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LOCOMOTIVE DESIGN

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Part III

SMOKEBOX, FRAMES, AND DRIVING MACHINERY

SECOND EDITION

CONTENTS

CHAPTER I

SMOKEBOX AND EXHAUST PIPE ARRANGEMENT*†

The arrangement of the smokebox and the draft appliances contained therein ranks among the more important of the details in the designing of a locomotive, for upon it largely depends the efficiency of the whole machine. Perhaps it is on account of this very importance, coupled with the wide range of variables that must be taken into consideration, that there is as yet no standard construction that is acknowledged to be the one that will produce the highest efficiency under all conditions.

The problem to be solved, with some modifications, is to secure the largest possible exhaust nozzle, and uniform action on the fire over the whole surface of the bed.

The action of the exhaust is, for the most part, that of a jet of steam drawing air with it by the friction of its sides; though, at very low speeds, the plunger action is also influential since there is a perceptible interval of time between the exhausts. When the plunger action predominates, the form and diameter of the stack as well as the heights of the top and bottom of the same above the exhaust nozzles are matters of importance. The jet action is, however, the one that for the most part controls in all the working of the engine, but the smokebox details must be arranged to suit both. In considering the jet action, the length of the stack, and, consequently, the height of its top above the nozzle, is of minor importance. It is evident that the shorter the stack the less will be the frictional resistance of the gases, so that it is useless to extend it beyond the point where they have obtained their maximum velocity. On the other hand, were the steam allowed to escape into the atmosphere through too short a stack, the interval between exhausts would be sufficient to permit the air to rush back into the smokebox and firebox and, by destroying the partial vacuum that had been created, add very materially to the work that would have to be done. For this reason it is necessary to sharpen the exhaust by contracting the nozzle, thus prolonging the time of its action and increasing the velocity of the steam. On the other hand, if this contraction is made too great, there will be an excessive action

* The present number of MACHINERY'S Reference Series is the third 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. In order to give concrete form to the examples and theoretical considerations, it is assumed that a consolidation freight locomotive and an Atlantic type passenger engine are being designed. It is further assumed that these locomotives are designed for a division 150 miles long, laid with rails weighing 75 pounds per yard, and with a ruling grade of one per cent ten miles in length.

† MACHINERY, Railway Edition, October, 1905.

and a breaking up of the bed of the fire, coupled with an undue back pressure in the cylinders.

The stack should be of such length that at moderate speeds one exhaust is entering at the bottom before the last of the preceding one has escaped at the top. At the same time it should be of such shape that resistances are reduced to a minimum. That this may be done it should increase in area in proportion to the loss of speed of the gases, by which means the inertia of those leading will be utilized to reduce the resistance of the succeeding ones; which, in turn, serves to increase the efficiency.

The exact amount of retardation of the gases in their passage from the nozzle to the air is only obtained by experiment, but it is evident that the stack should be flared, with the enlarged portion at the top. For practical purposes, however, the straight taper will answer every requirement with no noticeable difference from that of a form theoretically correct.

The most exhaustive experiments along this line that have thus far been made are probably those of Von Borries and Troske that were carried out in Germany in the early nineties and subsequently published in this country, from which it appears that the proper taper for a stack is approximately one in twelve and that its length should be three or four times the diameter at its smallest point or choke, a proposition that was confirmed by the committee of the American Railway Master Mechanics Association on Exhaust Pipes and Steam Passages in 1896.

The results of the Von Borries and Troske experiments may be approximately expressed by the formulas:

$$d = 0.156 \sqrt{\frac{S \times R}{S + 0.3\,R}} \qquad (1)$$

in which

d = diameter of the exhaust nozzle,
R = area through tubes,
S = grate area,

all expressed in inches.

$$h = 14\,d \qquad (2)$$

in which

h = height of the top of the stack above the top of the exhaust nozzle of a straight pipe.

$$D = 3.8\,d \qquad (3)$$

in which

D = diameter of the stack at the top.

$$D' = 0.65\,D \qquad (4)$$

in which

D' = an imaginary diameter which the bottom of the stack would have were it to be drawn down to the level of the top of the exhaust nozzle.

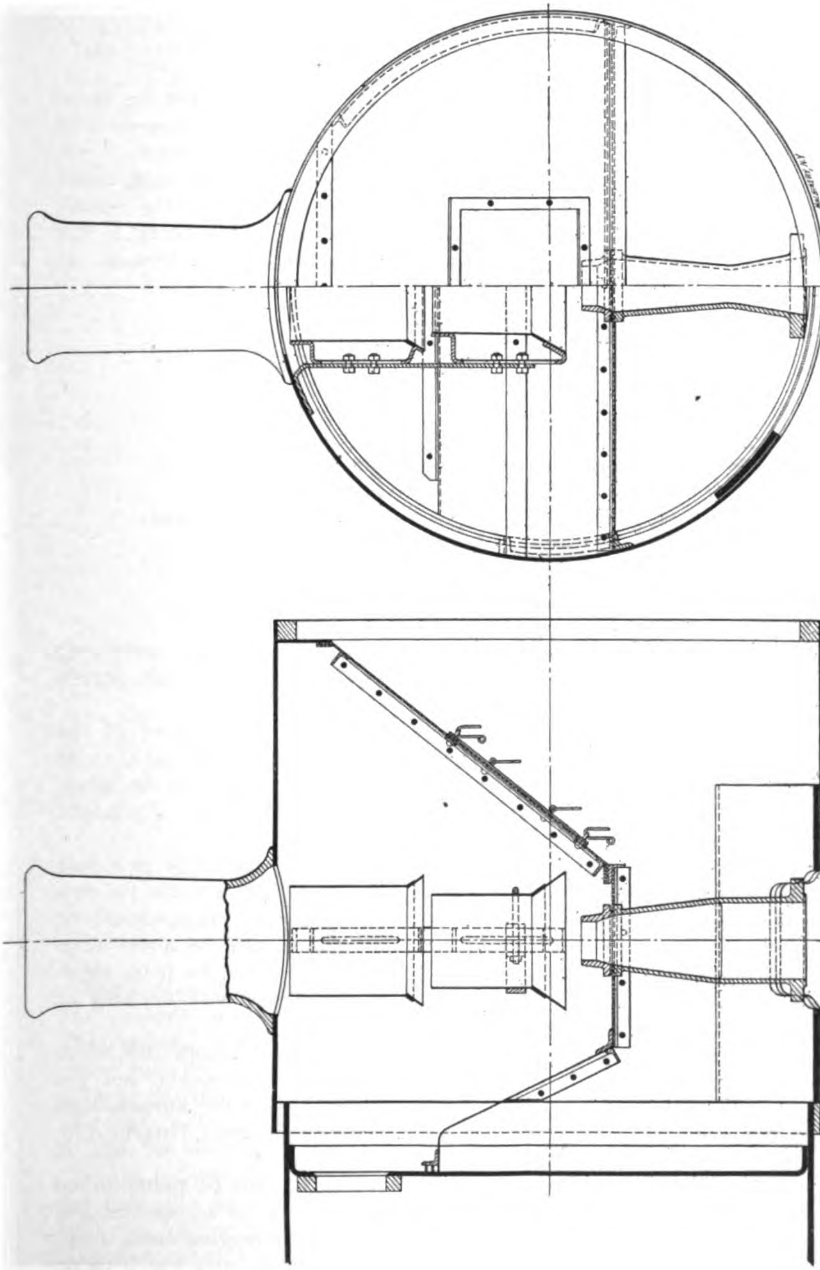

Fig. 1. Longitudinal and Cross-section of Smokebox of Express Passenger Locomotive

The height C of the choke above the top of the nozzle should not be more than 0.4 h or

$$C = 0.4\,h \qquad (5)$$

When circumstances do not permit the heights required by these formulas to be used, it is good practice to use at least the diameters of stack obtained by following the rules that have been laid down.

As for the effectiveness of the exhaust, it has been found that, under certain conditions, one pound of steam, discharged from the nozzle, will displace about 2.5 pounds of smoke-box gases, according to the following formula, that has been deduced from earlier experiments.

$$A = D\sqrt{\frac{2\left(\frac{T}{S}\right)^2\left(1-\frac{V}{S}\right)}{\frac{V}{S}\left[l+2\left(\frac{T}{S}\right)^2\right]}}$$

in which

A = weight of gases displaced in pounds,
D = weight of steam discharged through the nozzle in pounds,
V = area of exhaust nozzle in sq. inches,
S = area of stack in sq. inches,
T = area through tubes in sq. inches,
l = a coefficient (approximately 4).

The weight of air displaced is in direct proportion to the weight of the steam discharged through the exhaust nozzle, when all adjustments have been properly made.

According to the experiments referred to above, the area of the stack, if straight, should be about twelve times the area of the exhaust nozzle, and in one that is well proportioned and tapered, the effect of the nozzle can be increased 12 per cent above that of a straight stack.

The base of the straight stack should be from 30 inches to 36 inches above the top of the nozzle, and the length about three times its own diameter. The choke or smallest diameter of a taper stack should be about 24 inches above the nozzle and its area should be about nine times the nozzle area. The length of the stack should be from three and a half to four times the diameter of the choke, with an area at the top not more than twice that of the choke.

These rules run remarkably close to each other, though the latter is considerably older than the former. They have, however, not yet secured the recognition in this country that they have abroad, and we find a variation of smokebox arrangements in use. This is due, in all probability, to the great variety of fuel burned.

Taking up the smokebox details in general, it will be found to be advantageous to locate the nozzle as low as the area across the box will allow. The baffle plate or diaphragm should be carried down, from above the top row of tubes, to the top of the nozzle, from which point it should be continued horizontally forward well to the front of the

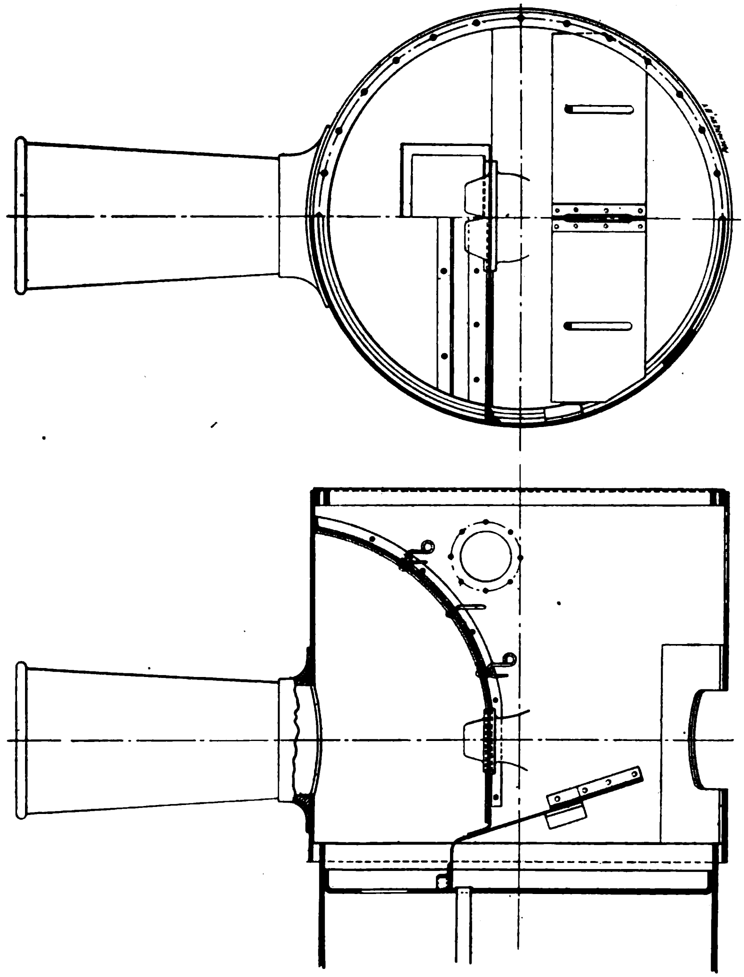

Fig. 2. Longitudinal and Cross-section of Smokebox of Consolidation Freight Locomotive

exhaust pipe, from which the inclined spark-arresting netting starts and extends to the top of the smokebox as shown in Figs. 1 and 2. This general arrangement has come into common use, and, in many cases, constitutes the so-called self-cleaning front end.

In its entirety, it consists of an adjustable diaphragm plate so located as to allow but a limited passage for the gases and sparks between it and the bottom of the smokebox. The diaphragm is carried well forward to limit the area of the passage so that the cinders and gases are kept in a state of constant circulation by the draft, with the result that the former are ground so small that they pass through the netting and out at the stack at a temperature below the igniting point, to which they have been cooled by the process.

The size of the smokebox is not a matter of much importance, except that a large one modifies the effect on the fire caused by the pulsations of the exhaust, on the same principle that an air chamber on a pump will cause a more uniform flow of water than would otherwise be obtained, whereas with a small box the fire will be more disturbed. On American locomotives the size of the smokebox has been reduced to a practical uniformity in the matter of length, ranging, as it does, from 6 feet to 7 feet, which is considered to be large enough to effect the desired uniformity in draft and affording sufficient room for the location of the diaphragms, netting and deflectors. Any length less than 4 feet would be considered small on a standard engine.

With the use of the low exhaust pipe, it is frequently found to be advantageous to introduce one or more draft or petticoat pipes between the exhaust nozzle and the bottom of the stack, as shown in Fig. 1, whereby, what amounts to several jets milder than that of the original issuing from the nozzle, are obtained. This holds especially for simple engines. For compounds it is advisable to extend the stack down into the smokebox to within 20 inches or 24 inches of the top of the nozzle and to secure the increased length of stack by which a more uniform action on the fire is obtained. In this arrangement of the smokebox no specific rules or proportions can be given for the working out of the details that will be of any value. The specific dimensions of the several parts will depend not only upon the size of the engine and the work that it is intended to do, but most particularly upon the fuel that is to be burned. For this a special adjustment is needed, and one that will give perfect satisfaction with a certain grade of coal will not be found suitable for another of a different character.

In Figs. 1 and 2 are given the longitudinal and cross-sections of the smokeboxes for the two engines that are being developed, from which the general proportions and arrangement of the parts according to modern approved practice may be determined.

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CHAPTER II

THE FRAMES*

Although the frames are the foundation upon which the locomotive is constructed, they are not the first thing to be taken into consideration in the designing of the machine. As a support they can be varied in form to suit the requirements of the boiler, machinery and other parts and so only take on their final shape when these other parts have been arranged, though their attachments and presence must be borne in mind throughout the whole of the preparatory work. That they must receive careful consideration goes without saying, for upon them depends much in the matter of strength and rigidity, and perhaps even more in the way of maintenance, as broken frames are usually expensive to repair. In short, weak frames are a source of constant trouble in the way of failures and repairs, since they are frequently the cause of abnormal wear of the moving parts of the machine as well as a never ending, though sometimes obscure, agency in producing hot boxes.

As the width of the frame is necessarily limited, and, as the lateral stresses to which it is subjected are very great, it follows that it must not only be of ample size, but must be thoroughly braced so as to be able to withstand the shocks to which it is exposed when in service. It is impossible, however, to present any absolute formula by means of which the dimensions of a frame can be calculated. It is, of course, comparatively easy to calculate the stresses to which it will be subjected under the direct action of the steam in the cylinder, but this falls far short of being sufficient to provide for the diagonal and lateral stresses that are set up when the machine is worked heavily. These are of a very serious nature, and cannot be estimated when the speed is high or the track rough.

Experience shows that the lack of proper bracing is more often the cause of frame failures than the actual size of the sections used. For, if the frames are held rigidly in position both horizontally and vertically, these high running stresses become merely those of compression and tension. It is, therefore, apparent that the bracing or construction of the transverse frame work is a matter equal in importance to that of providing longitudinal strength. Hence, it cannot be too strongly enforced upon the attention of the designer that it is impossible to estimate the components of the forces that are set up in every direction, as in the case of the derailment of one or more pair of drivers. Nor can all of the varying conditions be ascertained as they exist at high speeds or on a rough track, where stresses of a momentary character are set up, that differ widely from second to

* MACHINERY, Railway Edition, November, 1905.

second, and where the momentum of the engine is an important factor in determining the intensity of the side blows that are delivered.

As already stated the stresses imparted by the working of the engine are comparatively easy to estimate and this can be done by the following formula:

$$F = \frac{P \times \pi d^2 c}{4 e} \qquad (7)$$

in which F = stress produced,

P = boiler pressure,

$\frac{\pi d^2}{4}$ = piston area,

c = distance from cylinder center to frame center on the opposite side of the engine in inches,

e = distance between frame centers in inches.

Practice has shown, however, that no matter what may have been the attempt to hold the frames in line, it is impossible to eliminate the bending moments, and that an allowance must be made for them as well as for the other incidental stresses that have been referred to. The most practical way of accomplishing this is to assume a suitable fiber stress for that portion of the frame section which is above the pedestals and this may be done by the formula:

$$A = \frac{F}{S} = \frac{P \pi d^2 c}{4 e S}, \qquad (8)$$

in which A = the section area of the frame,

F = total stress as obtained by Formula (7),

S = fiber stress.

For the point in question the fiber stress should be placed at 4,000.

The same Formula (8) holds good for the upper frame section between the pedestals as well as for the lower rail. The value of S, the fiber stress, should be changed to 5000 for the former and 7500 for the latter. These values give the lowest practical limit that it is advisable to use for wrought iron frames. For cast steel frames, these values of S had better be made 3000, 4000 and 5500 respectively. In all cases the nearest larger even dimensions should be used, so that the sectional area may not fall below the requirements of these formulas.

For practical reasons the width of the frame section should not vary more than ½ inch for engines having cylinders more than 18 inches in diameter, and the ratio of depth to width should be kept as near as possible to 5 : 4, the section above the pedestal being taken as a base. The greatest width of the pedestal legs should not be less than ½ inch. This, naturally, causes a deviation from the figures obtained by the formulas, and it is here that the necessity for good judgment comes in; for the effect of the reduction of sectional area by bolt holes must be considered, and this is particularly true in the case of small engines.

When larger holes than those used for the ordinary frame bolts are put in to take such parts as the equalizer or brake hanger bolts, an

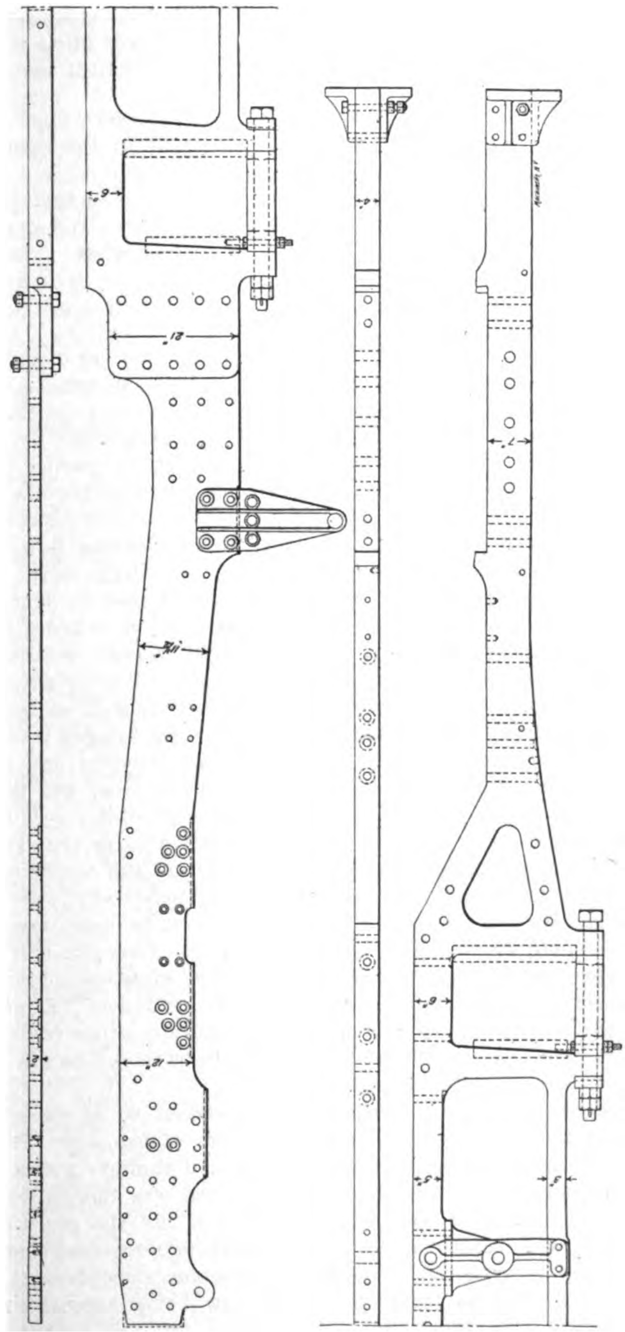

Fig. 3. Frame for Atlantic Type Passenger Locomotive

additional depth should be used at such places, to compensate for the material drilled out; and, in every instance, large fillets should be used at all unions between the horizontal and vertical members of the frame.

The front extension that passes beneath the cylinders when a single bar frame is used, should be of an area equal to the sum of the areas of the top and bottom rails of the main frame. When top and bottom rails are used at the cylinders, the former should have the same sectional area as the top rail of the main frame and that of the latter should not be less than that above the pedestals. Finally the splices or connections between the front extension and the main frame should be made as long as possible, and be well bolted and keyed together.

At all points where there is any fitting at the corners of the frames for cylinders or braces and especially for the pedestal shoes and wedges, these corners should be well rounded not only on the frame, but on the piece to be fitted to the same. As a rule, the wedges are fitted to the back leg of the pedestal so that the greater pressure exerted on the shoe and the vertical front leg, in running ahead, may bear perpendicularly against the face of the same. The slope of the rear leg where the wedge has a bearing usually ranges from ¾ to 1 inch in 12, which gives ample opportunity to adjust for wear.

Reverting now to the bracing for the frame, it may be again reiterated that it can hardly be made too substantial. The cylinder castings, which usually bear the greater portion of the burden of keeping the frames in line, should be relieved as far as possible by broad, well-ribbed cast steel cross-ties, having both horizontal and vertical extensions so as to form a rigid diagonal and transverse bracing between the upper and the lower rails, as well as by the horizontal and diagonal bracing that is needed between the frames themselves; and this bracing should be placed as close to the pedestal as possible.

No specific rule can be given either for the extent or the exact location of such bracing, because the requirements of the Stephenson link motion that is placed between the frames practically prevent the application of such bracing at points where it is most needed and where it would do the most good. The frequent results of this prohibition are to be found in a constant increase of frictional resistance and an ultimate breaking of the frames and cylinders. At the back end of the frames the footplate should extend along the frame for as great a distance as possible, as no adequate bracing can be put beneath the ashpan on the ordinary type of engine.

The pedestal binders on heavy engines should be of wrought iron with deep notches to receive the lower ends of the legs. For small and medium-sized engines, however, a bolt and thimble makes a satisfactory binder. For many years wrought iron was the only material used in the frames of American locomotives, and the preceding discussion has been based on this practice, although the cast steel frame has also been considered. The latter possesses some decided advantages over the forged frame and it is now being extensively intro-

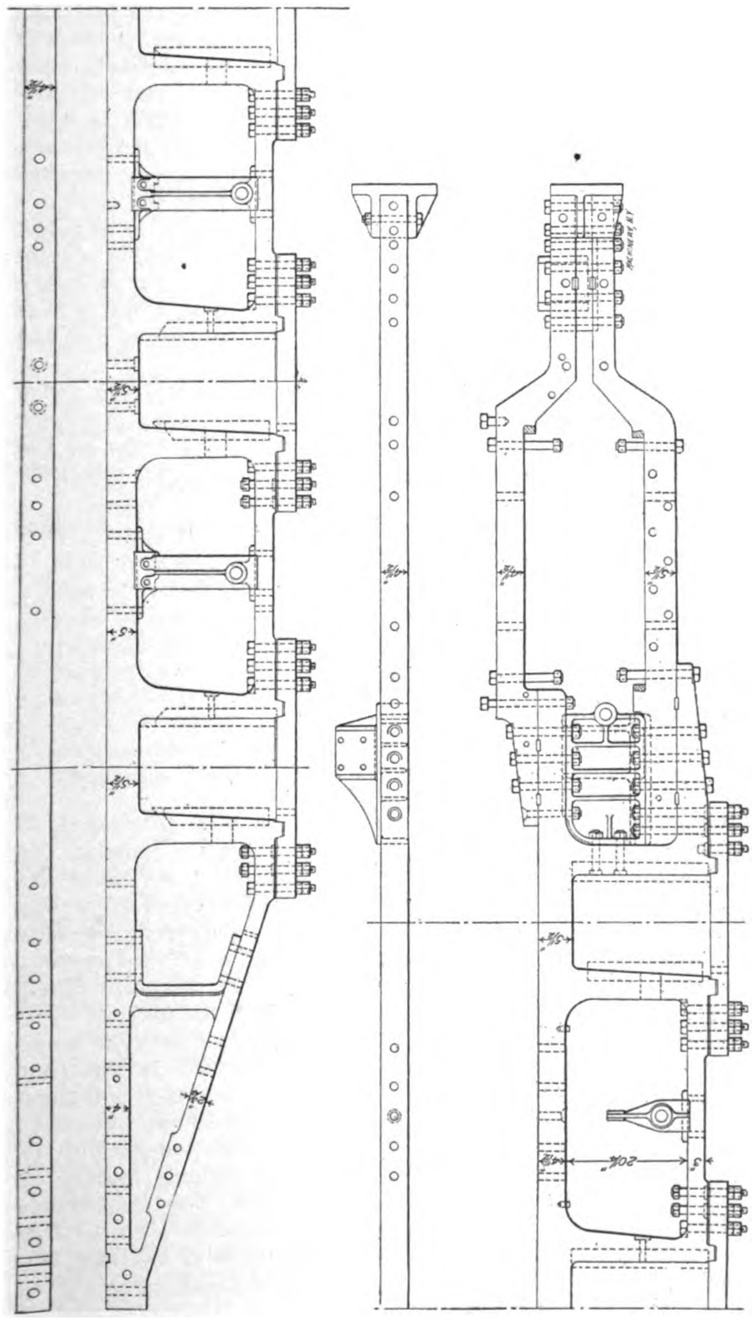

Fig. 4. Frame for Consolidation Freight Locomotive

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duced. The chief of these advantages is the ability to use I-sections and otherwise to so distribute the material that it is disposed to the best advantage to resist the stresses to which it will be subjected. This makes a stronger and a lighter structure, but has the resultant disadvantage in the difficulty which this lighter section offers to welding, if it is broken. This disadvantage more than offsets the primary advantage of the best disposition of the metal, and current practice is now leaning strongly toward the use of the rectangular section.

Another advantage in the use of the cast steel frame is to be found in the possibility of casting on all brackets and of making special arrangements for the attachment of auxiliary parts such as brake cylinders, braces, hangers, and the like; so that it is probable that cast steel frames of a section corresponding to that of those of wrought iron will be increasingly used.

Turning now to the frames of the two engines that we have under consideration, let us see how their designs conform to the principles that have been thus far set forth. Fig. 3 shows the frame of the Atlantic type express locomotive, and Fig. 4 that of the consolidation locomotive.

In the case of the former an application of Formula (7) becomes

$$F = \frac{200 \times 298.65 \times 64.5}{43} = 89{,}579 \text{ pounds,}$$

in which

$$\frac{\pi d^2}{4} = 298.65.$$

Then from Formula (8) we have

$$A = \frac{89{,}579}{4{,}000} = 22.375 \text{ approximately.}$$

In this case 22.375 square inches is the lowest limit that would be allowable for the sectional area of the frame above the pedestal. In order to have even figures for the frame dimensions, the latter are given a depth of 6 inches and a width of 4 inches, producing a sectional area of 24 square inches. Likewise the upper frame section should, according to the formula and a fiber stress of 5,000 pounds, have an area of 18 square inches, which it has in the depth of 4.5 inches and width of 4 inches. The lower rail, with the fiber stress of 7,500 pounds should have an area of 11.9 square inches. It is made 12 by a depth of 3 inches and a width of 4 inches. In this case the single bar at the cylinder connection is made 7 inches deep instead of the 6 inches that is found over the pedestal, so as to compensate for the holes that are drilled there. In the consolidation locomotives the areas are less than those called for by the formula because of the distribution of the stresses through four pairs of driving wheels instead of two. Large fillets are used at the union between vertical and horizontal members, and in the case of the frame of the consolidation engine the junction of the front and main sections is made very secure by means of bolts and wedges.

CHAPTER III

CROSSHEAD AND GUIDE BARS[*]

The piston rods and the pistons are at the origin of the motion of the machinery, which must now be considered. Starting with the outer connection of the piston rod, the crosshead will be found to exist in various forms, from which a selection can be made to meet the requirements of any particular case. It should be borne in mind, however, that for heavy or high-speed engines the double-bar guide, or what is ordinarily known as the alligator type of guide and crosshead, will give the most satisfactory results, and is, consequently, the one that has been most extensively adopted. In fact, where conditions will admit of its being used, this type of crosshead will probably give the best satisfaction for any kind of an engine, even though it does necessitate the use of more metal by which the weight is greater than in a single-bar structure. In the designing of these parts it is also desirable to have the guides as close together as possible so as to reduce the distance to the piston rod to a minimum, and this holds for either a single- or double-bar construction.

As for the crosshead pin, it might be very small so far as its power to resist the working stresses is concerned, since these are applied in shear only, but it must be made large enough so as to provide an ample bearing surface for the brasses in the main rod. In short, it should be of such diameter and length that the load does not exceed 5000 pounds per square inch of projected area. The formula for these dimensions may be expressed as:

$$dl = \frac{P}{S} \tag{9}$$

in which

d = diameter of crosshead pin,
l = length of crosshead pin,
P = total pressure upon the pin,
S = allowable pressure (5000 pounds) per square inch of pin.

In this it is necessary to assume either d or l, and the pressure P may be calculated by multiplying the area of the piston by the boiler pressure. The length is the dimension most commonly assumed and this is taken to suit the width of the crosshead.

In connection with the designing of the pin it should be kept in mind that, because of the action of the engine on curves and the side play in the driving boxes, the bearing on the crosshead pin is liable to wear faster at the ends than in the center. Under such conditions there will be a bending moment in addition to the shear, so that it is advisable to calculate the stress to which it will be subjected under

[*] MACHINERY, Railway Edition, December, 1905.

full boiler pressure from the formula:

$$S = \frac{P\,l}{4\,m} \qquad (10)$$

in which

S = the fiber stress to which the metal will be subjected,
P = total piston pressure,
l = length of crosshead pin,
m = moment of resistance of the circular section of the pin.

The moment of resistance of circular sections may be found from the formula

$$\frac{\pi\,d^3}{32},$$

and for the ordinary diameters of crosshead pins ranging from 3 to 4½ inches the moments of resistance are as follows:

3	inches,	2.66	4	inches,	6.28
3¼	"	3.38	4¼	"	7.53
3½	"	4.21	4½	"	8.95
3¾	"	5.18			

An examination of these figures will show that they vary as the cubes of the diameters.

Owing to the great amount of motion between the guide and the crosshead and the exposure of the surfaces to external influence, the pressure at this point should not be allowed to exceed 70 pounds per square inch, while it is desirable to drop well below these figures if possible, even cutting it down to one-half that amount, which is done in many instances.

In this, as in other cases, it is well to remember that large bearing surfaces are a good investment. The pressure against the guide can be calculated from the formula:

$$P' = \frac{P\,r}{L} \qquad (11)$$

in which

P' = the total maximum pressure against the guide,
P = maximum pressure on the piston,
r = radius of the crank in inches,
L = length of connecting-rod in inches.

This is an approximate formula that gives results sufficiently accurate for steam engine practice where the ratio between the connecting-rod length and crank radius is not less than 6 : 1. The formula for thrust which gives the theoretical reaction on the guide is $P' = P \tan \theta$ in which P' = thrust against guide; P = maximum pressure on piston; θ = greatest angle made by connecting-rod with axis of cylinder.

The area of the sliding surface of the crosshead will then be determined by

$$A = \frac{P}{t} \qquad (12)$$

where A = the required area of the sliding surface,
t = the allowable pressure per square inch.

According to the statement already made t should not be more than 70.

The strength of the guide should be such as to reduce the deflection to a minimum, which under no condition should be allowed to exceed 1/32 inch and should be kept down to 0.01 inch or less when circumstances will allow. In cases where the cylinder center is raised above, but is still parallel with, a line drawn through the centers of the driving wheels, this distance should be added to the radius of the crank r in Formula (11) as well as in the determination of the thickness of the guides, an allowance that will somewhat increase the value of the pressure against the guide, P', above what it would be were the crank radius alone used.

When the guides are forged the section is usually rectangular, in which case the moment of resistance is calculated from the well-known formula:

$$m = \frac{bh^2}{6} \tag{13}$$

in which

m = moment of resistance,
b = width of guide,
h = thickness of guide.

Having given the width, length and allowable fiber stress S, it is possible to calculate the thickness. The width is usually determined by the requirements of the construction, and the bars are fastened at both ends. This places the calculation on the basis,

$$\frac{P'l'}{4} = \frac{Sbh^2}{6}, \text{ or } h^2 = \frac{6\,P'l'}{4Sb}, \text{ and } h = \sqrt{\frac{3\,P'l'}{2Sb}} \tag{14}$$

in which l' = length of guide.

$P' = \dfrac{Pr}{L}$ of Formula (11) with the necessary correction for the height of the center of the cylinders above the driving wheel centers.

It is always desirable and will usually be found necessary to check off this determination of the value of the thickness of the guides h, in order that the deflection may not exceed the amount given above. This deflection may be determined by the formula:

$$f = \frac{P'l'^3}{4Ebh^3} \tag{15}$$

in which

f = the deflection in inches,

$P' = \dfrac{Pr}{L}$, as before,

E = the modulus of elasticity, which for steel may be placed at 30,000,000, and b, h and l' have the same values as in Formulas (13) and (14).

The thickness of the guides having thus been determined, it is always well to add from ⅛ to ¼ inch to the amount so as to compensate for wear.

When the guide-yoke is set ahead of the rear ends of the guides the length of the latter should be considered as the distance between the yoke bolts and those in the lugs of the cylinder head. The guide-yoke to which the back ends of the guides are fastened and by which they are supported must be strong enough to sustain the guide pressure P'. The special form that should be given to the yoke is largely dependent

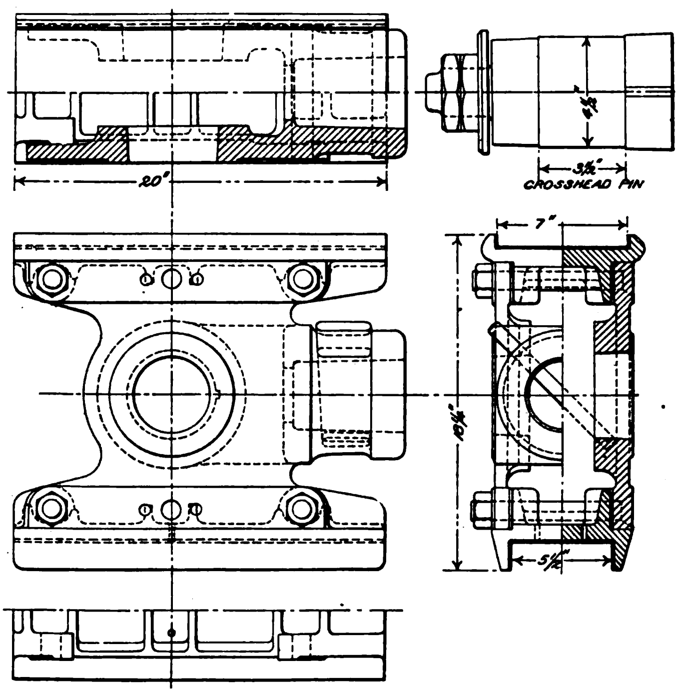

Fig. 5. Crosshead for Passenger Locomotive

upon the other conditions of the design of the engine as a whole, and no definite shape can be recommended that would not call for wide variations. The points to be borne in mind in this connection are that the guide-yoke should not only be strong enough to carry the guides and hold them rigidly in position but should also be made to serve as one of the most valuable and efficient braces for the frame. Its position is one where a substantial support for the frame is needed, and it is customary to take advantage of the opportunity thus afforded and utilize it to the utmost.

With the principles thus enunciated it is possible to make a direct application to the engines that have been kept under consideration, always bearing in mind that exigencies of construction may require a deviation, more or less pronounced, from the mathematical deductions, and that such a deviation is invariably made for the purpose of avoiding unusual or awkward dimensions.

Starting with the crosshead pin, by the substitution, in Formula (9), of the values for the cylinder dimensions and steam pressures assumed to have been decided upon, and by taking a safe margin from the maximum allowable stress of 5000 pounds, using 4000 for the passenger engine, we have:

$$dl = \frac{59{,}730}{4000} = 14.93,$$

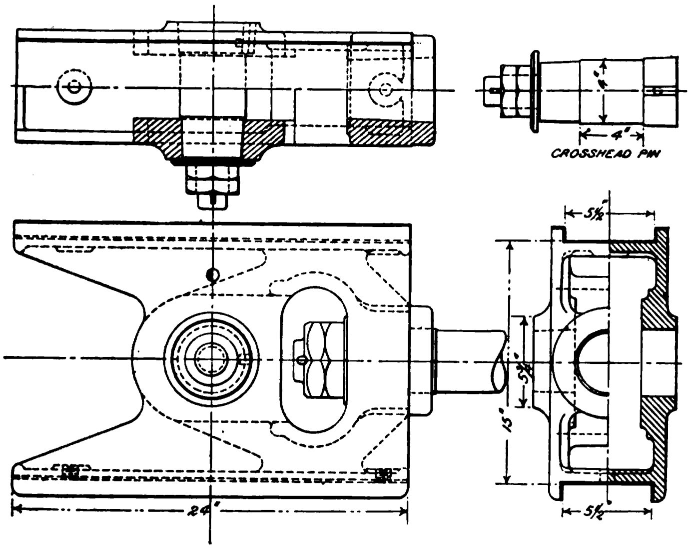

Fig. 6. Crosshead for Consolidation Freight Locomotive

while for the freight engine, where the speed is less, the 5000 pounds may be retained, giving

$$dl = \frac{69{,}276}{5000} = 13.85.$$

Figs. 5 and 6 illustrate the crossheads that have been designed for the passenger and freight engines respectively. In the case of the former, a pin 3½ inches long and 4.3 inches in diameter would satisfy the requirements; but in order to allow for some wear it has been made 4½ inches.

If these figures are checked by Formula (10) we have

$$S = \frac{59{,}730 \times 3.5}{4 \times 8.95} = 5840 \text{ pounds.}$$

This stress is much below what is required for strength in the pin, so that the size in this and other similar cases is governed by the area of the bearing surface rather than by strength. Pursuing a similar course for the freight engine to whose crosshead a length of 4 inches is suited to the other exigencies of the design, it will be found that with a fiber stress allowance of 11,000 pounds a diameter of 4 inches will be required.

Passing to the area of wearing surface on the crosshead, and substituting the values that we have already obtained in Formula (11) together with the length of the connecting-rod and noting that, in the case of the passenger locomotive, the crosshead center is 3 inches above

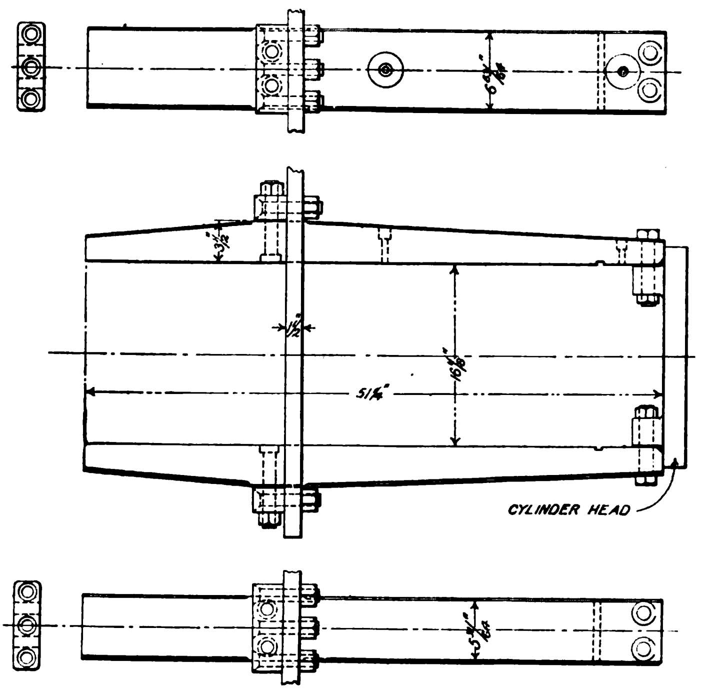

Fig. 7. Guide Bars for Passenger Locomotive

the center of the driving wheels, while in the freight engine it is 2 inches above, we have:

$$P' = \frac{59{,}730 \times 16}{125} = 7645 \text{ pounds for the passenger engine,}$$

and

$$P' = \frac{69{,}276 \times 15}{133.5} = 7784 \text{ pounds for the consolidation engine.}$$

As there is an opportunity in these cases to use ample wearing surfaces, the crossheads are made 20 inches long and 7 inches wide for the passenger, and 24 inches long and 5.5 inches wide for the freight engine, dimensions which reduce the pressure per square inch to about 55 pounds and 67 pounds respectively.

The guides for the consolidation locomotive, illustrated in Fig. 8, have their width fixed by that of the bearing surface of the crosshead, while their length is determined by the length of the crosshead, the stroke and their own fastenings. With Formula (14) a thickness would be obtained very much less than that used. The dimensions adopted

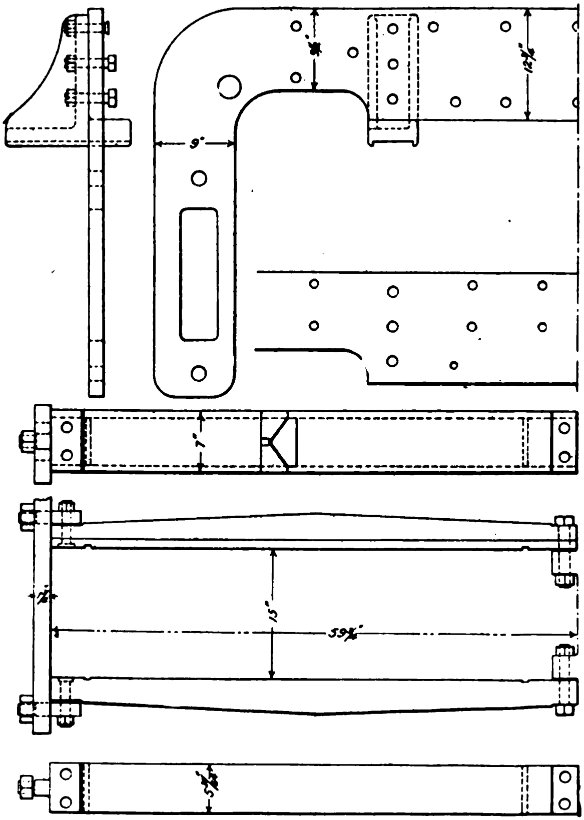

Fig. 8. Guides and Guide-yoke for Freight Locomotive

are rendered necessary by the desirability of securing rigidity in accordance with Formula (15), which then becomes

$$0.022 = \frac{7784 \times (59\frac{3}{4})^3}{120{,}000{,}000 \times 7 \times h^3}; \; h = 4.5 \text{ inches approx.}$$

In examining the engraving Fig. 8, it will be observed that the upper bar is widened above the lip of the crosshead in order to increase the strength, and at the same time reduce the thickness to a minimum, as well as for a protection against grit, by preventing dust and cinders working in between the crosshead and the top guide.

Two designs of guide yokes are shown for the two engines in Figs. 7 and 8. The one for the passenger engine, Fig. 7, holds the guide near the center while the other holds it at the ends. Both are much heavier than the calculations for the mere static load would call for, to allow for incidental stresses due to various conditions such as derailments and other accidents that will put excessive stress on the yoke. In fact, the guide-yoke must be designed in accordance with good judgment and past experience of what is suitable and not as the result of mathematical work. It should be always borne in mind that these formulas are to be used as outlines and approximations rather than for the final and unchangeable determination of the dimensions that are to be used.

Before leaving this subject attention may be called to a few details of current practice in crosshead construction. Steel castings have replaced cast iron and forgings, and when made from a steel casting the crosshead is usually made in one piece. This is the case on the consolidation locomotive, Fig. 6. For the passenger engine, however, it is built up with plates bolted in position. A decided advantage gained by the use of steel castings for this work is that it makes it possible to very materially lessen the weight of the crosshead. This is an important item that the designer should always remember, that the weight of the reciprocating parts should be kept down to the minimum, without a sacrifice of strength. In this, skill and judgment must be used and no invariable rule can be given for the disposition of the material where the stresses of operation must be cared for, while every effort must be made to avoid the internal stresses that may be set up in the process of casting and cooling.

The ideal location for the crosshead pin is at the center of the length of the crosshead, a condition that is realized in the case of that of the consolidation engine. But where this is impossible, as in the case of the passenger engine, it should be placed as near the center as possible. Especial attention should also be paid to the boss around the seat of the piston rod to see that it is of ample strength and not liable to be cracked by keying.

These precautions must be borne in mind in the designing, else the result will be a broken crosshead or piston rod which never ends with a simple break of the first part yielding, but invariably involves other portions of the engine in a break-down that is crippling and costly, including, as it usually does, at least a cylinder head, if not the cylinder itself.

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CHAPTER IV

CONNECTING AND SIDE RODS*

The main or connecting-rod is the part of the mechanism by which the reciprocating motion of the crosshead is converted into the rotating motion of the crank-pin. It is subjected not only to the tensile and compression stresses of the piston-rod but also to other stresses due to the vertical motion as well as to buckling loads imposed by the compression thrust on long columns. In short it is subjected to the stresses of compression, tension, horizontal deflection or bending due to compression, and vertical deflection due to compression, centrifugal force and inertia at high speeds. It is, therefore, of the greatest importance that the section should be as light as possible and yet of ample strength to carry the loads imposed. That is to say, it should be of such form and dimensions as to make the most economical utilization of the material, a consideration that is being more and more severely imposed with the increasing powers and speeds of locomotives. Both experience and mathematical calculations show that these requirements are best fulfilled by the I-section. In the matter of the material, it should be the best obtainable.

As to the area of the section, it should be large enough to withstand the crushing and pulling stresses to which it will be subjected; but, because of the importance of securing lightness of construction a comparatively high fiber stress can be allowed.

As the vertical bending moment is far greater than the horizontal, the section should be deep with broad top and bottom webs, in order that the material may be placed in the most favorable position for resisting these stresses. The vertical web must be of sufficient thickness not to buckle under the compressive load.

An analysis of the condition under which the rod works will show that, when the compression stresses have been cared for, there will be ample material to withstand the tensile loads. A further analysis will show that in compression the horizontal bending moment is simply that of a column with square ends while the vertical is that of one with round ends, and that the centrifugal inertia due to the motion acts as a load practically at right angles to the axis. The first consideration is for the crushing stress which is obtained from the formula:

$$S = \frac{P}{A}$$

in which

P = pressure on the piston,

A = area of the rod,

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* MACHINERY, Railway Edition, February and April, 1906.

S = allowable pressure per square inch,
or where S is given the formula becomes

$$A = \frac{P}{S} \qquad (16)$$

The same formula also holds for tension, and S may be taken to be the same, since the resistance of wrought iron and steel is about the same for both compression and tension.

The vertical bending moment is usually calculated on the basis of the piston pressure working at an estimated maximum speed in miles per hour equal to the number of inches in the diameter of the driving wheels. At such a high speed the piston pressure at mid-stroke will not be more than one-third of the maximum, so that the bending moment due to compression will be small, though that due to inertia will be maximum. The horizontal bending moment is based upon the full piston pressure, P, and it is at its maximum at slow speeds.

The calculation of the various stresses to which the connecting-rod is subjected is a complicated matter, but must be carefully worked out in order to avoid not only weakness, but excess of weight; for on the one side, a weak rod is exceedingly dangerous and is apt to cause a disaster if it breaks at high speed when it is under the greatest stress due to the centrifugal inertia; while, on the other hand, too much material will set up other disturbing elements that affect the smooth running of the engine.

After calculating the direct tension and compressive stresses in accordance with the formula given, the next step is to determine the lateral stresses on the rod when considered as a column with square end supports for which the following formula can be employed:

$$F = \frac{B}{1 - \dfrac{N B l^2}{10 E R^2}} = \frac{10 B E R^2}{10 E R^2 - N B l^2} \qquad (17)$$

in which

F = maximum compressive stress per square inch of concave side of column,

B = load per square inch of section of the rod $= \frac{P}{A}$,

E = modulus of elasticity = 30,000,000,

N = constant = ¼ for square bearing,

l = length of rod in inches,

R = radius of gyration of the section of the rod under consideration.

The same formula may be used for the vertical bending moment, except that as the rod has here become a column with rounded ends the value of $N = 1$ and the radius of gyration for the section must be taken about the horizontal axis. In both cases B is based on the full piston pressure.

The radius of gyration R is obtained by extracting the square root of the quotient of the moment of inertia I divided by the area of the section.

$$R = \sqrt{\frac{I}{A}} \tag{18}$$

in which

$$I = \frac{BH^3 - bh^3}{12}$$

where the several symbols of the second term of the equation are equal to the dimensions called for by the corresponding letters in Fig. 9.

When the speed is at the maximum it becomes necessary to combine the two bending forces due to compression and the inertia of centrifugal action respectively.

The general formula for centrifugal force is

$$C = \frac{Gv^2}{gr} \tag{19}$$

in which

C = centrifugal force,
G = weight of rod in pounds,
v = velocity in feet per second,
$g = 32.2$ = velocity acquired by gravity at the end of one second,
r = radius of motion in feet.

If the rate of revolution per minute is n then

$$v = \frac{2\,\pi\,r\,n}{60} = 0.1047\,r\,n$$

and

$$v^2 = 0.0109\,r^2\,n^2$$

whence

$$C = \frac{0.0109\,G\,r^2\,n^2}{32.2\,r} = 0.00034\,Grn^2 \tag{20}$$

Then taking the assumed maximum speed in miles per hour, V, as equal to the diameter, D, of the driving wheels in inches we have

$$n = \frac{V \times 5280 \times 12}{D \times \pi \times 60} = \frac{336\,V}{D} = 336$$

whence $n^2 = 112{,}896$.

Formula (20) then becomes

$$C = 38.38\,Gr \text{ or } 38.4\,Gr.$$

For convenience this may be converted into terms of the stroke, s, of the piston in inches.

$$s = 2r \times 12 = 24r$$

whence

$$C = \frac{38.4\,G\,s}{24} = 1.6\,Gs. \tag{21}$$

This is the simplest form in which the centrifugal force can be expressed and is as applicable to the side as to the main rods, though the effects will differ with the difference in the motion of the two rods.

Taking the main rod first, the centrifugal force is at zero at the crosshead and at the maximum at the crank-pin. If the centrifugal motion were to be considered as though the whole rod were in circular motion, as in the case of the side rod, with the length of the rod $= l$ and the centrifugal force $= C$, then the load can be represented by a rectangle as indicated by the dotted lines in Fig. 10, in which case the rod would be supposed to be loaded uniformly throughout its whole length with a burden equal to the centrifugal force. But as there is no load at the crosshead end, the rectangle may be divided diagonally, forming a right-angled triangle whose apex is above the crank-pin and which may be taken to represent the centrifugal force as applied. The center of gravity of this triangle is at a distance of $l/3$ from the crank-pin, and it is at this point that the whole of the centrifugal force

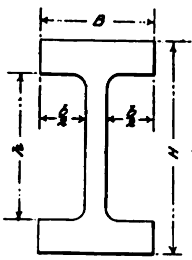

Fig. 9

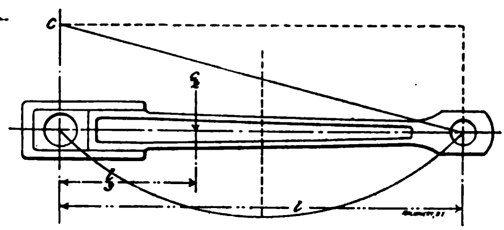

Fig. 10

$(C/2)$, as represented by the triangle, may be considered to be applied to the rod.

As, however, the main rod is heavier at the crank-pin than at the crosshead end, this is not strictly true because the load would fall somewhat nearer the crank-pin than $l/3$, but the maximum stress on the rod will fall between 1/3 and 1/2 the length of the rod from the crank-pin. The exact determination of this point would involve complicated calculations of no real value, in that a safety margin must be provided in any event, and the movement of the point of load application nearer the center merely decreases the margin.

It is, therefore, safer and better to assume the inertia load of $C/2$ to be at the middle of the rod or on the same section as the maximum vertical bending moment due to compression, working as indicated by the curved line in Fig. 10. As the rod is supported at both ends, the maximum moment (M) due to centrifugal action will then obtain when

$$M = \frac{C\,l}{2 \times 8} = \frac{1.6\,G\,s\,l}{16} = 0.1\,G\,s\,l \qquad (22)$$

The fiber stress (T) of the rod at this point due to inertia will then be

$$T = \frac{0.1\,G\,s\,l}{W} \qquad (23)$$

where $W =$ modulus of section.

The combined vertical bending stress will then be equal to the sum of that obtained by Formulas (17) and (23)

$$K = \frac{10\,B\,E\,R^2}{10\,E\,R^2 - N\,B\,l^2} + \frac{0.1\,G\,s\,l}{W} \tag{24}$$

When the speed has reached the indicated maximum the value of B falls considerably below the full boiler pressure, because at mid-stroke the steam in the cylinder is, as already stated, rarely more than one-third the initial pressure. But as a further precaution in the case of temporary spurts of speed, as in the slipping of the wheels, B may be taken as equal to $\frac{P}{2A}$.

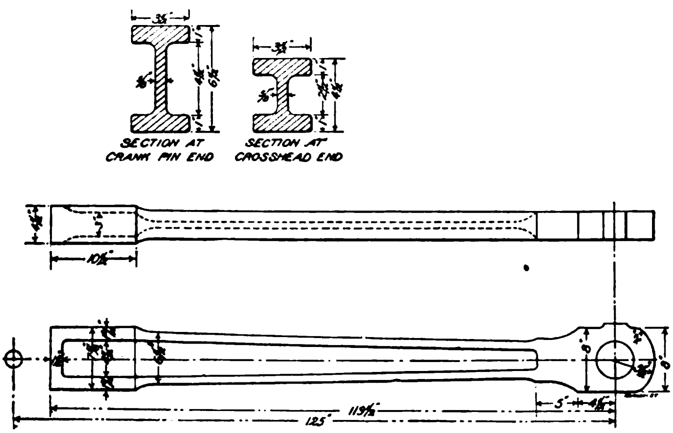

Fig. 11. Main Rod for Passenger Locomotive

The meaning of the symbols used in Formula (24) is the same as that previously indicated for Formulas (17), (19), (21), and (23).

The vertical fiber stress K to which the rod will be subjected in service having thus been determined, the dimensions and proper section to be used can only be ascertained by the assumption of some definite size and then applying the formula to determine its strength. If the section is found to be too light or too heavy, it must be increased or decreased accordingly until the proper figures are found that will meet the requirements of the case in hand.

The stresses at the several points of the connecting-rod can be shown graphically by laying out a momentum curve on the lower side of the rod and an inertia triangle above it as shown in Fig. 10.

In the case of the side rod, the inertia effect due to centrifugal action is represented correctly by the rectangle of Fig. 10, which is equivalent to a load uniformly distributed over its whole length. As for the compression stresses due to the thrust of the piston, these are probably greater at low than at high speeds. The strain on the side rod must always be that of overcoming the slip of the coupled wheels to which it is connected, due to the slight inequalities of circumference that always exist, and the resistance of its own inertia stresses.

Turning now to the application of the principles that have been laid down for the rods of the engines whose parts have been used as a basis of comparison, the main rod of the passenger engine is shown in Fig. 11, the side rod for the same in Fig. 12. The main and side rods of the consolidation freight locomotive will be shown later.

In the case of the passenger locomotive the diameter of the cylinder is 19½ inches, and the boiler pressure 200 pounds per square inch. This makes the pressure per square inch of area $B = \frac{59,730}{A}$ pounds, and for the consolidation locomotive with 21-inch cylinders $\frac{69,276}{A}$ pounds.

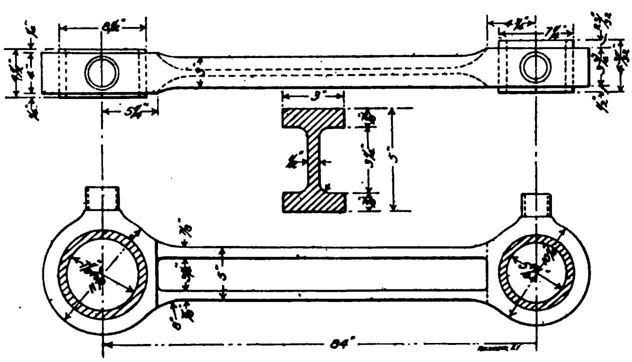

Fig. 12. Side Rod for Passenger Locomotive

The diameters of the driving wheels (D) of the two engines are 72 inches and 57 inches respectively so that the maximum speeds for which the rod stresses should be calculated are 72 miles per hour for the Atlantic and 57 miles per hour for the consolidation engine.

The first thing to be done is to determine the minimum area of the section of the rod according to Formula (16) in which it is necessary to assume the value of S which may be placed at 7000 as an approximation to meet requirements for the other stresses. For the passenger locomotive the formula then becomes:

$$A = \frac{59,730}{7000} = 8.53 \text{ square inches.}$$

As already stated, the method of determining the values of the various factors in Formula (24) is to assume a given section and weight of rod. When the radius of gyration and section modulus of this section have been substituted in the formula, the value of K, as so determined, must not exceed the allowable stresses that are settled empirically.

If, then, we take the main rod of the Atlantic locomotive as shown in Fig. 11, the weight will be found to be 404 pounds.

From Formula (18) the radius of gyration R will be found to be equal to 2.03 and the section modulus

$$W = \frac{I}{q} = \frac{38.2539}{2.75} = 13.9$$

in which q = the distance from the neutral axis to the outer fibers.

By the substitution of these values Formula (24) becomes

$$K = \frac{10 \times 3500 \times 30{,}000{,}000 \times 4}{10 \times 30{,}000{,}000 \times 4 - 1 \times 3500 \times 15{,}625} + \frac{1 \times 400 \times 24 \times 125}{14}$$
$= 12{,}239.$

$K = F + T = 3667 + 8572 = 12{,}239$ in the above demonstration, or

$$F = \frac{10 \times 3500 \times 30{,}000{,}000 \times 2^2}{10 \times 30{,}000{,}000 \times 2^2 - 1 \times 3500 \times 125^2} = 3667$$ pounds per square inch for vertical compressing bending stress under high speed, where

$$B = \frac{P}{2A} = \frac{60{,}000}{2 \times 8.5} = 3500 \text{ pounds.}$$

$$T = \frac{0.1 \times 400 \times 24 \times 125}{14} = 8572.$$

This falls within the stresses allowable for steel in this position which should not exceed 14,000 pounds fiber stress, and should be held as much below this as possible. In this calculation the area and section at the center of the rod are used as the basis of the work. The formulas can be applied in the same way to the side rods as well as to all the rods of the consolidation locomotive. The details of the construction of these rods will be discussed later.

Side Rods

It will be seen by a reference to Fig. 12, which represents the side rod of a passenger locomotive having but two pairs of driving wheels, that the construction is exceedingly simple and that all of the keys and gibs that formerly constituted so unimportant a part of the rods of engines have been entirely dispensed with. The rod in the present construction consists merely of a fluted bar with solid ends into which brass bushings are pressed. These are made to fit over the crank pins with an easy play and afford no means of adjustment to take up the wear. Such rods are the present universal practice on American locomotives. They are used until they become so worn that the pound on the pins is objectionable, when the bushings are renewed.

The side rods of a consolidation locomotive as well as those used on engines having more than two pairs of driving wheels, such as the ten-wheel and mogul classes, require a special arrangement. It is evident that the moving of a locomotive over a rough or uneven track involves a variation in the height of the driving wheels so that a rigid rod extending the full length of the wheel base would be impossible to operate safely. This necessitates the use of a horizontal joint at each

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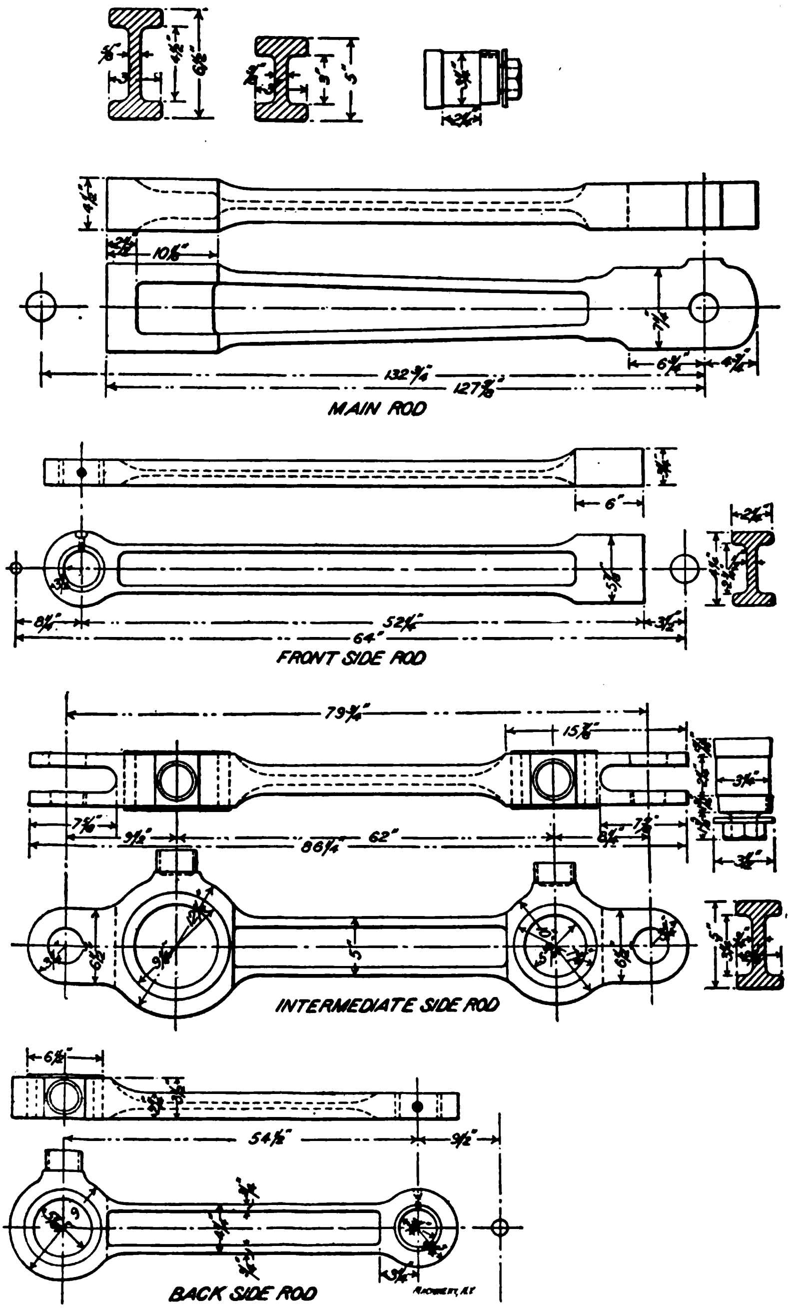

Fig. 13. Main and Side Rods for Consolidation Locomotive

wheel so that there may be a free vertical movement between the wheels without causing any cramping of the rod.

The general form of these rods is the same as that shown for the passenger locomotive with four driving wheels. That is to say, the

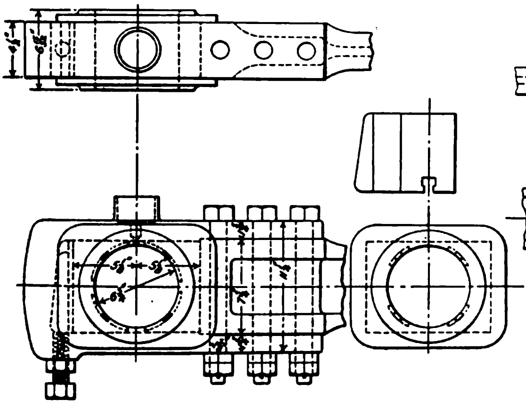

Fig. 14. Main Crank-pin Connection for Connecting-rod

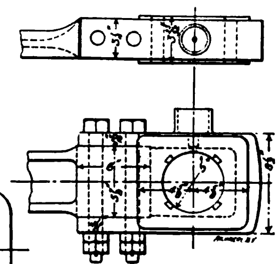

Fig. 15. Front End of Front Section of Side Rod

body of the rod is fluted and the bearings are solid brasses that admit of no adjustment; but in the case of the consolidation locomotive that we have in view, the side rod consists of three sections. The principal rod reaching from the main crank-pin to the pin in the third pair of

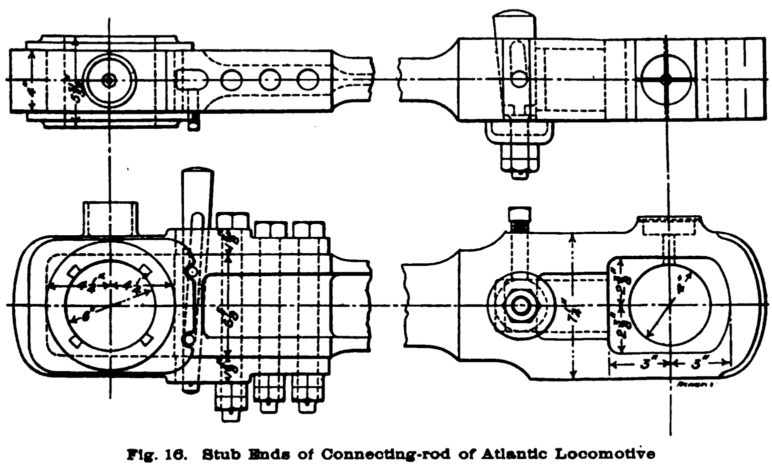

Fig. 16. Stub Ends of Connecting-rod of Atlantic Locomotive

wheels is solid like the passenger engine rod; and, as far as its action in connection with these two pairs of wheels is concerned, is the same. Each end of this rod carries a prolongation forming a knuckle joint with the stud ends attached to the front and back sections of the rod. The pin for these places has a tapered head at the back to fit into a counterbore in the back lip in order to clear the face of the wheel. The rear section of the rod is also solid as far as the connection between the two rear pairs of drivers is concerned and is pivoted in the fork of the intermediate section. Thus, as far as adjustment is concerned, there is none possible longitudinally between the three rear pairs of wheels of this consolidation locomotive.

This is as far as it has been found to be advisable to carry this scheme of non-adjustment. Hence the front section of the side rod is provided with a stub and strap at its front end, by which the distance between the centers of the front and back crank-pin brasses can be adjusted. Even here, however, the adjustment is made with liners and not with keys and wedges. Fig. 13 shows the general construction of the main and side rods on the consolidation locomotive. Fig. 15 shows the arrangement of the details of the stub end at the front end of the front section of the side rod. The brass is shown with babbitt pockets, and is in a strap to which the oil cup is forged solid and held by two bolts of 1⅛ inch diameter. Any adjustments as to length of this rod necessitates the removal of the strap, the filing of the brass and the insertion of liners. Fig. 14 shows the arrangements of the details of the stub end that is used for the main crank-pin connection of the connecting-rod. In this the strap is held to the end of the rod by three bolts of 1⅜ inch diameter each, and an adjustment is provided at the outer end by a wedge acting against the brass, which can be moved up and down by a screw bolt. Of course no adjustment can be made here without taking the brass out and filing it. There are many forms of stub ends that vary in detail from those shown, and which are usually based on a matter of choice in construction rather than any inherent merits of one over the other, the general principles of all being about as stated here.

CHAPTER V

CRANK-PINS AND AXLES*

Turning to the subject of the crank-pin the two considerations that are most prominent are the bending stresses to which it is subjected and the necessity for the provision of a proper bearing surface. The latter is usually the one that determines the size of the pin; but whether it be the limitatiion of the fiber stress due to bending, or the necessity of providing an ample bearing surface, the one calling for the larger diameter of pin is the one that should be selected to control. As regards the distribution of the work to the several pins, it is a problem which has not yet been solved, as the loads vary constantly, so that it is impossible to lay down any hard and fast rule in regard to it. If the bearings were perfect and without any play, the pins might have a truly proportional load to carry; but, as this is never the case, the best that can be done is to agree upon an arbitrary division. There are moments, for instance, when the main crank-pin will have to sustain the whole load, as when it is on the dead center and the side-rod bearings have considerable play. In other positions of the crank, the side rod is counteracting the stress on the main pin, so that if this assistance from the side rod is considered, a comparatively high fiber stress can be allowed. This pin should have a wheel fit somewhat larger than the bearing. The length of the leverage of the main rod on the pin is usually taken from the wheel hub to the center of the bearing, and the diameter at the former point, that is, just outside the wheel hub, as shown in Fig. 18, may be found by the formula

$$D = \sqrt[3]{\frac{P\,l \times 32}{\pi\,S}} = \sqrt[3]{\frac{10.2\,P\,l}{S}} \qquad (25)$$

In which

D = diameter of pin just outside the wheel fit,
P = the total pressure exerted on the pin by the piston,
S = the allowable fiber stress for the material,
l = the length of the pin as shown in Fig. 18.

The fiber stress should be limited to 16,000 pounds per square inch for steel and 14,000 pounds for wrought iron.

There should be only a small reduction of diameter of the pin for the bearing so that the requisite surface can be obtained without making the pin too long. This usually calls for so large a diameter that calculation of strength at the shoulders is unnecessary. The required projected area for the bearings must be such that the pressure per square inch does not exceed 1600 pounds per square inch, or

* MACHINERY, Railway Edition, April, 1906.

$$\frac{P}{D\,l'} = \text{or} < 1600 \qquad (26)$$

in which the letters have the same meaning as in the case of Formula (25), except that l' = length of bearing on the pin.

The side-rod crank-pins will never have any greater pressure brought upon them than that needed to slip the pair of wheels to which they belong so that the general principle is established that the maximum stress to which a side-rod pin can be subjected is that required to slip

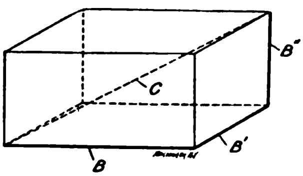

Fig. 17. Crank-pin Stress Diagram

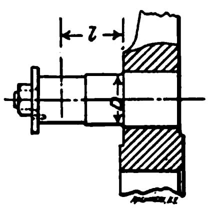

Fig. 18. Dimensions for Formula (25)

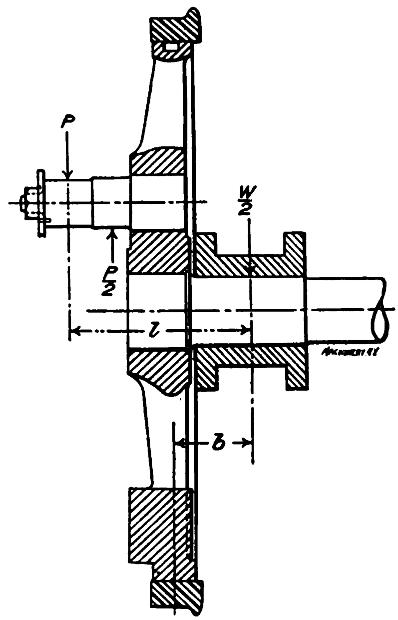

Fig. 19. Dimensions for Formulas (28) to (32)

the wheels to which it belongs. This may be calculated by the formula

$$P' = \frac{0.3\,W'\,D}{g} \qquad (27)$$

in which

P' = the frictional resistance of the driving wheels against the rails, transferred to the crank-pin,

W' = weight of the wheel with its load on the rails,

D = diameter of the driving wheels, in inches,

g = stroke of the piston, in inches.

In order to find the diameter of the pin, the value of $P' \times l'$ is substituted in Formula (25) for $P \times l$. In this the fiber stress should be placed at 12,000 pounds for a side-rod pin instead of the 16,000 pounds used with the main pin, inasmuch as certain conditions, for which it was allowable there, do not exist here in the case of the side-rod pins. Having found the diameter required to meet the required fiber stress, the bearing surface may be proportioned according to Formula (26) substituting, however, the value of P' as given in Formula (27). It will be observed that the values of P and P' as given in these three formulas indicate the highest stress that can be applied to the pins, so that the working pressure of 1600 pounds per square inch can only be exerted at very low speeds when the full boiler pressure is put upon the piston.

It is quite apparent, without argument, that the driving axles are among the most important of the details of a locomotive. Like the main rod, they are subjected to many and varying forces, such as the horizontal and vertical bending moments and torsion. The vertical bending moment, when considered as static, is the smallest to which the axle is subjected and is readily calculated. On the other hand, the shocks to which the part is exposed at high speeds are almost impossible to determine. At the same time the torsional stress under those same conditions, is comparatively slight, so that the increase of one may be considered to compensate for the decrease in the other. The horizontal bending force is due directly to the pressure exerted by the piston. The same is true of the torsional stress, whereas the vertical bending moment is due to the weight on the axle.

The torsional stress is never in excess of P' in Formula (27) since anything above this is transferred through the side rods to the other drivers, as one crank is in a favorable position to slip the wheels, when the other is on the dead center, thus taking up all of the slack in the rods, and relieving the axle from the full bending force of the piston. It is customary, therefore, to consider this bending moment as equal to one-half that exerted by the piston or $\frac{P\,l}{2}$, and the formula for the horizontal bending moment alone would be

$$B = \sqrt[3]{\frac{32\,P\,l}{2\,\pi\,S}} = \sqrt[3]{\frac{5.1\,P\,l}{S}} \qquad (28)$$

in which

$B =$ the required diameter of an axle that would resist the horizontal bending moment,

$P =$ the piston pressure,

$l =$ distance between center of main rod and center of journal box,

$S =$ allowable fiber stress.

The weight on the axle is a constant load and must, therefore, always be taken into consideration in connection with the horizontal bending moment, whereby the resulting bending moment of these two forces is equal to the hypotenuse of a right-angled triangle of which they are, themselves, the two sides. Consider, then, the total load on the axle as equal to W and the horizontal distance between the center of the rail and the center of the driving box as b, as indicated in Fig. 19. The vertical bending moment for $\frac{W}{2}$ or that for each box

13.

will be obtained when

$$B' = \sqrt[3]{\frac{32\,W\,b}{2\,\pi\,S}} = \sqrt[3]{\frac{5.1\,W\,b}{S}} \qquad (29)$$

in which

B' = the vertical bending moment.

When the crank stands on the quarter the torsional stress is to be added and may be expressed in the same way as the bending forces. If this stress of torsion is indicated as B'', we then have

$$B'' = \sqrt[3]{\frac{16\,t\,r}{2\,\pi\,S}} = \sqrt[3]{\frac{5.1\,t\,r}{2\,S}} \qquad (30)$$

in which

t = the resisting force at the crank-pin to the turning or slipping of the wheels,

r = radius of crank, in inches.

This value t is to be found in the same manner as that of P' in Formula (27).

Thus

$$t = \frac{0.3\ W'\ D}{g} \qquad (31)$$

in which

W' = total load of onè pair of wheels on the rail,

D = diameter of drivers, in inches,

g = stroke of piston, in inches,

0.3 = coefficient of friction.

It will be noted, of course, that all of the power required to turn or slip a pair of wheels does not go through the axle. One-half is absorbed directly by the wheel to which the crank is applied, and the other half, or only enough to turn the wheel at the opposite end of the axle, goes through the latter, thus reducing the torsion by one-half, a condition that is provided for in Formula (30) by the constant 2 in the demoninator of the general formula for torsion.

By combining Formulas (28), (29), and (30), it is possible to determine the required diameter D of the axle, thus:

$$D = \sqrt[3]{\sqrt{\left(\frac{5.1\,P\,l}{S}\right)^2 + \left(\frac{5.1\,W\,b}{S}\right)^2 + \left(\frac{5.1\,t\,r}{2\,S}\right)^2}}$$

This reduces to the form

$$D = \sqrt[3]{\frac{5.1}{S}\sqrt{(P\,l)^2 + (W\,b)^2 + \left(\frac{t\,r}{2}\right)^2}} \qquad (32)$$

which is preferable as decreasing the size of the number to be squared.

The whole problem can be indicated graphically by measuring the diagonal C of a parallelopiped, where the value of B obtained in Formula (28) is the length; that of B' in Formula (29) the width, and that of B'' in Formula (30) the height, as shown in Fig. 17.

In applying Formula (32) to the determination of the diameter of the axle for the consolidation locomotive, the following values for the several symbols will be obtained:

$P = 69{,}270$, or in round numbers 70,000 pounds,

$l = 22$ inches,

$W = 40{,}000$ pounds less the weight of the wheels and axles or 32,000 pounds,

$b = 10$ inches = horizontal distance from center of rail to center of box,

$$t = \frac{0.3\, W'\, d}{g} = \frac{0.3 \times 40{,}000 \times 57}{26} = 26{,}000 \text{ pounds,}$$

$r =$ radius of crank $= 13$ inches,

$S =$ allowable fiber stress $= 16{,}000$ pounds.

With these values the diameter of the axle (D) becomes

$$D = \sqrt[3]{\frac{5.1}{16{,}000}\sqrt{(70{,}000 \times 22)^2 + (32{,}000 \times 10)^2 + \left(\frac{26{,}000 \times 13}{2}\right)^2}}$$

This may be simplified by cancellation if written

$$D = \sqrt[3]{\sqrt{\left(\frac{5.1 \times 70{,}000 \times 22}{16{,}000}\right)^2 + \left(\frac{5.1 \times 32{,}000 \times 10}{16{,}000}\right)^2 + \left(\frac{5.1 \times 26{,}000 \times 13}{2 \times 16{,}000}\right)^2}}$$

and

$$D = \sqrt[3]{\sqrt{\left(\frac{5.1 \times 70 \times 11}{8}\right)^2 + (5.1 \times 2 \times 10)^2 + \left(\frac{5.1 \times 13 \times 13}{16}\right)^2}}$$

$$D = \sqrt[3]{\sqrt{(491)^2 + (102)^2 + (53.8)^2}} = \sqrt[3]{\sqrt{241{,}081 + 10{,}404 + 2894}}$$

$$D = \sqrt[3]{504} = 7.96 \text{ inches.}$$

By making a suitable allowance for wear and truing, the axle is made 9 inches in diameter at the journal.

The front and rear axles or all except the main driving axle may, of course, be made smaller, but it is customary to make them all of the same diameter, though exceptions to this rule are frequent. The same formulas may be applied to their determinations by the substitution of the proper values for the several symbols. Attention should be called to the fact that the horizontal bending moment, which is the greatest in the case of the main axle, is reduced to less than one-third on the other axles of a consolidation locomotive.

Turning now to the crank-pin and applying Formulas (25), (26), and (27) to the consolidation locomotive, the bending forces of the main pin at the wheel seat will give a diameter

$$D = \sqrt[3]{\frac{10.2\, P\, l}{S}} = \sqrt[3]{\frac{10.2 \times 70{,}000 \times 8.375}{16{,}000}} = \sqrt[3]{373} = 7.19 \text{ inches.} \quad (25)$$

The diameter of this crank-pin is made 7¼ inches, which is quite sufficient when the liberal allowance made in the formula for the pressure on the piston is considered, as this is rarely reăched, and the statement is emphasized by the fact that no allowance has been made for the reaction of the side rod when the wheels are slipped into bearing by the opposite crank, which is on the quarter at the time the maximum stress comes on the one at the dead center where the calculation is based.

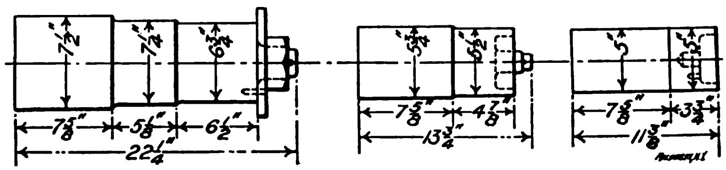

Fig. 20. Crank-pins used on Consolidation Locomotive

This diameter will be that of the side-rod bearing which, because of its large diameter, can be made very short. The pressure will seldom be more than three-quarters that of the main pin, and usually will be less. However, it will be found that, in the case of this consolidation locomotive the bearings are made longer than the 1600 pounds pressure per square inch required, for reasons of construction. Thus

$$l = \frac{3 \times 70{,}000}{4 \times 7.25 \times 1600} = 4.5 \text{ inches approx.}$$

But for the reasons just given it is made 5⅛ inches long.

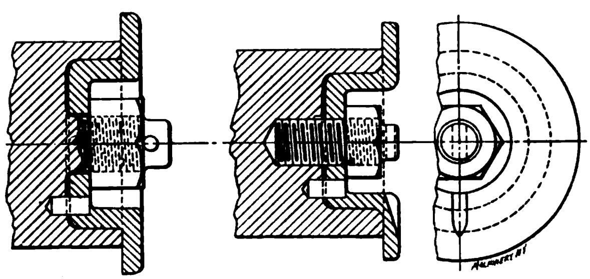

Fig. 21. Method of Attaching Crank-pin Collar

The pins are usually reduced in diameter for the main bearing, in order that the connections may be as light as possible. In the present case the projected area called for amounts to

$$D\,l' = \frac{P}{1600} = \frac{70{,}000}{1600} = 43.75 \text{ or a bearing about } 6.25 \times 7 \text{ inches.} \quad (26)$$

Taking Formula (27) for the side-rod pins we have

$$P' = \frac{0.3\,W'\,D}{g} = \frac{0.3 \times 40{,}000 \times 57}{26} = \text{about } 26{,}000 \text{ pounds}$$

and

$$\frac{26{,}000}{1600} = 16.2 \text{ square inches of bearing area}$$

and

$$D = \sqrt[3]{\frac{10.2\,P\,l}{S}} = \sqrt[3]{\frac{10.2 \times 40{,}000 \times 2}{16{,}000}} = \sqrt[3]{51} = 3.7 \text{ inches, or a}$$

4×4 inch pin. (25)

Here we have a variation in diameter in practice, demanded by the clearances at the front limiting the length of the pin which is accordingly made 5 inches diameter and 3¾ inches long.

As for the forms of the crank-pins used on such a locomotive as this consolidation engine, they are simple and are shown in Fig. 20.

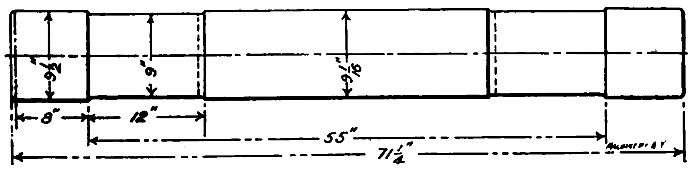

Fig. 22. Driving Axle for Consolidation Locomotive

In this the three forms shown are for the main, intermediate and front and back pins respectively. Fig. 21 shows the method of attaching collars at the ends so as to occupy a minimum of space. As all of the rods are practically fitted with solid bearings, it is necessary that the pins should have washers at the outer ends to hold them in place.

Fig. 22 illustrates the common form of driving axles used at present. There is an enlarged wheel seat at the end. The journals are inside this and are slightly smaller in diameter than the wheel seat, while the body of the axle at the center is slightly larger. Abrupt changes of diameter are avoided and square shoulders are not allowed. The same process in the determination of dimensions may be followed in the case of the passenger locomotive.

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CHAPTER VI

DRIVING WHEELS AND COUNTERBALANCING*

The driving wheels, on which the engine is carried and by which it is propelled, are subjected to stresses far beyond those due to the mere requirements of carrying the load put upon them or transmitting the thrust, exerted by the steam, into rotation and pull upon the rails. Among the first in importance of these secondary stresses are those due to the shrinkage of the tires upon the center. In order that the wheels may withstand these stresses, the rim should be heavy between the spokes, and the latter stiff enough to carry the load without bending. In the shrinking on of the tire under ordinary conditions an allowance of 0.010 inch to the foot is made. That is to say, the tire is bored out 0.010 inch smaller than the diameter of the center upon which it is to be placed for each foot of diameter of the latter. This allowance is modified somewhat according to the proportions of carbon entering into the composition of both the tire and the center.

When the center is made of cast steel in accordance with prevailing practice, it should be well annealed so that the greatest possible degree of toughness may be obtained. In effecting this annealing the utmost care should be taken to do the work uniformly throughout the whole extent of the material, and the same degree of hardness obtained as far as it is possible. It is essential that this should be done in order that the application of the tire may be successfully made, for the allowance for shrinkage must be met by the compression of the center and the elongation of the tire in such proportions that the latter will be securely held in place without overstraining either.

When soft material is used either in the tire or the center, a greater allowance must be made for shrinkage than where one or both are hard. These terms are, however, indefinite as to their limits, and there is, as yet, no standard of reference by which the work has been scientifically laid down. The result is that while the figures given for the proper shrinkage are those recommended and extensively used, experience and good judgment must enter into the work, and no hard and fast rule can be laid down that will meet all of the exigencies of varying conditions. In the design of the center, the spokes should be placed as close together as practicable and should be of such a section that they can withstand, without bending, the stress required to elongate the tire, in accordance with the amount of shrinkage allowed; this stress to be divided among all of the spokes.

If we place the shrinkage at 0.010 inch per foot of the diameter of the wheel, and assume that the center compresses as much as the tire elongates, there will be a total elongation of the tire of about 1/64 inch for each foot of circumferential length, and a corresponding com-

* MACHINERY, Railway Edition, May, 1906.

pression of the rim of the center. It is evident, then, that in order that such a compression may take place in the rim, the spokes must be able to be compressed a corresponding amount without changing form. This requirement does not determine the form of the section of the spoke but does influence its thickness. The width of the spokes laterally should be as great as other conditions will permit, as there is no means of determining the stress in this direction that may be brought to bear upon the wheel by a rough track, a derailment or the movement at high speeds around curves and over frogs and switches. Furthermore, the wheel should be dished as little as possible, because this, with the stresses induced by the shrinkage of the tire, tends to increase the transverse stress on the spokes.

If the strength alone were to be considered, the best form of spoke section would be that of an I-beam; but as this is impracticable, because cracking would occur in the cooling of the casting, the next best, or rectangular form, is used, rounded at the corners to an elliptic section. Again, an allowance must be made in the spoke lengths for rough usage and indeterminate stresses, at the same time using the utmost discretion in the work lest the weights run up to excessive amounts, as may very easily occur.

The hub should be of ample strength to resist the stresses to which it is subjected by the pressing in of the axle and crank-pin. It is essential, not only that there should be sufficient thickness to resist these unknown stresses but that this thickness should be as uniform as possible so as to avoid shrinkage cracks and unequal stresses due to the cooling of the metal after casting.

In boring for the axle and pin the work should be done with the utmost care and the dimensions so proportioned that the pressure required to force the wheel and axle together will be from 12 to 14 tons per inch of diameter of the axle. Where steel or wrought iron wheel centers are used, the fit should have a slight taper which will result in a better fit and less crushing of the surfaces of the material than when the fit is straight, when this may be so great as to exceed the elastic limit of the material. As a general rule the taper should be not more than 3/64 inch or less than 1/64 inch to the foot. Of the two, the larger taper is to be preferred where comparatively soft steel is used in the wheel centers; while, when the steel is hard, the smaller taper should be used. The taper for the crank-pin fit may be made 1/32 inch to the foot and the pressure regulated to the same amount per inch of diameter as in the case of the axle.

There is another detail in the designing of the wheels that is of great importance and which has received the closest examination for many years, and that is the counterbalancing. To do this all parts connected to each crank-pin should be carefully calculated, or better still, weighed, if it is possible to do it before the counterbalance weights are determined.

In doing this, the side rods, the rear end of the main rod, together with all pins and straps, are to be considered as revolving parts and the weight of each assigned to the wheel and pin with which they are connected. An approved method of weighing is to couple the side rods together and, supporting each bearing on a knife-edge with the rod horizontal, take the weights on these knife-edges in succession. The weight of the end of the main rod is obtained in the same way. In addition to these revolving weights those of the reciprocating parts, including the front end of the main rod, the crosshead, piston and piston-rod must be taken.

Of these the counterbalance in the wheel must be the equivalent of all of the revolving parts and a portion of the reciprocating. Regarding the latter the rules and practice have not yet taken such shape that a fixed proportion of the reciprocating parts to be counterbalanced is established. In this the designer is between two evils. The greater the proportion of reciprocating parts that are counterbalanced the smaller will be the longitudinal motion of the engine, but the greater will be the vertical disturbance. This puts an excessive pressure on the rail when the counterweight passes the lower center and tends to lift the wheel when it passes the upper; while, on the other hand, if too little of the reciprocating parts are counterbalanced, there will be an excessive longitudinal motion or nosing of the machine. As a matter of fact, an attempt is made to strike a happy medium between these extremes by which the minimum vertical effect on the rail is produced with the maximum longitudinal disturbance that can be tolerated.

To accomplish this, it is well to take the weight of the whole engine into consideration when calculating that of the counterbalance; for the relation between the two has a marked influence on the perceptible horizontal disturbance. Hence it comes about that the question has to be reversed and decided along the lines of what proportion of the reciprocating weights, that of the engine will permit to be left unbalanced.

The rule adopted by the American Railway Master Mechanics Association is to allow 1/400 part of the weight of the engine to remain unbalanced in the reciprocating parts. This, while entirely empirical, works well in practice and is quite generally used throughout the United States. It is, however, modified in some instances by limiting the counterweights to from 55 per cent as a minimum to 65 per cent as a maximum of the weights of the reciprocating parts for road engines, though the practice in this respect is not universal. This weight is equally divided among all of the driving wheels of the locomotive. In case the wheel centers are so small that there is not room to put the whole of the counterbalance that should be apportioned to the main driver on the side opposite the crank-pin, and get it within a reasonable area that does not exceed the area of half of the half-circle, the remaining wheels should not be balanced to compensate therefor, unless the extra weight to be balanced is less than 65 per cent of the weight of the reciprocating parts divided by the numbers of wheels.

This may be expressed by the formula

$$B = \frac{0.65\ W}{n} \qquad (33)$$

in which

B = the maximum allowance balance weight which is added above that needed to counterbalance the revolving parts,

W = the total weight of the reciprocating parts,

n = the number of coupled wheels on each side of the engine.

If this does not make up for the deficiency of balance of the main wheel, the total balance should be left that much short, and the maximum speed limit of the engine should be correspondingly reduced.

In designing the wheel, it is best to cast the counterweight in solid with the center. Should the weight be too great for this, pockets may be cast in symmetrically on the side opposite the crank to be filled with lead in the final adjustment, since by the use of lead, a greater weight can be obtained in a smaller compass than with iron. In other cases the major portion of the weight may be cast in, and the deficiency made up by the addition of lead in the final adjustment.

This is usually done after the axle and crank-pin have been pressed in. The journals of the axle are placed upon horizontal straightedges, and the wheel turned so that the crank-pin is on a horizontal line with the center of the axle. A weight is then hung upon the pin by a ring somewhat larger in diameter than the pin so as to secure a central bearing. This weight should be equal to the sum of the weights of the ends of the side rods that will be attached to the pin, and the proportion of the reciprocating weights that is to be apportioned to the pin. In the case of the main pin the weight of the rear end of the connecting-rod is to be added.

The reciprocating weight to be balanced on each wheel may be found by the formula:

$$r = \frac{R - \dfrac{W}{400}}{n} \qquad (34)$$

in which

r = the reciprocating weight to be balanced at each wheel,

R = the total weight of the reciprocating parts,

W = the total weight of the engine in pounds,

n = the number of coupled wheels on each side.

Still it must be borne in mind that this weight is subject to the conditions imposed previously, that the total weight counterbalanced or $R - \dfrac{W}{400}$ should lie between 0.55 R and 0.65 R and r will be the same for each wheel.

As for the driving boxes, their dimensions, like so many other parts of the locomotive, are not determined by any special rule except that in a general way they are proportioned according to the diameter of the journals, compounded with those indispensable requisites for good designing in all branches of mechanics—experience and good judgment. The prime requisites are that they shall have sufficient strength to sustain the load that they have to carry and sustain the pressures to which they may be subjected in the working of the engine. Nearly all of these forces partake of the nature of compression stresses, which, of themselves, would not seriously strain the material, but they are coupled with shocks and blows both lateral and longitudinal, especially when the wedges become worn or loose in service.

Here again the importance of reducing all weights to a minimum manifests itself and so cast steel has come to be the usual material used for driving boxes. The crown brass is usually made with a thickness at the top equal to one-quarter the diameter of the axle and is forced into place by a hydraulic pressure of from 20 to 25 tons. The length of the bearing is governed by the diameter of the journal and

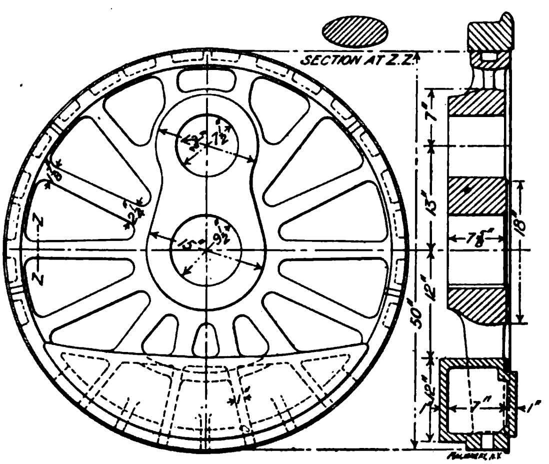

Fig. 23. Cast Steel Driving Wheel Center for Consolidation Locomotive

the weight that it has to carry and is so proportioned that on fast passenger engines the load carried does not exceed 170 pounds per square inch of projected area, and 180 pounds for freight engines, while 190 pounds per square inch can be allowed on switching engines.

In order to meet these conditions it frequently becomes advisable to increase the diameter of the axle by a fraction of an inch, above the calculated requirements, so that the bearing may be shortened and the center of the box be brought centrally beneath the springs. There should be sufficient side play between the flanges of the boxes and the wedges to permit one box to rise until it strikes the frame while its mate at the other end of the axle remains down in its normal position. The reason for this is mainly that the flanges may not be broken in case a spring breaks on one side.

The oil cellar is usually a simple cast-iron box fixed to the lower shanks of the driving box. It is held in position by horizontal pins that may be easily removed so that the cellar may be lowered for repacking or replacing the pads, which should always be high enough so as to press against the axle journal and thus supply lubrication for a time should the feed from the top of the box be cut off.

Turning now to the practical application of these principles, Fig. 23 illustrates a cast steel driving wheel center for the consolidation locomotive in which pockets are cast in the counterbalance for the use of lead filling, and Fig. 24 shows a wheel of the same material, but of larger diameter intended for the Atlantic engine with the counterbalance cast solid with the rim.

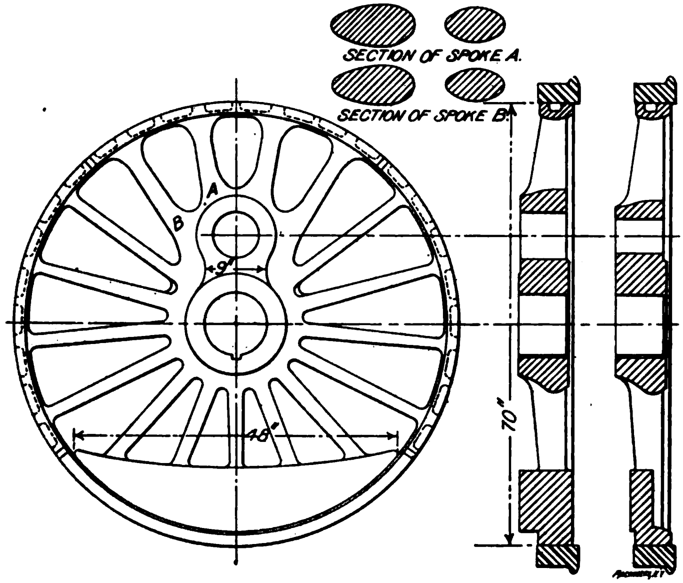

Fig. 24. Cast Steel Driving Wheel Center for Atlantic Type Locomotive

In the former case the weight to be counterbalanced on the main driver is, according to the rule of the Master Mechanics Association:

$$966 - \frac{176,000}{400} = 526 \text{ pounds,}$$

and for the Atlantic engine

$$1057 - \frac{168,000}{400} = 637 \text{ pounds,}$$

in both of which cases the weights are from 55 to 65 per cent of the weights of the reciprocating parts which are 966 pounds and 1057 pounds respectively.

Again, if it should have so happened that there had not been room on the main wheel for this amount of counterbalance, the distribution of the weights among the other wheels would have been, according to Formula (34), for the consolidation engine

$$r = \frac{966 - \dfrac{176,000}{400}}{4} = 131.5 \text{ pounds,}$$

and for the Atlantic engine

$$r = \frac{1057 - \dfrac{168,000}{400}}{2} = 318.5 \text{ pounds.}$$

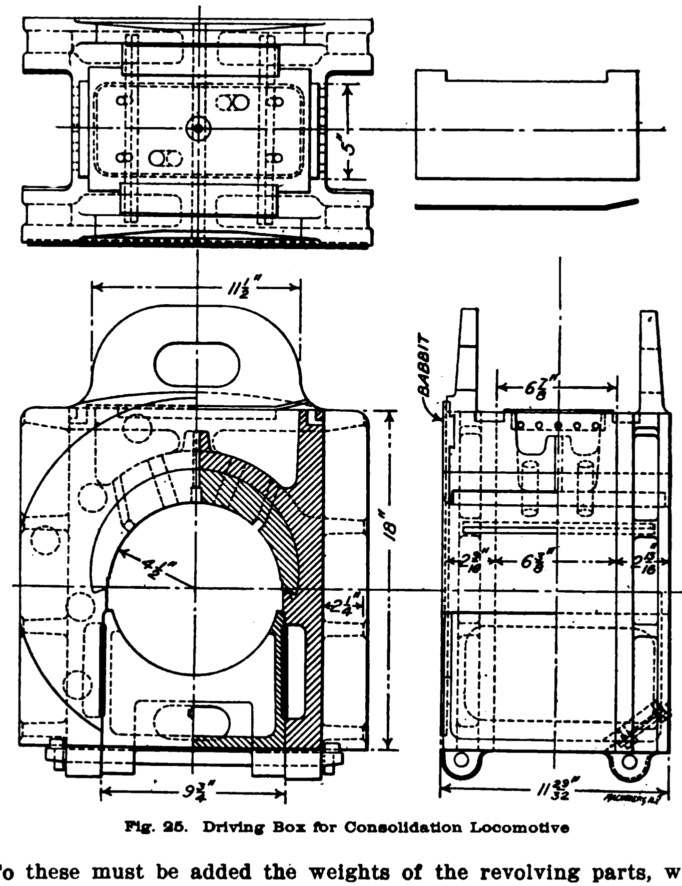

Fig. 25. Driving Box for Consolidation Locomotive

To these must be added the weights of the revolving parts, which, aside from the crank hubs, are on the consolidation engine for

the forward pin.............................. 100 pounds
the main pin.................................. 545 pounds
the third pin................................. 215 pounds
the rear pin.................................. 100 pounds

in the case of the Atlantic engine for

the forward pin.............................. 140 pounds
the main pin.................................. 490 pounds

All of these weights must be reduced in proportion to their distance from the center of the axle as compared with that of the crank-pin.

For example, if the center of gravity of the counterbalance of the main driver of the Atlantic engine is two and one-half times the distance of the crank-pin from the center of the axle, the actual weight to be used becomes

$$\frac{490 + 637}{2.5} = 450.8 \text{ pounds.}$$

Fig. 25 shows the form of box that is used for the consolidation locomotive and the same general features obtain in that for the Atlantic. Particular attention is called to the form of the flanges, where they bear against the wedges. They are tapered from the center to the ends, the distance between them widening, so that they can have a rocking motion without binding, and thus fulfill the conditions imposed that the box shall be able to rise and strike the frames and not bind while its mate is in the normal position. It will be seen that the flanges are quite heavy and that the depth is sufficient to give a good bearing on the wedge faces. The length of the bearing is made 11¾ inches long, and as the diameter of the axles is 9 inches, this makes 105¾ square inches of projected area of bearing. As the weight on the drivers is 19,375 pounds, from which must be deducted the weight of the wheel and one-half the axle, which will amount to at least 3000 pounds, the load on the journal falls within the limit of 180 pounds per square inch, or will be about 155 pounds per square inch, a margin that leans to the side of safety and cool running.