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REVIEW OF COMBUSTION PROCESS ON OIL BURNING STEAM LOCOMOTIVES AND PROPOSED STANDARDIZATION OF DESIGN

Belongs to
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FOREWORD AND ACKNOWLEDGEMENTS

Early in 1949, the Canadian Pacific Railway initiated a study to investigate the combustion process of oil-burning steam locomotives, the object being to determine the necessary requisites for efficient operation so the combustion arrangement could be standardized. The National Research Council was asked to participate in this study and much of the technical data quoted herein was developed by that organization.

This report is an overall summary of the various studies carried out and provides sufficient information to substantiate the recommendations made. In view of the present dieselization program, the test series were of necessity carried out on a modest scale. In addition, the recommended changes in construction are aimed at reducing only the major sources of fuel waste.

In addition to the assistance rendered by the National Research Council, valuable information was provided by the Imperial Oil Company, the Battelle Memorial Institute, the Southern Pacific Company and many other railways and organizations.

Full cooperation at all times was rendered by the staffs of the Chief of Motive Power and Rolling Stock and the Superintendent of Motive Power and Car Department, Prairie & Pacific Regions.

SUMMARY

The results of the combustion investigation of oil-burning steam locomotives have been so encouraging that it is proposed to continue the development and application study by considering the following features:

1. An air inlet system utilizing a sliding plate damper and low-loss entry in place of the present style using butterfly-type dampers.

2. The modification of the firebox geometry of certain classes in accordance with the requirements listed on page 12 of this report.

3. A "closed" type fuel oil heater in place of the present "open" style, designed on the exit heating principle whereby the fuel oil is heated as it leaves the oil tank.

4. The application of an automatic steam regulator to control the oil temperature.

5. The enlargement of the oil supply line between engine and tender to 2 and $2\frac{1}{2}$-inch piping in place of the present $1\frac{1}{4}$ and $1\frac{1}{2}$-inch size, insulated with sectional lagging.

6. The automatic regulation of the atomizing steam pressure by a mechanical linkage to the oil regulating valve.

7. The installation of a pressure gauge on the atomizer line near the burner.

8. An automatic control valve in the blower line to prevent unnecessary use of the blower.

9. An exhaust gas analyzer and an air fuel ratio indicator to assist the fireman in operating the firing valves with greater efficiency.

10. A rapid-response steam gauge to indicate whether or not equilibrium is being maintained between boiler heat input and output.

11. A light beside the smokestack to illuminate the exhaust at night and facilitate adjustment of the air-fuel ratio.

12. The introduction of the sand ahead of the arch for cleaning the tubes and flues.

An indication of the range of potential savings possible with these features is shown graphically in figure 10, page 40. The successful application of all features would reduce fuel oil consumption by an estimated 14 percent. Based on the oil consumption per locomotive in 1950, this saving would amount to $2800 per locomotive per year, or $740,000 annually for the 264 oil-burning locomotives in service on January 1, 1951. In addition, it is estimated that $69,000 per year could be saved in firebox maintenance costs by the proposed changes in the firebox and air entry design and the method of introducing sand.

It is recommended that the following applications involving a total of six oil-burning locomotives be made at the earliest opportunity:

1. To a T-1 class locomotive:

    a. A sliding-type damper.
    b. A cam and level operated atomizer control valve linked to the oil regulating valve. This application will require a steam pressure reducing valve and an atomizer pressure gauge.
    c. An enlarged oil supply line.
    d. An exit oil heater and automatic temperature regulator.
    e. A light at the smokestack to illuminate the exhaust at night.

2. To an H-1cd class locomotive:

    a. A sliding-type damper.
    b. A modified sanding arrangement which will inject sand through the combustion chamber for cleaning the tubes and flues.

3. To a second T-1, a second H-1cd and two H-1e class locomotives:

    a. A sliding-type damper.

TABLE OF CONTENTS

	Page
Foreword and Acknowledgements	(i)
Summary	(ii)
Index of Figures and Tables	(iv)
Main Report	
Air Inlet System and Firebox Design	1
Recommended Arrangement for Butterfly-type Dampers	4
Recommended Arrangement for Sliding Plate Damper	8
Comparison of Butterfly and Sliding-type Damper Arrangements	15
Von Boden Burner	17
Fuel Oil Atomization	19
Regulation of Atomizing Pressures	23
Fuel Oil Piping	26
Fuel Oil Temperature	30
Fuel Oil Heating	31
Blower	32
Air Fuel Ratio Adjustment	34
Sanding of Tubes and Flues	38
Economic Considerations	39
Further Development Work	42

INDEX OF FIGURES

Figure No.		Page
1	Engine 2382 - Firebox Air Flow Pattern Butterfly-type Damper	3
2	Engine 2843 - Firebox Air Flow Pattern Butterfly-type Damper	7
3	Standard Air Entry System Using Butterfly Dampers Air Entry System Using Sliding-type Damper	9
4	Engine 2832 - Firebox Air Flow Pattern Sliding Plate Damper	13
5	Performance Comparison of Air Inlet Systems Using Butterfly and Sliding-type Dampers	18
6	Photographs of Atomization Process	21
7	Calibration of Standard Oil Regulating Valve and Burner	25
8	Oil Valve Calibration Curves	27
9	Graph Showing Reynold's Number as a Function of Flow Rate, Pipe Diameter and Oil Temperature	29
10	Graphical Summary of Possible Economics by Redesigning Combustion Arrangement on Oil-burning Steam Locomotives	40

INDEX OF TABLES

Table No.		
1	Oil Wastage Due to Faulty Blower Operation	36
2	Oil Wastage Due to Incorrect Air Fuel Ratio Control	36

REVIEW OF COMBUSTION PROCESS ON OIL BURNING STEAM LOCOMOTIVES AND PROPOSED STANDARDIZATION OF DESIGN

Air Inlet System and Firebox Design

When oil was first adopted as a locomotive fuel by the Canadian Pacific Railway, the combustion arrangement was modelled after designs incorporating front burners. Later on it was found that more satisfactory results could be attained by placing the burner in the back position below the mudring, firing towards the front of the firebox. As more conversions from coal to oil burning were made, various modifications were tried using both front and back burners with the result that the firebox arrangements on different classes of power varied considerably.

A thorough field investigation covering each class of power was carried out to determine the most satisfactory arrangement of burner, arch and air inlet ports. As the main basis for comparison, the firebox airflow pattern for each design was studied so correlation could be made between this characteristic and the general performance of the locomotive. A fairly strong draught was induced by operating the blower as close to full boiler pressure as possible. The firedoor was closed and at various damper positions the general circulation pattern was traced by means of a wool tuft on the end of a slender rod. By this means an observer in the firebox was able to plot the relative strengths of the various currents. Although this technique did not provide definite values for the amount of turbulence induced, it did show the effectiveness of the various air ports and provided means whereby the correctness of location and size could be assessed.

It was found that the locomotives which steamed most freely and gave the most satisfactory results had an air flow pattern similar to that developed in the G-3 class firebox, illustrated in Figure 1, page 3. This design, with the burner located at the back of the firebox, admits primary air around the burner and secondary air through the floor below the crest of the brick arch. It will be noted that this arrangement induces fairly equal distribution of air throughout the firebox.

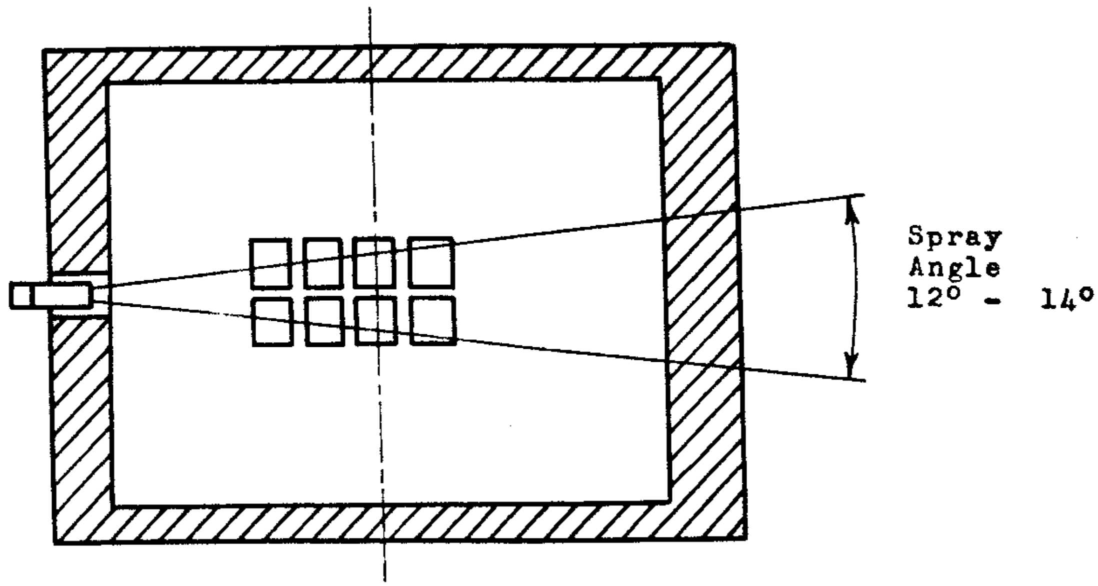
Several factors favour a firebox design patterned after that of the G-3 class power. When front burners were used on earlier locomotives, considerable trouble was experienced with staybolt failures and cracking of the faceplates. It was found that this condition was caused by unequal expansion of the firebox sheets, since the flame had an opportunity to impinge directly on the back sheet. In addition, the back sheet was subjected to sudden cooling whenever the load was decreased since it was usual practice to admit secondary air at the back of the firebox. By using the back burner and a central location of the floor air inlet, this condition was averted since the incoming air was heated by the hot arch bricks before being distributed throughout the firebox.

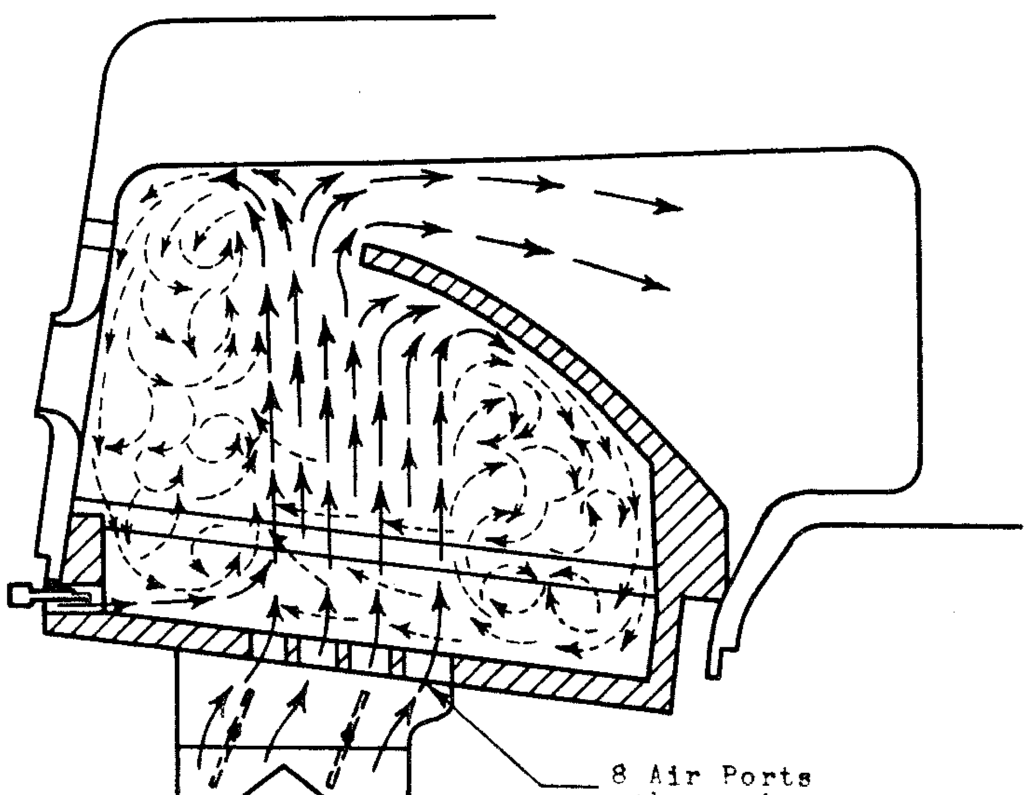
The brick arch provides protection against damage to the crown sheet as well as increasing the available time for combustion to take place. Its use not only intensifies but also affords control of firebox turbulence. The heat retaining characteristic of the brickwork has a tendency to reduce fluctuations in firebox temperature as well as to provide radiant heat for igniting the incoming oil spray.

When a front burner is installed, it must be set inside the throat sheet because of structural limits and to provide the necessary clearance for installation and servicing, the effective length of the firep

Fig. 1

ENGINE 2382
FIREBOX AIR FLOW PATTERN
BUTTERFLY TYPE DAMPER





must be reduced. This condition normally results in carbon formation at the flashwall since a combustion length of approximately eight feet is desirable when using the Von Boden type burner. In addition, the front burner position increases the length of oil piping required.

Although many railroads favour the induction of nearly all the combustion air around the burner, the results attained are not as satisfactory as those using inlet ports in the floor. With the former design the air and oil enter the firebox in parallel streams and mixing is largely dependent on firebox circulation. By using floor inlets, the incoming air meets the oil spray at right angles and thorough mixing is thereby ensured soon after the air is inducted.

Recommended Arrangement for Butterfly-type Dampers

Following the general survey covering the various classes of oil-burning locomotives, modifications were made to several engines in an effort to determine the necessary relationships between the firebox variables governing the combustion efficiency. These tests were conducted on engines equipped with the standard butterfly-style dampers and were aimed at fixing the location of the air inlet ports and brick arch so the firebox geometry could be standardized.

The relative merit of each alteration was assessed by its effect on the air flow pattern and arch brick deterioration. It was found that the arch and flashwall bricks deteriorated rapidly if the inlet was located too close to the flashwall where an intense vortex was created. Conversely, when the opening was moved back past the crest of the brick arch there was very little circulation at the front section. By locating the opening in a central position a fairly equal distribution pattern was attained throughout the firebox.

An undesirable characteristic of the butterfly-type dampers is that the air flow pattern is altered as the damper position changes, since in all but the fully open position the incoming air is deflected to either the front or the back of the bootleg depending on which direction the dampers operate. When a single floor opening is used, this may allow the incoming air to blast directly on the underside of the arch resulting in rapid brick deterioration. This effect may be reduced by using a checkerboard of air ports similar to those on the G-3 class power. Besides providing protection for the bootleg steel against the firebox heat, the grid type openings tend to straighten and distribute the incoming air. By increasing the $2\frac{1}{2}$-inch spacing of the air ports on the standard grid sections, the blast effect of the incoming air may be further minimized. This alteration may be accomplished by plugging one pair of holes in an eight-hole grid, which also has the effect of increasing the contact area between the incoming air and the oil spray.

With a single floor opening, the butterfly dampers may become fouled if large pieces of firebrick are dislodged and fall into the bootleg. The danger of this occurring is reduced by the application of a grid section.

Combustion air should be provided around the burner for cooling purposes and to ensure early ignition of the incoming oil spray. The effect of the burner opening was studied so its dimensions could be specified. It was found that the induction of too much air at the burner retarded the ignition of the oil spray as well as causing undesirable effects on the firebox airflow pattern. In addition, the air supply became difficult to control at very light loads, resulting in the induction of excess air. On the other hand, when the opening was made too small it became subject to plugging. Since the oil temperature at the burner frequently rises above the boiling point of water under light loads, vaporization and bubbling may occur at the lip of the burner. Unless combustion air is provided, the oil may contact the hot bricks around the burner forming carbon deposits.

Considering all factors, the most satisfactory results were attained by using a burner opening whose net free area was approximately 10 percent of that of the floor openings. Most of this area should be placed below the burner to ensure the combustion of stray oil droplets which might cause carbon formation if they struck the floor.

The burner should be set sufficiently clear of the floor so sand or slag deposits which may occur will not obstruct the oil spray. This is usually determined by the clearance to the back mudring. Present practices which are followed in aligning burners are to direct the centerline of the spray at a point 14 inches to 18 inches above the floor at the flashwall. The former figure should be favoured since the oil spray forms a circle approximately 24 inches in diameter in a firing length of 8 feet.

For air inlet systems utilizing butterfly-type dampers and the standard grid section, test results indicate that the most satisfactory firebox geometry is that shown in Figure 2, page 7. This design embodies the following points:

1. The Von Boden burner should be set at least eight feet from the flashwall.

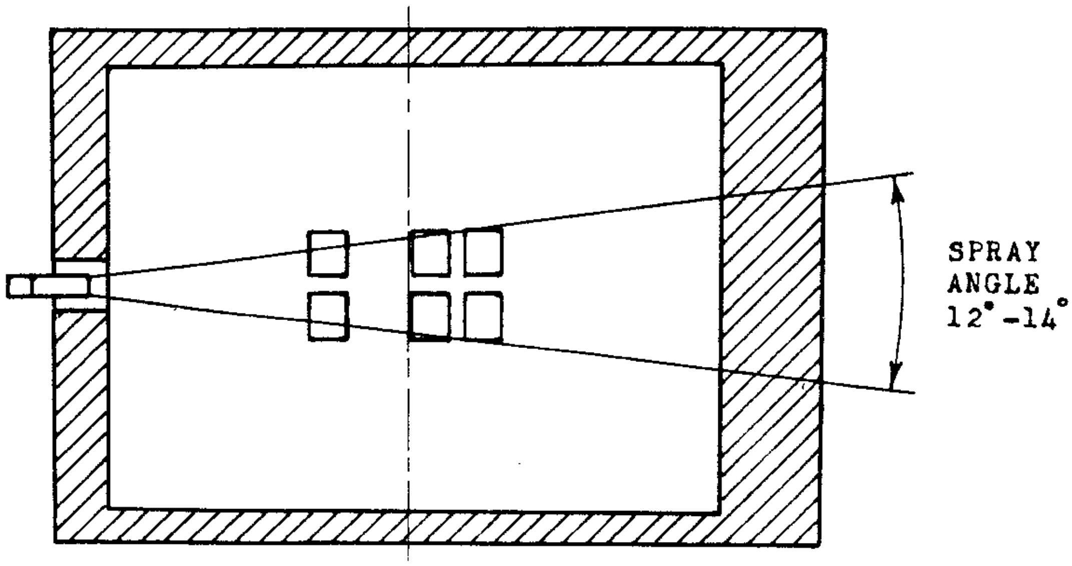
2. The bootleg should be centered between the burner and the flashwall.

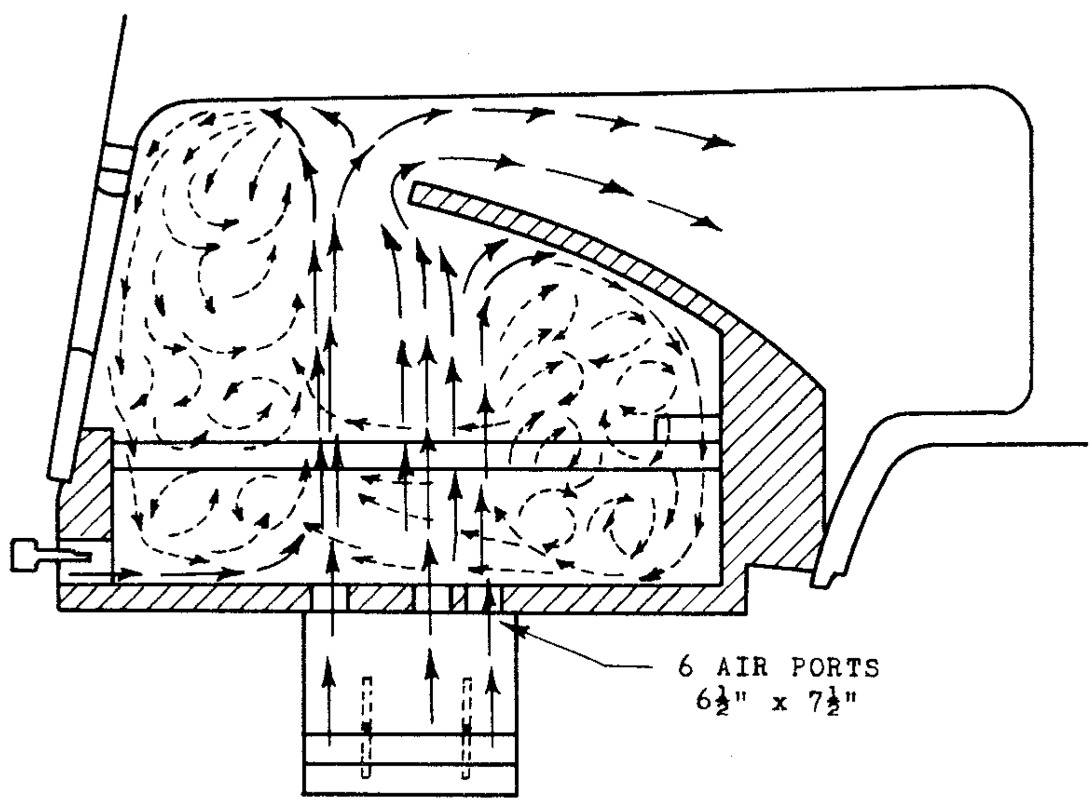
3. The crest of the brick arch should be located above the center of the bootleg.

4. The floor openings should be spaced by plugging one pair of holes in an eight-section grid, placing four holes beneath the arch and two holes behind the arch.

ENGINE 2843
FIREBOX AIR FLOW PATTERN
BUTTERFLY TYPE DAMPER

FIG. 2





5. The total area of the floor openings should be approximately 300 sq in.

6. The burner opening should approximate 10 percent of the area of the floor inlet with most of the opening below the burner.

Fuel consumption tests were carried out to determine the relative merit of this firebox arrangement and the results indicated that its application would effect definite savings, particularly when the grid-type spaced ports were used in place of a single floor opening. The general performance of locomotives of the P-2k, F-1 and F-2 classes equipped with this design has been very satisfactory.

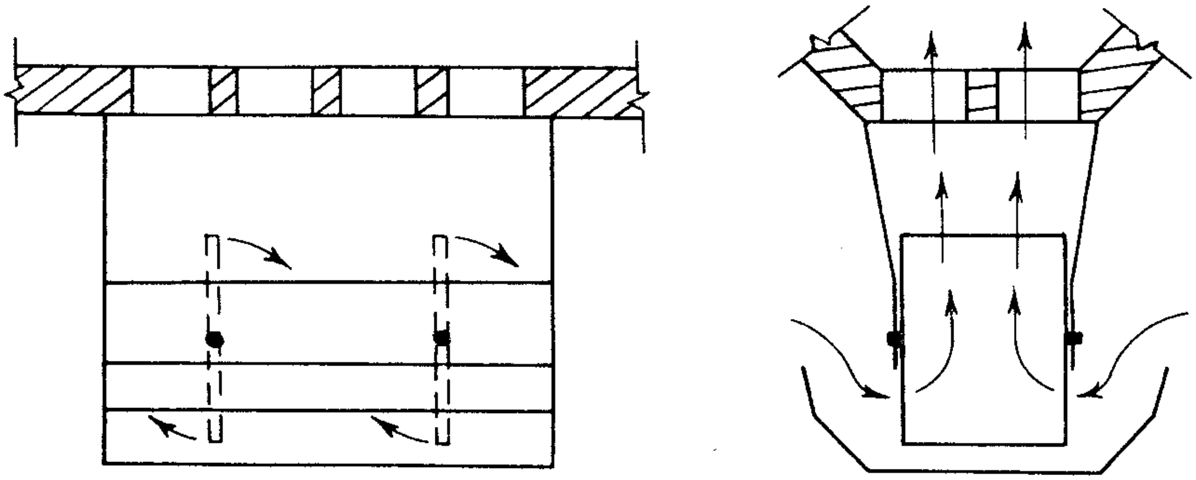
In view of the still more satisfactory results which may be attained by the use of a sliding-type damper, as pointed out in the next section, it is not recommended that any locomotives be converted in accordance with the above specifications.

Recommended Arrangement for Sliding Plate Damper

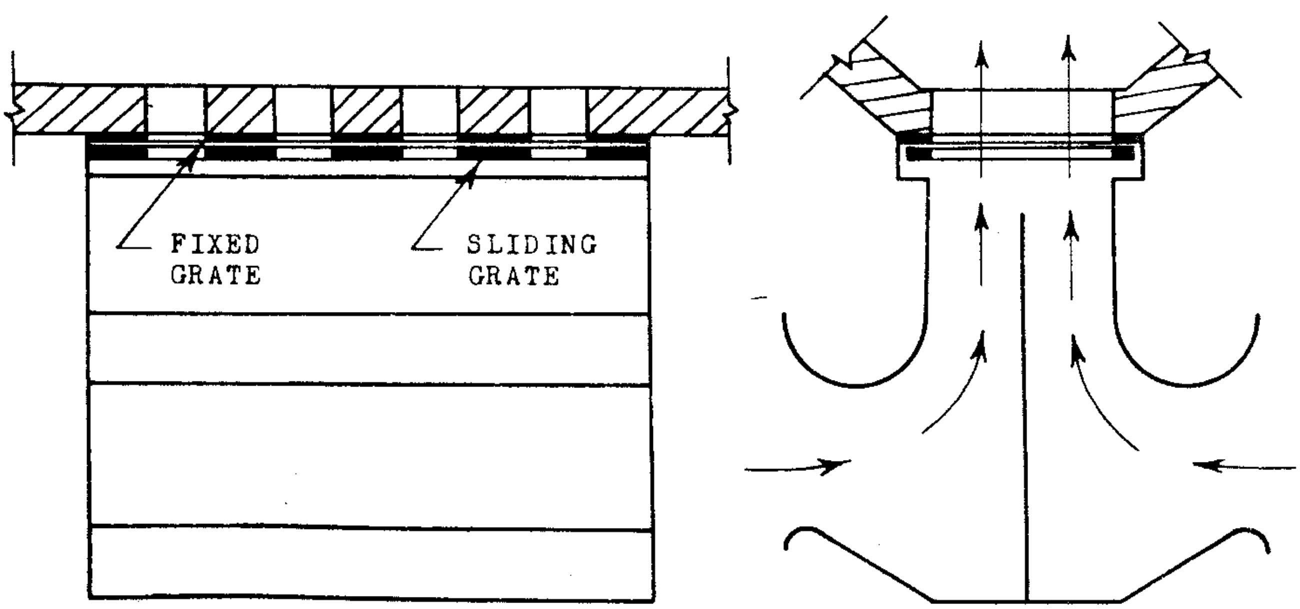
The field investigation of oil-burning steam locomotives, carried out under the supervision of National Research Council personnel, indicated that a redesign of the air inlet system comprising the bootleg and butterfly dampers should result in a substantial improvement in the combustion efficiency. This form of air inlet ducting is illustrated in Figure 3, page 9. Draught measurements showed that the ratio of firebox to smokebox draught had a minimum value of 0.6, with the average at 0.7 or higher. In other words, at least 60 percent of the smokebox draught was expended in drawing air into the firebox. A further analysis of air flow measurements showed that a major portion of the firebox draught was expended in high energy losses of the air stream as it passed through the bootleg. These losses constituted from 80 to 95 percent of the available firebox draught. It was therefore obvious that the required draughts could be considerably reduced if these losses were not so great.

STANDARD AIR ENTRY
SYSTEM USING BUTTERFLY
DAMPERS

FIG. 3



AIR ENTRY SYSTEM
USING SLIDING TYPE
DAMPER



The turbulence created by the pressure losses in the bootleg and around the dampers was largely dissipated before entering the firebox. This energy change is nearly all wasted since maximum turbulence should occur in the firebox itself where it will serve a useful purpose by intensifying circulation.

It was also shown that as the air stream passed through the standard grid openings, the jets so formed did not retain their identity and had in most cases reunited within 12 inches of the openings. In effect, the air was entering the firebox as a single large jet which normally had sufficient energy to cause impingement on the underside of the arch, causing arch brick deterioration. This observation indicated that deterioration could be appreciably reduced if the spacing of the air ports were increased so the air stream would be broken up into separate jets. This would also increase the path length of oil droplets over the air ports and would tend to improve the distribution pattern throughout the firebox.

In order to determine the necessary requirements for a more satisfactory design of air intake system, a quarter-scale model of a firebox was set up in the National Research Council laboratory. This model was used to study the effect of variations in the firebox geometry and the merit of possible modifications which could be made.

The energy losses sustained in the air entry system of an actual locomotive were verified on the quarter-scale model. Another type of damper and grid arrangement was set up so comparison could be made with the standard design. This test design is also shown in Figure 3, and consisted of a sliding-plate damper set flush with the floor of the firebox, having four openings 5 in. by 17 in. spaced 6 in. apart.

The air ports of the sliding-type damper were designed to break the air stream up into separate jets which would maintain their identity until they had passed through the oil spray region. Suitable spacing was selected to increase the path length of oil droplets over the air ports, ensuring thorough mixing. The port dimensions were also such as to minimize the impingement effect of the incoming air on the underside of the arch, indicating that the arch bricks should stand up longer.

Comparison between the standard and proposed air entry designs was made on the quarter-scale model at equal mass flows of air and at equal firebox draughts. It was found that the energy loss across the new air inlet and sliding damper did not exceed fifteen percent of the available firebox draught, whereas with the butterfly damper this loss was always greater than 60 percent. In the combustion region, where the pressure loss is a measure of the intensity of turbulence, tests with the sliding type damper showed that the firebox pressure losses sustained were four to five times as great as those obtained with the butterfly type damper. This in effect means that a comparable combustion efficiency will be attained for a lower value of firebox draught. Thus where twelve inches of firebox draught were required with the butterfly type damper, five inches should be sufficient with the sliding type damper.

The effect of varying the longitudinal position of the air ports was studied and it was found that in comparison to any other position, firebox turbulence was 10 to 12 percent higher for a grid location 14 inches from the flashwall where the ports were completely covered by the arch. The type of air flow attained by this arrangement is shown in Figure 4, page 13. A tuft exploration showed that intense turbulence occurred throughout the entire firebox volume. This was attributable to a strong under arch vortex which was most intense with the grid in the forward position and which tended to direct air towards areas where turbulence was normally fairly weak.

A study of the circulation pattern indicated that the arch should be of sufficient length to completely cover the grid openings. It will be noted that the entire flow at the back of the firebox passes upwards with very few down currents. Shortening of the arch created a vortex at the back of the firebox which caused a down flow over the firedoor. Excessive firedoor heating which occurs on many locomotives, particularly those with short fireboxes, is brought about by this condition. In addition, the shorter arch tended to reduce the intensity of the under arch vortex, which should be as strong as possible.

For air inlet systems utilizing the sliding-type plate damper, the laboratory tests indicated that the firebox geometry for most efficient operation was that shown in Figure 4 which embodies the following points.

1. The brick arch should extend for 40 percent of the firing length between the flashwall and the burner, completely covering the floor openings.

2. The floor inlet should consist of four openings, each 5 by 17 inches, spaced 6 inches apart.

3. The total area of the floor openings should be 340 sq in.

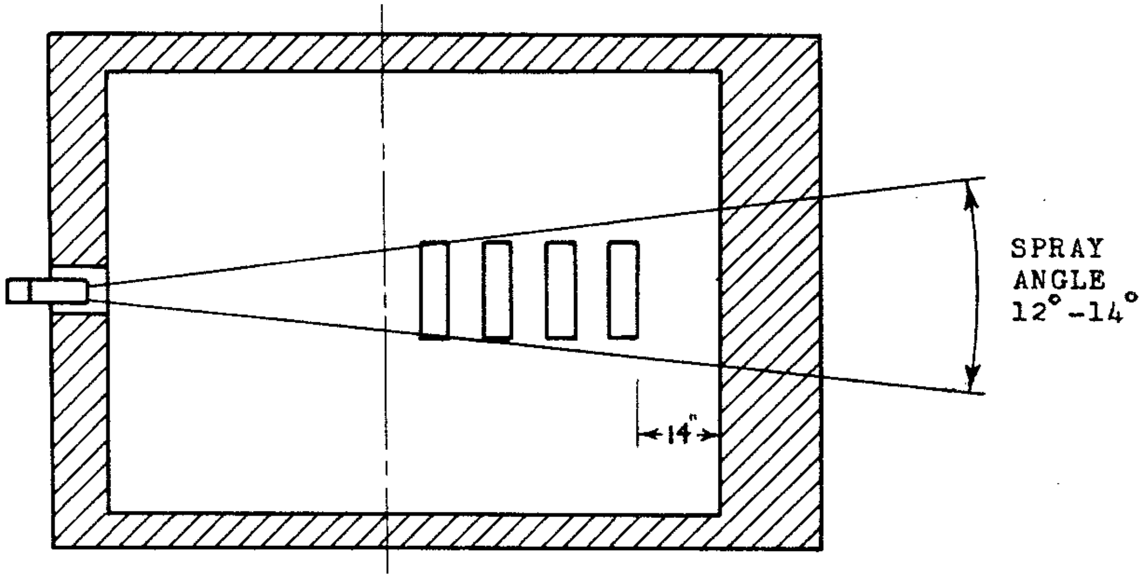
4. The forward opening should be located 14 to 16 inches from the flashwall.

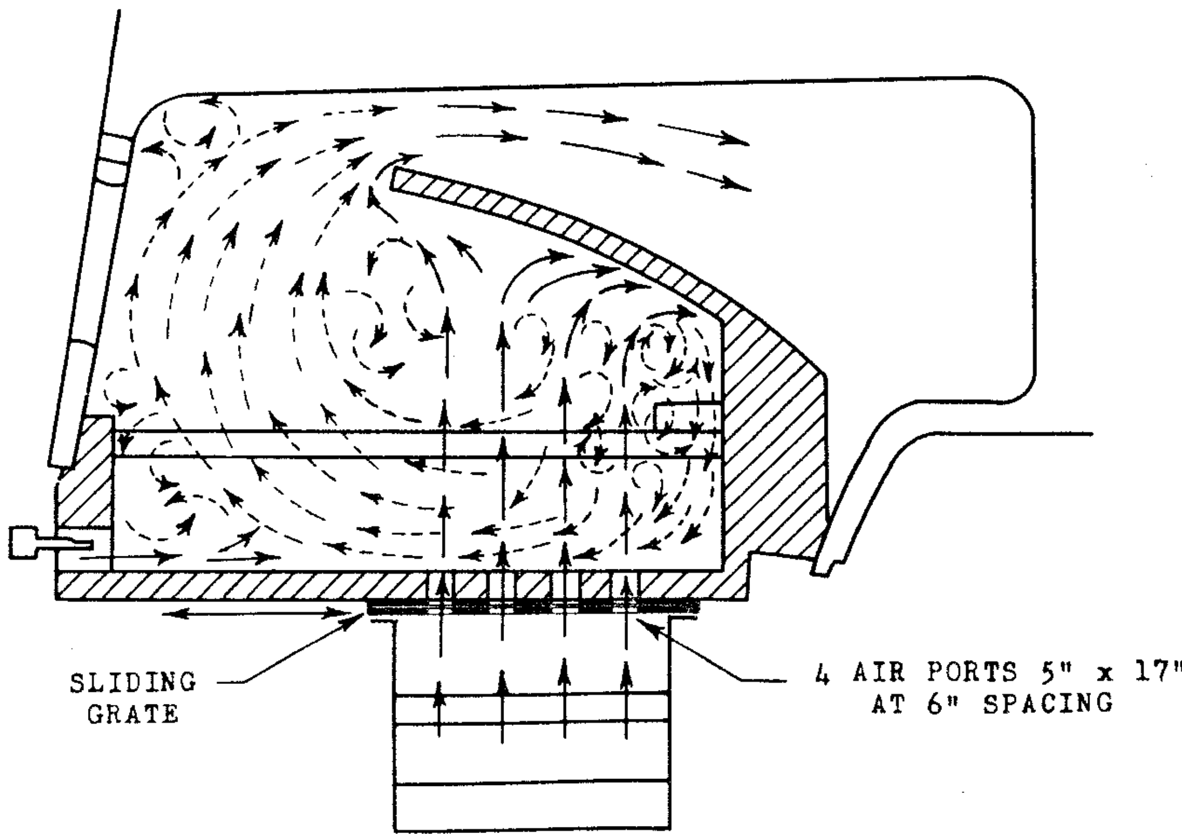
5. The sliding damper plate should close towards the rear of the firebox.

6. The burner opening should have the same dimensions as previously recommended for the butterfly-type damper.

ENGINE 2832
FIREBOX AIR FLOW PATTERN
SLIDING PLATE DAMPER

FIG. 4





A locomotive of the H-1 class was equipped with a sliding-plate damper and air inlet system in accordance with the recommendations outlined. This engine was operated in regular passenger service for approximately three months during which time its steaming qualities were entirely satisfactory. Fuel consumption figures were developed so comparison could be made with designs incorporating the butterfly-type dampers.

It was found that the application of the recommended sliding damper design had altered the appearance of the flame in the firebox from the bright shade normally associated with good combustion to a dull orange colour. This was caused by the altered circulation pattern which resulted in a flow of partially burned gases upwards past the firedoor and sandhole. This condition was not indicative of good or bad combustion, however, since observation was made halfway through the combustion process. The best indication is the colour of the exhaust gases and during the tests described it was not necessary at any time to operate the engine with more than a very light haze at the smokestack. The disadvantage of the hazy flame colour was that firemen had a tendency to use excessive atomizer pressures to brighten up the fire. Under certain conditions, this misadjustment could cause carbon formation or rapid arch brick deterioration.

Modifications were made to the spacing of the air inlet ports in an effort to clear up the hazy colour of the flame. Experimentation along this line was also desirable on an actual locomotive to corroborate the experimental results on the quarter-scale model. With the modified spacing no change was observed in the colour of the flame. In addition, it was found that undesirable alterations had occurred in the circulation pattern since spaces could be observed at the back of the firebox which were void of flame. Under this condition, the steaming qualities of the locomotive deteriorated and it was found necessary to replace the experimental design by a standard arrangement.

The road tests indicated that the most satisfactory arrangement of the sliding type damper was that complying with the recommended design developed through laboratory tests. The only apparent disadvantage was the hazy condition of the flame and the possibility of the use of excessive atomizing pressures. It would therefore be desirable to automatically regulate the atomizing pressure on locomotives equipped with this design.

Under a recent conversion program, the Northern Alberta Railways equipped three locomotives with the sliding-type damper design. The steaming quality and general performance of these locomotives has been very satisfactory. In view of the favourable results attained, this arrangement has been adopted as standard for application to future conversions.

Comparison of Butterfly and Sliding-type Damper Arrangements

As already discussed, the fuel burning capacity and efficiency of combustion may be appreciably increased by the application of sliding plate dampers. Although tests were not carried out over an extended period, the results did indicate that this design would effect savings of some 4 percent of the fuel rate based on the consumption per 1000 egtm, over and above that possible with the most satisfactory arrangement using butterfly dampers.

Other factors, in addition to possible fuel savings, favour the adoption of this design. Air flow considerations indicate that its application will minimize the blast effect of the incoming air on the underside of the arch and should therefore reduce brick deterioration. In operation, the arch had a reddish appearance in comparison to the normal bright white tint, indicating that it was remaining cool while combustion proceeded. There are some 264 oil-burning locomotives in service for which the material cost of arch brick renewals amounts to approximately $3,200 per month. Since labour costs will double this figure, it is evident that worthwhile savings might be realized.

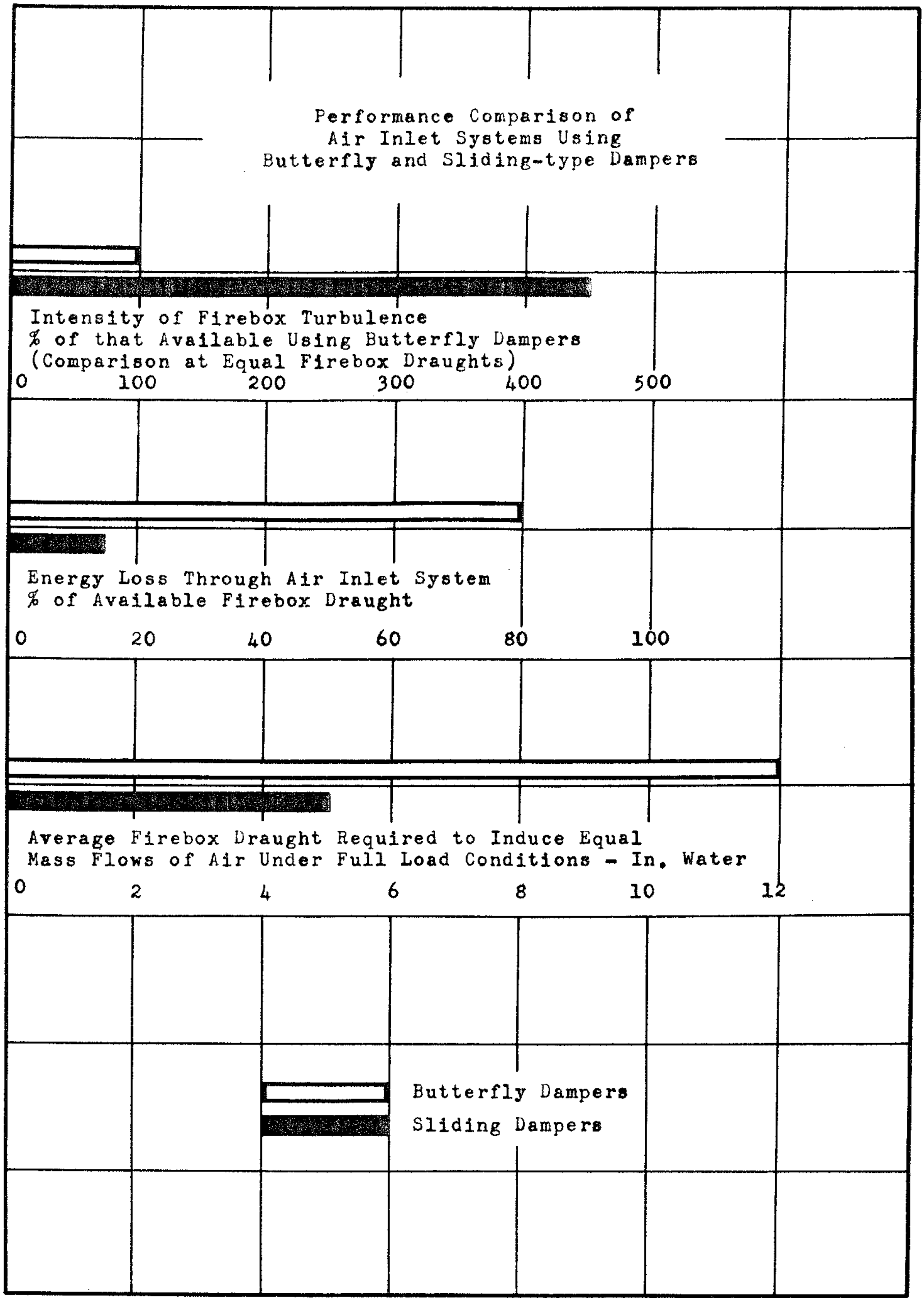
To provide operational and service clearance, the butterfly-type dampers are usually set well below the base of the firepan. This means that in the fully open position the plates swing within 3 to 4 inches of the bottom of the bootleg. During winter operation when snow and ice may pack in the bootleg, it has been found that the dampers frequently freeze in the open position, making it difficult to maintain steam pressure due to the induction of excess air. To remedy this condition, it has been found necessary to replace the bottom plate of the bootleg with a grid during the winter months. Although the open bottom overcomes the freezing problem, it sometimes allows snow to swirl into the firebox, if drifting has occurred across the right of way. This occasionally extinguishes the fire and must induce appreciable stresses in the firebox sheets. It has also been reported that difficulty in controlling the air supply has frequently been experienced in severe sidewinds.

Since the sliding-plate damper is installed flush with the floor of the firepan, it is well protected against freezing and does not necessitate the removal of the bottom plate of the bootleg as outlined. This therefore prevents snow from swirling into the firebox. The design of the accompanying bootleg is also such that a baffle may be installed to minimize the effect of strong sidewinds, as shown in the lower sketch of Figure 3.

The performance of air inlet systems using butterfly and sliding-type dampers is graphically illustrated in Figure 5, page 18. Considering turbulence factors, the function served by the sliding-type damper is that of transferring the high pressure loss in the air entry system, which is largely wasted, into the firebox where it will intensify circulation and improve combustion. High turbulence has the effect of inducing thorough mixing of the air and oil and providing a longer period for combustion to take place. This means that the available firing length is not too critical, in comparison to the butterfly damper arrangement where a combustion length of eight feet is desirable In addition, it also means that burner misalignment, which causes localized disruption in the air fuel ratio over the air ports, is not as critical. The test application of the sliding-type damper indicated that its adoption should eliminate carbon formation in the firebox, if the firing controls are properly handled.

Von-Boden Burner

In view of the possibility that a more efficient design might be adopted, the characteristics of the standard Von Boden burner were thoroughly investigated to determine its suitability for locomotive service.



Performance Comparison of
Air Inlet Systems Using
Butterfly and Sliding-type Dampers
Intensity of Firebox Turbulence
% of that Available Using Butterfly Dampers
(Comparison at Equal Firebox Draughts)
0
100
200
300
400
500
Energy Loss Through Air Inlet System
% of Available Firebox Draught
0
20
40
60
80
100
Average Firebox Draught Required to Induce Equal
Mass Flows of Air Under Full Load Conditions - In. Water
0
2
4
6
8
10
12
Butterfly Dampers
Sliding Dampers

For this purpose, a box of comparable proportions to a locomotive firebox was set up in the National Research Council laboratory. Cold tests (without flame) were carried out to determine the characteristics of the spray pattern under varying conditions of oil flow rate, oil temperature and viscosity, atomizing steam pressure and steam slot width. The spray pattern was obtained by exposing a suitable screen at different distances from the burner.

The results indicated that the standard Von Boden burner adequatel fulfills the requirements for efficient combustion in a locomotive fire box and that little benefit could be gained by going to a more expensiv burner with a finer degree of atomization. In addition, the Von Boden burner is simple and rugged and since only gravity feed is required, it has proven very dependable. The most favourable characteristic is its high capacity and ability to function satisfactorily over a wide range of fluctuating loads, which on a locomotive may vary from 40 to 750 Imp. gal of oil per hour.

Fuel Oil Atomization

The laboratory study showed that a definite relationship exists between oil flow rate and atomizing pressure required. Sufficient pressure produces a satisfactory spray consisting of a wide range of droplet sizes, resulting in good distribution of fuel throughout the firing length. For a particular fuel rate, the spray quality deteriorates rapidly below a minimum critical value of steam pressure and with any further decrease the oil gurgles irregularly from the lip of the burner. This action is quite pronounced, projecting large blobs of oil erratically throughout the firebox with obvious detrimental results on the combustion process. This phenomenon is shown in the photographs of Figure 6, page 21, where a 5 psi increase in atomizing pressure has corrected the gurgling condition.

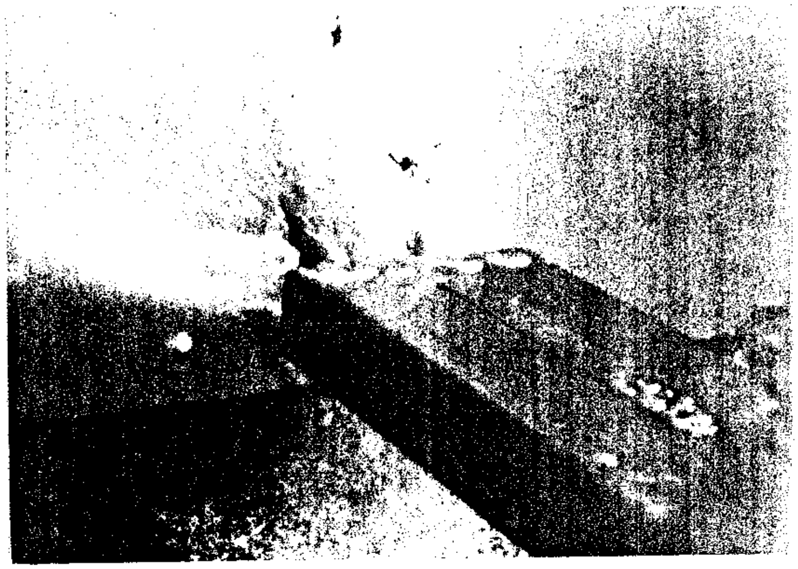
Although pressures on the low side adversely affect the atomization process, pressures in excess of those required for non-gurgling flow do not appreciably improve the quality of the spray but rather project droplets forward at higher velocities due to the excessive kinetic energy imparted. This causes no change in the conical spray angle produced, which varies between 12 and 14 degrees. Proper location and design of air inlet ports provide sufficient combustion length and turbulence that pressures greater than necessary have no beneficial effect. In fact, on locomotives equipped with short fireboxes, high atomizer pressures under certain conditions cause rapid deterioration of arch and flashwall bricks as well as excessive deposits of carbon.

With the present oil burning arrangement, the fire as observed at the backhead of the firebox is normally a bright orange colour. This colour serves as an indicator for determining the proper setting of the atomizing steam valve, although, as their experience increases, firemen can usually control this pressure by the sound of the fire and the colour of the exhaust. The colour of the fire indicates when sufficient atomizing pressure is being used, since lower pressures cause a darkening of the flame due to the gurgling effect described. As already discussed, the application of the sliding-type damper causes a darkening of the flame colour and in an effort to clear up the fire there is a tendency to use excessive atomizing pressures, particularly by inexperienced firemen.

Firing would be greatly facilitated if the firemen were provided with a pressure gauge on the atomizing steam line, located adjacent to

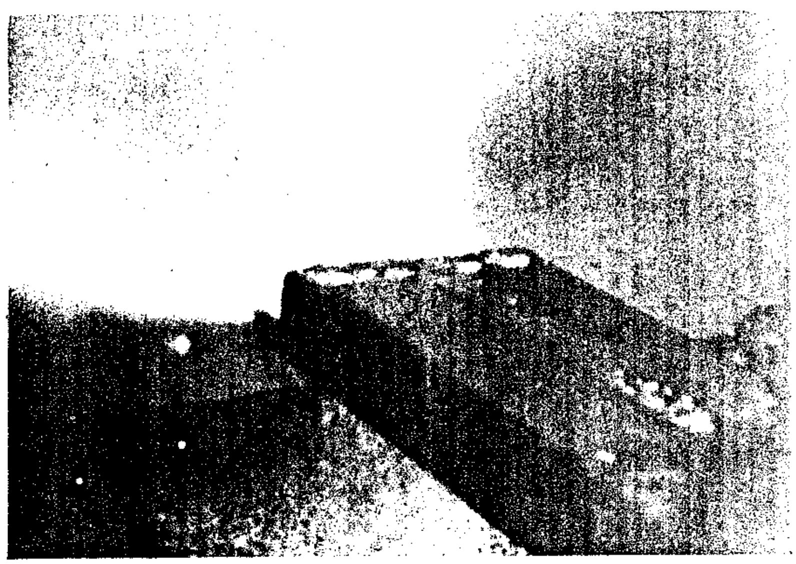
OIL SPRAY ISSUING
FROM VON BODEN BURNER

GURGLING CONDITION



ATOMIZER
PRESSURE
20 PSIA

NON-GURGLING CONDITION



ATOMIZER
PRESSURE
25 PSIA

OIL FLOW - 150 IMP. GAL/HR.
1/32" STEAM SLOT BURNER
OIL TEMPERATURE - 170 F
MONTREAL "BUNKER C" OIL

the burner. At present their only indication of the atomizing pressure is the degree of opening of the control valve. Adjustment may therefore become difficult if slack occurs due to wear.

Examination of test data for locomotives equipped with the present standard arrangement shows that no definite relationship is maintained between oil flow rates and atomizing pressures used. The general tendency is to carry pressures well above those required for proper spray production, often being two to three times what is actually required and consequently automatic control would be in order. The possibility of maladjustment is accentuated with the sliding damper arrangement and if consideration is to be given to the use of this design it is desirable that the atomizing pressure be automatically regulated.

An idea of the quantity of steam wasted when excessive atomizing pressures are used may be gained from a consideration of flow requirements. The maximum oil rate encountered on road tests was of the order of 750 Imp gal per hour. For proper spray production using the standar Von Boden burner with a 1/16 inch steam slot, an atomizing pressure of approximately 43 psig is required, which results in a steam flow rate of 675 lb per hour. With improper regulation, this pressure may be more than doubled, resulting in an additional steam flow of at least 500 lb per hour. Rough calculations show that approximately 60 lb of oil her hour are required to supply this loss, or in other words, 0.8 percent of the fuel rate. The actual cost of improper regulation is almost impossible to estimate, but it must be appreciable, due to poor combustion when pressures are too low, and to steam wastage when pressures are too high.

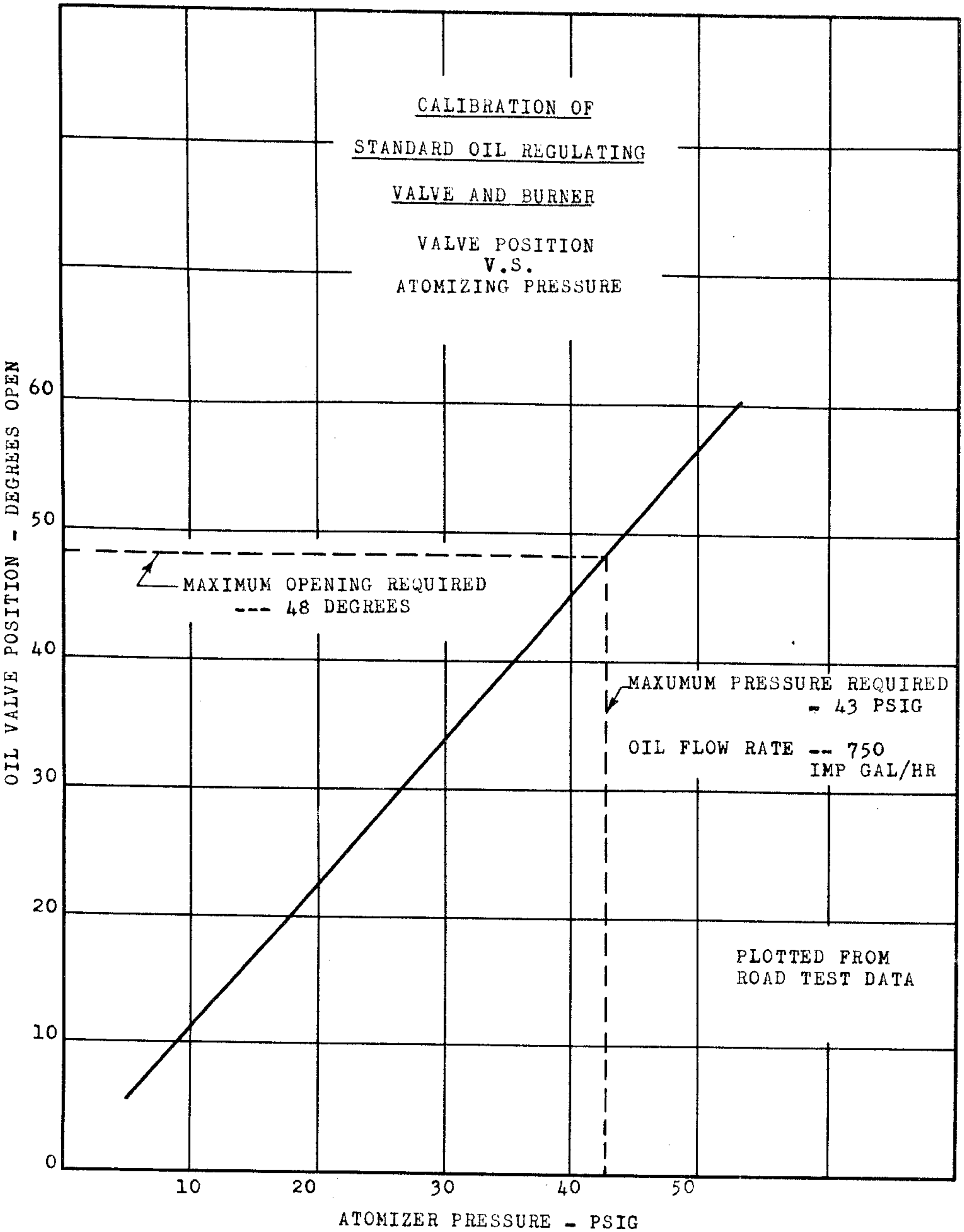
Regulation of Atomizing Pressures

The atomizing steam flow is governed directly by the pressure at the burner, since the steam passage is of constant area. Therefore, the logical method of regulation is to control the pressure by the volume of oil flowing. This is not practical, however, since accurate measurement of oil flow using simple commercial instruments is impossible to attain. Consequently, the simplest and most dependable method of control is a mechanical linkage between the oil regulating valve and a pressure control valve on the atomizing line. This requires that for a certain pressure head, the oil flow be directly dependent on oil valve position. This may be attained through restriction of viscosity variations by fuel oil temperature control and reduction of pressure loss by using a large enough oil line, these moves being covered in subsequent sections.

In an improperly designed fuel system viscosity changes may have adverse effects on the fuel flow, causing excessive pressure loss in the piping or fluctuations in the rate of flow for a fixed setting of the oil regulating valve. In the atomization or spray formation process itself, the amount of steam required is essentially dependent on the volume of oil flow as long as viscosity variations are kept within a reasonable range. This is true since the main function of the steam is to supply kinetic energy to the oil droplets in order that they will leave the lip of the burner at sufficiently high velocity. It is characteristic of the Von Boden type burner that this process results in a wide range of droplet sizes, rather than in a predominance of minute particles. This is quite satisfactory for locomotive practice due to the high firebox turbulence and ample combustion length available.

Test data for oil flow versus regulating valve position and oil flow versus required atomizing pressure have been correlated into the graph of Figure 7, page 25, showing required atomizing pressure versus oil valve position for a properly designed fuel system. The oil regulating valve consists of two discs having suitable passage ports, one fixed and one operated by a spindle which rotates through 48 degrees from zero to full flow. The mechanical linkage for atomizer control can be operated directly from this spindle. The spindle would be modified to incorporate a lug which picks up the arm to the atomizer control valve when the oil passage begins to open. A suitable cam and lever operated pressure regulating valve has been selected for this purpose. Again referring to Figure 7, it will be noted that the atomizing pressure will vary linearly from 0 to 43 psig. This is effected by the valve as its control lever operates through the same sector as the spindle of the oil regulating valve.

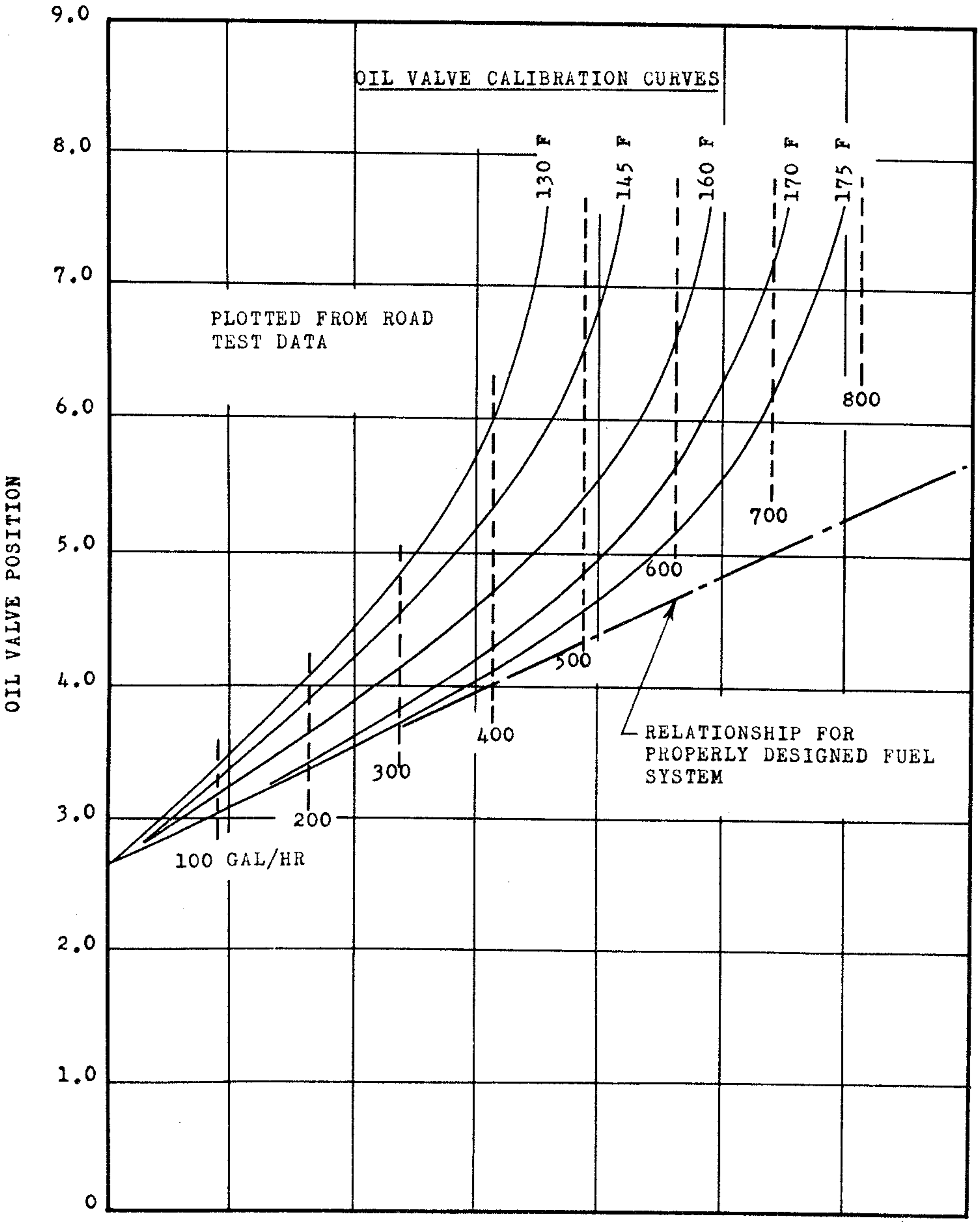
In the application of the atomizing control valve, it may be neces sary to develop some form of compensation for the change in level of the fuel oil. Tests indicate that the rate of flow is governed by the pressure head exerted by the tank of oil and that very little inspirating action is produced on the oil by the atomizing steam as it issues from the burner. One move would be to pressurize the fuel tank at a suitable pressure such as 10 psig which would cause an increase of some 23 feet in the pressure head, making any changes in the fuel level insignificant. Without pressurization, the oil regulating valve will open somewhat further when the tank level is low, in turn increasing the movement on the pressure control valve. However, this added movement, or in other words, increase in atomizing pressure, may not be appreciable, due to the rapidly increasing area of the passage


CALIBRATION OF
STANDARD OIL REGULATING
VALVE AND BURNER
VALVE POSITION
V.S.
ATOMIZING PRESSURE
MAXIMUM OPENING REQUIRED
--- 48 DEGREES
MAXUMUM PRESSURE REQUIRED
- 43 PSIG
OIL FLOW RATE -- 750
IMP GAL/HR
PLOTTED FROM
ROAD TEST DATA
OIL VALVE POSITION - DEGREES OPEN
0
10
20
30
40
50
60
10
20
30
40
50
ATOMIZER PRESSURE - PSIG

port in the oil regulating valve in the upper range of flow rates. It is therefore likely that satisfactory compensation would be attained by limiting the maximum atomizing pressure at that required for the maximum fuel rate, in other words, 43 psig. This limit is easy to attain with the type of valve selected.

Fuel Oil Piping

To provide a linear variation of oil flow rate with regulating valve position, modifications to the fuel supply system are necessary. In Figure 8, page 27, experimental data has been plotted showing rate of oil flow at different temperatures versus oil valve position, this information having been developed for a locomotive equipped with a standard fuel supply system. It will be noted that in the upper load range the unit increase in fuel rate falls off rapidly for a unit movement of the oil regulating valve, this effect being more pronounced at low temperatures (high viscosity). This condition is attributable to excessive friction loss in the fuel line which is governed by variations in the velocity and viscosity of the oil. The velocity for a particular volume of flow is dependent on the transverse area of the pipe while the viscosity is dependent on the fuel oil temperature. It can be shown that for laminar or smooth flow the pressure drop varies directly as viscosity and inversely as the fourth power of the pipe diameter. Thus reduction of viscosity by increasing the oil temperature and enlargement of the fuel oil line to reduce the velocity of flow will lower the pressure drop, the latter change having the greater effect since it varies as the fourth power. Again referring to Figure 8, the ultimate effect of the modifications outlined will be that the variation of flow with oil valve position will approach the straight line relationship indicated.

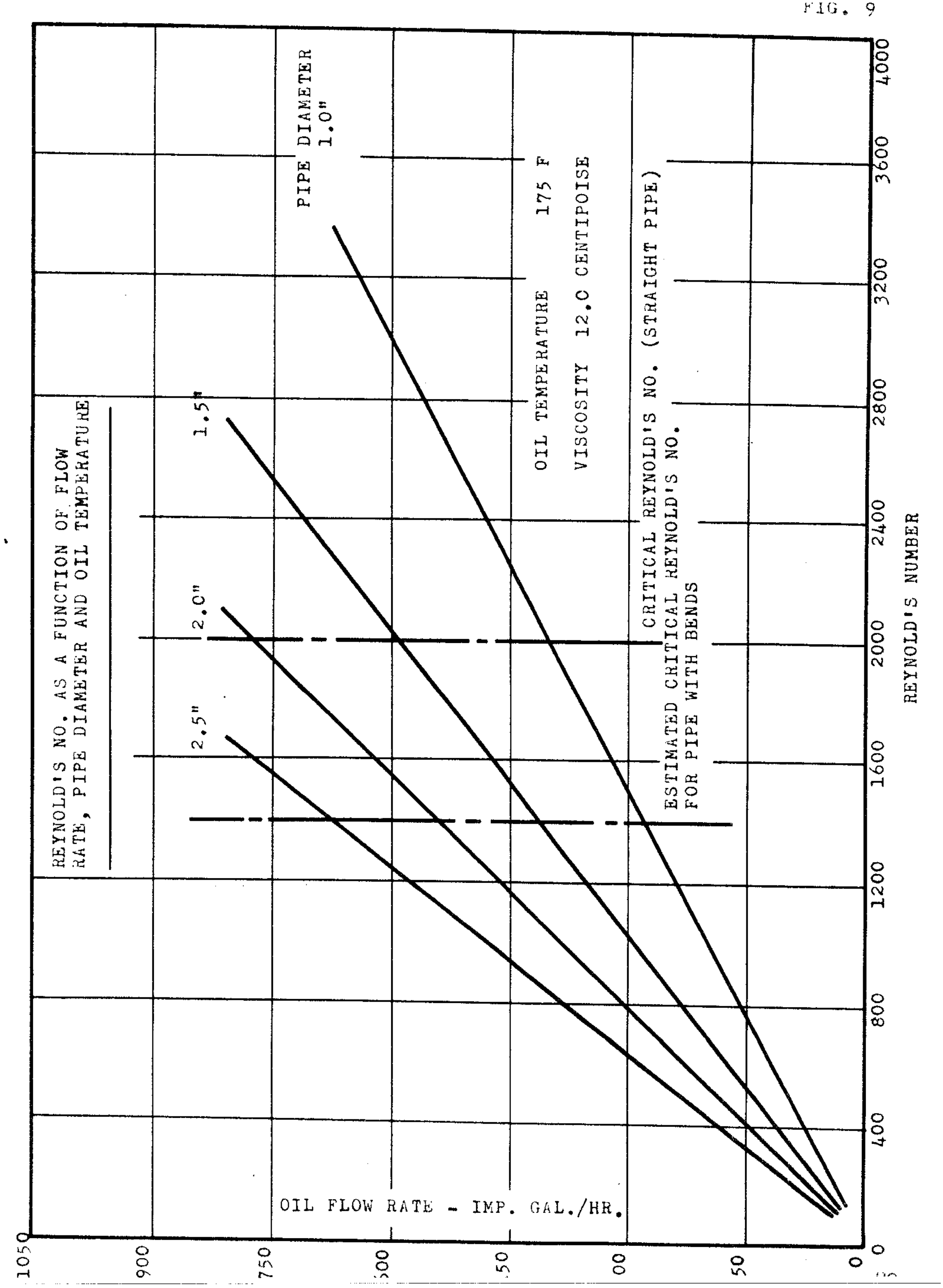


OIL FLOW RATE - IMP. GAL/HR.

Another undesirable condition which was experienced on road tests but which is not indicated in Figure 8, is an appreciable fluctuation in fuel rate which became most pronounced at high loads. An analysis of the flow conditions in the standard arrangement of piping indicates that the flow becomes turbulent at high rates. This produces a variable friction loss which causes irregular rates of flow for a particular setting of the oil regulating valve, making manual control of the air fuel ratio difficult. The nature of flow, whether laminar or turbulent, depends upon a numberical value known as Reynold's number, which in turn depends upon the pipe diameter, the density and viscosity of the liquid and the velocity of flow. Since velocity is related to pipe diameter, it can be shown that in the application we are considering Reynold's number varies inversely as the product of viscosity and pipe diameter. In Figure 9, page 29, Reynold's number at a viscosity corresponding to a fuel oil temperature of 175 F, has been plotted against oil flow rate for various pipe diameters. The critical Reynold's number at which laminar or smooth flow ceases is indicated. In order that this value of 1400 is not exceeded, the oil pipe should be $2\frac{1}{2}$ inches in nominal diameter in comparison to the present standards of $1\frac{1}{2}$ and $1\frac{1}{4}$ inches.

Considering the variables involved, suitable control of viscosity for the varying grades of oil can be attained by regulating the fuel temperature at 170 F and enlarging the oil line to $2\frac{1}{2}$ inches. The number of bends or Barco joints should be kept to a minimum and their passage area and that of the telescoping expansion joint between engine and tender should be at least equal to the minimum area of a $2\frac{1}{2}$ inch pipe. The present oil regulating valve, although having $1\frac{1}{4}$ inch unions, will be satisfactory since the major pressure loss

FIG. 9


REYNOLD'S NO. AS A FUNCTION OF FLOW RATE, PIPE DIAMETER AND OIL TEMPERATURE
PIPE DIAMETER 1.0"
1.5"
2.0"
2.5"
OIL TEMPERATURE 175 F
VISCOSITY 12.0 CENTIPOISE
CRITICAL REYNOLD'S NO. (STRAIGHT PIPE)
ESTIMATED CRITICAL REYNOLD'S NO. FOR PIPE WITH BENDS
OIL FLOW RATE - IMP. GAL./HR.
1050
900
750
0
REYNOLD'S NUMBER
0
400
800
1200
1600
2000
2400
2800
3200
3600
4000

occurs across the valve seat and this control area must of course be maintained. The oil line should be reduced on either side of the regulating valve. At the burner, the line should be reduced to the $1\frac{1}{4}$ inch oil connection. To minimize radiation losses during cold weather, the oil line and joints should be covered with sectional insulation enclosed in a sheet metal casing.

Fuel Oil Temperature

At present, the temperature at which the fuel oil is burned varies at the discretion of the fireman, subject to the advice of the road foreman of engines. As would be expected, wide differences of opinion exist which stem from the assumption that practices followed in industrial installations may be applied to locomotives. High temperatures are normally required with mechanical atomizing burners and this no doubt leads to the malpractice followed by some firemen of maintaining locomotive oil temperatures as high as 200 F.

The variations in grade and source of bunker fuel oil available from Canadian refineries, make rigid viscosity control impossible. Optimum temperatures for ensuring a free supply of oil to the burner would range between 130 and 180 F. As previously discussed, the effect of varying viscosity can be minimized by regulating the oil temperature and modifying the supply piping. It is therefore necessary to define a standard operating temperature at which the fuel oil should leave the supply tank. For the type of oils received, sludge formation increases rapidly above 190 F. In addition, radiation and evaporation losses become appreciable. The primary consideration, however, is the minimum flash point specified by the Canadian Pacific Railway, which is 175 F. To ensure that this limit is not exceeded, the fuel oil temperature should be maintained at 170 F. This temperature should be automatically controlled by a temperature regulator.

Fuel Oil Heating

At present, heating of the fuel oil is accomplished by blowing live steam directly into the oil tank. Although this method is simple and meets the fundamental requirements of a heating system, it has certain inherent disadvantages.

The accumulation of water in the fuel oil due to condensation has a detrimental effect on the combustion process. Examination of oil temperatures from experimental data shows that at all but the higher flow rates, the oil temperature rises above the boiling point of water at the burner entry. At low flows this temperature increases above 250 F. Since further heating occurs in the burner itself due to the proximity of the atomizing steam passage, it is evident that violent vaporization takes place. This action must cause the oil to flow in spurts and is one of the factors contributing to firebox drumming which is most pronounced at low rates of flow. Under certain conditions, vaporization at the lip of the burner will also cause carbon formation.

Water is a contributing factor to oil tank boil-over. On heating fuel oil itself will expand approximately 1% by volume for each 25 F change in termperature. If water is not present in the oil, a 100 F rise will only cause a change in level of approximately 2½ inches, in which case boil-over may occur if the tank is filled right to the top. The specific gravity of the fuel oil in current use is practically the same as that of water and therefore condensation accumulations in the tank tend to form an emulsified mixture with the oil. If excessive heating occurs, foaming may result, causing a rapid increase in volume with the danger of boil over. Another less serious factor is that tank measurements are not dependable under this condition.

Another drawback to the direct heating method is that the full body of oil in the tender must be maintained at the combustion temperature. Thus, steam requirements are stepped up considerably during cold weather due to radiation losses. Although rather difficult to estimate, rough calculations show that these losses require an additional steam flow of at least 200 lb per hour during cold weather. This amount of steam is produced by approximately 24 lb of oil per hour and is, of course, almost constant. Since fuel is consumed at an average rate of 300 gal per hour, radiation losses amount to roughly 0.8 percent of the fuel rate.

Besides overcoming the problem of water accumulation, fuel oil heating requirements may be reduced considerably by the use of a closed type heater from which the condensed steam is discharged to atmosphere through a trap. By designing the heater on the exit principle, whereby only the oil that is used is heated, radiation losses become negligible since the main body of oil is maintained at a low temperature. Due to the rapidly varying flow rates and consequent steam demands, this type of design necessitates the use of an automatic temperature regulator. As outlined previously, the application of this type of control is also necessary to facilitate regulation of oil flow rates and atomizing steam pressures.

Experimental work along this line was carried out during the past winter. The results indicated that oil temperature regulation is quite feasible.

Blower

The blower is a very necessary auxiliary on the steam locomotive, being used to create draught under drifting or standing conditions when exhaust steam is not available for this purpose. Generally speaking, the handling of this important control leaves much to be desired. During periods when its use is necessary, the blower pressure is frequently much higher than need be. In addition, firemen are always prompt in turning the blower on, but are usually negligent about turning it off when no longer required.

Examination of test data for a typical run in freight or passenger service shows that of the total amount of steam supplied to the blower, approximately 50 percent is wasted. Under drifting conditions, the blower is frequently operated at such a high pressure that 4 times as much air is inducted as is required to burn the amount of fuel being fed to the burner. This source of waste may be partially prevented by limiting the maximum blower pressure through the introduction of additional resistance in the blower line. It also lends itself to correction by the application of air fuel ratio controls.

Under load conditions, the smokebox draught increases rapidly until at 3 to 5 inches of water the blower contribution becomes insignificant. After this point is reached, it frequently takes considerabl time before the fireman realizes that the blower valve should be closed This forgetfulness is most wasteful and data from a typical run indicates its seriousness.

Elapsed Time Before Blower Valve Closed Minutes	Average Smokebox Draught Inches Water	Average Rate of Steam Flow to Blower lb per hr
12	15	2,750
32	13	2,800
10	17	2,880

During periods when the blower is not required, most firemen "crack" the control valve to varying degrees. When questioned, they advance the theory that they may not be able to open it quickly enough if it is closed right off. Although this practice does not appear too serious on the surface, its wastefulness is again borne out by the following data from the run mentioned previously.

Time Under Load Conditions Blower Valve Cracked Minutes	Average Smokebox Draught Inches Water	Average Rate of Steam Flow to Blower lb per hr
31	18	480
20	19	340
22	16	840

Table I, page 36, is a summary of blower operation for four typical runs. The indicated wastage is that due to cracking of the valve and negligence in turning it off when not required. No account is taken of the loss due to the use of higher blower pressures than required.

The wastage indicated in Table I may be prevented by the installation of an automatic control valve in the blower line. This valve would be actuated by the smokebox draught and would be set to operate at 3 to 4 inches of water, in other words, at that draught under load conditions above which the blower contribution is insignificant. The additional resistance imposed by this valve would also reduce the maximum blower pressure, thereby limiting the loss of steam when excess pressure is used.

Air Fuel Ratio Adjustment

The efficiency of manual control on the air fuel ratio was thoroughly analyzed from operational data. Under full load conditions when firing must be reasonably good to maintain steam requirements, the air fuel ratio approached the theoretically correct value. At light loads and under changing loads, however, firing was usually bad and the air supply varied from 50 percent to 1000 percent of requirements.

Table II, page 36, was prepared to illustrate the fuel wastage due to improper regulation of the air fuel ratio. It takes into account only that loss due to the induction of excess air over correct requirements.

Since possible savings would amount to approximately 10 percent of the total fuel burned, consideration was given to the application of automatic air fuel ratio controls. A suitable system of regulation was proposed by the National Research Council which would necessitate the positive measurement of the rates of air and oil flow. This would entail, in addition to points already covered in this report, pressurization of the fuel supply tank and the installation of airfoil members in the air entry ducting. The former move, although requiring structural strengthening of the oil tank, appeared feasible since pressurized fuel systems on locomotives have been used by some United States railways. The latter application, however, was not considered practical since it would be subject to error and failure during the winter months when snow and ice may accumulate in the bootleg. In addition, it was not felt that the sensitive controlling mechanism required would be suitable for the rugged service to which steam locomotives are subjected.

An alternative method of reducing the loss due to poor regulation of the air fuel ratio was suggested. This would entail the

TABLE I
OIL WASTAGE DUE TO FAULTY BLOWER OPERATION

Run	Period of Run Considered	Total Oil Consumed During Period	Blower On		Blower Req'd On		Blower Cost - in Terms of Oil Used: Actual		Required		Oil Wasted % of Total Consumption
	Hr.	Gal.	Hr.	% of Total	Hr.	% of Total	Gal. of Oil	% Total Oil Con.	Gal. of Oil	% Total Oil Con.	
F-A	9.0	2642	6.95	77.3	4.4	48.9	119.3	4.53	91.4	3.45	1.1
F-B	8	3080	7.3	91.2	3.42	42.7	116.6	3.8	44.2	1.4	2.4
P-A	9.3	2610	7.85	84.5	4.41	47.5	120.5	4.6	60.6	2.33	2.3
P-B	6.3	1883	2.9	46	1.04	16.5	49.8	2.65	21.2	1.13	1.52

TABLE II
OIL WASTAGE DUE TO INCORRECT AIR-FUEL RATIO CONTROL

Run	Miles	Load	Total Oil Con.	Oil Rate	Equiv. Oil Wasted Due to Improper Air Supply	
			Gal.	gal./ton mile	Gal.	% of Tot.
Freight A	136.6	2989 tons	2642	.00648	253	9.6
Freight B	136.6	2412 tons	3080	.00927	280	9.4
				gal./car mile		
Pass. A	246.8	15 cars	2610	.070	313.7	12
Pass. B	145.7	12 cars	1883	.108	210	11

application of indicating gauges to aid the fireman in the setting of his controls. The main instrument would be a simple and rugged air fuel ratio indicator placed adjacent to the firing valves. The control mechanism for this instrument would be placed in the smoke-box to give a continuous analysis of the flue gases (by density readings). By this means the fireman would have positive information to tell him when he is handling the controls properly. The most desirable feature of this set up is that any failure on the part of the control equipment would not affect the performance of the locomotive.

Although not absolutely necessary, the use of a rapid response steam gauge in conjunction with the air fuel ratio indicator would facilitate the duties of the fireman. After boiler pressure is attained, some time usually passes before the fireman knows whether he is gaining or losing pressure and can adjust his controls accordingly. Under such conditions, a rapid response gauge showing rate of rise or fall of steam pressure would be an asset. It would in effect indicate whether equilibrium was being maintained between the heat input and output of the boiler.

Another auxiliary that would definitely be most advantageous is a light placed beside the smokestack so the fireman could view the exhaust at night. At present, the only indication that the firemen has of the air fuel ratio is the colour of the exhaust, which should be only slightly hazy. Satisfactory observations may be made during daylight hours, but it has frequently been observed at night that maladjustment of the air fuel ratio may exist over extended periods until lights from outside sources show that black smoke is issuing from the smokestack. This is particularly true on rainy or foggy nights. A satisfactory light with a control switch on the fireman's side of the cab could be installed at moderate cost. In view of the wastage due to poor regulation it is felt that any expense in this respect would soon be offset.

Sanding of Tubes and Flues

Sanding should be done when a locomotive is working hard, creating a strong, steady draught. The most satisfactory results are attained at high speeds when the mass flow of gas becomes greatest. In no case should sanding be done at speeds below 15 mph when the draught is intermittent. Since the inrush of air from the bootleg assists in directing the sand over the brick arch, the dampers should be left in the normal running position (dampers are frequently closed to varying degrees). This is also desirable so that the air fuel ratio will not be altered. The scoop should be held just above the sand hole and the sand allowed to flow into the air stream at a moderate rate. Placing the scoop in the sandhole is not desirable since the air stream will pass above the sand tending to force it downwards below the arch. If the sand is introduced too rapidly, it will also have a tendency to fall to the floor of the firebox.

Due to the deflecting effect of the arch and to malpractice in sanding methods, sand deposits on the floor of the firebox frequently become appreciable (often 12 to 14 inches in depth), and with exposure to the intense firebox heat, extensive slagging occurs. When the fused sand is removed from the firebox, the brickwork is often damaged to the extent that portions of it must be replaced. The cost of firepan brick repairs amounts to approximately $8,000 per month for the 264 locomotives in service. Of this total, less than one eighth is direct material cost.

A suggestion has been advanced that more satisfactory results could be attained by introducing the sand in the combustion chamber, thereby eliminating the possibility of deposits on the firebox floor. The proposed set-up will require an auxiliary sand dome on top of the boiler with two injection nozzles, one on either side of the combustion chamber, operated by an air valve on the fireman's side of the cab. By placing the nozzles just ahead of the brick arch, uniform distribution over the tube sheet, as is attained by the standard method, should be ensured.

Although the application of an auxiliary sand dome and operating equipment may prove expensive, it is felt that this arrangement should be adopted in view of the possible reduction in maintenance costs.

Economic Considerations

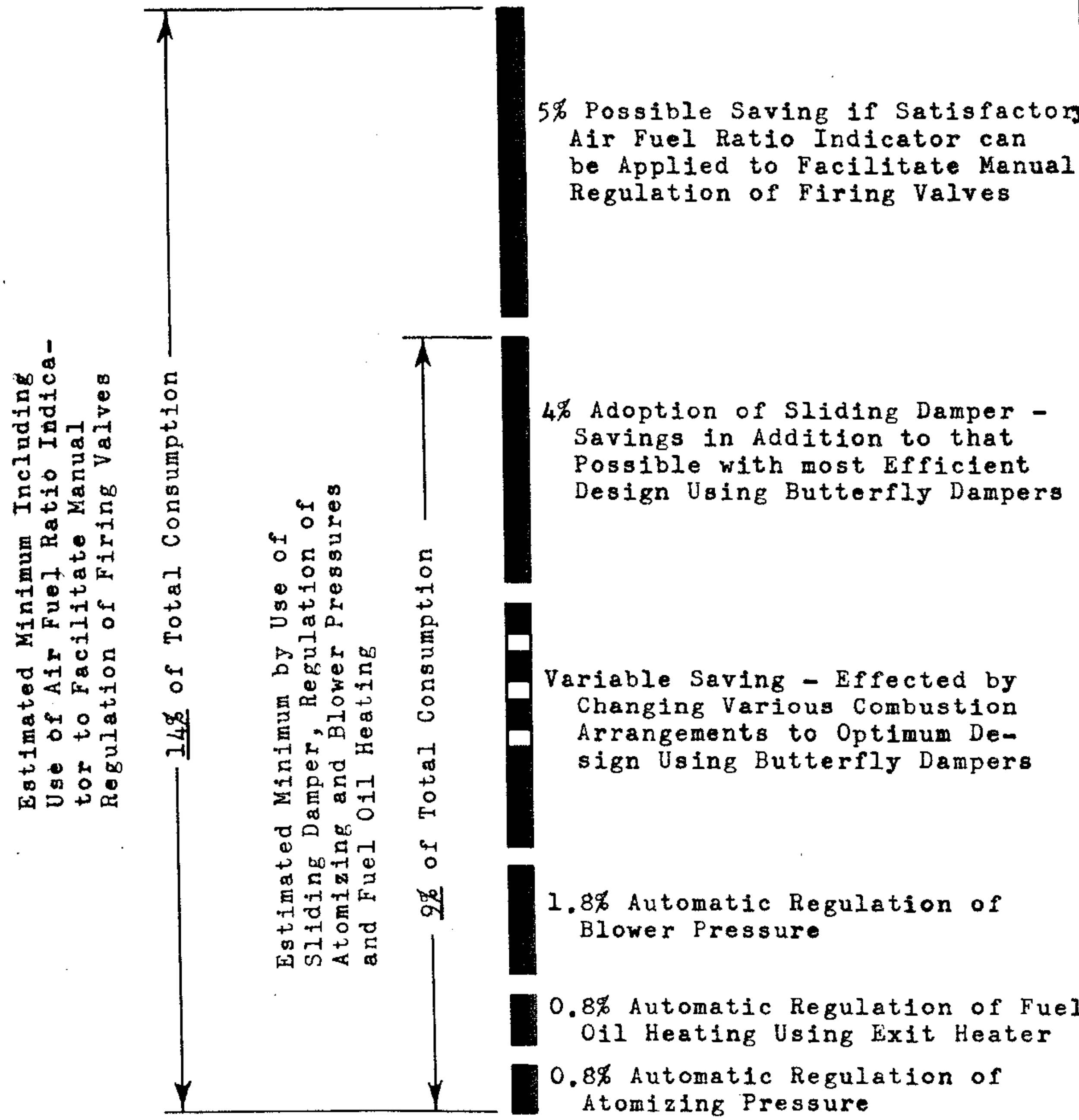
The possible savings which may be effected by the adoption of the recommendations outlined in this report have been summarized in the graph of Figure 10, page 40. One item has been shown as a variable saving since fuel consumption tests which were carried out on different classes of locomotives indicated that savings through modification of design were dependent on the type of arrangement in use.

The percentage fuel savings shown were developed through extensive field and laboratory tests and are generally conservative estimates. It will be noted that a minimum reduction in fuel consumption of 9 percent may be expected by the adoption of the sliding plate damper and the regulation of atomizing and blower steam pressures and fuel oil heating.

FIG. 10

Graphical Summary of Possible Economics by Redesigning Combustion Arrangement on Oil-burning Steam Locomotives

Fuel Savings



Maintenance Savings

Reduction of Arch Brick Deterioration and Formation of Sand Slag Effected by use of Sliding Damper and Modification of Sanding Method.

Following Figures Show Source of Savings:

Arch Brick Renewals $6,400 per month

Floor Brick Renewals and Removal of Sand Slag $8,000 per Month (264 locomotives)

Estimated Minimum Saving -
$69,000 per Year (264 locomotives)

As was pointed out in the section entitled "Air Fuel Ratio Adjustment", a further fuel saving of some 10 percent can be effected by the use of positive air fuel ratio control equipment. This was not felt feasible, however, and as an alternative improvement, the use of air fuel ratio indicators to facilitate the manual operation of the firing valves is advocated. It seems reasonable that an additional 5 percent saving could thereby be effected, providing of course, that adequate instruction covering the purpose of the indicator is given. Thus, if all recommendations outlined are introduced, a minimum reduction in fuel consumption of 14 percent may be expected.

The following economics are based on operations for the year 1950:

<u>Prairie and Pacific Regions</u>

Average number of oil-burning locomotives	228
Total number of locomotive miles	11,284,126
Average number of miles per locomotive	49,492
Total fuel consumption	3,058,562 bbl
Consumption per locomotive mile	0.27105 bbl
Consumption per locomotive per year	13,415 bbl
Weighted cost of fuel (excluding OCS transportation)	$1.51 per bbl
Fuel cost per locomotive per year	$20,257
14% fuel economy - approximate saving per loco. per year	$2,800
For the 264 locomotives in service as of Jan. 1, 1951	
Saving per year	= $740,000

In addition to direct fuel savings, the test results indicated that an appreciable reduction in firebrick maintenance costs should be effected. It is felt that these expenses, amounting to some $14,400 per month ($173,000 per year) for the 264 oil-burning locomotives in service, can be reduced from 40 to 50 percent. This will result from a reduction of arch brick deterioration and formation of sand slag by the adoption of the sliding damper and the modification of sanding methods, which will mean a minimum additional saving of $69,000 per year.

Although complete information on the necessary capital outlay has not been developed, the results of the work completed to date would indicate that necessary expenses should be offset well within a period of one year.

Further Development Work

The necessary details of construction and assembly of the air inlet system using the sliding plate damper have been developed. Tests which were carried out during the past winter indicate that automatic regulation of fuel oil heating is quite feasible. There were several defects in the operation of the heater unit applied, notably in the sensitivity of control, but this may be corrected by changes in construction.

The control equipment for the regulation of the atomizing and blower steam pressures will require further development work to adapt it to locomotive use. A cam and lever operated pressure regulating valve has been selected for the atomizer control. Calibration tests will be necessary so that the required cam profile may be determined. It appears that a solenoid type valve will be most suitable for the blower control, actuated by some form of draught switch. The National Research Council has looked into this type of control and it is felt that a satisfactory system can be developed.

The Research Council has been doing work towards the development of a practical air fuel ratio indicator that lends itself to adaptation for locomotive use. It is felt that the Council should be requested to carry this work through so that a test application can be made at an early date.

Consideration has been given to the possibility of failure of any of the recommended control units. The steam supply line for the atomizer and fuel oil heating systems will be taken off the present supply line to the auxiliary manifold (which feeds manually controlled atomizer, tank heater, blow back and blow forward valves). If failure occurs in either system (as indicated by an atomizer pressure gauge or the thermometer in the oil tank), the fireman will immediately be able to revert to the normal manual controls. A by-pass line will be installed around the solenoid valve on the blower line. This control, however, should stand up satisfactorily since it has proven dependable for use in the electromatic blowoff system. If a defect occurs in the air fuel ratio indicating system, it will not affect the operation of the locomotive.

In view of the encouraging results obtained to date, it is recommended that the following applications involving a total of six oil-burning locomotives be made at the earliest opportunity:

1. To a T-1 class locomotive:

   a. A sliding-type damper.

   b. A cam and lever operated atomizer control valve linked to the oil regulating valve. This application will require a steam pressure reducing valve and an atomizer pressure gauge.

   c. An enlarged oil supply line.

   d. An exit oil heater and automatic temperature regulator.

   e. A light at the smokestack to illuminate the exhaust at night.

2. To an H-1cd class locomotive:

   a. A sliding-type damper

b. A modified sanding arrangement which will inject sand through the combustion chamber for cleaning the tubes and flues.

3. To a second T-1, a second H-1 cd and two H-1e class locomotives:

a. A sliding-type damper.

Following this work, it would be advisable to carry out further tests on a locomotive of the N-2 or P-1n class having a smaller combustion volume. This test program would provide sufficient information so that a complete statement of possible economies, including capital outlay, could be prepared. If the controls advocated prove feasible for locomotive use, a program of standardization of design could then be initiated so maximum efficiency and economy of operation will be assured.