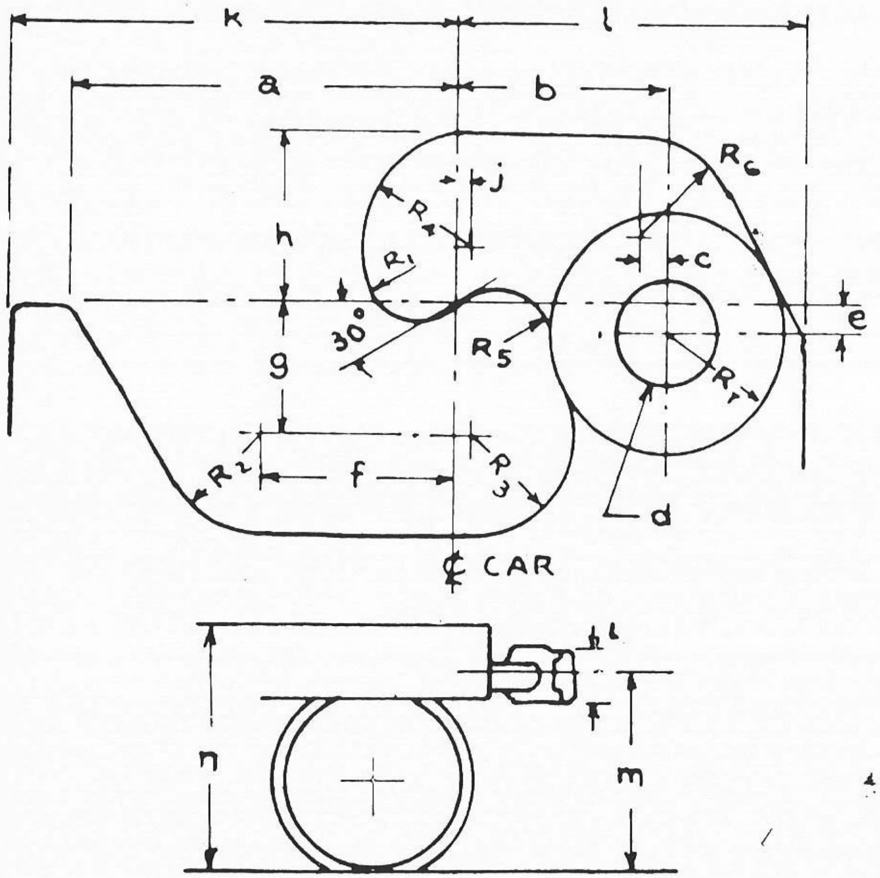
TABLE B Automatic Coupler Standards

REVISED 4-13-63

GOLDEN GATE LIVE STEAMERS, INC.



From a safety standpoint the most important dimensions are the 30^0 angle, R_1 and R_5 designed to give a good "hook" to the knuckle.

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

Wheel and Track Gauging

The subject of wheel gauging would not need to be discussed except it has become somewhat complicated in 1-1/2" and larger scales. For example, in 1-1/2" scale there are two accepted gauges: 7-1/4" and 7-1/2". In 3" scale there are three gauges: 14", 15" and 16". In 4" scale there appears to be four gauges in use: 18", 18-1/2", 18-3/4" and 19". Above 4" scale there appears to be standard gauging at 24", 30" and 36". The purpose of this article is not to take sides, but just highlight where each standard operates.

1-1/2" Scale:

The true gauge for 1-1/2" scale would be 7-1/16", however this is not a generally used gauge. The two predominate gauges are 7-1/4" and 7-1/2" and these differences are primarily regional. The following is the list of regions where each gauge predominates:

7-1/4" Gauge	7-1/2" Gauge
Northern Eastern USA	Mid Atlantic USA
Eastern Canada	Southern USA
England	Mid Western USA
Europe	Western USA
Australia	Western Canada
	Japan

Due to the "oversize" track of 7-1/2", some people in 7-1/2" gauge regions have begun modeling in 1.6" scale and call this "Finescale" 1-1/2" scale.

3" Scale:

The true 3" scale track gauge would be 14-1/8", however as in 1-1/2" scale, three gauges tend to predominate: 14", 15" and 16". There does not appear to be any predominant gauge regionally; it is more a function of the builder. The most common gauge appears to be 15",, though it is difficult to actually place numbers on this.

4" Scale:

The true 4" scale gauge (one-third full size) would be 18-5/6". Again, there appears to be a mixture of gauges: 18", 18-1/2", 18-3/4" and 19". No gauge appears to predominate here as in California alone 18", 18-3/4" and 19" gauge railroads exist.

Truck Design

The design of scale trucks must be carefully considered due to some unique loading characteristics.

Specifically, a pair of 1-1/2" scale trucks when running light may have a 50 lb. car riding on top of them. Later on, they may have four adults weighing up to 850 lbs. for a total of 900 lbs. between riders and car. This 18 to 1 ratio in weight means that the trucks must operate well both at 50 lbs. and 900 lbs. without derailing. As a comparison, a full scale car such as a gondola or hopper may weigh 40,000 lbs. and fully loaded may weigh up to 150,000 lbs. which is a ratio of under 4 to 1.

Scale trucks, therefore, must be flexible so they will run light without derailing on uneven track. If they are sprung they must be rugged enough to handle up to 450 lbs. per truck (900 lbs. per pair). They must be free rolling both at light and full weight, their wheel contours must be designed to reduce friction on curves, and they must be mounted to their cars so that they are flexible when running light on uneven track, yet avoid "wobbling" as so often happens in model railroad cars. Specific suggestions to meet each of these problems follow:

Truck Equalization & Springing

The most flexible truck is one that is equalized **without** springs, e.g., the bolster is loosely attached to one side frame and the bearings are loose. However, an unsprung truck is very uncomfortable to ride on. Therefore, if they are sprung, the springs when fully extended (car running light) must exert very little pressure between the bolster and the side frame. This flexibility allows the truck to handle any track irregularity with ease.

Truck Bearings

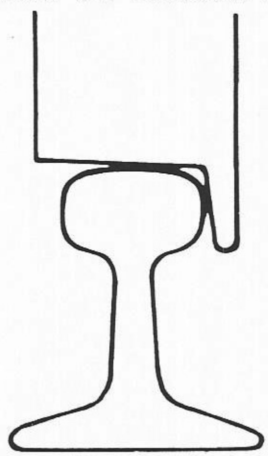
A truck when fully loaded with passengers must now be rugged enough to handle 450 lbs. per truck. First, the journal bearings must be able to handle the weight. The most common bearing used is a ball bearing which is free rolling. These are capable of carrying much heavier loads for the same axle diameter than bronze bearings. For example, a 7/8" diameter bronze bearing can carry a load of 314 lbs. where a ball bearing of the same size can carry 700 lbs. Therefore, smaller ball bearings may be used as the most a particular bearing would carry is estimated to be 150 lbs. in a four wheel truck. It is our opinion that ball bearings are most preferable, their rolling characteristics are good and if sealed resist dirt well. We recommend 3/8" ball bearings as they will support at least 200 lbs. at normal running speed. If space is a limitation for ball bearings, then teflon bearings are our next preference as they are reasonably free rolling and are not affected by dirt. Our last preference are either needle bearings and bronze bearings. Needle bearings, though free rolling, are difficult to install and align, during operation can "cross" roll and jam, and if not well sealed are affected by dirt. Bronze bearings, though low cost and easy to install, do not roll well, must be continuously oiled, and are affected by dirt.

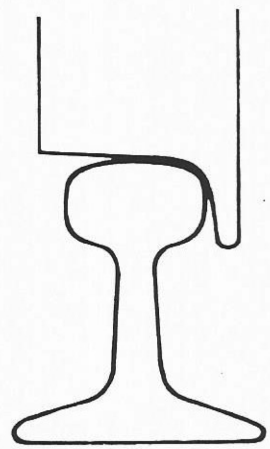
Truck Spring Planks

Next, as part of ruggedness on sprung trucks, a good design characteristic is to install a spring plank across the bottom of the truck for the springs to seat in. This plank helps to keep the side frames vertical, and if these trucks are to experience considerable weight, this plank should be loosely pinned to the side frame, again to even more insure verticle stability. Even though this was not done on many real freight car trucks, the lighter weight design of scale trucks benefit from this spring plank.

Truck Wheel Contours

The number of cars a locomotive can haul on level track is a function of truck friction with train weight becoming an additional factor. On truck friction, this is primarily a function of two characteristics: first being bearing friction and second being wheel contours on curved track. Therefore, the finest running bearings practical should be selected, and this was covered in the prior sections. If bearings do have significant friction, this friction is increased with more weight, again no surprise. However, the subject of wheel contour on friction is often overlooked.


Flange with
no fillet


Flange with
fillet

When a truck runs on a straight track, it generally runs in the center of the track if it is not banked. However, when the truck enters a curve the rail must "steer" the wheel to turn the truck. If the wheel flange is straight up, it tends to rub against the rail, creating friction as shown in the no fillet flange example. If the flange fillet is well rounded where it meets the tread, the wheel starts to slowly climb the radius. The act of "climbing" tends to "steer" it back before the flange ever makes contact with the rail. This is easily observed by rolling two trucks around a curve, one with a flange, one with no fillet on the flange, and one with a curved fillet leading into the flange. You will hear the scraping with the straight flange indicating increased friction, and you will not hear it with the flange having a good fillet.

Truck Mounting

Finally particular attention must be paid to how trucks are attached to cars. The reason is that if they are not attached properly, the car will either "wobble" or be "too stiff". Trucks should be attached to cars using body bolsters that have outboard bearing surfaces to mate with the bearing surface on the truck bolster. On one truck the bearing blocks should nearly touch on both sides. This will keep the car from wobbling. On the other truck there should be a gap of 3/32" between the bearing blocks on each side, only the pivot section of the car bolster touches the truck bolster. This allows for track variations when the cars run light. The approach is like a three legged milk stool. On one truck the "two" legs are the bearing blocks, and on the other end the "one" leg is the pivot section only.

A SIMPLE TRAIN BRAKE SYSTEM

By Chet Peterson

In an emergency, can I stop my train? This question had plagued me many times. Discussions concerning this problem were frequent and many contemplated systems were outlined in detail by various operators. I wanted my own cars and it seemed that brakes should be a basic part of the design.

With all the talk in various Live Steam quarters I had never seen anyone put brakes on cars and then make them work down through a train. It was apparent that the complications of installation and maintainability were the deterent.

One day during a visit with Carl Purinton in the East he showed me a system he used. The concept was borrowed directly from English railroads and was so basic and straightforward that I made an immediate decision to use it. The installation was a simple vacuum system where the train line was bled down, on application of the brake valve, by a steam ejector under the cab. Each car had one or two vacuum cylinders which through linkage, pulled the brake shoes in against the wheels. Carl even used this on his engines.

The ejector I used turned out to be very efficient and small enough for even 1" engines. The design is copied directly from an article in May, 1966 Model Engineer. If made carefully it will suck a train line down to 27 inches of mercury (approx. 13 lbs. per sq. inch) at 110 lbs. per sq. inch steam pressure. Tests made with a five car train (60 ft. train line) proved the effectiveness. It took one and one-half seconds for this tiny ejector to evacuate the entire train line (3/16" dia. tubing) and have the brakes fully applied. A ball valve was used on the steam inlet to the ejector to get full application in a quarter turn of the brake handle. The exhaust steam from the ejector is overboard. The assembly of the three parts of the ejector can be made with soft solder.

Connections between cars are merely soft rubber hose 5/16 OD by 3/16 ID. These are merely wetted and slipped over standard 3/16 copper tubing at the end of the cars. No clamps are necessary because we are dealing with a vacuum and not a pressure.

The vacuum cylinders or cylinder can be mounted either on the bottom of the car or directly on the trucks. I chose to mount directly on the trucks with two cylinders on each truck and shoes against the four outboard of the six wheels. The cyclinder is mounted so that the cylinder actuating arm pulls upward on application. This pulls up on the long arm of a bell crank. The short arm of the bell crank pulls back to bring the shoes in against the wheel. The cylinders used were obtained at an automotive store and are Ford distributor vacuum advance cylinders made by Delco-Remy, Part No. F-1302. They are 3 inch diameter giving 7 sq. inches of diaphragm area which exerts approximately 90 lbs. pull. This in turn, must be be reduced by a diaphragm efficiency factor of .70 resulting in a diaphragm force of 63 lbs. Multiplying this by two and a half for the advantage through the bell crank results in approximately 80 lbs. force against each of the two shoes it actuates.

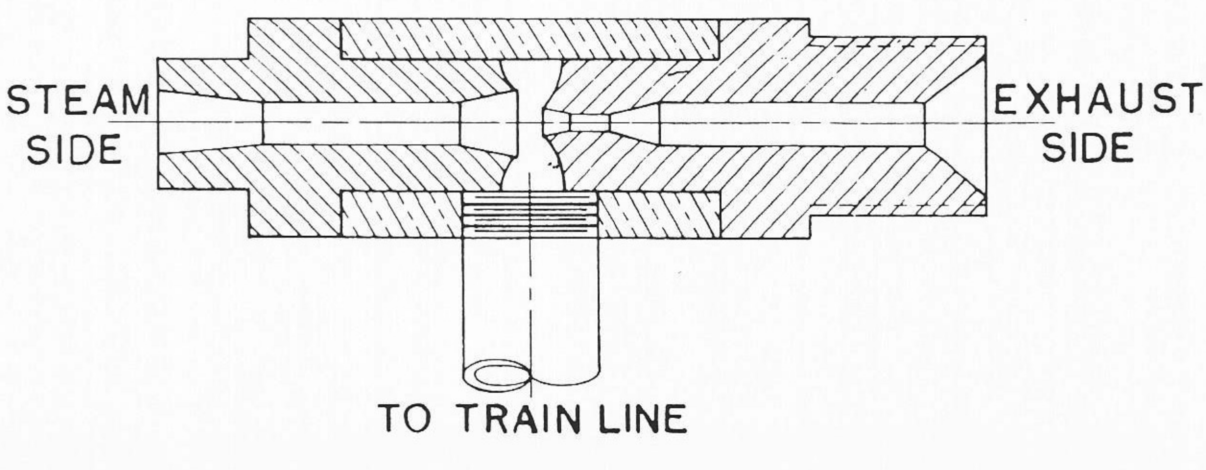
On a four wheel truck, one cylinder per truck should do the job fine by increasing the bell crank advantage to come out with the same force against the shoe. My cars were designed to weigh out totally loaded at 1000 lbs. (four heavy adults) so the forces were calculated to obtain effective braking without sliding the wheels. Apparently the calculations were fairly accurate because at loads of less than 500 lbs. (4 children) the wheels do slide. It is suggested that a car weighing 600 lbs. total (3 adults) should have approximately 70 lbs. per wheel.

In conclusion, when you throw the train brake handle 90 degrees, a ball valve opens and lets steam pass through the ejector. This unit, acting as a venturi, sucks the air out of the train line and the vacuum cylinders on each truck pulling the shoes in against the wheels and applying the brakes. When the brake handle is returned to "Off", the line vacuum bleeds back to atmosphere through the ejector and the brakes are immediately released. The system is easily installed, positive in operation and so simple as to require almost no maintenance. Occasional take-up on the brake linkage clevis to allow for shoe wear is about all that is necessary.

A refinement of the above system would include an arrangement to allow constant operation of the ejector with constant vacuum on the train line holding the shoes away from the wheels. Interruption of this vacuum by operation of the train brake handle or a break in the train line would immediately apply the brakes. This refinement was decided against for the following reasons:

1. More complicated design and installation.
2. Substantially increased maintenance and service attention to attain equal reliability.
3. Experience has proven that to brake a complete train to a quick stop is a big step forward in safety and the small added feature of insuring against a broken train line is difficult to rationalize in the interest of simplicity.

Even if you never have to make a panic stop, your effort will all be repaid the day you throw the train brakes on for the first time and drag your train into the station with the exhaust barking to the last "chuff."



LOCOMOTIVES PRINCIPLE TYPES OF STEAM LOCOMOTIVES

WHYTE CLASSIFICATION

SWITCHING LOCOMOTIVES

0—4—0	OO	Four-wheel Switcher
0—6—0	OOO	Six-wheel Switcher
0—8—0	OOOO	Eight-weel Switcher
0—8—2	OOOOo	
0—10—0	OOOOO	Ten-wheel Switcher
0—10—2	OOOOOo	Union

TANK LOCOMOTIVES

0—4—0T	OO	
0—4—2T	OOo	
0—4—4T	OOoo	Forney
0—6—4T	OOOoo	Forney
4—4—2T	ooOOo	
4—6—4T	ooOOOoo	Baltic
4—6—6T	ooOOOooo	

TWO-WHEEL LEADING TRUCKS

2—4—0	oOO	Four-coupled
2—4—2	oOOo	Columbia
2—6—0	oOOO	Mogul
2—6—2	oOOOo	Prairie
2—8—0	oOOOO	Consolidation
2—8—2	oOOOOo	Mikado
2—8—4	oOOOOoo	Berkshire
2—10—0	oOOOOO	Decapod or Russian
2—10—2	oOOOOOo	Santa Fe
2—10—4	oOOOOOoo	Texas

FOUR-WHEEL LEADING TRUCKS

4—4—0	ooOO	American
4—4—2	ooOOo	Atlantic
4—4—4	ooOOoo	
4—6—0	ooOOO	Ten-wheeler
4—6—2	ooOOOo	Pacific
4—6—4	ooOOOoo	Hudson
4—8—0	ooOOOO	Twelve-wheeler
4—8—2	ooOOOOo	Mountain or Mohawk
4—8—4	ooOOOOoo	Northern or Niagara
4—10—0	ooOOOOO	Mastadon
4—10—2	ooOOOOOo	Southern Pacific or Super Montain
4—12—2	ooOOOOOOo	Union Pacific

ARTICULATED LOCOMOTIVES

0—6—6—0	OOO OOO	
0—8—8—0	OOOO OOOO	
2—6—6—2	oOOO OOOo	
2—6—6—4	oOOO OOOoo	
2—6—6—6	oOOO OOOooo	Allegheny
2—8—8—0	oOOOO OOOO	
2—8—8—2	oOOOO OOOOo	
2—8—8—4	oOOOO OOOOoo	Yellowstone
2—10—10—2	oOOOOO OOOOOo	Virginian
4—6—6—4	ooOOO OOOoo	Challenger
4—8—8—2	ooOOOO OOOOo	Cab Forward
4—8—8—4	ooOOOO OOOOoo	Big Boy-World's Largest Locomotive

SPECIAL ARTICULATEDS

2—8—8—2	Beye - Garratt
4—8—8—2	Cab Forward Articulated

OIL FIRING YOUR LOCOMOTIVE

By Chester G. Peterson

By taking into consideration firebox size, boiler capacity, and tube length, there is a definite breakover point where oil firing should be considered. This is, of course, directly related to the size of the engine. Engines smaller than 1-1/2" scale Atlantics or Moguls may have combustion problems. However, starting with this size and progressing on up in the 1-1/2" scale, the operational and economic advantages increasingly outweigh either Liquid Petroleum Gas (LPG) or the coal method. Operationally, the following considerations are important:

1. Complete control of the fire. Simple light off and adjustment of the fire at will for heat level and shut down if necessary. This is very useful where you may be pulling a grade with a heavy load and a few minutes later, be idling in the station.
2. On a large engine, the grate area is rather extensive, and the firedoor rather small for efficient coal firing. The engineer could be quite busy (with a small shovel) keeping up with the fire and effectively distributing it over the grate.
3. Oil (or LPG) firing requires minimum attention while running, and it should be remembered that on a full size locomotive there was always a fireman to take care of such duties as firing and water level, leaving the engineer to concentrate on the track and signals. Many of the accidents on our Live Steam tracks are a direct result of the engineer having his "head buried in the cab" attending to what in full scale, would be fireman's duties.
4. Logistics: A couple of five gallon cans of Diesel fuel will give all the running that most of us can handle for the day in even the largest engines.
5. Safety: Diesel fuel is extremely difficult to ignite unless it is atomized.
6. Cleanliness: If anything in the way of exhaust amounting to more than a very light haze is noticeable, or if the boiler tubes or smokebox have to be cleaned frequently, either the oil burner nozzle is at fault, by dripping and not supplying a vapor haze, or the engineer is not on the ball to adjust for efficient oil-to-air ratio; this author brushes out his boiler tubes about every 100 hours of running.

Economically, the advantages of firing over either LPG or coal are startling in the larger engines. An example is an engine in 1-1/2" scale, 4–8–4 class (which this author is familiar with), and assuming a given load (4 tons) operating for one hour. The heat required to keep steam up is 312;000 BUT/hr. Diesel oil is rated at 19,500 BTU/lb. and costs 2.1¢/lb. LPG is rated at 21,000 BTU/lb. and costs 7¢ per lb. and coal is rated at 15,500 BTU/lb. and costs 6.5¢/lb. (West Coast). By dividing the required heat value of 312,000 BTU by the heat values per pound of the three fuels, you have the number of pounds of each fuel used per hour. Then the real shock comes when one multiplies these values by their respective costs per pound. It is evident that for this application, it would cost you three times as much per hour to use LPG as for oil, and nearly four times as much for coal. A day's run, of say five hours, could look like this:

Oil	80 lbs.	x 2.1 =	$ 1.68		1966 figures
LPG	73 lbs.	x 7.0 =	5.11		when article
Coal	100 lbs.	x 6.5 =	6.50		was written.

It is' to be acknowledged that there will be all kinds of minor variables, but they will not have any major effect on the above figures.

The oil firing system is relatively straight forward. Oil flows by gravity from the tender tank through a filter and tube to the oil control valve under the cab. The oil control valve (needle valve) meters the oil that flows to the burner nozzle where high pressure steam atomizes it to allow combustion. The steam supply to the burner nozzle is metered by a valve in the line from the boiler. Draft into the firebox is normally achieved by a center firepan damper door just ahead of and below the firebox door, and a small rectangular vent toward the back of the firepan on either side. It is a good practice to line the firebox floor and sides with about one half inch of fire clay. The burner is positioned through the center of the front of the firepan pointing level over the clay and directly back at the base of the firebox door. The draft vents, being at the back of the firepan, tend to roll the body of the flame up and turn it back along the crown sheet and into the boiler tubes. If the draft vents are adequate and the stack draft is good, very little flame will come out the firebox door, even when it is open.

A discussion of the various components in the system is in order. If certain fundamental facts are kept in mind, optimum steaming will result.

The tender oil tank should be approximately five gallons in size, and it is very important to keep it as high as possible to insure maximum head for gravity flow. It should also be accessible for occasional cleaning.

A good automotive or aircraft type filter should be in the fuel line between the tank and the needle valve. It should be of sufficient size to allow reasonable periods between cleaning or replacement, and, for this reason, should be readily accessible.

The efficiency of this filter can be improved if reasonable care is used when filling the tank; this is where a funnel and a good felt strainer pays off.

The oil control valve should be a needle with a fairly long taper to allow fine adjustment of the flow. It can be mounted under the cab with the control handle accessible up in the cab.

From the oil control valve, the line goes to the burner nozzle located in the front of the firepan. It should be noted that all of the oil line from the tank to the burner should be at least 1/4" diameter, and preferably 5/16" to insure adequate free fuel flow.

The steam atomizing line should be at least 1/4" diameter and have a good valve to allow control of atomization. It can come from the valve mounted on the boiler backhead and on to the burner nozzle.

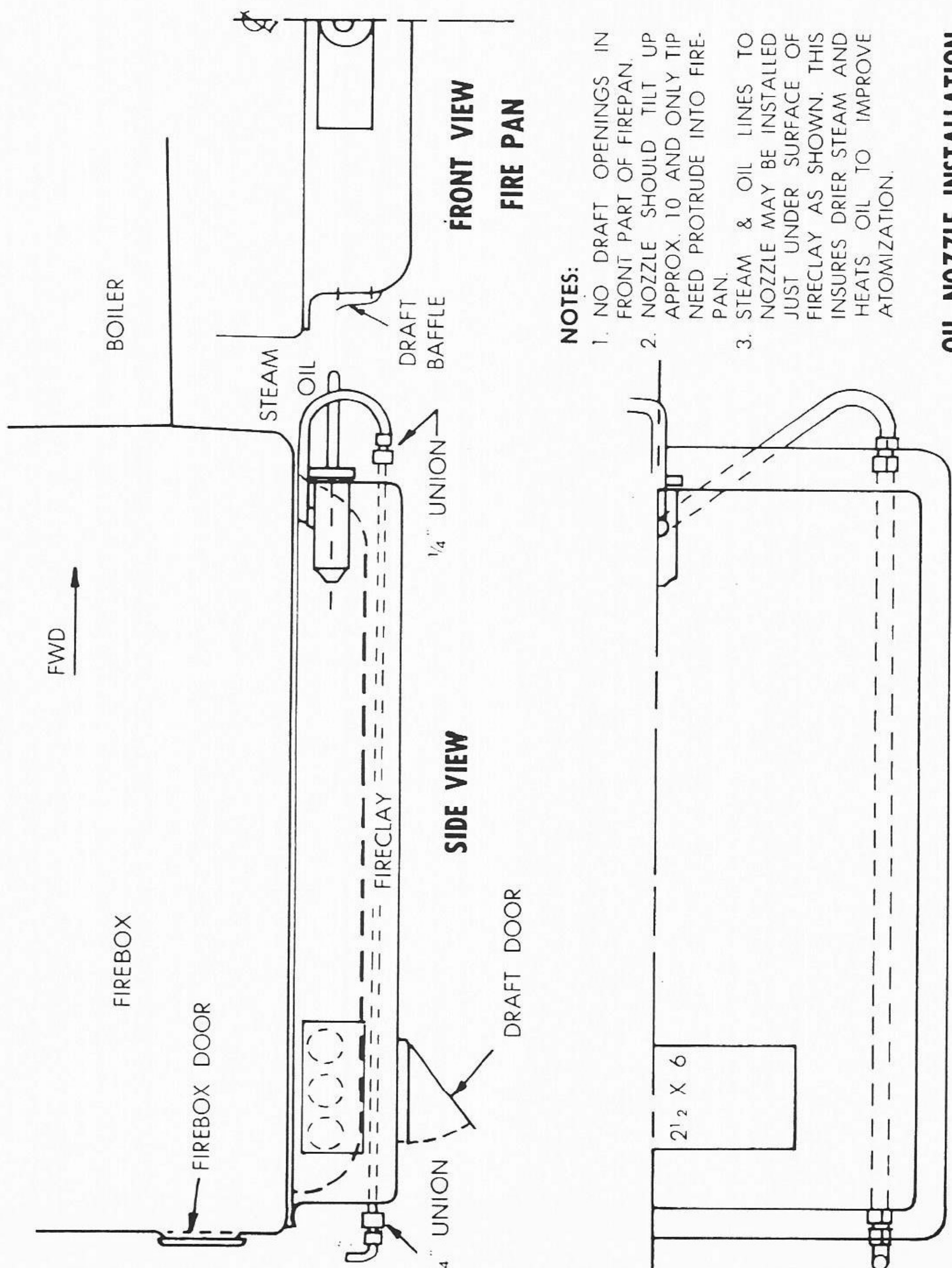
The burner nozzle is the heart of the system and is very important to good combustion. This writer has tried many different types and has found the simple one illustrated with this article to be the most reliable and efficient. The concentricity of the oil nozzle tip in the steam nozzle opening is important because if it is not concentric it will drip.

For the average size firebox, one burner-nozzle should suffice, but it may be gauged to a twin or even a three-nozzle assembly manifolded together with a single oil line and a single steam line split just behind the burner to feed the oil and steam in evenly. In this writer's Northern (1-1/2" scale), a two-jet worked very well but now, a three-jet burner fills the firebox completely with fire, even when turned down to two-thirds capacity. When turned up full for heavy loads or for quick steam it scares one a little.

The lighting off procedure is usually quite simple. Open the blower valve to provide good draft. Then open the atomizing valve partially followed by opening the oil control valve gradually. By looking in the firebox door, a grey haze can be seen which signifies that the atomized vapor is ready for light-off. A light-off rod made from wire about eight inches long with a wick of asbestos on the end is now important. Dip the wick in fuel, light it, and thrust it through the firebox door into the grey haze. When the firebox is cold on initial light-off, it may take a little patience because the draft tends to suck the wick fire out before the burner fire gets going.

Occasionally, during operation, because of quick throttle opening, or high water, the fire may suck out and have to be relighted. Some operators use a pilot wick fed from a little can of alcohol installed below the firebox with an asbestos wick leading up through a hole in the center of the firebox. If the main fire is snuffed out, this pilot light reignites it. This writer uses a different system which also works well. A common automotive spark plug is installed through the bottom of the fire pan so that it sticks up midway between the nozzles and the firedoor. A Ford Model "T" coil and a small 6-volt battery is carried in the tender. If the fire should suck out, a touch of the starter switch on the tender panel makes for instant fire. However, if one tarries more than a few seconds, the combustible build-up in the firebox may reach a level which will give one a surprise (and a black face) when he does get around to pushing the button. It isn't all bad, though, because the engine will have clean tubes for a while.

In summary, the use of oil firing on larger engines makes for greater enjoyment and safer operation. There is enough to do after you set your fire and forget it. Then you can concentrate on your waterglass and the track ahead.



OIL NOZZLE INSTALLATION
TYPICAL FIREPAN

HORSEPOWER

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

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

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

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

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

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

Now filling in our formula it looks like this:

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

TRACTIVE EFFORT

The tractive effort is found from the following formula:

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

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

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

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

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

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

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

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

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

CYLINDER, PORT SIZES AND STRENGTH OF BOLTS

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

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

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

In our example d = 2
Pb = 100

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

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

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

BOLT SIZES

In model engineering work small hexhead 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, we prefer not less than eight. More can be used if you desire. Now that we know the number of bolts we are going to use, we can easily determine the size. As our cylinder is of 2-in. bore we have a total area of 3.1416 square inches of area x 100 boiler pressure which totals 314 or call it 315 pounds push against the cylinder head. Now

by dividing 315 by the number of bolts, 8, we have a load of 40 pounds per bolt. Looking at Table IV we find that a bolt which can carry a load of 40 pounds corresponds to a 4-40. Therefore 8, 4-40 bolts will do the job.

In formula style it appears:

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

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

AREAS OF STEAM PORTS AND PIPES

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

The formula for piston speed in feet per minute is:

$$PSFM = RPM \times 2L$$

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

$$PSFM = 600 \times 2 \times .2083 = \text{feet per minute}$$

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

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

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

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

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

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

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

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

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

PISTON ROD SIZE

DIAMETER OF PISTON ROD

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

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

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

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

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

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

CROSSHEAD GUIDE

THRUST OF THE CROSSHEAD AGAINST THE GUIDES

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

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

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

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

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

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

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

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

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

DRIVING AXLE JOURNALS, DIAMETER AND LENGTH

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

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

Pb = Boiler pressure
A = area of piston

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

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

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

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

CRANKPIN 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 x ½ = ½ square inch. To determine the projected area we take the boiler pressure in pounds per square inch and multiply it by the area of the piston and divide the product by 1000. Written as a formula it looks like this:

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

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

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

A good ratio between the diameter of a bearing and its length is between 1:1 to 1½:1. In our example we will use a ratio of 1-1/8 to 1 so our problem now looks like this:

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

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

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

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 piston and steam pressure to give us the load we can substitute these values by assuming we will carry 6 adults at 150 pounds on a car. This gives us 900 pounds plus the weight of the car. Assuming this to be 100 we therefore have a total load of 1000 pounds to be carried on 8 journals. The length should be made 1¼ x the diameter.

Comments on different bearing types are covered under Trucks.

BOILER CONSTRUCTION

The design of model locomotive boilers is limited to specific outside dimensions to conform in scale to the general outline of the prototype. The calculations given are for determining the strength, grate area, tube area, stack size, exhaust nozzle area and general strength requirements.

The most common material used in model locomotive boilers is copper. In general this material has excellent corrosion resistance, heat transfer and fabrication qualities. However, its strength is rather low, especially at elevated temperatures. Joining the parts together is accomplished by silver solder, brazing, riviting or welding. With the proper hard solders joints can be made equal to the parent material strength. However, this requires joints to be fitted to a maximum of .003-in. clearance. Increased clearance in spaces between parts greatly reduces the joint strength. We feel that in figuring the strength of joints only 50% of the strength of the material should be used in calculations.

The average tensile strength of copper in the soft or annealed state is 30,000 pounds per square inch. Steel used in boilers has a tensile strength of 65,000 pounds per square inch.
Copper sheet can easily be shaped around forming blocks by pounding. During this working of the copper it work hardens and will crack. To eliminate this the part should be heated to a dull red heat and quenched in cold water. This process will soften the copper so it can be further worked without cracking. Steel should be worked while it is a dull red in heat. This makes the fabrication more difficult. However, the steel may be easily welded. All steel boilers that are welded should be stress relieved after final welding to eliminate highly stressed areas and possible future failure. Many model engineers are afraid of steel boilers because of rust and its consequent problems. However, with today's chemicals, any steel boiler will last a lifetime if properly taken care of.

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

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

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

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

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

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

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

$$\text{For copper } T = \frac{100 \times 3.5}{2,000} = .175''$$

$$\text{For steel } T = \frac{100 \times 3.5}{5,000} = .070''$$

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

This is about 3/16 diameter from Table I.

GRATE AREA

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

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

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

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

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 is that the tubes would be so small that they would soot up almost immediately so we must be practical. A tube of 1/2" outside diameter is very common and works quite well.

The general ratio of grate area to total tube area which has worked quite well is 8:1. That is, for each 8 square inches of grate area we will have a combined tube area of 1 square inch. From example above our total tube area will be $\frac{38}{8} = 4\text{-}3/4$ square inches. Using a tube which has an outside diameter of 1/2" and wall thickness of .35 our inside diameter is approximately 7/16". From Table I the area of a circle 7/16" diameter is .15 square inches. Now by dividing the total area 4-3/4 square inches by .15 square inches we have $\frac{4.75}{.15} = 32$ tubes in our boiler.

WATER CONSUMPTION

In order to determine the water consumption and size of our water pump we must know how much water is to be used in a given time. When water is turned into steam it occupies a specific volume in relation to the original volume of water and the final pressure of the steam. Table VII has been compiled to give this information.

Continuing with our 1½ - Mogul which has a 2" bore and 2½" stroke, we find that the volume of steam used per one revolution of the drive wheel is--

$Vs = A \times S \times N$

A = area of piston in square inches

S = stroke of piston in inches

N = number of working strokes per one revolution of the drive wheel

$A = 3.1416$

$S = 2.5$

$N = 4$

$Vs = 3.1416 \times 2.5 \times 4 = 31$ cubic inches of steam per revolution. Note we have not taken into account that the steam is cut-off before the full stroke of the piston. The reason is that due to the many mechanical deficiencies in model locomotives the extra amount of steam will be used.

Looking at Table VII we find that at 100 pounds (gage) pressure one cubic inch of water evaporated into steam will give us 237 cubic inches of steam.

Therefore $\frac{31}{237}$ = .13 cubic inches of water per revolution of the drivers is required.

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

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

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

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

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

SMOKE BOX

a. STACK DIAMETER AND LENGTH

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

b. EXHAUST NOZZLE

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

POWER TO OPERATE PLAIN SLIDE VALVES

The plain slide-valve is used in many model locomotives because of it's simple construction and ease of making it steam tight. However, it has one serious drawback and that is it takes a lot of power to operate it back and forth. This causes a lot of wear on parts such as the links, rocker, rocker pins and the eccentrics. The example below will illustrate the power required to make the plain slide valve move.

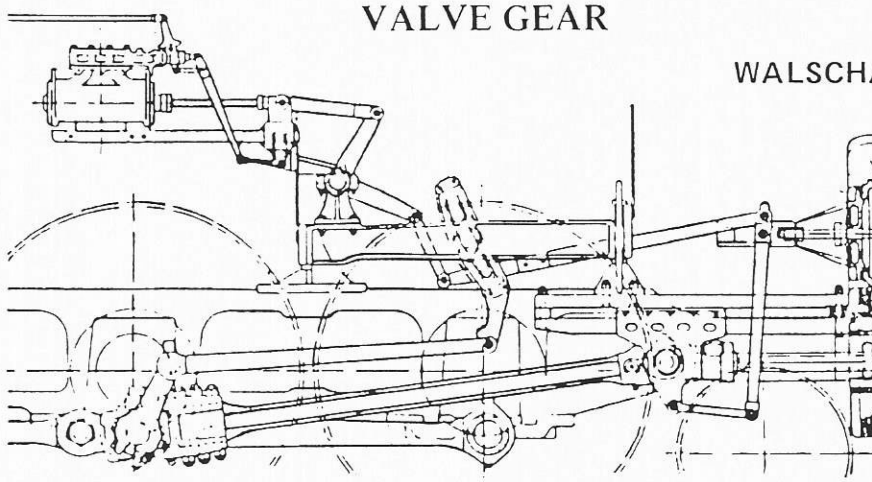
The resistance which must be overcome in moving any slide-valve is simply the friction between the valve and its seat. This friction depends upon the pressure of the valve against the seat, and this pressure is equal to the total steam pressure upon the back of the valve, minus the reaction of the steam pressure in the steam and exhaust ports.

We shall take for our example a valve which is 1¼" x 1½" which has an area of 1.875 sq.in. using a boiler pressure of 125 lbs we have a force of 1.875 x 125 = 233 lbs. pushing against the valve. This is not the actual pressure of the valve against the seat as we have a back pressure due to the steam acting against the valve. Tests have shown that the total back pressure is about ¼ of the pressure on the live steam side. This means that the pressure pushing the valve away from the seat is $\frac{233}{4}$ = 58 Lbs.

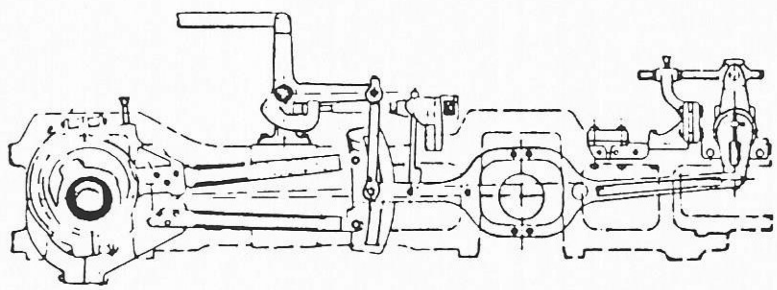
Now to find the total pressure of the valve against the seat we take 233-58 = 175 lbs. One way of looking at it is that we have to push the valve back and forth with 175 lbs. on it. It Looks like a real job for the valve gear to do. Now the actual force the valve gear has to operate against is found by dividing the total load on the valve by the friction valve, which for smooth -iron surfaces well oiled is about 10:1, thus to find the resistance the valve gear has to overcome we divide 175 by 10 = 17.5 lbs.

This 17.5 pound load has to be started, moved, stopped and reversed at every wheel revolution. This creates a real strain on the valve mechanism and for this reason railroads adapted balanced valves to reduce the maintenance in the valve gear assemblies. With a piston valve a load of only a few ounces will do the same job.

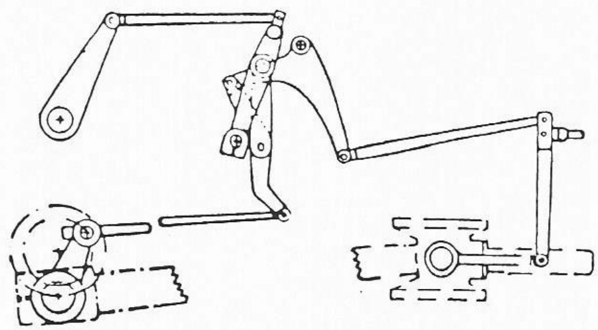
VALVE GEAR



WALSCHAERTS



STEPHENSON



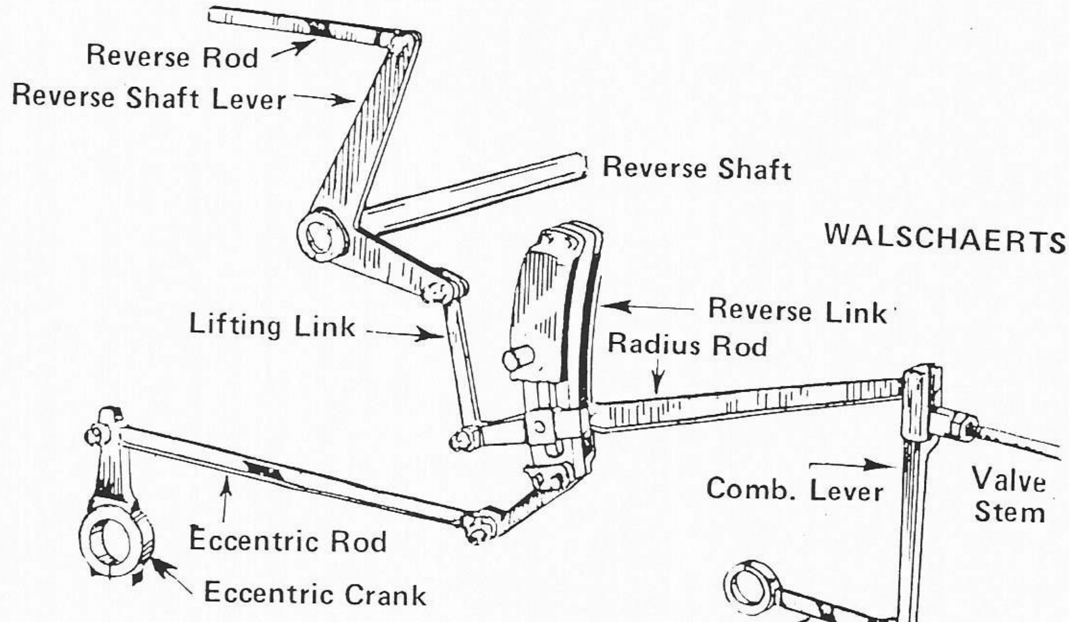
BAKER

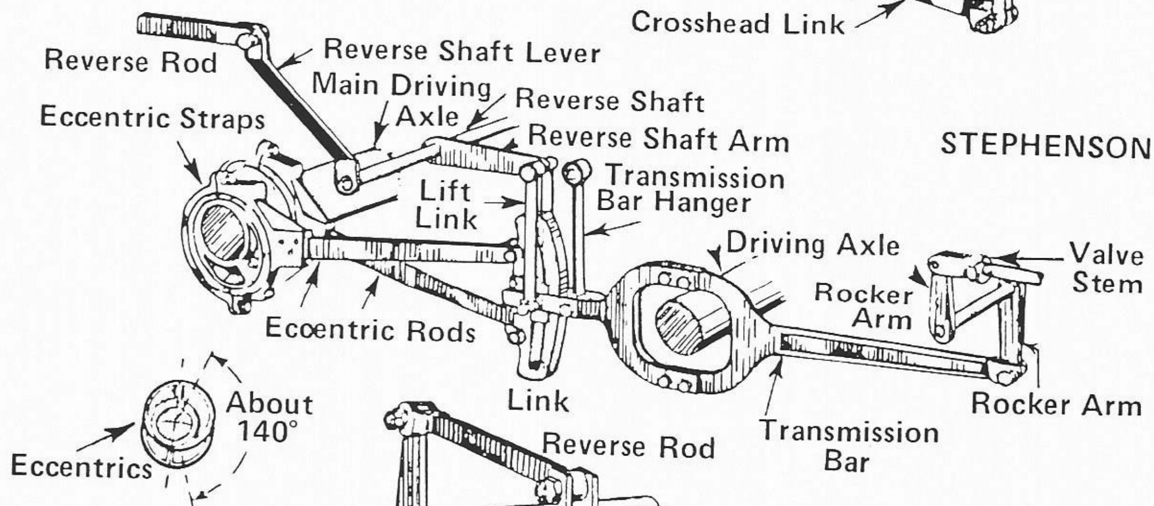
WALSCHAERTS: Originally introduced in the U.S. in the 70's by William Mason, this valve gear did not gain wide acceptance until it was introduced again, in 1904. Its virtues are simplicity and accuracy (in positioning and holding valve settings), and it eventually became the most extensively used of any of the modern gears. Part of the motion is derived from return crank on the outside of the main driver, part from the movement of the crosshead. The direction the loco runs is determined by whether the radius rod is above or below center in the link. Walschaerts gear has been used on locomotives with modern piston valves, and on many with older style slide valves. It's also been applied with and without a power reverse mechanism.

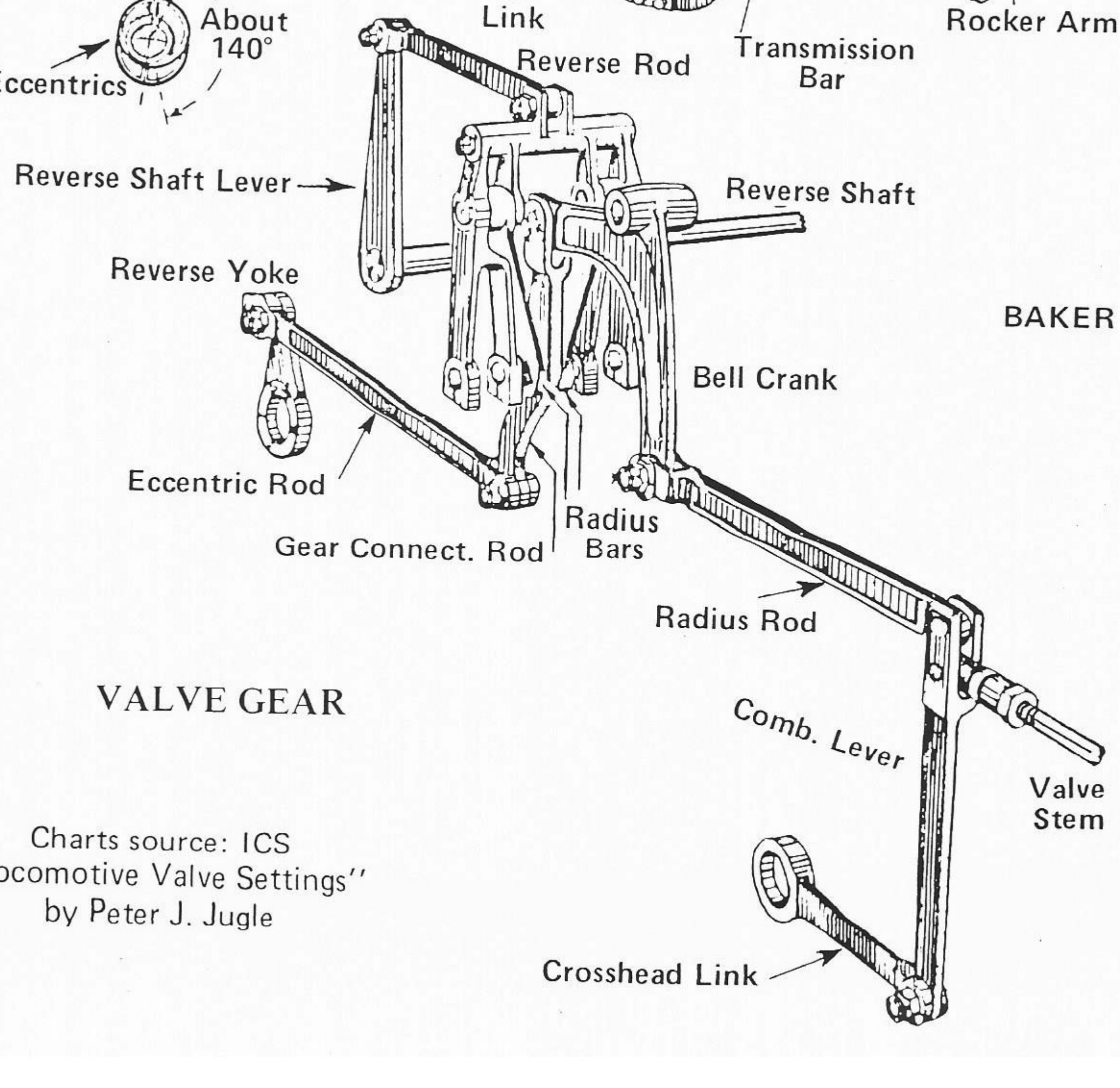
STEPHENSON: Here is the classic gear of the old-time locomotive, adopted widely in the 1850's and used almost universally until about 1900. It is an "inside" gear; that is, most of its parts are located inside the locomotive frame, the motion being transmitted to the valves by a system of rods and cranks. A pair of eccentrics on the main driver axle provides the motion for the gear. Direction of the locomotive is determined by whether the link is in a raised or lowered position. Stephenson gear is usually associated with slide-valve locomotives, but it has been used in some instances with piston-valves.

BAKER: Introduced about 1908, the Baker gear is a relatively simple, if heavy, gear despite its complicated appearance. Instead of a link and link block with sliding surfaces that are difficult to maintain as they wear, this gear has a double-bell, crank-shaped rocker that performs the same function of the link and block. By shifting the assembly up or down the locomotive direction is reversed. Motion is provided by a return crank on the main crank pin, plus a combination lever and link connected to the crosshead. Most Baker applications were on piston-valve locomotives with power reverse equipment.

See following page.







VALVE GEAR

Charts source: ICS
"Locomotive Valve Settings"
by Peter J. Jugle

VALVE SETTING

The purpose of listing valve setting procedures in this handbook is to provide a "short form" method for setting the valves on an engine which has an already established valve gear design. For those who wish to design their own valve gear, several sources of instructions have been printed in other books, magazines, etc. Suggested sources would be Joe Nelson's book "So You Want To Build A Live Steam Locomotive", or one of the English publications on valve gear design.

This handbook will list procedures for setting Stephenson and Walschaerts valve gears, as those are the most commonly used. The procedure for setting the Baker valve gear is similar to that used for the Walschaerts valve gear.

STEPHENSON VALVE GEAR

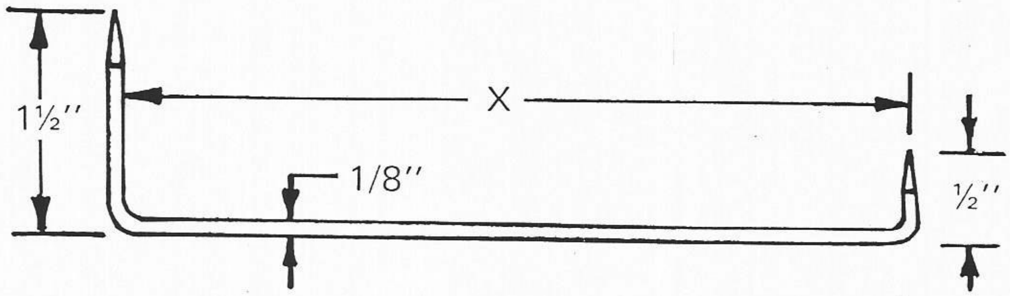
1. Check that the valve gear has been properly assembled and is free of binds.

2. Mark the valve port surface to show an exact center position of the valve when the valve is resting on the ports.

3. Adjust the length of the valve rod as follows:
   a) Position the upper rocker arm so that a line thru the center of the arm is perpendicular to the locomotive frame at the rocker box.
   b) Adjust the position of the valve in the steam chest so that it is directly centered over the ports as shown by the center marks and tighten adjusting nuts.
   c) Recheck to make sure the settings are correct.

4. Find the two dead centers of each piston. Definition:— the locomotive is said to be on a dead center when the center of the main axle, the main crankpin, and the crosshead wrist pin fall in one straight line. Find the dead center this way
   a) Place a sharp prick punch mark on the upper frame bar between the main driver and the next driver to the rear. This mark will be the location point for one end of the tram (see b).

b) Make a tram as shown from a relatively hard piece of 1/8" diameter steel rod. Make the "X" dimension the distance from the punch mark to the axle center of the main driver.



c) To find the forward dead center, rotate the main driver until the crosshead reaches a maximum forward stroke. Measure the gap between the crosshead and the cylinder head and make a spacer slightly longer than the gap. Place the spacer between the crosshead and the cylinder head and rotate the driver forward until the spacer is contacted by the crosshead and the cylinder head.

d) Without moving anything, scribe an arc on the main driver with the tram.

e) Remove the spacer and rotate the driver to the other side of dead center. Reinsert the spacer and repeat procedure (c). With dividers scribe an arc which crosses the two tram marks, using the driver center as the swing point for the dividers. Find the point on this arc which is exactly midway between the two tram marks, and place a sharp prick punch at this point. This point now locates exact forward dead center when the tram touches this point and the point on the frame.

f) To find back dead center, rotate the main driver until the crosshead reaches maximum reverse stroke. Measure the gap between the crosshead and the cylinder head and again make a spacer this time slightly shorter than the gap. Follow the procedure for finding forward dead center (c).

g) Repeat these procedures for the other dead centers.

h) To position the pistons, add the piston stroke and the piston thickness. Then subtract the result from the cylinder length and divide by 2 to get the piston clearance. Set the drivers in the forward dead center position and locate the piston from the front cylinder face by the piston clearance.

5. To find the neutral position and the correct length of the reverse reach rod, set the eccentrics to the angular dimension indicated by the valve gear design for your engine, – usually about 20° angle of advance. The angle of advance is the angle between the eccentric centerlines and a line perpendicular to the crankpin/axle centerline when the drivers are in forward dead center. This is a close but not final setting, so tighten the set screws enough to hold them to the axle, but not so tight as to upset the surface of the axle. With the reverse stand secured, place the reverse lever in neutral. Rotate the drive wheels to either dead center on the right hand side and raise or lower the right hand link block until the link block pin is exactly midway between the forward-gear and the back-gear eccentric rod (link blade) pins on the link. This is now the neutral setting, and the hole in the reach rod for the connection to the reverse lever (tumbling shaft) may be located and drilled. It is advisable to check the left hand link in the same way as the right hand to see if proper alignment of the reverse shaft (tumbling shaft) and link hangers exists.

6. Setting Eccentrics

   a) The eccentric rod (link blade) length should be the same for all eccentric rod assemblies within .002". This is the length between the center of the eccentric to the center of the hole for the link connection.

   b) Set the reverse lever into the full-forward notch. Starting with the right side, rotate the drivers to the forward dead center and observe the position of the edge of the valve to the port opening. Continue rotating the drivers until the back dead center is reached. Observe the position of the edge of the valve to the other port opening. The relationship between the edge of the valve and the port opening should be the same for both dead centers. If it is not, the eccentric rods are not equal in length and will have to be corrected. Assuming that equal movement is observed, but that the lead (the opening between the valve edge and the port) is not as required by the design of your valve gear, adjust the go-ahead eccentric to increase or decrease the lead. The go-ahead eccentric is the outboard eccentric on either side of the engine and controls the motion of the valve gear for forward motion of the locomotive. To increase the lead, rotate the go-ahead eccentric in the direction of the drivers and to decrease the lead, rotate the go-ahead eccentric in the opposite direction. Set the left hand go-ahead eccentric in the same manner.

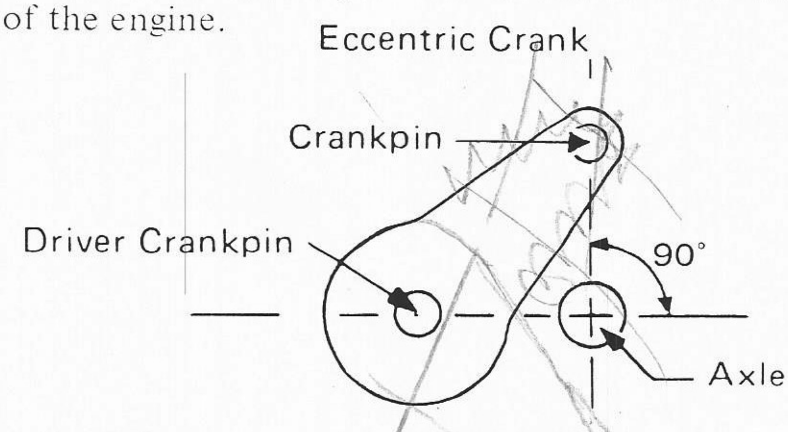
c) Setting the back-up eccentric (the in-board eccentric on either side of the engine) is done in an identical manner except that the reverse lever is placed in the full reverse position and the wheels are rotated for reverse movement. Having set all eccentrics, check positioning again. If all is correct, then tighten set screws firmly in place.

WALSCHAERT VALVE GEAR with PISTON VALVE

This procedure established by William Cooper in 1964, we have found to be most effective in setting the timing of a Walschaert valve gear.

1. Following procedures 1 and 4 for the Stephenson valve gear, locate the forward and back dead centers.

2. To set the eccentric crank position, rotate the drivers to forward dead center and adjust the eccentric as shown for both sides of the engine.



3. To set the valve, rotate the drivers to forward dead center. By looking through the front valve chest inspection port (centered over the valve liner port), check that the valve lead (the opening between the piston ring and the port) is proper for your valve gear design. Note the position of the lead.

4. Repeat procedure 3 for the back dead center and check the lead. The lead should be equal for both forward and back dead centers. If it is not, adjust the valve connecting rod until the lead is equal in both directions.

5. Repeat procedures 3 and 4 for the opposite valve. If the pin locations of the rods in your valve gear design are correct, these procedures should result in a dead square engine.

LOCOMOTIVE HAULING POWER AND ADHESION

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

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

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

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

LOCOMOTIVE SPRINGS

a. Size

In order for a locomotive to operate properly and to stay on the track it is necessary to have the locomotive fitted with equalizing levers and springs. The purpose of the equalizing levers is to distribute the weight equally on the driving axles. also to reduce the effects of shocks caused by the rails, and to allow the wheels to adjust themselves readily to any unevenness in the track without throwing an undue strain on the frames and other parts of the locomotive.
The formula below will give a good approximation of springs to use to give proper riding qualities to the locomotive.
First of all several things have to be known about the spring