LOCOMOTIVE DESIGN AND CONSTRUCTION

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OF MECHANICAL ENGINEER

BY

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PREFACE

This work is prepared to meet a need which has arisen due to the inaccessibility of a large amount of the existing information concerning locomotive design and construction.

Improvements in locomotive practice are being made as rapidly as in any other field of engineering work. The desire on the part of the government and many individuals to standardize locomotive practice may, in some respects, curtail advancement that has been made up to the present time but the demand of the railroads is ever for heavier, more powerful, more economical, and more reliable locomotives. The past ten years has seen greater development and advancement per ton mileage per locomotive than any past period. The advantages obtained from standardization are by far offset by the advantages gained in improvement and development of the locomotive. Each railroad has, in the past, had its own standards and types of locomotives for its individual use. This condition has led to a vast variety of types of locomotives but there is a vast variety of conditions which must be met by locomotives in their operation. Locomotives must be operated and built to suit the roads on which they are run. A locomotive which is economical on one run may not be on another.

Thus it may be seen that the development of various types and styles by various builders has its advantages.

There have been a great many books, bulletins, handbooks, reports, and magazine articles written on subjects pertaining to locomotive design and construction. A large part of locomotive design has been standardized but this information is very difficult to obtain as no one of these deals with more than one or a few subjects concerning the locomotive. There are so many broad subjects on which a vast amount of experimental work has been done that it is difficult to write without going into specific detail which cannot of course be done in a work of this sort. My idea is to give the information which is essential in order to proceed with technical questions arising in the design and construction of locomotives. The text presupposes definite knowledge of the details of construction of the locomotive. This work cannot go into details sufficiently to give one with no knowledge of the construction of the locomotive a thorough understanding of the subject.

In this work the definite information needed to plan a locomotive suited to meet the demands of trackage and haulage on any road or for any type of service will be found. It also contains the information necessary to write the specifications which must be furnished the builders of such locomotives and that necessary for the design of the locomotive by the locomotive builder to suit the the specifications given and the methods used in building the locomotive and making it ready for operation. The whole process, from the beginning to the final product is here given.

It can be seen that this is a vast subject to be covered by one work but a locomotive builder must be familiar with every step here given and it is very essential that no part be omitted in the consideration of the problem in hand. My purpose, then, is to put this information in compact and easily accessible form.

James Clyde Maris.

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TYPES OF LOCOMOTIVES.

The service which locomotives are called upon to give in practice may be divided into three classes: road service, switching service, and industrial service. Each of these three classes of service calls for many different types and sizes of locomotives, depending of course upon the track upon which the locomotive is operated; fuel and water supply; and length, tonnage and schedule of haul.

Road service is divided into two classes: freight and passenger. As a general rule passenger service demands greater speed and less tonnage. Passenger service is frequently subdivided into heavy passenger and fast passenger and the same is true of freight service, the latter service demands being very different for the various classes of freight hauled. Mountainous country demands other types of locomotives than are used in ordinary practice in level country, this service being over roads with heavy grades, sharp curves, and frequently long hauls, Suburban service in densely populated communities demands still other types for quick starting and light, short distance hauls. Such classes of service demand locomotives of a much different type than are used in ordinary railroad practice. Branch and stub lines usually take care of the discarded motive power from the trunk lines. Frequently discarded passenger and freight engines are converted into switchers.

Switching service demands heavy tractive effort with short, quick runs on sharp curves. The fact that switching engines usually have the entire load on the drivers accounts for their increased tractive effort.

Industrial service may be divided into a great many different classes as light locomotives are used in a great many fields, such as: mining, logging, plantation work, construction work of various kinds, around large mills and factories, at docks and shipyards. Locomotives for all of these various classes of service are more or less standardized, economy of operation being of minor consideration. It is customary for purchasers of these locomotives to buy standard types of the manufacturer, usually no attempt being made to standardize the locomotives being used by any one concern. This makes the manufacture of such locomotives much less complicated than the manufacture of locomotives for road service. Variation in design usually is limited to weight, power, and clearance limits.

Locomotive nomenclature for various types of service is standard in American practice. The Whyte Locomotive Nomenclature is as follows:

(1) Mogul	2- 6-0	Light Freight
(1) Prairie	2- 6-2	Heavy Passenger-Fast Freight.

(1)Consolidation	2- 8-0	Freight
Mikado	2- 8-2	Freight
(1)Atlantic	4- 4-2	Fast Passenger
(1)American	4- 4-0	Ordinary Passenger
(1)Pacific	4- 6-2	Fast and Heavy Passenger
Forney 6 Coupled	0- 6-4	Switching
Forney 4 Coupled	0- 4-4	Switching
Decapod	2-10-0	Freight on Heavy Grades
Articulated	- - -	Freight on Heavy Grades
Twelve-wheel	4- 8-0	Freight
(1)Ten-wheel	4- 6-0	Heavy Passenger-Fast Freight
(1)Six-wheel	0- 6-0	Switching

About 95% of the locomotives used are of the above types marked (1). From 30 to 35% of all locomotives used at the present time are of the Consolidation type. For passenger service the Pacific type with a tractive effort of from 40,000 to 45,000 pounds is ordinarily used for level country. For mountain work the Mikado is becoming popular. In freight service the Mikado type is now largely used in place of the Consolidation where a tractive effort of from 50,000 to 60,000 pounds is required. On heavy grades with heavy loads the Santa Fe type with a tractive effort of from 65,000 to 85,000 pounds is most popular. In mountain freight service the Mallet Articulated locomotives are used. In switching service the 0-6-0 and 0-8-0 are most commonly used, the tractive effort being up to 75,000 pounds or more. Industrial service demands locomotives of the 0-4-0 and 0-6-0 types, of all the way from 6 inch by 10 inch cylinders to 17 inch by 24 inch cylinders. Frequently industrial locomotives call for either front or rear trucks or both, depending, of course, on the service.

Practically all locomotives for road and switching service have tenders used in connection with the locomotives while industrial locomotives, as a rule, carry their fuel and water supply on the engine itself by means of tanks and coal boxes.

The road bed on which a locomotive must be used determines to a very large extent the kind of a locomotive which can be used upon it. The guage of the track, the weight of the rails, the ballasting of the roadway, the strength of the weakest bridge in the line, clearances of obstructions near and over the track, length and degree of the grades, and the degree of the sharpest curve are factors which must be determined and considered before attempting to specify the locomotive which can be used. The strength of the rails will determine the maximum wheel load that can be used. The sharpest curve determines the running gears. The maximum grade and speed which the locomotive must hold to, to keep up schedules, determine the tractive effort for a given tonnage. The weakest bridge in the line will determine the outside dimensions of the locomotive.

The locomotive must be operated on fuel which is obtainable near the location in which it is used. For this reason there are many varieties of fuel used, such as: various kinds of coal, oil, and wood. The kind and quality of the fuel determines the size and type of the grates, fire-boxes, front end arrangement in the boiler, and the stack.

The water supply will determine the necessary quantity of water which must be carried in order to make a run between watering places. Also the quality of the water, whether it contains mineral salts, acids or other impurities, should be considered, in order to know just what should be provided for.

The length of a run which a locomotive is called upon to make, the load which it must pull, and the rate which must be maintained in order to hold to the schedule determine the necessary tractive effort which a locomotive must exert and the fuel and water which must be carried. In order that a locomotive may perform these functions as as desired all of these factors must be considered.

TRAIN RESISTANCE

In order to determine the size and type of locomotive which should be used in any particular case it is first necessary to know just what load it will be called upon to pull and what speed it must maintain while pulling this load. The resistance which a train will meet depends of course upon the various conditions of the track, the weight and number of wheels of the train, the necessary rate of acceleration, and the rate at which the train must be pulled.

The factors which go to make up the resistance met by a train are as follows: resistance due to wind pressure; resistance due to friction on a straight, level track; and the additional friction due to grades and curves. The resistance met in starting a train must also be considered in getting the necessary tractive effort which a locomotive must give.

There has been a great deal of experimental work done to determine just what train resistances are in various cases. These results may be used in order to calculate the resistance which a proposed train will meet.

Frequently train resistances are determined by using a dynamometer car connected between the locomotive and the train. In this way the exact resistances for all conditions may be determined accurately and a locomotive designed to suit the conditions. But where it is impossible to use such a car one must calculate the resistances from the known information here given:

Wind resistance is indeterminate.

Curve resistance is found to be from .5 to .8 lbs. per ton per degree of curve.

Grade resistance:

Momentary resistance at a point: Rg=G .379

Average resistance over a section: $Rg=2000(E_2-E_1)\div S$

Rg=Resistance due to grade in pounds per ton.

G=Grade in feet per mile.

E_1=Elevation of mass of train in feet at the start.

E_2=Elevation of mass of train at the end.

S=Length of track.

Resistance due to the acceleration of the train can be found by means of the following formula:

$$Ra = \left(91.05 + 145.5\left(\frac{N}{W}\right)\right) A$$

N=Number of cars in train.

W=Total weight of train in tons.

A=Acceleration in miles per hour per second. This is in two parts: accelerating the train as a whole and accelerating the wheels around.

$$A = .733(V_2^2 - V_1^2)\,S$$

V_1 and V_2 represent the speed in miles per hour.

Freight train resistance:

$$R = (V + 39.6 - .031w) \div (4.08 + .152w)$$

w=The average gross weight of the cars.

R=Net resistance on level straight track in still air in pounds per ton. This formula is empirical.

The net resistance for trains may also be found from the following:

Momentary resistance:

$$R = \frac{P}{W} - .379 - \left[91.05 + 145.5\left(\frac{N}{W}\right)\right] A.$$

Average resistance over a section:

$$R = \frac{P}{W} - \frac{2000\,(E_2 - E_1)}{S} - \left[91.05 + 145.5\left(\frac{N}{W}\right)\right] A.$$

P=Total gross resistance or draw-bar pull.

W=Total weight in tons.

N=Number of cars.

A=Acceleration in miles per second.

This resistance does not include the resistance due to wind, grade acceleration, and curvature resistance.

Another rule which has been worked out is as follows:

Net resistance for passenger trains is:

$$R = 5.4 + .002(V-15)^2 + 100 \div (V+2)^3,$$

V=Velocity in miles per hour.

R=The pounds per ton for the heaviest cars, on good track and the best conditions.

For ordinary cars and common track the net resistance is:

$$R = 4 + .1V + .278\left(\frac{V^2}{W}\right).$$

The Baldwin Data Book gives the following:

The resistance due to curvature is .7 of a pound to 1 pound per ton per degree of curvature, 1 pound for light loaded cars and .7 of a pound for heavy loaded cars. This is on account of the wheels per ton.

The train resistance as found by experiment on the I.C.R.R.:

$$R = 1.8T + 100N.$$

R=Total resistance in pounds for the train not including the engine and tender.

T=Total weight of the train.

N=Number of cars in the train.

In order to determine the tractive effort or draw-bar pull which a locomotive must give, calculate the maximum train resistance which it will be called upon to overcome and the result is the necessary tractive effort.

TRACTIVE EFFORT

The tractive effort which a locomotive can exert determines the size of the train which it can handle. By tractive effort is meant the pull on the draw-bar which a locomotive can make. This tractive effort is dependent on several elements, all of which should be carefully considered in the design of a locomotive to suit specified conditions.

The elements which determine tractive effort are: the style of the locomotive, its weight on the drivers, and the condition of the track on which it is operated. A great deal of experimental work has been done in order to determine the tractive effort which various locomotives can exert under various conditions. It has been found that the locomotive with the necessary power can pull on dry, sanded rails one third of its weight on the drivers; on dry clean rails it can pull one fourth of its weight on the drivers; and on wet rails one fifth of its weight on the drivers. With frost or snow and ice on the rails only one sixth of its weight on the drivers can be pulled. For this reason it can be seen that the ratio between power of the engine and the weight on the drivers should be held to suit conditions. This ratio is in practice usually made to suit dry, clean rails.

Having determined the train resistance which a locomotive will be called upon to meet, the size of the locomotive can be determined by means of the following formulae:

For simple engines the tractive effort can be found from the following:

$$T = \frac{d^2 \times S \times .85P}{D}.$$

d=The diameter of the cylinders.

S=Stroke of the piston.

P=Boiler pressure.

D=The diameter of the drivers in inches.

.85=The mean effective pressure of the boiler.

For two cylinder compound locomotives the tractive effort may be found by means of the following:

$$T = \frac{d_1^2 \times S \times .66P}{D}.$$

d_1=The diameter of the high pressure cylinder.

The maximum tractive effort for four cylinder locomotives is found by means of the following:

$$T = \frac{d_1^2 \times S \times 1.6P}{(R+1) \times D},$$

R=Area of the low pressure cylinder divided by the area of the high pressure cylinder.

When all the cylinders are working on direct steam the tractive effort can be found by means of the following:

$$T = \frac{d_1^2 \times S \times 1.6P}{D}.$$

The tractive effort of a Mallet Compound:

$$T = .52\, d_L^2 \left(P \frac{S}{D}\right).$$

d_L=Diameter of low pressure cylinder.

The tractive effort of a four cylinder compound working compound is also found from the following:

$$T = \left[(2 d_1^2 \div 3) + (d^2 \div 4)\right] \frac{PS}{D},$$

The tractive effort of two cylinder compound locomotives is found from the following, working simple:

$$T = .80 d_1^2 \left(\frac{PS}{D}\right).$$

The tractive effort of the Mallet compound:

$$T = \frac{d^2 \times S \times 1.6P}{\left(\frac{\text{Ratio of cyl.}}{\text{Volumes}}\right) + 1 \times D},$$

The tractive effort of the Vaulclain balanced compound:

$$T = \frac{S \times P}{D} \left[\left(\tfrac{2}{3} d_1^2\right) + \left(\tfrac{1}{4} d^2\right)\right],$$

Cross compound and also Mallet:

$$T = \frac{d^2 \times S \times 1.2P}{D}.$$

For the cross compound divide the above by two.

In the locomotives of recent date the tractive effort divided by the weight on the drivers varies from .18 to .25 for all classes of locomotives. The Railway Master Mechanics Association recommends the following ratios for standard practice in design:

.25 for passenger locomotives. ,

.235 for freight.

.22 for switching.

The maximum tractive effort is obtained only with maximum steam pressure, full throttle, maximum cut-off, and at a speed of from 6 to 8 or 10 miles per hour. As the speed increases the tractive effort decreases. This is the natural result of the earlier cut-off at the higher speeds.

The internal resistance uses up from 8% to 10% of the total power developed.

The formulae above given are developed from the old common "P.L.A.N." formula so commom in steam engine practice.

In the use of the above formulae there are a number of factors, such as the diameter of the drivers, dimensions of the cylinder, and steam pressure, which must be determined before getting the necessary weight of the locomotive but these factors may be determined to some degree from the style of locomotive which is wanted.

RUNNING GEARS

The running gears of a locomotive must be arranged to suit the track upon which it runs. The factors which determine the running gears are: the strength of the rails, the curves in the track, and the speed at which the locomotive is run.

By running gears is meant the wheels upon which the locomotive is carried. These consist either of a system of driving wheels alone, which will make up a rigid wheel base, or driving wheels in combination with front or rear trucks or both. The trucks are so arranged that they swing on either side of the center line of the locomotive, carrying part of its load and serving to guide the locomotive around curves.

The number of wheels which must be used usually is determined by the strength of the rails. The strength of the rails is proportional to the weight of the rails. Each ten pounds weight per yard, properly supported on ties, is capable of sustaining a safe load per wheel of 3000 pounds.

The curvature of the track determines the length of the rigid wheel base which can be used. This, of course, is because the flanges on the wheels must bear against the side of the rails and not on top.

The length of the rigid wheel base which can be used may be determined from the following formula:

$$R = \frac{.7646W}{2P}, \quad \text{in which}$$

R=The radius of the curve in feet.

P=The play in the driving wheels.

W=The rigid wheel base.

The center line of a locomotive is perpendicular to the radius of the track at the mid-point of the rigid wheel base and the same holds true with four wheel trucks. For pony trucks the axle always remains radial with the curve.

The swing each side of the locomotive which a truck must have may be determined from the following formula:

$$S = \frac{(T-W) \times T}{2R}, \quad \text{in which}$$

S=The swing each side of center.

T=The total wheel base from the rear driver to the center-pin of the truck.

W=The rigid wheel base.

R=The radius of the curve, all dimensions being in the same units.

For small locomotives which must run on very crooked tracks, such as are found in many classes of industrial work where the radii of the curves are 50 feet or less, the swing of the truck should be calculated a little more accurately. For four wheel trucks the swing may be **found** from the following formula:

$S = \sqrt{R^2 + B(W+B)} - R$, in which

W=The rigid wheel base.

B=The distance from the center of the front driving wheel to the center-pin of the truck when the engine is on a straight track.

S=One half the total swing of the truck.

R=The radius of the track,all dimensions being in the same units.

For a pony truck the following formula is used:

$S = \frac{db^2 + bcR}{b^2 + R^2}$, in which

S=One half the total swing of the truck.

d= $\frac{1}{2}\sqrt{4R^2 - W^2}$

b=The length of the radius-bar= $\frac{(W+B)B}{W+2B}$

c= $\frac{W}{2} + a$

R=The radius of the track.

W=The length of the rigid wheel base.

a=The distance from the center of the front driving wheel to the radius-bar pin.

B=The distance from the center of the front driving wheel to the center-pin of the truck when the engine is on a straight track.

In order that the trucks may swing properly the length of the radius-bar must also be determined from the curve of the track and the distance from the front drivers to the truck center-line, the swing which the truck must have each side of the center, and the length of the rigid wheel base. This radius-bar may be determined from the following formula:

$$L = \frac{T^2}{2T+W} + 3 \text{ or } 4 \text{ inches,}$$ in which

L=The distance from the center of the front drivers to the radius-bar pin.

T=The distance from the front drivers to the center line of the truck.

W=The rigid wheel base.

R=T-L=The length of the radius-bar.

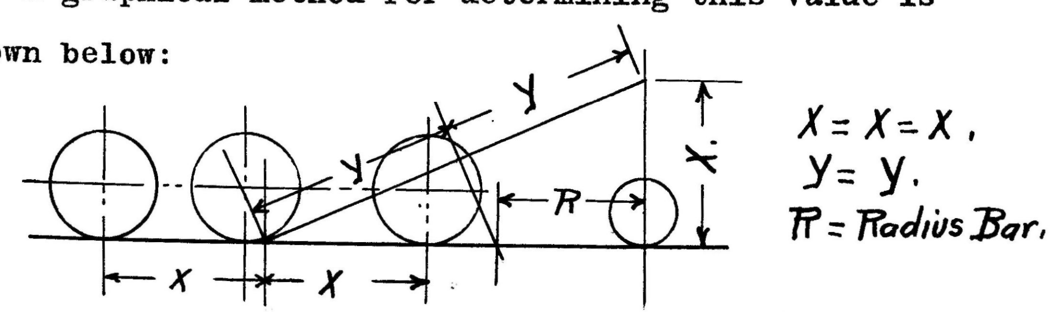
Another formula for the length of the radius-bar is as follows:

$$R = \frac{AT}{A+T} \times .85,$$ in which

A=The wheel base from the rear drivers to the center of the truck; for Consolidation and Mikado types A=The total wheel base.

R and T=Same as above.

A graphical method for determining this value is shown below:



In all of the above calculations the standard M.C.B. standard flanges and rails are assumed to be used. Also the tracks are assumed to have the proper spread on the curves which should be as follows: curves 8 degrees and under should not be widened; for curves over 8 degrees, 1/8 of an inch should be added to the guage for each 2 degrees of curvature, up to a maximum of 4 feet 9 1/4 inches for standard guage when new.

The track is also assumed to have the proper elevation of the outer rail around curves which should be according to the following formula:

$E = .00066 D V^2$ in which

E=The elevation of the outer rail in inches.

D=The degree of the curve.

V=The velocity of the train in miles per hour.

For locomotives of six drivers or more which must be used on tracks with sharp curves it is usually customary to use wheels with flat tires for the middle drivers. This allows the locomotive to take a much sharper curve than if there were a flange on these wheels, as all that is necessary is to make sure that the flat wheels will not drop off the rails and that they are wide enough and have sufficient bearing on the rails at the sharpest curve.

It is usually customary to allow from 1/8 to 1/4 of an inch in lateral motion for the driving wheels.

From the above it can be seen that care should be taken that a sufficient number of wheels should be used to carry the load of the locomotive so that the rails are not overloaded; also that the wheel base must be short enough to allow the locomotive to make curves; and that the trucks should be designed to have the proper swing on each side of the center and that the length of the radius-bar for the trucks should be such that the trucks will swing so that the center-line of the truck is radial to the curve.

PROPORTIONS OF LOCOMOTIVES

Locomotive design has been so thoroughly worked out and such a vast amount of experimental work done that there are a great many ratios between various elements going to make up the whole which should be held to the values already determined and which give good results in practice.

There are a few terms in use in locomotive practice which show the relation of various elements going to make up the locomotive, such as the Modulus of Propulsion which may be found by the following formula:

$M = \frac{A \times S \times 4}{C \times W}$, in which

M=The Modulus of Propulsion.

A=The area of the piston in square inches.

S=The stroke of the piston in inches.

C=The circumference of the driving wheels in inches.

W=The weight in tons on the driving wheels.

For Consolidation locomotives this Modulus should be about 2.85, for Pacific types about 2.

The Modulus of Traction is determined by the following formula:

$T = P \times p$, in which

T=The Modulus of Traction.

P=The Modulus of Propulsion.

p=The boiler pressure.

This value (T) varies from 400 to 600. About 550 gives the best results.

Other locomotive ratios are:

$$\frac{\text{Weight on drivers}}{\text{Tractive effort}} = \text{From } 3\ 1/2 \text{ to } 6\ 1/2.$$ 5 is usually used.

$$\frac{\text{Total weight}}{\text{Tractive effort}} =$$ This value is various but descriptive of the locomotive.

$$\frac{\text{Tractive effort} \quad \text{Diameter of drivers}}{\text{Equivalent heating surface}} = 750.$$

$$\frac{\text{Weight on drivers}}{\text{Equivalent heating surface}} = \text{From } 43 \text{ to } 65.$$

$$\frac{\text{Equivalent heating surface}}{\text{Cylinder volume}} = \text{Around } 220.$$

$$\frac{\text{Grate area}}{\text{Cylinder volume}} = \text{From } 3 \text{ to } 5.$$

$$\frac{\text{Heating surface}}{\text{Grate area}} = \text{From } 50 \text{ to } 90.$$

To get the values of the various factors in these equations, look up each one separately under its proper heading and it can be readily calculated. These ratios are used a good deal in giving the specifications of locomotives, both by the company ordering the locomotives and also by the sales department of the locomotive builders.

BOILERS

The locomotive is in itself a complete power plant, being made up of a boiler,with all the necessary appliances used with it,and the engine used to create the necessary power and tractive effort.

The boiler of this power plant is then one of the most important features as the effeciency of the machine as a whole is largely determined by the efficiency of the boiler.

Locomotive boilers are made in a great many styles and sizes. The styles that were used in the past are quite different from those used at the present, both in shape and size.

Some of the most important types of boilers used, some of which are now almost obsolete, are:

The wagon top boiler is one having the dome over the fire box and having the first course of the barrel sloping towards the front end.

The extended wagon top boiler is one having one or more straight barrel courses next to the fire box on which the dome is located and with the course in front of the dome or the other courses sloping toward the front end. Usually there are one or more straight courses next to the smoke box.

The straight top boiler is one having the barrel cylindrical and equal in diameter over its entire length.

The Belpaire boiler is one having a flat crown sheet and flat and parallel outside sheet. These flat plates are stayed with straight, direct, vertical, and transverse horizontal stays. No crown bars are used. The Wootten fire box is sometimes used with this type of boiler.

The Wootten fire box is wide and shallow. Usually a curved crown sheet is used with this style of fire box. This type of fire box is used for burning anthracite coal where a very large grate area is necessary. This type is used of course only where it is necessary to burn this kind of coal.

The radial stay boiler is one which has radial stays between the inside and outside fire box sheets. This style of staying may be used with many types of boilers.

The crown bar boiler is one which has reinforcing bars above the fire box which are used to support the fire box sheet. The ends of the bar are supported by the side sheets and the sheet is connected to the bars by stay bolts.

In the preliminary design of the locomotive it is necessary to determine the size, type, and capacity of the boiler to be used. There are a number of factors which must be considered before it is possible to determine just the size and type of boiler that is to be used: first the steam, in pounds per hour, that it will be necessary to develop and the pressure at which this steam must be delivered; second the kind of fuel that is to be burned; and third the necessary weight of the boiler which, when full of water, should be equal to about 30% of the total locomotive when ready to operate.

In order to detail properly a locomotive boiler from the rules and tables laid down by the A.S.M.E. Code Committee or according to whatever code the boiler is to be designed to suit, it is best to get a good idea of locomotive boiler performance and efficiency as regards operation and upkeep.

The maximum boiler pressure for the boiler should not exceed 225 pounds as any pressure above that adds greatly to the maintenance cost. Little is gained by having the boiler pressure over 200 pounds unless the space is limited and cost not important. For simple engines 160 to 200 pounds have been found to give the bést results.

The maximum equivalent evaporation from and at 212 degrees Fahrenheit per square foot of heating surface is 12 pounds. The average value is lower, being from 6 tp 8 pounds. The heating surface is not very often driven above 12 pounds but it is possible to drive the boilers to 15 to 16 pounds. It is evident that one square foot of heating surface should deliver .2 horse power when properly driven. The boiler can be driven to .35 or .45 boiler horse power. In late designs of locomotives from 35% to 50% of the heating takes place in the fire box of the boiler while only from 4% to 8% of the heating surface is in the fire box. This fact shows the importance of the design of the fire box.

With the quality of the steam from 98% to 99% it requires 1 pound of coal to evaporate from 10 to 12 pounds of water when working at maximum efficiency. Ordinarily only from 6 to 8 pounds are evaporated.

The temperature in the fire box of a locomotive boiler varies from 1400 to 2000 degrees when working under ordinary conditions. The temperature can be raised to 2300 degrees Fahrenheit when the boiler is forced.

The temperature in the smoke box should vary from 500 to 700 degrees Fahrenheit.

The draft in the smoke box should vary from 5 inches to 8 inches of water.

Large grates give the best satisfaction for temperature and draft.

Fire brick arches help combustion ~~but do not~~ and add to the capacity of the boiler.

Fuel saving in boilers using high super-heat averages about 25%. The best temperature for super-heat is about 500 to 625 degrees Farenheit. It is well to consider this fact when making the preliminary design.

All of these factors should be carefully considered before proceeding with the detail design of the boiler.

Before attempting to design a boiler for a locomotive one should get the report of the A.S.M.E. Boiler Code Committee, study every article carefully, and make every detail conform to the laws laid down in that report. This report does not give sufficient information to design a boiler however; so it is necessary to go a little more into detail.

2 inch tubes have been found to give about as good results as any for boilers of the ordinary size. They should not be too long as it has been found that long tubes are very little more effective than those of ordinary size. These tubes should be used with as close spacing as possible and the rows of tubes should be vertical and not horizontal as the vertical rows give the best circulation of water.

The tubes are put in through the front tube sheet after both tube sheets are in place. The holes in the front sheet are slightly larger than the tubes so they can be inserted through it. On the fire box end the tubes are swaged down the thickness of the tube and copper ferrules are used in the tube holes in the sheet. The thickness of the ferrules should be 1/32 of an inch for new work and up to 1/8 of an inch on repair work. The allowance for clearance before rolling the tube should be 1/16 of an inch. This will give a good easy fit when putting the tube in place. The copper ferrules should be put in the holes and the edges of the ferrules should be left 1/32 of an inch from the fire box side of the sheet. Then when the tube is rolled the edge of the sheet is about even with the ferrule. When the tube is put in it should be rolled or expanded in the front tube sheet and rolled and beaded or rolled, beaded, and prossered which is expanding the tube on the inside of the tube sheet as well as the outside. The tube should be beaded until it forms a perfect roll back against the tube sheet.

The tubes may be welded in the fire box end as well as rolled. This welding may be done either by oxy-acetylene or electricity. When the tubes are welded they are extended through the tube sheet only part way. They should extend all but 1/32 of an inch through the tube sheet, the balance of the space being filled up with welding metal. The front ends of the tubes are rolled and beaded.

The spaces between the tubes should be from 5/8 of an inch to 7/8 of an inch. These figures are about the minimum and the maximum. The tube sheet can be layed out by locating the first tube, then describing a circle about this tube with a radius equal to the diameter of the tube plus the spacing between the tubes. Divide the circle into six equal spaces, starting with the top or bottom center line; the points found are the centers of the nearest tubes. Other tubes can be located by taking some of the tubes thus found and drawing other circles. It is best to make a template of the tube spacing which can be used for getting the maximum number of tubes in the available space by shifting it about.

The tubes should slope about 1/4 of an inch toward the back end of the boiler so that any water which might get into the tube will run into the fire box and not stay in the tube to rust it out.

The longitudinal seams on boilers, of course, depend on the pressure which they must carry and the boiler code under which they are built. The most common and probably the best is the butt joint with inside and outside butt straps and double riveting, the outside taking the inside row of rivets only. This type of joint gives the highest efficiency for the cost of construction.

A lap joint should never be used on the barrel of a boiler because of the bending action to and fro which must take place at the end of the inside lap which causes it to be grooved and cut at this place. This is because the barrel cannot be made circular and the pressure in the boiler tends to make it circular.

This holds true on the dome of a boiler and, although the dome may be of much smaller diameter and of higher factor of safety, it is, when over 24 inches in diameter, not allowed to have a lap joint.

When reinforcing plates are applied to boilers for steam pipe connection wash out plugs and the like, it is best where possible to put the patches on the inside of the shell. This is because they are much easier to apply and it makes a cheaper and better job.

In the design of the water leg of the fire box care should be taken to see that the sheets, on both the inside and outside, slope a little away from the mud ring, thus increasing in width as they ascend. This will insure the steam properly rising and not forming bubbles on the sheet, causing it to keep the heat from being taken up by the water.

The fire doors in locomotive boilers should be located with their lower edges 22 inches above the deck of the cab for the best average firing conditions. This was established by the Pennsylvania R.R. by taking motion pictures and making a study of the motions of the fireman.

When fusible plugs are used they should be placed in the highest point of the crown sheet of the fire box. These plugs are not very reliable as the impurities in the gasses unite with the fusible part of the plug and cause the melting point to rise and the plug will not melt out when it should. For marine work the U.S.governemnt specifies Banca tin with a melting point of 445 degrees Fahrenheit. This is also commonly used on locomotive work. These plugs are usually not required on locomotive boilers.

Locate washout plugs at every place about a boiler where there is any chance of mud accumulating. Plugs should be located so that every part of the boiler can be easily washed out. If the boiler is kept properly cleaned it will give much better results.

Boilers must be properly braced and stayed on the flat surfaces and great care should be taken to allow for expansion and contraction. It is usually customary to use flexible stay bolts over the crown sheet where there is sufficient room. The heads of the boiler should be braced from the sides of the barrel. For small boilers the best bracing consists of a rod with a T-shaped end and the end riveted to the head and flattened to the barrel end. These braces come in stock sizes of various lengths. They are heated and bent to the proper angle when being applied in the shop.

The most important part of a locomotive boiler is the fire box.

The function of the fire box of a locomotive is two-fold: first, to burn the coal and liberate the heat; second, to absorb the heat into the water. It will be seen, therefore, that both of these functions must be taken into consideration when a fire box is designed.

About 50% of the heat liberated by the coal is liberated by the burning of the volatile gasses over the coal bed. These gasses must be properly mixed with the air before they will burn; so it is very important that there be a convenient place for them to mix. A proper combustion chamber or space is, then, as important as the grates of the fire box.

It is a common practice to design locomotives to deliver their maximum tractive effort when burning 120 pounds of coal per square foot of grate area per hour. It has been determined by experiment that at this rate of combustion we get the following values:

Heat liberated by the furnace, 72%.

Heat absorbed, 53%.

Heat lost by the furnace, 27%.

Heat lost in the front end gas and sparks, 14%.

Heat lost in ash and by radiation, 8%. (This includes part of the furnace loss).

Heat lost by rejection by the heating surface, 6%.

Heat lost as CO, 5%. This is included in the front end loss.

The fire box losses in these tests were as follows:

Combustible in the ash, 1% to 1 1/2%. This cannot be reduced a great deal. The carbon monoxide should be burned in a long combustion chamber. The unburned hydrocarbons should be mixed with an excess of air in the flame ways. The loss by sparks can be avoided by reducing the velocity of the draft and by increasing the area of the grate.

Summed up, the losses most common in locomotive practice can be reduced by increasing the grate area and by increasing the fire box and combustion chamber space. It is also very necessary to provide effectual baffles and mixing devices and by supplying at least 33% excess of air or approximately 16 pound of air per pound of coal.

Tests show that at times 80% of the fixed carbon is incompletely burned to carbon monoxide. It is very difficult to burn all of the carbon monoxide to carbon dioxide in the ordinary fire box.

It is very desirable to have a very large fire box and combustion chamber and to use a very high fire box temperature so that a large part of the heat can be given as radiant heat to the heating surface. This is true of course when high efficiency is to be obtained at the expense of space in the boiler. Unless the combustion chamber is full of flame it is of practically no value.

Long grates for locomotive fire boxes should have a slope of about 2 inches in 12 inches. Grates should have as nearly as possible 50% of their area open for the passage of air.

Grates for burning wood are the simplest of all types. They consist of simple stationary grate bars that usually run the long way of the fire box. These bars are of course made in sets for one fire box, with the necessary side and end bars to rest on. The supporting bars on the end are fastened to the inside of the fire box sheet with studs. The spaces and bars are usually about equal in width.

Grates for bituminous coal are usually made of cast iron bars, made in such a way that they will rock on trunnions on the sideand the front bar made so that it will dump the ashes in the front of the fire box. The other bars rock to shake out the ashes. This shaking and dumping should be done from the cab; so it is necessary to make a rigging by which this can be done.

Grates for anthracite coal: Place at the side of the fire box 1 3/4 inch wrought iron pipes, allowing about 4 inches between the center lines of these pipes. Between these tubes are to be placed cast iron grates fastened at the ends. In the center of the grate and about 3/4 of the area of the grate should be a rocker grate similar to that used for bituminous coal.

Water grates for anthracite coal: Use 2 inch wrought iron pipe, sloping toward the front end for circulation. Every fifth space use a removable rod which permits the ashes to be dumped. These rods should run through the back water leg so they can be removed. Screw tubes in the front sheet and fasten with tapered thimbles in the back end. These thimbles are driven into holes in the back plate so as to make a tight joint around the tube. The rods are to be supported on cast iron bars in the front end of the fire box.

The area which the grate for a locomotive should have has been the subject for a great deal of investigation and there are a great many different rules and tables by which the area can be determined. I will give some of the most reliable of them:

Grate area in square feet: $G=\frac{W}{fh}$. in which

W=Pounds of water evaporated per hour.

f=Pounds of water per pounds of coal.

h=Pounds of coal per hour per square foot.

75 pounds of coal burned per square foot of grate area is the best value to be used for design purposes, al-

though this volume is often exceeded. For the amount of water evaporated per pound of various kinds of coal per hour see the table on coal:

The Master Mechanics' Association (1897) gives this:

Kind of coal	Water evaporated per lb. of coal	Coal burned per sq. ft. of grate area per hr.
Large anthracite	8 lbs.	60 lbs.
Fine anthracite	6 1/2 lbs.	35 lbs.
Semi-bituminous	9 lbs.	65 lbs.
Bituminous	7 lbs.	90 lbs.

	Ratio of cyl. vol. in cu. ft. to grate area in sq.ft.	Cyl.vol.in cu. ft. to heating surface in sq.ft.	Heating surface to grate area.
Large anthracite	1 : 4	1 : 180	40 : 1
Fine anthracite	1 : 9	1 : 200	20 : 1
Semi-bituminous	1 : 3	1 : 200	60 : 1
Bituminous	1 : 3	1 : 200	60 : 1

The Railway Master Mechanics' Association (1902) gives this:

Kind of coal	Freight engine Simple	Freight engine Comp.	Passenger engine Simple	Passenger engine Comp.
Free burning bituminous	70-80	65-85	65-90	75-95
Average burning bituminous	45-70	50-65	50-65	60-75
Slow burning bituminous	35-45	45-50	40-50	35-60
Slack burning bituminous	30-35	40-45	35-40	30-35
Free burning anthracite	30-35	40-45	35-40	30-35
Slow burning anthracite	25-30	30-40	28-35	24-30

Here is a good table showing the necessary grate area for various sizes of locomotives. The diameter and stroke of the piston are given.

Piston: Diam.	Stroke.	Grate area for anthracite coal	Grate area for bituminous coal
12	20	1591	1217
13	20	1873	1432
14	20	2179	1666
15	22	2742	2097
16	24	3415	2611
17	24	3856	2948
18	24	4321	3304
19	24	4810	3678
20	24	5337	4081

In ash pan design it is always best to give a good pitch to the sides of the pan so the ashes will slide to the bottom of the pit. On narrow guage locomotives this is accomplished by sloping the mud ring of the boiler and also the sheets of the pan.

The air space between the ash pan and the boiler and through the air spaces in the pan itself should be at least 15% of the grate area and may well amount to 25% for good results. Usually 15% is allowed over the top of the ash pan; then draft doors are placed in the front and back for the extra draft.

Ash pans are usually made of about no. 10 sheet iron riveted together and bolted to the mud ring of the locomotive boiler. They are made with either one or two hoppers. When two hoppers are used it is usually because an axle passes under the fire box. The pan should always be provided with a slide dumping arrangement at the bottom of each pit. This slide should be fitted up so that it is possible to dump it either from the cab or by means of a convenient lever on the side of the frame under the cab. The arrangement of this rigging is dependent on conditions and the judgment of the designer.

There is one very important item in the design of ash pans and that is that there should be no holes through the walls of the pan which are sloping and through which ashes might fall. By these holes I mean holes for grate rigging rods and the like.

The petticoat pipe of a locomotive is the pipe in the front end of the boiler which serves the purpose of directing the steam from the exhaust up the stack and distributing the suction caused thereby equally over the boiler tubes. By this means the tubes will each do their share in carrying away the gasses and in heating the water in the boiler.

The petticoat pipė must be so designed that it will cause a vacuum both at the top and at the bottom of the pipe. This accomplished by making the bottom of the pipe funnel shaped and having the body of the pipe about the same size as the stack. The funnel is at the bottom while at the top an open space is left through which gasses are drawn by the suction caused by the steam rushing up the pipe. The space left at the top and the height at the bottom of the pipe above the exhaust nozzle are the means by which the draft is regulated.

Usually with the petticoat pipe a diaphram is used. This diaphram is simply a sheet steel plate for directing the gasses out of the tubes. The top of the plate is above the tubes and the smoke is drawn down under the lower edge of the plate. This plate causes a vacuum at the bottom the same as the petticoat pipe. In order to regulate the draft through the tubes the lower edge of the plate is raised or lowered. Usually 1/4 of an inch is sufficient to regulate the draft.

The effect of having no draft through part of the flues is to cause them to choke up with ashes. If the draft is stronger through the bottom tubes than through the top ones the fire burns more in the front end; while if the draft is stronger through the top tubes the fire burns more at the back end of the grate. If the fire burns too much at the front end of the grate the petticoat pipe is too high; if it burns too much at the back end the pipe is too low. For adjustment of the diaphram, when the fire burns too much at the back end, lower the sheets; when it burns too much in the front end, raise the sheets.

Heavy wire netting is often used with the diaphram. The smoke must pass through this netting, thus breaking up the cinders that might pass through the stack and couse fire otherwise.

Before an attempt is made to design a front end arrangement for a locomotive a careful study should be made of the existing designs in order to get an exact idea of how these arrangements are made in common practice. This arrangement has been the result of practical experience and not the result of any theoretical development.

In the design of locomotives it is customary to make the highest point of the structure the top of the stack. This is for looks only. On the larger types of locomotives of to-day the stack is very small and often there is more of it inside the smoke box than outside. It must necessarily be short on account of the clearance.

There is a number of rules for the design of stacks which have been worked out from practical experience. Most of the rules have been established a number of years. These rules must be used with a full knowledge of the present day practice in the design of stacks and front end arrangements used on various types of locomotives to-day.

Stacks are in most cases tapered from the choke, which is just a few inches above the base, to the top with a taper of about 1 inch in 12. They are also made straight.

The diameter of the choke should be the same size as the diameter of the cylinder of the locomotive or an inch or two less.

The height of the stack should be three or four times the smallest diameter.

The height above the exhaust nozzle should be determined by means of the following formula:

$H=14\ d$ in which

H=The height and d=the dimeter of the exhaust nozzle.

The diameter at the top of the stack should be:

$D=3.8\ d$ in which d is the same as above.

The imaginary diameter of the stack if it were extentended down to the top of the exhaust nozzle:

$D_1=.65\ D$ in which

D_1=The diameter spoken of above and D=the same as above.

The height of the choke above the exhaust nozzle:

$C=.4\ h$ in which h=the same as above.

Some of the older types of stacks and stacks for wood burning locomotives and other locomotives where there is danger of causing a fire from the sparks use wire netting to break up the cinders. This is sometimes used in what are known as diamond stacks, which are stacks with large tops. The netting used in these stacks and also in front end netting is 1/10 to 1/30 of an inch in diameter and has from 3 to 4 meshes per inch.

Stacks are usually made from cast iron although many of them are made of sheet steel or pressed steel. Sometimes the upper and lower parts of the stack are made in two pieces. This is for manufacturing purposes only.

The exhaust nozzles of locomotive are for the purpose of directing the steam from the cylinders up the stack and causing a draft for the fire. These nozzles are made in two styles, single and double columns. Usually they are cast in one piece and are made in such a way that a bushing can be placed in the top of such a size as will give the proper draft for the fuel that is used. Several bushings are usually furnished with each engine so that the proper size may be fitted in at any time.

Varying the size of the nozzle varies the size of the stream of steam up the stack and also the back pressure on the pistons. The blast should be just strong enough so that it will give a good draft and still not be strong enough to cause too much back pressure on the pistons.

The bushings are bolted or fastened into the nozzle with set screws. The nozzle is bolted onto the cylinder castings with studs. The size nozzle that should be used with various sizes of engines and the size of the stack that is used are shown in the following table which represents the sizes actually used on locomotives that are already in use:

Size of cyl. Diam.	-Stroke.	Diam. of stack.	Diam. of double nozzle.	Diam. of single nozzle.
12	20	9 1/2	2	2 13/16
13	20	10 1/2	2 1/8	3
14	20	11 1/4	2 5/16	3 1/4
15	22	12 1/2	2 9/16	3 11/16
16	24	14	2 7/8	4 1/16
17	24	15	3 1/16	4 5/16
18	24	15 3/4	3 1/4	4 5/8
19	24	16 1/2	3 7/16	4 13/16
20	24	17 1/2	3 5/8	5 1/16

All dimensions in the above table are given in inches.

Boilers should have their fittings and appliances to suit the boiler code under which they are designed. By fittings and appliances are meant such things as safety valves, steam and water gauges, injectors and the like.

On top of the boiler in the cab a steam gauge stand or steam fountain should be located. This steam fountain can be used for all steam connections for boiler appliances such as injectors, steam pipe for air brake pump, blower, lubricator, and steam gauges.

All of the valves for operating the appliances should be located in a place convenient for their operation. The steam gauge should be located so that it can be seen easily by both the fireman and the engineer. Usually one water glass is used on one side of the back head and on the opposite side within the range of the glass three drip cocks are placed.

The injector is usually placed so that it can be operated by the fireman. The blow off cock is usually placed at the bottom of the water leg of the boiler, on the front side and in the middle, with an extension handle for operating it.

The safety valve should be located on top of the dome and must be large enough to discharge all of the steam generated by the boiler when forced and should be set so that the boiler pressure will not rise higher than 6% above the working pressure. Safety valves should be not less than 1 inch in diameter nor more than 4 1/2. The minimum capacity of a safety valve should be figured on the basis of 5 pounds of water being generated into steam per square foot of heating surface per hour.

The efficiency of boilers may be increased somewhat by obtaining better combustion of the fuel and also black smoke may be prevented by using fire brick arches. These arches are built in the front end of fire boxes and extend down towards the lower tubes at an angle of from 30 to 45 degrees. They are supported either on studs in the case of small fire boxes or by water tubes which are connected to the two ends of the fire box.

In tests made by the Pennsylvania Railroad on a Mikado engine the following results were obtained when using a fire brick arch of the security sectional type:

The evaporation was increased 15 1/2% at maximum capacity. The economy in the coal ranged from 6 to 8%, this being evident at all rates of combustion. A higher boiler efficiency at all rates of evaporation was obtained; this varied from 6.9 to 11.6%. The rate of firing over the tests was from 35 to 120 pounds of coal per square foot of grate area per hour. In road limits of firing the boiler efficiency was increased from 7 to 8 1/2%. Arch tubes add 1% for each tube involved alone. The maximum draw bar pull at 29 miles per hour was increased 6.4%.

BOILER MAKING

The first step in the construction of locomotive boilers is that of making templates for marking the various sheets which go to make up the boiler. It is usually customary to make these templates of about number 10 sheet steel and they are usually made exactly the shape of the sheet with all of the holes located. In laying out these templates it is necessary to develop the sheet on a flat surface. This sheet is then sheared and punched and when rolled or flanged must conform to the shape and size given on the boiler blue print with all the holes properly located.

In sheet metal pattern drafting there are three general methods employed for laying out templates: by means of parallel lines, by radial lines, or by the method known as triangulation. Triangulation is usually preferred by most layer outs as it can be used in a great many cases.

Triangulation in boiler making is the division of the surface of any irregular object into triangles, determining the length of the sides from the drawing and transforming them in regular order in the pattern. The method used is as follows:

If necessary lay out both the plan view and elevation full size on an extra sheet . Divide the ends of the plate that is to be layed out into equal numbers of spaces . Connect these points of division with lines and number the end points. On another place on the sheet lay out either the vertical projection or the horizontal, whichever is common to all the connecting lines. At right angles to this line at one end lay out another line and on this lay out the opposite or variable projection. Get the hypotenuse of the triangle thus formed when the ends of the lines are joined and you have the true length of the line. These true lengths are found for all lines and layed out on the sheet in order as they come.

Such sheets as the outside and inside fire box sheets are made by the above method. Also tapered courses of the barrel are usually made in this way. The dome bodies of a locomotive boiler that follow the contour of the boiler barrel and tank thimbles are layed out as follows:

Draw a sketch of the plan and elevation of the dome section . Divide the plan view into, say, eight equal parts. Project these points of division onto the sectional elevation and draw lines across the elevation at these locations. The lengths of these lines on the elevation are the true lengths of the lines. Lay out the sheet and divide it into the same number of equal spaces as the plan is divided and lay off these lengths that were found. Bend straight edge and connect the points. The rows of rivets are parallel to these lines.

There are a few rules in common use which should be used for getting the length of various shaped sheets, such as:

When two courses are to be fitted one inside the other for boiler or tight joints add six times the thickness of the metal for the outer course length or seven times for non-water-tight joints or loose fits.

For finding the length of the inside course of the barrel use the inside or outside diameter and calculate the circumference. Then add or subtract three times the thickness of the metal.

Inside corners of lap joint courses which come on the outside should be scarfed and outside corners of inside courses of lap joints should be scarfed.

For getting staggered rivets in back row, tram with dividers from each pair of rivet holes in the front line, using diagonal pitch.

When making a layout for a flanged head subtract twice the thickness of the metal from the height of the flange for the flange allowance on the flat plate. In marking for corners in flanged heads less metal is needed to make the flange so more than twice the thickness may be subtracted on the corners.

When the head is flanged the holes should be located on the guage line to correspond with the holes in the plate. This is usually done by making a tape and transfering to the head.

These templates should be made for all sheets that go to make up the boiler. After this is done all that is necessary is to take a sheet of the proper thickness and quality and mark it from the template so the shear line can be seen and all the holes should be center punched.

Having marked the sheets the plates are sent to the shear and sheared to the proper dimensions. Next they go to the punch and the holes are punched. The next operation on most of the sheets is planing the edges which trues up the edge and gives it the proper bevel for caulking. From the planes the plates go either to the bending roll or to the flange press to be bent to the proper shape.

The bending rolls for bending boiler plates are made up in a number of sizes and lengths. The larger sizes are always operated by power. The rolls themselves are three in number and so placed that their position or relation to each other is adjustable and can be altered so that the plate in passing between them is bent.

The sweep is usually made to exactly the shape which the sheet should be rolled to. This sweep is used for a guage for rolling the plates. The sheet is rolled back and forth between the rollers until it is made to canform to the necessary dimensions.

The flanging of the various sheets is accomplished in three different ways: by hand flanging while held in flange clamps; machine flanging in various kinds of flanging machines; or under a hydraulic press by means of dies. The greater part of boiler flanging is done in the later way. The sheets are heated in an oil furnace and then forced through dies which give them their proper shape. Usually after they are run through these dies a little hand work is necessary.

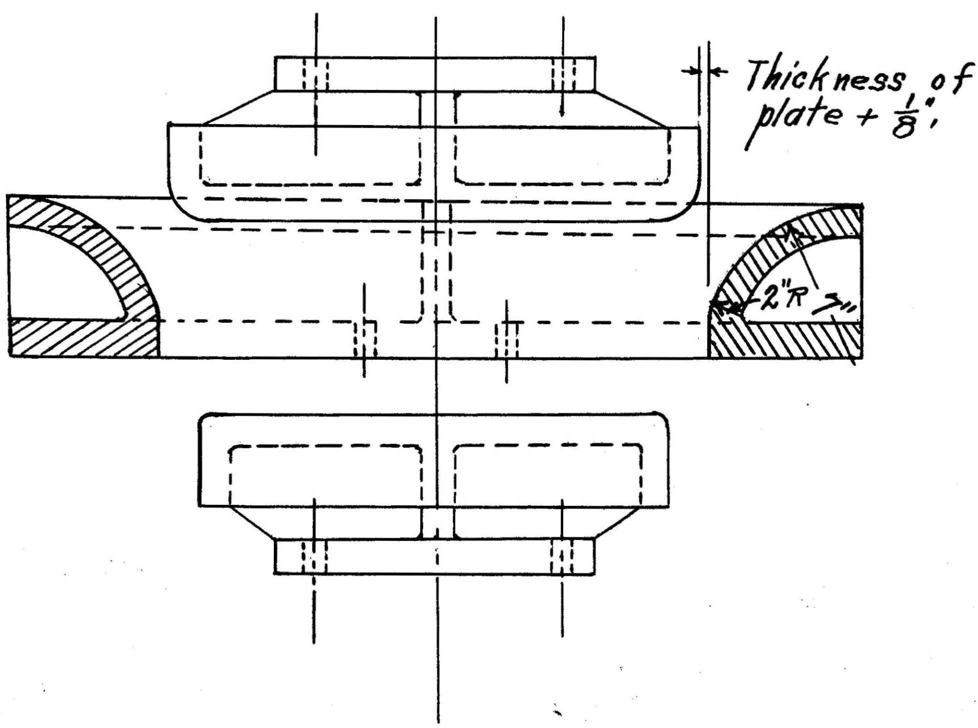
The flange press which is used for this kind of work has two rams on the top arm of the press or head and one in the bottom midway between the two upper rams and one in the back jaw of the press which moves horizontally with the bed of the press. The movement of each plunger is controlled by a lever at the side of the press.

The dies used for flanging boiler plates under the hydraulic press must be made in sets for each sheet. The lower plunger usually is fitted with a cast iron block which holds the plate tight against the male block which is fastened to the upper rams. The female block is placed upon cast iron stands which are high enough to allow the sheet to be withdrawn from under it after being flanged. The operation, then, of flanging the boiler sheet with this set of die blocks is as follows:

The sheet is withdrawn from the fire when properly heated and placed on top of the female block. The ram in the bed of the press is then moved upward and the plate is located on it either by measurement or by dowel pins in the face of the block. The top rams then are forced down upon the sheet which rests on the lower block, clamping the sheet firmly between the blocks and holding it in position. The plate and plungers are then forced down through the female block and the sheet is given its form.

Just under the female block in the stands which support it are adjustable strippers. These strippers are then moved into position so that when the male block is drawn up the plate is stripped from the male die. It is then withdrawn through the stands holding the female block.

Examples of the sheets flanged in this way are: front and back tube sheets, inside and outside fire door sheets, and one operation on the throat sheet. The die blocks for these operations should be made of cast iron, the walls of which vary from 1 to 2 inches. The depth and strength of the ribs for the blocks should be made of about equal thickness. The top male die should have a contour on the outside the same as the boiler sheet should have after being flanged. The female die should be of about equal thickness and the contour of the die should be a smooth curve, starting with about a 7 inch radius at the top and should be about 7 or 8 inches wide in section, the curve gradually becoming sharper towards the bottom and inside until it forms about an inch or two of straight perpendicular surface at the bottom. This contour allows the sheet to be drawn through the die without tearing and it bends the sheet as it slides through.

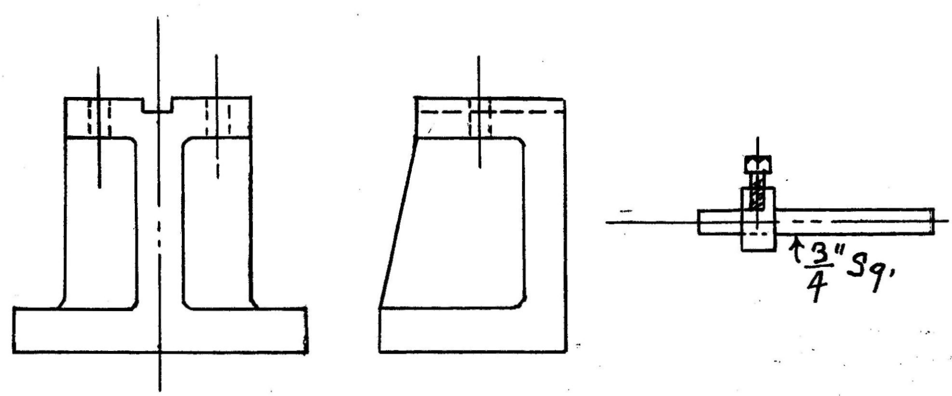


The male block should be made 1/16 of an inch smaller and the female block 1/16 of an inch larger in radius than the sheet which is being flanged. This allows 1/8 of an inch clearance. This clearance should be made larger for plates over 1/2 inch thick.

The throat sheet is flanged in two operations: first the water leg is flanged and then the barrel. For flanging the barrel flange one upper cylinder is used to clamp the sheet down on the forming block while the other plunger is used with the die to make the barrel flange through the jaws of the lower block.

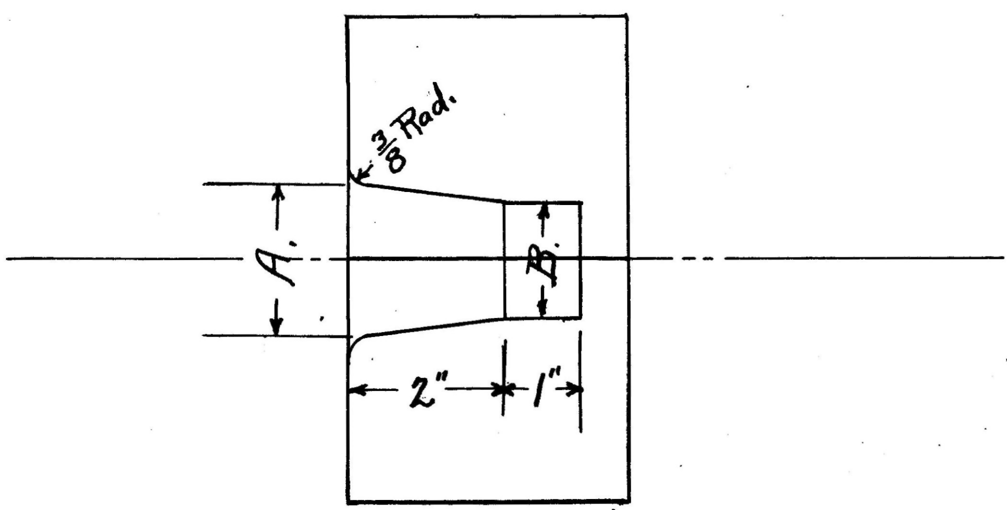
Butt straps are also formed under the flange press. For this work the bottom die is made concave and the upper block convex to the radius of the butt strap. The blocks should be wide enough for the butt strap and at least 36 inches long. The bottom block rests on the bed of the press.

Below is a sketch of the stands used for the female die blocks and the stripper which works in the slot of the stand



The female die is bolted to the stands and the stands are clamped onto the bed of the press, in this way holding the block and preventing it from moving.

For swaging boiler tubes a small air hammer is used which operates the swag blocks, the bottom block being stationary and the top block moving up and down. The ends of the tubes are heated in an oil furnace and swaged down in this manner. Below is a sketch, showing the dimensions of the impressions in the swage blocks and it represents the form which the tube will take after being swaged. These blocks are made of high carbon steel and machined to exact sizes.



A = Outside diameter of the tube.
B = A - thickness of the tube.

After the boiler sheets are flanged the edges of the sheets are sheared off so their edges are smooth or they are machined off if it is impossible to shear them in such a way that the edge of the sheet is beveled for caulking. After this is done the boiler can be fitted up.

The butt straps on the barrel are usually riveted on with a hydraulic riveting machine. The inside and outside fire box sheets and the throat sheet are fitted up and riveted separately; then the two parts are fitted together with the mud ring in position and the stay bolts are then put in. The barrel is next fitted onto the fire box; then the front tube sheet and dome are fitted on, the braces for the back head of the boiler being put in before the fire box is fitted up, and the braces for the front tube sheet are put in before the tubes are put in place, the dome with its liner being put on at any convenient stage of the process. The tubes are put in last. All of the edges of the sheets should be caulked to prevent them from leaking and also the rivets.

All of the workmanship such as drilling, reaming, riveting, caulking, and driving stay bolts should be done in accordance with the code under which the boiler is made.

After the boiler is completed it must be tested to make certain that there are no leaks and that all stay bolts and braces are securely fastened and will not give way.

First-the cold water test: Fill the boiler with water and force the water to working pressure plus about 50% of the working pressure.

Second:Kindle a fire under the boiler in the grate, when the water is at zero pressure. Water expands 1/24 from 60 to 212 degrees. Warm water can be forced in with an injector supplied with steam alone. This test should run to about 1 1/2 times the working pressure.

Third: A steam test should next be made to 25% over the working pressure.

Do not caulk leaky stays or rivets while pressure is on as it may cause an explosion.

Cold water leaks through crevices that may be closed by expansion when the boiler is heated.

Warm water may cause the boiler to expand unequally causing leaks.

Apply a straight edge to flat surfaces while under pressure to note bulges.

Examine stay bolts by tapping with a small hammer while the pressure is on.

After hydraulic tests the inside should be examined thoroughly to see whether stays are broken. After this is done the steam test can be made.

When it is desired to get up steam in a locomotive boiler care should be taken not to heat the boiler too rapidly as in time this will cause leaky joints. To fire up a boiler proceed as follows:

Cover the grates with coal banked to 15 to 20 inches along the sides and door sheet. Slope the coal toward the center leaving a light bed of coal in the center for air draft. Add shavings in the center of the grate and pile on enough coal to hold the shavings in place. Start the fire in the rear of the fire box as the blower will draw the fire ahead. Drop oily waste inside the door to start the fire and use a light blower. Increase the draft as the fire increases and the coal starts to burn. Use enough draft to keep down the smoke. With cold water in the boiler steam can be got up to 100 pounds in 1 hour and 40 minutes; with hot water and no pressure in 1 hour.

SUPERHEATED STEAM FOR LOCOMOTIVES

In locomotive practice where saturated steam is used there are some conditions which, if eliminated, would greatly increase the efficiency of the engine. One of the greatest losses and trouble makers is the initial condensation in the cylinders of the engine. This is due to the fact that the steam, on entering the cylinder, comes in contact with the comparatively cold cylinder wall. Part of the steam is condensed. This steam is later reevaporated and there is a consequent loss of energy. In order to avoid this loss of energy it has been found very effective to superheat the steam, that is, to heat the steam above the point at which it becomes steam under the given pressure.

In modern locomotive practice this superheating process has been accomplished in two types of superheaters. One takes only the heat from the smoke after it has left the boiler tubes and is ready to be exhausted up the stack. The other type takes heat from the gasses in the tubes, that is, it has tubes for the steam to be superheated inside the fire tubes of the boiler. These two types are standard and they may be found of various designs.

It has been found by experiment that from 200 to 250 degrees superheat gives the best results. Heat over this point does no good and heat below this point is not sufficient to prevent condensation in the cylinders. The degree of superheat used on locomotives varies from 150 to 275 degrees.

On simple locomotives, by comparative tests under constant load, it has been found that there is a saving of from 15 to 35% while using steam at a superheat of from 150 to 250 degrees. In road tests and in actual service it has been found that there is a saving of from 10 to 20% in coal and 15 to 25% in steam.

Superheaters should be placed in the front end of the locomotive. The pipes should be so arranged that they will not retard the flow of the gasses unduly. The pipes should be so arranged that they can expand freely. All superheaters should be provided with independent safety valves of the outside spring type and set slightly below the pressure of the boiler safety valve. All superheaters should be also provided with drains so that they can be drained before starting up. Care should be taken that steam flows in all the tubes of the superheater, as, if the steam is by-passed and does not flow in the tube, the tube will become heated and may melt out.

The velocity of the steam in the superheater should be about the same as allowed in the steam pipes. This will not cause any too much friction in the pipes. The rate of heat transfer that may be expected will be at the rate of about 1500 B.T.U. per square foor of heating surface per hour. This is of course under good conditions.

Superheaters should have two compartments, one for saturated steam and one for the superheated steam. These should be connected in such a way that there is an easy flow and also plenty of allowance made for the expansion of the tubes. When tubes are used in the boiler tubes they are usually 1 1/2 inches outside diameter. The boiler tubes are usually about 5 1/2 inches outside diameter. It has been proved in other countries that this size can be successfully reduced.

Superheater locomotives, having greater economy, may be designed within given limits and weight to have much greater capacity than others. This is sometimes more important than the saving of fuel.

Roughly stated, it may be said that the superheater increases the capacity of the boiler 35%. The fuel economy is from 20 to 25%; the steam economy, 30% for average practice. At the present time almost all locomotives are built with superheaters which are to be used on the road. There are not so many for switching but there may be as high as 40% of switchers now built with superheaters.

On Mallet locomotives the superheater is sometimes placed between the high and the low pressure cylinders. In this case it is known as a reheater. Sometimes both a reheater and a superheater are used on Mallet engines.

LOCOMOTIVE FEED WATER HEATING

In recent years there have been a number of feed water heaters designed, using the heat which is ordinarily lost in exhaust steam and flue gasses. The distribution of heat in a locomotive has been given as follows:

Total heat in coal-100%.

Loss in unconsumed coal and heat in ashes-5.3%.

Loss in dry smoke box gasses and vapor of combustion -18%.

Heat through boiler heating surfaces, including superheater-76.7%.

Total-100%.

These are fairly typical values.

The heat that is taken up by the steam is distributed as follows:

Heat in steam------------------76.7% (see above).

Loss by radiation--------------- 3.5%

Loss by friction in locomotive-- 1%

Useful work at draw-bar--------- 7%

Heat discharged up stack--------35.2%

Total----76.7%

To use this last item we must heat the feed water from its low temperature up to that near the boiler temperature.

Two sources are available for feed water heating: exhaust steam and smoke. Because of the large size and maintenance the method of heating water with smoke is not practical; so we must use exhaust steam. The most important item is to produce a high agitation in water; so it will take up heat. The water is agitated by passing it through between two spirally corrugated tubes made of copper. This keeps the water agitated and hence it takes up more heat. With this system the heat transfer reaches 900 B.T.U. per square foot per degree temperature difference. This heater can be fitted under the deck and cylinders and takes up very little room. It may be placed crosswise with the exhaust steam entering at each end and going to the nozzle at the center. There should be 3/16 of an inch spaces between the tubes . To force water through the heater a Westinghouse hot water pump should be used.

Rule of thumb:

For every 11 degrees feed water is heated by waste fuel there will be 1% saving in fuel.

It appears that an economy of at least 10% can be expected by using feed water heaters.

Feed water heaters using smoke must have such a large heating surface due to the slow transmission of heat that it is impossible to get a heater of this type on a modern locomotive that will give over 5% economy. It is impossible to use a combination of both exhaust steam and smoke due to the fact of insufficient room.

In a heater using exhaust steam copper tubes should be used in place of iron or steel on account of their higher conductance. The tubes should have walls just as thin as possible. Exhaust steam can be diverted for heating feed water without decreasing the power of the locomotive as the area of the exhaust nozzle is made the proper size to give the required draft. This can be done without increasing the back pressure on the pistons. Not over 15% of the exhaust steam should be used for heating the feed water.

ALLOWABLE BEARING PRESSURES

There are a number of authorities which give the allowable bearing pressure in pounds per square inch of projected area for locomotive bearings of all kinds. These values vary somewhat but they represent the best American and English practice in the design of locomotives. These values if used will give entire satisfaction.

Locomotive Part	Authority	Pressure.
Crank pins	Unwin Machine Design	1200-1800
	Halsey Hand Book	1500-1700
	British Railway Practice	up to1400
	Kinball and Barr Mac. Des.	1500-1700
	Machinery Reference Series-	up to1600
Wrist pins	Unwin Machine Design	1500-2000
	Halsey Hand Book	3000-4000
	British Railway Practice	up to2000
	Kinball and Barr Mac. Des.-	3000-4000
	Machinery Reference Series-	up to5000
Crossheads on Guides		35-70
Driving Journals	Unwin Machine Design	200-250
	Halsey Hand Book	up to550
	British Railway Practice	250-300
	Kinball and Barr Mac. Des.	190-220

Machinery Reference Series		
	Fast Passenger	170
	Freight	180
	Switching	190
Reciprocating motion parts		
Cast iron on babbit metal		200-300
Cast iron on cast iron slow		60-100
Cast iron on cast iron fast		40-60
Tender and railway carriage journals		
	Unwin Machine Design	250-450
	Halsey Hand Book	300-425
	British Railway Practice	up to350
	Kinball and Barr Machine Des.	-300-325
Maximum bearing pressures		
Hardened steel on hardened steel		2000
Unhardened steel on gun metal		800
Mild steel on iron or bronze		500
Mild steel on wrought or cast iron		350

These pressures are all in pounds per square inch of projected area of the bearing. The authority for the pressures given for reciprocating motion parts and maximum bearing pressures, which is not given in the table, is Unwin Machine Design.

EQUALIZATION AND SPRINGS

The spring rigging of locomotives and the equalization of the load of the locomotive on the various springs is a problem in the design which should receive very careful consideration. There are no short cuts in getting the figures which must be used in calculating these factors.

The weight of each part of the locomotive should be very carefully calculated and the moment of each separate part should be calculated about the front truck or other convenient point. In this way the center of gravity is determined. With this known it is next necessary to so arrange the spring rigging that the center of gravity of the locomotive will fall at the center of equalization. The boiler may be shifted to alter the center of gravity to quite an extent.

The front group of wheels should be cross-equalized and the others equalized longitudinally. By so doing any shock on any set of wheels or on any wheel will be transmitted to the other springs of the same set.

Where no front truck is used it is usually customary to calculate the moments of the various parts about the center line of the cylinders.

Locomotive springs of the semi-elliptic type are fastened together in the middle with a band which is welded on while hot. When this band cools it contracts and draws the leaves of the spring tighter together. The spring should be so designed that it will be drawn out straight when the maximum load is placed on it. The endwise compression on the leaves of the spring depends on the curvature; so it should be as small as possible.

Sometimes springs are designed with double curves. This should in all cases be avoided.because when the spring bends the leaves must slide on each other. If there is but one curve they will do so but if there is more than one they will pull apart or "gape" as it is called.

Spring plates should all be the same thickness in the same spring. The ends of the leaves should be cut off in such a way that when the spring is drawn out in a straight line the ends of the leaves will form a straight line. This means that the ends must be cut off at equal intervals. By so doing the spring is made of equal strength throughout its entire length. Enough long plates should be used on top to give the spring the necessary strength at the hangers. It is a mistake to make the top plates thicker because thicker plates are not so elastic and hence will carry too large a proportion of the load. The plates should all be of the same thickness, simply using more of them on top.

The carrying capacity of helical coil springs made up of round wire can be found by means of the following formula:

$$W = .3929 \frac{5d^3}{D}$$

The deflection of a helical spring of the same style is found from the following formula:

$$F = 8 \frac{PD^3N}{G d^4}$$

W=The carrying capacity in pounds.

S=The fiber stress in pounds per square inch.

d=The diameter of the wire in inches.

D=The mean diameter of the coil in inches.

G=The torsional modulus of elasticity of the steel which is, in American practice, 12,600,000; in British, 11,000,000.

P=The load in pounds.

N=The number of coils.

The ratio of the diameter of the coil to the diamater of the wire should never be less than 5.

The fiber stress of small piano wire may with safety be put at 100,000 pounds per square inch. For larger diameters with tempered steel the following may be used:

Diameter of wire	Pounds per square inch
up to 1/8 of an inch	75,000
up to 1/4 of an inch	70,000
up to 3/8 of an inch	60,000
up to 1/2 of an inch	50,000
Larger springs	40,000 or more.

This value is variable on various roads. The Pennsylvania Railroad uses from 60,000 to 70,000. For phosphor bronze 15,000 is used; for brass, up to 5,000. The modulus of elasticity for phosphor bronze is 6,200,000; for brass, 3,400,000.

The deflection of a helical spring can also be found by the following formula which is similar to the previous one given:

$$\frac{d^3}{D^4 \times M} \times 8L = K$$

D=The diameter of the steel wire.

M=The modulus of elasticity which for good steel is 12,000,000.

L=The load.

K=The deflection in fractions of an inch for one coil.

N=The number of complete coils.

X=The total deflection in inches.

KxN=X. d=The mean diameter of coil.

To find the diameter of the spring with the size of the steel, number of coils, and deflection given:

$$d = \sqrt[3]{\frac{X \times M \times D^4}{8L \times N}} - D$$

To find the number of coils when the deflection, load, and size of spring and steel are given:

$$N = \frac{X \times M \times D^4}{d^3} \times 8L$$

The carrying capacity of a helix spring can be found from the following:

$$\text{Carrying Capacity} = \frac{.3927 \times \text{Ult. Ten. Stg. of Steel} \times D^3}{d},$$

The carrying capacity or strength and the deflection of elliptic and semi-elliptic springs can be found from the following formulae:

$$D = \frac{n b t^2 f}{3L}$$

$$D = \frac{4L^2 f}{tE} K = \text{Full elliptic.}$$

$$D = \frac{2L^2 f}{tE} K = \text{Semi elliptic.}$$

P=The safe load in pounds.

n=The number of leaves (one side for full elliptic)

b=The breadth of leaves in inches.

t=The thickness of leaves in inches.

f=The safe fiber stress in pounds per square inch which is usually 60,000 pounds.

L=Free length or the projection of one end from the center band in inches.

D= Deflection in inches.

E=Modulus of elasticity-30,000,000

$$K=\frac{1}{(1-r)^3}\left[\frac{1-r^2}{2}-2r(1-r)-r^2\log_e r\right].$$

$$r=\frac{\text{Number of full length leaves}}{\text{Total number of leaves}}$$

Another well known formula for getting the deflection of semi-elliptic springs is the following:

$$R=\frac{S^3}{b\times t^3\times N}\times 1.66.$$

b=The breadth of the plates.

t=The thickness of the plates in sixteenths of an inch.

n=The number of plates.

S=The span of the spring in inches.

R=The deflection of the spring in sixteenths of an inch per ton of load.

The total number of plates necessary can be found by the following formula, using the same notation:

$$N=\frac{S^3\times 1.66}{R\times b\times t^3}$$

The greatest working strength, L, can be found by means of the following formula:

$$L = \frac{b \times t^2 \times n}{S \times 11.3}$$

L is in 2240 pound tons.

Use the following formula to find the number of plates when the strength of the plates, span, and size of the plates are all given:

$$\text{Number of Plates} = \frac{L \times S \times 11.3}{b \times t^2}$$

To find the curvature or set required for the spring before loading:

$$R_1 \times S_1 + \text{Desired Working Compass.}$$

R_1=Deflection in inches per ton.

S_1=Working strength in tons.

The sum given by the formula is the whole original set to which should be added from 1/8 to 3/8 of an inch for permanent set.

The size of the locomotive springs can be determined from the following formulae which are similar to the others but are those given by another authority:

Let:

P=The load on one end of the spring.

S= Allowable fiber stress, 8,000 pounds per square inch.

b=The width of the springs.

h=The thickness of the spring leaves.

n=The number of the spring leaves.

l=The length of the spring from edge of spring band to the point of suspension of the spring hanger.

The size of the springs is determined from the following:

$$P = \frac{S b h^2 n}{6L},$$

Deflection of the spring is found from the following:

$$Def. = \frac{2PL^3}{E b h^3 n}, \qquad \text{in which}$$

E=31,500,000.

36 to 42 inches gives the best length of spring.

3/8 to 7/16 of an inch is the best thickness.

For helical springs carrying power is determined from the following:

$$\text{Carrying Power} = \frac{S \pi d^3}{8D} = P.$$

The deflection of helical springs is found from the following:

$$Def. = \frac{8PD^2}{\pi G d^4} = F. \qquad \text{in which}$$

G=The modulus of elasticity for torsion-12,000,000.

The length of the wire between the ends of the coil is found from the following formula:

$$L = \frac{FG\pi d^4}{8PD^2}, \quad or \quad \frac{FdG}{Ds},$$

LOCOMOTIVE BRAKES

Locomotive brakes are of three types: air, steam, and vacuum. There are several types of air brakes: straight, automatic, and straight automatic which is a combination of the first two. Straight air brakes work very similarly to steam brakes in that the air pressure is applied directly to the brake cylinder through the engineer's valve. In the automatic air brake system the air which goes to the brake cylinder is stored in the auxiliary tank. When the brakes are applied the engineer works his air lever which releases the air in the train line. This release of the pressure in the train line causes the triple valve connected in the system to allow the air in the auxiliary tank to flow into the brake cylinder and apply the brakes. With this system of brakes if there happens to be any break in the train line the brakes of the entire train will operate. This system is best for freight and passenger service while straight air is best for switching where the engine is not as a general rule connected with the train it is pulling.

The braking power applied to any wheel on a locomotive should be proportional to the load on the wheel. The proportions which should be used for good results are: 75% of the load on the drivers, 60% of the load on the trucks, and 100% of the light load on the tender.

The equalized pressure on the brake cylinder should be 60 pounds per square inch; the maximum pressure, 85 pounds per square inch.

The maximum stress in brake levers should be 23,000 pounds per square inch; in the rods, 15,000; except in the jaws where it should not be over 10,000 pounds per square inch. No rod should be less than 7/8 of an inch in diameter. The maximum shear on the pins should not exceed 10,000 pounds per square inch and the bearing value of the jaw, 23,000 pounds per square inch.

The retarding moment of a block brake that has worn smooth may be calculated by means of the following formula, M being the retarding moment in inch pounds:

$M=1/2JQdf$ in which

Q=The force pressing the block on the wheel.

d=The diameter of the wheel in inches.

f=Coefficient of friction. Metal on dry metal=.15.

$$J=\frac{1}{1-\frac{1}{3}\,\text{sine}^2\left(\frac{b}{2}\right)},$$

b=The angle subtended from the tips of the blocks to the center.

COUNTER BALANCING LOCOMOTIVES

The counter balancing of the reciprocating parts of a locomotive is done to accomplish two things: first to keep the locomotive from nosing or moving from side to side and second to keep the wheels from pounding on the rails. The rotating parts that are unbalanced must be balanced also.

In four cylinder locomotives the reciprocating parts are counter balanced by placing the high and low pressure cylinders on each side opposite to each other so they counter balance each other. In this way the high and low pressure pistons are always traveling in the opposite directions on the same side and in this way they counter balance each other. Here it is very important that the riciprocating parts for both the high and low pressure cylinders are the same. The high pressure cylinder is usually placed on the inside of the frames.

By counter balancing the reciprocating parts with revolving weights the counter balance is in a horizontal direction only. The counter weight will cause an up and down unbalanced force. This force is greatest at the upper and lower crank pin positions and becomes zero when the crank is on the dead center position. In some experiments where the counter weight was made excessive it was found the unbalanced force was enough to cause the wheels to leave the rails at each revolution.

In the new design of counter weights place the center of gravity opposite the crank pin and as far from the center as possible. Have the center of gravity of the counter weight come as near the plane of the rods as possible, allowing the necessary clearance. To get the weight of the reciprocating parts take the weight of the piston, rod, and crosshead, and the weight of the front end of the connecting rod. The other end of the connecting rod should be figured as rotating. It is necessary to weigh both ends of the connecting rod in order to get this accurately. For the rotating parts take all of the truly unbalanced parts and add the weight of the back end of the main rod to it. The weight of the unbalanced weights for each wheel should be figured.

In high speed locomotives of the larger sizes the centrifugal forces caused by the parts which must be counter balanced amounts to as much as 90,000 pounds. This is first forward, then backward, of course.

In getting the loads carried by the crank pins they should be taken either at the top or at the bottom position on account of the angularity of the rods.

The counter balance need not, according to some authorities, be distributed equally between the drivers but may be distributed in order to get equal wheel loads. This, of course, means the counter weight of the reciprocating parts only.

On small driving wheels it is usually customary to cast the counter weight in the driver but on the larger drivers and where more precision is needed the wheel is cast with a pocket or several pockets for lead counter weight. For large fast locomotives the wheels are accurately counter balanced by hanging the actual weight on the crank pin and adding the counter weight until the wheels are balanced. This precision is not followed on slow locomotives however.

Centrifugal and reciprocating forces should be figured assuming that the maximum speed will be equal in miles per hour to the diameter of the driving wheels in inches. This speed is commonly known as the "diameter speed". Due to inertia the reciprocating parts tend to continue their motion at the end of each stroke with a force about equal to forty times their weight. The overbalance exerts a centrifugal force equal to about forty times its weight and is maximum at the top and bottom position of the crank. This force, of course, tends to cause pounding on the rails when at the bottom crank position and to lift the wheels from the rails when at the top.

From these facts it can be seen that 1/40 of the static wheel load will be the maximum counter balance for the reciprocating parts without causing the wheels to leave the track. A simple rule for counter balancing which will give good results for locomotives of most any class of service is as follows: Keep the total weight of the reciprocating parts on each side of the locomotive less than 1/160 part of the total weight of the locomotive in working order and then balance 1/2 the weight of the reciprocating parts. This rule takes into consideration the dynamic augments at diameter speeds.

Another counter balancing rule is: to set an arbitrary percentage which the dynamic force of the overbalance will be allowed to increase the static weight.

The Railway Master Mechanics Association reported in 1915 that 50% increase in the static weight on the drivers at diameter speed would represent good average practice while much less than this percentage is desired. With modern methods it is possible to cut down the total weight of the reciprocating parts to 1/240 part of the total weight of the locomotive in working order and such a decrease aids greatly in maintenance cost both of the locomotive and of the road bed.

VALVE MOTION MECHANISMS

There are to-day only two kinds of locomotive valve motion in general use. These two are the Stephenson link motion and the Walschaert valve gear. On all of the later types of locomotives the Walschaert is almost universally used. This is due to the fact that the Walschaert motion is outside the frames and is more accessible for repairs and care and it also allows the space inside the frames to be used for bracing the feames. The Walschaert motion gives constant lead and the Stephenson gives a variable lead but as far as operation and economy are concerned there is little difference between the two.

Before attempting to design valve motions make a careful study of Zeuner's diagram which can be found in a great many technical books. This diagram should be used in every step of the design. The elements of this method are too well known to need explanation here.

The following is a list of rules for general practice:

The connecting rod of the locomotive should be at least six times the length of the crank.

The eccentric rods should be at least eight times the throw of the eccentric in length.

The ports in the valve face should be equal to from

1/10 to 1/12 the cross sectional area of the cylinder, in area.

The width of the port should be about 1/12 of the length of the port. The length of the valve is usually a little less than the diameter of the cylinder.

The travel of the valve should be 3 1/2 times the port opening.

The lap of the valve may be calculated or arbitrarily made from 1/5 to 1/6 the travel of the valve in full gear.

The frictional resistance of the valve on the seat may be calculated to be 20% of the load on the valve. The valve stem and other parts should be designed to withstand this load with not over 10,000 pounds per square inch. For the Stephenson link motion the rocker arm should be calculated in two places to determine the size, one near the end for bending and the other for the boss. The eccentric straps should be calculated as a beam fixed at one end and loaded in the middle. Use column formulae for the eccentric rods.

The links are usually made of wrought iron, case hardened and ground to size. Cast steel has been used. Cast steel permits of better distribution of the metal and the cost is reduced to quite an extent, but cast steel is in most cases too apt to be full of blow holes and does not make as good looking a job. The holes in the link should be protected with case hardened bushings.

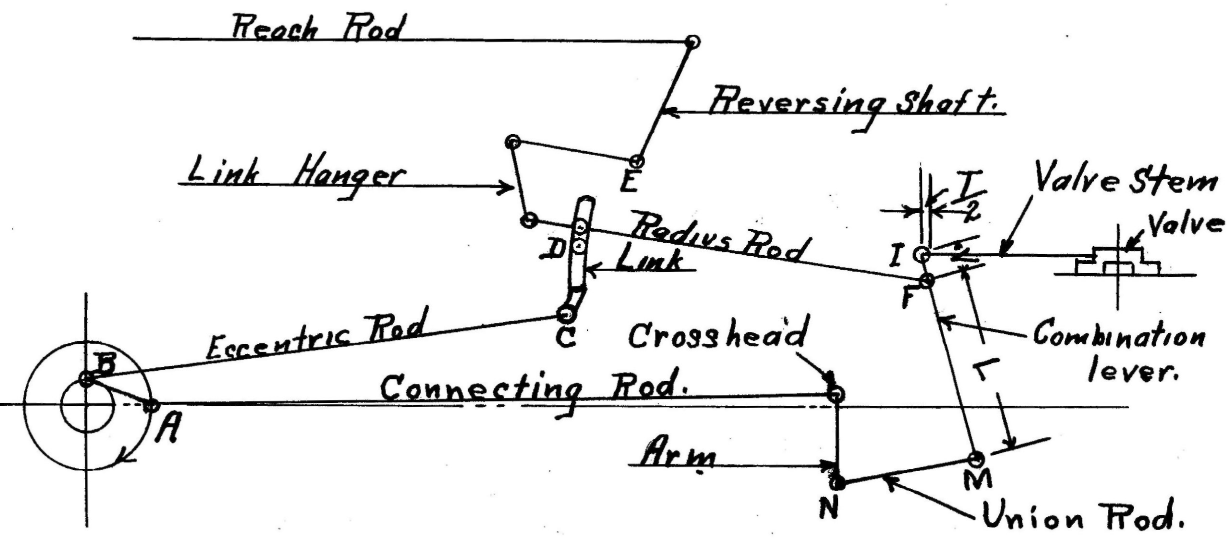
In the Walschaert valve motion the holes in all the parts should be protected with case hardened bushings and the pins should all be case hardened and ground to size. The pins on Stephenson valve motion should also be case hardened. These pins should also be doweled to prevent them from turning. Oil holes should be provided for properly lubricating the pins and other working parts.

The lead on the Walschaert valve gear is usually 1/4 of an inch. On Stephenson valve gears the lead varies from 1/16 of an inch at full gear to 1/2 an inch at mid gear.

The eccentric rod pins are commonly placed 3 inches back of the link arc on ordinary locomotives; the saddle stud, from 5/8 to 1 1/2 inches back. The saddle stud is commonly set by trial in the shop for the first engine.

The maximum economy of operation is obtained with the cut-off not later than 25% of the stroke when in full gear. This should be borne in mind when designing the valve motion.

When it is desired to lay out a Walschaert valve gear the following points should be very carefully noted:



T=Twice the sum of the lap plus the lead. This factor remains constant. By reversing B to lead the crank 90 degrees the conditions of the valve operation are reversed. The same thing is accomplished by moving the link block from one end of the link to the other. The link in the case of this valve gear remains stationary, being pivoted on a trunnion on each side, while the link block is moved from one end of the link to the other. The length of the combination lever is obtained by the following formula:

$S:t=L:V$ or $V=\frac{Lt}{S}$ in which

S=The stroke of the piston.

t=Twice the sum of the lap and the lead.

L & V are as shown in the sketch.

For outside admission of steam as in the case of slide valves the point F falls below the valve stem.

Take great care in placing the link hanger and shaft so it will lift the link block properly.

The radius of the eccentric crank is found by means of the following:

(a) Outside admission, that is, slide valves:

$$b = \frac{R\sqrt{a^2-c^2}}{R+c},$$

(b) Inside admission, that is, piston valves:

$$b = \frac{R\sqrt{a^2-c^2}}{R-c},$$

a=One half the travel of the valve.

b=One half the travel of the link block at F.

c=The lap and lead of the valve.

R=The radius of the main crank.

1/2 the travel of the point F can be assumed as 1/2 the travel of the link block.

The link should not be allowed to swing through more than 45 degrees.

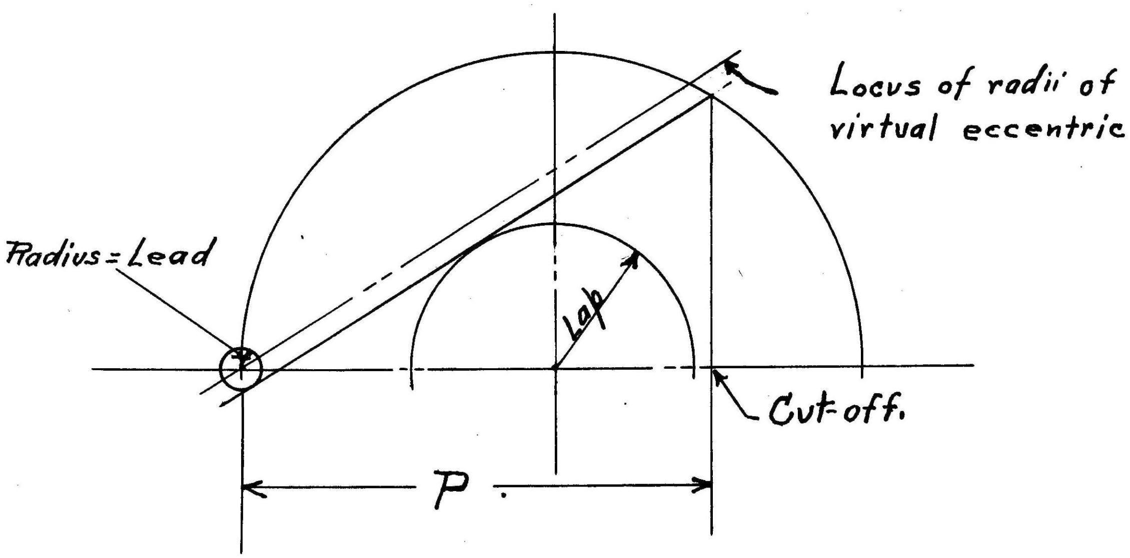
The point C should be back of the tangent to the link passing through D. In ordinary conditions this distance will be from 2 to 5 inches.

Locate the center of the reversing shaft so that the nearest correct position of the hanger will be between

30% and 60% cut-off on the forward stroke and at **half** gear on the reverse operation.

Make the union rod at 90 degrees to the combination lever when it is in a vertical position.

To get the lap of the valve for a given lead when the Walschaert valve gear is used:



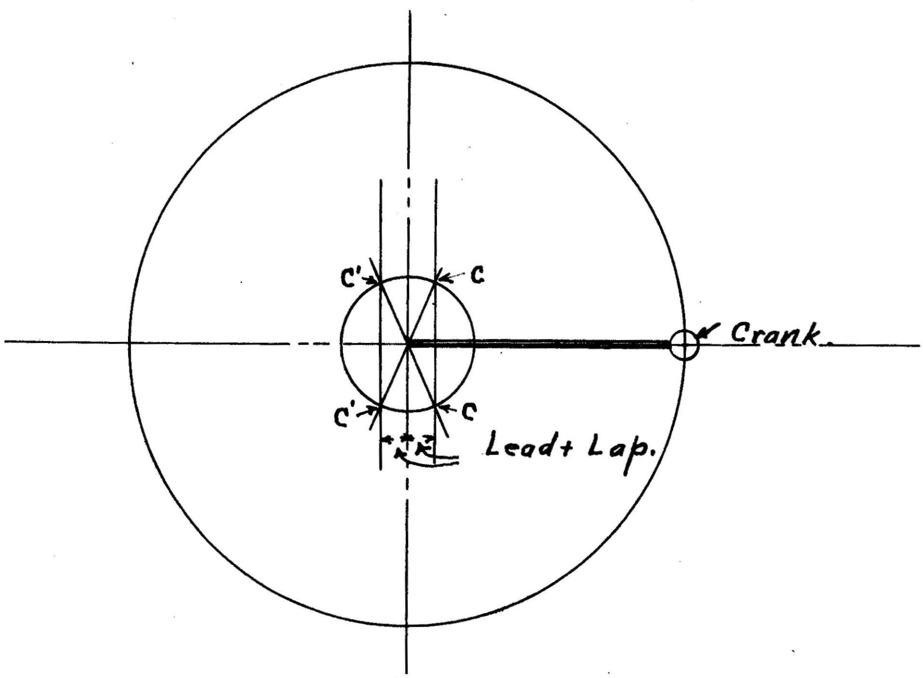
The combination lever should be at least 2 1/4 times the stroke in length from point F to the lower end.

To lay out a Stephenson valve gear the following rules should be borne in mind:

The center line of the cylinders is placed slightly above the center line of the wheels to partly adjust errors due to the angularity of the connecting rod.

"Line of Motion" is a line from the center of the main driver to the lower end of the rocker arm.

The center points of the eccentrics on the dead center will be each placed horizontally ahead of the center line of the wheel a distance equal to the lead plus the lap.



To get the proper length of the eccentric rods: When the points cc get around to c'c' the eccentric blades are crossed. The fact that they are crossed means that they will move the link a little farther back than they do in the front position. This difference in the amount they move the link should be found and added to the lengths of the eccentric blades to make the link move equally on each side.

The correct point of suspension or location of the saddle stud is determined as follows: The point of cut-off is changed by moving the saddle stud. The inequalities of motion are greatest at half stroke. It is then best to adjust the saddle pin so the point of cut-off is equal for both strokes at this point. Locate the eccentric centers for both half strokes; draw the center line of motion. Next locate the rocker arm and describe the rocker arm pin circle. Next describe the link pin arc from the centers of the eccentrics with the length of the eccentric blade. Now on the arc of the rocker pin lay off each side of center the distance the valve must travel to be equal to the lead plus the lap. Locate the two links for these positions. These links stand at the points of cut-off. With these two links located as before locate a point on each one by trial on the center line of the link in such a way that these points are equally distant from the link arc and a line through them is parallel to the line of motion. In other words where the center lines of the links cross is the correct position for the saddle stud, provided they are equally distant from the link arc. They must also be equally distant from the hanger suspension pin.

There are a few errors in link motion:

(1) Offset of the eccentric rod pins back of the link arc.

(2) Angularity of and angular vibration of the eccentric rods. (3) Angularity and angular vibration of the connecting rod. These tend to cause unequal events in the two strokes.

The angularity of the connecting rod tends to compensate for the error due to the offset of the eccentric rod pins. As the rod is made shorter the offset of the stud is decreased until with a very short rod the stud is on the link arc. Therefore the stud is offset to take care of the residual error due to the offset of the eccentric pins.

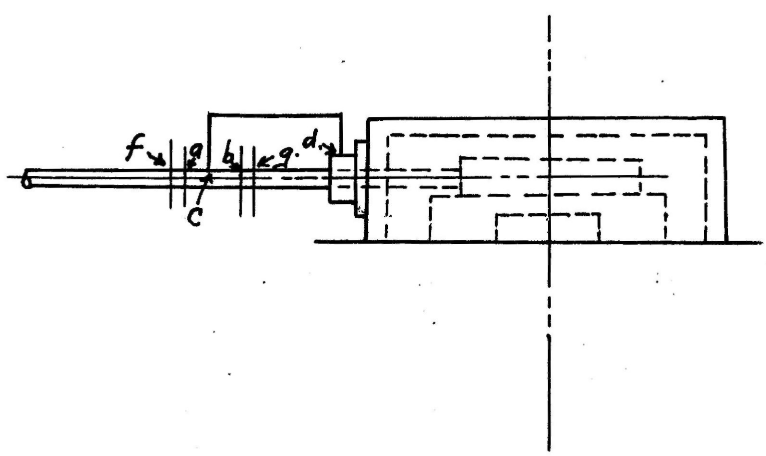
Ordinarily the offset of the saddle stud is determined in the shop after the locomotive is built by using an adjustable stud and saddle. After it is once found for a certain valve motion it can be duplicated on other engines.

The reverse shaft arm should stand horizontal at mid-gear so the adjustment of the saddle stud will be correct for both forward and backward running. The point of suspension of the hanger should be over the saddle stud so the stud will be in a horizontal line when in the two positions for each stroke.

In setting the Walschaert valve gear the process consists principally in determining whether or not the parts were correctly made and checking the various operations of the valve.

First put the crank on center and also the valve by using a tram on the wheel and frame and on the valve stem.

Second place the link block in center and tram a point on the valve stem from the gland. Turn to the opposite center and tram another point on the stem from the same origin. The distance between the points should be equal to the lap plus the lead of the valve. If not there is a mistake in the design or workmanship of the lap and lead lever.



Next, examine the distances fa and bg. They should be equal to the lead of the valve, fg being equal to the lap plus the lead times 2. If they vary from that value correct the length of the valve stem. Trouble may be caused by the radius rod length.

Next, put the gear in the forward position with the block placed for maximum travel. This point can be determined only by experiment. Place the crank on the forward center and tram to the valve stem. Set in back gear and note whether the valve moves or not. This point should check with the mid-gear position and not move for changing from full forward gear to full back gear the same way.

In designing the Walschaert valve motion the irregularities of travel should not be more than 1/4 of an inch. The valve should be squared for the position at which it will most frequently work: that is, for passenger engines from 1/3 to 1/2 the stroke for cut-off is the best place for the valve motion to be equal on both ends or square. At the full stroke irregularity will be unavoidable when squared at 1/3 or 1/2 the cut-off but this will be all right. The particular case of each engine determines at what point the valve should be squared.

In setting the Stephenson valve gear proceed as follows:

First remove the steam chest cover and with a tram on the valve stem and stuffing box for the valve at both ends for the cut-off place the valve in center by dividing the distance between the punch marks and using the tram to the same point on the stuffing box.

Second adjust the length of the valve rod so that the rocker stands exactly vertical or at 90 degrees to the valve stem with the valve in center.

Next, adjust the lengths of the eccentric rods, after the valve stem is tightened. Fasten the eccentrics to any position on the axle and turn the engine over. Note: Put the link in the extreme forward cut-off position first. Note whether the valve travels an equal amount on each side of center. If not, adjust the forward eccentric rod until such equal travel is obtained.

Next, raise the link to the extreme backing position and adjust the length for equal travel in the same way for the forward rod. Use a tram on the valve stem for making the previous test. When set, check to make sure they are accurately set.

Next, set the eccentrics. They are usually keyed to the axle so this is not necessary. First put the crank on forward center, using a tram on the wheel and guide yoke, moving the crosshead 1 inch each side of center to locate the points on the wheel. This should be set exactly. Always move the wheel in one direction to avoid lost motion in setting the eccentrics.

Set the crank on forward center. Drop the link to its extreme forward position. Loosen the eccentric. If the link block is attached to the same rocker arm as the valve rod the eccentric leads the crank; if a reversing rocker arm is used the eccentric follows the crank. Turn the eccentric until the valve is opened by the amount of lead that it is desired that it should have. Then fasten in position. Raise the link and set the backing eccentric in the same way. Again drop the link and see if the lead has been changed. If it has, reset and follow the same rules.

Next, turn the wheel until the other dead center is reached and see if the same lead is given there. If not go over the lengths of the rods and the valve stem again. If they are set properly the same lead should be obtained unless the design is faulty. If this does not occur lengthen and shorten the rods of the eccentric until it gives equal lead. Set the opposite eccentrics in the same manner.

LOCOMOTIVE CYLINDERS

Cylinders for locomotives should be made of cast iron which should have a tensile strength of 25,000 pounds per square inch minimum. The casting should be fine grained and free from blow holes.

The center line of the cylinders should be placed slightly above the center line of the drivers in order to partially overcome the effect of the angularity of the connecting rods. This helps although it does not entirely overcome this effect as can be seen by laying out the line of motion.

To get the necessary diameter of the cylinders use the following formula:

$$d = \sqrt{\frac{T \times D}{P \times .85 S}},$$ in which

d=The diameter of the cylinders.

T=The required tractive power.

D=The diameter of the driving wheels.

P=The boiler pressure.

S=The stroke of the piston.

85% is the mean effective pressure in the cylinders.

The length of the working barrel of the cylinders is the length of the stroke plus the width over the rings minus 1/4 or 1/8 of an inch for over-run on each end.

The inner edge of the ports should be opposite the outer edge of the packing ring at the end of the stroke.

The ends of the cylinders outside the working barrel should be from 1/4 to 1/2 of an inch more in diameter than the barrel.

The clearance between the piston and the cylinder head should be not less than 1/4 of an inch nor more than 3/8 of an inch. The clearance on the end toward which the piston is drawn when keyed up should be from 1/6 to 1/8 of an inch more than the opposite end.

The steam ports into the cylinders are usually made about 2 inches less in length than the diameter of the cylinder.

The thickness to be used for the cylinder walls can be found from the following formula which should be increased about 50% for wear and for shock:

$$T = \frac{dP}{2S},$$ in which

T=Thickness in inches.

d= Diameter of the bore in inches.

P=The steam pressure in pounds per square inch.

S=The safe stress in the metal in pounds per square inch. This is usually allowed as about 2,500 pounds per square inch.

The cylinder heads should be made about the same thickness as the walls of the cylinders if they are made of cast iron. If they are made of cast steel they may be made thinner in proportion to the strength of the two. The back head should be made slightly heavier to bear the guide. For fastening the head to the cylinder 7/8 to 1 1/8 inch studs should be used at a pitch of about 5 inches. For small engines this may be made smaller.

Cylinder castings are usually made in halves, the parting line being on the center line of the locomotive. The machine operations which the cylinder undergoes are about as follows:

The cylinders are set up in gangs on a planer and the surface for bolting the cylinder together is planed and part of the valve face also. Next, the set up is changed and the frame fit is planed. Then the ports of the valve are milled. The casting is then bolted to a fixture on a boring mill which locates the center of the bore from the frame fit, in this way holding an accurate dimension with relation to the finished surfaces to which the frame will be bolted . During this set up the ends of the cylinders and the counter bores on each end are finished.

Next, the cylinder is drilled for the cylinder head studs. This drilling operation is done with a jig which locates from the counter bore of the cylinder and from the horizontal frame fit. The drilling for the frame bolts, exhaust nozzle, and holes for connecting the cylinders is done with jigs, Usually only one cylinder is drilled for the connection bolts, the ones in the other cylinder after the cylinders have been properly lined up together.

The steam joints of the cylinder heads are ground in on both ends with the cylinder heads that are used in them. This is done with oil and emery and may be done by rotating the cylinder head back and forth through about 45 degrees either by power or by hand.

The cylinder saddle is drilled from a lay out and is chipped to fit the barrel of the boiler either by marking the radius with a fixture or by lining up the boiler in place and marking the cylinder from it. The chipping is done with a pneumatic chipping hammer.

The holes in the smoke box of the boiler are drilled through the holes in the casting after it is in place. In this way there is no necessity for keeping the holes accurate. When a cylinder is replaced on an old engine the saddle holes are drilled from the holes in the smoke box, they being left blank by the manufacturers.

PISTONS AND PISTON RODS

The thickness necessary for strength for pistons can be found from the following formula for any point out to the rim of the piston:

$$A = \frac{\pi R^2 P - \pi r^2 P}{S},$$ in which

A=The area of the metal on given circle.

R=The radius of the cylinder.

r=The radius of the section sought.

P=The boiler pressure.

S=The allowable stress in the metal.

The piston may be considered as a beam, that is a section of it may be so considered, with one half of the load on the section concentrated at the center of gravity of each half of the section. The distance of the center of gravity of each half section from the center is .42 of the radius of the piston. With this information the modulus of section can be found and the necessary thickness figured. The following formula can be used in getting this modulus:

Let M be the modulus of section.

$$S = \frac{.42 R \times R^2 \times P}{2M},$$

For disk pistons the thickness can be found with the following formula which gives the thickness near the boss:

$$t = CD\sqrt{P}.$$ in which

D=The diameter of the piston.

P=The boiler pressure in pounds per square inch.

t=The thickness of the metal.

c=.0046 for cast steel, .008 for cast iron.

The thickness at the rim should be .6 of the thickness at the boss.

For conical pistons the thickness may be found by the following:

$$t = .003D\sqrt{P}.$$

This gives the thickness for cast steel only.

The dimensions for piston rings should be about as follows: 18 to 20 inch pistons-1/2 of an inch wide and 5/8 of an inch deep; 20 to 24 inch pistons-5/8 of an inch wide and 3/4 of an inch deep. An approximate rule for finding this size is: The ring should be 1/8 of an inch less than 1/30 of the diameter of the cylinder in width and 1/8 of an inch more in thickness. Ordinary rings are square, being from 3/8 of an inch square to 7/8 of an inch square.

Either two or three rings should be used on pistons. This number depends on the conditions and no special rule can be given.

Piston rods are usually about 1/6 of the diameter of the cylinder in diameter. The stress in the rod should never exceed 10,000 pounds per square inch at the root of the thread or across the key section. The crosshead key should hold the full load of the steam pressure with not to exceed 10,000 pounds per square inch shearing stress or 24,000 pounds per square inch in bearing.

The body of the piston rod should be figured for compression by the column formula for columns with free ends. There are several acceptable formulae for finding this value:

$$\frac{P}{A} = \frac{S}{1 + \frac{g l^2}{r^2}} \quad \text{in which}$$

P=The total load in pounds per square inch.

A=The area of the section in square inches.

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

l=Length of the rod in inches.

r=The radius of gyration of the section, d/4.

g=4/25,000 for steel, 4/36,000 for wrought iron.

The Merriman formula:

$$C = \frac{B}{1 - \frac{nB}{10E}\,\frac{l^2}{r^2}}, \quad \text{in which}$$

C=The maximum compressive stress in pounds per square inch.

B=Load per square inch on the rod.

E=The modulus of elasticity, 30,000,000 for steel and 25,000,000 for wrought iron.

n=1 for round end bearings and 1/4 for square end bearings.

After getting the diameter of the rod by means of these formulae it should be checked for strength in tension through the section at the threads on the piston end and through the section through the crosshead key slot. If these values are correct the rod may be considered properly designed.

The machine operations on the piston are as follows:

Pistons are bored, faced, and turned; also the grooves for the piston rings are turned and then ground to a smooth accurate finish. The tapered ends, both for the crosshead and piston ends, are ground to gauge. The gauges are made so that a press fit will be had for the piston head end and a drive fit for the crosshead end. The slot for the key in the crosshead end of the piston rod is either drilled in a jig and then chipped or it is milled.

The piston rings are made from a cylindrical casting long enough to make a number of rings with flanges on one end for bolting it to the face plate of the lathe. This casting is bored and turned, allowance being made in the diameter for the spring fit. The solid rings are then cut off from the casting the proper thickness. Next the slot is milled in the ring. Then a number of the rings are put in a fixture, sprung together, and clamped. The outside diameter is then turned accurately to fit the bore of the cylinder. Frequently the outside diameter is ground.

As the ring is taken out of the fixture which holds it so the slot is closed, the ring springs out and takes its normal position. By machining the ring in this way it will fit the cylinder perfectly and spring out, making a steam tight joint.

LOCOMOTIVE CROSSHEADS

There are several types of locomotive crossheads in use, all of which give good service when used in the proper way. The most common of these is the alligator crosshead which is made for two guide bars, one above the crosshead and the other below. This crosshead is used on large, high power locomotives in which care must be taken to see that perfect alignment and bearing are obtained. There are several others that are used to a greater or less extent, some on one class of engines and some on another.

The Laird crosshead is probably the next in importance. This crosshead is used in connection with two guide bars or with only one but in this type both bars, or the one, are on top, or, in other words, both are above the wrist pin. This crosshead is used on smaller locomotives which are usually less exacting about the line-up of the crosshead.

The single bar crosshead is used almost universally on small locomotives and on cheap light road engines. There are several styles of single bar crossheads, the main difference being whether they open on the side or the top for the bar.

The least important style is the four bar crosshead which uses two bars above and two below. This style is not seen very often.

A crosshead should be designed in such a way that the proper bearing pressure is obtained both on the top and bottom side. It is first necessary to figure the necessary size of the guide bar in order to know how wide the bar will be; them by calculating the maximum pressure and dividing by the allowed bearing pressure in pounds per square inch the necessary area of the crosshead is obtained. The same area is used above and below.

The bearing on the guide is obtained, in the case of small crossheads, by means of brass gibs above and below that lip over the ends of the crosshead. The sides are usually babbited. In the case of the larger locomotives the crosshead is lined with blocked tin or in many cases the large crossheads have brass gibs also.

The hole in the crosshead pin or wrist pin is tapered. This taper varies from 3/4 of an inch in 12 to 1/8 of an inch in one inch. 1/4 of an inch in five is a well known standard. The wrist pin should be held with a dowel and two nuts or castle nut, and case hardened.

The slot for the crosshead key for the piston rod is usually made straight and the key is made tapered about 1 inch in 32 or 1/8 of an inch in 5, or something similar.

The key and slot are made the same thickness throughout their length.

The seat in the crosshead for the piston rod is tapered from 1/2 of an inch in 12 to 1 inch in 16. The hole should be made to give a light drive fit for the piston.

Crossheads are made both of cast iron and also of cast steel. The material used is usually determined by the cost and by the necessary strength. Cast steel crossheads are, of course, much lighter for the same strength and therefore are to be preferred where possible because they reduce the weight of the reciprocating parts.

The crosshead is oiled from the top of either the crosshead or the guide-bar by means of an oil cup. The oil is allowed to run down between the gib and the top of the bar. From there it finds its way down between the bottom of the guide and the bottom gib. The crosshead pin is oiled through the side of the crosshead either by means of an oiler or an oil hole.

In machining crossheads the guide bar fit should first be planed. This will give locating surfaces for holding fixtures which must be used for slotting for the main rod fit and for boring the piston and wrist pin fits. The fixtures for doing these operations should be made of heavy rigid cast iron with an arm to fit inside of the guide bar fit and with the center located in such a way that the crosshead will be centered in the lathe for doing the above operations. Holding appliances for securely fastening the crosshead in position should be made integral with the fixture.

All of the drilled holes in the crosshead should be drilled with jigs and also located from the guide bar fit. Any plates or gibs used in connection with the crosshead should also be drilled in jigs, thus insuring interchangibility.

LOCOMOTIVE GUIDE BARS

Guide bars for locomotives are made of cold rolled steel and ground perfectly smooth on a plane surface grinder before they are used. The bars are sometimes single and sometimes double. On large locomotives there are usually two bars used and on switchers it is usually customary to use only one.

The guide bars should be kept as close to the center as possible. The pressure between the guide and the crosshead should not exceed 70 pounds per square inch of projected area. It is best to cut this down by half if possible. The pressure against the guide bar can be figured by means of the following formula:

$$P' = \frac{Pr}{L}, \quad \text{in which}$$

P'=The pressure against the guide.

P=The pressure of the steam against the piston.

r=The radius of the crank.

L=The length of the connecting rod in inches.

In using this formula the distance between the center line of the cylinders and the wheels should be added to r.

The strength of the guide should be such that the deflection of the bar will never exceed 1/32 of an inch and should be kept as close to .01 of an inch as possible for the maximum value. When the necessary size of the bar has been calculated to give this deflection 1/8 to 1/4 of an inch should be added to the thickness of the bar for wear.

The guide yoke should be of the same strength as the guide and it should be so designed that it will give cross bracing to the frame. The yokes are usually made of cast steel for the part to which the guide bars are fastened and the slab across the frame, of ordinary mild steel.

The holes in the guide bar should be drilled in a jig which locates from the ground sides of the bar and from the end which bolts to the cylinder head. This jig should be made in the form of a long bar with stops arranged at the sides and ends. The holes should be drilled 1/64 undersize and reamed in position when assembling. The offset ends of the guide bar should be milled to give them a smooth finish.

LOCOMOTIVE DRIVING BOXES

Locomotive driving boxes are all of very similar design, differing principally in size which is, of course, dependent on the size of the locomotive. They are made of both cast iron and cast steel. Cast steel boxes are now used on large locomotives almost exclusively. Cast iron boxes are used to some extent on dinkies. In most cases where they are used it is desirable to save in first cost.

The bearing of the box on the journal of the drivers is always a half round moon shaped brass bearing. This bearing is usually made about 1/4 as thick at the top as the diameter of the journal. The thickness decreases toward the sides. On small boxes this bearing is cast in the box solid but on larger boxes the box is slotted or shaped out and the brass turned for a press fit. The brass should be pressed into the box with a pressure of about 15 or 20 tons. The brass is bored out after it has been pressed in place.

The top of the box is made with two pockets for the driving spring saddle. These are placed on each side of a pocket in which is placed waste into which oil is poured for lubrication. The oil finds its way into the bearings by way of an oil tube in the center or two oil tubes, one on each side. There is always an oil tube on each side to oil the shoe and wedge. The oil, after it has run over the bearings, finds its way into a cellar at the bottom of the box. This cellar is separate from the box, being fastened to it by means of bolts. The cellar fastens between the bottom jaws of the box and is filled with waste which catches the oil, and the waste, being in contact with the journal, keeps the journal lubricated.

To protect the frames from wear from the constant motion of the driving boxes, shoes and wedges are used. The wedge is also used to take up the loss due to wear. The boxes are made with channels on each side, front and back, which fit over the sides of the shoes and wedges. The shoes and wedges, in turn, fit over the frame pedestal jaws. These are made of cast iron and can be readily replaced or relined when worn. The wedge is adjusted by means of an adjusting screw through the pedestal binder. The shoe is stationary.

On large boxes it is customary to allow for a slight longitudinal and transverse rotation. This is accomplished by allowing on the center of the wearing faces of the boxes, all around, a flat space of about 2 inches. On each side of this flat space the box tapers away on a slope of about 1/8 to 1/4 of an inch in 12.

In many cases slots are cut across the faces of the brass bearings and babbit linings are put in. These linings are held by means of babbit anchors in the brass. The groove for the oil is usually cut straight across the top of the brass and is large and deep to insure a distribution of the oil across the face of the bearing.

The size and length of the bearing in the box is, of course, determined from the weight on the box and the speed at which the locomotive will run. The allowable bearing pressures can be found under that subject.

In order to machine driving boxes the sides of the boxes should first be planed; next the boxes are bolted to a fixture with the planed surfaces for locating the boxes. The channels and edges of the boxes are then milled with a gang of milling cutters. The boxes are then turned over and the other channels and edges milled. The fit for the brass may either be slotted or shaped before or after the sides of the boxes are finished, provided that if it is done afterwards the finished surface is held central with the channels. After the channels have been milled the boxes are bored in a fixture which locates the boxes from the channels. The bore must be held central between the channels. After the boxes have been bored the cellars are put in position in the boxes and the holes drilled for the bolts or cotters which hold them in position. The oil holes are drilled and oil grooves in the channels, milled; then the boxes are ready to be fitted up with the journals.

LOCOMOTIVE FRAMES

At the present time about the only kind of frames that are used is cast steel. Before cast steel became so generally used the frames were made of wrought iron or structural steel. It has been found that better and cheaper frames can be made of cast steel and also that they can be made of any type or style desired. Ordinarily the frames are made in two parts, one on each side. At the present time frames are being made in one piece with rigid cross-ties between the two sections. This gives a very rigid and good frame but may hot become generally used on account of the manufacturing difficulties encountered.

When designing wrought iron frames do not allow more than 4,000 pounds per square inch stress and for cast steel 3,000 pounds per square inch. By this stress is meant the stress due to the steam pressure. With this factor of safety a sufficient allowance will be had for the other stresses.

The ratio of the depth of the frame to the width should be as nearly as possible 5:4.

The back face of the pedestal leg is always made sloping so that a tapered wedge can be used between it and the driving box for taking up the play between the box and the frame. This slope varies from 3/4 of an inch to 1 inch in

12. The pedestal binders below the driving boxes should be made of wrought iron. In small engines a bolt is sometimes used in place of the ordinary bar.

In the design of frames it is customary to avoid all sharp corners and to make the connections between the various parts of the frame with smooth curves.

In order to machine the frames plane the sides, top and bottom; next the frames must be marked for slotting from a steel template. This template should be made, the exact shape of the frame, of about 1/8 inch steel. It is necessary to mark only one frame for a guide. Three or four or more frames may be slotted at one time by fastening them one on top of the other. The pedestal jaws should be slotted to gauge between centers and from the cylinder fit. The top of the frames should be drilled with a long bar jig which locates from the sides of the frame and from one of the perpendicular surfaces of the pedestal jaws. The holes for the binders on the bottom should be drilled in a jig which locates from the lug used to locate the pedestal binders. The binders should be drilled with a similar jig. The holes for bolting the frames to the cylinder should be drilled in a jig which locates from the back end of the jaw. All of the holes should be drilled 1/64 undersize and reamed in place when assembling for the the standard taper bolts.

CONNECTING AND SIDE RODS FOR LOCOMOTIVES

The connecting rods for locomotives are made from steel forgings and carefully machined to length and dimensions. In the design of these rods it is necessary to consider only the stresses due to compression and bending. For bending stress the load must be considered along both axes. On the vertical axis figure the stress due to loading a column with round ends and due to the inertia of the rod caused by the acceleration of the rod. Both of these must be added together to get the total stress. Along the horizontal axis the rod must be considered as a column with square ends and also the bending due to the load along the horizontal axis must be added.

First find the compressive stress in pounds per square inch.

The vertical bending moment is calculated by assuming a maximum rate of speed in miles per hour.

The lateral stress on the rod is calculated by the following formula, assuming a column with square ends:

$$F = \frac{B}{1 - \frac{NBl^2}{10ER^2}} = \frac{10BER^2}{10ER^2 - NBl^2}.$$

In the above formula:

F=Maximum compressive stress in pounds per square inch of concave side of rod.

B=Compressive stress in pounds per square inch due to direct loading.

E=30,000,000.

N=1/4 for square ends.

l=Length of the rod in inches.

r=Radius of gyration of the section under consideration.

For the vertical bending use the column formula for a column with round ends which is the same as the other only that n=1 and r is about the other axis.

For considering the bending due to the load and the inertia of the rod the centrifugal force of the rod must be calculated. This can be calculated by means of the following:

$$C = \frac{G v^2}{g r}$$

in which

G=Weight of the rod in pounds.

v=Velocity in feet per minute= $\frac{2\pi R N}{60}$, =.1047 rn.

r=The radius of motion in feet.

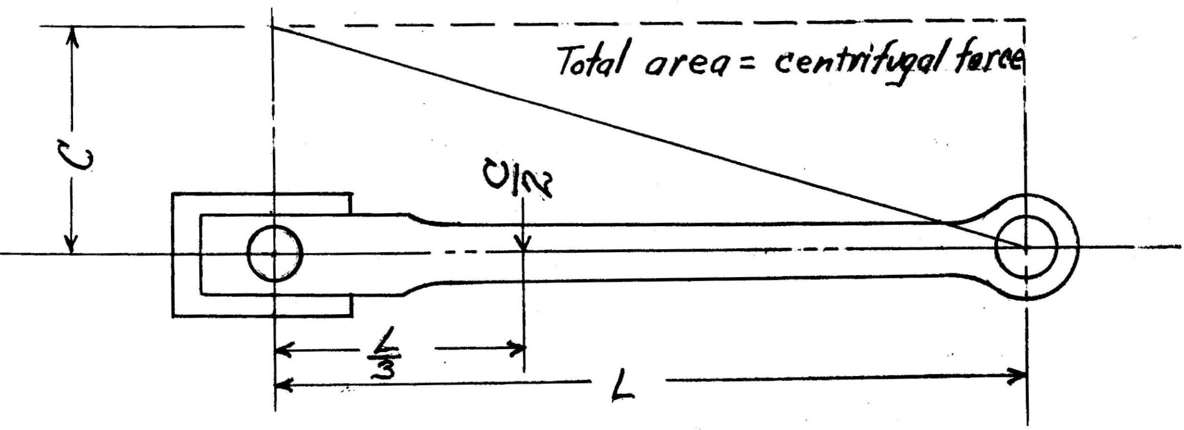
g=32.2.

n=R. P. M.

This may also be calculated in terms of the stroke, which is applicable to the side rods as well as the main rods:

$$C = \frac{38.4\,G \times Stroke}{24} = 1.6\,G \times Stroke.$$

The action of the centrifugal force on the main rod may safely be assumed to be as follows:



For all practical purposes 1/2 of the centrifugal force is assumed to act at 1/3 of the total length of the rod from the crank pin end of the rod.

The machine operations on connecting rods are as follows: Mill or plane the sides of the rods when holding them in fixtures. Next the ends of the rods should be drilled in a fixture which is made to serve for drilling pilot holes and also to serve as a template for marking the ends of the rods for slotting. After the pilot holes have been drilled the holes are either slotted or the holes are drilled to size with core drills and slotted afterwards. The bolt holes for straps and oil cups may also be drilled in jigs. The same surface which is used to locate the jig for drilling the holes for the straps should also be correspondingly used for the jig used for drilling the holes in the strap.

CRANK PINS AND AXLES

The sizes of crank pins and axles are usually determined by the wearing surface necessary. In order to get the size it is necessary to calculate the maximum load which will be required to slip the wheel; then by dividing this load by the allowable bearing pressure the square inches of projected area of the journal are obtained. After calculating the size of the journal in this way the maximum fiber stress should be figured to make sure that it is within the allowable limits. These limits are: for steel 16,000 pounds per square inch and for wrought iron 14,000 pounds per square inch. These parts are figured as beams, assuming that the loads fall at the centers of the bearings. The torsional stress in the axle should also be added to the bending stress.

Crank pins are made from forgings, rough turned and ground accurately to size, allowing for a press fit of ten tons per inch of diameter in the driving wheels. The pins are ordinarily made so they can be riveted over after being pressed in. The journals should be ground to a smooth finish.

On the outer end of the pins a washer and nuts are always used to hold the rods on the pins. The washer is always doweled to the pin to prevent its turning. There are numerous designs for these washers but the purpose is always the same.

The axles are rough turned and for the wheel fit an allowance for a press fit of ten tons per inch of diameter is also made. The journals are rolled to give them a hard smooth finish. From .006 to .010 of an inch is allowed for this rolling.

For crank pins used in connection with Walschaert valve motion on which the eccentric cranks are used, the method of locating the keyway so that it will come at the proper angle for holding the eccentric crank is as follows: The keyway is milled in the pin. A template is made for marking on the wheel a line which will be in line with the center line of the crank pin which locates the keyway. The crank pin is set up on V blocks and a line scribed on it through the keyway for the eccentric crank. This line will line up with the line on the wheel when the crank pin is pressed in.

The keyway in the eccentric crank is scribed from the keyway in the crank after the valves have been set. This will take care of any slight errors which might occur in the valve motion parts.

THROTTLES DRY PIPES AND STEAM PASSAGES

In order to obtain the proper flow of steam from the steam dome of the boiler through the throttle, dry pipe, and steam passages, through the valve and cylinders, then out again through the valve and into the exhaust nozzle, it is first necessary to consider the rate of flow which will be obtained in each case to keep the locomotive properly supplied at the necessary running speed. The maximum speed of the piston of a locomotive for maximum pulling power is 250 feet per minute. In order to obtain faster locomotives the size of the driving wheels is increased, thus maintaining the same piston speed. The proper velocity of steam passing through the steam passages is as follows: steam pipes, 8,000 feet per minute; exhaust port, 6,000 feet per minute; exhaust pipe, 4,000 feet per minute.

Assume the piston diameter to be equal to 1; the following table gives the relative size of the steam passages for various piston speeds in feet per minute:

Piston Speed	Dia. Steam Pipe	Dia. Exhaust Pipe	Area Exhaust Passage
200	.158	.223	.033
250	.176	.248	.042
300	.194	.272	.050

There are a number of empirical rules for obtaining the necessary size of the steam and exhaust pipes:

Make steam pipe 1/4 of diameter of cylinder.

Make exhaust pipe 1/30 of diameter of cylinder.

Make area of throttle and dry pipe 1/15 that of both cylinders.

The steam pipes in the smoke box should be 1/12 of the area of the cylinder.

The steam should have as smooth a course as possible throughout its course.

The front end steam pipes, on account of expansion and contraction and irregularities of construction, should have a small amount of flexibility and adjustability. For this reason both the T connection at the top and the cylinder connection are made by means of ball joints. To get both lateral and vertical flexibility, and as a vertical adjustment of 1/8 of an inch or more may be necessary, a ring should be used as the male part of the ball joint. One side of the ring should be spherical and the other flat. Sometimes only one end of the pipe is made with this joint, the other being the simple ball and socket joint. These joints are made steam tight by grinding them in place.

The throttle valve in the dome of the boiler is always of the partially balanced type, one side of the valve being made larger in order to hold the valve closed. These valves are usually made of two cast iron disks with conical edges which fit into corresponding conical seats in the opening in the upper part of the throttle pipe. In order to make these valves work together they are ground in with emery.

The valve seats are machined as follows: A fixture is made for holding the body of the throttle pipe in line with the spindle of the drill press or boring mill and a guide is provided on the fixture at the proper distance to guide a boring bar with a double cutter for boring the two seats at one operation. Next the two cone seats are reamed with a reamer made up in such a way that the distances between the two cone fits are held constant. The reamer is guided on the lower end with a guide which runs in the bored hole. In this way the two cones can be held with exact relation to each other. The valve is turned to gauge in a lathe. After the machining operations the valves are fitted up and ground with emery to make them steam tight.

The male parts of the ball joints and the steam pipes are turned to gauge; the female parts are reamed with ball reamers. The joints are also ground in place to make them steam tight.

REVERSING LINKS

The reversing links used on all locomotive valve motion mechanisms must receive very careful consideration in their design. Links are made in two styles: solid and of the two bar type with blocks at the ends. The dimensions which should be used for a link operating a slide valve engine are as follows:

Let L=The length of the valve

B=The breadth of the valve

P=The absolute steam pressure in pounds per square inch.

R= The factor of computation=$.01\sqrt{L \times B \times P}$

The breadth of the link=R $\times$ 1.6

The thickness of one bar of the link=R $\times$.8

The length of the link block=R $\times$ 2.5

The diameter of the eccentric rod pins=(R $\times$ .7)+1/4 inch.

The diameter of the suspemsion rod pins=(R $\times$ .8)+1/4 inch.

The diameter of the link block pin when overhung= R +1/4 inch.

Links are made of forgings and first planed on the sides to exact thickness. Then for links of the two bar type the bars are set up in a radius planing or radius slotting fixture and the exact radius is planed. The bars are then shaped on the ends to the proper length for the block fit. They are then drilled in jigs. The outside contour of the link is shaped or slotted. This completes the machining operations. The links are then fitted up by hand and sent to be case hardened. After case hardening the link is set up on a fixture on a vertical grinder which swings it in the proper arc and the link block fit is ground accurately to size. The rest of the link is polished.

For links of the solid type the sides are planed to the proper thickness; then the connection bolt holes and holes at the ends of the slot for the link block are drilled in a jig which serves the purpose both of drilling and reaming the holes and also as a template for marking the link for slotting on the outside. The metal in the slot for the link is drilled out on each side by means of two jigs which locate from the drilled holes which were drilled in the previous operation. These two jigs are made in such a way that the holes in the one fall between the holes in the other; thus when the link has been drilled with both jigs the metal in the slot can be knocked out. After drilling the links are set up in a radius slotting fixture and the slot machined to size. Also the outside surface is slotted, having been marked with the jig used in the first operation. The link then undergoes operations similar to those undergone by the link of the double bar type, being polished and ground accurately to size.

JIG DESIGN

In the design of jigs for locomotive parts it is well to keep in mind the necessary accuracy of the part: that is, the jigs need not be any more accurate than the part yet they must be sufficiently accurate to insure that there need be no fitting up on the assembly floor of parts that have been jigged.

On all jigs keep the ends of the drill bushings as near the work as possible, yet where there is clearance between the bushing and the work there should be sufficient room to allow the chips to slip out without crowding the bushing or the jig.

Always keep nearest the jig the ends of the holes which must be most accurate. The jig should always be used in such a way that the ends of the drilled holes which should be most accurate are next to the jig when drilling.

The minimum length of the drill bushings is the same as the diameter of the drill, but where the hole must be accurate it is well to have the bushings much longer. Twice as long or longer is well for accuracy. The length and necessary accuracy and fit of the bushings of course depend on the piece that is to be drilled.

The positive stops on the jigs should locate the work with reference to the holes so that the parts which are bolted together will be located from the surfaces which must fit together or be in line with each other, or the jig will not serve its purpose. It is well to make a careful study of the way the parts are machined and how they must go together before deciding where the positive stops should be and how the piece should be held when it is drilled.

Where plates must be moved on a jig it is necessary to have dowel pins to locate the plates. If this is not done the plate will soon be so loose that it will not be properly located with reference to the work.

If possible the jig should be self cleaning and so it will not have to be moved to change the work. This will allow the jig to be clamped down and will save time in locating it on the table.

Make jigs in all cases so that they can be easily clamped down and unclamped or so that the work can be changed as quickly as possible as one minute saved on one piece will be one hundred minutes saved on one hundred pieces. This holds true on all fixtures and tools of all kinds. Too much care cannot be taken in deciding just how the work should be held, replaced, and the locating points kept clean and free from chips. All these factors must be considered before a good jig can be made.

All positive stops should be case hardened and also the other stops where necessary. The bushings ahould all be made of tool steel, ground to size, hardened, then ground or lapped to exact size. Where it is necessary to prevent a slip bushing from turning a notch should be ground in the side of the collar after it is hardened and a dowel pin used. If the slot is cut before the bushing is hardened it will warp.

There are so many styles and types of jigs for different kinds of pieces that a general outline of how they should be designed is all that can be given here. In the description of the machine work of the various pieces the particular type of jig that is necessary is explained.

All jigs for locomotive work should be made heavy so they may be thrown around as they probably will be. Also make all the parts strong enough that a large wrench may be used on them as it probably will be.

FORGE BLOCK DESIGN

In order to make the forgings for various locomotive parts such as links, connecting and side rods, valve motion mechanism parts, and numerous other forgings of odd shapes and sizes it is necessary to make special forging dies to be used in drop forges, steam hammers, and forging machines. These dies are made either of cast iron or of hardened steel. When made of cast iron the V on the outsides of the dies are usually chilled to save machining. the impressions in the die are cast, then filed and fitted up. It is customary to leave 1/8 of an inch finish on the faces of the dies so that they can be machined to mate together properly.

The dies may form a piece in one, two, or more impressions until the completed shape is obtained. The blocks must part on the center line of the work and the forming or working part of the die should follow the exact contour of the work with the exception of the sides which should be made with a taper of 4 or 5 degrees or about 1 inch in 12, for withdrawing the piece and preventing it from sticking in the die.

When making a pattern for a cast iron die a 1/4 inch shrink scale should be used for measuring up the working part of the die. Where the forging calls for a finish,1/8 of an inch allowance should be made on the pattern. For the tong hold on the work there should be a half round opening which tapers towards the outside of the blocks and which is not less than 5/8 of an inch in diameter next to the work.

When steel dies are used it is customary to machine the die from a solid high carbon steel forging. The impressions are machined in the face of the die, using a 1/8 inch shrink scale for measuring the working parts. Otherwise the impressions are similar to those in the cast iron dies. After the steel blocks have been machined they are tempered before using.

In the design of dies for forging machines where the operation consists of upsetting bar stock it is customary to allow 1/64 of an inch for the grip of the stock when the two blocks are forced together to hold the work. The impressions in the dies are similar to those in the other types of dies, 1/8 inch shrink scale being used in machining the impressions to allow for the shrinkage of the forging. These blocks are usually made of cast iron finished all over with or without hardened steel faces.

ASSEMBLING INSPECTION AND TESTING

The various parts of the locomotive having been made up in the various shops, the last operation in the production is the assembling and making the locomotive ready for service. The cylinders should be set up on blocks in a convenient place in the erecting shop. The frames should then be brought and bolted to the cylinders and allowed to rest on jacks. They should be carefully leveled and lined up with the center-lines of the cylinders. This lining up is accomplished by means of a line located with a spider in the front end of the cylinder. The line is fastened at the other end of the frame by means of a movable post. With a pair of dividers the line is made in the exact center of the cylinder. The distance from the line to the frame is made equal along its entire length on both sides. In this position the cylinder castings are bolted firmly to the frames, thus holding them in proper alignment. The distances from the back faces of the cylinder to the perpendicular face of the pedestal jaws should be checked on each side to make sure that they are the same. This will insure alignment of the axles, provided there is no error in machining the driving boxes, shoes, and wedges.

The frames and cylinders being now in proper relation to each other, all of the crossties, guide yoke, and frame braces are put in position in the frames and their location checked or located from the back ends of the cylinders. The measurements should be taken on both sides to insure their being square with the frames. The crossties and cross members should be firmly bolted with temporary bolts after which the holes should be reamed and bolts fitted, allowing for a drive fit a distance about equal to the diameter of the bolts.

The boiler should next be put in position and located accurately both laterally and vertically, using the frame and mud ring for measurements. The cylinder saddle should then be marked from the barrel of the smoke box for chipping, using a pair of dividers, unless it has already been chipped, having been marked with a fixture. On large locomotives it is usually customary to mark the cylinder saddle from the boiler to be sure that the barrel of the smoke box will fit properly. Before removing the boiler the barrel of the smoke box should be scribed through the holes in the cylinder saddle in order that they may be drilled when the boiler is removed to chip the cylinder saddle.

After the cylinder saddle has been chipped and the cylinder saddle bolt holes in the smoke box drilled the boiler is again set in position and the holes for the saddle bolts are reamed and it is bolted up. At the same time the furnace bearers and waste ~~box~~ sheets are put up, marked, then taken down and drilled, then put up again, reamed, and bolted, thus securing the boiler in position.

The locomotive being thus far assembled, the guide bars and guide yokes are put in position and lined up. This is accomplished in the following way: The back cylinder heads are put in position and bolted up. A spider with tapered conical feet is placed in the front end of the cylinder and in the back cylinder head a conical bushing is then inserted. Through the center of the spider and the conical bushing a long bar is placed. The guide bar is then placed in position and accurate measurements made from both ends of the bar , or bars as the case may be, to the frames and also to the rod which corresponds to the piston rod. Liners are placed under the ends of the bars to bring them to exact position vertically and the cylinder head is turned slightly one way or the other and the guide bearer moved in order to locate the guide laterally on both ends. Having the bars thus properly located, the holes are reamed and they are bolted in position. Next the pistons are put in position and at the same time the crossheads. The front cylinder heads may then be bolted up. In the meantime the spring rigging should be put up, also the grates and ash pan and other accessories may be added at convenient times. The locomotive is then ready to be wheeled.

The driving boxes with their cellars packed and ready for use are assembled on the driving axles. The driving wheels and truck, or trucks as the case may be, are put in position on a track in proper relation to each other. The locomotive is then picked up with overhead cranes and carried to position over the running gears. It is then lowered into position, the driving spring saddles being adjusted properly in position in the driving boxes, and the truck fastenings are moved to position simultaneously.

The side rods and valve motion parts are now put up and adjusted after which the valves are set for operation. The boiler lagging and jacket are put on when convenient. Usually this is done before the boiler is put in position; also most of the boiler fittings and all the piping possible are put on previous to the assembly.

The brake rigging with all of the accessories is put on after the locomotive has been wheeled; also the frame binders, smoke stack, bell, head light, sand boxes, cab, etc.

The locomotive is now thoroughly inspected to make sure that every part is properly fitted up, that all bolts and nuts are tight and in place, and also to make certain that there are none of the parts missing. After all corrections have been made the locomotive is then ready for the road test. The locomotive is fired up and steam pressure got up so that all steam connections, valves, joints, and glands can be tested to make sure that they do not leak and that they work properly. The engine is then tried out on the road to make sure that there are no bearings that run hot and that all parts function properly. After this test any necessary corrections, alterations, and additions are made and the locomotive is a finished product.

SUMMARY

In the design and construction of a locomotive the following elements deserve most careful consideration:

1. Make a very careful study of the existing conditions relative to the operation.

2. Estimate as nearly as possible the size and type of the locomotive required.

3. Determine the number of wheels and their proper relation to each other and running gears.

4. Estimate the size and style of the boiler and of the accessories and appliances used in connection with it.

5. Calculate the probable weight of the locomotive and thus determine the center of equalization and the method of supporting this weight from the driving axles.

6. Design and lay out the various elements going to make up the engine such as valve motion, cylinders, and power transmitting mechanisms.

7. Make a careful study of the detailed design of every part to suit manufacturing.

8. Design and make flange blocks, forging dies, jigs, fixtures, tools, and gauges to be used in the manufacture of the locomotive parts.

9. Lay out and make templates and sheet metal patterns for the boiler shop work.

10. Make a complete set of patterns for all cast iron and cast steel parts.

11. Proceed with all boiler shop work, forgings, and castings.

12. Perform all machine operations and deliver parts to the assembly shop.

13. Assemble and test the locomotive.

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