Engineering Notes

INTRODUCTION

Perhaps no invention of man has had more impact on changing his way of life than has the STEAM LOCOMOTIVE. This marvelous wonder of mechanical motion opened new frontiers throughout the world, furnishing transportation not only for man but for the multitude of items needed to make our life more enjoyable, more plentiful, and to create vast metropolitan areas and industrial giants.

Now that the puffing, smoking mechanical wonder has passed into oblivion it is only natural that it should have a final resting place which is in the form of model live steam locomotives built to scale and operated as a true steam locomotive. With this in mind WINTON Engineering dedicates this catalog and engineering notes to the fallen, but still mighty in the minds of all, 'IRON HORSE'.

The selection of a locomotive to be modeled is simply what the individual desires in the way of the prototype. Since models are not required to handle any given load it is not necessary to choose a type for a particular service but only to select what appeals to him. However, the general mechanical function is the same for all types; hence certain engineering fundamentals are common to all.

Once the decision has been made on which type of locomotive is to be modeled and the cylinder diameter, stroke, drive wheel diameter and operating steam pressure are determined, one is ready to work out some of the basic mechanical details. It is of importance to state here that in some cases a part can be correctly designed from calculations but will fail to please the eye because it appears to be disproportionately small. It is necessary to use common sense in the design of all parts.

In the calculations to follow, the 1½-inch scale Mogul 2 - 6 - 0 locomotive will be used as an example.

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BASIC DESIGN ENGINEERING

LIST OF TABLES

1. HORSEPOWER

Indicated horsepower = IHP = $\frac{PLAN}{33,000}$

P = average steam pressure in the cylinder
L = stroke of the cylinders in feet
A = area of the piston in square inches
N = number of working strokes per minute

P = the average steam pressure in the cylinder is found by using table VI which gives the constant to multiply the boiler pressure by to get the average or mean effective pressure. As our gear cuts-off at 3/4 of the stroke we find that for this cut-off the constant is .966. 100 x .966 = 96.6 or 96 pounds per square inch is the average or mean effective pressure.

L = stroke in feet which from table II shows that a 2½-inch stroke is 0.2083 of a foot.

A = area of the piston as we have a 2-inch diameter cylinder table I shows the area to be 3.1416 square inches.

N = number of working strokes per minute. We will assume an RPM on the drivers of 200. As we have two cylinders and they are double-acting this gives us 4 power strokes per revolution of the drivers. At 200 RPM this is 4 x 200 = 800 working strokes per minute.

Now filling in our formula it looks like this:

$$IHP = \frac{96 \times .2083 \times 3.146 \times 800}{33,000} = 1.5$$

2. TRACTIVE EFFORT

The tractive effort is found from the following formula:

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

d = diameter in inches of the cylinder
P = average pressure in the cylinder (see preceeding example)
S = stroke in feet (see L in preceeding example)
D = diameter of driving wheel in feet (see table II)

To fill in our equation we will have to know the value of each function which is found as follows:

$$d^2 = 2 \times 2 = 4$$

$$P = 100 \times .96 = 96$$

$$S = \frac{2.5}{12} = .208$$

$$D = \frac{7}{12} = .5833$$

Therefore $T = \frac{4 \times 96 \times .208}{.5833} = 136$ pounds

This means that our model theoretically will have a pull of 136 pounds. However, due to friction, steam condensation and angularity of the main rod this valve will be reduced to approximately 110 pounds.
The factor of adhesion for locomotives operating on dry rail is approximately 1/5-1/6 the weight on the drivers. This means that if the weight on the drivers is 300 pounds the adhesive force will be 50-60 pounds.

3. CYLINDER, PORT SIZES AND STRENGTH OF BOLTS

The thickness of the cylinder wall to withstand a given internal pressure is determined by the formula:

$$Tcyl = (d \times Pb \times .0001) + .15\sqrt{d}$$

d = diameter of the cylinder in inches
Pb = boiler pressure

In our example d = 2
Pb = 100

Therefore $Tcyl = (2 \times 100 \times .0001) + .15\sqrt{2}$

= .02 + .21
= .23-in. the nearest even fraction is 1/4-inch so we will use this.

To find the square root of 2 we use table III to make the job easier.

BOLT SIZES

In model engineering work small hexhead brass cap screws are almost always used as their appearance adds so much to the overall beauty of the model. We have calculated the working strength of these small bolts and put them in table form for easy reference. Table IV is for the coarse thread series and Table V for the fine thread series. It will be noted that the fine thread is somewhat stronger than the coarse thread. This is due to the greater stress area as the depth of the thread is less. However, the fine thread tapped into iron, aluminum or other soft material is weaker than the coarse thread of equal diameter.

To figure the size and number of bolts we need for our cylinder we have, first of all, to decide on how many bolts we are going to use. For sake of appearance only I prefer not less than eight. More can be used if you desire. Now that we know the number of bolts we are going to use, we can easily determine the size. As our cylinder is of 2-in. bore we have a total area of 3.1416 square inches of area x 100 boiler pressure which totals 314 or call it 315 pounds push against the cylinder head. Now by dividing 315 by the number of bolts, 8, we have a load of 40 pounds per bolt. Looking at Table IV we find that a bolt which can carry a load of 40 pounds corresponds to a 5-40. Therefore 8, 5-40 bolts will do the job.

In formula style it appears:

$$\text{load per bolt} = \frac{A \times Pb}{Nb}$$

A = area in square inches of piston
Pb = boiler pressure
Nb = number of bolts to be used

AREAS OF STEAM PORTS AND PIPES

It is very important that the area of all steam ports and pipes leading to and from the cylinders be of ample size so the flow of steam is not restricted thereby causing a drop in pressure. In order to determine the area it is necessary to know what the maximum piston speed in feet per minute will be. Of course, piston speed is a function of the RPM and length of stroke. Since we know the stroke of our model it becomes necessary to to choose the RPM. 600 RPM is about the maximum you can run so we will use this value.

The formula for piston speed in feet per minute is:

$$PSFM = RPM \times 2L$$

RPM = revolutions per minute
L = length of stroke in feet (Table II gives the decimal values of feet from inches)
PSFM = 600 x 2 x .2083 = feet per minute

The area of the steam port in the cylinder is found from the formula:

$$Pa = \frac{PSFM \times .1}{600} \times A$$

PSFM = piston speed in feet per minute
A = area of the piston in square inches (Table I)

$$Pa = \frac{250 \times .1 \times 3.14}{600} = .13 \text{ square inches}$$

In model work the length of the port should not be less than .4 the cylinder diameter due to the difficulty in making small cores which will give good castings. Since our model has a 2-in. bore this gives us a port length of .4 x 2 = .8-in. Now we divide the area by the length to get the width. .13 ÷ .8 = .162. Making our port an even fraction we will have the port 7/8-in. x 3/16-in.

To determine the size of steam pipe to use leading from the boiler to the cylinders to properly feed the cylinders without undue loss of pressure we use the formula:

$$ASP = PSFM \times .00013 \times A$$

PSFM = piston speed in feet per minute
A = area of piston in square inches
= 250 x .00013 x 3.14 = .1 approximately

From Table I we find that for an area of .1 square inch the inside diameter would be 3/8-in.

4. PISTON ROD SIZE

DIAMETER OF PISTON ROD

The diameter of the piston rod is a function of the maximum steam pressure, area of the piston and strength of the material.

$$APR = \frac{A \times Pb}{S}$$ = cross section of area of the piston rod

A = area in square inches of the piston
Pb = boiler pressure
S = tensile strength of the material

$A = 2^2 \times .7854 = 3.1416$ (See Table I)
Pb = 100
S = 5000

$$\frac{3.1416 \times 100}{5,000} = .0628 \text{ square inches}$$

From Table I we find that an area of .0628 is close to 9/32. In our example we will choose a piston rod 5/16-in.diameter.

5. CROSSHEAD GUIDE

THRUST OF THE CROSSHEAD AGAINST THE GUIDES

To design the crosshead sliding areas we must know what load is placed upon them. This is found by taking the total pressure on the piston in pounds, length of the connecting rod and length of the crank throw, both in inches and putting them in the formula.

$$\text{Load} = L = \frac{Lc}{LcR} \times Tp$$

Lc = length of crank (½ stroke)
LcR = length of connecting rod
Tp = total load on the piston
(piston area x boiler pressure)
= Lc = 1.25 (our stroke is 2.5)
LcR = 11
Tp = 314 pounds ($2^2 \times .8754 \times 100$)

$$L = \frac{1.25}{11} \times 314 = 35\tfrac{1}{2} \text{ pounds load placed on the sliding surfaces}$$

In designing the sliding areas the load which is allowable per square inch is 50 pounds. Since our load is 35½ pounds we find the area needed by dividing 35½ by 50 which written in formula style is:

$$CA = \frac{35.5}{50} = .71 \text{ square inches}$$

The width of the crosshead guides is determined by mechanical clearances for the style of locomotive being built. In our case we use a guide 1/2-in. wide. With this one known dimension we find the length by taking the area CA and divide it by the width 1/2-in.

$$= \frac{.71}{.5} = 1.4\text{-in. long or in even inches}$$

we will make it 1½-in. long. This gives us a sliding area ½-in. x 1½-in.

6. DRIVING AXLE JOURNALS, DIAMETER AND LENGTH

The normal method of determining the size of driving axle journals in full scale practice cannot be used in model work as the ratio of weights of the full sized locomotive and model are not in the same ratio of the scale being used and, in addition, the steam pressure may be equal to the life size engine. The load imposed on the bearings by the steam pressure is higher than that due to the weight of the locomotive on the drivers. Remember the weight on the drivers is the weight of the locomotive minus the weight of the driver axle assemblies. Our Mogul weighs about 200 pounds which distributed over six bearings is only 34 pounds per bearing. The load due to the steam pressure is 314 pounds or 52 pounds per bearing.
Driving boxes with bronze bearings should be limited to a projected area loading of 500 pounds per square inch. The formula is the same as for side rods except the value 1,000 is reduced to 500.

$$Pa = \frac{Pb \times A}{500}$$

Pb = Boiler pressure
A = area of piston

$$Pa = \frac{100 \times 3.14}{500} = .628 \text{ square inches}$$

Length to diameter ratio is 1 to 1-1/8 Thus

$$\text{Length} = \frac{Pa \times 8}{9} = \frac{.628 \times 8}{9} = .749\text{-in.}$$

Again using standard size bearings we will use one .75 or 3/4-in. long. The diameter is 1-1/8 x the length so .75 x 1.125 = .843. Again using a standard size we go to 7/8-in. diameter. Therefore our main driver axle is 7/8-in. diameter x 3/4-in. long.

7. CRANK-PIN DIAMETERS AND LENGTH

The basic rule to follow is to allow 1000 pounds pressure per square inch of projected area. By projected area we mean the square inch of projected area. By projected area we mean the diameter multiplied by its length. For example a bearing 1-inch diameter and ½-inch long has a projected area of 1 x ½ = ½ in diameter and ½-inch long has a projected area of 1 x ½ = ½ square inch. To determine the projected area we take the boiler pressure in pounds per square inch and multiply it by the area of the piston and divide the product by 1000. Written as a formula it looks like this:

$$Pa = \frac{Pb \times A}{1000}$$

Pb = boiler pressure
A = area of the piston in square inches

$$Pa = \frac{100 \times 3.14}{1000} = .314 \text{ square inches of projected area}$$

A good ratio between the diameter of a bearing and its length is between 1:1 to 1¼:1. In our example we will use a ratio of 1-1/8 to 1 so our problem now looks like this:

$$\text{length} = \sqrt{\frac{Pa \times 8}{9}} = \sqrt{\frac{.314 \times 8}{9}} = .53$$

$$\text{diameter} = .53 \times 1.125 = .595\text{-in.}$$

In order to use a standard size bearing we chose one that is 5/8-in. diameter and 1/2-in. long. The value of 1000 is quite low as in actual practice the value varies from 1600 to 2500 pounds. The reason we use the lower value is to make the overall appearance more pleasing.
The same formula can be used for knuckle pins except the value 1000 is changed to 7000. The reason is that the pin has only a shear load to resist.

8. TENDER AND CAR JOURNAL SIZES

The pressure per square inch of projected area for tender and car journals can be taken between 350 and 500 pounds. Use the same formula as above to find the required projected area. Since we do not have a piston and steam pressure to give us the load we can substitute these values by assuming we will carry 6 adults at 150 pounds on a car. This gives us 900 pounds plus the weight of the car. Assuming this to be 100 we therefore have a total load of 1000 pounds to be carried on 8 journals. The length should be made 1¼ x the diameter.
The use of ball bearings are quite common in model work. These bearings are capable of carrying much heavier loads for the same axle diameter than conventional bronze bearings. In our example we calculated the bronze bearing to be 7/8-in. diameter to carry a load of 314 pounds now if we use a ball bearing on the same axle its load capacity would be 700 pounds.
The writers' opinion is that for model locomotive work bronze bearings of the oil-bearing type are preferred as they come in a great variety of sizes, are precision manufactured and low in cost. Oiling is simplified as they require no costly oil grooves. Under normal conditions they will not run dry and their replacement is simple as it only requires pushing out the old one and installing the new one. No boring is required. Grease packed ball bearings are preferred on locomotive axles as they never need any attention or replacement.

9. BOILER CONSTRUCTION

The design of model locomotive boilers is limited to specific outside dimensions to conform in scale to the general outline of the prototype. The calculations given are for determining the strength, grate area, tube area, stack size, exhaust nozzle area and general strength requirements.
The most common material used in model locomotive boilers is copper. In general this material has excellent corrosion resistance, heat transfer and fabrication qualities. However, its strength is rather low, especially at elevated temperatures. Joining the parts together is accomplished by silver solder, brazing, riviting or welding. With the proper hard solders joints can be made equal to the parent material strength. However, this requires joints to be fitted to a maximum of .003-in. clearance. Increased clearance in spaces between parts greatly reduces the joint strength. I feel that in figuring the strength of joints only 50% of the strength of the material should be used in calculations.
The average tensile strength of copper in the soft or annealed state is 30,000 pounds per square inch. Steel used in boilers has a tensile strength of 65,000 pounds per square inch.
Copper sheet can easily be shaped around forming blocks by pounding. During this working of the copper it work hardens and will crack. To eliminate this the part should be heated to a dull red heat and quenched in cold water. This process will soften the copper so it can be further worked without cracking. Steel should be worked while it is a dull red in heat. This makes the fabrication more difficult. However, the steel may be easily welded. All steel boilers that are welded should be stress relieved after final welding to eliminate highly stressed areas and possible future failure. Many model engineers are afraid of steel boilers because of rust and its consequent problems. However, with today's chemicals, any steel boiler will last a lifetime if properly taken care of.

The basic formula for determining the strength of a tube is--

$$\text{Bursting Pressure} = \frac{T_1 \times Tw}{D \times .5}$$

T_1 = tensile strength of the material
(if the tube has joint or seam use 50% of T_1)
Tw = thickness of the tube wall
D = outside diameter

To find the thickness of the material to use the formula--

$$T = \frac{P \times R}{C}$$

P = boiler pressure
R = ½ diameter of the boiler
C = strength of material of construction
(copper = 2,000, steel = 5,000---
these include strength of joint)

We will work out a typical example of a boiler with a 7" outside diameter and operating on 100 pounds pressure--

For copper $T = \frac{100 \times 3.5}{2{,}000} = .175"$

For steel $T = \frac{100 \times 3.5}{5{,}000} = .070"$

The one consideration in a boiler which is so often overlooked in designing is the flat areas. There are the sides of the firebox, backhead, front flue sheet and crown sheet. What happens is that the steam pressure tends to buldge out these areas and thus, rupture occurs. The way to overcome this is to use stays which hold together the flat surfaces. As each square inch of these areas are pushed on by the boiler pressure each square inch must be resisted by stays. Stays can be placed approximately 1" apart for materials of not less than 1/8 thick. Thicker sheets can have stays placed further apart if the builder wishes. 3/16" sheets may have stays 1½" apart. The stress in stays should be limited to maximum 4000 pounds per square inch. This means that if you have a stay supporting one square inch at a pressure of 100 pounds the area of the stay should be $\frac{100}{4000} = .025$ square inch.
This is about 3/16 diameter from Table I.

10. GRATE AREA

The grate area is based on information gathered over many years of operation of all classes of locomotives. The general rule is to take the tractive effort and divide it by 500. In our example the tractive effort was calculated at--

$$T = \frac{d^2 \times P \times S}{D} = \frac{4 \times 93.7 \times 2083}{.5833} = 133 \text{ pounds}$$

$$\text{Grate area} = \frac{T}{500} = \frac{133}{500} = .266 \text{ square feet}$$

$$.266 \times 144 = 38 \text{ square inches}$$

11. DIAMETER AND NUMBER OF TUBES

The diameter and number of tubes in model locomotives is not a direct scale as is the rest of the model. The reason being that the tubes would be so small that they would soot up almost immediately so we must be practical. A tube of 1/2" outside diameter is very common and works quite well.
The general ratio of grate area to total tube area which has worked quite well is 8:1. That is, for each 8 square inches of grate area we will have a combined tube area of 1 square inch. From example above our total tube area will be $\frac{38}{8} = 4\text{-}3/4$ square inches. Using a tube which has an outside diameter of 1/2" and wall thickness of .35 our inside diameter is approximately 7/16". From Table I the area of a circle 7/16" diameter is .15 square inches. Now by dividing the total area 4-3/4 square inches by .15 square inches we have $\frac{4.75}{.15} = 32$ tubes in our boiler.

12. WATER CONSUMPTION

In order to determine the water consumption and size of our water pump we must know how much water is to be used in a given time. When water is turned into steam it occupies a specific volume in relation to the original volume of water and the final pressure of the steam. Table VII has been compiled to give this information.
Continuing with our 1½ - Mogul which has a 2" bore and 2½" stroke, we find that the volume of steam used per one revolution of the drive wheel is--

$$Vs = A \times S \times N$$

A = area of piston in square inches
S = stroke of piston in inches
N = number of working strokes per one revolution of the drive wheel

A = 3.1416
S = 2.5
N = 4

Vs = 3.1416 x 2.5 x 4 = 31 cubic inches of steam per revolution. Note we have not taken into account that the steam is cut-off before the full stroke of the piston. The reason is that due to the many mechanical deficiencies in model locomotives the extra amount of steam will be used.
Looking at Table VII we find that at 100 pounds (gage) pressure one cubic inch of water evaporated into steam will give us 237 cubic inches of steam.
Therefore $\frac{31}{237}$ = .13 cubic inches of water per revolution of the drivers is required.

As this is only the theoretical volume needed and knowing that pumps do not operate at 100% volumetric efficiency we make them 50% greater in capacity to make up for losses due to leakage, waterslip and line restriction.

This means that the amount of water we must design a pump to handle is .13 x 1.5 = .2 cubic inch per revolution of the driver as our pump is connected to the crosshead. In our case the stroke of the pump is equal to that of the engine which is 2.5". Now by dividing the volume of water we need in cubic inches by the stroke of the pump we will have the area of the plunger in square inches

$$\text{Pump area} = \frac{Vw}{S} \quad \frac{.2}{2.5} = .08 \text{ square inches}$$

Vw = volume of water per stroke in cubic inches
S = stroke of plunger

From Table I we find that an area of .08 the corresponding diameter is 5/16". Our pump has a plunger 3/8" diameter.

13. STACK DIAMETER AND LENGTH

Practice indicates that the smallest internal diameter of the stack should not be less than 1/17 of the grate area. Expressed as a decimal 1/17 = .059. Our grate area is 38 square inches so to find the internal stack diameter we multiply 38 x .059 = 2.24 square inches. Table I shows that an area of 2.24 = 1-11/16" diameter. The length of the stack should be 4 diameters which is 4 x 11/16" = 6-3/4".

14. EXHAUST NOZZLE

In general the area of a single exhaust nozzle is 1/200 of the grate area. 1/200 = .005 as a decimal our grate area is 38 square inches so we multiply this by .005 which is .190 square inches. Again from Table I the area .190 - ½" diameter. It is recommended that several diameters of nozzles be tried so the most optimum results will be obtained.

15. POWER TO OPERATE PLAIN SLIDE-VALVES

The plain slide-valve is used in many model locomotives because of it's simple construction and ease of making it steam tight. However, it has one serious drawback and that is it takes a lot of power to operate it back and forth. This causes a lot of wear on parts such as the links, rocker, rocker pins and the eccentrics. The example below will illustrate the power required to make the plain slide valve move.
The resistance which must be overcome in moving any slide-valve is simply the friction between the valve and it's seat. This friction depends upon the pressure of the valve against the seat, and this pressure is equal to the total steam pressure upon the back of the valve, minus the reaction of the steam pressure in the steam and exhaust ports.
We shall take for our example a valve which is 1¼" x 1½" which has an area of 1.875 sq.in. using a boiler pressure of 125 lbs we have a force of 1.875 x 125 = 233 lbs. pushing against the valve. This is not the actual pressure of the valve against the seat as we have a back pressure due to the steam acting against the valve. Tests have shown that the total back pressure is about ¼ of the pressure on the live steam side. This means that the pressure pushing the valve away from the seat is $\frac{233}{4}$ = 58 Lbs.

Now to find the total pressure of the valve against the seat we take 233-58 = 175 lbs. One way of looking at it is that we have to push the valve back and forth with a 175 lbs. on it. Looks like a real job for the valve gear to do. Now the actual force the valve gear has to operate against is found by dividing the total load on the valve by the friction valve, which for smooth -iron surfaces well oiled is about 10:1, thus to find the resistance the valve gear has to overcome we divide 175 by 10 = 17.5 lbs. This 17.5 pound load has to be started, moved, stopped and reversed at every wheel revolution. This creates a real strain on the valve mechanism and for this reason railroads adapted balanced valves to reduce the maintenance in the valve gear assemblies. With piston valve a load of only a few ounces will do the same job.

16. LOCOMOTIVE HAULING POWER

Many people ask how much can a locomotive pull up grades of varying percentages. This can not be answered in so many pounds as there are many influencing factors such as locomotive tractive effort, rail conditions, rolling friction in the car journals and general locomotive performance. However, over a period of years railroads found average value for grades which when applied will give good results or at least some idea of what a given locomotive will do. The following table and example will illustrate why railroads keep all track as level as possible.

	GRADE IN %					
Level track	½%	1%	1½%	2%	2½%	3%
100%	44 %	26%	18 %	13%	10 %	8%

To illustrate the use of the above chart we will say a given locomotive will pull a load of 4,000 pounds. If we go up a 2% grade we can only pull 13% of 4000 or 520 pounds.

17. SIZE OF LOCOMOTIVE SPRINGS

In order for a locomotive to operate properly and to stay on the track it is necessary to have the locomotive fitted with equalizing levers and springs. The purpose of the equalizing levers is to distribute the weight equally on the driving axles, also to reduce the effects of shocks caused by the rails, and to allow the wheels to adjust themselves readily to any unevenness in the track without throwing an undue strain on the frames and other parts of the locomotive.
The formula below will give a good approximation of springs to use to give proper riding qualities to the locomotive.
First of all several things have to be known about the spring before we can work out the details. These things are the length, width, thickness and the weight the spring has to support. The length can be scaled from the prototype or some arbitrary length chosen. The width can also be scaled or taken as the frame width or less. Thickness is a matter of choice remembering that a thick spring leaf will make the locomotive ride stiff and a thin one will cause it to bounce. I feel that on 1½" locomotives of average size a spring of from .035 to .060 will give good results. Weight is something that is hard to come by but with a little calculations it can be approximated close enough to give good results. Now for the formula

$$\frac{\text{Load in tons x length of spring in inches x 11}}{\text{width in inches x (thickness of one leaf in sixteenth)}^2}$$

gives the number of leaves needed for each spring.

Working out a typical problem we have:

Total weight on drivers .... 400 lbs
Number of springs.. 6
Weight each spring has to support 400/6= 66 lbs
Length of spring.. 3.5"
Width of spring.. .375"
Thickness of one leaf.................................. .050"

number of leaves - $\frac{Lt \times l \times 11}{w \times t^2} - \frac{.033 \times 3.5 \times 11}{.375 \times .69} = 5$ leaves

Lt----66/2000-----.033
l----------------- 3.5
w----------------- .375
t^2--------------- .69 $(.050/.062)^2$

18. DEFLECTION OF LOCOMOTIVE SPRINGS

The amount a spring will deflect is calculated by the use of the formula below. This is understood to be only approximate but will serve as a guide as to how much to allow for in the set when in the free state.

$$\frac{\text{(lengths of spring in inches)}^3 \text{ x 1.5}}{\text{width in inches x (thickness in sixteenths)}^3 \text{ x number of leaves}} =$$

deflection in sixteenths per ton.

In our example we will fill in the known quantities

l = 3.5 inches
w = .375 inches
t = .050/.062 = .8
n = 5

We now have $\frac{3.5^3 \times 1.5}{.375 \times 8^3} = \frac{64.3}{.94} = 6.8$ sixteenths of an inch deflection per ton. 6.8 x .062 = .42"

Since our load is only 66 pounds this 66/2000 = .033 of one ton. To get the deflection we will take .42 x .033 which gives us .014" total deflection.

In order that the spring will remain straight under load we will make the ends of the spring 1/32" higher than the center.

19. ADHESION

The effort to haul a train which a locomotive can exert is limited by the adhesion between the driving wheels and the rail. This adhesion is simply friction between the driving wheels and the rails acting so as to prevent slipping. If, for instance, the train resistance exceeds the adhesion, the driving wheels will slip, or, in other words, turn around without advancing.
The adhesion depends upon the weight placed on the drivers. When the rails are dry and in comparatively good condition, we may assume that the adhesive force is equal to 1/5 of the weight on the drivers. Thus, for instance, if the weight on the drivers is 400 pounds, the adhesive force will be $\frac{400}{5} = 80$ pounds. This adhesive force enables an engine to pull a train, and must not be less than the train resistance.

20. CONCLUSION

We have omitted any mention of valve gears as it is a very deep study and many fine publications are available for those who wish to pursue the subject in detail.
It is our hope that the information contained herein will answer some of the questions which come up in the designing of live steam locomotives and enable the model engineer to better enjoy the world's finest hobby.

TABLES

TABLE I

Circumference and Areas of Circles

DIAMETER	CIRCUM-FERENCE	AREA	DIAMETER	CIRCUM-FERENCE	AREA
1/64	0.0491	0.0002	1 1/2	4.7124	1.7671
1/32	0.0982	0.0008	1 9/16	4.9087	1.9175
1/16	0.1964	0.0031	1 5/8	5.1051	2.0739
3/32	0.2945	0.0060	1 11/16	5.3014	2.2365
1/8	0.3927	0.0123	1 3/4	5.4978	2.4053
5/32	0.4909	0.0192	1 13/16	5.6941	2.5802
3/16	0.5890	0.0276	1 7/8	5.8905	2.7612
7/36	0.6872	0.0376	1 15/16	6.0868	2.9483
1/4	0.7854	0.0491	2	6.2832	3.1416
9/32	0.8836	0.0621	2 1/16	6.4795	3.3410
5/16	0.9817	0.0767	2 1/8	6.6759	3.5466
11/32	1.0799	0.0928	2 3/16	6.8722	3.7583
3/8	1.1781	0.1105	2 1/4	7.0686	3.9761
13/32	1.2763	0.1296	2 5/16	7.2649	4.2000
7/16	1.3745	0.1503	2 3/8	7.4613	4.4301
15/32	1.4726	0.1726	2 7/16	7.6576	4.6664
1/2	1.5708	0.1964	2 1/2	7.8540	4.9087
17/32	1.6690	0.2217	2 9/16	8.0503	5.1572
9/16	1.7672	0.2485	2 5/8	8.2467	5.5119
19/32	1.8653	0.2769	2 11/16	8.4430	5.6727
5/8	1.9635	0.3068	2 3/4	8.6394	5.9496
21/32	2.0617	0.3382	2 13/16	8.8357	6.2126
11/16	2.1598	0.3712	2 7/8	9.0321	6.4918
23/32	2.2580	0.4057	2 15/16	9.2284	6.7771
3/4	2.3562	0.4418	3	9.4248	7.0686
25/32	2.4544	0.4794	3 1/8	9.8175	7.6699
13/16	2.5525	0.5185	3 1/4	10.2102	8.2958
27/32	2.6507	0.5595	3 3/8	10.6029	8.9462
7/8	2.7489	0.6013	3 1/2	10.9956	9.7211
29/32	2.8471	0.6450	3 5/8	11.3883	10.321
15/16	2.9452	0.6903	3 3/4	11.7810	11.945
31/32	3.0434	0.7371	3 7/8	12.1737	11.793
1	3.1416	0.7854	4	12.5664	12.566
1 1/16	3.3379	0.8866	4 1/8	12.9591	13.364
1 1/8	3.5343	0.9940	4 1/4	13.3518	14.186
1 3/16	3.7306	1.1075	4 3/8	13.7445	15.033
1 1/4	3.9270	1.2272	4 1/2	14.1372	15.904
1 5/16	4.1233	1.3530	4 5/8	14.5299	16.800
1 3/8	4.3197	1.4849	4 3/4	14.9226	17.721
1 7/16	4.5160	1.6230	4 7/8	15.3153	18.665
			5	15.7080	19.635

DIAMETER	CIRCUM-FERENCE	AREA	DIAMETER	CIRCUM-FERENCE	AREA
5 1/4	16.9934	21.648	8	25.1327	50.265
5 1/2	17.2788	23.758	8 1/4	25.9181	53.456
5 3/4	18.0642	25.967	8 1/2	26.7035	56.745
6	18.8496	28.274	8 3/4	27.4889	60.132
6 1/4	19.6340	30.680	9	28.2743	63.617
6 1/2	20.4204	33.183	9 1/4	29.0597	67.201
7	21.9911	38.485	9 1/2	29.8451	70.882
7 1/4	22.7765	41.282	9 3/4	30.6305	74.662
7 1/2	23.5619	44.179	10	31.4159	78.540
7 3/4	24.3473	47.173			

TABLE II

Inches to Decimals of a Foot

INCHES	FOOT	INCHES	FOOT	INCHES	FOOT
1/4	0.0208	2 3/4	0.2292	6 1/4	0.5208
5/16	0.0260	2 7/8	0.2396	6 1/2	0.5417
3/8	0.0312	3	0.2500	6 3/4	0.5625
7/16	0.0365	3 1/8	0.2604	7	0.5833
1/2	0.0417	3 1/4	0.2708	7 1/4	0.6042
9/16	0.0469	3 3/8	0.2813	7 1/2	0.6250
5/8	0.0521	3 1/2	0.2917	7 3/4	0.6406
11/16	0.0573	3 5/8	0.3021	8	0.6666
3/4	0.0625	3 7/8	0.3229	8 1/4	0.6875
13/16	0.0678	4	0.3333	8 1/2	0.7083
7/8	0.0730	4 1/8	0.3437	8 3/4	0.7292
15/16	0.0781	4 1/4	0.3542	9	0.7500
1	0.0833	4 3/8	0.3654	9 1/4	0.7708
1 1/8	0.0937	4 1/2	0.3750	9 1/2	0.7917
1 1/4	0.1042	4 5/8	0.3854	9 3/4	0.8125
1 3/8	0.1146	4 3/4	0.3958	10	0.8333
1 1/2	0.1250	4 7/8	0.4062	10 1/4	0.8542
1 5/8	0.1354	5	0.4167	10 1/2	0.8750
1 3/4	0.1458	5 1/8	0.4271	10 3/4	0.8958
1 7/8	0.1562	5 1/4	0.4375	11	0.9167
2	0.1666	5 3/8	0.4479	11 1/4	0.9375
2 1/8	0.1771	5 1/2	0.4583	11 1/2	0.9583
2 1/4	0.1875	5 5/8	0.4687	11 3/4	0.9792
2 3/8	0.1979	5 3/4	0.4792	12	1.0000
2 1/2	0.2083	5 7/8	0.4896		
2 5/8	0.2187	6	0.5000		

TABLE III

Inches to Square Root to Square

INCHES	SQ. ROOT	SQUARE	INCHES	SQ. ROOT	SQUARE
1.000	1.00000	1.00000	6.375	2.52488	40.64063
1.125	1.06066	1.26563	6.500	2.54951	42.25000
1.250	1.11803	1.56250	6.625	2.57391	43.89063
1.375	1.17260	1.89063	6.750	2.59808	45.56250
1.500	1.22474	2.25000	6.875	2.62202	47.26563
1.625	1.27475	2.64063	7.000	2.64575	49.00000
1.750	1.32288	3.06250	7.125	2.66927	50.76563
1.875	1.36931	3.51563	7.250	2.69258	52.56250
2.000	1.41421	4.00000	7.375	2.71570	54.39063
2.125	1.45774	4.51563	7.500	2.73861	56.25000
2.250	1.50000	5.06250	7.625	2.76134	58.14063
2.375	1.54110	5.64063	7.750	2.78388	60.06250
2.500	1.58114	6.25000	7.875	2.80624	62.01563
2.625	1.62019	6.89063	8.000	2.82843	64.00000
2.750	1.65831	7.56250	8.125	2.85044	66.01563
2.875	1.69558	8.26563	8.250	2.87228	68.06250
3.000	1.73205	9.00000	8.375	2.89396	70.14063
3.125	1.76777	9.76563	8.500	2.91548	72.25000
3.250	1.80278	10.56250	8.625	2.93684	74.39063
3.375	1.83712	11.39063	8.750	2.95804	76.56250
3.500	1.87083	12.25000	8.875	2.97909	78.76563
3.625	1.90394	13.14063	9.000	3.00000	81.00000
3.750	1.93649	14.06250	9.125	3.02076	83.26563
3.875	1.96850	15.01563	9.250	3.04138	85.56250
4.000	2.00000	16.00000	9.375	3.06186	87.89063
4.125	2.03101	17.01563	9.500	3.08221	90.25000
4.250	2.06155	18.06250	9.625	3.10242	92.64063
4.375	2.09165	19.14063	9.750	3.12250	95.06250
4.500	2.12132	20.25000	9.875	3.14245	97.51563
4.625	2.15058	21.39063	10.000	3.16228	100.00000
4.750	2.17945	22.56250	10.125	3.18198	102.51563
4.875	2.20794	23.76563	10.250	3.20156	105.06250
5.000	2.23607	25.00000	10.375	3.22102	107.64063
5.125	2.26385	26.26563	10.500	3.24037	110.25000
5.250	2.29129	27.56250	10.625	3.25960	112.89063
5.375	2.31840	28.89063	10.750	3.27872	115.56250
5.500	2.34521	30.25000	10.875	3.29773	118.26563
5.625	2.37171	31.64063	11.000	3.31662	121.00000
5.750	2.39792	33.06250	11.125	3.33542	123.76563
5.875	2.42384	34.51563	11.250	3.35410	126.56250
6.000	2.44949	36.00000	11.375	3.37268	129.39063
6.125	2.47487	37.51563	11.500	3.39117	132.25000
6.250	2.50000	39.06250	11.625	3.40955	135.14063

INCHES	SQ. ROOT	SQUARE	INCHES	SQ. ROOT	SQUARE
11.750	3.42783	138.06250	15.000	3.87298	225.00000
11.875	3.44601	141.01563	15.125	3.88909	228.76563
12.000	3.46410	144.00000	15.250	3.90512	232.56250
12.125	3.48210	147.01563	15.375	3.92110	236.39063
12.250	3.50000	150.06250	15.500	3.93700	240.25000
12.375	3.51781	153.14063	15.625	3.95285	244.14063
12.500	3.53553	156.25000	15.750	3.96863	248.06250
12.625	3.55317	159.39063	15.875	3.98434	252.01563
12.750	3.57071	162.56250	16.000	4.00000	256.00000
12.875	3.58818	165.76563	16.125	4.01559	260.01563
13.000	3.60555	169.00000	16.250	4.03113	264.06250
13.125	3.62284	172.26563	16.375	4.04660	268.14063
13.250	3.64005	175.56250	16.500	4.06202	272.25000
13.375	3.65718	178.89063	16.625	4.07738	276.39063
13.500	3.67423	182.25000	16.750	4.09268	280.56250
13.625	3.69121	185.64063	16.875	4.10792	284.76563
13.750	3.70810	189.06250	17.000	4.12311	289.00000
13.875	3.72492	192.51563	17.125	4.13824	293.26563
14.000	3.74166	196.00000	17.250	4.15331	297.56250
14.125	3.75832	199.51563	17.375	4.16833	301.89063
14.250	3.77492	203.06250	17.500	4.18330	306.25000
14.375	3.79144	206.64063	17.625	4.19821	310.64063
14.500	3.80789	210.25000	17.750	4.21307	315.06250
14.625	3.82426	213.89063	17.875	4.22788	319.51563
14.750	3.84057	217.56250	18.000	4.24264	324.00000
14.875	3.85681	221.26563			

TABLE IV

Strength of Model Engineers Hex Head Bolts made of Brass

-- Course Thread --

SIZE	O.D.	STRESS AREA	SAFE LOAD LBS.	TAP DRILL SIZE	CLEARANCE DRILL SIZE
1-64	0.073	0.0026	13	53	5/64
2-56	0.086	0.0036	18	51	3/32
3-48	0.099	0.0048	24	5/64	7/64
4-40	0.112	0.0060	30	43	1/8
5-40	0.125	0.0079	40	39	9/64
6-32	0.138	0.0090	45	36	23
8-32	0.164	0.0139	70	29	15
10-24	0.190	0.0174	87	25	5
1/4-20	0.250	0.0317	150	8	17/64
5/16-18	0.3125	0.0522	261	F	21/64

Continued on next page

TABLE IV (continued)

SIZE	O.D.	STRESS AREA	SAFE LOAD LBS.	TAP DRILL SIZE	CLEARANCE DRILL SIZE
3/8-16	0.375	0.0773	387	5/16	25/64
7/16-14	0.4375	0.1060	530	U	29/64
1/2-13	0.5000	0.1416	708	27/64	33/64
9/16-12	0.5625	0.1816	908	31/64	37/64
5/8-11	0.6250	0.2256	1128	17/32	41/64
3/4-10	0.7500	0.3340	1670	21/32	49/64
7/8-9	0.875	0.4612	2306	49/64	57/64
1-8	1.000	0.6051	3026	7/8	1-1/32

TABLE V

Strength of Model Engineers Hex Head Bolts made of Brass
-- Fine Thread --

SIZE	O.D.	STRESS AREA	SAFE LOAD LBS.	TAP DRILL SIZE	CLEARANCE DRILL SIZE
0-80	0.0600	0.0018	9	3/64	51
1-72	0.0730	0.0027	13	53	5/64
2-64	0.0860	0.0039	19	50	3/32
3-56	0.0990	0.0052	26	46	7/64
4-48	0.1120	0.0065	32	42	1/8
5-44	0.1250	0.0082	41	37	9/64
6-40	0.1380	0.0101	50	33	23
8-36	0.1640	0.0146	73	29	15
10-32	0.1900	0.0199	99	21	5
1/4-28	0.2500	0.0362	181	3	17/64
5/16-24	0.3125	0.0579	289	I	21/64
3/8-24	0.3750	0.0876	438	Q	25/64
7/16-20	0.4375	0.1185	592	W	29/64
1/2-20	0.5000	0.1597	798	29/64	33/64
9/16-18	0.5625	0.2026	1013	33/64	37/64
5/8-18	0.6250	0.2555	1277	37/64	41/64
3/4-16	0.7500	0.3724	1862	11/16	49/64
7/8-14	0.8750	0.5088	2544	13/16	57/64
1-14	1.0000	0.6624	3312	15/16	1-1/32
1/4-32	0.2500	0.0377	188	7/32	17/64
5/16-32	0.3125	0.0622	311	9/32	21/64
3/8-32	0.3750	0.0929	464	11/32	25/64
7/16-28	0.4375	0.1270	635	13/32	29/64
1/2-28	0.5000	0.1695	847	15/32	33/64

TABLE VI

Mean Effective Pressure Constants

1/4	stroke equals boiler pressure X	.597
1/3	" " " " "	.670
3/8	" " " " "	.743
1/2	" " " " "	.847
5/8	" " " " "	.919
2/3	" " " " "	.937
3/4	" " " " "	.966
7/8	" " " " "	.992

TABLE VII

Volumes of Saturated Steam

Gauge Pressure	Cubic inches of steam per cubic inch of water	Temperature F
50	406	298
60	355	308
70	316	316
75	299	320
80	285	324
85	271	328
90	258	332
95	247	335
100	237	338
110	219	345
120	204	350
125	197	353
135	184	358
150	169	366
175	150	378
200	131	388
225	117	397
250	107	406

TABLE VIII

Strength of Materials

Material	Tensile strength in pounds per sq. in.
Aluminum	
cast	15,000
cast high strength	26,000
bar stock 2011-T3 excellent machining	54,000
bar stock 2011-T8 excellent machining	59,000
bar stock 2024-T4 good machining	68,000
structural shapes 6061-T6 excellent for car frames	45,000
Copper	
sheet hard	46,000
sheet soft	33,000
rod hard	45,000
rod soft	32,000
Brass	
bar stock free cutting	58,000
bar stock high lead	73,000
Bronze	
#1012 Everdur	95,000
Tobin bronze	63,000
Steel	
C1018 cold finished bar	82,000
Stressproof - excellent machining, fine finish	125,000
Stainless	
Type 303 free machining	75,000
Type 321 fair machining	75,000
Type 416 good machining - annealed condition	75,000
Type 416 poor machining in heat treated condition	90,000-200,000
Iron	
cast grey	18,000
cast malleable	28,000

TABLE IX

O-Rings used as Valve Seats

O-rings are particularly suited for use as valve seats. They absorb shock loads, and are soft enough to seal at all pressures, even when dirt and grit are present in the system. They are ideal for check valves where the fluid pressure helps to make the seal. High-pressure check valves can maintain 20,000 psi for weeks. Properly applied, they can be used on relief and angle-valve seats for all pressures.

One of the design problems with O-ring valve seats is to prevent the ring from blowing out of the groove. This will happen with a square or rectangular groove, if a high-pressure differential exists across the valve seat at the moment of opening, Fig. 1.

In most cases, blowout occurs if the differential pressure is more than 100 psi. Since blowout is similar to extrusion, it helps to use harder O-ring compounds that can withstand higher pressures before elongating. One way of preventing blowout is by use of a dovetail groove design, Figure 3. Other methods of preventing blowout are to mechanically spin metal around the O-ring and secure it in the groove, or, vulcanize and bond the synthetic rubber into the valve-seat groove. By venting the groove, pressure cannot build up underneath the O-ring, and it remains in its seat, Fig. 2.

Fig. 1

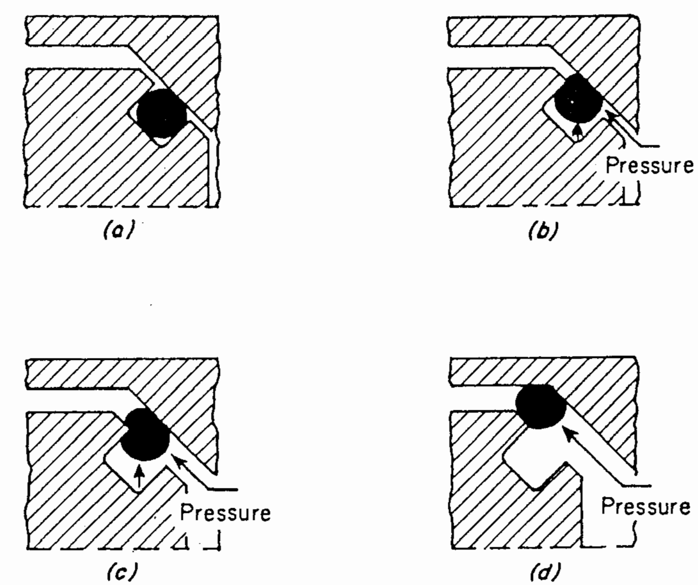


Fig. 1 -- Blowout of O-ring used as a valve seat. As the valve opens, the space between the two faces becomes larger. The pressure acts on the O-ring. The ring continues to seal the opening until it is completely stretched out of the groove.

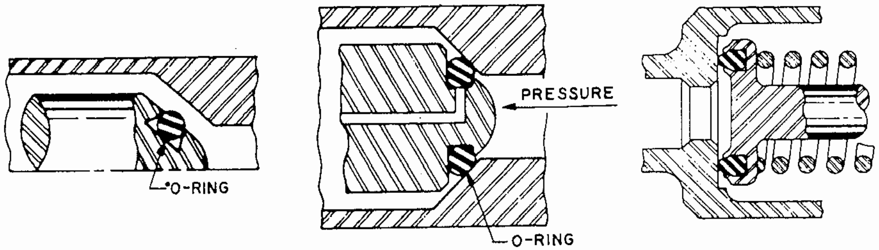


Fig. 2. Groove Designs to Prevent Blow-Out.

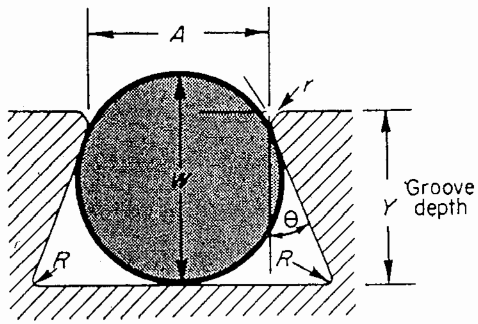


Fig. 3. Standard Dovetail Groove Sizes for O-Ring Seals

Standard Size and Installation Data

O-Ring Size No.	O-Ring W		A Groove Length Sharp Edge tolerance ±0.002	A Groove Length Rounded tolerance ±0.002	Y Groove Depth ±0.000-0.002	Radius r	Radius R
004 thru 028	0.070	0.003	0.057	0.063	0.052	0.005	1/64
110 thru 149	0.103	0.003	0.085	0.090	0.083	0.010	1/64
210 thru 274	0.139	0.004	0.115	0.120	0.115	0.010	1/32
325 thru 349	0.210	0.005	0.160	0.170	0.180	0.015	1/32
425 thru 460	0.275	0.006	0.220	0.235	0.234	0.015	1/16

$\theta = 24° \pm 1°$

First cut groove, leaving sharp edge at corners, then round off to *A* dimension.

TABLE X

Pattern Shrinkage Allowance

Cast iron	1/8	inch per foot
Brass	3/16	" " "
Steel	1/4	" " "
Aluminum	3/16	" " "

TABLE XI

Standard Keyways and Setscrews

Diameter of Hole	St'd Keyway W	St'd Keyway d	Recommended Setscrew
5/16 to 7/16"	3/32"	3/64"	10-32
1/2 to 9/16	1/8	1/16	1/4-20
5/8 to 7/8	3/16	3/32	5/16-18
15/16 to 1 1/4	1/4	1/8	3/8-16
1 5/16 to 1 3/8	5/16	5/32	7/16-14
1 7/16 to 1 3/4	3/8	3/16	1/2-13
1 13/16 to 2 1/4	1/2	1/4	9/16-12
2 5/16 to 2 3/4	5/8	5/16	5/8-11
2 13/16 to 3 1/4	3/4	3/8	3/4-10
3 5/16 to 3 3/4	7/8	7/16	7/8-9
3 13/16 to 4 1/2	1	1/2	1-8
4 19/16 to 5 1/2	1 1/4	7/16	1 1/8-7
5 9/16 to 6 1/2	1 1/2	1/2	1 1/4-6

TABLE XII

Table giving Proportionate Weight of Castings to Weight of Wood Patterns

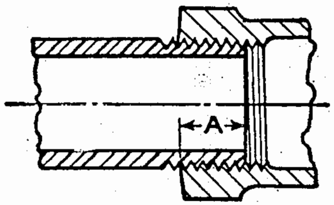
A Pattern Weighing One Pound (Less Weight of Core Prints)

Made of	Cast Iron	Brass	Copper	Bronze	Bell Metal	Zinc
Pine or Fir	16	15.8	16.7	16.3	17.1	13.5
Oak	9	10.1	10.4	10.3	10.9	8.6
Beech	9.7	10.9	11.4	11.3	11.9	9.1
Linden	13.4	15.1	16.7	15.5	16.3	12.9
Pear	10.2	11.5	11.9	11.8	12.4	9.8
Birch	10.6	11.9	12.3	12.2	12.9	10.2
Alder	12.8	14.3	14.9	14.7	15.5	12.2
Mahogany	11.7	13.2	13.7	13.5	14.2	11.2
Brass	0.85	0.95	0.99	0.98	1.0	0.81

TABLE XIII

Length of Thread

LENGTH OF THREAD ON PIPE SCREWED INTO VALVES OR FITTINGS TO MAKE A TIGHT JOINT



Size Inches	Dimen. A Inches	Size Inches	Dimen. A Inches
1/8	1/4	2 1/2	15/16
1/4	3/8	3	1
3/8	3/8	3 1/2	1 1/16
1/2	1/2	4	1 1/8
3/4	9/16	5	1 1/4
1	11/16	6	1 5/16
1 1/4	11/16	8	1 7/16
1 1/2	11/16	10	1 5/8
2	3/4	12	1 3/4

Dimensions given do not allow for variation in tapping or threading

TABLE XIV

Approximate Weight of Various Metals

To find the weight of various metals, multiply the contents in cubic inches by the number shown below. The result will be the approximate weight in pounds.

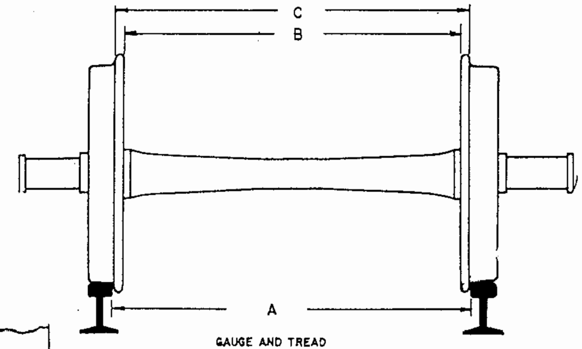
Iron	.27777	Brass	.3112
Steel	.28332	Lead	.41015
Copper	.32118	Zinc	.25318
Tin	.26562	Aluminum	.09375

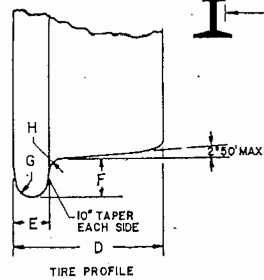
LIVE STEAM MODEL LOCOMOTIVE STANDARDS

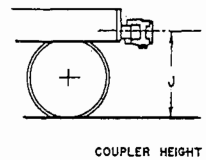
TABLE A

Wheel and Gauge Standards

¾ - 1 - 1½





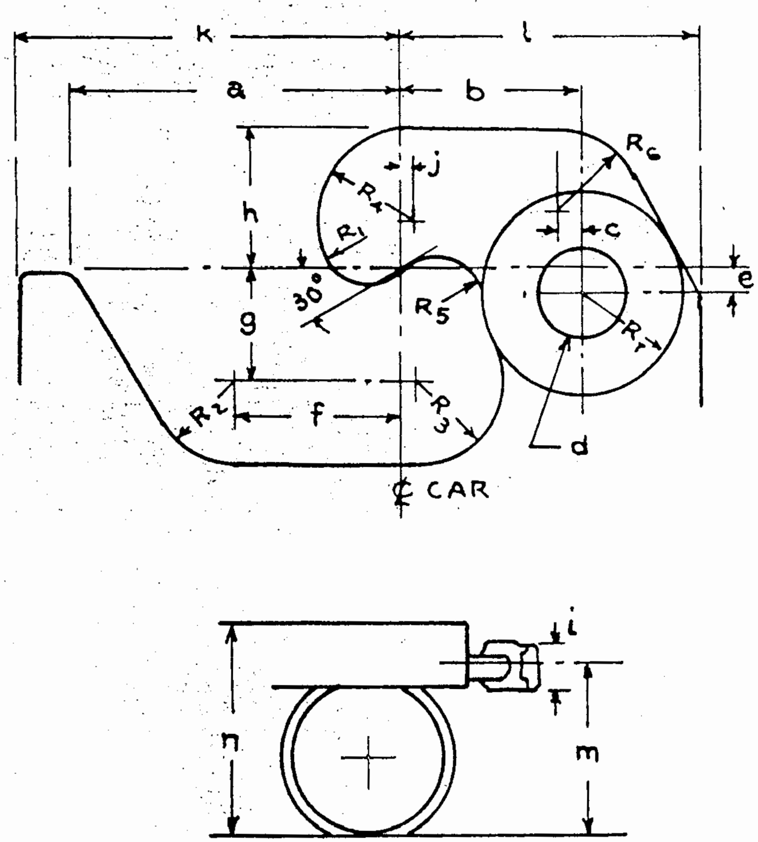


	SCALE	1-1/2"	1"	3/4"
A	Track gauge: Tangent	7 1/2+1/16-0	4 3/4+1/16-0	3 1/2+1/16-0
	Curves	7 9/16+1/16-0	4 13/16+1/16-0	3 17/32+1/16-0
B	Back to back of flanges	7 1/8+1/64-0	4 1/16+1/64-0	3 9/32+1/64-0
C	Wheel gauge ---- max.	7 7/16+0-1/64	4 11/16+0-1/64	-15/.2+0-1/64
D	Width of tire -- min.	3/4	9/16	7/16
E	Flange thickness- max.	.156	.125	.094
F	Flange depth --- max.	.187	.156	.094
G	Flange radius File to contour			
H	Gauging radius-- max.	1/16	3/64	1/32
J	Coupler height	4-7/16	2-29/32	2-7/32

TABLE B

Automatic Coupler Standards

REVISED 4-13-63
GOLDEN GATE LIVE STEAMERS, INC.



SCALE	3/4	1	1 – 1/2	3
a	27/64	9/16	27/32	1 11/16
b	15/64	5/16	15/32	15/16
c	1/32	3/64	1/16	1/8
d	7/64	9/64	7/32	7/16
e	1/32	3/64	1/16	1/8
f	7/32	19/64	7/16	7/8
g	9/64	3/16	9/32	9/16
h	3/16	1/4	3/8	3/4
i	11/16	29/32	1 3/8	2 3/4
j	1/64	1/64	1/32	1/16
k	1/2	43/64	1	2
l	25/64	17/32	25/32	1 9/16
m	2 5/32	2 7/8	4 5/16	8 5/8
n	2 3/4	3 11/16	5 1/2	11
R_1	1/16	5/64	1/8	1/4
R_2	7/64	9/64	7/32	7/16
R_3	7/64	9/64	7/32	7/16
R_4	1/8	5/32	1/4	1/2
R_5	1/16	5/64	1/8	1/4
R_6	7/64	9/64	7/32	7/16
R_7	9/64	11/64	17/64	17/32

From a safety standpoint the most important dimensions are the 30° angle, R_1 and R_5 designed to give a good "hook" to the knuckle.