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LOCOMOTIVE DESIGN

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Part I

BOILER AND CYLINDERS

SECOND EDITION

CONTENTS

CHAPTER I

PRELIMINARY CONSIDERATIONS*†

Of all the various departments constituting the working elements of a railroad system, that of the motive power is undoubtedly one of the most important, and it is becoming more and more recognized that the greatest care and the highest skill must be exercised in order to obtain that degree of efficiency which the modern management demands. That this may be done necessitates not only extensive experience and an intimate knowledge of the general principles of mechanics and engineering, but a sound judgment as to the peculiar characteristics of the locomotives and rolling stock that shall adapt them to the conditions of the road over which they are to be operated.

It is, therefore, the duty of this department to thoroughly investigate the character of the freight to be moved, the strength of the bridges as well as the peculiarities of the roadway, as a basis for forming an intelligent opinion as to the power to be provided to comply not only with the present conditions imposed by the operating department, but to allow for future expansion and development.

The object, then, of this treatise is to prepare a simple and comprehensive guioe, supplemented by examples from practice, by which the type and size of an engine for a definite service may be determined. In order to convey some idea of what it means to design a locomotive that will be well adapted to economically perform the work which it is desired that it shall do, a few words will be of value, to set forth some of the many and varying influences that have a bearing on the problem. The officers in charge of a railroad may have preconceived ideas as to the economical value of the heaviest types of engines for the hauling of a high tonnage, but they will almost invariably find themselves limited in the matter of the application of such engines by the weight of the rail that is down, the strength of the bridges, and the clearances of the permanent way.

Added to these, the grades, the quality of the coal that will be available, the speeds and character of the traffic are modifying factors that will affect not only the final details but the type of the design. It must be understood also that, despite the scrutiny and study to which the locomotive has been subjected, there is still much to be learned and a vast amount of research work must still be made before the designer will have all of the data that he needs for the task to which

* The present number of MACHINERY'S Reference Series is the first part of a treatise on complete Locomotive Design, covered by Nos. 27, 28, 29, and 30 of the Series, and originally published in RAILWAY MACHINERY (the railway edition of MACHINERY). Each of the four parts of the complete work treats separately on one, or more, special features of locomotive design; and while the four parts make one homogeneous treatise on the whole subject, each part is complete by itself.

† MACHINERY, Railway Edition, October, 1904.

he has set himself. It is, therefore, in view of these limitations impossible to formulate any set of hard and fast rules that can be made to serve and which will be accepted as absolutely correct in all quarters.

In order, then, that what follows may serve somewhat as a guide to show what has been done in certain concrete cases, a number of assumptions will be made as a basis of work, and two locomotives will be worked out and developed that will meet the requirements of this supposititious road and the assumptions that will be made in connection with its operation. It will be understood that any variation from the assumed conditions may cause modifications of design that will be more or less extensive according to the character of the variations.

In order that a start may be made, we will suppose that two locomotives designed respectively for freight and passenger service are to be designed for a division one hundred and fifty miles in length, that is laid with rails weighing 75 pounds to the yard, whose bridges are of such strength that they are up to the full capacity of the rail,

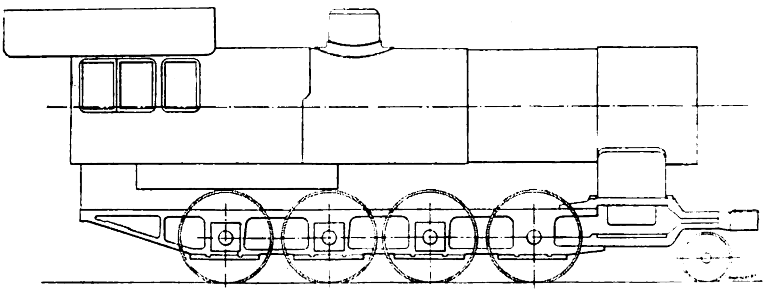

Fig. 1. Outline of a Consolidation Freight Locomotive

with clearances that permit the usual widths and heights of design, and finally with a ruling grade of one per cent 10 miles in length. These figures, scanty as they are, will suffice as a guide to indicate to the locomotive designer the conditions that he must meet. It is, of course, beyond the province of this work to enter into a discussion of the construction of the track, so that it will be merely taken for granted that this is of the most approved character and is kept in first-class condition.

With this data at hand, the requirements usually sent to the locomotive designer are that he shall supply an engine that will haul a given tonnage over a ruling grade at a minimum speed. In the drawing up of the specifications in this form, judgment, backed by experience, must be exercised that the requirements do not call for a heavier engine than the rails are able to carry.

The first steps, then, in the determination of the size and character of the engine that can be used is to ascertain the weight per wheel that can be safely carried upon the rail. This depends upon the metal of which it is formed, the shape, and size or weight. If we take the fiber stress to be put on the rail under a static stress as 12,500 pounds, we find that a wheel load of 22,000 pounds will meet the requirements for a 75-pound rail, and this experience has proved to be good current practice.

For heavy freight work on roads of ordinary curvature, it has been found that four driving wheels coupled are about as many as can be satisfactorily worked. Engines with larger numbers have been built, but even those roads using them have reverted to the consolidation type, having four pairs of wheels coupled and a pony truck in front.

In the case of our supposititious division then, the type that current practice would suggest for adoption would be a consolidation engine, and the weight upon each driving wheel according to the assumed conditions would be about 19,400 pounds, with 21,000 pounds on the truck, or about 176,000 pounds for the total weight of the whole machine.

Experience has shown that the tractive power of an engine can be made from 22 to 26 per cent of the weight on the drivers. In the

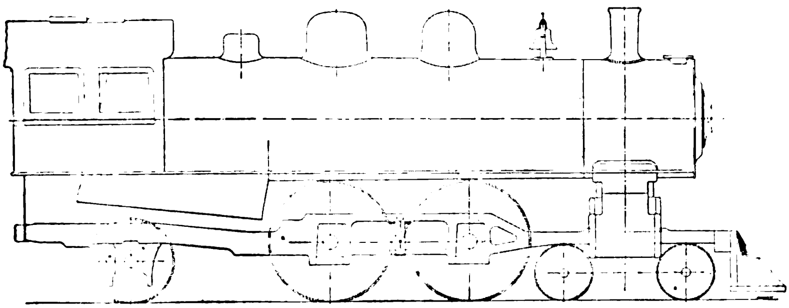

Fig. 2. Outline of an Atlantic Type Passenger Locomotive

present case we will use the former as suitable to the designs to be presented, which would make a total tractive force of 34,200 pounds from which 10 per cent should be deducted for the internal friction and resistance of the engine, leaving 30,800 pounds or about 31,000 pounds available tractive power. When an attempt is made to refer this drawbar pull to the weight of train that can be hauled, we find at once a mass of variable resistances that again render accurate calculations an impossibility. Train resistances will vary with the number of cars, the conditions of the journal lubrication, the direction and force of the wind, and many other minor details of the construction. The best that can be done then is to take an approximate formula for train resistance and make due allowance for the excess resistances resulting from emergencies.

There are some variations in the formulas given by different authorities. The work of foreign specialists is of little or no value to American designers on account of the difference in the rolling stock experimented with, and of all the work done in this country the formula known as that of the *Engineering News* is the most widely accepted for approximate reliability. This formula is

$$R = \frac{1}{4} v + 2 \qquad (1)$$

in which R is the resistance in pounds per ton of 2000 pounds in the weight of the train, and v the speed of the train in miles per hour. By substituting a value for v of 10 miles per hour, in the example under consideration, we find the rolling resistance to be

$$R = \frac{10}{4} + 2 = 4.5 \text{ pounds per ton of 2000 pounds.}$$

The resistance due to grade is found by the formula

$$R' = l \times \frac{p}{100}, \text{ in which} \qquad (2)$$

R' = resistance in pounds,
l = load in pounds to be carried up the grade,
p = percentage of gradient,

From this it appears that the resistance per ton of load on a 1 per cent grade is

$$R' = 2000 \times \frac{1}{100} = 20 \text{ pounds.}$$

To these resistances should be added the resistance due to curves, which is found to be about 0.5 pound per degree of curvature per ton of load where the curves are not compensated for in the reduction of the grades. On modern roads, however, the curves are generally compensated for in the gradient and it will therefore be omitted in the example and the rolling and grade resistances only used. Hence $R + R' = 4.5 + 20 = 24.5$ pounds to the ton, leaving the minor uncontrollable resistances to be taken care of by a certain margin provided for such purposes and other emergencies.

We have seen that the available tractive power of an engine to do the required work will have to be 30,800 pounds under the driving wheels, but it would not be advisable to load the engine to these figures under all conditions without allowing a reasonable margin for the uncontrollable resistances, previously referred to, in the variation of car resistances and weather conditions of about eleven per cent, which leaves 27,400 pounds for moving the train, inclusive of the weight of engine and tender.

By dividing this amount by the resistances obtained from Formulas (1) and (2) we get the total weight of the train to be hauled up the one per cent grade at the rate of ten miles per hour, *viz.:*

$$\frac{27{,}400}{24.5} = 1118 \text{ tons of 2000 pounds.}$$

Before it is possible to determine the load behind the tender, it is necessary to work out the dimensions of the engine and thus ascertain the weight of the engine and tender. With the requirements before us, experience has taught that the following dimensions of wheels and their related parts will most satisfactorily meet the conditions, namely:

Diameter of driving wheels, 57 inches.
Boiler pressure, 200 pounds.
Strokes of pistons, 26 inches.

With these dimensions given, we get the cylinder diameter from the formula:

$$d = \sqrt{\frac{T \times D}{P \times 0.85 \times S}}, \text{ in which} \qquad (3)$$

d = diameter of cylinder,
T = the required tractive power,
D = diameter of driving wheels,
P = boiler pressure,
S = stroke of pistons.

As 85 per cent is the generally adopted coefficient of the boiler pressure for average cylinder pressure at low speed, that is taken as a constant factor. By substituting the values decided upon in tne formula we obtain the cylinder diameter

$$d = \sqrt{\frac{34{,}200 \times 57}{200 \times 0.85 \times 26}} = 21 \text{ inches.}$$

The weight allowed on the driving wheels is 155,000 pounds and by ordinary proportions in the design of an engine of this type about 21,000 pounds will come on the truck, making a total weight of the engine alone of 176,000 pounds, and the weight of the tender filled with coal and water will be about 110,000 pounds, which added together, makes 286,000 pounds or 143 tons.

This amount will now be deducted from the previously obtained total weight of the train to find the load behind the tender, namely: $1118 - 143 = 975$ tons.

A rational specification for an engine to work on such a grade, then, would be one capable of hauling a train of 975 tons at a speed of ten miles an hour, leaving a reasonable margin to be utilized under favorable conditions. The result will be a consolidation locomotive of the general outline shown in Fig. 1.

Turning now to the matter of the passenger locomotive, there are three types in common use in this country. They are the eight-wheeled American or 4-4-0 type, the Atlantic or 4-4-2 type, and the ten-wheelers or 4-6-0 type. The latter is heavier, and has a greater tractive power than the other two, and is intended for what might be called special services.

To these may be added a fourth, the Pacific type (4-6-2). The last, having the same number of drivers as the ten-wheeler, has greater boiler power in proportion to its adhesive weight, and is used in what might be called exceptionally heavy service. The first two classes bear, in a general way, the same relation to each other as the two last, namely, that of having the same number of driving wheels. The Atlantic type has the greater boiler power and is capable of maintaining a higher speed than its predecessor, the eight-wheeled engine.

The Atlantic type, first introduced in 1893, possesses so many advantages for heavy and fast passenger service, that it has been rapidly introduced for that purpose, supplanting the first type in many places where it was originally used.

In order, then, to simplify matters, it will be decided at the outset that the design will be made for the Atlantic type; and, as the work progresses the advantages possessed by the same will be set forth.

With the same weight of rail and the same conditions of track as those set forth in the determination of outlines of the freight engine, we would naturally have the same weight upon the driving wheels. But as there are but two pairs instead of four, the available tractive force drops to 18,250 pounds.

In the case of the passenger engine, there are some complications introduced into the calculation that do not enter into that of the freight engine. The most important one is that of speed. It is evident that ten miles an hour would not at all answer the requirements of passenger service even on the grade mentioned, and the work should be based on a speed of at least thirty-five miles an hour. Owing to the shorter cut-off that will be involved by such a speed, the full adhesive weight of the locomotive cannot be used so that a very liberal reduction will have to be made. The designer knows from analysis that only about 60 per cent of the total adhesive weight is available at such speeds, and as this should be cut down still more to allow a suitable margin for wind and other uncontrollable resistances there is left but little more than 10,000 pounds tractive power that can be used at the speed decided upon.

Referring back to the formula of the *Engineering News* for the train resistance at 35 miles per hour, we find it to be 10.75 pounds per ton, to which should be added the 20 pounds due to grade, making a total of 30.75 pounds per ton. If we divide the available tractive power of 10,000 pounds by 30.75, we obtain 325 tons as the weight of the train, inclusive of the engine, that can be hauled. If the speed were to be dropped to 25 miles per hour, this weight would be raised to something more than 350 tons. It would, therefore, be a reasonable specification on the part of the railroad officers to call for a locomotive capable of hauling 350 tons up a 1 per cent grade at a speed of 25 miles an hour, and, if such a specification were to be made, an engine like that shown in Fig. 2, having about 20,000 pounds on each of the driving wheels, and weighing in all 168,000 pounds, would be offered for the service.

CHAPTER II

THE BOILER*

In the preliminary considerations regarding the designing of the locomotive two main points have been approximately settled: the weight of the engine that can be made to produce the tractive power needed to perform the work that is required and the size of the cylinders that will be needed in order to utilize the adhesive weight of the engine, with the steam pressure of 200 pounds per square inch that it has been assumed the boiler is to carry.

It may be remarked here, that in the designing of a modern locomotive all things are made subordinate to the boiler and cylinders, and of these two the boiler is the more important. On it depends the whole action of the engine. If it fails to supply the requisite amount of steam the engine either cannot haul its train without losing time upon the schedule on which it is supposed to run; or, if it supplies the steam, its grate area and heating surfaces may be too small to do the work properly, and the result will be that the engine is extravagant in the use of fuel. For these reasons, then, it is the end and aim of every designer to use as large a boiler as possible in order to obtain an ample supply of steam, and at the same time secure that supply on a minimum fuel consumption. At the same time he is limited by the allowable total weights which must include not only the boiler itself but the cylinders, wheels, axles, machinery and other parts.

In this, as in the work that has already been done, it is impossible to lay down any hard and fast rules, and the designer will frequently find himself thrown back on his own judgment and experience in default of formulated data bearing upon the subject that he has in hand. With this understanding of the matter attention may now be turned to the determination of certain points connected with the boiler of the consolidation freight locomotive that we have in hand, and of which a preliminary outline has been laid down in Figs. 3 and 5.

The two important elements in the boiler are the heating surface and the grate area. The former takes the precedence and is usually based upon some assumed service that the engine is to render. In the case in hand this has been arbitrarily placed at the hauling of 975 tons as a speed of 10 miles an hour up a 1 per cent grade, or of moving 1118 tons including the weight of the engine and tender.

For some time the empirical rule for the determination of the amount of heating surface was to make it, in square feet, 400 times the cubic contents of a single cylinder in cubic feet. This rule is,

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* MACHINERY, Railway Edition, November and December, 1904, and January, 1905.

however, only approximately followed and is regarded merely as a rough guide as to what should be aimed at as a minimum. For, as already stated, it is the desire of the designer to make the heating surface as large as possible, so that this ratio is exceeded wherever it is possible to do so and still keep within the limitations of weights. This is especially true of work in connection with passenger engines, where the demand for steam is apt to be excessive.

Then, too, after the general dimensions of the boiler have been decided upon it is quite possible to vary the heating surface through comparatively wide limits by a variation in the spacing of the tubes. It is in this particular especially that good judgment must be exercised. There is the constant temptation, backed by desire to run the heating surface up, to use a large number of tubes, but it must be borne in mind that it is sometimes advisable to space the tubes more widely apart and put in a smaller number than it is to crowd them; because it is necessary that the steam formed in contact with the lower rows should be free to rise to the surface of the water, otherwise poor evaporation, damp steam or even priming may be the result. It is, therefore, usually better to sacrifice some of the heating surface that it might be possible to obtain, as this will give an actual increase in the evaporative efficiency.

Turning now to the determination of the amount of heating surface and regarding the rule given merely as an approximate guide, it is considered that a more correct basis for the estimate will be to ascertain the weight of steam that will be required to maintain a given speed and tractive power, and from that calculate the amount of heating surface that will be needed to produce it. In short, it is necessary to determine the amount of water that is to be evaporated per minute or per hour, and this, in turn, swings back to the cylinder, where the point of cut-off and the pressure will have to be assumed. This assumption should be based on the records of performances of other engines, and should be critically scrutinized in order to determine the influences that necessary variations in valves, valve motion and steam passages may have upon the result. This means a thorough investigation and the securing of reliable data if a close degree of accuracy is to be obtained.

For the solution of the specific problem that we have before us, the data available have made possible the development of the following formula for the determination of the amount of water to be evaporated per minute by an engine in heavy freight service:

$$W = 2uvnpc \times 1.25 \qquad (4)$$

in which

W = pounds of water to be evaporated per minute,
u = the volume in cubic feet of the two cylinders,
v = the percentage of the stroke at which cut-off takes place,
n = the number of revolutions per minute of the driving wheels,
p = the weight of 1 cubic foot of steam at the cut-off pressure,
c = the factor of evaporation from and at a temperature of 212 degrees F.

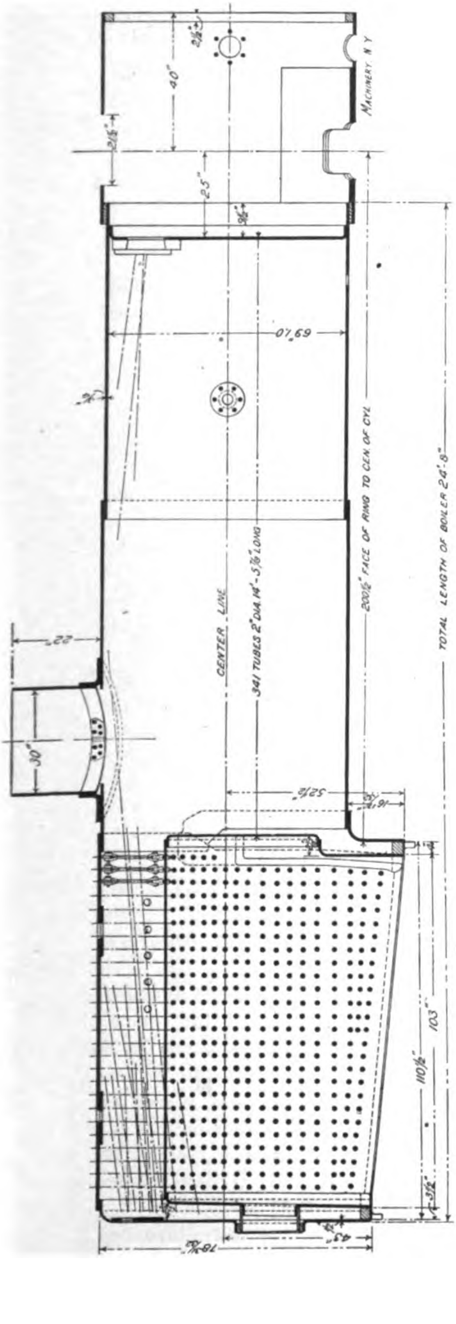


Fig. 3. Section of Boiler of Consolidation Freight Locomotive to haul Train of 975 tons up Grade of one per cent at Ten Miles per Hour

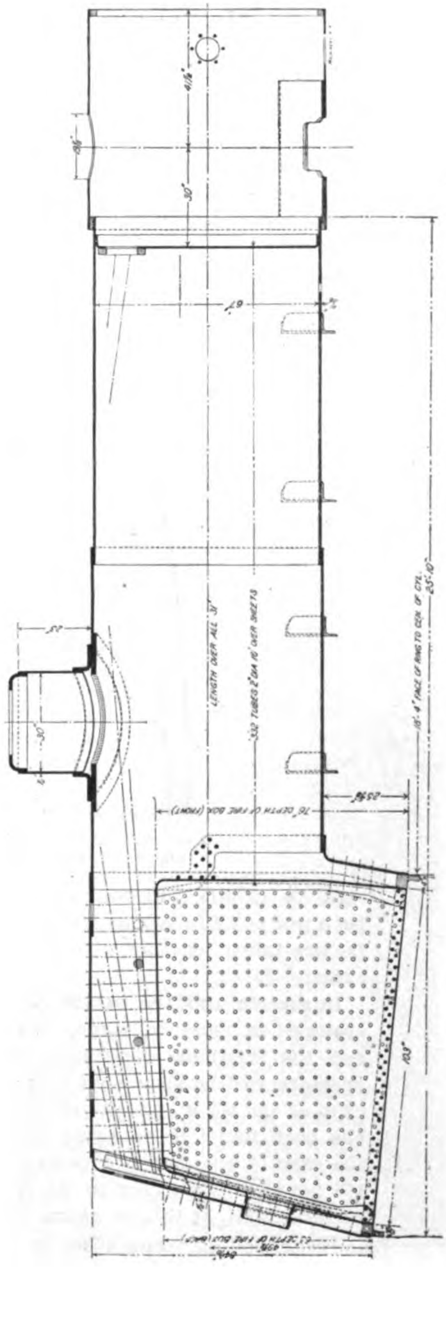


Fig. 4. Section of Boiler of Atlantic Type Passenger Locomotive to haul Train of 350 tons up Grade of one per cent Twenty-five Miles per Hour

It will be seen that this formula involves a few points that nave not yet been settled in the course of our work, and which must be decided upon before further progress can be made.

Taking up Formula (4) seriatim, the coefficient 2 is used because the cylinders must be filled with steam twice at each revolution. The factor u is readily determined, and for the case of the cylinders for the consolidation locomotive that are 21 inches in diameter and of 26 inches stroke it is 10.4 cubic feet. For an engine working at the speed and conditions that have been assumed, practical experience shows that the valves may be made to cut off at about 70 per cent of the stroke.

The speed of the engine and the diameter of the drivers are needed in order to determine n. We have already assumed that the speed

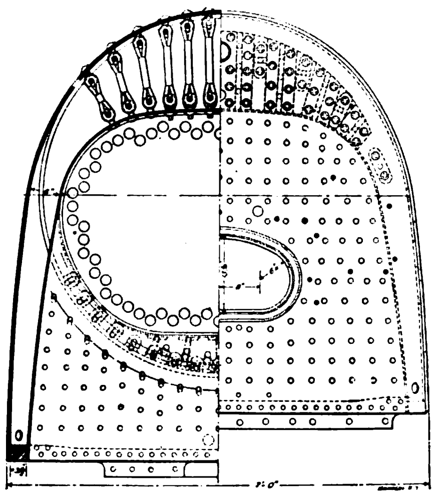


Fig. 5. Cross-section of Boiler shown in Fig. 3

shall be 10 miles an hour and the diameter of the driving wheels has been put at 57 inches as being well suited to the stroke adopted. These factors taken together give about 60 revolutions per minute, or, more exactly, 59.

In ascertaining the weight, p, of the steam at cut-off pressure another assumption must be made. Owing to the slow closing of the valve, and the frictional resistance of the steam pipes and passages, this pressure will be somewhat below that of the boiler, and for the 70 per cent of cut-off at which the engine will be worked at this speed, it has been found to be about 80 per cent of full boiler pressure, or in the case of our assumed pressure of 200 pounds per square inch it is 160 pounds. Referring to the steam tables as ordinarily published we will find that, at this pressure, the weight will be 0.3873 pounds.

The factor of evaporation is obtained by dividing the difference between the heat of the steam at the observed pressure (in this case 200 pounds), and the total heat of the feed water (which is taken at 32 degrees) by the factor of 965.7 which gives in the case in hand about 1.24. The 1.25 at the end of the term is an allowance made for clearances, leakages and the like. With these quantities obtained they may be substituted in Formula (4) with the following result:

$$W = 2 \times 10.4 \times 0.70 \times 60 \times 0.39 \times 1.24 \times 1.25 = 528 \text{ pounds.}$$

That is, the requirements of the engine are such that it must be supplied with the equivalent of 528 pounds of steam per minute from and at 212 degrees F., or 31,680 pounds per hour. The actual evaporation requirement is the amount obtained without the factor 1.25, or 25,350 pounds hourly.

Having determined the amount of work that the boiler will be required to do, the next step is to ascertain the details of dimensions

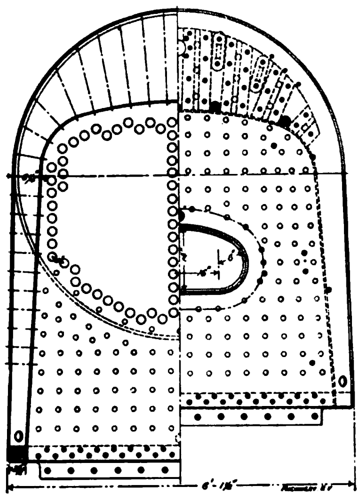


Fig. 6. Cross-section of Boiler shown in Fig. 4

of that boiler to meet the requirements. The first movement in this matter is one that varies with every road and every locality for which an engine can be designed. It involves a determination of the heating capacity of the coal that is to be used. For this reason, when a builder is called upon to design a locomotive whose performance is to be guaranteed he requires a sample of the coal that is to be used in order that its value may be ascertained and the proper proportions of grate area and heating surface be arranged. In the case in hand, it will be necessary to make an assumption and this will be done on the basis of a good quality of bituminous coal, that can be depended upon to evaporate 7½ pounds of water per pound of fuel in a properly proportioned firebox with a suitable ratio between heating surface and grate area.

Experimental examination has shown that, for this grade of coal, one square foot of heating surface is capable of absorbing the heat developed by 1½ pounds of coal burned per hour. We can now employ the following formula for the determination of the heating surface:

$$S = \frac{W_1}{f\,g} \tag{5}$$

in which

S = the area of heating surface in square feet,
W_1 = pounds of water to be evaporated per hour,
f = pounds of water to be evaporated per pound of coal,
g = the number of pounds of coal allowed per square foot of heating surface per hour.

Substituting the values already obtained the formula becomes,

$$S = \frac{31{,}680}{7.5 \times 1.5} = 2816$$

The grate area may be determined by the substitution, in Formula (5), for g a factor expressing the amount of coal to be burned per square foot of grate per hour. The equation then becomes:

$$G = \frac{W_1}{f\,h} \tag{6}$$

in which

G = the grate area in square feet,
h = the amount of coal in pounds to be burned per square foot of grate area per hour.

It is common practice in the freight service of American locomotives to burn 100 or more pounds of coal per square foot of grate per hour. This means that the fires are to be forced to a greater or less extent, and the conservative designer who wishes to make sure that his engine will meet all the requirements of the specifications will assume a much lower rate of combustion than this, in order to have a reserve power for use in cases of emergencies. It will be well, then, to take 75 pounds per square foot per hour as the rate of combustion. By substituting this and the other values that have been obtained in Formula (6) it becomes

$$G = \frac{31{,}680}{7.5 \times 75} = 56.3\text{, or } 57$$

The boiler called for should therefore have 2816 square feet of heating surface, and a grate area of 57 square feet. On comparing the ratio of grate to heating surface obtained in this way it will be found to be as 1 to 50, which is below the average of construction. It must be borne in mind that these are not hard and fast dimensions, but may be varied according to the exigencies of the case, the limitations of construction and the modifications suggested by the experience of the designer. The distribution of the heating surface and the determination of the weight of the boiler is the next step.

In this, much depends upon the type of the boiler, the wheelbase and other features of general construction. These limit or indicate the length of the tubes. This may range anywhere from 13 feet to 20 feet, although about 15 feet is now an approved dimension. In a general way it may be said that the longer the tube the more heat is extracted from the gases of combustion, though it has been found that if the heating surface remains the same the length of the tube is a matter of indifference within certain limits. The reason for this is that with a long tube the gases move rapidly but remain in it for the same length of time as in a short tube where they move slowly, there being more short than long tubes for the same heating surface. As for the weight of the boiler this may be taken to average from 30 to 33 per cent of the total weight of the engine.

The type of the boiler is usually settled by preference and past experience with a limiting factor of present requirements. The general types of boilers are, broadly speaking, the straight and wagon-top. In either of these the crown of the firebox may be stayed with crown-bars or radial stays, or be of the Belpaire type, and the firebox itself may be broad or narrow; that is, it may be extended over the wheels, set on top of the frames or drop down between them. The straight-top boiler is the more common of the two first divisions because of the inconvenience of utilizing the wagon-top on large modern engines, where everything is set well up, on account of the limitations of the overhead clearance of the permanent way.

Of the methods of staying the crown-sheet, the crown-bars have been practically discarded. The Belpaire type has not been widely adopted, while the use of radial stays is most widespread. Finally the wide firebox set above the wheels is a necessity for large engines in order to obtain the large grate area required in modern motive power, and the narrow construction between the frames may be regarded as a thing of the past. So in the boilers for the engines under construction, the straight-top type, with a wide firebox held by radial stays will be taken as best suited to the requirements of this case.

Referring back to the general outline of the consolidation engine of Fig. 1, it will be seen that the wheels are spaced rather near together. The object of this is to secure as short and rigid a wheel-base as possible. It is evidently out of the question to bring the wheels so near together that there is a bare clearance between the flanges, on account of the necessary attachments of the frames. It has been found, however, that from 5 inches to 7 inches between the treads will answer, so that in this engine, with wheels 57 inches in diameter, the distances between centers are taken at 64 inches, 62 inches and 64 inches respectively.

With a cylinder having a piston stroke of 26 inches the total length of the same will be about 36 inches. Allowing necessary clearance between the tread of the forward driving wheels and the cylinder casting, the distance from the center of the forward driving wheel to the center of the cylinder may be taken as the case requires. As the boiler proper usually ends at the cylinder casting, in this case it will be 42 inches ahead of the forward center. Finally in order that the boiler and engine may balance well on the wheels and not tend to put an excessive load on the rear drivers or front truck, the overhang of the firebox back of the rear drivers should be made with due consideration to the position of the wheels and distribution of the weight. On this basis the approximate length of the boiler from the back head to the front tube sheet would be about 24 feet 6 inches or 104 inches more than the wheelbase. Now comes the proper distribution of this distance into firebox and tube lengths.

The requirements of the service will demand that the firebox extend out over the wheels, so that constructional limitations will decide the total width. With a firebox tapering in at the top, a total width of about 7 feet at the mud-ring can be used. Allowing for the thickness of metal and a width of ring on each side of 3½ inches will leave about 75¼ inches, or 6¼ feet, for the width of the grate. In order to obtain 57 square feet of grate area the length should be a little more than 9 feet, and deducting 4 inches at the rear for the water leg, would leave a length of about 15 feet for the tubes.

As already stated, these dimensions are not fixed, so that in the designing of the boiler some modifications are possible, to suit the requirements and conveniences of construction. If the work were to be undertaken from the start a great many trials would have to be made on the drawing board in order to secure the proper adjustments. Without reviewing these steps one by one, it will be permissible to state that if a firebox of the width given be used, and it be made 6 feet high above the mud-ring, it will have a heating surface of about 15½ square feet per foot of length in addition to about 54 square feet for the ends, if no allowance is made for tube and door openings. Suppose, then, this firebox be made 8 feet 6 inches long inside; the heating surface will be 186 square feet, leaving 2630 to be made up in the tubes. With 0.5 square feet per lineal foot as the approximate heating surface of a 2-inch tube, this would require a total tube length of 5260 feet. As the available length is 15 feet, this would require 351 tubes.

With this as a guide, it will be found, in laying out the best form of firebox, and allowing the space needed above it for water and steam, and by keeping within the limits imposed, that 341 tubes 2 inches in diameter will be the most convenient number and that best adapted for service. They will have a length of 14 feet 6 inches over the tube-sheets and the inside diameter of the smallest ring of the shell will be 69 inches, all of which is shown in the general outline of Fig. 3. On taking this boiler and calculating the actual heating surface it will be found that there are 2570 square feet in the tubes to which the 186 square feet in the firebox is to be added, making 2756 square feet in all.

Without entering into the details of the proportioning of the parts in order to secure the proper strength, which is really the next step, but which is outside the province of this work, attention is to be turned to making an estimate of the weight of such a boiler as that outlined. It will be found that the weight of the materials in the boiler itself with its firebox and bracing complete, but without the tubes will be about 30,000 pounds to which must be added the weight of the latter or 11,000 pounds, thus bringing the total weight up to 41,000 pounds.

With the weight and dimensions thus established, it will be possible to go on and work out the other parts of the mechanism.

Turning now to the passenger locomotive, the boiler must be worked out somewhat differently. Here we are not confronted with the hauling of a very heavy load at a low speed, but with the work to be done at a comparatively higher speed with a lighter load. This will involve the working of the engine at a shorter cut-off and here again experience in the working of an engine will be called into play.

In the preliminary considerations of this matter the requirements were laid down that the engine should be capable of hauling a passenger train weighing 350 tons up a grade of 1 per cent at a speed of 25 miles an hour. The engine selected for this purpose was of the Atlantic type, as being that best adapted to the work. As in the case of the consolidation locomotive the diameter of the driving wheels must be arbitrarily determined. In this the designer is to be guided by the necessity of obtaining a reasonably good speed on lighter grades and level; and for this a diameter of 77 inches will be found to be well suited.

From Formulas (1) and (2) it will be found that the resistance of the train will be $R = \frac{25}{4} + 2 = 8.25$ pounds, and $R' = 2000 \times \frac{1}{100} = 20$ pounds, or that the total will be 28.25 pounds per ton, and $350 \times 28.25 = 9888$ pounds, or that, in round numbers, a drawbar pull of 10,000 pounds will be required to do the work.

It is evident that the varying resistances of the train do not become any the less when there is an increase of speed, but rather increase. On the other hand, the power of the engine diminishes with the increase of speed. So instead of the 11 per cent margin allowed for uncontrollable resistance, in the case of the freight locomotive, an allowance of at least 15 per cent must be made for the passenger locomotive. By allowing a 15 per cent increase of resistance and a similar loss of power in connection with the needed drawbar pull we have

$$\frac{10{,}000 \times 0.15}{1 - 0.15} + 10{,}000 = 1765 + 10{,}000 = 11{,}765 \text{ pounds.}$$

Further, the internal friction of the engine is not reduced in proportion to the fall in the mean effective pressure in the cylinders. So instead of taking this at 10 per cent, as in the case of the consolidation, it has been found by tests and experience that an allowance of 18 per cent of the theoretical tractive power must be made. Hence the minimum tractive power needed would be $11{,}765 \times 1.18 = 13{,}882.7$ pounds, or in round numbers, 13,900 pounds.

For the determination of the piston speed, to be used in calculating the proper diameter, the following formula may be used:

$$S = \frac{12 \times 5280\, v \times 2\, p}{3.1416\, D \times 60 \times 12} \qquad (7)$$

in which

S = the piston speed in feet per minute,
v = speed of the train in miles per hour,
D = diameter of the driving wheels in inches,
p = the stroke of the piston in inches.

In order to fill in the required factors in this case, it is necessary to assume a piston stroke that will be well adapted for use with the wheel diameters that have been given and the service to be performed. This will be placed at 26 inches. Then by substitution Formula (7) becomes:

$$S = \frac{12 \times 5280 \times 25 \times 2 \times 26}{3.1416 \times 77 \times 60 \times 12} = 473 \text{ feet per minute.}$$

Indicator diagrams have shown that at a piston speed of 500 feet per minute the mean effective pressure in the cylinder will be about 62 per cent of the boiler pressure.

By substituting the values which have now been obtained in Formula (3) it is possible to obtain the diameter of the cylinder. This is then:

$$d = \sqrt{\frac{13{,}900 \times 77}{200 \times 0.62 \times 26}} = 18.5 \text{ inches, nearly.}$$

In order that there may be an ample margin of power the cylinder diameter, in this case, will be increased to 19½ inches, as the weight allowed will make it possible to use a boiler of sufficient capacity to supply such a cylinder.

The boiler dimensions may now be determined in the same way as for the freight engine by the substitution of the several values in Formula (4), for the calculation of the equivalent amount of water to be evaporated per minute. This then becomes:

$W = 2 \times 9 \times 0.55 \times 110 \times 0.30 \times 1.24 \times 1.25 = 506.4$, or 500 pounds per minute, or 30,000 pounds per hour.

Taking the same quality of coal as before, and allowing 1½ pound of coal per square foot of heating surface per hour, the latter, according to Formula (5) should be:

$$\frac{30{,}000}{7.5 \times 1.5} = 2666 \text{ square feet,}$$

and from Formula (6), with a rate of combustion of 85 pounds per square foot per hour of grate area, the latter will be:

$$G = \frac{30{,}000}{7.5 \times 85} = 47 \text{ square feet.}$$

Proceeding as before, it will be found that the most convenient dimensions for tubes and surfaces will give a boiler with 2767 square feet of heating surface in the tubes and 162 square feet in the firebox, making a total of 2929 square feet and a grate area of 46.3 square feet.

This is somewhat in excess of that called for by the formula in the way of heating surface, but it will be found to be of great advantage in a passenger engine and can only be obtained in this type of locomotive, though there is some objection to the extra weight on the forward truck and trailing wheels.

The length of the firebox will be 102 inches by 65½ inches wide. There will be 332 tubes of 2 inches diameter, 16 feet long, and the diameter of the smallest ring of the shell will be 67 inches. Finally, the weight of the boiler will be 39,300 pounds.

In the matter of the evaporative qualities of the various grades of coal and the consumption per hour, the following table is presented as deduced from a report made to the American Railway Master Mechanics' Association in 1897:

	Water Evaporated per pound of Coal	Coal to be Burned per sq. ft. of Grate Area per hour
Large Pennsylvania anthracite............	8 lbs.	60 lbs.
Fine Pennsylvania anthracite.............	6½ lbs.	35 lbs.
Virginia semi-bituminous	9 lbs.	65 lbs.
Illinois bituminous	7 lbs.	90 lbs.

Finally the following ratios were suggested:

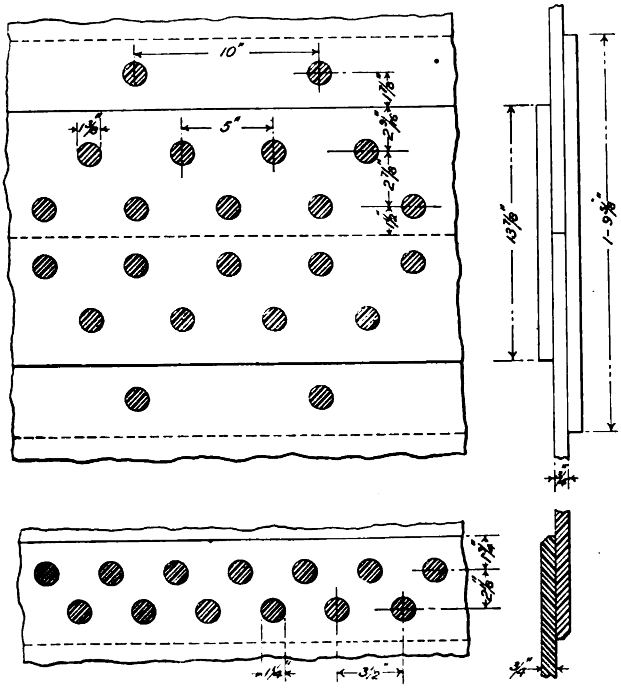
	Cyl. Volume in cubic feet to grate Area in square feet	Cyl. Volume in cubic feet to Heating Surface in square feet	Heating Surface to Grate Area
Large anthracite	1:4	1:180	40:1
Small anthracite	1:9	1:200	20:1
Bituminous	1:3	1:200	60:1

In the work that has preceded, the various steps leading up to the determination of the general dimensions and proportions of the boilers have been indicated. With this accomplished there still remains a great deal of work to be done in laying out the details of the several parts of the boiler itself. While it will be impossible to enter into a discussion of boiler construction in detail the importance of this part of the locomotive is such that some attention should be paid to it here.

It has already been remarked that the boiler is the life of the locomotive, and that upon it depends the efficiency of the whole machine. Hence it is of the utmost importance that the greatest care should be exercised in its design and construction to make sure that it is possessed of the requisite strength and power of endurance.

In this, attention is first turned to the shell whose strength depends upon the thickness of the plate of which it is formed and the type of longitudinal seam used to connect the edges of the same. In regard to the latter, that one should be selected that will give the highest percentages of strength, as compared with that of the solid plate, consistent with practicability of construction and maintenance in service.

Without entering into a discussion of the relative merits of the various types of joints it may be stated that the sextuple riveted, double butt strap joint with welts inside and outside the main plates as shown in Fig. 7, is best adapted for this work and will, therefore, be used. In this joint there are three rows of rivets on each side of the joint, all of which pass through the inner strap, and but two through the outer. For an analysis of the strength of such a joint, the reader is referred to the various handbooks on boiler construction, where it will be found that the calculated strength of such a seam is a little more than 86 per cent of that of the solid plate and that 85



Figs. 7 and 8. Sextuple Riveted Butt Strap Seam, and Double Riveted Lap Seam

per cent can be counted upon in regular working practice and construction.

With this as a preliminary basis, the strength and thickness of the shell can be calculated from the following formula:

$$L = \frac{S\,C}{f} \tag{8}$$

in which

$L =$ the working stress that can be put upon the plate per square inch of section,

$S =$ the ultimate tensile strength of the steel in pounds per square inch,

C = the percentage of strength of the seams of the plate,
f = the factor of safety that it is necessary to allow.

The specifications for boiler steel of the American Railway Master Mechanics' Association state that "the desired tensile strength is 60,000 pounds per square inch, with minimum and maximum limits of 55,000 and 65,000 pounds." As, in the design it will be necessary to work to the minimum, 55,000 pounds per square inch will be taken as the tensile strength. Practice has further shown that a factor of safety of 5 is well adapted to working conditions. Hence by the substitution of the values obtained in the second term of the equation we have

$$L = \frac{55{,}000}{5} = 11{,}000 \text{ pounds}$$

per square inch when referred to the solid portion of the plate, although when referred to the efficiency area of the seam this is

$$L = \frac{55{,}000 \times 0.85}{5} = 9350 \text{ pounds.}$$

The actual thickness of the shell plate is calculated by the formula

$$T = \frac{D' P}{2 L} \qquad (9)$$

in which

T = the thickness of the plate in inches,
P = the working pressure,
D' = the diameter of the boiler in inches.

The coefficient 2 is in the denominator to allow for the stress carried by two sheets, at opposite ends of the diameter. Referring to the boiler intended for the consolidation engine, Formula (9) becomes, by substitution,

$$T = \frac{69 \times 200}{2 \times 9350} = 0.738.$$

We therefore take ¾ inch as the proper thickness of the smallest ring of the boiler shell.

In the case of the boiler for the Atlantic type engine, the inside diameter is 67 inches and Formula (9) becomes

$$T = \frac{67 \times 200}{2 \times 9350} = 0.716.$$

Owing to the liberal factor of safety that has been adopted, it will be allowable to take the sheet in nearest sixteenths which will give 11/16 inch as the thickness of the smallest ring.

In these calculations the rings of the sheet have been considered as though they were integral and unbroken. This is true of the front sheet but in the second, the opening for the dome weakens it to such an extent that it is generally made about 1/16 inch thicker than that calculated. Hence, in the case of the consolidation locomotive this sheet should be 13/16 inch thick, while for the Atlantic ¾ inch may be used, although this falls a little short of the extra 1/16 inch called for.

The dome opening should be made as small as possible, besides being thoroughly braced by a heavy dome base and an inside liner around it, as clearly shown in Fig. 4 of the passenger engine boiler. This opening should be limited to that actually needed for the entrance and adjustment of the standpipe, and for its use as a manhole—a rule that applies equally well to the diameter of the dome. The idea that the dome can serve any useful purpose as a storage reservoir for steam has long since been discarded as it is merely a means of elevating the throttle above the water and thus securing dry steam for the cylinders.

Although the neutral part of the shell within the lines of rivets holding the dome in place should not be taken into consideration in calculating the strength of the former, it nevertheless does serve a very useful purpose, and materially adds to the strength of this part, in that the flexible edge around the opening transfers the stress from the edge to a line inside the inner row of rivets of the dome base, and thus to the solid portion of the shell, and thereby lessens the tearing effect that would exist if the opening were cut out so as to leave only the usual margin inside the rivets.

In the designing of the circumferential seams, the steam pressure may be disregarded, and the work done with consideration only to tightness and structural strength. A brief consideration of the stresses to which the boiler is subjected will show that, when the tubes are in position they, together with the bracing of the front and back heads, so relieve the longitudinal seams from the stresses due to seam pressure alone that the latter becomes a negligible quantity. For that reason a double riveted lap seam is used, which, while it has perhaps less than 70 per cent the strength of the solid plate, is ample for its work. Such a seam is shown in Fig. 8.

Turning now to the firebox, the inside sheet should be made thin so as to offer the minimum resistance to the transmission of the heat of the fire to the water beyond. The proper spacing of the staybolts will, of course, make it possible to use almost any thickness of sheet; hence the choice must be made as the result of experience rather than from any mathematical calculations. This experience has shown that when the steam pressure ranges from 180 pounds to 200 pounds per square inch, the best results can be obtained with side sheets ⅜ inch thick. This does not hold for the tube sheets, where, on account of the necessity of expanding and fixing the tubes, the thickness should never be less than ½ inch and the use of ⅝ inch will frequently be found advisable. So, basing the choice on this general principle, the thickness of the metal in the side and tube sheets of the two boilers will be taken at ⅜ inch and ½ inch respectively, the crown-sheets being given the same thickness as the sides.

The general practice at the present time for supporting the crown-sheets is by means of radial stays, while the flat surfaces are held by the ordinary staybolts. In some cases the latter are replaced at the upper forward portions of the firebox, where there is the greatest difference between the expansion of the inner and outer sheets, by flexible staybolts of various designs. Owing to this difference in the expansion of the two sheets, and the bending of the staybolts resulting therefrom, it is desirable that the latter should be made of a comparatively small diameter. In practice, on boilers carrying a high steam pressure, the diameters range from ⅞ inch to 1⅛ inch as a maximum, and they are usually spaced so that the stress put upon them does not exceed 5500 pounds per square inch of section. The spacing runs from 3¾ to 4½ inches from center to center of the bolts. If, then, 4 inches is adopted in the boilers under consideration, the total stress on the 16 square inches held by each bolt will be 3200 pounds, which will call for a staybolt ⅞ inch diameter and load it to about 5330 pounds per square inch of section.

To provide for the injurious stresses imparted on the flue-sheet and shell by the expansion of the former in advance of the latter in raising steam in the boiler, sling stays of a telescopic nature are applied over the forward part of the firebox so that the latter is allowed to rise slightly until the shell of the boiler is heated by the water and steam, which then expands and gradually takes up the slack thus formed and brings the stays under tension as the temperature and pressure increases.

The width and shape of the water legs has a most important bearing on the efficiency and durability of the boiler. These two points should be so related to each other that there is room for the inflowing current of water and the escaping steam that is generated in contact with the inner sheet. Carefully conducted experiments have shown that, in some cases, where the sheet is vertical there is hardly any water in contact with the upper portions, but that they are covered with a layer of steam bubbles ascending to the surface. Such a condition has the double disadvantage of lowering the rate of evaporation and leading to an overheating of the sheets. In order to avoid this, the sheet should be so sloped in that the steam, in rising vertically, tends to leave it and give free access to the water. It will be noticed from the cross-sections in Figs. 5 and 6 that this has been done in both boilers.

Again, owing to the fact of there being more steam at the upper portion of the leg than the lower, it should be widened at the top, and this has also been done in both cases. At the mud-ring the width used is 3½ inches while at the top 6 inches is required. Such a space will be found to work well in practice, and is founded on sound reasoning rather than on mathematical calculations. There are many other points of more or less importance that could be referred to, but available space requires that the discussion should be limited, in the main, to general principles. Finally a few recommendations should always be borne in mind, and the work designed in accordance therewith. Among these, one of the first importance is that flat surfaces should be avoided as far as possible, and where they cannot be done away with entirely, they should be reduced to the lowest dimensions, so as to cut down staybolt stresses to a minimum. Wherever flanging or bending of the sheets is required, it should be done with a liberal radius of curvature, so as to secure flexibility and avoid undue internal stress of the material. In the application of the bracing great care should be exercised that it gives strength without adding too much to the rigidity. A boiler is subjected to such varying temperatures in its different parts that it must expand and contract differently; hence it is of the utmost importance that it should be capable of this internal movement of its parts relatively to each other,

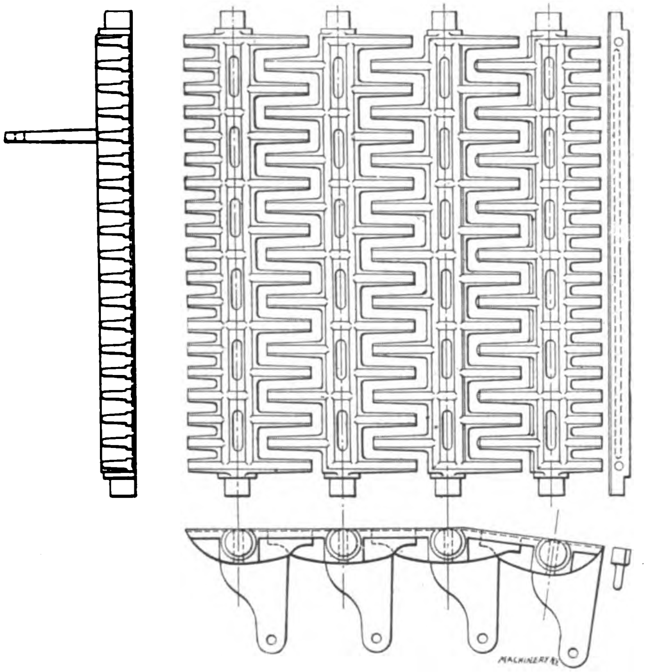


Fig. 9. Grate for Bituminous Coal

without putting an undue stress upon the metal of which they are composed.

A few words may be added regarding the actual work of constructing the boiler, for this, while not strictly belonging to the work of the designer, should nevertheless, be borne in mind by him, and may well form a part of his specifications. Without taking up the matter in all its detail, a few points will be touched upon.

Next to the formation of a tight joint at the sheets, it is of the utmost importance that the tubes should be well and efficiently set. To do this it is advisable to make the holes in the front tube-sheet 1/16 inch larger than the diameter of the tubes, while those at the firebox end should be of the exact size. Both edges of the holes should be chamfered to a radius of 1/16 inch. Further, a copper ferrule should be used at the firebox end, and this should have the same outside diameter as the tubes, a thickness of about 1/16 inch and a length ¼ inch greater than the thickness of the sheet. The ferrules should be lightly rolled and expanded in place before the tubes are inserted, while these should be swaged down so that they will just enter the ferrules. At the front end, on the other hand, they should be expanded and rolled out to fill the holes, care being taken that the ends are annealed before they are put in place. As to length, the tubes should be cut so that they will project from ⅛ to 3/16 inch beyond the sheets at each end. A satisfactory system of tube-setting is to turn back about 10 per cent of the tubes at each end to be beaded over in order to act as stay tubes, and then to round up the edges of all the tubes with a mandrel. Of course it goes without saying that the rolling of the tubes should be carefully done, and if necessary, should be repeated at the firebox end after the boiler has been fired and tested.

Turning back to summarize the stresses that have been given as allowable for staybolts and other related and similar parts, experience has shown that for those in the side, front and back water legs the load should not exceed 5500 pounds per square inch of section. For radial stays they are longer and are not subjected to such excessive bending stresses, due to the variation in the expansion of the two plates that they connect, the load may be raised to 8000 pounds per square inch, while on the other bracing on which there is little or no bending stress, a working load of 10,000 pounds per square inch may be imposed.

In the firebox the common American practice on all engines burning bituminous coal is to use the finger-grate type of bar having openings between each bar of from ¾ inch to ⅞ inch in width. Such a grate is shown in Fig. 9. As for the brick arch it may be considered to be practically out of use on all wide firebox engines like those which we are now considering in detail. For the narrow firebox type of boiler, however, the brick arch offers a very important advantage. The reason for this variation of practice in the two types is due to the fact that the movement of gases in the large box is slower and combustion more perfect than it would be in the narrow construction, if this latter be without the assistance of the arch for mingling and maintaining a high temperature of the gases.

CHAPTER III

THE CYLINDERS*

Closely allied with the boiler in importance are the cylinders. As the boiler is the important element in converting the potential energy of the fuel into that of the steam, so the cylinders serve as the means of converting this potential into dynamic energy and thus produce the useful work for which the machine, as a whole, is designed. The matter of the size of the cylinders has already been considered, in the determination of the general dimensions of the engine, where it was found that a diameter of 21 inches, and a piston stroke of 26 inches would be suited to the work that it is intended that the freight or consolidation engine should perform. At the same time the diameter of the cylinder and stroke of the piston of the passenger engine were calculated to be 19½ inches and 26 inches respectively. The cylinders are invariably made of cast iron, and it is of great importance that the metal should be of a character suitable for the work that it has to perform. It should be of a fine grain and as hard as can be worked, the latter quality being needed in order that it may withstand the hard wear of the pistons and valves. A common form of specification is to require that the "cylinders shall be made of a hard, compact, tough iron, of not less than 25,000 pounds tensile strength per square inch, and so cored as to produce uniform shrinkage in cooling." A metal that will meet these requirements can be made from 50 per cent pig, 25 per cent of old carwheels, and 25 per cent of high grade machinery scrap.

The practice of the several builders varied somewhat in the past in the matter of the form of the cylinder and saddle. At one time the saddle was made a separate casting, with the cylinders bolted on outside the frames. In this case the steam and exhaust connections to the smokebox were made either direct from the cylinder casting or from the side of the steam chest. This practice was succeeded by the almost universal adoption of the cylinder and half saddle cast in one piece as shown in Figs. 10 and 11. It will be observed in the two cylinders here illustrated, that Fig. 10 is adapted to the use of a flat slide valve, while Fig. 11 is fitted with a bore for a piston valve, the former being intended for use on a consolidation freight locomotive and the latter on the Atlantic passenger engine. The reasons for this variation in practice will be explained later. The length of the cylinder is dependent upon four factors: the stroke, the thickness of the piston, the clearances at the end, and the amount of counterbore allowed for the inset of the cylinder heads. The length of the working barrel of the cylinder is usually made equal to the stroke of the piston plus the width over the piston packing rings less ¼ inch. This last

* MACHINERY, Railway Edition, January, 1905.

subtraction is made so that the piston rings will travel beyond the edge of the counterbore at each end by ⅛ inch and thus prevent the formation of shoulders at the end of the stroke due to the wear of the cylinder barrel. This, of course, involves the determination of the widths of the packing rings, their number and the spaces between them, but this will be referred to later, it being sufficient to state

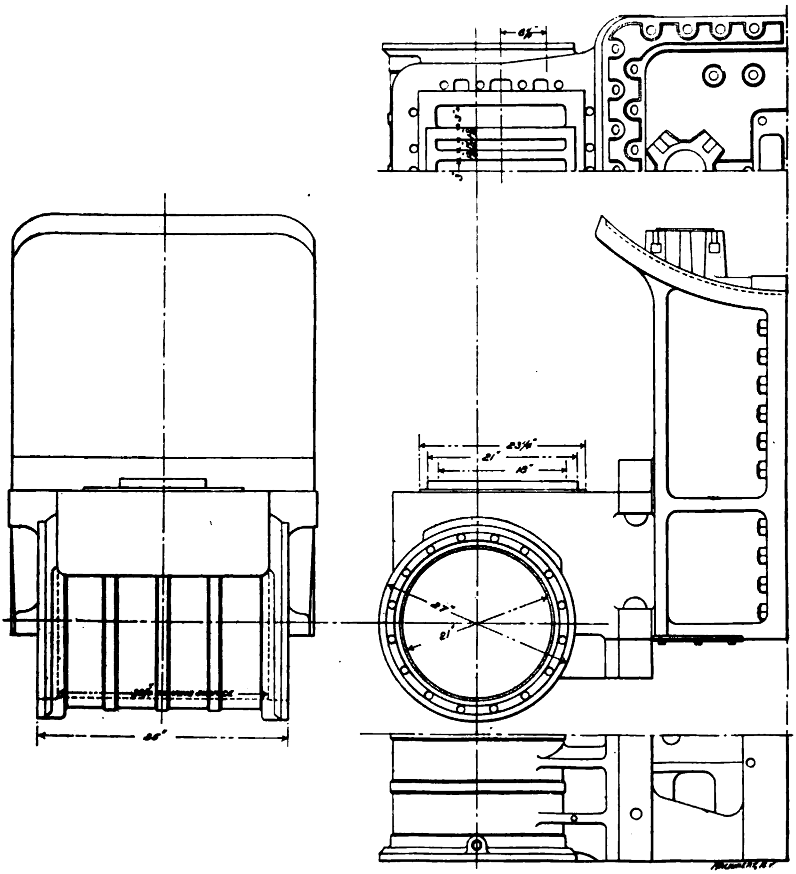


Fig. 10. Cylinder and Half Saddle for Consolidation Locomotive

here that the width of the pistons over the packing rings, in the case of those used in the two cylinders here shown is 3½ inches, which, with the stroke of 26 inches and the subtraction of ¼ inch, will make the bore of the cylinder 29¼ inches long in each case. The inner edge of the ports should come nearly opposite the outer edge of the packing rings at the end of the stroke, which places them 29¾ inches apart. The ends of the cylinders outside the working barrel are usually bored from ½ inch to 1 inch larger in diameter than the barrel itself to allow for wear and reboring without interfering with the fit of the cylinder heads. Sometimes, as in the case of Fig. 11, the cylinder is bored through from end to end to the full diameter of the counterbore and a bushing is inserted which can be renewed when it is worn, without disturbing or changing the dimensions of the pistons and packings.

As already indicated, the steam ports and cylinder heads enter the counterbore at the ends which usually extend from 4 to 5 inches beyond the actual stroke of the piston. In Fig. 10, this extension is 4½ inches, making the total length of the cylinder 35 inches, while in Fig. 11 it is 5 inches, making that cylinder 36 inches long. Subtracting from this distance the total thickness of the pistons (5¼ inches for the freight and 5½ inches for the passenger locomotive), we will have 3¾ inches and 4½ inches respectively for the clearances and the counterbore for the heads. It is desirable that the clearances should be as small as possible, though it is necessary that they be large enough to avoid all possibility of the pistons striking the heads. This will frequently be sufficient to take care of the port and steam requirements when the possible variations of motion due to faulty workmanship, wear, and changes effected by the keying of the rods, are guarded against. At times, however, this is not the case, since it has frequently been found to be desirable to give a high-speed engine a somewhat greater clearance than a slow one because of the longer period during which the steam is worked expansively, followed by an earlier compression. The cylinder heads should be so designed and proportioned that they will extend into the counterbore of the cylinder sufficiently for the clearance left between their inner faces and the piston, when the latter is at the extreme end of the stroke to be not less than ¼ inch nor more than ⅜ inch. This clearance should be so divided that when the new engine is turned out of the shop there will be from 1/16 inch to ⅛ inch more clearance allowed at that end of the cylinder toward which the wear of the rod brasses has a tendency to draw the piston, than at the opposite end. As to whether this will be the front or the back will depend on the details of the ends of the main rods.

The design of the cylinders is so closely allied to that of the valve motion, and so much depends upon the proper distribution and flow of the steam, that the size and arrangement of the ports will be considered in connection with the valves. In the actual construction of the cylinder, however, the ports as they enter the barrel are usually about 2 inches shorter than the diameter of the bore, increasing somewhat in length at the valve in flat slide valve engines, so that they are a trifle longer or about the same length as the nominal diameter of the cylinder. In the case of the piston valve the length of the port as it enters the cylinders is of about the same proportion as before, but is narrowed as it approaches the steam chest on account of the diameter of the latter being less than that of the cylinder. Here the steam ports extend entirely around the steam chest, being interrupted at a few points for the insertion of bridges to carry the packing rings of the valve over the openings. In laying out the

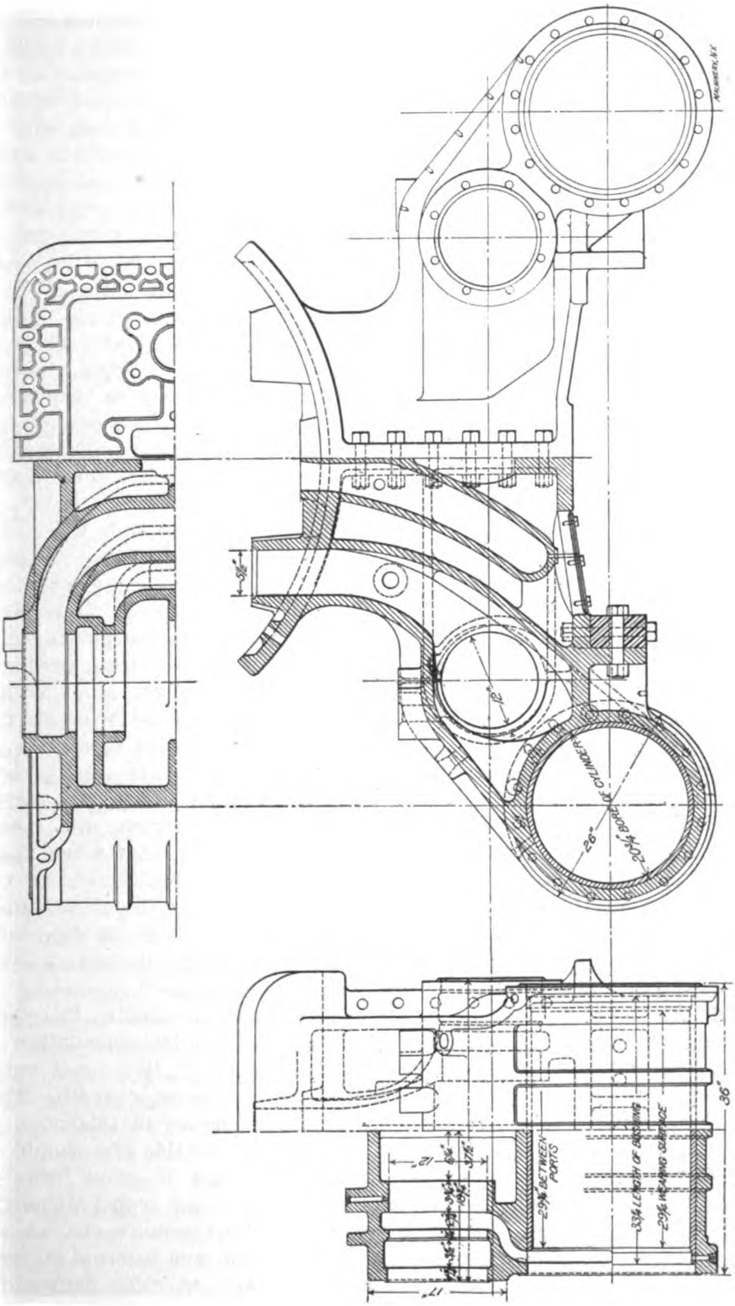
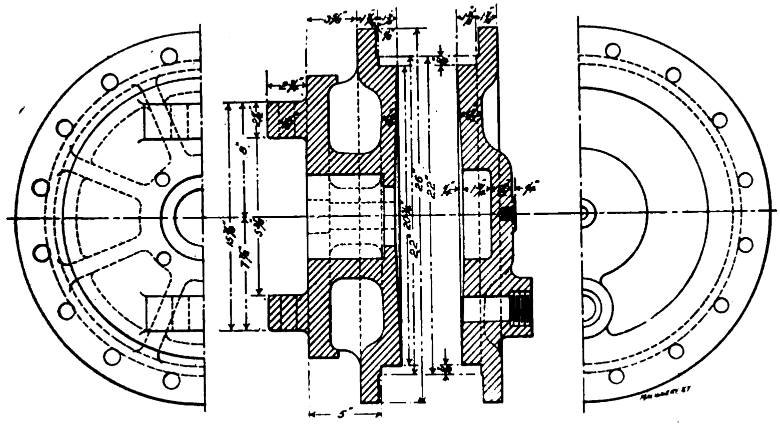


Fig. 11. Cylinders and Saddle of Atlantic Type Locomotive

course of the ports care should be taken that they are of a uniform section, as short and direct as possible, well rounded on all curves and turns, and with the turns of as long a radius as possible. These are matters of the utmost importance, since it is essential, for an efficient action of the machine, that the flow of the steam from the valve to the cylinder should be free and unimpeded, and devoid of eddies and cross currents which will tend to reduce the speed of such flow. This is especially important in the case of high-speed engines where any checking of movement of the steam will have a very marked effect on the work than can be done. Where the ports widen out in the saddle to meet the steam and exhaust pipes, they should be clean and roomy, and this is especially true of the exhaust ports. It has been found to be fully as difficult, if not more so, to get rid of the steam after it has done its work in a high-speed locomotive as it is to get it into the cylinder in the first place; hence the exhaust passages should be designed with ample spaces and easy curves that there may be the minimum amount of resistance set up to the escape of the slow-pressure steam found in the cylinder at the end of the stroke. In short, every effort should be made to reduce back pressure to the lowest point.

The steam passages should be protected in every possible way from radiation. Under no circumstances should they be allowed to pass along the outer walls of the saddle, but should always be protected by an air space insulating them from the cold outer air. It is well, too, to add to this the further protection of some non-conducting material. Not only should these air spaces separate the steam passages from the external walls, but they should be so arranged as to isolate them, whenever practicable, from the exhaust passages wherein the temperature is much below that of the steam at boiler pressure. In fact, the end and aim should be to deliver as many heat units as possible to the cylinder, for it is upon this that the efficiency of the engine largely depends. In addition to this care by insulation, the steam passages should be laid out in an elastic curve, so that when their walls are heated and cooled by the admission and withdrawal of the steam, the expansion of the metal may not be such as to put any additional stress upon the other parts of the casting. This is especially necessary at that portion of the saddle near the frame fits, which must be ribbed in order to withstand the working stresses.

As for the thicknesses of metal to be used in the saddle, there are no reliable formulas available whereby a mathematical calculation of this can be made. Experience has shown, however, that too great a thickness is detrimental to the securing of the greatest strength. This is particularly true where there is an abrupt change in thickness, as the casting is apt to crack at such a point. All outside ribs should be avoided, since they have the double disadvantage of being liable to fracture and serving as a good radiating medium for such heat as the saddle necessarily takes up from the steam. Furthermore, the corners should be well rounded so that both the external and internal stresses may be evenly distributed over the large surfaces. The flanges for bolting the cylinders and half-saddles together and to the boiler must be made heavy and be ribbed so as to withstand the pressure and driving of the bolts. Here, too, the metal should be arranged so as to join the walls with curves of long radius and large fillets. The walls should be well braced internally with ribs so as not to yield at the root of the flanges. No formula is available for calculating the thickness of these ribs since no one knows exactly, or even approximately, the stresses to which they are subjected. They are ordinarily made from 2½ inches to 3 inches thick, which, in the cases before us, is taken at 2¾ inches.



Figs. 12 and 13. Back and Front Cylinder Heads for Atlantic Type Locomotive

About the only point in the design of the cylinder that is subjected to a mathematical analysis is the thickness of the shell of the barrel, which may be calculated from the formula

$$t = \frac{dp}{2S} \qquad (10)$$

in which

t = thickness of the shell in inches,
d = diameter of the bore in inches,
p = steam pressure in pounds per square inch,
S = safe stress in metal in pounds per square inch.

Allowing 2500 pounds per square inch as a safe working stress for the metal, and substituting the values that have been already found for the two engines, the formula becomes

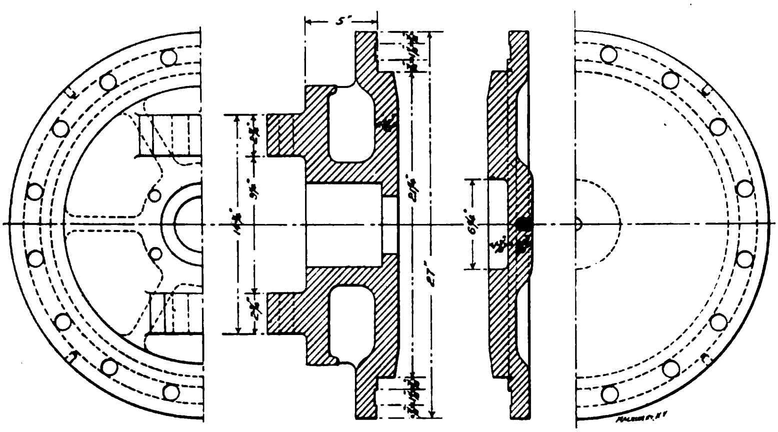
$$t = \frac{21 \times 200}{2 \times 2500} = 0.84 \text{ inch}$$

for the consolidation engine, and

$$t = \frac{19.5 \times 200}{2 \times 2500} = 0.78 \text{ inch}$$

for the Atlantic type. Unfortunately the resistance to the pressure of the steam is not all that is required of a cylinder. It is subjected to wear, and provision must be made for reboring, besides which it must sustain many of the shocks to which the locomotive is subjected while in motion. Hence, while the calculated thicknesses are a trifle less than ⅞ inch it will be found advisable to increase this by 50 per cent or more and make the shells from 1¼ to 1½ inch thick.

The cylinder heads are usually simple in construction and are bolted in the flanges cast at the ends of the cylinder by studs spaced about 5 inches apart from center to center. In the case of the front head these studs have no work to perform other than to withstand the pressure of the steam against the head. At the back they have not only this, but the indeterminate stresses due to the support of



Figs. 14 and 15. Back and Front Cylinder Heads for Consolidation Freight Locomotive

the front end of the guides and the varying loads that they transmit to the heads. These studs are usually made 1 inch in diameter for the cylinder sizes that are here considered, so that with sixteen studs to withstand the pressure in a counterbore 21¼ inches in diameter, the load on each stud would amount to less than 5700 pounds per square inch of section of the metal, hence there is an ample margin beyond for safety. The thickness of metal in the heads must be taken independently of any stress that may be put on them by the steam pressures. The guides must be supported, and there must be provision on the back head for the packing box of the piston rod. Besides this the heads are exposed to external shocks and blows, so that while far in excess of the requirements for resisting the steam pressures, the heads can well be made from 1⅛ to 1½ inch thick. As the heads are especially exposed to the action of the wind, they must be well protected against radiation of heat. At the front, a plain disk head is generally used which is protected by a casing, enclosing an air space in front of the flat surface. Sometimes this space is filled with a non-conducting material that serves to considerably lessen the radiation. At the back, by the use of a two-bar guide and the flange of the packing box, the construction lends itself very readily to the formation of an air space and pockets for insulating material in the head itself. This is clearly shown in Figs. 12 and 14 of the back heads for the two engines.

Let us briefly review the work to be done on the cylinder and heads. The seat for the smokebox is cast with chipping strips that are chipped to the proper radius, while the two parts (smokebox and saddle) are held together by bolts. The connection between the steam passage and the steam pipe in the smokebox is made by means of a ground joint. The abutting surfaces of the two half-saddles are accurately planed and are held together by bolts that are usually about 1¼ inches in diameter. In common practice these bolts are tapered about 1/16 inch to the foot and they are driven home. The surfaces for the bearing of the frames are planed and the frames and cylinders bolted together with tapered bolts turned to a driving fit. The details of these fastenings will be considered under the subject of the frames. The bore and valve faces of the cylinder are finished with a smooth-cutting tool and are left as they come from the machine. As the joint between the heads and the cylinder ends must be steam-tight, the two surfaces in contact are ground so that no packing is required. Of course in all this work the utmost care must be exercised that a true alignment of every surface is obtained, for without that not only would the working of the moving parts be defective, but the stresses set up would be excessive and the liability of the fixed parts to work loose be greatly increased.

CHAPTER IV

THROTTLE VALVE—DRY AND STEAM PIPES*

With the four principal items in the construction of the engine decided, namely: the weight, wheel arrangement, boiler, and cylinders, the following steps will be that of the working out of the various details upon which the successful operation of the locomotive depends. Closely connected with the work done in the cylinders and boilers are the means by which the steam generated in the one can be conducted to the other so as to perform its proper functions. In this it is of the utmost importance that the flow of the steam from the boiler to the cylinder should be free and unimpeded and that there should be the minimum drop in pressure when it reaches the steam chest. There must necessarily be some drop, else there would be no flow, but this will decrease as the size of the passages is increased.

The proper sizes of the throttle and the dry-pipe are determined, like so many other things in locomotive practice, not so much by a theoretical calculation as by an empirical formula deduced from practice, that has been shown to give satisfactory results. As already stated, the flow of the steam must be free enough to maintain a high pressure in the steam chest, and yet the limits of space in the boiler and smokebox cut down the available dimensions of the pipes to a low figure. In this, too, the designing of a locomotive differs from that of a stationary engine in that the latter is built to run at a constant speed, and with a comparatively uniform degree of admission in the cylinder, so that the steam pipe can be proportioned accordingly and be made to deliver a constant quantity of steam at a uniform rate of flow. In the locomotive, on the other hand, the steam consumption varies between wide limits and is greatest at high speeds when the earliest cut-off is used.

It has been found, then, that the sectional area of the throttle and dry pipe should not be less than one-fifteenth (1/15) that of both cylinders, while the steam pipes in the smokebox should not be less than one-twelfth (1/12) the sectional area of one cylinder. The reason for this apparent discrepancy is that when the engine is working slowly, an area of one-fifteenth can deliver all of the steam required; while, when it is running at a high speed, the point of cut-off is moved back until it never occurs later than at half stroke, so that only one cylinder is taking steam at a time. This makes the dry pipe proportionately larger or raises it to two-fifteenths of the area that it is obliged to supply.

Referring these proportions to the two engines that we have in hand, it will be seen that in a consolidation freight locomotive having wheels 57 inches diameter, and a piston stroke of 26 inches, the maxi-

* MACHINERY, Railway Edition, March, 1905.

mum velocity of the piston with the engine running at a speed of ten miles an hour will be about 6 2/3 feet per second, so that at times the flow of steam through the dry pipe would be fifteen times this, or about 100 feet per second. Were the engine to be running at the rate of thirty-five miles an hour the velocity of flow would only be increased to 175 feet per second, because under these circumstances only one cylinder would be taking steam at a time. In the case of the Atlantic passenger locomotive, with 77-inch drivers and a 26-inch

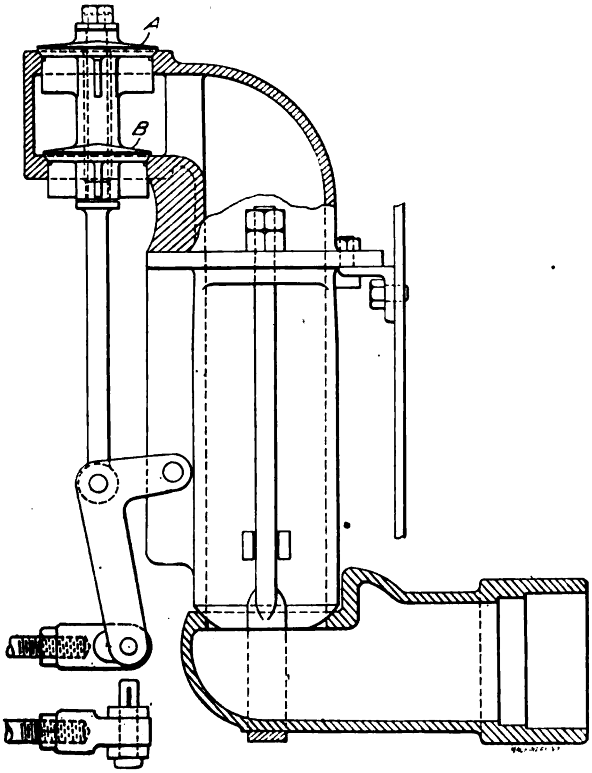


Fig. 16. Section of Ordinary Type of Throttle Valve

stroke of piston, the velocity of flow through the dry pipe would be, at speeds of fifteen and sixty miles an hour, 111 feet and 223 feet per second respectively.

Such a velocity does not obtain in practice, however, even though the proportions given are maintained, since the steam contained in the steam chest and pipes expands somewhat during admission and is later replenished during expansion and compression; the result is a practical uniformity of velocity of flow through the throttle valve and pipes. Consequently the actual speed is much below that given by the figures above and does not increase in the direct proportion of the piston speed, as might be expected.

As for the types of throttle valve, dry pipe and steam pipes that are to be used, there is little variation in current practice. The double-seated balanced type of poppet valve is universally used with some slight variations in the details of its construction.

The ordinary throttle valve is shown in section in Fig. 16. In this the two valves, *A* and *B*, close from the top and when raised admit steam to the throttle casting at the top and bottom. In order to assemble this valve it is necessary that the upper should be the larger

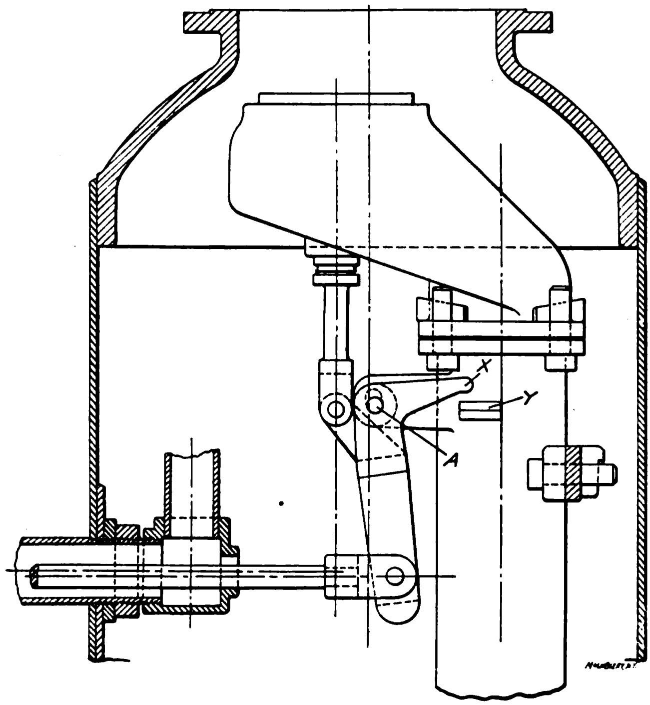


Fig. 17. Section of Throttle Valve with Variable Fulcrum

of the two, so that the lower may be put in from the top. This destroys the perfect balance, as the steam, acting upon the greater area of the upper disk, tends to close the valve and will overcome that against the bottom face of the lower disk. This prevents the valve from opening accidentally. The casting may be supported from a bracket bolted to the inside of the dome. As for the inside diameter of this casting, the proportions given above make it a little more than 7½ inches for the consolidation locomotive with 21-inch cylinders, and a little less than 7⅛ inches for the Atlantic type with 19½-inch cylinders. Hence a casting with 7¼ inches diameter of opening can well be adopted for the two. As a matter of fact these dimensions will be found to vary slightly in practice to meet the exigencies of other requirements that may come up in the working out of the details.

In this connection attention may be called to the fact that the departure from an exact balance of the throttle valve due to the variation in the diameters of the two parts, may cause a considerable preponderance of closing load, especially when high steam pressures are used. This frequently makes it somewhat difficult to open the throttle. In order to avoid this, a system of levers has been designed like that shown in Fig. 17. In this the bell-crank has a double fulcrum. When closed and about to be opened, the bearing is on a pin, *A*. This gives a long leverage and a powerful purchase to assist in the opening of the valve. As the bell-crank turns, the arm *X* comes in contact with the

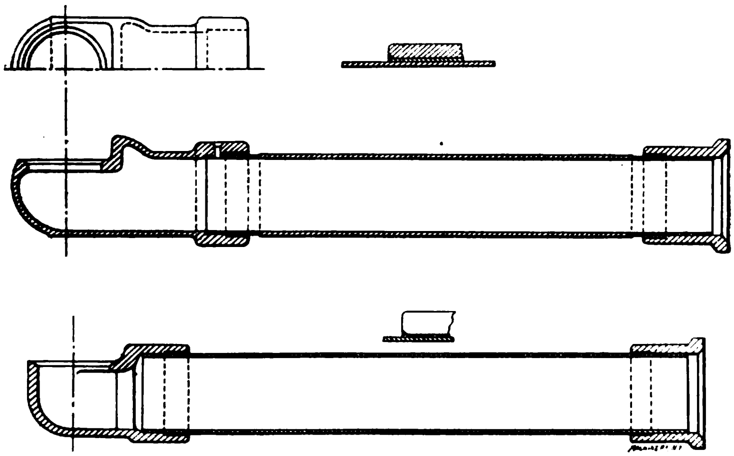


Fig. 18. Dry Pipes for Locomotives

lug, *Y*, after the valve has started and then serves, by the shorter leverage given, to cause the valve to move more rapidly for the same amount of movement of the throttle stem, the pin *A* rising from its bearing in the slot in the main casting.

The dry pipe is usually made of wrought-iron pipe with cast-iron ends riveted on. The dimensions called for will be the same as those of the throttle casting. But as wrought-iron pipe of these approximate dimensions rises by inches in diameter there is nothing available for the purpose between the nominal diameters of 7 inches and 8 inches. It will, therefore, be necessary to use the former as being nearest to the estimated size, though slightly smaller. A common form of dry pipe is shown in Fig. 18.

It is evident that such a pipe as this cannot be put in through the dome, so it must be run into the hole in the front tube sheet which is, therefore, made large enough to pass the castings riveted to the ends. For a fastening and joint, the arrangement shown in Fig. 19 is used. The dry pipe, *A*, with its castings, is put through the hole in the tube sheet, a heavy reinforcing ring, *E*, having first been fastened against the inside of the sheet in order to increase its stiffness. The hole in the sheet itself is beveled to match the bevel on the casting. The latter is cut with a spherical ground joint on the outside to take the brass ring, *B*, which is also ground with a flat outer face to bear against the tee-head, *C*. The latter has flanges cast upon either side to which corresponding flanges of the steam pipes, *D*, are bolted. All of these joints are ground and fitted as no soft packing would be able to withstand the intense heat of the smokebox to which they are subjected.

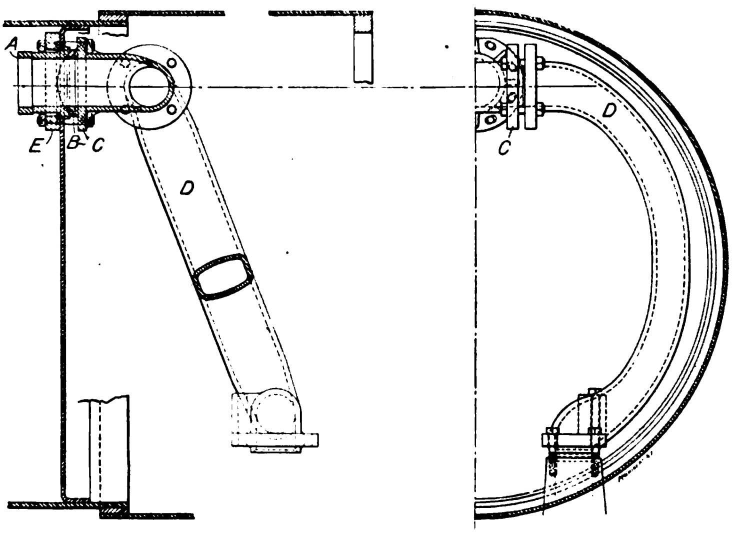


Fig. 19. Tee-head and Steam Pipes

It has already been stated that the area of the steam pipes is put at about one-twelfth that of one cylinder. Hence, for a cylinder 21 inches diameter, the corresponding area of steam pipes would be a little less than 29 square inches and for a 19½-inch cylinder it would be a little less than 25 square inches. Owing to the desirability of encroaching to as slight an extent as possible upon the diametral dimensions of the smokebox, these steam pipes are not made round except at the ends, but are flattened, as shown, so that they lie close to the shell with the longer dimension corresponding with the longitudinal dimensions of the smokebox.

The connection between the foot of the pipe and the cylinder casting is also made by means of a ground joint. The rest of the passage to the steam chest and cylinder has already been treated.

CHAPTER V

PISTON AND PISTON ROD*

Piston design for locomotive work has been subjected to many changes and there are several forms in use, all of which may be included in two classes: the built-up and the solid. The built-up form of spider bull ring with a follower permits of the removal of the packing without the necessity of removing the piston from the cylinder. Where, however, reduction of weight is of the first importance, a solid piston of a double Z-section, and made of either cast iron or steel casting is used, in which case the packing rings are usually made in one piece and sprung into place. In pistons of this type, when made

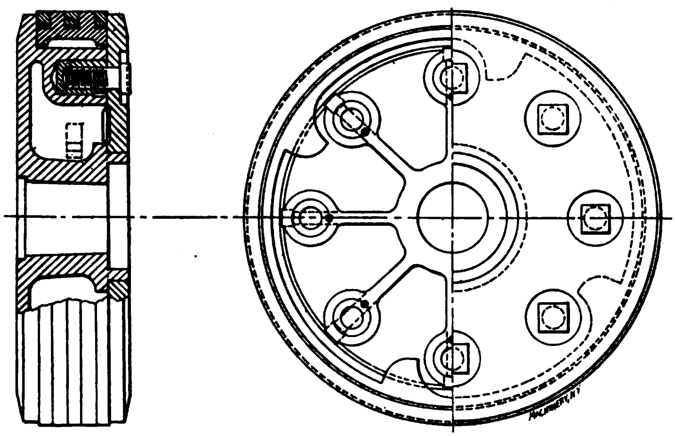


Fig. 20. Piston with Follower and Bull Ring

of cast steel, the rim is usually surrounded by a cast iron or brass ring, the former being fused or bolted to the body, so as to secure a better wearing surface against the cylinder than the steel would afford. The fused rim is, however, much to be preferred as bolting weakens and adds to the weight of the piston. The first type with the follower and bull ring is shown in Fig. 20, and the second in Fig. 21.

The stresses to which a piston is subjected are of two kinds: one is that of the punching or shearing of the disk about the boss, due to the steam pressure exerted upon it between the rim and the hub; and the other the breaking stress exerted across the diameter. The first should, therefore, be treated as a combination of bend and shear but must be estimated separately. The direct shear alone may be easily provided for by laying out several concentric circles with radii of

* MACHINERY, Railway Edition, March, 1905.

r, r', r'', etc., and subtracting the area multiplied by the pressure inside the circle under consideration from the total pressure on the piston and dividing the remainder by the allowed strength of material per square inch of section. This will give the total area of metal to be used; which, in turn, being divided by the circumference of the circle will leave a quotient equal to the thickness of the metal to be used at the point in question, or the minimum thickness of metal to be used in that particular circle. The first part of the problem may be expressed by the formula:

$$A = \frac{\pi R^2 P - \pi r^2 P}{S} \tag{11}$$

in which

A = the required area of metal on the given circle,
R = radius of cylinder,
r = radius of section sought,
P = boiler pressure,
S = allowable stress on the material.

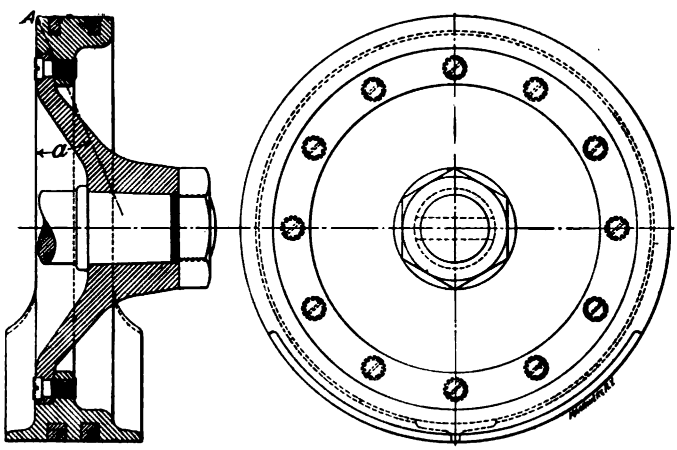


Fig. 21. Cast Steel Z-shaped Piston

The radial bending stress should be provided for by ample internal ribbing so as to maintain the requisite stiffness when double-bottomed pistons are used. The disk or single-walled pistons upon which no ribs can be used, should be so designed that the bending stresses are transformed into tension and compression as the loading alternates, and thus transfer them to those sections that would otherwise be subjected to a low stress, such as the rim. This may be accomplished by making the disk conical or of a double Z-section in which a smaller amount of material can be used than in any other form of piston in proportion to the strength developed. It is best, therefore, that the rim should be cast solid with the main body and be made of brass or cast iron so as to secure the full advantage of its wearing qualities.

In providing for the diametral breaking stress across the disk, which by the way, usually provides for the other stresses, as in the one illustrated, the piston may be considered as a beam whose section is that of the piston through its center, with one-half of the total load concentrated on each half at the center of gravity of the same. The distance from the center of gravity to the center of the piston is 0.42 R. Then

$$S = \frac{0.42\,R \times R^2 \times P}{2M} \tag{12}$$

in which

S = working stress of the metal,
R = radius of cylinder,
M = modulus of section.

M must be calculated for each section of piston to which it is desired to apply the formula.

For disk pistons there are a number of empirical formulas in use. One is to take the thickness near the boss as:

$$t = c\,D\,\sqrt{p} \tag{13}$$

D = diameter of piston in inches,
p = pressure per square inch,
t = thickness of metal, and
c = 0.0046 for cast steel, and
c = 0.008 for cast iron.

The thickness of the plate near the rim may be taken as 0.6 the thickness at the boss. Calculation now gives for a piston 21 inches diameter a thickness at the boss of about 1⅜ inch and 13/16 inch at the rim for steel; and 2⅜ inches and 1⅝ inch at the boss and rim respectively for cast iron. For the calculation of the thickness of the conical portion of a Z-shaped piston, Unwin gives the following formula for cast steel pistons:

$$t = 0.003\,d\,\sqrt{p} \tag{14}$$

in which

t = thickness of metal,
d = diameter of the cylinder,
p = pressure in pounds per square inch.

For a 21-inch piston working under 200 pounds per square inch of steam pressure the thickness would be 0.88 inch.

After having passed through the various forms of rings expanded by steam and springs, practice has returned to that of a simple split ring held out by its own elasticity. The formulas that are given for the width and thickness of these vary between wide limits, ranging on a 20-inch piston ring from ⅜ inch to 3 inches. In practice it has been found that on pistons of 18 to 20 inches diameter the width of the rings may be about ½ inch, and for pistons of from 21 inches to 24 inches it may be about ⅝ inch with a thickness ⅛ inch greater than the width; or the width may be expressed approximately as about ⅛ inch less than one-thirtieth the diameter of the cylinder. In a general way it may be stated that the piston packing in ordinary use is that of a snap ring type of from ½ to ⅞ inch square section to suit the size of the cylinder and of a reasonable tightness. Such rings possess the advantage of simplicity of manufacture and ease of maintenance, and are therefore to be preferred. The only other packing in use is the Dunbar, which still remains a favorite on some roads. It is made in small segments of one L-shaped and one rectangular section, fitted into each other so as to form a ring of square section with the several parts overlapping at the joints. The main objection raised to it is its cost. Great width of packing rings is not required, it being simply necessary to get a good contact around the whole surface, and this can be obtained if the elasticity of the ring is sufficient to give an outward pressure when sprung into position in the cylinder. The size, therefore, is somewhat dependent upon the character of the metal of which the ring is made. It is, of course, desirable to avoid any excess of pressure on account of the wear that would result. The number of rings used varies from two to three with different designers.

The total width of the piston may be placed at approximately one-quarter the diameter of the cylinder, though there are variations from this, the thickness usually being less where but two packing rings are used, when it may be one-fifth.

Piston Rod

Closely allied with the calculations for the piston are those for the rod, the section of which is, to some extent, governed by the areas through the cotter hole or the bottom of the thread at the smallest part of the rod, where it is attached to the crosshead. As this section of the rod must withstand the full boiler pressure on the piston, it should be of such section that the stress will not exceed 10,000 pounds per square inch of metal. As the piston is usually tapered in its fit in the crosshead the main body will be large enough for truing or turning up for wear in the stuffing-box. Still the dimensions should be carefully checked for safety, and the determination of the size of the cotter or key is of the first importance. These cotters are usually subjected to a stress in one direction only, as the taper or shoulder on the rod takes that in the other. It is, however, well to calculate this stress as equal to the steam pressure on the whole surface of the piston. The proportions of the cotter must, then, be such as to sustain this full load both in shear and compression. For shear a steel cotter may be safely subjected to 10,000 pounds per square inch of section and to 24,000 pounds in compression. Taken in single shear, the stress will be one-half the total load and the section resisting this will be equal to the thickness multiplied by the width. We thus have the formula:

$$W = 2btS_1 \qquad (15)$$

in which

W = load on the cotter,
b = width of cotter,
t = thickness of cotter,
S_1 = shearing stress on metal.

In the case of a 21-inch cylinder working under 200 pounds pressure this formula becomes:

$$69{,}272 = 20{,}000bt,$$

or

$bt = 3.46$ square inches.

If then t is assumed to be ¾ inch, b becomes 4.61 or 4⅝ inches.

The next and most important step in the matter is to determine the diameter of the body of the piston rod. The stresses to which this member is subjected are both in tension and compression; and each must be taken into consideration in calculating the proper diameter. For tensile stress we have

$$d = 2\sqrt{\frac{W}{\pi S}} \tag{16}$$

in which

d = diameter of the rod,

W = total load on the piston,

S = allowable working stress on the material = 10,000 pounds per square inch.

By substitution, we have, for a 21-inch piston

$$d = 2\sqrt{\frac{69{,}272}{31{,}416}} = 3 \text{ inches nearly.}$$

The determination of the compressive strength is a much more complicated matter owing to the fact that the piston rod cannot be considered as a strut pure and simple, because of its motion and because of the bending stresses that are put upon it by the looseness of the crosshead in the guides and the piston in the cylinder. A number of empirical formulas have been proposed for this work, but none of them has as yet been accepted as adapted to all conditions on account of the variation in the results that have been obtained in experiments conducted upon a large scale. It is generally considered, however, that when the length of a column is less than twelve times its diameter, the compression can be safely placed at the same figure as the tension, and this is the case with nearly all locomotive piston rods. Cases, however, may arise when the compression would be excessive and, in such, it is well to refer to Rankine's formula for columns with round, free ends, which is

$$\frac{P}{A} = \frac{S}{1 + \frac{q\,l^2}{r^2}} \tag{17}$$

in which

P = the total load,

A = area of the section of the rod,

S = ultimate compressive strength of the material = 150,000 pounds per square inch,

l = length of rod in inches,

r = radius of gyration $= \frac{d}{4}$,

$$q = \frac{4}{25{,}000} \text{ for steel,}$$

$$q = \frac{4}{36{,}000} \text{ for wrought iron.}$$

This formula can be readily changed to terms indicating the diameter, by substitution, and a proper allowance of a factor of safety for S.

A similar formula of Merriman is

$$C = \frac{B}{1 - \frac{n\,B}{10\,E} \times \frac{l^2}{r^2}} \qquad (18)$$

in which

C = the maximum compression stress per square inch of area,

B = the load per square inch of section of the rod,

E = modulus of elasticity = 30,000,000 for steel; 25,000,000 for wrought iron,

n = 1 for round end bearings and ¼ for square ends.

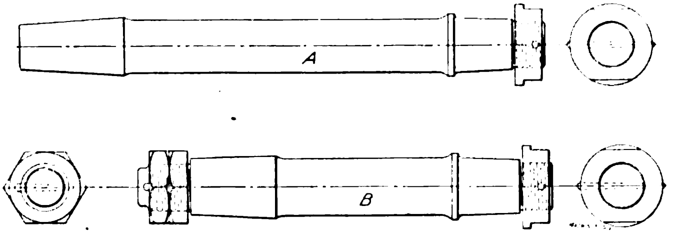


Fig. 22. Piston Rods for Locomotives

For an approximate estimate, we may use the formula:

$$W = \frac{1125\,\pi\,d^4\,S}{4500\,d^2 + 4\,l^2} \qquad (19)$$

in which

l = the length of the rod in inches,

S = allowable stress per square inch of section.

For the determination of l it will be necessary to lay out the rod on the drawing board so as to allow for the necessary clearances, and the length, in the case of the consolidation cylinder with 26 inches stroke, will be found to be about 38 inches from the back face of the piston to the boss on the crosshead.

Then assuming the maximum fiber stress to be 9000 pounds per square inch Formula (19) becomes

$$69{,}272 = \frac{1125 \times 3.1416 \times d^4 \times 9000}{4500\,d^2 + 4 \times 1444}, \text{ from which } d = 3.3.$$

As some allowance must be made for the truing up of the rod on account of wear the addition to this amount will depend upon the personal opinion of the designer, modified, to an extent, by the character of the road upon which it is to work, a greater addition being made for a sandy track than for one that is rock ballasted, because of the greater wear likely to take place on the former. Taking ⅜ inch as a proper allowance for this purpose, the rod becomes 3 11/16 inches in diameter and had therefore best be made 3¾ inches to avoid odd measurements.

Fig. 22 gives the two forms of piston rods in most common use in the United States. In the upper elevation, *A*, is shown the form used when it is attached to the crosshead by a cotter; the lower, *B*, shows one fastened by nuts. It may be added in conclusion that piston rods and keys should be made of a high grade steel whose tensile strength does not fall below 65,000 pounds per square inch.

It now remains to be seen whether there will be metal enough left in the rod, after cutting out a keyway ¾ inch wide, to allow sufficient strength to carry the load imposed. This may be expressed by the formula:

$$W = \left(\frac{\pi d^2}{4} - d\,t \right) \times S \qquad (20)$$

in which

d = diameter of piston rod,

t = width of keyway,

S = allowable tensile stress to be put on the metal.

Assuming $S = 10{,}000$, the formula becomes

$$W = (11.04 - 2.81) \times 10{,}000 = 82{,}300,$$

a result far in excess of the load of 69,272 pounds that it will be required to sustain; and this margin is still further increased by the enlargement of the ends as shown in Fig. 22.