$	$D_s = 1.296\ D_b$
5 inches	$D_s = .415\ D_b$	$D_s = .830\ D_b$
6 inches	$D_s = .288\ D_b$	$D_s = .576\ D_b$
7 inches	$D_s = .211\ D_b$	$D_s = .422\ D_b$
8 inches	$D_s = .160\ D_b$	$D_s = .320\ D_b$
9 inches	$D_s = .126\ D_b$	$D_s = .252\ D_b$
10 inches	$D_s = .102\ D_b$	$D_s = .204\ D_b$

The relations expressed in the table are only approximate as no correction has been made for the weakening of the sheet due to punching and tapping for staybolt application. However, the results are of the proper order of magnitude and serve to show the advantage of the flexible staybolt in regard to its ability to relieve the fire sheet of undue bending, by allowing it greater freedom of longitudinal movement.

This is a characteristic which is highly desirable when excessive expansion and contraction is involved. In other words, with the flexible assemblage, the deflection of the bolt is greater and the warping of the sheet is less than that which is encountered when a rigid installation is used.

The table also shows that for positions where the longer bolts are required, the movement is almost entirely absorbed by the bolt without any appreciable distortion of the fire sheet; whereas, in the positions requiring very short staybolts, severe bending or warping of the fire sheet is involved, especially when rigid staybolts are used.

A complete installation of flexible staybolts should therefore effect a material increase in the life of the side sheets, for where the side sheets are located, the water space is narrow and the bolts are very short.

The Linear Thermal Expansion of the Flues and Tubes of a Locomotive Boiler

WE HAVE computed the maximum amount of the relative thermal expansion which we may expect to get in our 4–8–2 locomotive firebox when the engine is being fired at its highest rate. Turning our attention to the forward portion of the boiler we will next determine, by means of analytical methods, what relative change in length the flues and tubes of such a locomotive undergo, with respect to the barrel of the boiler, under similar operating conditions. When this has been accomplished we will have the total amount of differential expansion, along the length of the boiler, which must be accounted for and distributed in the form of deformations of the materials of which the heating surface is composed.

If we first obtain the difference in temperature between the tubes and the barrel of the boiler, averaged over the entire length of the flues, for the maximum rate of firing of the boiler, the actual differential or relative expansion between the inner and outer portions follows immediately providing we know the distance between the front and back flue sheets and the coefficient of thermal expansion of the steel of which the flues and tubes are composed.

The desired difference in temperature can readily be computed with sufficient accuracy for present purposes by considering separately the individual factors which bring about the temperature difference. This method is identical with the one previously used in connection with the differential or uncompensated expansion of the fire sheet with reference to the wrapper sheet. When the separate parts, which make up the total temperature difference, have been determined they are added together and the sum is used in the computation of the relative thermal expansion. In accordance with this plan we will first determine the temperature drop which takes place across the metal walls of the tubes and flues in a direction perpendicular to their length.

THE FALL OF TEMPERATURE THROUGH BOILER TUBES AND FLUES

In an investigation referred to earlier it was found that the distribution of the heat absorption in the boiler as determined by the water evaporation is somewhat as follows:

Evaporation from firebox surface	45.1%
Evaporation from arch tube surface	4.4%
Evaporation from surface of tubes and flues	50.5%

The maximum rate of evaporation from the entire heating surface of our 4–8–2 locomotive boiler, as found by test, is 66,491 pounds of water per hour, so the maximum amount which can be evaporated from the surface of the tubes and flues is

$$66{,}491 \times 0.505 = 33{,}578 \text{ pounds per hour.}$$

The total surface of the tubes and flues is equal to 4,110 square feet. If then, we let E represent, in pounds per square foot per hour, the highest overall rate of evaporation which can be obtained from this surface we have;

$$E = \frac{\text{Tube and flue evaporation in pounds per hour}}{\text{surface area of tubes and flues}}$$

$$= \frac{33{,}578}{4{,}110} = 8.12 \text{ pounds per square foot per hour.}$$

Near the back flue sheet where the hot flames enter the tubes from the firebox the rate of evaporation is several times this overall average, while at the front flue sheet, which is 21.5 feet ahead of the rear sheet the evaporation drops off to such a small value that additional flue length would be uneconomical from the standpoint of heat recovery when disadvantages, such as added weight and sagging of flues, are considered.

Laboratory tests, carried on by a number of different investigators, have shown that the rate of transfer of heat from a gas to a water-backed metal tube through which the gas flows is given fairly accurately by the following exponential equation,

$$F_x = K_e^{-cx}$$

in which F_x is the rate of flow of heat to the wall of the tube at a point located at a distance X from the end of the tube at which the gas enters. K and c are constants, and e is the natural base of logarithms.

If the boiler tubes are very much in excess of two inches in diameter, this equation has been found to give results which differ materially from the measured values. Furthermore it is not ordinarily applicable to the extreme rear portion of the flues on account of the effect of direct radiation from the brick arch and the very hot flames in the firebox. This transfer of heat by direct radiation causes the first few inches of the tubes to absorb heat nearly as rapidly as does the flue sheet, but as the distance into the tube becomes greater the solid angle through which direct radiation may be received, from the hotter regions, becomes less and less until this end effect is negligible. It may, however, be noticeable for as great a distance as two feet in tubes of approximately two inches inside diameter and for even greater distances when larger tubes are used.

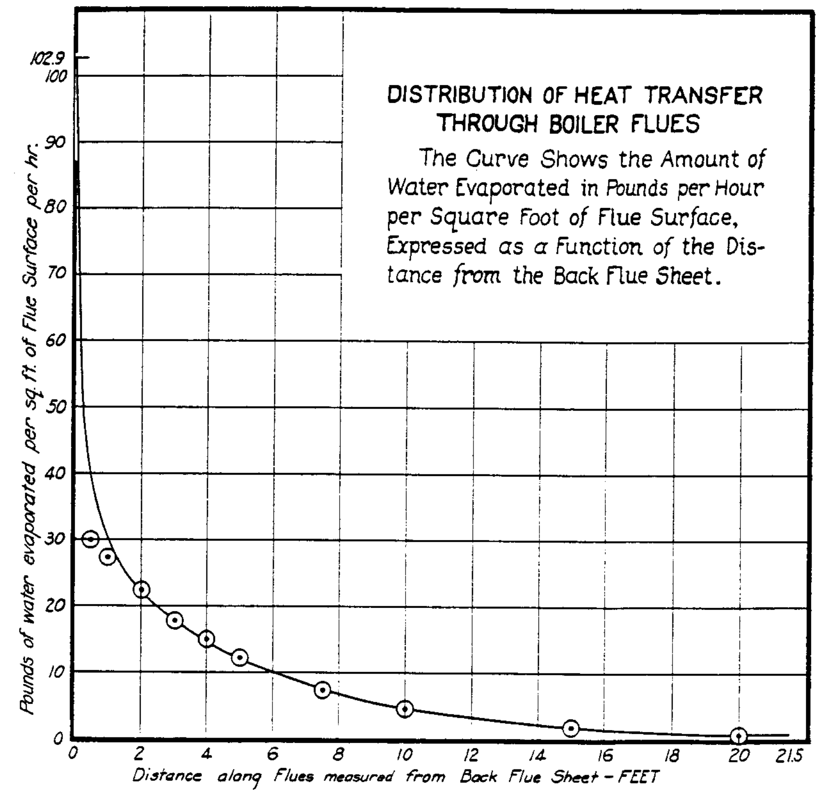


FIGURE 9

The curve of Figure 9 represents approximately, for the locomotive which we have been considering, the rate of transfer of heat from the combustion gases to the boiler tubes expressed as a function of the distance from the back flue sheet. Since the rate of evaporation of boiler water is proportional to the rate of heat transfer, we have expressed this variable, in the Figure, as pounds of water vaporized per hour per square foot of tube surface. The area of the surface between the curve and the horizontal base, measured in terms of the vertical and horizontal scales of units indicated in the figure, is equal to the rates of water evaporation summed up over the entire length of the tubes. If then, we divide this area by the length of the tubes (i.e. 21.5 feet) we will obtain the maximum overall rate of evaporation from the tubes. This will, of course, be equal to 8.12 pounds per hour per square foot which is the value of E obtained above.

The heat transfer curve (Figure 9) will be found to agree closely with the curve given by the equation

$$F_x = K_e^{-cx}$$

for values of X lying between 2 and 21.5 feet.

The following table of values of F_x computed from the formula for various distances from the back flue sheet has been prepared to illustrate this agreement. The constants K and c required in applying the formula are 33.3 and 1/5 respectively.

Values of X in Feet	F_x Expressed as Evaporation in Lbs. per Sq. Ft. per Hr.
½	30.13
1	27.29
2	22.3
3	18.3
4	14.98
5	12.2
7½	7.44
10	4.51
15	1.66
20	0.61

For comparison purposes the computed values of F_x given in the table have been indicated on the diagram (Figure 9) by means of circles. Not only do they show the applicability of the formula to the curve for values of X in excess of 2 feet, but they also show the influence of direct radiation in producing, near the back flue sheet, the relatively greater rate of heat transfer indicated by the portion of the curve between X = 2 and X = 0 feet.

Coming back to the problem of determining the overall temperature difference between the inner and outer surfaces of the flues and tubes we will make use of the conductivity equation

$$\Delta t_t = \frac{Q}{Az} \times \frac{b}{K}$$

in which Δt_t is the average or overall temperature difference between the inner and outer tube surfaces. Q is the heat in B.T.U. passing

through the walls of the tubes in Z hours, A is the total area of tube and flue surface, K is the thermal conductivity of the steel, and b is the wall thickness in feet.

Now the quantity $\frac{Q}{Az}$ is equal to the heat passing through a square foot of tube surface per hour. It can, therefore, be expressed as follows,

$$\frac{Q}{Az} = E \times \text{(B.T.U. per pound of water evaporated)}.$$

It requires 1,181 B.T.U. to heat one pound of feed water from an average out-door temperature to the boiling point, (388° F.) and evaporate it, so,

$$\frac{Q}{Az} = 8.12 \times 1{,}181 = 9{,}590 \text{ B.T.U. per hour per square foot.}$$

We will now determine, by means of the thermal conductivity equation given above, the overall average temperature drop through the tube wall for the maximum rate of firing.

The boiler tubes in this case are of number eleven B.W. Gage steel which is 0.120 inches thick or $\frac{0.12}{12}$ feet thick. The thermal conductivity of the steel is approximately 26 B.T.U. per square foot of area per hour per degree Fahrenheit based on a plate one foot thick.

Substituting in the equation we have for the temperature difference between the inner and outer tube surfaces,

$$\Delta T_t = \frac{9{,}590 \times 0.120}{26 \times 12} = 3.69° \text{ F.}$$

The superheater flues are made of number 9 B.W. Gage material which is of considerably greater thickness than that of the smaller flues. In this case $b = \frac{0.148}{12}$ feet, so if we use the same value for $\frac{Q}{Az}$ as before we have, from our conductivity equation,

$$\Delta T_f = \frac{9{,}590 \times 0.148}{26 \times 12} = 4.55° \text{ F.},$$

the temperature difference between the inner and outer surfaces of the large flues.

Since these temperature differences, Δt_t and Δt_f, are small and since, furthermore, they do not differ greatly from one another it is hardly worth while to consider them independently in the work which is to follow. We will therefore say that for conditions requiring heavy firing the average value of the temperature drop through the walls of the tubes and flues is 4° F. This quantity we will call Δt_t.

THE TEMPERATURE DROP CAUSED BY SCALE ON THE WATER SIDE OF THE FLUES

The formula for this is

$$\Delta t_s = \frac{Q}{Az} \times \frac{b'}{K'}.$$

in which b′ is the scale thickness in feet, and K′ the thermal conductivity of the boiler scale. We will assume the latter to be 1.0 as in the previous work:

Then

$$\Delta t_s = 9{,}590 \times \frac{b'}{1.0}$$

It is a well known fact that the scale which forms on the water side of the boiler flues and tubes usually accumulates to a greater thickness than it does on the firebox sheets. There is a certain amount of breaking up and shedding of the scale from the fire sheets due to the combined effects of the intense heat, the warping of the sheets, and the violent agitation of the water where it contacts the hot surfaces.

The tubes are much cooler than the fire sheets, the bending which takes place is not as concentrated, and the circulation at the surfaces is not as violent. These factors tend to allow the scale to build up so that it is frequently found to be ⅛″, 3⁄16″, or even ¼″ thick. There are, of course, cases in which the feed water is so soft, either naturally or through treatment, that little or no scale formation occurs. Such cases, however, are relatively few. In view of this let us use 1⁄16″ as a conservative value of the thickness of scale averaged over the entire surface of the flues.

Substituting this value for b′ in the equation we have,

$$b' = \frac{1}{16 \times 12} \text{ Ft.}$$

and

$$\Delta t_s = \frac{9{,}590}{16 \times 12} = 50° \text{ F.}$$

THE TEMPERATURE DROP THROUGH THE FILM ON THE WATER SIDE OF THE TUBES

As nearly as we can estimate from the data available the film conductance, k, of the water surrounding the tubes would be approximately 600 B.T.U. per square foot per hour per degree Fahrenheit for a heat flow of 9,590 B.T.U. per hour per square foot of tube surface for a water temperature of 110° F. Since, however, the water temperature during operation is 388° F. its fluidity is much higher, being 7.05 as compared with 1.6 at the lower temperature (See Figure 4, Page 11).

Under conditions of good circulation the film conductance, k, varies as the fluidity of the water to the 0.8 power, therefore the conductance at the higher temperature is,

$$k' = \frac{\overline{7.05}^{\,0.8}}{\overline{1.6}^{\,0.8}} \times k = 3.27\ k$$

$$k' = 3.27 \times 600 = 1{,}962 \text{ B.T.U. per sq. ft. per hr. per } ° \text{ F.}$$

Investigations have shown that the film conductance, k, is also a function of the shape and volume of the container which is conveying the heat to the water. The greater the wetted surface across which the heat travels as compared with the volume of moving water, and the greater the stirring or mixing of the water, the greater will be the value of k. This shape factor which we call S varies approximately as the quotient of the conducting surface area divided by the volume of the water space, in this case, the net volume of water in the barrel of the boiler. So S is roughly proportional to the reciprocal of the hydraulic radius and, therefore, to what we will call the effective diameter D. That is,

S is proportional to $\left(\frac{D°}{D}\right)$. Then, since it has been found that for reasonably good circulation k is proportional to $S^{0.2}$, we have for the final corrected value of k which we will call k″,

$$k'' = S^{0.2} \times k' = \left(\frac{D°}{D}\right)^{0.2} \times k',$$

in which D° is unity, being the value of the D which was used in obtaining the film conductance, k, used above.

As nearly as can be determined for the present case $D = 6.46$, so we have, by substituting this value for D and 1,962 for k' in the equation,

$$k'' = \left(\frac{1}{6.46}\right)^{0.2} \times 1{,}962$$

$$= \frac{1{,}962}{1.452} = 1{,}351$$

This is, of course, only a rough approximation, but since film conductance varies so widely with operating conditions, it is impossible to obtain an accurate value for such a complex case. However, since the temperature drop across this film is relatively small, as compared with the total temperature difference between the tubes and the barrel of the boiler, the value of k'' which we have arrived at will be sufficiently accurate for our use.

If then we substitute in the heat flow equation,

$$\Delta t_f = \frac{Q}{Azk''},$$

in which Δt_f equals the drop of temperature across the steam and water film on the tubes, we have, since $\frac{Q}{Az}$ is equal to 9,590, as before,

$$\Delta t_f = \frac{9{,}590}{1{,}351} = 7.1^\circ \text{ F.}$$

The Temperature Difference Between the Tubes and the Barrel of the Boiler

We will now sum up the temperature differences along the line of heat flow from the hot gases in the tubes, as determined above, in order that we may obtain the temperature difference between the entire tube system, which we will call T_T, and the outer sheet, T_B. From this quantity we will be able to compute the relative change in length of the tubes as compared with the barrel of the boiler.

One-half the temperature drop through the tubes, $\frac{1}{2}\Delta t_t$	2° F.
The temperature drop across the scale deposit, Δt_s	50° F.
The temperature drop across the steam and water film, Δt_f	7° F.
The temperature drop across the still water film at the outside sheet	3° F.
One-half the temperature drop across the outside sheet	5° F.
Therefore, $(T_T - T_B)$ is equal to	67° F.

The reason for using one-half of the first and last items is that the temperature at the central position of the cross-section of a metallic member represents the effective temperature of the member as far as the computation of linear thermal expansion is concerned. The value of 5° F. used for $(T_T - T_B)$ has been taken from the temperature summation given on Page 16. It represented, in that case, one-half of the temperature drop across the wrapper sheet. Since the outside conditions governing the rate of heat loss are practically the same over the entire length of the boiler, the temperature drop across the outside sheet may also be considered to be the same.

The Total Uncompensated Expansion Within the Boiler

The equation which represents the differential expansion of the tubes with reference to the barrel of the boiler is,

$$l = a\,L\,(T_T - T_B).$$

This is the same as the expansion equation used on Page 24, except that different values of temperatures are indicated. The coefficient of expansion, a, will be the same as before, but the length, L, will now be 21.5 feet (258″), or the length of the tubes. Substituting the numerical values for the symbols gives, for the differential expansion,

$$l = .000007 \times 258 \times 67 = 0.12''$$

In a similar manner we may compute the differential expansion along the length of the combustion chamber resulting from the same temperature difference between sheets as that which was found to exist at the firebox section (Page 16). The length of the combustion chamber of this locomotive is 51″, so,

$$l = .000007 \times 51 \times 279 = 0.10''$$

To obtain the total uncompensated linear expansion of the inner parts of the boiler, with reference to the outer structure, we have only to sum up the separately computed values for the firebox, the tubes, and the combustion chamber:

Relative Expansion of:

Firebox	(2 × 0.117)	0.23″
Tubes		0.12″
Combustion Chamber		0.10″
Total		0.45″

From the manner in which a locomotive boiler is constructed it is obvious that an expansion of this magnitude, over and above the expansion of the boiler as a whole, cannot take place freely in a lengthwise direction due to the restraining effect of the anchorages. It must, therefore, be largely distributed over the flues, flue sheets, and firebox plates in the form of bends, folds, and undulations. We cannot expect any material amount of relief of the forces involved through the stretching of the outer structure, since the latter is much superior in mechanical strength. In the discussion which follows later on, the distribution of the differential expansion will be considered in greater detail.

If we wish to find out how much the entire boiler structure increases in length when it is fired up to working pressure, we may do so by means of the same expansion equation used before. For instance, let us assume that the average temperature of the boiler shell, from end to end, is 380° F., the same as the temperature of the wrapper sheet (Page 17). In doing this we are supposing that the temperature of the smoke box does not differ materially from that of the wrapper and barrel. It is true that the gases in the smoke box are hotter than 380° F., but, since there is no lagging on the outer surfaces, the heat losses from radiation and convection are high. This tends to produce a temperature, in this portion of the boiler shell, somewhere near the value assumed. Taking 50° F. as the outdoor temperature, as before, we will have a resulting net temperature rise of 330° F. Now the length of the boiler, obtained by adding the dimensions taken from back to front through rear water space, firebox, combustion chamber, tubes, and smoke box, is 46 feet (552″). Substituting these new values in the expansion equation gives,

$$l = .000007 \times 552 \times 330 = 1.27''$$

as the overall expansion of the boiler due to its being brought up to operating temperature.

Similarly, the entire linear expansion (compensated and uncompensated) for the firebox, combustion chamber, and tubes would be,

$$l = .000007\,[(120 \times 609) + (51 \times 609) + (258 \times 397)] = 1.45'',$$

of which 1.00″ is compensated by expansion of the wrapper sheet and barrel (not including the smoke box), and the remaining 0.45″ is the total uncompensated expansion obtained on the previous page.

The Distribution and Mechanical Effects of Firebox and Tube Expansion in a Locomotive Boiler.

If the computed temperature differences between the inner and outer parts of the boiler exist, then the various relative expansions, which have been enumerated, exist also, even though they may not be fully evident at the locations which we are about to study.

The expansion diagram shown on Page 25 represents the manner in which the relative expansion of the side sheets of a firebox would take place if the sheets were unrestrained in their movements. Since, however, as already explained, the staybolts exert forces which tend to bend the sheet, a warping and waving results which takes up and redistributes a part of this expansion so that the actual linear displacement of the fire sheet with respect to the wrapper sheet is frequently altered in addition to being materially diminished. Furthermore, the thermal expansion of the boiler tubes with respect to the barrel of the boiler causes the tubes to exert a backward thrust on the back flue sheet and thus tends to crowd the entire firebox toward the rear end of the boiler. Although this backward movement is not very large, it nevertheless causes a further deviation from that which would represent the natural free movement of the fire sheet due to thermal expansion.

The latter influence, as well as other characteristics of firebox expansion, may be observed by studying the solid line curves, which we will call "F", in the accompanying Figures 10, 11, 12, and 13, in which are plotted, from experimental data, the movements of the staybolts and fire sheet with reference to the wrapper sheet at four different places in the firebox of a locomotive boiler equipped with a complete installation of flexible staybolts. The positions selected for illustrative purposes are those of the front, middle, and rear bolts of the top row of stays in the firebox side sheet and that of one additional bolt located practically at the center of the side sheet.

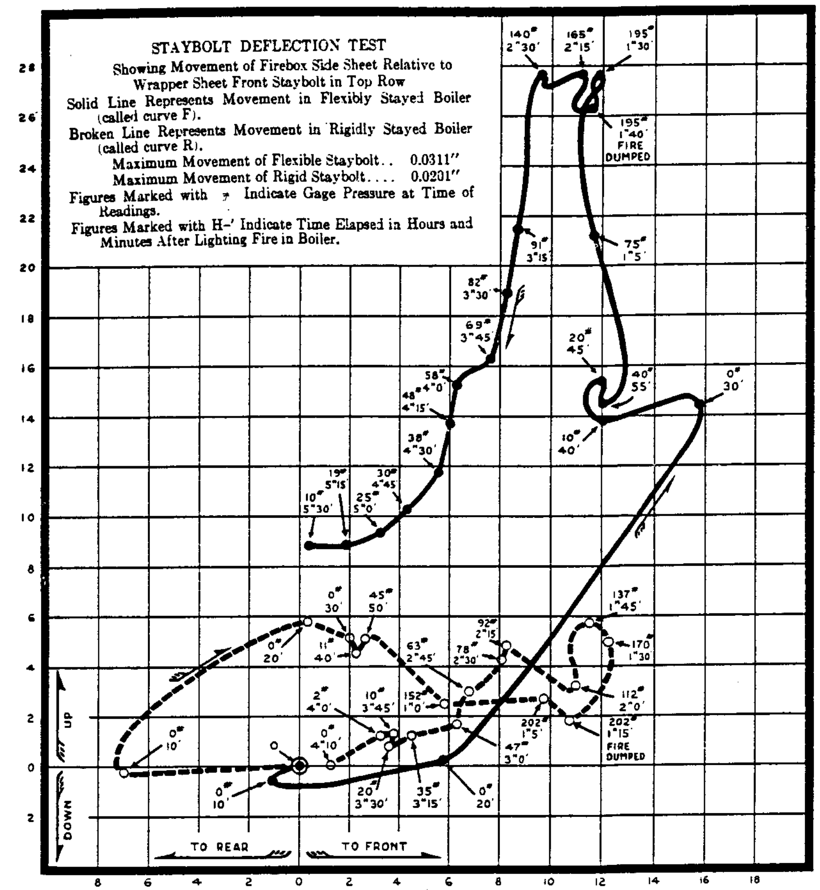
Accompanying each of the four solid line curves (F), is another curve plotted as a broken line, which we will call "R". The latter curves represent the staybolt and sheet movements for the same positions in a rigidly stayed firebox of a locomotive which is otherwise identical with the one used in obtaining the solid line curves, "F". These locomotives are not of the same type as that which was used in computing expansions, the main difference being that they do not have combustion chambers. However, we would not expect this difference to materially change the amount and effect of the expansion. The operating conditions, for the two locomotives, were made as nearly alike as possible during all test runs in order that comparisions might be made between the sheet movements of flexibly stayed and rigidly stayed boilers. It may be noted that, in general, the freedom of movement of the sheets is materially restricted by the use of rigid staybolts.

In taking the data for these, and for a great many other similar curves, the movements were measured graphically by an optical system employing a pair of pivoted mirrors mounted on the wrapper sheet and actuated by an arm which passed through the wrapper sheet and terminated in the water space where it engaged the end of a staybolt which had previously been cut off and drilled centrally to receive the end of the arm.

Each curve represents the staybolt deflection, and also approximately the sheet deflection or movement, at the particular location described, taken over a complete cycle consisting of heating up, raising boiler pressure, dumping the fire, and blowing down the steam pressure. The zero of reference in the curves, that is, the point at which both the vertical and the horizontal deflections are indicated as zero, represents, in each case, the beginning of the test when the fire is started in the firebox and the forced draft is turned on. Time is measured from this instant and is indicated at various points along the curve together with the corresponding boiler pressure. At each point the upper figures, marked #, represent the steam pressure in the boiler, and the lower ones, marked H and ', indicate the time in hours and minutes measured from the starting point.

It is interesting to note that at no time is the firebox stationary with reference to the wrapper sheet. From the time that the fire is lighted until the test is over and the boiler has had an opportunity to cool down, the fire sheet and, in consequence, the staybolts also are in continual motion. Every variation in flame intensity and draft produces some change in the expansion of the sheets. Even after the fire has been dumped and the boiler has reached its original temperature, the firebox sheets are seldom found to have returned entirely to their original positions.

In Figure 10, which represents the bolt and sheet movements at the position of the front staybolt in the top row, we observe, in curve "F", that, instead of a forward movement as great as the vertical movement, a relation which might be expected in this location if free thermal expansion could take place, the forward movement is only about one-half

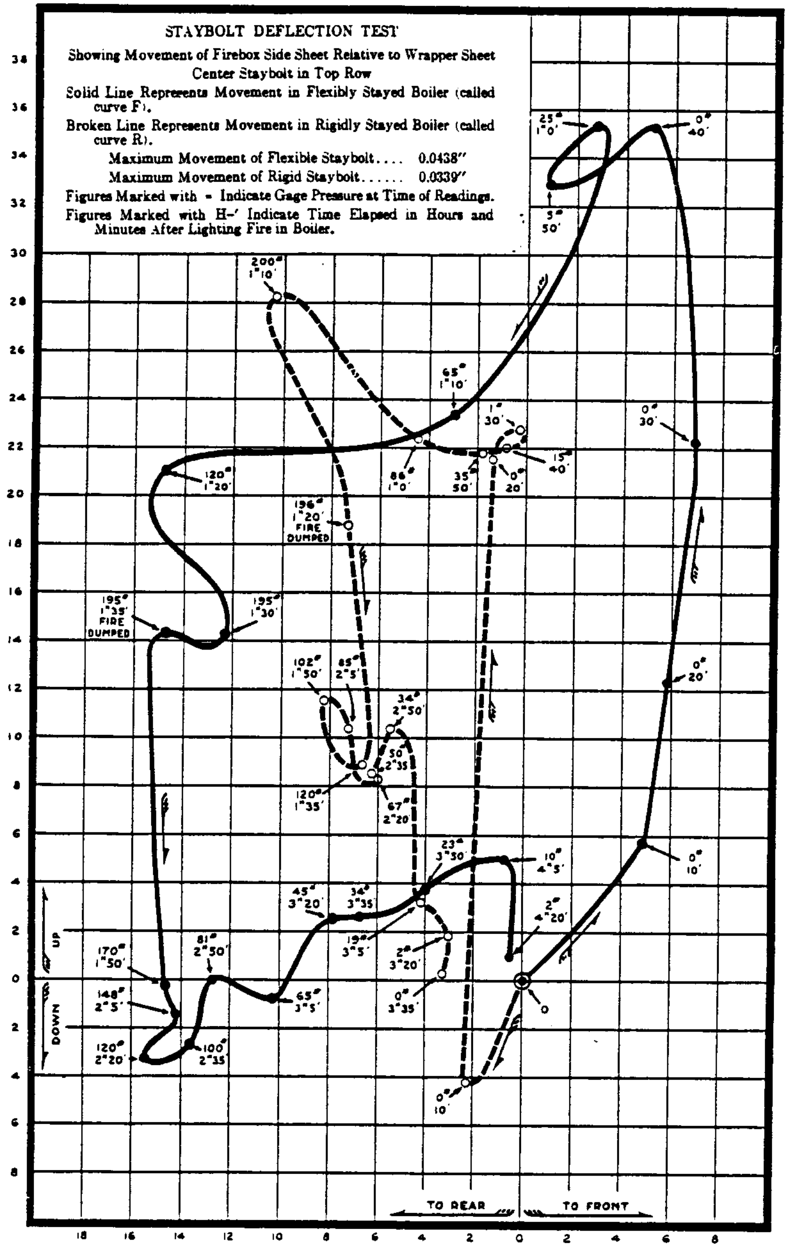


MOVEMENT, BOTH VERTICAL AND HORIZONTAL, IN THOUSANDTHS OF AN INCH

FIGURE 10

of that of the vertical movement. This deviation from the natural movement of expansion of the sheet is undoubtedly due in part, as mentioned above, to tube expansion. It may also be due in some degree to the irregular heating of the sheet itself.

At the top row central position (Figure 11, Curve F) we would expect to find essentially a vertical movement of the staybolt and of the fire sheet with reference to the wrapper sheet, but here again the



MOVEMENT, BOTH VERTICAL AND HORIZONTAL, IN THOUSANDTHS OF AN INCH

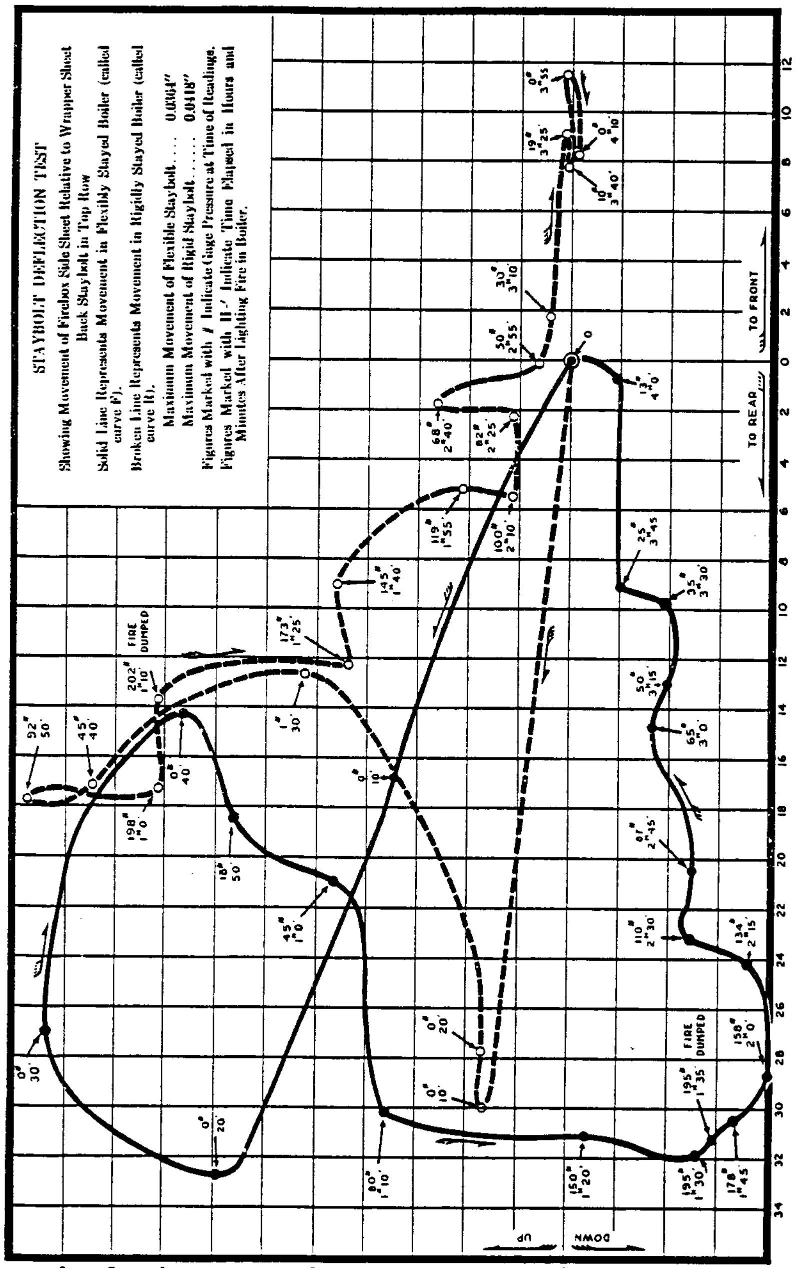
FIGURE 11

effect of the backward thrust due to the expanding tubes and flues seems to be in evidence. As soon as the fire is started and the flames begin to heat the fire sheet a rapid expansion upward and slightly forward occurs, but as the fire gets hotter so that the tubes and flues are strongly heated the increased backward thrust introduced by the expansion of the tubes probably has considerable influence in producing the movement to the rear which is so plainly seen in the curve.

In this figure the "R" curve, which represents the bolt and sheet movements of the rigidly stayed boiler, seems to show slight evidence of this backward thrust but inasmuch as the "R" curves do not, in general, justify any such interpretation we must attribute the movement, in this particular case, to some extraneous cause. The conclusion is that in this type of boiler the importance of tube expansion, insofar as concerns its ability to force the side and crown sheets backward, is confined to flexibly stayed fireboxes in which the opportunity for adjustment is greater. Since, however, the expansion of the tubes is as great in the rigidly stayed boilers as it is in those which are flexibly stayed, the greater rigidity of the firebox of the former type of boiler may introduce greater sagging of the flues, and a greater buckling action near the flanges of the flue sheets than that which would ordinarily exist in boilers which are fully equipped with flexible staybolts.

The same theory is given weight by the movements of the back staybolt of the row, (Figure 12, Curve F) and, for that matter, by the movements at a considerable number of other positions, in the flexibly stayed firebox, for which curves might be included. The back staybolt position of the top row, being located at approximately equal distances above the mud ring and to the rear of the center of the firebox, might be expected, in general, to expand upward as much as backward but from the start the backward movement is seen to predominate. There is one interval of about fifteen minutes, just prior to the time at which steam pressure began to develop, which is characterized by a pronounced decrease in the backward displacement, but from then until the fire was dumped it steadily increased even though the vertical component dropped back to a position lower than that of the starting point.

One more illustration of the movements produced by the difference in the expansion of the fire and wrapper sheets of both flexibly and rigidly stayed boilers is given in curves "F" and "R" of Figure 13. These curves are included mainly because they give the deflections at practically the central position of the side sheet. The movement, represented by the "R" curve, being essentially an up and down movement, is such as we might call a typical theoretical movement for this region if the



MOVEMENT, BOTH VERTICAL AND HORIZONTAL, IN THOUSANDTHS OF AN INCH

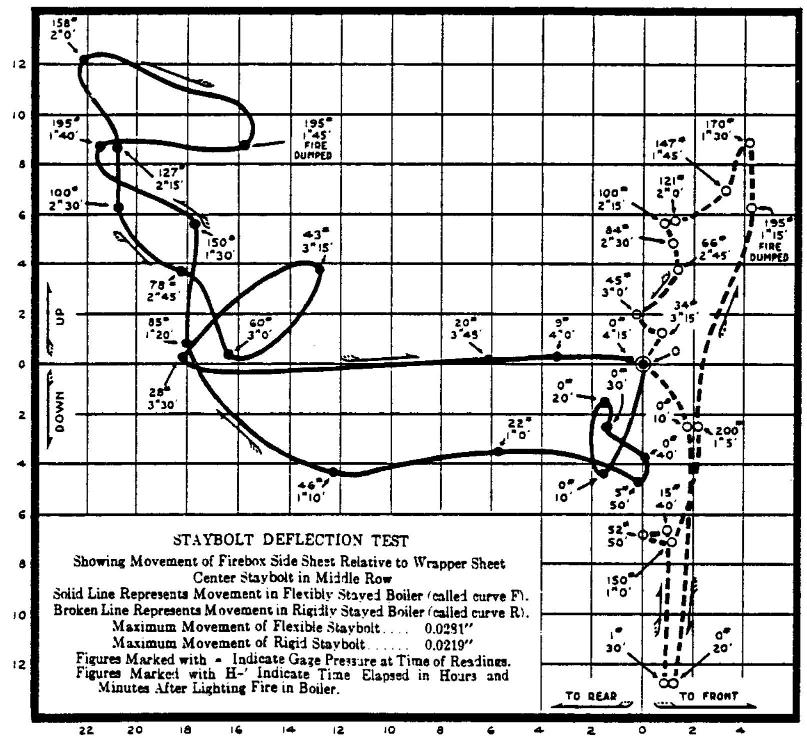
FIGURE 12

sheet were assumed to be uniformly heated and if longitudinal thrusts or end effects were eliminated. In this case the elimination of the effect of such thrusts has been accomplished, or largely accomplished, by rigid staying. The freer movement of the parts in the flexibly stayed boiler is quite evident in curve "F" where also the effect of the back thrust of the flue sheet is easily seen.

In connection with the experimental work which furnished these expansion curves the backward movement of the back flue sheet, which seems to have affected the fire sheet movements, was measured by means of a specially designed gage with a micrometer indicator and was found in some cases to be as great as 0.02″ at a position near the upper level of the throat sheet not far from the outer circumference of the flue sheet where it joins the firebox. Since this is relatively near to the mud ring where the structure is fairly rigid, it would be reasonable to expect considerably larger movements of the flue sheet at positions higher up and nearer to the middle. The actual backward movement, even though very small, is significant since the action of firebox expansion alone would result in a forward movement.

The Flannery Bolt Company has conducted a considerable amount of experimental research along the lines of firebox, staybolt, and boiler tube movements, and the results of these investigations indicate quite clearly that there are so many variables which effect the expansion of the various metal parts that it is impossible to duplicate the data obtained during a particular run no matter how much care is taken to produce the same operating conditions. Some of the most important of these variables are: the kind of fuel used, the manner of firing, the thickness and distribution of the fuel bed, the nature and amount of the draft, the temperature and rate of intake of the feed-water, and the temperature of the outside air. The expansion movements, such as those illustrated by the eight curves of Figures 10, 11, 12, and 13, which are shown and discussed here, are so dependent upon these variables that the mere opening of the fire door for a few seconds brings about a readjustment of firesheets, staybolts, flue sheets, and tubes. From this it can easily be understood why it is that all tests show that the parts of the boiler, which are exposed to the flames and gaseous products of combustion, are subject to continual and varied movements.

The method of conducting the experiments which furnished all of our data on deflections of staybolts and fire sheets gives very accurately the actual movements of the ends of the staybolts where they were cut off and where the arm of the indicating instrument was connected, but these deflections, though highly accurate in themselves, do not necessarily represent, to the same degree of accuracy, the vertical and horizontal



MOVEMENT, BOTH VERTICAL AND HORIZONTAL, IN THOUSANDTHS OF AN INCH·

FIGURE 13

movements of the sheet at the point of attachment of the staybolt. The reason for this is that a tilting movement of the sheet at this particular point without vertical or horizontal shift would cause a certain amount of deflection of the bolt at the cut off end. The plotted curves can therefore, in general, be considered as representing only approximately the movements of the sheet.

It is essential, therefore, in view of the above facts, that, in drawing conclusions from the data, we refrain from undertaking to interpret too specifically the movements which are obtained from a single experimental run, and that we confine ourselves to characteristics which are common

to a large number of tests. Acting on this basis, after a careful study of all of our experimental data, we are justified in making the following summary.

FIRST: There is considerable buckling of the steel in the vicinity of the junction of the back flue sheet with the fire sheet, or with the combustion chamber sheet, as the case may be.

SECOND: A fairly large portion of the thermal expansion of the fire sheet is taken up by a warping or waving deformation of the sheet between staybolts.

THIRD: This bending of the fire sheets between staybolts is much greater in rigidly stayed than in flexibly stayed boilers.

FOURTH: The movement of the fire sheet with reference to the wrapper sheet at any point is, in general, considerably greater in a flexibly stayed boiler than in one of the same type rigidly stayed.

FIFTH: The linear thermal expansion of the flues and tubes produces, in flexibly stayed boilers, a noticeable backward thrust of the upper portion of the firebox.

Research Laboratory
Flannery Bolt Company
Bridgeville, Pennsylvania
October 15, 1940