Winton Brown's Engineering Data

An IBLS
Wandering
Locomotive
Book

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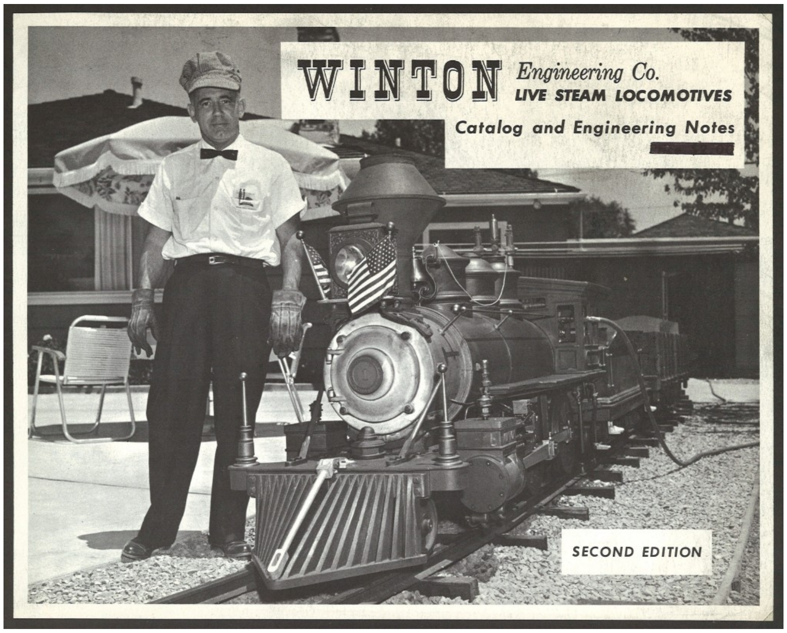
PREFACE

Winton Brown founded the Winton Locomotive Works Company in Danville, California in the early 1960's. Winton was a mechanical engineer by trade. He brought as set of finely honed engineering skills to the company, designing his locomotives for maximum strength, maximum weight on the drivers and easy, modular construction.



The Winton catalog included a section simply entitled "Engineering Notes". Behind that humble title lay 17 pages of notes, drawings, formulas and tables essential for designing and building model steam locomotives. In the 1960's, if you were building a loco from scratch which wasn't adaptable from a Little Engines design the Winton catalog engineering section was considered the bible and many used it to design parts of their locos, especially boilers.

Winton's engineering data is just as valuable today. It offers, among other things, tips on improving the performance of practically any steam locomotive. An additional article, written by Winton which originally appeared in the Nov/Dec 1954 edition of "The Model Locomotive", is included in this publication. It points out the importance of superheated steam and works through an example to compute the ratio of water to steam to maintain maximum operational capacity.

Kenneth Shattock, IBLS secretary of the North American Region, encouraged the creation of this document. He realized the importance of preserving this important asset for the live steam community, and increasing its general availability.

The data has been checked against modern standards and updated as needed. Additional helpful tables have also been included. Other modifications have been made to aid the reader in understanding the concepts described by Winton.

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.

HORSEPOWER

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

where

IHP = Indicated horsepower
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 ¾ of the stroke we find that for this cut-off the constant is 0.966. 100 x 0.966 = 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:

$$IPH = \frac{96 * 0.2083 * 3.1416 * 800}{33{,}000} = 1.5$$

TRACTIVE EFFORT

The tractive effort is found from the following formula:

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

where

d = diameter in inches of the cylinder
P = average pressure in the cylinder (see preceding example)
S = stroke in feet (see L in preceding 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:

$$d^2 = 2 * 2 = 4$$

$$P = 100 * 0.96 = 96$$

$$S = \frac{2.5}{12} = 0.208$$

$$D = \frac{7}{12} = 0.5833$$

Therefore

$$T = \frac{4 * 96 * 0.208}{0.5833} = 137 \ \textit{pounds}$$

This means that our model theoretically will have a pull of 137 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 to 1/6 the weight of the drivers. This means that if the weight on the drives is 300 pounds the adhesive force will be 50 to 60 pounds.

CYLINDER, PORT SIZES AND STRENGTH OF BOLTS

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

$$Tcyl=(d*Pb*0.0001)+0.15*\sqrt{d}$$

where

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

In our example

d = 2
PB = 100

Therefore

$$Tcyl=(2*100*0.0001)+0.15*\sqrt{2}$$

= 0.02 + 0.21
= 0.23-inches

The nearest even fraction is ¼-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-inch 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 eight 5-40 bolts will do the job.

In formula style in appears:

$$Lb=\frac{A*Pb}{Nb}$$

where

Lb = load per bolt (psi)
Pb = boiler pressure (psi)
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 feed 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 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:

$$PSFM=RPM*2*L$$

where

PSFM = piston speed in feet per minute
RPM = revolutions per minute
L = length of stroke in feet (Table II gives the decimal values of feet from inches)

Example:

PFSM = 600 * 2 * 0.2083 = piston speed in feet per minute

$$Pa=\frac{PSFM*0.1}{600}*A$$

where

A = area of the piston in square inches (Table I)

Example:

$$Pa=\frac{250*0.1*3.14}{600}*0.13\,square\,inches$$

In model work the length of the port should not be less than 0.4 the cylinder diameter due ot the difficulty in making small cores which will give good castings. Since our model has a 2-inch bore this gives us a port length of 0.4 x 2 = 0.8-inches. Now we divide the area by the length to get the width. 0.13 / 0.8 = 0.162. Making our port an even fraction we will have the port 7/8-inch x 3/16-inch.

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:

$$ASP=PSFM*0.00013*A$$

Example:

250 * 0.00013 * 3.14 = 0.1 approximately

From Table I we find that for an area of 0.1 square inch the inside diameter would be 3/8-inch.

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.

$$APR=\frac{A*Pb}{S}$$

where

APR = 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

Example:

A = 2 * 2 * 0.7854 = 3.1416 (See Table I)
Pb = 100
S = 5000

$$\frac{3.1416*100}{5,000}=0.0628\,square\,inches$$

From Table I we find that an area of 0.0628 is close to 9/32. In our example we will choose a piston rod 5/16-inches in diameter.

CROSSHEAD GUIDE

THRUST OF THE CROSSHEAD AGAINST THE GUIDES

The 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.

$$L=\frac{Lc}{LcR}*Tp$$

where

L = load placed on the sliding surfaces
Lc = length of crank (½ stroke)
LcR = length of connecting rod
Tp = total load on the piston (piston area * boiler pressure)

Example:

Lc = 1.25 (our stroke is 2.5)
LcR = 11
Tp = 314 pounds (2 * 2 * 0.8754 * 100)

$$L=\frac{1.25}{11}*314=35.5$$

or 35-½ 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:

$$CA=\frac{35.5}{50}=0.71\,square\,inches$$

where

CA = crosshead area

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 ½-inch wide. With this one known dimension we find the length by taking the area CA and dividing it by the width ½-inch, as follows:

$$CA=\frac{0.71}{0.5}=1.4\,inches\,long$$

We will make it 1½-inches long. This gives us a sliding area ½-inch by 1½-inches.

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 diameter multiplied by its length. For example a bearing 1-inch in diameter and ½-inch long has a projected area of 1 * 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:

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

where

Pa = projected area in square inches

Pb = boiler pressure in pounds per square inches

A = area of the piston in square inches

$$Pa=\frac{100*3.14}{1000}=0.314\,square\,inches$$

A good ratio between the diameter of a bearing and its length is between 1:1 to 1.5:1. In our example we will use a ratio of 1-1/8 to 1 so our problem now looks like this:

$$length=\frac{\sqrt{Pa*8}}{9}=\frac{\sqrt{0.314*8}}{9}=0.53$$

$$diameter=0.53*1.125=0.595\,inches$$

In order to use a standard size bearing we chose one that is 5/8-inch diameter and ½-inch long. The value of 1000 psi is quite low as in actual practice the value varies from 1600 to 2500 psi. 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 psi is increased to 7000 psi. The reason is that the pin has only a shear load to resist.

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 crank-pins, above, except the value 1000 psi is reduced to 500 psi.

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

where

Pa = projected area in square inches

Pb = Boiler pressure in psi

A = area of piston in square inches

$$Pa=\frac{100*3.14}{500}=0.628\,square\,inches$$

Length to diameter ratio is 1 to 1-1/8, therefore

$$Length=\frac{Pa*8}{9}=\frac{0.628*8}{9}=0.749\,inches$$

Again using standard size bearings we will use on 0.75 or ¾-inches long. The diameter is 1-1/8 by the length, so 0.75 * 1.125 = 0.843. Again using a standard size we go to 7/8-inch diameter. Therefore our main driver axle is 7/8-inch diameter by 3/4-inch long.

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 100. We therefore have a total load of 1000 pounds to be carried on 8 journals. The length should be made 1¼ times 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-inches 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 writer's 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.

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, riveting 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 0.003-inches 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:

$$Bursting\ Pressure = \frac{T1 * Tw}{D * 0.5}$$

where

T1 = tensile strength of the material (if the tube has joint or seam use 50% of T1)
Tw = thickness of the tube wall
D = outside diameter

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

For Copper:

$$T = \frac{100 * 3.5}{2{,}000} = 0.175\ inch$$

For steel:

$$T = \frac{100 * 3.5}{5{,}000} = 0.070\ inch$$

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 inch apart for materials of not less than 1/8 inch thick. Thicker sheets can have stays placed further apart if the stress in stays should be limited to maximum 4,000 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

$$\frac{100}{4{,}000} = 0.025\ square\ inch$$

This is about 3/16 inch diameter from Table I.

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:

$$T=\frac{d^2 * P * S}{D}=\frac{4 * 93.7 * 2083}{0.5833}=133\ pounds$$

$$Grate\ area=\frac{T}{500}=\frac{133}{500}=0.266\ square\ feet$$

Conversion from square feet to square inches:

$$\frac{12\text{in} * 12\text{in}}{1\text{sqft}}=\frac{144\text{sqin}}{sqft}$$

Continuing with the example:

$$0.266\text{sqft} * 144\frac{sqin}{sqft}=38.3\ square\ inches$$

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 ½ inch 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 are we will have a combined tube area of 1 square inch. From example above our total tube area will be

$$\frac{38.3}{8}=4.79\text{sqin}$$

Using a tube which has an outside diameter of ½ inch and wall thickness of 0.35 our inside diameter is approximately 7/16 inch. From Table I the area of a circle 7/16 inch diameter is 0.15 square inches. Now by dividing the total area 4-3/4 square inches by 0.15 square inches we have

$$\frac{4.79}{0.15}=32\ tubes$$

in our boiler.

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 scale Mogul which has a 2 inch bore and 2½ inch stroke, we find that the volume of steam used per one revolution of the drive wheel is

$$Vs = A * S * N$$

where

Vs = volume of steam per revolution of the drive wheel
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 * 2.5 * 4 = 31 cubit 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 ocomotives 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

$$Vs = \frac{31}{237} = 0.13 \, \textit{cubic inches of water per revolution of the drivers is required}$$

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 0.13 * 1.5 = 0.2 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 inches. 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

$$Pump\ area = \frac{Vw}{S} = \frac{0.2}{2.5} = 0.08\ square\ inches$$

where

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

From Table I we find that an area of 0.08 the corresponding diameter is 5/16 inch. Our pump has a plunger 3/8 inch diameter.

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 1/7 = 0.059. Our grate area is 38 square inches so to find the internal stack diameter we multiply 38 * 0.059 = 2.24 square inches. Table I shows that an area of 2.24 = 1-11/16 inch diameter. The length of the stack should be 4 diameters which is 4 * 11/16 inch = 6-3/4 inch.

POWER TO OPERATE PLAIN SLIDE-VALVES

The plain slide-valve is used in many model locomotives because of its 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 its seat. This 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 inch by 1½ inch which has an area of 1.875 square inches using a boiler pressure of 125 pounds we have a force of 1.875 * 125 = 233 pounds 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

$$\frac{233}{4} = 58\ pounds$$

Now to find the total pressure of the valve against the seat we take 233 – 58 = 175 pounds. One way of looking at it is that we have to push the valve back and forth with a 175 pounds 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 value, 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 pounds. This 17.5 pounds 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.

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 4,000, or 520 pounds.

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 offsets 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 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 inch scale locomotives of average size a spring of from 0.035 to 0.060 inch thick will give good results. Weight is something that is hard to come by but with a little calculation it can be approximated close enough to give good results. Now for the formula:

$$N=\frac{L*S*11}{W*T^2}$$

where

N = the number of leaves needed for each spring
L = total weight on drivers in tons
S = length of spring in inches
W = width of spring in inches
T = thickness of one leaf in 1/16 inch

Working out a typical problem we have

Total weight on drivers …........................ 400 pounds
Number of springs …............................... 6
Weight each spring has to support …....... 400/6 = 66 pounds
Length of spring ….................................. 3.5 inches
Width of spring …................................... 0.375 inch
Thickness of one leaf …........................... 0.050 inch

L = 66 pounds / (1 ton / 2,000 pounds) = 0.033 tons
S = 3.5 inches
W = 0.375 inch
1/16 inch = 0.0625 inch
T = 0.050 inch / 0.0625 inch = 0.8

$$N=\frac{0.033*3.5*11}{0.375*(0.8)^2}=5\,leaves$$

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.

$$D=\frac{L^3*1.5}{W*T^3*N}$$

where

D = deflection in 1/16 inch per ton

In our example we will fill in the known quantities.

L = 3.5 inches
W = 0.375 inches
T = 0.050 / 0.0625 = 0.8
N = 5

We now have

$$\frac{3.5^3 * 1.5}{0.375 * 8^3} = \frac{64.3}{0.94} = 6.8 \quad \text{sixteenths of an inch deflection per ton}$$

$$6.8 * 0.0625 = 0.42 \quad \text{inch deflection per ton}$$

Since our load on a single spring is only 66 pounds this is 66/2,000 = 0.033 ton. To get the deflection we will take 0.42 * 0.033 which gives us 0.014 inch total deflection.

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

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 driver 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.

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

Editors note: Winton included several tables with pre-computed values that simplified the use of the formulas he provided. His document was originally written in the era prior to the availability of calculators, computers and spreadsheets. Tables that were computed from a formula, such as Table I square roots and squares, have been replaced by their formulas. Many of these functions are built into most calculators and spreadsheet programs.

Table I: Circumference and Areas of Circles

$$c=\prod * d=2*\prod * r$$

and

$$a=\prod * r^2$$

where

c = circumference of a circle
a = area of a circle
r = radius of the circle, equal to ½d
d = diameter of the circle
$\prod$ = 3.1416

Table II: Inches to Decimals of a Foot

$$f=i*12$$

and

$$i=f/12$$

where

i = inches
f = feet

Table III: Inches to Square Root to Square

From Wikipedia:

> In mathematics, a **square root** of a number a is a number y such that $y^2 = a$, or, in other words, a number y whose *square* (the result of multiplying the number by itself, or $y \times y$) is a. For example, 4 and -4 are square roots of 16 because $4^2 = (-4)^2 = 16$.

and

> In algebra, a **square** is the result of multiplying a number, or other expression, by itself. In other words, squaring is exponentiation to the power 2.
>
> The squaring function is denoted by a superscript 2, as in $(x+1)^2$. However when superscripts are not available, as for instance in programming languages or plain text files, the notations *x*^2 or *x***2 are commonly used.

Table IV: Strength of Model Engineers Hex Head Bolts Made of Brass – Coarse Thread

Size	Outside Diameter	Stress Area	Safe Load Pounds	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
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	Outside Diameter	Stress Area	Safe Load Pounds	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	0.500	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
7/18-14	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

Valve Gear Cut-Off	Boiler Pressure Multiplier
¼	0.597
1/3	0.670
3/8	0.743
½	0.847
5/8	0.919
2/3	0.937
¾	0.966
7/8	0.992

Table VII: Volumes of Saturated Steam

Gauge Pressure	Cubic inches of steam per cubic inch of water	Temperature F
50	406	298º
60	355	308º
70	316	316º
75	299	320º
80	285	324º
85	271	328º
90	258	332º
95	247	335º
100	237	338º
110	219	345º
120	204	350º
125	197	353º
135	184	358º
150	169	366º
175	150	378º
200	131	388º
225	117	297º
250	107	406º

Table VIII: Strength of Materials

Material	Tensile strength Pounds per square inch
Aluminum: cast	15,000
Aluminum: cast high strength	26,000
Aluminum: bar stock 2011-T3 excellent machining	54,000
Aluminum: bar stock 2011-T8 excellent machining	59,000
Aluminum: bar stock 2024-T4 good machining	68,000
Aluminum: structural shapes 6061-T6 excellent for car frames	45,000
Copper: sheet hard	46,000
Copper: sheet soft	33,000
Copper: rod hard	45,000
Copper: rod soft	32,000
Brass: bar stock free cutting	58,000
Brass: stock high lead	73,000
Bronze: #1012 Everdur	95,000
Bronze: Tobin bronze	63,000
Steel: C1018 cold finished bar	82,000
Steel: Stressproof – excellent machinging, fine finish	125,000
Stainless: Type 303 free machining	75,000
Stainless: Type 321 fair machining	75,000
Stainless: Type 416 good machining – annealed condition	75,000
Stainless: Type 416 ppor machining in heat treated condition	90,000-200,000
Iron: cast grey	18,000
Iron: 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, Figure 1.

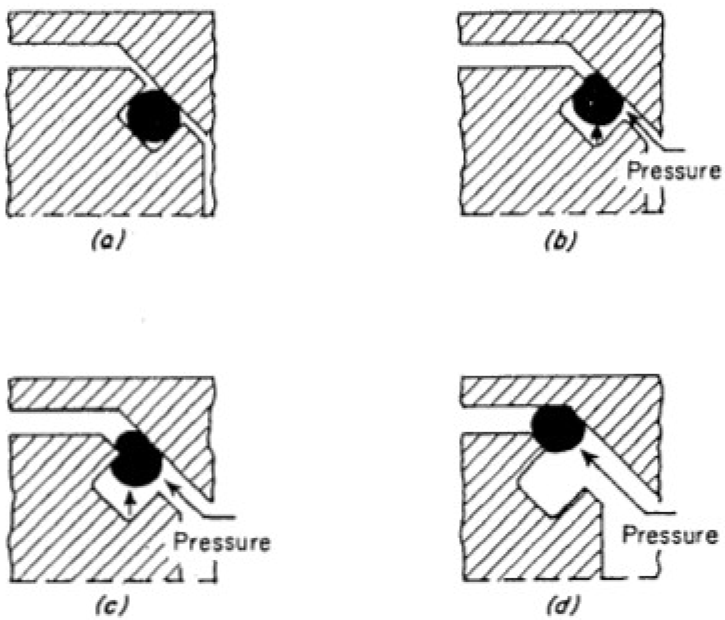


Figure 1. Blowout of O-ring used as a valve seat. As the valve opens, the space between the two faces becomes larger. The pressure acts on the O-ring. The ring continues to seal the opening util it is completely stretched out of the groove.

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, Figure 2.

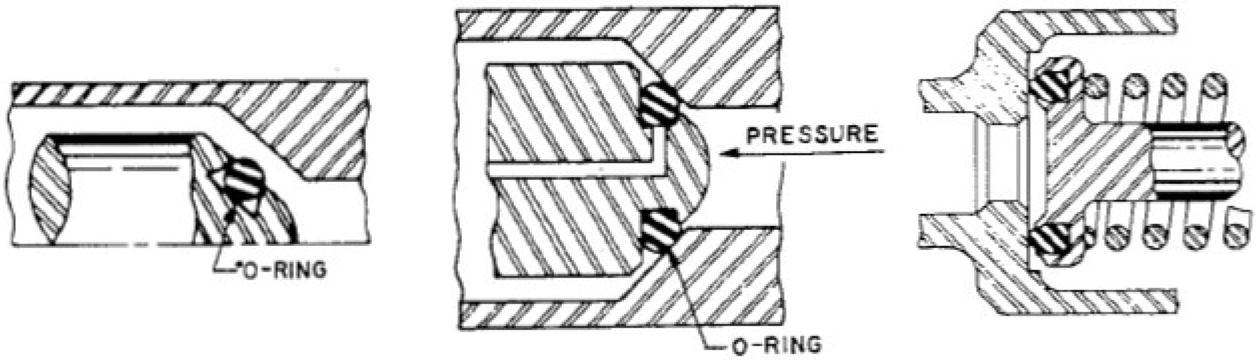


Figure 2. Groove Designs to Prevent Blow-Out

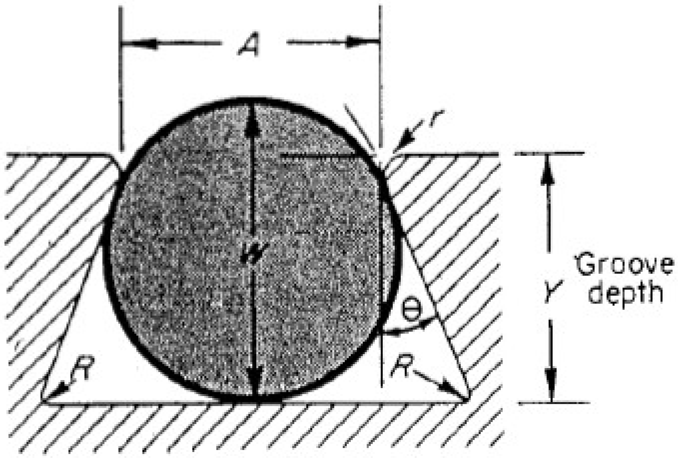


Figure 3. Standard Dovetail Groove Sizes for O-Ring Seals

Standard O-ring Size and Installation Data

O-Ring Size Number	**O-Ring *W***	***A* Grove Length**		***Y* Groove Depth ±0.002**	**Radius *r***	**Radius *R***
		Sharp Edge tolerance ±0.002	**Rounded tolerance ±0.002**			
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	0.063	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

Material	Shrinkage allowance – inches per foot
Cast iron	1/8
Brass	3/16
Steel	1/4
Aluminum	3/16

Table XI: Standard Keyways and Setscrews

Diameter of Hole inches	Standard Keyway inches		Recommended Setscrew
	w	d	
5/16 to 7/16	3/32	3/64	10-32
½ 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 ¼	¼	1/8	3/8-16
1 5/16 to 1 3/8	5/16	5/32	7/16-14
1 7/16 to 1 ¾	3/8	3/16	1/2-13
1 13/16 to 2 ¼	½	¼	9/16-12
2 5/16 to 2 ¾	5/8	5/16	5/8-11
2 13/16 to 3 ¼	¾	3/8	3/4-10
3 5/16 to 3 ¾	7/8	7/16	7/8-9
3 13/16 to 4 ½	1	½	1-8
4 19/16 to 5 ½	1 ¼	7/16	1 1/8-7
5 9/16 to 6 ½	1 ½	½	1 ¼-6

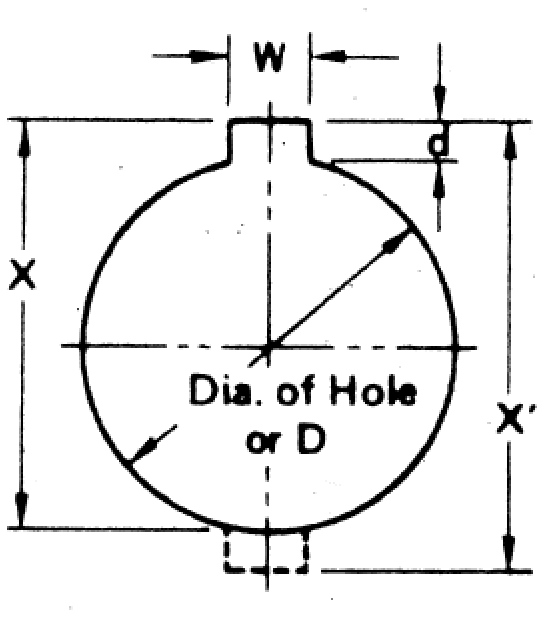


Figure 4. Keyway Dimensions

$$X=\sqrt{(D/2)^2-(W/2)^2}+d+D/2$$

$$X'=2\mathrm{X}-D$$

Example: Hole 1"; Keyway 1/4" wide by 1/8" deep

$$X=\sqrt{(1/2)^2-(1/2)^2}+1/8+1/2=1.109\,inches$$

$$X'=2.218-1.000=1.218\,inches$$

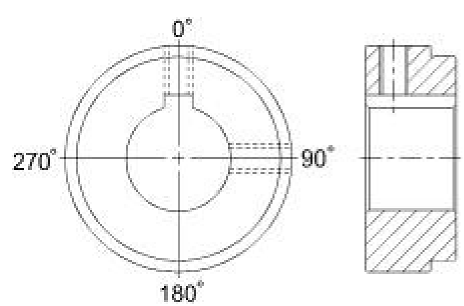


Figure 5. Two set screws should be set at 90°

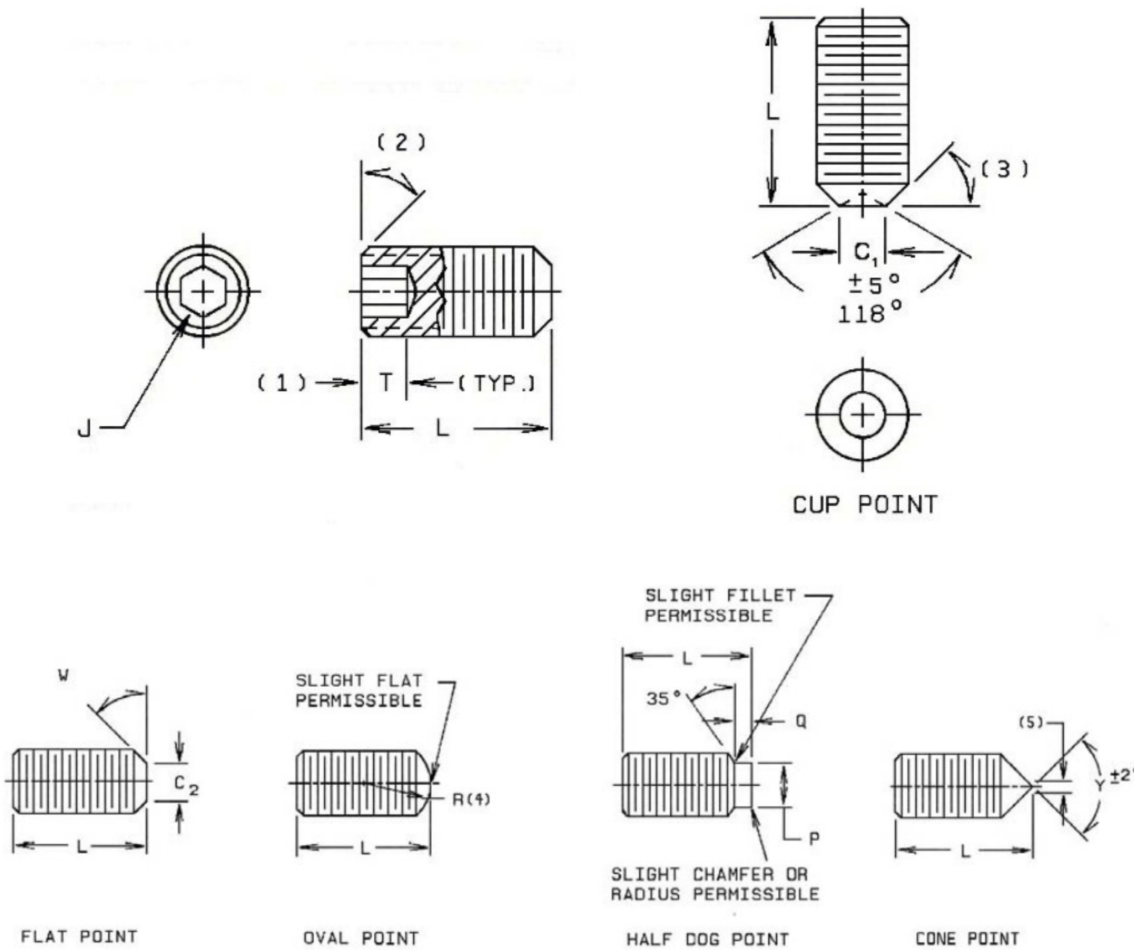


Figure 6. Inch Series Socket Set Screws

Inches Series Socket Set Screws

Nominal Size	Basic Screw Diameter	J Hex Socket Size	T Minimum Key Engagement	Threads per inch	
				UNRC	UNRF
#0	0.060	0.028	0.035	-	80
#1	0.073	0.035	0.035	64	72
#2	0.086	0.035	0.060	56	64
#3	0.099	0.050	0.070	48	56
#4	0.112	0.050	0.070	40	48
#5	0.125	1/16	0.080	40	44
#6	0.138	1/16	0.080	32	40
#8	0.164	5/64	0.090	32	36
#10	0.190	3/32	0.100	24	32
¼	0.250	1/8	0.125	20	28
5/16	0.312	5/32	0.156	18	24
3/8	0.375	3/16	0.188	16	24
7/16	0.437	7/32	0.219	14	20
½	0.500	¼	0.250	13	20

Table XII: Proportionate Weight of Castings to Weight of Wood Patterns

A pattern Weighing one Pound (Less Weight of Core Prints)

Pattern Material	Cast Iron	Brass	Copper	Bronze	Bell Mateal	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	0.81

Table XIII: Length of Pipe Thread

Length of Thread on Pipe Screwed Into Vales or Fittings To Make A Tight Joint

Size Inches	Dimension A Inches
1/8	¼
¼	3/8
3/8	3/8
½	½
¾	9/16
1	11/16
1 ¼	11/16
1 ½	11/16
2	¾
2 ½	15/16
3	1
3 ½	1 1/16
4	1 1/8
5	1 ¼
6	1 5/16
8	1 7/16
10	1 5/8
12	1 ¾

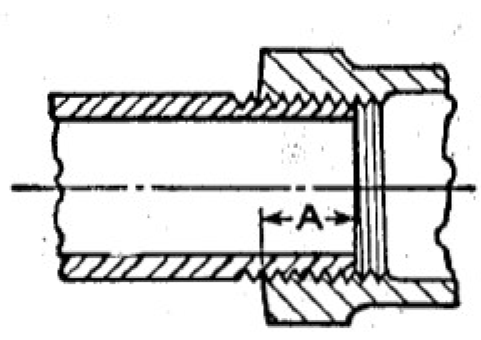


Figure 7. Pipe Thread Length A

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 b e the approximate weight in pounds.

Material	Pounds per cubic inch
Iron	0.2777
Steel	0.28332
Copper	0.32118
Tin	0.26562
Brass	0.3112
Lead	0.41015
Zinc	0.25318
Aluminum	0.09375
Bronze	0.2947

Table XV: Approximate Weight of Various Liquids

Liquid	Pounds per cubic inch	Pounds per cubic foot	Pounds per gallon
Hydraulic oil	0.0312	53.9	7.2
Diesel Fuel	0.0303	52.1	7.0
Kerosene	0.0295	51.0	6.8
Water	0.0361	62.4	8.3

How to Figure Water Consumption for Miniature Steam Locomotives

Published in “The Miniature Locomotive” November/December 1954 edition.

Properties of Saturated & Superheated Steam

When water is heated in an open vessel it will boil at 212°F at sea level. This is as hot as the water will get regardless of how long the heat is applied. At this temperature the water turns to a vapor called steam. If we close the vessel and keep the steam generated confined the pressure and temperature will gradually rise.

When steam remains in contact with water its temperature will be that of the water. As long as these two remain together the steam will have a certain amount of suspended particles of water in it. Steam in this condition is called Saturated Steam. This is the lowest temperature at which steam can exist and when used in this condition it will condense rapidly when it comes in contact with the cool cylinder walls. This rapid condensing causes the pressure to drop and consequently a loss in power.

If the saturated steam is removed from contact with the water and further heated its pressure will not rise, only the temperature will increase. The higher the temperature the dryer the steam will be and greater the volume. This increased volume comes from speeding up the molecules thus making the steam lighter in weight and thereby increasing its volume. This increased volume is obtained without additional fuel consumption by passing the saturated steam through coils of pipe located in the flues called superheater tubes.

The steam table shows the relationship between saturated steam temperature and volume to superheated steam volume at various degrees of superheat.

Example

At 100psi the temperature of water and saturated steam is 338°F. Its corresponding volume per cubic inch of water is 239. This means that for every one cubic inch of water evaporated 239 cubic inches of steam will result.

At the same pressure of 100psi but at 400°F of superheat the corresponding volume will be 265 cubic inches of steam per one cubic inch of water. Thus by raising the temperature of the steam from the steam dome to the cylinders will have an additional 26 cubic inches of steam to use without burning additional fuel. This is an increase of 11%. Thus it can be seen that superheated steam is more efficient than saturated steam. For this reason all modern steam locomotives and steam generating plants use high temperature superheated steam. Also the higher temperature steam reduces cylinder condensation which in turn increases the engine's efficiency.

In brief, superheating steam gives more power without expenditure of additional fuel.

Table XVI: Cubic Inches Steam per Cubic Inch Water

Steam Pressure psi	Temp °F	Cubic Inches Saturated Steam Per Cubic Inches Water	Superheated steam temperature °F Cubic inches superheated steam per cubic inches water			
			350°F	**400°F**	**450°F**	**500°F**
25	267	648	728	777	823	874
50	298	410	444	475	499	534
75	320	292	316	339	362	382
100	338	239	246	265	283	244
125	353	197		180	194	207
150	366	170		157	166	177
175	378	150		135	146	156
200	388	131		117	129	138
225	397	117			116	125
250	406	107				

Formula

V = Cubic inches of steam required per revolution of drive wheels

V = ∏ * R * R * S * N * Co

where

R = radius of cylinder in inches
∏ = constant pi (3.1415)
S = stroke of piston in inches
N = number of working strokes per revolution of drivers
Co = % cut-off , part of stroke at which the steam is shut off. This is usually 85% at full gear.

Example

This example illustrates how to calculate the amount of water required to generate sufficient steam to keep a locomotive running under full load.

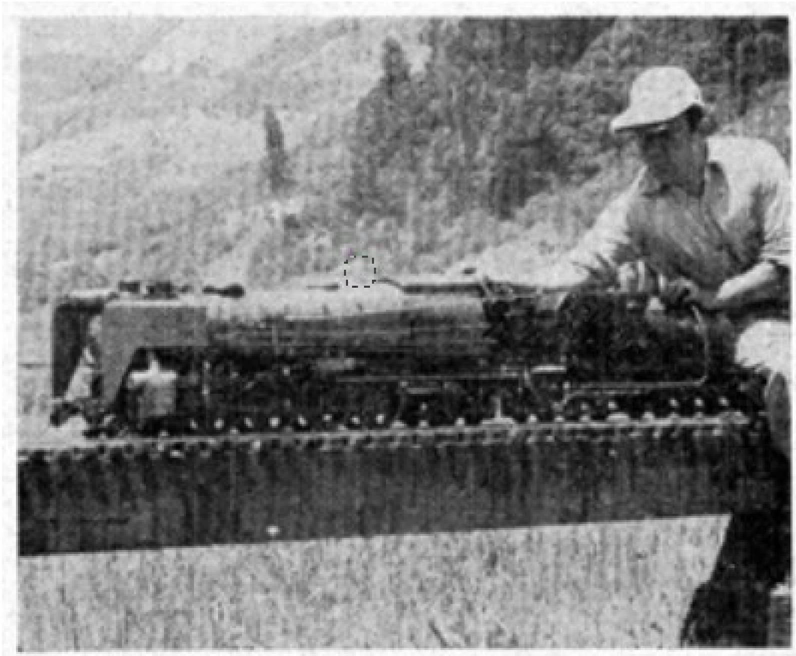
R = 0.6875
S = 1.75
N = 4
Co = 0.85

V = 3.1415 * 0.6875 * 0.6875 * 1.75 * 4 * 0.85
V = 8.8 cubic inches steam used per revolution of drive wheels

- 60 scale mph = 250 rpm on 5” diameter drivers
- Cubic inches steam at full speed = 250 * 8.8 = 2,200 cubin inches per minute at 125 psi at 400°F
- 1 cubic inch water generates 216 cubic inches steam

2,208 / 216 = 10.18 cubic inches water per minute to run locomotive at full load full speed

The above example is the theoretical steam consumption. To this should be added steam for the steam blower (about 10% to 15% of the engine's running steam consumption). Also an additional 10% to 15% for steam driven water pump or injector. In my calculations I add 25% to my steam consumption figures to take care of the blower, feed water pump, injector and whistle.


Winton Brown and his loco on the Golden Gate Live Steamers track at Oakland, California.