# L.S.M.R.E.

## LARGE SCALE MODEL RAILWAY ENGINEERING

MILWAUKEE
LIGHT ENGINEERING
SOCIETY
*Ltd.*

# STEAM

## Volume II

**Milwaukee Light Engineering Society**
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MLES



December 15, 1994

Fellow members & modelers:

In the following volumes of L.S.M.R.E. you will find reprints of the articles that Tom Artzberger has written for the GAZETTE starting back in 1991.

These reprints are compiled pretty much in the order in which Tom has submitted them for print, with some changes.

On the following page you will find a reprint from an old *M.L.E.S.* BULLETIN on Tom's background. I think you'll find it interesting as well.

Tom continues to write articles for the GAZETTE, and additional volumes will follow, or updates will be made available as necessary.

I'm sure you'll find these volumes extremely helpful and easy to follow to get you started on some easy and nice looking equipment.

Happy Steaming

Ken Rodig

Ken Rodig, Sec.

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From the Secretary's Desk

In the future I promise to get a little more thorough on the minutes of the last meeting's events in the BULLETIN for those who didn't make the meeting. I'm not used to taking notes yet. But your best bet would be to make the meetings. Tom A's talk on hydraulics was very enlightening, and with more talks in the future like his, you wouldn't want to miss any more free knowledge. I did video the last meeting talk by Tom and we're thinking of starting a video library of the special talks coming up too.

Now that the holidays are behind us I can get going on the new "New Membership Information" packet I started a month ago. It will contain the revised club bylaws and other pertinent information on the club. This packet will be available to all present members also in the event they never received one when they joined. Just call or write me to order one.

This month in the "Getting to Know Your Fellow Member" column I'm covering a gentleman that's well known to many of us in the club for a long time. However with all the new members joining recently, and for the organizations that receive the news letters outside the club, it might be nice to have a little background on him with the series he's writing for the GAZETTE and the talks he gave and will be giving.

Getting to Know Your Fellow Member: THOMAS ARTZBERGER

Tom has been a member of the M.L.E.S. almost from the beginning. He missed the very first or charter meeting, but from the second meeting on, he's been instrumental in the organization and formation of the society to date. Tom received his Bachelors degree in Engineering at the Milwaukee School of Engineering in 1971. That was two years after the club's formation in 1969. Today he's the Chief Design Engineer of the M.B.W. Ground Pounders Corp. in Slinger, WI which produces industrial concrete finishing equipment.

In 1970 when the R.R. Div. of the M.L.E.S. was formed this was right down Tom's alley. His interest in trains and Steam Engines goes back to his childhood. Tom's grandfather was even the shop superintendent for the Porter Locomotive Works and after his retirement was commissioned to construct a 2" scale 4-8-4 Northern loco for a private party, which is still running today.

In 1971 Tom started writing the technical articles for the then R.R. Div. STACK TALK paper. At the time Tom was the Division Chief and has held that and a number of other offices in the club's history. Then when the land was acquired for the R.R. Park, Tom did most of the surveying and grading of the mainline with his own small crawler loader and through the years has been a major contributor in park maintenance and construction with his personally owned equipment. He's always been willing to bring it out any time to help with projects and grass cutting.

Tom's first live steam engine was a light two truck shay he built some 20 years ago. Since then he's been setting his machine shop up and has built a number of engines, including a EMD SW-1200 diesel switcher, a Southern Pacific 4-6-0 Ten-wheeler, and a 40 Ton Climax geared logging loco. At this time he's in the process of building two EMD GP-7 diesels, one gas powered, one electric powered--all 1-1/2" scale, along with a 15" gauge diesel locomotive for the Riverside and Great Northern Society, intended for the Milwaukee County Zoo Line R.R. nearing completion for summer 1992 operation. He's also built about a dozen cars to include flat cars, gondolas, hoppers, a caboose, and even a drop bottom gondola, along with helping with the major rebuilding of a 0-4-2, 2-1/2" scale plantation loco into a 2-4-4, 1-1/2" scale logging engine.

Those who know Tom, know that he's always willing to share his knowledge or lend a helping hand, and the club wouldn't be the same without him.

Ken Rodig

Rule One - SAFETY FIRST !
Rule Two - HAVE FUN !

## Large Scale Model Railway Engineering

### Introduction

Designing a good performing model locomotive is a relatively simple job. During the next several months I will describe the procedure that I use when I build a diesel or steam locomotive. This month we will begin by calculating the train weight and from this determine the weight and horsepower required. In future months design parameters for a diesel and steam locomotive along with construction suggestions will be covered.

So lets get started.

### Section 1 The load

The first thing one needs to know is the load that one wants to pull. We will use 1 1/2 scale as an example, but this procedure also applies to the small and larger scales. One may wish to pull 6 cars with 2 people in each car. A 1 1/2 scale car weighs about 100 lbs and we will use a average weight of 125 lbs. for each person. We can then calculate the total train weight as follows:

1 car 100 lbs.
2 people 250 lbs.
(1) total for each car = 350 lbs.

350 lbs / car X 6 cars = 2100 lbs.

We must also add the weight of the locomotive which we will say is 400 lbs. The total train weight is therefore:

locomotive 400 lbs.
train 2100 lbs.
(2) total train weight = 2500 lbs.

The rolling resistance of a train on straight level track is approximately 10 lbs. per ton of train weight. Using our results from equation 2 above we can calculate the rolling resistance as follows:

(3) 2500 lbs / 2000 = 1.25 tons
(4) 1.25 tons x 10 = 12.5 lbs rolling resistance

To the rolling resistance we must add the resistance due to grade and curvature. At high speed (over 30 mph) we must also add air resistance, but since our speed are much slower than this we can ignore this factor. The grade resistance is approximately 20 lbs. per ton for each 1% of grade (1 ft in 100 ft.). In our example we will assume a 2% grade.

(5) 20 lbs per ton X 2% grade = 40 lbs. per ton

Since our train weighs 1.25 tons (equation 3).

(6) 1.25 tons x 40 lbs. per ton = 50 lbs. grade resistance

The curve resistance has been found to be the following:

(7)

| | | |
|---|---|---|
| 35 ft. radius | = | 16 lbs. per ton |
| 45 ft. radius | = | 12 lbs. per ton |
| 60 ft. radius | = | 10 lbs. per ton |

We will use a 45 ft radius in our example. To find the total train resistance we will add the values for the rolling resistance (equation 4), the grade resistance (equation 6) and the curve resistance (equation 7).

| | | | |
|---|---|---|---|
| | rolling resistance | = | 12.5 lbs. |
| | grade resistance | = | 50 lbs. |
| | curve resistance | = | 12 lbs. |
| (8) | total train resistance | = | 74.5 lbs. |

This means that the locomotive must have at least 74.5 lbs. of tractive effort to pull our train. In order to generate this tractive effort we need two things. First we need enough power on the wheels and second enough weight to prevent the wheels from spinning. This last factor is a function of the friction between the wheel and the rail. This is known as the coefficient of adhesion, which for steel or cast iron is approximately 0.25 under ideal conditions. This value drops off rapidly under wet conditions. The maximum tractive effort that can be generated is equal to the weight on the driving wheels times the coefficient of adhesion. In this discussion we will assume that all the weight are on the drivers.

(9) 400 lbs x .25 = 100 lbs. tractive effort

In equation 8 we calculated that we need 74.5 lbs of tractive effort to pull our train and equation 9 tells us that we can generate 100 lbs.

The next thing we must consider is the speed that the train is to run. This is typically 4 - 6 miles per hour with a maximum speed of 8 mph when dealing with 1 1/2 scale. The speed is a function of the wheel diameter and revolutions per minutes(rpm). In this example we will use a 40 inch diameter wheel which is 5" in diameter in 1 1/2 scale. To calculate the rpm required for a top speed of 8 miles per hour we proceed as follows:

(10) 1 mile per hour = 88 ft per minute

(11) 88 ft per minute = 1056 inches per minute

The circumference of a wheel is found by multiplying the diameter by the value of "pi" (3.1416).

(12) 5 in diameter x 3.1416 = 15.70 inches

(13) 8 miles per hour = 8,448 inches per minute

To find our rpm divide 8448 inches per minute by the circumference of the wheel (equation 12).

(14) 8,448 / 15.70 = 538 rpm

We now have the speed (rpm) and the required tractive effort. The last step is to calculate the axle torque required to generate this tractive effort. To do this we will use the maximum tractive effort calculated in equation 9 which was 100 lbs., and the radius of the wheel (1/2 the diameter). The total axle torque is tractive effort times the wheel radius.

(15) 100 lbs x 2.5 inches = 250 in lbs of torque

Our last step is to calculate the approximate horsepower required to generate this torque at this speed. This will be a rough approximation which we will refine latter. The horsepower is found by multiplying the torque (in in-lbs) by the speed (rpm) and dividing the result by 63025.

(16) hp = 250 in-lbs x 538 rpm / 63025 = 2.13 hp.

To sum up we have determine that we will need 250 in lbs of torque at 538 rpm to meet our performance expectations.

That's it for this month. Next month we will examine the power train for a miniature diesel locomotive.

## Section 4 STEAM

After a couple of months off I'm back at it again. This month I am going to start dealing with Engineering a steam locomotive. Starting with the Basics, our first subject of discussion is STEAM. This will give us a good foundation when we talk about boilers and cylinder sizes, etc.

Steam is generated by evaporated water. The point that is occurs is called the Boiling Point which is a function of the pressure exerted on water. Pure water at sea level (standard atmospheric pressure = 14.7 PSI) boils at 212 degrees F. This temperature increases as the pressure goes up and drops if the pressure decreases (vacuum conditions). Table 1 shows the boiling point at various temperatures.

There are two types of steam that is used in power applications: saturated and superheated. Saturated steam is defined as steam that is in contact with and in thermal equilibrium with the water. The temperature when this occurs is called the saturation temperature, ie 212 degrees F at standard atmospheric pressure. If either the pressure or temperature is known, the other can be determined from the tables or curves that are published in many technical hand books.

Superheated steam is defined as any steam at a temperature above its saturation temperature. To define the state of supeheated steam we usually state the pressure and the temperature. Often we speak of the degrees of superheat, which is the difference between the actual steam temperature and the saturation temperature.

Another term that is often used is the quality of the steam. The quality is the percentage of vapor in the vapor-water mixture. For example 95% quality would mean 95% vapor and 5% water by weight.

On our miniature locomotives we usually use saturated steam ( no super heaters) because of the problems involved with small superheater tubes and the fact that it is difficult to get a large amount of superheat in a small boiler. The amount of work that can be performed by a steam engine depends on the amount of heat that can be released from the steam. the higher the pressure the greater this heat. With saturated steam the heat available depends entirely on the pressure, while with superheated steam there is additional heat that was added in the superheating process that can be released as useful work.

Table 1 shows the properties of saturated steam at the pressures we use in our locomotives. This data will be used later when we start sizing the boiler, cylinders, water pumps and other components. The first column of table 1 is the boiler pressure, the second column is the amount of heat required to generate the steam starting with 32 degree F water. The third is the volume of one pound of steam, the fourth is the steam temperature and the last column is the volume of steam generated by one cu inch of water.

Next month we will start sizing components.

| boiler pressure | btu | cu ft./lb | temp F | cu in steam per cu in water |
|---|---|---|---|---|
| 80psi | 1187 | 4.60 | 325 | 285 |
| 90psi | 1188 | 4.30 | 330 | 258 |
| 100psi | 1189 | 3.88 | 338 | 237 |
| 110psi | 1190 | 3.59 | 344 | 219 |
| 120psi | 1191 | 3.33 | 350 | 204 |
| 130psi | 1193 | 3.11 | 355 | 190 |
| 150psi | 1195 | 2.75 | 366 | 169 |

TABLE 1

## Section 4 The Steam Locomotive

This month we are going to continue our discussion on designing a steam locomotive by determining the cylinder and drivers size required to pull a given size train.

To make things simple we will use the same parameters that we used back in section 1 when we talked about the amount of tractive effort required to move our train over the railroad. We calculated that we needed 74.5 lbs of tractive effort and we wanted a top speed of 8 miles per hour. The only other piece of information that we need to size our cylinders and wheels is the boiler pressure. We will use 125 PSI in our example which is typical for a 1 1/2 scale model.

Our first step is to pick the size of the drivers, which are usually determined by the size of the drivers on the prototype locomotive or by what is available from the live steam suppliers. One should remember that as a general rule of thumb the wheel speed should be kept below about 300 RPM. Using 300 RPM and a 8" diameter wheel we can calculate our track speed by finding the circumference of the wheel and multiplying by the RPM.

8" dia X3.1415 = 25.13" per revolution

25.13"/revolution X 300 RPM = 7539 in per minute

7539" per minute /1056in/min./MPH = 7.13 MPH

This is close to the 8 MPH that we seek, so our drivers will be 8" in diameter. The cylinder stroke is usually about a third of the driver diameter which in our case is 2.66". we will round it up to an even 3".

Now that we know the driver size, cylinder stroke, boiler pressure and the required tractive effort required we can find the cylinder bore that we should use.

Before we calculate the cylinder bore however we have to talk about the boiler pressure and cylinder pressure. First of all we have pressure drops between the boiler and the cylinders and back pressure on the piston due to restrictions in the exhaust. In addition to this we have to consider the fact that the steam is not being admitted to the cylinder for the full stroke. The valve motion cuts off the incoming steam before the piston gets to the end of its stroke to allow the steam to expand before being exhausted up the stack. This amount of cut off is dependent on the valve gear type and geometry and the position of the reverse level. The pressure that we have to use in our calculations is the average pressure or the mean effective pressure (MEP) as it is usually called.

The actual MEP can be measured by a device called an engine indicator which measures the cylinder pressure vs. the piston position. The engine indicator creates a plot on a piece of paper called an indicator chart. From this chart much information can be gathered including the mean effective pressure.

One can however estimate a value if the valve cut off is figured by making a full size drawing of the valve gear motion. More on this latter.

| cut off | MEP |
|---|---|
| 87% | 0.99 X inlet pressure |
| 75% | 0.96 X inlet pressure |
| 66% | 0.93 X inlet pressure |
| 50% | 0.84 X inlet pressure |

Pressure drop between the boiler and cylinders can be estimated at 15%, therefore the MEP for our example using 75% cut off would be:

125 PSI X 85% efficiency = 106 PSI

from above
106 PSI X .96 = 101 PSI MEP

The cylinder size can be found using the equation:

$$d^2 = \frac{\text{TRACTIVE EFFORT (LBS)}}{\text{MEP} \times \text{STROKE(FT)} / \text{Driver DIA (FT)}}$$

$$d^2 = \frac{74.5 \text{ LBS}}{101 \text{ (MEP)} \times .25 \text{ (FT)} / .687 \text{ (FT)}}$$

$$d^2 = 1.96$$

$$d = \sqrt{d^2} = 1.40 \text{ BORE}$$

In the above equation we get a bore size of 1.40" which seem a little small. Well it is because our original value for the tractive effort that we needed was in fact low for a locomotive with 8" drivers. A more realistic value would be about twice this or 150 lbs. Using this value we find that we require a bore of 1.99" which we will round up to 2.00". For the rest of our discussion we will use cylinders with 2 inch bore and 3 inch stroke with 8 inch drivers.

Next month, THE BOILER.

Large Scale Model Railway Engineering

Section 4 The Steam Locomotive (continued)

Last time we calculated the cylinder and drive wheel size for our locomotive; the next thing we want to do is size our boiler to supply enough steam.

There is no exact formulas for determining the size of the boiler because of the many variables involved, however over the years I have found some general guidelines that if followed usually give satisfactory results. These are general guide lines and not hard and fast rules.

## I. HOW MUCH STEAM

The first calculation we must make is to determine the quantity of steam needed. This can be found by multiplying the area of the cylinder times the stroke times 4 strokes per revolution. For example:

Piston Area = (2" DIA / 2) X 3.1416 = 3.14 in sq.

Stroke = 3"

Volume = 3.14 in sq. x 3" stroke x 4 strokes/ Rev =
= 37.7 In cu. / Revolution

We will use this later when we size the feed water pumps or injectors.

## II GRATE AREA (fire box)

To find the grate area, one common method that I use is to take the maximum tractive effort and divide it by 3.5. From my example:

$$\text{Grate Area} = \frac{150 \text{ lb T.E.}}{3.5} = 42.8 \text{ sq. in.}$$

This has worked out well on the locomotive that I have built when using coal as fuel. A slight or larger area is preferred for oil or propane fired boilers.

## III NUMBER OF FLUES

I have found that the area of the flues should be approximately 1/8th the area of the fire box. For our example of 42.8 sq in. fire box area this would mean that we need a total flue area of 5.35 sq. in. Our next step is to determine the size of the flues we want to use and from this we can determine the number required. If one plans on burning coal it is important that we keep the ID of the flues big enough so they do not plug up. Usually it is recommended that they should not be smaller than 1/2 ID. On oil or gas fired boilers the flue size is not as important since there is less chance of plugging. In our example we will use a 3/4 OD X 5/8 ID tube which is common for an engine of this size.

The area of a flue tube is found by:

$$\begin{aligned} \text{Area} &= 3.1416 \times \text{rad}^2 \\ &= 3.1416 \times (.312)^2 \\ &= .305 \text{ IN SQ} \end{aligned}$$

If we need a total area of 5.35 in and each flue tube is .305 in we should use:

5.35 in / .305 in = 17.54 tubes

We will round this us to 18 flue tubes.

This is great assuming we can fit 18 tubes in our boiler. A quick layout on a piece of paper will give you the answer. One thing to remember is to allow sufficient steam space at the top of the boiler. I have found that it is often better to use fewer flue tubes and lower the crown sheet to gain space for steam and water than to squeeze in the maximum number of flues and have to have a very high crown sheet with little space for steam.

Next month we will finish our discussion on the boiler and take a look at fuels and the FRONT END (smoke box).

## Section 4 The Steam Locomotive (continued)

We will continue our discussion of the boiler by discussing the smoke box and how to properly layout the stack and exhaust nozzle.

The purpose of the smoke box is to create an area of vacuum to cause the flow of air up through the fire box and flues. This air flow aids in the combustion process of the fuel and greatly increase the heat transfer between the hot gases and the fire box and flues. The draft is created by ejecting the exhaust steam through a nozzle and up the stack, or by using steam from the boiler and an auxiliary nozzle(blower nozzle).

The first thing we will do is to design our stack and then the exhaust nozzle. Practice indicates that the smallest internal stack diameter should be approximately .06 times the great area. In our example our grate area is 42.8 sq in. , so our stack area is :

.06 X 42.8 sq in. = 2.56 sq in.

2.56 sq in. = 1.63 in diameter

So our stack should have a minimum id of 1.63" and a length of at least four diameters or 6.5" including the petticoat pipe.(see figure 1)

The exhaust nozzle is usually determined by trail and error but a good starting point is about .005 times the grate area. Therefore:

.005 x 42.8 sq in. = .214 sq in.

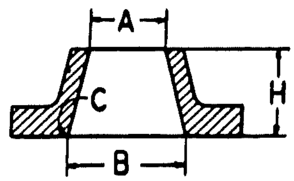
.214 sq in. = .522 in diameter

This is a good place to start but several nozzles should be tried to determine the best diameter. To small a nozzle will cause excessive back pressure on the cylinders and thus poor performance, a large diameter will reduce the draft resulting in problems keeping steam up. The nozzle shape also affect the amount of draft. The amount of exhaust blast exposed to the flue gases has a major affect on how much draft can be produced. Although a round hole is usually used as a nozzle in our small models, a star shaped or multi-ported design often gives better performance. Figure 2 shows nozzle shape that have been used over the years in full size locomotives. These shapes yields a greater surface area on the exhausting steam. This surface area causes drag on the smoke box gases which are pulled up the stack with the escaping steam thus creating a better vacuum and better draft. Fuel also has a major affect on the amount of draft needed. Much more air flow is required for solid fuel than for oil or gas and thus a smaller nozzle is usually required when burning solid fuel. The location of the nozzle with respect to the stack is also critical and will affect the efficiency of the smoke box.

The blower nozzle can be a multi-ported ring around the main nozzle or can be a single nozzle besides the main exhaust nozzle. The ring design is preferred because the jets can be directed directly up the center of the stack where as the single nozzle design be necessity must be off center so that the blower nozzle does not interfere with the main exhaust nozzle. Like the exhaust nozzle the blower nozzle is determined by trail and error to find one that generate enough vacuum for good steam generation but at the same time not be wasteful of steam.

Figure 1 shows a cross-section of a smoke box showing the exhaust stand and nozzle along with the stack.

Next month we will talk about the water supply.




All Dimensions in Inches

| *A* | *B* | *C* | *H* |
|---|---|---|---|
| 1 | 2.4 | 0 | 1¾ |
| 1¼ | 2.4 | 0 | 1¾ |
| 1⅜ | 2.4 | 0 | 1¾ |
| 1½ | 2.4 | 0 | 1¾ |
| 1⅝ | 2.4 | 0 | 1¾ |
| 1¾ | 2.4 | 0 | 1¾ |
| 1.5 | 2.3 | 0.4 | 4½ |
| 1.6 | 2.3 | 0.4 | 4½ |
| 1.7 | 2.3 | 0.4 | 4½ |
| 1.8 | 2.3 | 0.4 | 4½ |
| 2.0 | 2.3 | 0.4 | 4½ |
| 2.2 | 2.3 | 0.4 | 4½ |
| 1.5 | 1.5 | 0 | 1¼ |
| 1.5 | 1.5 | 0 | 3¼ |
| 1.5 | 1.5 | 0 | 4½ |
| 1.5 | 1.5 | 0 | 6¼ |
| 1.5 | 1.5 | 0 | 1¾ |
| 1.5 | 1.7 | 0 | 1¾ |
| 1.5 | 2.0 | 0 | 1¾ |
| 1.5 | 2.4 | 0 | 1¾ |

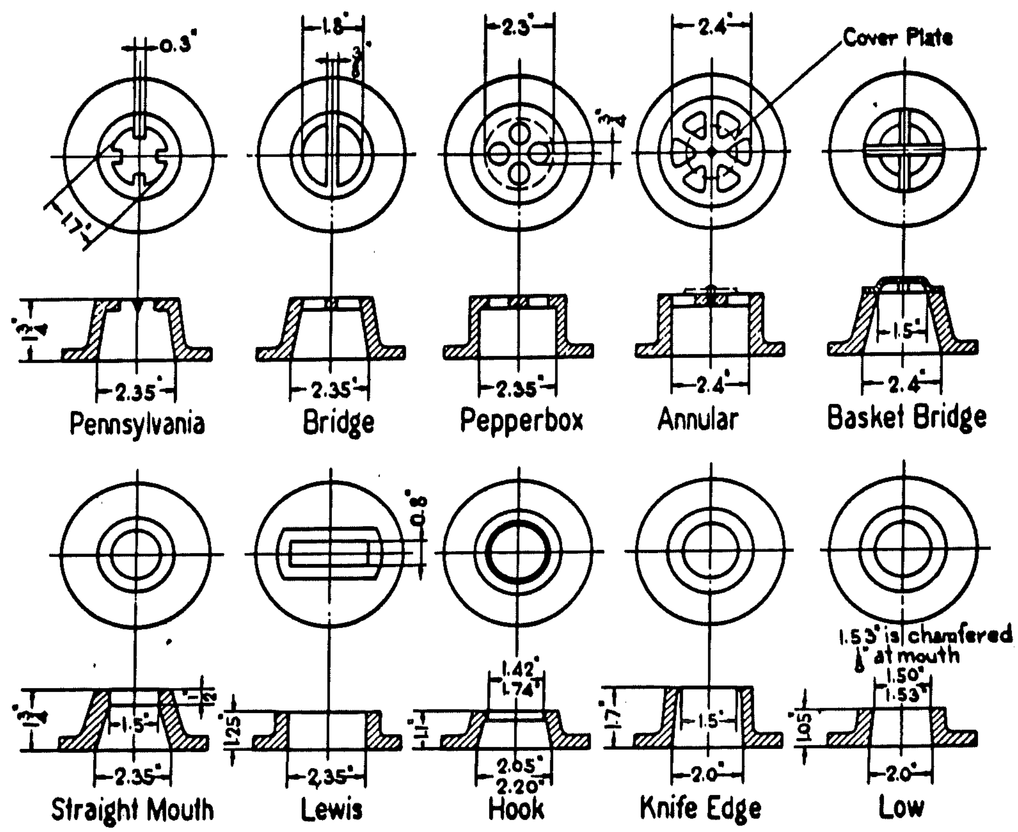
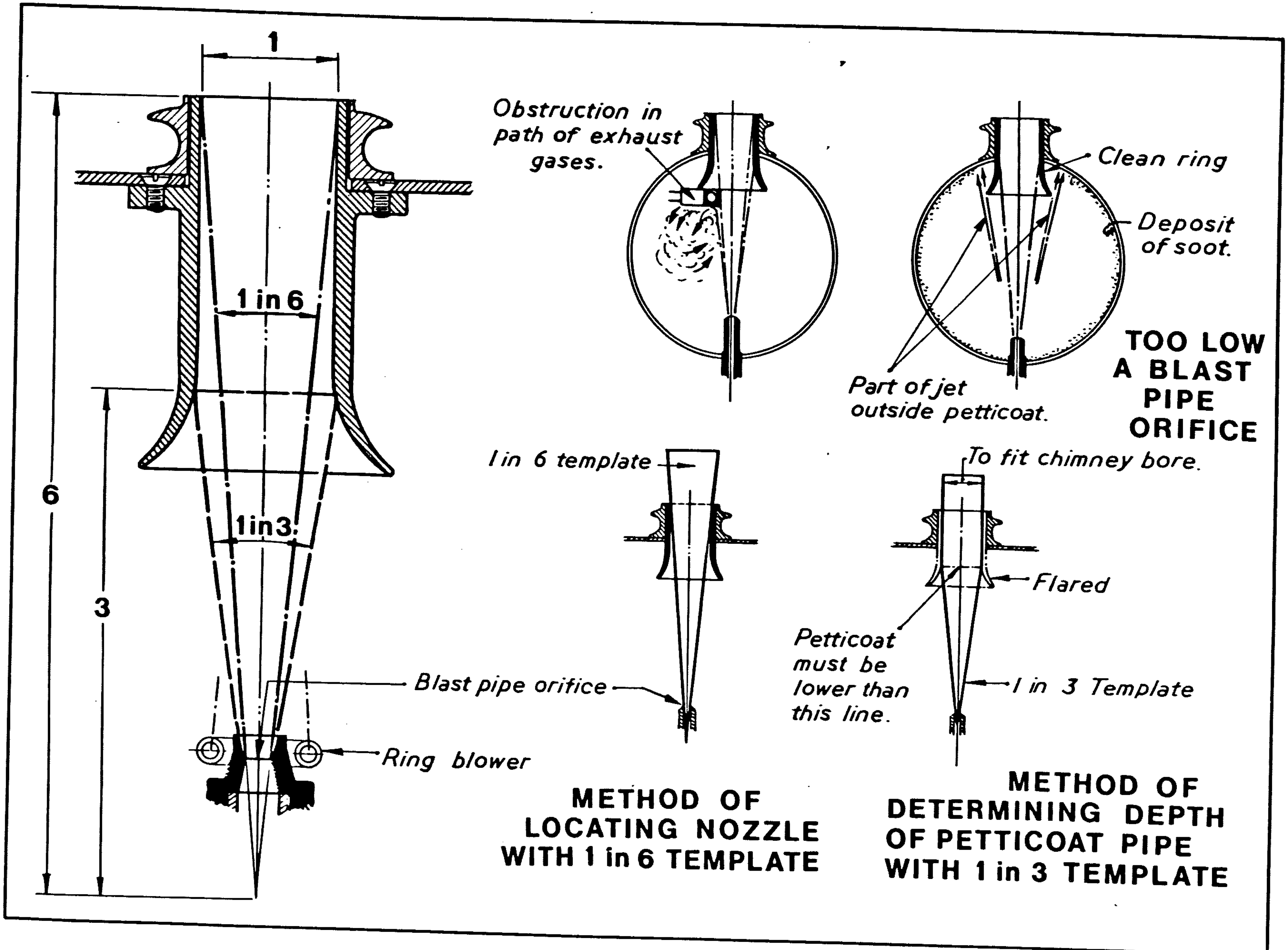



**Fig. 10—Nozzles Used in University of Illinois Tests**

Figure 2 "THE STEAM LOCOMOTIVE" by Ralph Johnson (Simmons-Boardman)




9 *The importance of blast-pipe setting out.*

Figure 1 (from Greenly's book "Model Steam Locomotives")

Large Scale Model Railway Engineering

## Section 4 The Steam Locomotive (continued)

In past months we discussed the properties of steam and the sizing of some of the major components to come up with a satisfactory steam locomotive. Our next topic will cover the methods of putting water into the boiler.

There must always be at least two ways of getting water into the boiler. On full size locomotives this was usually two injectors or an injector and a feed water pump combined with a feed water heater. On our models the same thing holds true; I usually recommend three, one that works when the locomotive is moving(axle pump), one that works when standing still(injector or steam pump, and a third(hand pump) if all else fails.

Each type of feed water device has advantages and disadvantages, along with different sizing requirements. We will look at three basic types, the mechanical pump, the injector and the steam driven pump. This month we will cover the mechanical pump(hand and axle) and next month the injector and steam driven pump.

Mechanical pumps:

The first thing we must do in sizing a pump is to determine the amount of water we have to put in the boiler under typical operating conditions. Several months ago we calculated the amount of steam that is needed per revolution of the drive wheels, which for our example was 37.7 cu in.

vol = area X stroke X strokes per revolution

vol = 3.14sq in X 3" X 4

vol = 37.7 in cu

Also from our discussion of steam we know that at 120 PSI, 1 in cu of water generates 204 in cu of steam. (table 1 section 4) Therefore we can calculate the amount of water required per revolution of the wheels by dividing 37.7 in cu by 204 in cu which gives us .185 in cu.

37.5 in cu / 204 in cu = .185 in cu

This assumes 100% efficiency and no leaks, I usually figure an extra 25% to make up for this.

.185 in cu X 125% = .23 in cu per revolution

We can now use this value to size our pump. We can find the displacement of a pump by multiplying the area of the piston by its stroke. We have to be careful however to select a piston size that is not overly big or a stroke that is too long. As the piston diameter increases it takes more force to push the water into the boiler against the boiler pressure , causing an uneven load on the locomotive which results in the locomotive lunging on each stroke. The best way around this problem is to use a pump with two cylinders with the the pistons 180 degrees out of phase so that one piston is on the pressure stroke while the other is on the suction stroke. A double acting pump where the piston pumps in both directions also helps.

Sizing the pump is a bit of trail and error. Lets try a single acting pump with a piston diameter of 1/2 inch, a stroke of 5/8 inch and 2 cylinders. The area of a 1/2 inch piston is .196 in sq and with a stroke of 5/8 inch gives us a volume of .122 in cu per piston and since we have two pistons in our pump our total output is .244 in cu per revolution of the drive wheels. We calculated that we needed about .23 in cu to keep up with our steam requirements, so this would be an acceptable design. In general I have found that a 3/4 dia piston is about as big of a piston as should be used on a 1 1/2 scale locomotive.

The output of a hand pump can be found using the same method as was used for an axle pump. Again keep in mind that a large diameter piston makes for hard work on the part of the operator. A 1" diameter is about as large as one wants to go and still maintain a reasonable pumping effort; at 120 PSI it takes about 100 lbs of force on the piston to force water into the boiler and even with a 5 to 1 mechanical advantage on the pump handle it would take 20 pounds of force on the handle.

Next month we will continue with the injector and steam pump.

## Section 4 The Steam Locomotive (continued)

This months topic concerns the design and installation of injectors. The main advantages of using an injector for putting water into the boiler is that they operate equally well while the locomotive,is stationary or moving, and is more efficient than'mechanical pumps, as there is no moving parts. The drawback is their reliability to always work when needed.

An injector works by converting the pressure energy of the steam into kinetic energy, imparting this energy to the feed water and finally converting the kenitic energy back to a pressure higher than the boiler pressure. This is accomplished by a series of tapered cones. When an unbroken flow of fluid is moving through a closed chamber, its velocity and pressure are interchanageble as the cross section area changes.

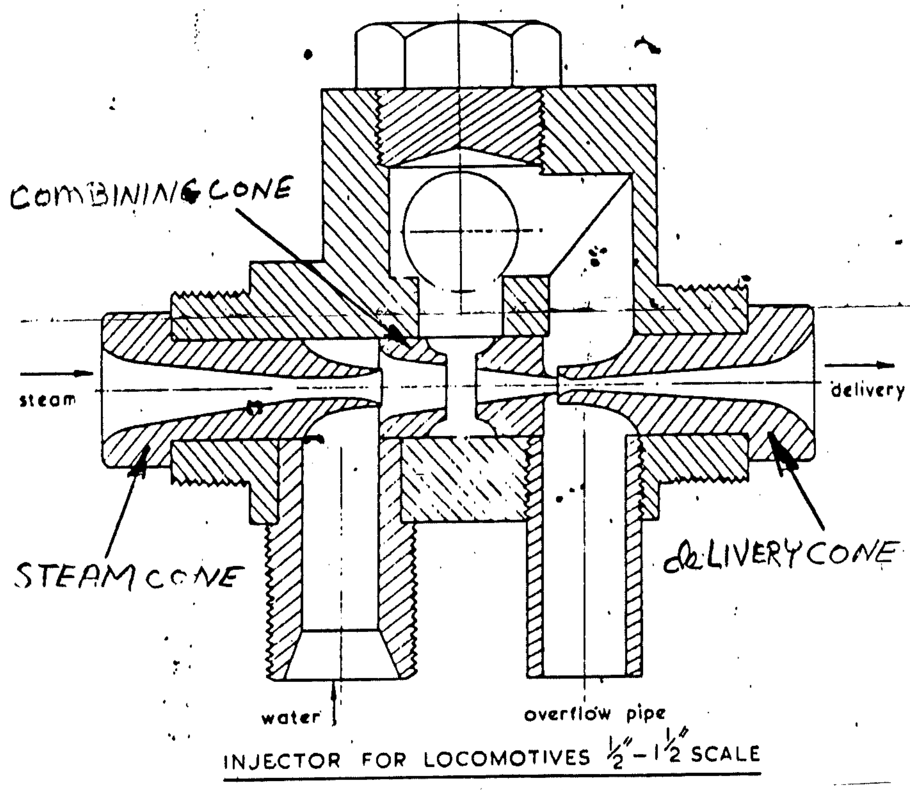



Figure 1

Figure 1 shows a typical injector as used in model locomotives. It consists of the steam cone, the combining cone and the delivery cone, each being mounted in line inside the injector body. Steam from the boiler is admitted to the steam cone while feed water is admitted between the steam and combining cones. The steam cone function is to convert as much of the pressure energy into kenetic energy. This is done by admitting steam into a cone which first converges to speed up the flow of steam and then diverges, so that the steam is expanded until its pressure is lower than atmospheric. The combining cone allows the feed water to mix with the steam again forming a high velocity flow at the end of the cone. The delivery cone then takes this high velocity low pressure flow and convert it to a lower velocity higher pressure flow, in fact convert it to a pressure higher than the boiler pressure thus allowing the water to enter the boiler. In the process the feed water is heated due to its contact and condensing of the steam.

The output of a small injector of this type can be found by dividing the delivery cones throat (smallest diameter) X 1000 squared and deviding by 40.

delivery (ounces/minute) = dia X 1000 / 40

For a delivery cone with a diameter of .060 one could expect a flow of approximately 90 oz per minute or .66 gal/min.

As one can see from above we are looking at some very small cone dimensions and that it does not take a very large piece of dirt or scale to plug up a cone and make the injector useless. This is why their reliability is not always as good as one would like. However well made the injector, the plumbing and valves are also important. It is best to mount it below the water level of the tender with the steam and delivery lines of ample size and with a minimum of sharp bends to minimize the pressure drops. Another common problem is that the injector and the feed water must not be hot or it just won't work. A leaky boiler feed check valve will prevent an injector from working.

Another factor that turns off many people is that an injector is a very precise device and is difficult to make and thus relatively expensive. Not only are the cones very small but their alignment and surface finish is very critical if they are to work.

Several books like L.B.S.C. SHOP SHED AND ROAD and Martin Evans book THE MODEL STEAM LOCOMOTIVE goes into great detail on how to build one and how to size the various cones for those who are adventitious enough to try.

Next month the steam driven water pump.

Large Scale Model Railway Engineering

## Section 4 The Steam Locomotive (continued)

The topic for this months series is the final installment of putting water into the boiler. We have discussed axle and hand pumps, injectors and this month steam driven pumps.

Steam driven pumps are the most complicated way of supplying boiler feed water, but also one of the most interesting. There is nothing like a pump hung on the side of the boiler slowly pumping away as the locomotive sits in the station awaiting its next trip.

These pumps can be divided into two basic types. The first is the single cylinder type with a single steam and a single water cylinder, and the second, the dual cylinder type with two of each. Of the two types, the dual cylinder configuration is usually considered more reliable because it can be made so that it is always self starting where the single cylinder configuration can sometimes get hung up when trying to start requiring the engineer to give it a nudge.

The enclosed drawing shows the construction of the single cylinder design vs. the dual cylinder. In the single cylinder design the piston assembly controls a small pilot valve which in turn controls the main valve that directs steam to either end of the steam cylinder. The reason this is done is that if the main valve was connected directly to the piston, as the valve moves it would eventually cut of steam to the steam piston and as soon as this happens the piston would stop moving and thus the main valve would not open the other port to reverse the piston travel and the pump would hang up. A single cylinder steam engine can solve this problem because it has a heavy flywheel to keep the engine going allowing the valve to open the other port. By using a pilot valve only a very small amount of steam is required to shift the main valve therefore causing the piston to reverse. Unfortunately this pilot valve can occasionally get hung up and not shift causing the pump to stall.

The dual cylinder pump gets around this pilot valve problem by having the valve on one cylinder controls the other. In addition to this they usually cross the ports on one cylinder so that when the piston of cylinder one reaches the top of its stroke cylinder two then is ported so that its piston is driven to the top at which time cylinder one's piston has steam admitted to the top driving its piston down. Upon reaching the bottom its valve shifts directing steam to the top of cylinder two. By doing this one can assumed that one valve is always fully open so that the pump will always start. This cross porting is usually accomplished by cross drilling the ports in the steam cylinder block.

To calculate the capacity of the pump one can use the procedure similar to what was used when calculated the output of a axle pump. If we find the displacement of the head end and the rod end of the water cylinder and multiply this total by the strokes per minute we get the displacement per minute. Dividing this by 234 cu in /gallon we get the pump output in gallons per minute.

For example a pump with a 1/2" diameter piston, a 3/16 rod diameter, a 1" stroke and operating at 200 strokes per minute would have an output of:

.196 cu in / stroke head end displacement

.176cu in / stroke rod end displacement

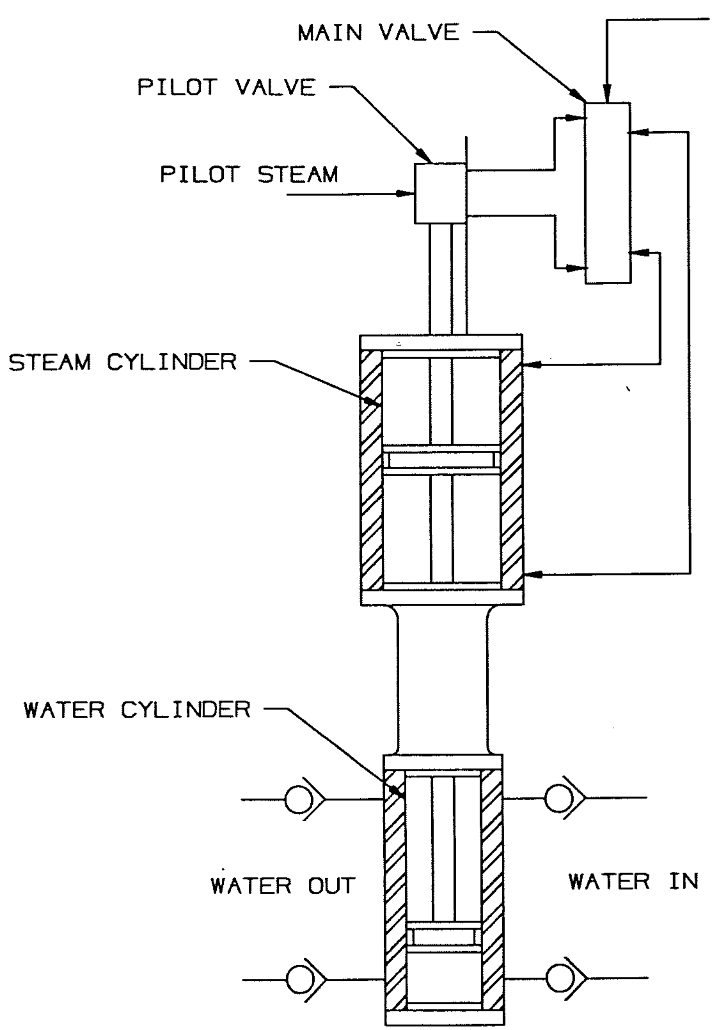
adding the two together we get the total displacement of .372cu in / stroke.

.372cu in X 200 strokes/minute = 74.4 cu in per minute

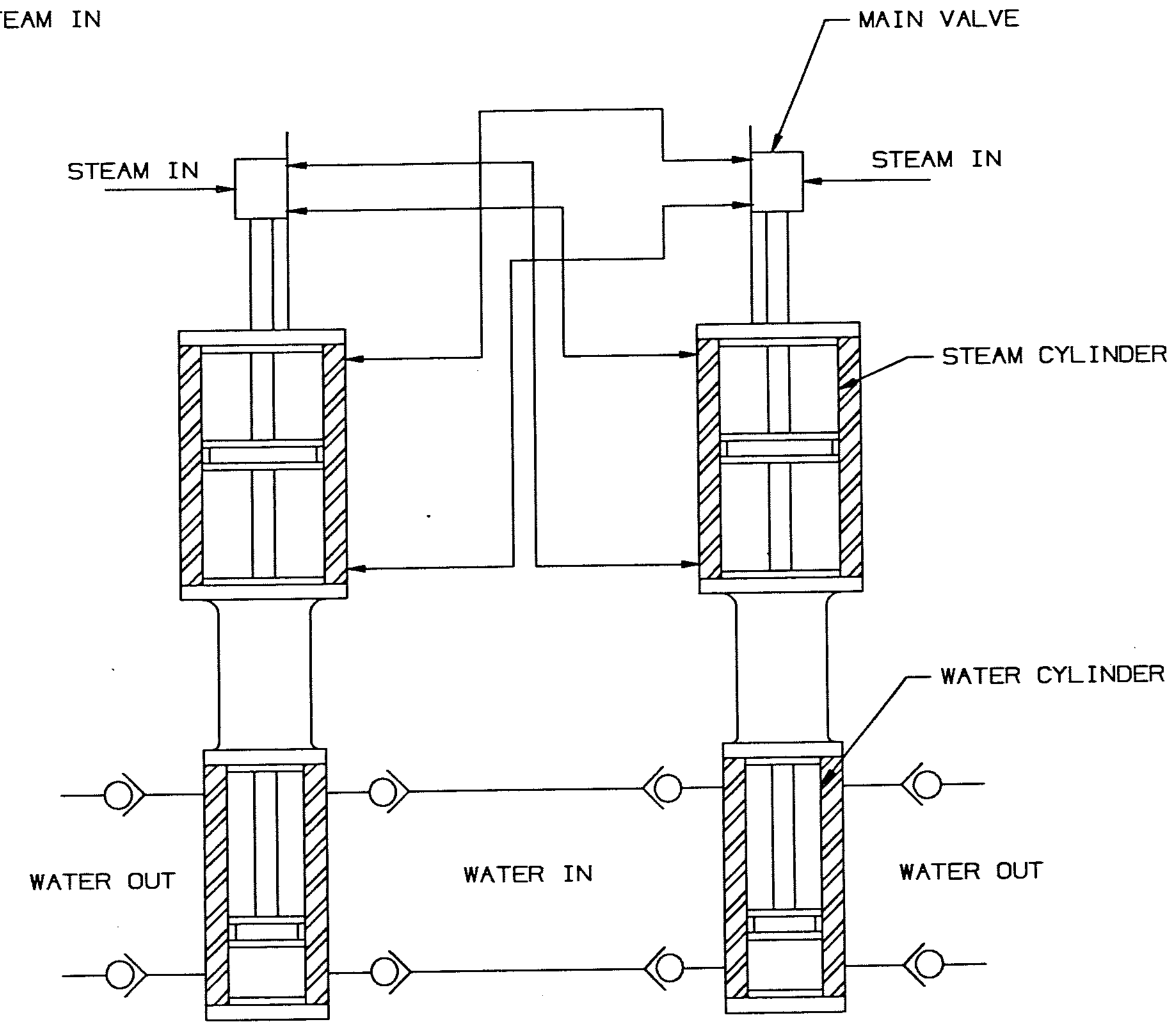
74.4cu in per minute / 234 cu in/gal = .31 gpm

Pumps usually are about 90% efficient so our pump would put out about .28 gallons per minute. Of course a dual cylinder pump would put out twice this amount.

This end our discussion on boiler feed water, at least for now. Next month we are going to talk about different kinds of bearings and how to size them.




SINGLE CYLINDER PUMP




TWO CYLINDER PUMP

**Tech Sheet** - Propane burners

Several times in the last couple of months Ken and I have been asked about the propane burners that we use in our locomotives, why they work so well and how we made them. So I am going to interrupt my engineering series this month and answer the questions that we are often asked.

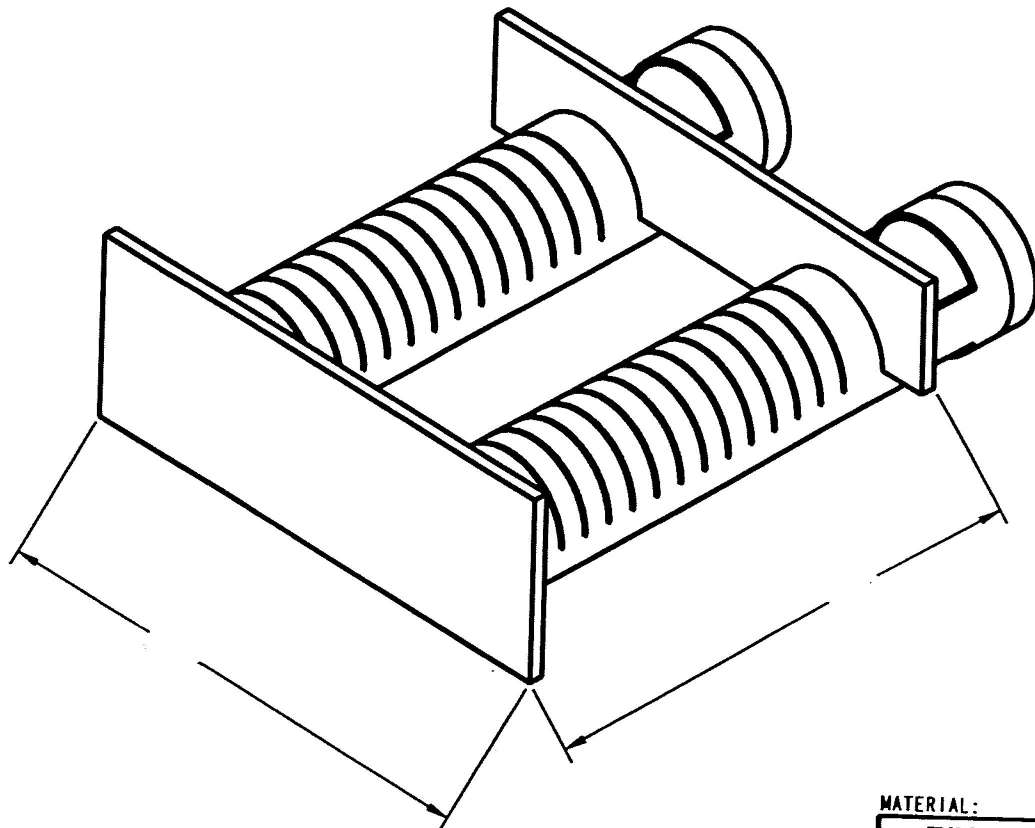
Thee drawings are included that show the construction of the burner. The first one is an isometric view of the complete burner assemble. The mounting bars are approximately the size of the inside of the fire box with the inlet end either to the front or back of the boiler. The actual mounting arrangement varies greatly and is left up to the builder to figure out how to apply it to his engine. The drawing shows two burner tubes but a very small boiler might have one and a large one three. No fire pan or ash pan is required.

The second drawing shows the top view and a side cross section view of a burner tube. The tube is made up out of four parts which are detailed in the third drawing. The four parts are the burner tube, the mixer tube, the tube head and the nozzle.

In this design the fuel enters the burner by way of the tube head and nozzle and directed down the center of the mixing tube. As the fuel exits the nozzle and enters the mixing tube air is drawn in through the two slots and is allowed to mix as the fuel air mixture passes down the tube. When the fuel air mixture reached the end of the mixer tube it enters the burner tube and flows back towards the inlet end. Slots in the burner tube allows the fuel air mixture to escape and burn. Note that the end of the burner tube must be caped (air tight) with a plug or one of the mounting bars as shown. The dimensions shown are reference only and I have made burners with smaller, and larger tubes as well as longer. The only thing to remember is that it takes some distance for the fuel and air to mix well , so there is a limit on how short a burner can be. I have found that a ratio of about 5:1 between the id of the mixer tube and its length is about the minimum for good mixing in this design.

This burner is a high pressure design requiring several psi to function properly, therefore us should use a high pressure regulator on your propane tank and use a fine firing valve on the locomotive that can be adjusted accurately. Of course proper safety precautions are required like ample ventilation for the car carrying the tank and of course safety chains between the locomotive or tender and the car caring the fuel tank.

One last word of caution, propane is heavier than air and will sink to any low area and can ignite.



DO NOT SCALE DRAWING
TOLERANCES
unless otherwise specified:
ANGLES: ±1°
DECIMALS: (3 PLACES) ±.005
(2 PLACES) ±.015

MATERIAL:

# PIKE LAKE and EASTERN R.R.

HARTFORD SHOPS

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| CHK. | DATE | DWG | |
| TITLE BURNER ASSEMBLY | | | NO. |

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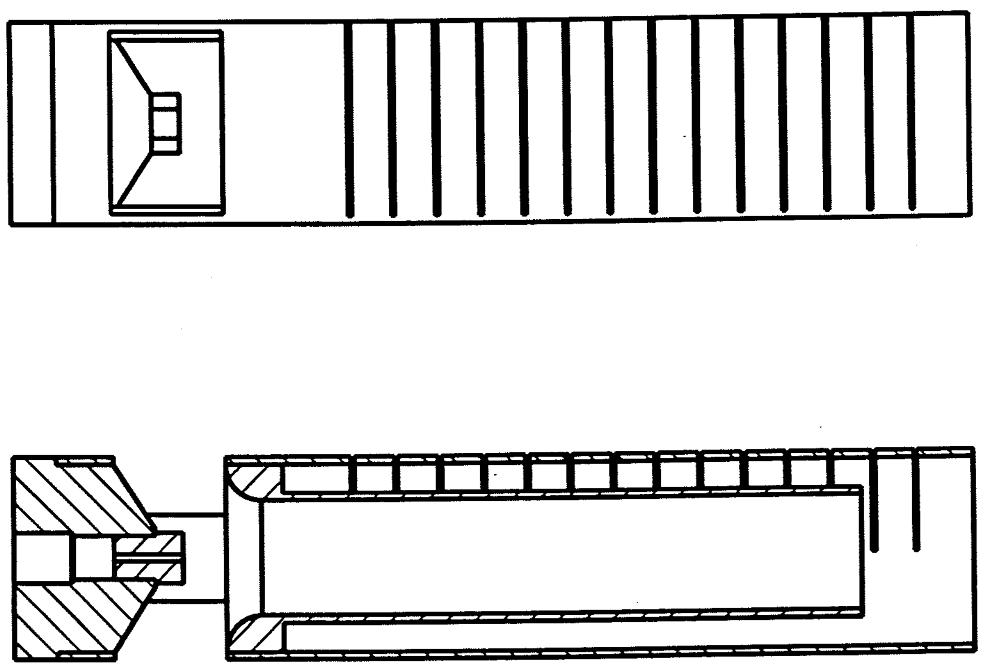

X-SECTION

DO NOT SCALE DRAWING
TOLERANCES
unless otherwise specified:
ANGLES: ±1°
DECIMALS: (3 PLACES) ±.005
(2 PLACES) ±.015

MATERIAL:

**PIKE LAKE and EASTERN R.R.**
HARTFORD SHOPS

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| DR. TA | DATE 8-26-92 | FILE | SCALE NONE |
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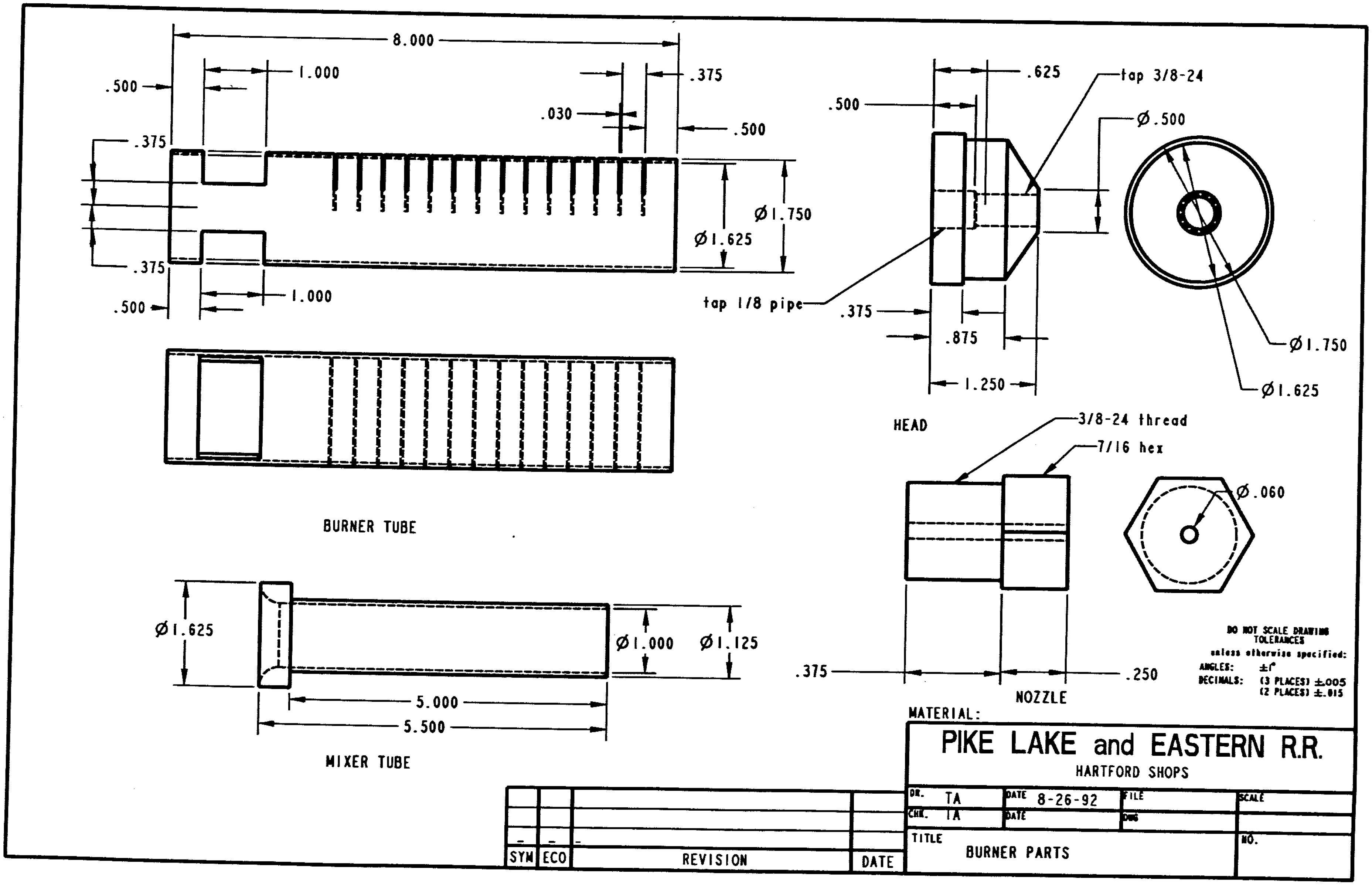
8.000
.500
1.000
.375
.030
.375
.500
.375
1.000
.500
Ø1.750
Ø1.625
BURNER TUBE
Ø1.625
Ø1.000
Ø1.125
5.000
5.500
MIXER TUBE
.625
.500
tap 3/8-24
Ø.500
tap 1/8 pipe
.375
.875
1.250
Ø1.750
Ø1.625
HEAD
3/8-24 thread
7/16 hex
Ø.060
.375
.250
NOZZLE
DO NOT SCALE DRAWING
TOLERANCES
unless otherwise specified:
ANGLES: ±1°
DECIMALS: (3 PLACES) ±.005
(2 PLACES) ±.015
MATERIAL:
PIKE LAKE and EASTERN R.R.
HARTFORD SHOPS
DR. TA
DATE 8-26-92
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BURNER PARTS
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## Large Scale Model Railway Engineering

This month Ken asked me to review the Locomotive Specification Questionnaire and explain how to fill it out. To start, many of the questions are self explanatory but a few are not and some do require some calculations. Also some assumptions are required which I will point out as we go.

The first item I want to look at is the driving wheel diameter. Wheels are measured over the tread and not over the flanges and if the tread is tapered the measurement is usually taken at the tangent point of the tread and the fillet radius between the tread and flange.

The wheelbase can be measured in several ways. The most important is the rigid wheelbase which can be defined as the length from the center of the first axle affixed to the frame to the last axle that is affixed to the frame. The overall wheelbase is the length from the first axle of the pilot truck to the last axle of the trailing truck. On our specification sheet the driving wheel is the rigid wheelbase and the engine wheelbase is the overall wheelbase.

Boiler diameter is defined as the outside diameter of the boiler barrel and the pressure is the pressure that the relief valves open.

Now we look at the tractive effort and Indicated Horsepower which both require some calculations. The tractive was covered a while back when we talked about locomotive design but to review, it is found by using the following formula:

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

where T= tractive effort
- d = diameter of cylinder in inches
- P = average steam pressure (mean effective pressure)
- S = stroke in feet
- D = diameter of drive wheels in feet

As one recalls from section 4 the mean effective pressure (MEP) is a function of the valve cutoff, which for a typical locomotive might be 75%. With a valve cutoff of 75% our mean effective pressure would be .96 times the inlet pressure to the cylinder. To fill out the specification sheet we will assume a 15% pressure drop between the boiler and the cylinders and a valve cutoff of 75%, unless you know the actual values for your locomotive. If your relief valves are set to 125 PSI. your mean effective pressure would be :

125PSI x 85% X .96 which would give you 102 PSI MEP.

This value can be plugged into the tractive effort formula to calculate the maximum tractive effort that can be generated. However as we talked about back in section 1 the maximum tractive effort is affected by the factor of adhesion which is a function of the weight on the drive wheels. In filling out the specification sheet we will use the calculated tractive effort.

The indicated horsepower is the maximum hourse power that can be generated at a given speed. It can be found by the formula:

$$HP = \frac{P \times L \times A \times N}{33000}$$

where P = mean effective pressure (same as above)
L = stroke of cylinders in feet
A = area of piston in square inches
N = number of working strokes per minute

In order to find a value for N we have to assume a locomotive speed and work backwards using the wheel diameter to find the wheel rpm and then the strokes per minute. To make things simpler we will use 200 RPM for our axle speed and 4 piston strokes per revolution of the drivers for a total of 800 strokes per minute ( N = 800).

For reference, the area of an rectangular area is the length times the width, the area of a circle or piston is 3.1416 times the radius squared while the circumference of a circle is its diameter times 3.1416. To convert inches into feet divide the inches by 12.

This should cover most of the information one needs to fill out the M.L.E.S. Locomotive Specification Questionnaire. What I presented covers non geared type locomotives and since the only geared locomotive currently owned by a member is my Climax I will not complicate this by covering the effects of gearing on the above calculations.

If any one has any questions or need help in filling out their sheet please let me know and I will be glad to help.

**MILWAUKEE LIGHT ENGINEERING SOCIETY**
**R A I L R O A D   D I V I S I O N**
Membership Steam Locomotive Specification Questionnaire

Name of Railroad: ____________________

Name of Builder: ____________________

Name of Owner: ____________________

Serial No.: __________ Years Built ____/____/____ to ____/____/____

Type Name: __________ Wheel Arrangement: __________

Road No.: ________ R.R. Class: ________ Builder Class: ________

Prototype Builder & Railroad: ____________________

Scale Built: ______ Gauge of Track: ______ Overall Length (E+T): ______

| DRIVING WHEEL Diameter | WHEEL BASE Driving | WHEEL BASE Engine | WHEEL BASE Engine & Tender | CYLINDERS Diameter | CYLINDERS Stroke |
|---|---|---|---|---|---|
| ______ | ______ | ______ | ______ | ______ | ______ |

| FUEL Kind | BOILER Diameter | BOILER Pressure | FIREBOX Length | FIREBOX Width | TUBES & FLUES Number | TUBES & FLUES Diameter | TUBES & FLUES Length |
|---|---|---|---|---|---|---|---|
| ____ | ______ | ______ | ______ | ______ | ______ | ______ | ______ |

| AVERAGE WEIGHT IN WORKING ORDER, POUNDS: On Drivers | Lead Truck | Trailer Truck | Total Engine | Tender 2/3 load | GRATE AREA, Sq.in. |
|---|---|---|---|---|---|
| ______ | ______ | ______ | ______ | ______ | ______ |

Tractive Effort __________ Indicated Horsepower __________

HEATING SURFACES, Square Inches

| Tubes & Flues | Arch Tubes & Syphons | Firebox & Comb. Chamber | Total | Super-heater |
|---|---|---|---|---|
| ______ | ______ | ______ | ______ | ______ |

TENDER

Type: __________ Number of Wheels: ____

CAPACITY: Water ________ Gallons Fuel ______ Lbs.