W. T. Huntley

Dec. 1st 1904

see R R Gazette
Dec 9/04 p 620
article by Henderson
on fuel consumption
of locomotives

LOCOMOTIVE OPERATION

A TECHNICAL AND PRACTICAL ANALYSIS

BY

G. R. HENDERSON

MEMBER AMERICAN SOCIETY MECHANICAL ENGINEERS

CHICAGO
THE RAILWAY AGE
1904

PREFACE.

The object of this work is to give a complete and systematic discussion of the theory and practice of locomotive operation. By this is meant the work or results accomplished by a locomotive in motion, together with the effect upon itself and the track, and the amount of fuel and water needed to perform such work, rather than an exclusive treatise upon the mere manipulation of the machine, though the latter naturally forms a part of the study in connection with the proper manner of procuring certain results.

The order in which the different conditions are taken up is somewhat at variance with the usual custom. Inertia is first considered, as this property is inherent in all bodies having motion of a variable character. The Action of the Steam is next examined in detail, as to its power of producing rotation and strains in the various members of the mechanism. Resistance naturally follows, as it must be overcome by the action of the steam. This leads us to Slipping and Braking, which are the logical outcome of the previous studies.

These prepare us for an examination of the Hauling Power at slow speeds, but in order to consider high speeds, the steam capacity of the boiler must be determined, which is first undertaken. As the principal business of the locomotive is to haul trains, this is the most important chapter of the treatise, and it is placed in this part of the work, so that the various factors by which it is governed might be studied in an elementary manner before the complication necessary for a complete understanding of it is encountered. After this follow determinations of the water and fuel consumption under different conditions, and their economical use.

In order to make the book complete, not only for the student, but also for the convenient reference of railway officers and employes, considerable matter is presented which has been taken from competent authorities on the subject, and which is not, of course, original; such reproductions have been properly acknowledged in the text; there are also presented numerous elementary formulæ, the intention being to avoid the necessity for making extended references to other works. It has been the author's endeavor to discuss the various laws of mechanics which govern the subject in a technical and a practical manner, and while formulæ are used for the benefit of those who desire to follow entirely through the different phases of the several problems presented, tables and diagrams have been freely introduced to represent graphically the important laws and deductions, so that those who do not care to read the work from a technical standpoint, may obtain all the necessary information on the subject by consulting the diagrams, these having been arranged with a view of providing a ready and convenient reference, so that the various and complicated problems of locomotive performance may be quickly solved without the aid of extended mathematical deductions.

G. R. HENDERSON.

Philadelphia, June 30, 1904.

TABLE OF CONTENTS.

CHAPTER I.

CHAPTER II.

CHAPTER III.

CHAPTER VIII.

CHAPTER IX.

LOCOMOTIVE OPERATION.

CHAPTER I.

INERTIA.

All bodies in motion are affected more or less by their inertia, the amount depending upon their weight and the change in velocity to which they are subjected. As locomotives are continually varying their rate of speed, and also the speed of all portions of their machinery, it follows that the effects of inertia are distributed throughout the machine, and that they operate in various directions, in addition to the forces due to the locomotive as a whole, and which act mainly in a horizontal direction, both longitudinally and transversely of the track.

The two forces last enumerated are by far the most important, especially in fast moving trains. When the motion is slow, inertia forces are always small and unimportant, but as their value varies with the weight of the body affected and the difference in the squares of the initial and final velocities, they should be carefully considered in connection with high speeds. Whenever a moving body changes its rate of motion in a straight line, the effects of inertia are present. For instance, when a locomotive running at a high speed has its brakes applied, the inertia of the machine must be overcome by the brakes before the engine can be brought to a standstill, even if the forces of gravitation and steam on the pistons are absent. Again, in rounding a curve, even at a uniform rate of speed, as far as the track is concerned, there is a variation in the motion in a straight line, and this is made manifest in the centrifugal force tending to overturn the locomotive.

It will also be obvious, that while the speed of the engine may be constant, that of various portions of the machinery may be exceedingly variable. The pistons and their attachments reverse the direction of their travel relative to the locomotive at every stroke, and have a constantly variable motion. The parallel rods, while revolving uniformly, develop inertia forces in various right lines. The connecting rod, having a combination of these motions, develops a combination of forces. Thus it is seen that, as soon as the locomotive begins to move, there are forces exerted that are non-existent while the machine is dormant, and that these forces increase rapidly as the motion is accelerated.

STARTING AND STOPPING.

The longitudinal inertia of the locomotive as a whole is a most important factor in its operation and should be carefully considered.

Let P = force producing acceleration or retardation, in pounds.
- S = distance in feet in which the acceleration or retardation takes place.
- v = velocity in feet per second.
- p = acceleration or retardation in feet per second.
- G = weight of body in pounds.
- g = acceleration of gravity = 32.2 feet per second.
- t = time in seconds, during which velocity or acceleration is in action.

Then $v = pt$, and also

$S = \frac{vt}{2}$, v being considered as the ultimate velocity produced by p in t seconds.

Substituting the value of $t = \frac{v}{p}$ in second equation, we have $S = \frac{v^2}{2p}$.

It is well known that $\frac{p}{g} = \frac{P}{G}$, or, in words, the acceleration produced upon a body by a force, bears the same ratio to the acceleration of gravity (32.2) that the force producing acceleration bears to the weight of the body moved. Therefore by combination we obtain

$$S = \frac{v^2}{2p} = \frac{v^2 G}{2Pg} \text{ and } P = \frac{v^2 G}{2Sg}$$

It will generally be more convenient to express the horizontal forces in pounds per ton of weight of machine, and the velocity in miles per hour.

Let V = velocity in miles per hour; then $v = V \times \frac{5280}{3600} =$ 1.466 V, and $v^2 = 2.15\, V^2$; also substitute 2,000 pounds (1 ton) for G, and we have

$$P = \frac{2.15\, V^2 \times 2000}{2 \times S \times 32.2} = 66.76 \frac{V^2}{S}$$

The formula just stated considers only the movement of translation of the engine as a whole, but the rotative energy of the wheels and axles must also be accounted for.

It is not unusual for a pair of driving wheels about 70 inches in diameter to weigh 6,000 pounds, and for the axle to weigh at least 1,000 pounds and to have a diameter of 8 inches.

In order to determine the effect of wheels and axles, the moment of inertia about their axis must be determined. This moment is defined as "the sum of the products of the weights of the elementary particles of which the body is composed, by the square of their distances from the axis," and its value for a cylindrical body is $\frac{1}{2} G r^2$, r being the radius of the cylinder.* As a concentration of metal occurs near the rim, we figure as below:

$$\frac{6000 \times 40^2}{2} = 4,800,000 \text{ for two wheels.}$$

For the axle $\frac{1,000 \times 4^2}{2} = 8,000$, or for one pair of wheels and axle 4,808,000 inch pounds. If the engine be of the 10-

* The demonstration is as follows: In figure 1, r is the outside radius of the cylinder, and the dark circle represents an elementary area

wheel or 4-6-0 type, or any style with six drivers, we must take three times 4,808,000, or 14,424,000 inch pounds.

The rods will weigh for such an engine about 1,500 pounds, and with 28-inch stroke will act at 14 inches radius. Their moment of inertia will be $1{,}500 \times 14^2 = 294{,}000$ inch pounds. Summing these moments, we obtain a total of $14{,}424{,}000 + 294{,}000 = 14{,}718{,}000$, and dividing by 35^2 (1,225) to reduce the effect to the wheel rim, we obtain $\frac{14{,}718{,}000}{1{,}225} = 12{,}000$ pounds (approximately) as the rotative inertia of driving wheels, axles and rods at the rim or tread of wheel.

For the tender we may assume

2 wheels, 36 inches in diameter, at 700 pounds each = 1,400 pounds.

1 axle, at 450 pounds = 450 pounds.

The moment of inertia of the wheels will be $\frac{1{,}400 \times 324}{2} = 226{,}800$ inch pounds, and of the axle $\frac{450 \times 6}{2} = 1{,}350$ inch

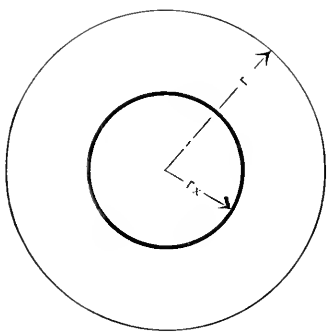


Fig. 1.

da_x at radius r_x. The elementary moment will be $dm = da_x\, r_x^2$. But the elementary area $da_x = 2\,\pi\, r\, dr \frac{r_x}{r} = 2\,\pi\, r_x\, dr$ and $dm = 2\,\pi\, r_x^3\, dr$. Integrating between o and r.

$$\int_0^r 2\,\pi\, r_x^3\, dr = \frac{1}{2}\,\pi\, r^4, \text{ but area} = \pi\, r^2, \text{ therefore}$$

$$\int_0^r dm = \frac{1}{2}\, A\, r^2, \text{ and } A \text{ may be replaced by G, giving } \frac{1}{2}\, G\, r^2$$

pounds; so for pair of wheels and axle = 228,150 inch pounds; and dividing by $18^2 = 324$, we have $\frac{228,150}{324} = 704$ pounds at tread of wheel, and for four pairs of wheels and axles equals 2,816 pounds. This, added to the value found for the locomotive, and allowing two pairs more for the engine truck wheels, we obtain a total inertia weight of

3 pairs locomotive wheels and rods..........	= 12,000 pounds
4 pairs tender wheels	= 2,816 pounds
2 pairs truck wheels	= 1,408 pounds
Total	= 16,224 pounds

The total weight of such an engine and tender will not be less than 300,000 pounds, and $\frac{16,224}{300,000} = .05+$, from which it is apparent that the revolving parts of the engine and tender will increase its effective inertia in a horizontal direction by 5 per cent. Adding this 5 per cent to our last value of P, we have

$$P_t = 105 \times 66.76 \frac{V^2}{S} = 70 \frac{V^2}{S} \qquad \text{(1)}$$

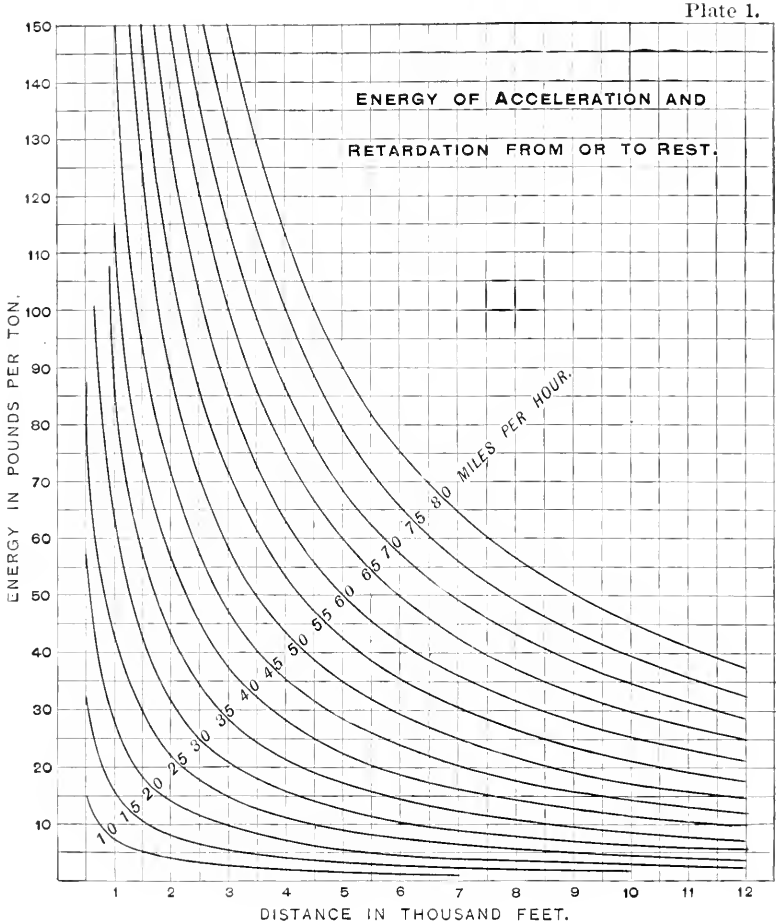
P_t being the accelerating or retarding force in pounds per ton of engine and tender, including the effect of wheels, etc.

The above formula (1) considers the force necessary to produce a desired velocity "V" in a certain distance "S." At times it may be desirable to determine the force required to produce the velocity "V" in a certain time. To evolve this form, we combine the fundamental formula $v = pt$ and $\frac{p}{g} = \frac{P}{G}$, or $v = \frac{Pgt}{G}$ and $P = \frac{Gv}{gt}$, and allowing 5 per cent increase for wheels, rods, etc., reducing to miles per hour and considering a weight of 2,000 pounds, we obtain

$$P_t = \frac{1.05 \times 1.466\,V \times 2,000}{32.2\,t} = 95.6 \frac{V}{t} \qquad \text{(2)}$$

The formulæ just developed consider either acceleration from a state of rest to a velocity V, or retardation from such velocity to a dead stop. If we wish to consider the effects of variation in velocity, we must substitute the difference of the squares, thus: Let V_1 = the smaller velocity and V_2 =

Plate 1.

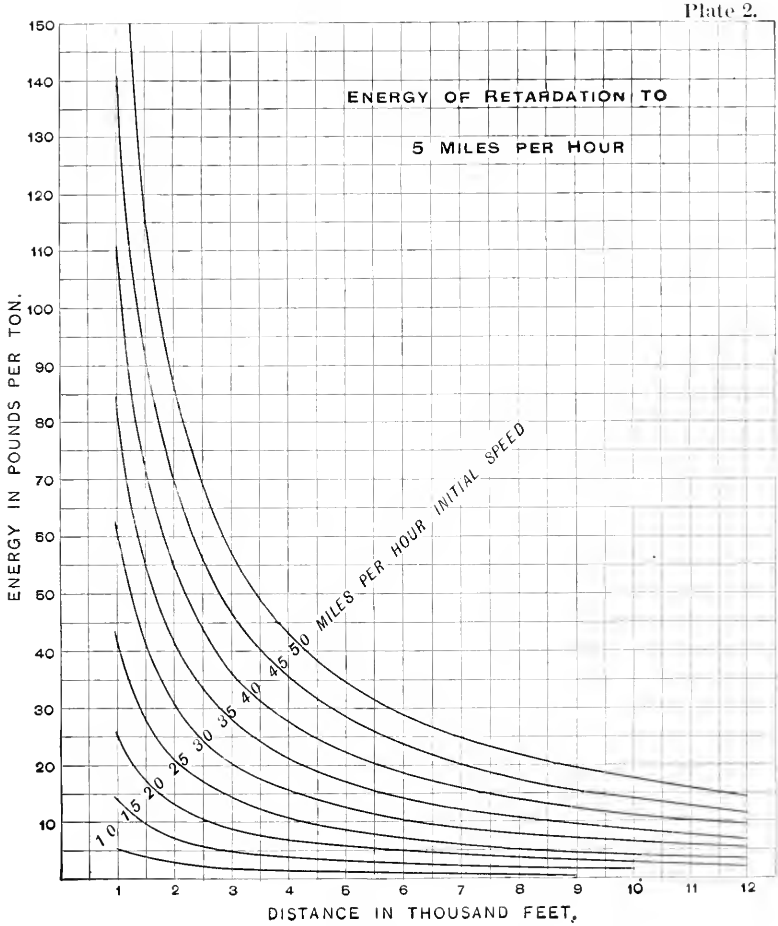


the greater velocity, both in miles per hour. Then for the force of acceleration or retardation, we have from formula 1,

$$P_t = 70 \frac{V_2^2 - V_1^2}{S} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots (3)$$

This form is particularly useful in considering the effect due to loss of velocity in ascending a short grade, and if we assume that a speed of at least 5 miles an hour must be maintained at the summit, we can write $P_t = 70 \frac{V^2 - 25}{S}$,

Plate 2.



V in this case being the velocity of approach at the foot of the grade, and P_t will be the assistance in pounds per ton which inertia will give to the steam power of the locomotive. It may here be stated that each ton in the train will be assisted in the same way, as the 5 per cent allowance for rotative inertia will apply without great error to the cars composing the train.

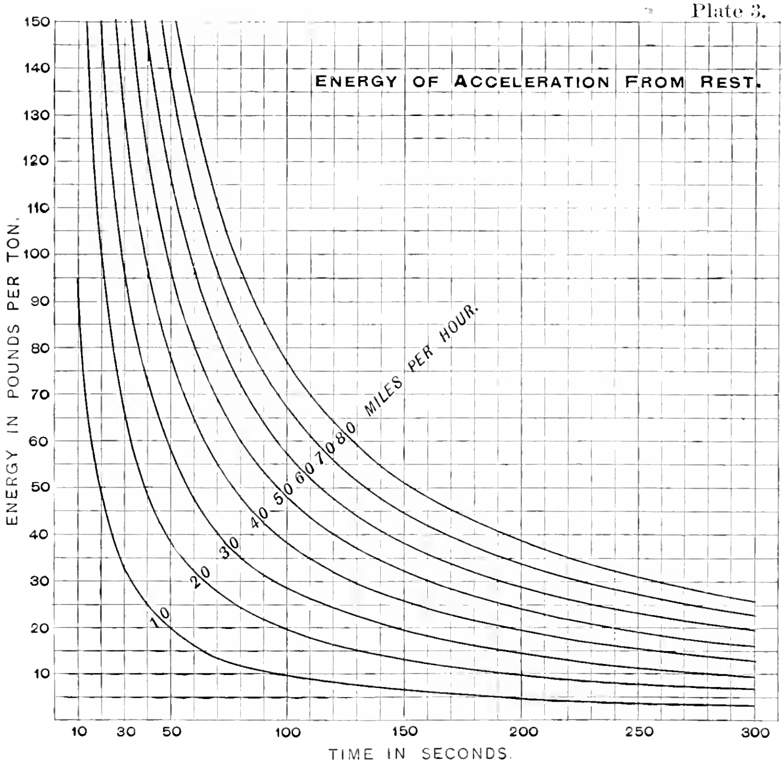
Plates 1 and 2 illustrate graphically the energy in pounds per ton due to acceleration and retardation in various distances, first, from and to a dead stop, and the second to a minimum speed of 5 miles per hour.

To illustrate the use of these tables, suppose that it be desired to find the amount of energy necessary to take a train from a state of rest, and bring it to a velocity of 25 miles an hour in a distance of 2,000 feet. By formula

$P_t = 70 \frac{V^2}{S} = \frac{70 \times 625}{2,000} = 21.9$ pounds, and by diagram, select the intersection of the curve marked "25 miles per hour" and the abscissa corresponding to 2,000 feet of distance, which will be found opposite the ordinate 21.9. So, for obtaining the benefits of momentum from 35 miles an hour to 5 miles at the summit of a grade 8,000 feet long, on diagram 2, where the 35-mile curve crosses the 8,000 feet line, we read 10½ pounds per ton, or by formula $P_t = 70 \frac{V^2-25}{S} = 70 \frac{1,225-25}{8,000} = 10.5$ pounds per ton, which means that inertia will assist the engine up the grade with a force of 10½ pounds for each ton in the train, and in the first case, the total weight of train in tons multiplied by 21.9 will give the average force necessary to produce the speed desired, of course being in addition to the resistance due to grade, curvature and friction. If it be desired to determine the force necessary to produce a particular velocity in a given time, we may use formula 2 and plate 3, which latter gives a graphical representation of the formula. For example, if a train is to be brought from rest to a speed of 30 miles per hour in 30 seconds, we find, by formula 2, $P_t = 95.6 \frac{V}{t} = 95.6 \frac{30}{30} =$ 95.6 pounds per ton average force required to overcome inertia only. This can be found from plate 3 in a manner similar to that previously explained.

A uniform force applied to a body produces a constant acceleration, and an increasing velocity, as demonstrated by the formula $v = pt$, this showing that the velocity attained is directly proportional to the time of the acceleration. In getting a train up to speed, however, the increasing resistance due to speed, and the diminishing power of the locomotive, prevent a uniform continuance of the force P, so that the acceleration is

Plate 3.



really a continually diminishing quantity, and this prevents the attainment of a high velocity as rapidly as would be the case were these variables absent, and which then would result in a constant rate of acceleration similar to the motion of a falling body. The term "velocity head" is frequently used in connection with moving bodies, and by it is meant the height which they would have to drop vertically in order to attain, by the influence of gravity, the speed under consideration.

If we let h = height in feet which a body must drop to attain the velocity "v," we have the equation $h = \frac{v^2}{2g}$, and reducing to miles per hour, $h = \frac{2.15\ V^2}{2 \times 32.2} = .0333\ V^2$ and adding 5 per cent for rotative effect of wheels, etc.

$h = .035\ V^2$ .... (4)

This formula, reduced to a tabulated form, appears as below:

VELOCITY HEAD.

V = 1	2	3	4	5	6	7	8
h = .035	.140	.315	.560	.875	1.26	1.72	2.24
V = 9	10	15	20	25	30	35	
h = 2.83	3.50	7.85	14.0	21.9	31.5	42.9	
V = 40	45	50	55	60	70	80	
h = 56.0	70.9	87.5	106.	126.	172.	224.	

Some idea of the effects of collisions at high speed may be gathered from the above; thus, if a locomotive running 50 miles an hour were to meet an impenetrable obstruction, the damage might be represented by supposing the engine to fall 87½ feet vertically.

CENTRIFUGAL FORCE ON CURVES.

When a locomotive, during its journey, traverses a curve, centrifugal forces are present, due to the inertia of the engine and tender, which inertia would, if unrestricted by the rail, compel them to continue on a tangent. These centrifugal forces tend to overturn the machine in a direction normal to the curve, and, of course, about the outer rail. Whether there will be sufficient force for that purpose depends upon the speed of the engine, the sharpness of the curve, the height of the center of gravity of the locomotive, the gauge of track, and the elevation of the outer rail.

The formula given in mechanical textbooks for centrifugal force is

$C = \frac{G\ v^2}{g\ r}$ .... (5)

where G = weight of body in pounds,
C = centrifugal force in pounds,
v = velocity in feet per second,
r = radius of curve in feet,
$g = 32.2$, as before.

It will generally be more convenient to express the speed in miles per hour = V, and the curvature in degrees = c.

Now we have already found that $v = 1.466\ V$ and $v^2 = 2.15\ V^2$, and we also know that $r = \frac{5.730}{c}$, so, by substitution, we derive

$$C = \frac{G\ v^2}{g\ r} = \frac{2.15\ V^2\ G\ c}{32.2 \times 5.730} = .0000117\ V^2\ G\ c$$

and for the proportion of the weight of the engine we have

$$\frac{C}{G} = .0000117\ V^2 c \qquad (6)$$

In Fig. 2, let C and G be represented by the arrows, then R will give the resultant of the two forces. As long as this resultant falls inside of the rail, stability is insured, but if it falls outside, or even on the rail, the engine will overturn. The center of gravity is represented by the black dot, and the forces of gravity and centrifugal act at this point.

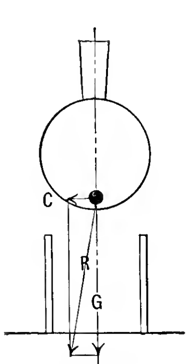


Fig. 2.

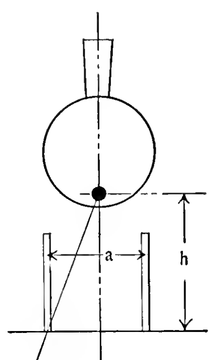


Fig. 3.

If, in Fig. 3, we let h = height of center of gravity of engine above the rail; and a = the gauge of track, we find that $\frac{a}{2h}$ is the tangent of the angle where the resultant force will just strike the rail, therefore, to insure stability, the value $\frac{C}{G}$ must be less than $\frac{a}{2h}$. However, it is customary to elevate the outer rail of curves, and this helps to increase the stability of the trains. In order to include this rail elevation in our calculations, let us proceed as follows:

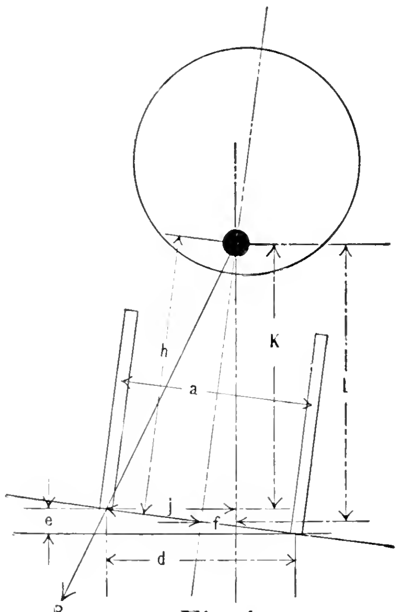


Fig. 4.

In Fig. 4, let a and h represent the gauge of track and height of center of gravity, as before, and e the elevation of the outer rail. The other letters represent certain dimensions in order to make the necessary calculations. Then from similar triangles, we have

$$\frac{a}{e} = \frac{h}{f} \text{ and } j = \frac{d}{2} + f = \frac{d}{2} + \frac{he}{a};$$

$$\frac{a}{d} = \frac{h}{l} \text{ and } K = l - \frac{e}{2} = \frac{dh}{a} - \frac{e}{2}, \text{ and}$$

$$\frac{j}{K} = \frac{\frac{d}{2} + \frac{he}{a}}{\frac{dh}{a} - \frac{e}{2}};$$

but as the angle of elevation (whose sin is $\frac{e}{a}$) is always quite small, we can write $d = a$ or

$$\frac{j}{K} = \frac{\frac{a}{2} + \frac{he}{a}}{h - \frac{e}{2}};$$

but $\frac{j}{K}$ is the tangent of the angle which the resultant force R should make with the vertical in order to pass directly through the rail, so that when

$$\frac{C}{G} = \frac{\frac{a}{2} + \frac{he}{a}}{h - \frac{e}{2}} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (7)$$

the point of unstability has been reached.

In order to give a practical example of the use of formulæ 6 and 7, we will take the case of a disastrous wreck that actually occurred. The height of center of gravity of the locomotive was 69 inches above the rail, the gauge was 4 feet 9 inches (57 inches), and the curve 6 degrees, with an elevation of 7 inches, therefore in formula 7, $a = 57$; $h = 69$ and $e = 7$. Substituting and reducing, we have

$$\frac{C}{G} = \frac{\frac{57}{2} + \frac{69 \times 7}{57}}{69 - \frac{7}{2}} = \frac{37}{65.5} = .56+$$

Now, placing this value in equation 6 and transposing, we have

$$V = \sqrt{\frac{.56}{.0000117 \times 6}} = \sqrt{8,000} = 90 \text{ approx.}$$

showing that at 90 miles an hour, the centrifugal force would overturn the engine. From the train sheets, it was figured that the locomotive must have been running near the speed mentioned.

Plate 4.

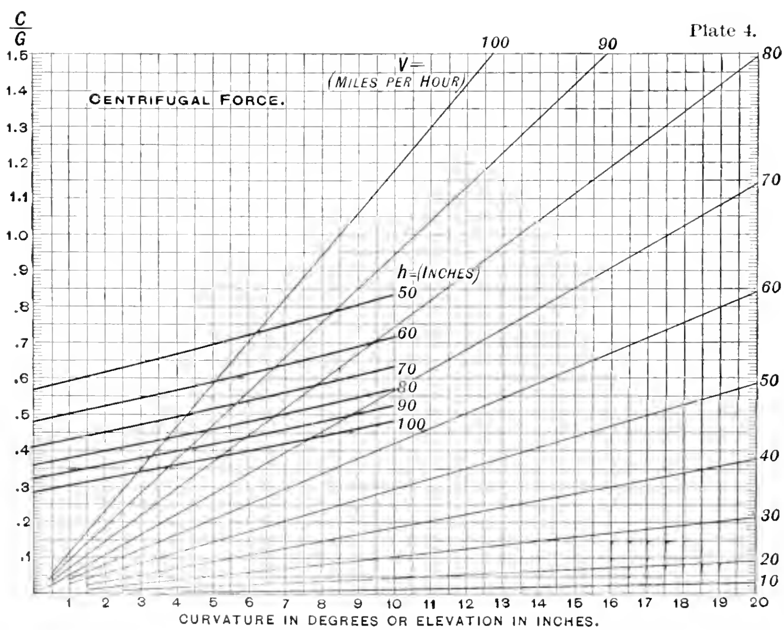


Plate 4 gives a graphical representation of formulæ 6 and 7, so that any combination of height of center of gravity, elevation of outer rail, curvature and velocity, with standard-gauge track, may be studied without calculations. For example, let us take the case just quoted.

As the height of center of gravity, select the line representing h at 70 inches, and at its intersection with ordinate through 7 at bottom (for rail elevation) read .56 for value $\frac{C}{G}$. As the curve in question was 6 degrees, we find that the intersection of the .56 lines and the ordinate passing through 6 degrees of curvature is also traversed by the radial line corresponding to a velocity of 90 miles per hour. (In using this dia-

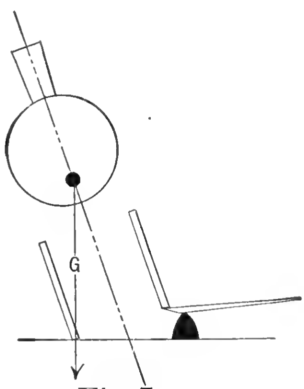


Fig. 5.

gram, care should be taken to associate the height of center of gravity and the elevation of rail together, also the velocity and curvature, as designated by formulæ 6 and 7.)

Thus 90 miles per hour is the speed at which the engine considered would turn over by centrifugal force on a 6-degree curve with a 7-inch elevation. If the elevation were only 4 inches, 85 miles speed would overturn. It must be borne in mind in using these formulæ, that the action of the springs carrying the engine will rather increase the overturning tendency, for, as the resultant approaches one side, the springs on that side will deflect beyond the normal, and allow the center of gravity also to move in the same direction, thus increasing the danger.

The effect of centrifugal force on trains is curious, and well worth considering as to its action in detail. It is often customary to think of it in the same way that the force of gravity would act if we were to jack up one side of a locomotive until it fell over to the other side. It will be seen by a little thought that there is really no analogy between the two cases.

If we lift a locomotive (or other body) by means of a bar or similar device, the effect of gravity will be to restore the body to its normal position, if the bar be removed, until the force of gravity acts outside of the width of base. As shown in Fig. 5, further effort on the bar will overturn the engine, as the line of weight G now passes through one rail, all the weight being on that side. In other words, the right-hand wheels must leave the rail by an amount (approximately) $= \frac{a^2}{2h}$, see Fig. 3, before the engine will roll over. When centrifugal force is acting, however, there is no leaving the rails on one side (except for jars and jolts), until the resultant R (Fig. 2) passes through the outside rail, although the pressure is constantly decreasing on the right side. The instant that the resultant R passes beyond the rail, the centrifugal force lifts the right side clear from the track, as all pressure has ceased on this side, thereby increasing the lever arm (h) of action of the force, and still further diminishing the lever arm $(\frac{a}{2})$ of gravity, and the engine rolls over immediately, without any period of "contemplation" analogous to that when turning over with a bar. This point is well worth bearing in mind.

In the above formulæ and diagrams, it is apparent that we must know the location of the center of gravity of the engine relatively to the rails. Ordinarily this point lies near to the bottom of the barrel of the boiler, but it should be determined closely. The old method of calculating the moments of the various parts about the rail and then dividing by the total weight of the engine, was an extremely tedious undertaking, if done with sincerity. The method of swinging the engine on trunnions (one at each end) is costly, and requires heavy cranes, not always convenient. A method was devised by the author a few years ago which can be cheaply and quickly followed at any point where there are track scales of sufficient capacity; this will now be explained.

Suppose that we have a body of symmetrical cross section, the weight of which is known, resting upon supports at the two lower edges only. Now, if this body be tipped slightly, so that one of the edges upon which it stands is lower than the other, the center of gravity will be displaced horizontally, and the lower support will sustain more weight than the higher one, due to the lateral displacement of the center of gravity; and if the angle of tipping be such that the center of gravity is vertical over the lower edge, the body will be in unstable equilibrium, as illustrated in Fig. 5, and the total weight of the body will be concentrated on the lower edge.

In order to determine the location of the center of gravity, however, no great amount of tipping is necessary—in fact, no more than that caused by the elevation of an 8 or 10 degree curve—so that the work is entirely devoid of danger.

The engine should first be carefully weighed upon the track scales, with a definite height of water in the boiler, and then be backed off. The rail must next be removed from the narrow side of the scale platform, and blocks laid close together, like ties, from the outer or pit wall frame to the fixed or dead rail support, care being taken to be sure that these blocks are entirely clear of the portion of the scale platform under them.

Then relay the removed rail on these blocks to gauge with the rail remaining on the scales, and slope off from one end to mate the track upon the ground. Balance the scale beam and, after bringing the water in the boiler to the previous level, run the engine upon the track just prepared. By now balancing the scales, the weight upon the lower rail will alone be indicated; while the locomotive is in this position, the difference in level between the two rails must be accurately determined by means of a spirit level. The frames of the locomotive should be blocked over the boxes, in order to keep the engine in a normal position. There will be a slight error due to the water in the boiler adjusting itself to the new angle, but this will be on the safe side. As the wheels on the elevated side will rest on the inside of the rail and those on the depressed side on the outside, the horizontal distance between points of contact must also be measured, also between contact on lower rail and center line of the engine.

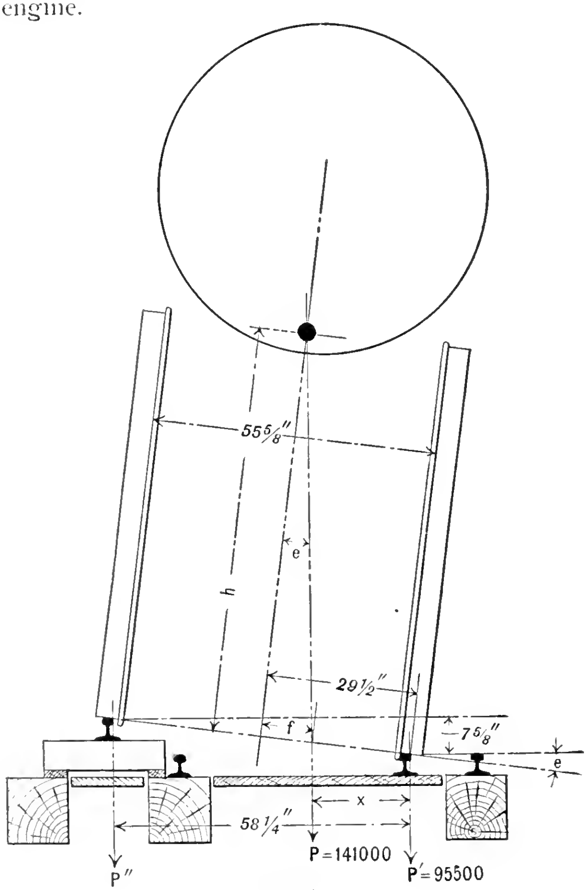


Fig. 6.

In order to make the process and calculations perfectly clear, an actual case will be worked out.

The weight of the engine was first found to be 141,000 pounds. The track scale was then treated as just explained, and the engine run upon it. Fig. 6 shows the measurements and weights actually taken in figures, and those that were calculated, in letters, so that the sketch acts as a guide in making future determinations of this kind. With a rail elevation of $7\frac{5}{8}$ inches, the scale beam indicated 95,500 pounds on the lower rail, and a measure $58\frac{1}{4}$ inches between points of rail contact, and $29\frac{1}{2}$ inches from center of engine to lower rail contact. Now if $P =$ total weight, and $P' =$ that on lower rail, while $P'' =$ that on higher rail, we must have $P = P' + P''$, and $P'' = P - P'$, or $141{,}000 - 95{,}500 = 45{,}500$ pounds. So, by equating the moments, we find $P'' \times 58\frac{1}{4} = P \times x$, and $x = \dfrac{P'' \times 58\frac{1}{4}}{P} = \dfrac{45{,}500 \times 58\frac{1}{4}}{141{,}000} = 18.8$ inches. In order to determine f accurately (which is necessary) we must figure $f = 29\frac{1}{2} - 18.8 \sec e$, e being the angle of inclination from the vertical. But $\tan e = \dfrac{7.62}{58.25} = .1325$, indicating that angle $e = 7° 33'$, and the secant of $7° 33'$, or $\sec e = 1.0087$, therefore $f = 29.5 - 1.0087 \times 18.8 = 10.5''$, and as $f = h \tan e$, we have $h = \dfrac{f}{\tan e} = \dfrac{10.5}{.1325} = 79.25$ inches, which is the height of the center of gravity above the rail.

The value of h may be determined graphically by laying out the construction shown in Fig. 6, but the mathematical method used above is more accurate.

EFFECT ON RODS.

After the effects of inertia on starting and stopping, and in rounding curves, which we have already studied, the most important are those developed by the rods and their related parts. We will here study only the inertia of these parts, and later, in connection with the steam action, we will complete our investigations.

The strains developed in the rods by inertia are unimportant, outside of those which put them in bending, or affect them transversely, and as the parallel or side rods maintain their longitudinal axis in a horizontal direction, it will be the inertia of the rod in a vertical direction that produces the transverse strains. It can be shown that the effect of the vertical inertia, when at its maximum, is the same as the centrifugal force, and is equal to $\frac{G\,v^2}{g\,r}$, as in formula 5.*

---

*The demonstration is as follows:

Let v = velocity in feet per second.
r = radius of motion in feet.
G = weight of body in pounds.
t = time in seconds.
g = 32.2.
p = acceleration or retardation.

In Fig. 7, let us represent v by the tangent to the circle, its direction at that instant being indicated by the tangent, and the vertical

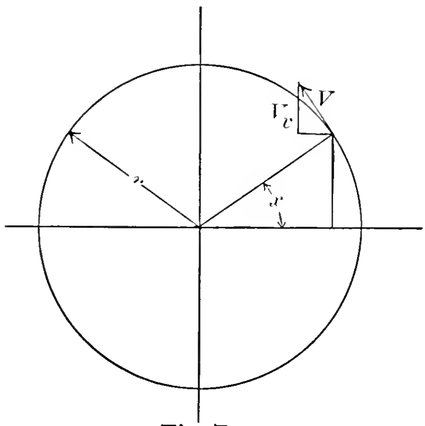


Fig. 7.

component of the velocity will be $v_v = v \cos x$. Differentiating, we have $d\,v_v = -v \sin x\, d\,x$, where $d\,v_v$ is the difference between two infinitesimally consecutive vertical velocities, therefore, $\frac{d\,v_v}{d\,t}$ = the vertical acceleration or retardation, and as we have already seen that the force of acceleration or retardation $= \frac{G\,p}{g}$, we can writes this force $P = \frac{G}{g} \times \frac{d\,v_v}{d\,t} = -\frac{G}{g} v \sin x \frac{d\,x}{d\,t}$; but the angular velocity $\frac{d\,x}{d\,t} = \frac{v}{r}$, therefore $P = -\frac{G}{g} \times \frac{v^2}{r} \sin x$. Thus the vertical force is a direct function of the sine of the angle made by the body measured from the horizontal axis of revolution and is a máximum when $x = 90°$ or at

When the side rods are at the top quarter, the centrifugal force is exerted upward, and if the rod be of uniform cross section throughout its length, the force will be uniformly distributed, and will be reduced by the actual weight (effect of gravitation) of the rod. At the bottom quarter, the force will act downward, and will be in addition to the normal weight. At the ends of the stroke, there will be no vertical forces, and at intermediate points these forces will be proportional to the centrifugal force multiplied by the sine of the angle made by the crank in its revolutionary movement from the dead center or ends of stroke.

It will be more convenient to transpose the formula No. 5, $C = \frac{G v^2}{g r}$ into a form containing the number of revolutions per minute in place of velocity in feet per second, and which we will do by inserting in the formula, in place of v^2, the value, $.0109 r^2 n^2$. This was derived as follows: Let n = number of revolutions per minute made by the body, then

$$v = \frac{2\pi r n}{60} \text{ and } v^2 = .0109 r^2 n^2. \text{ Then } C = \frac{G v^2}{g r} \text{ becomes}$$

$$C = .00034 G r n^2 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (8)$$

As the maximum forces are those of the greatest importance, let us consider that in a locomotive the greatest speed expected is generally about equal, in miles per hour, to the diameter of the driving wheel in inches. Representing this driving wheel diameter in inches by D, and the speed in miles per hour by V, as before, we have the revolutions,

$$n = \frac{V \times 5{,}280 \times 12}{\pi D \times 60} = 336 \frac{V}{D} \text{ and } n^2 = 112{,}896 \frac{V^2}{D^2}$$

so that, if we assume that $V = D$, we have for the maximum case simply $n = 336$ and $n^2 = 112{,}896$. Substituting this in equation number 8, we have $C = 38.4 G r$, or, if we let s = stroke of engine in inches, $C = 1.6 G s$. . . . . (9)

---

the top and bottom quarters, and as x for 90° = 1, we can write simply $P = \pm \frac{G v^2}{g r}$, Now, letting L = length of rod in inches, center to center of crankpins, for a load P uniformly distributed, we would have a maximum moment at center $= M = \frac{P\,L}{8}$, and inserting equation 9, considering G as the total weight of the rod between centers, we find that

$$M = \frac{P\,L}{8} = \frac{1.6\,G\,s\,L}{8} = .2\,s\,G\,L \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (10)$$

this moment being that due to centrifugal force only, and at a speed equaling the diameter of the drivers in inches. In order to determine the extreme fiber strain in the rod, which will be at the center of its length, let

S = modulus of section of rod around a horizontal axis,

f = extreme fiber strain in rod in pounds per square inch, then

$$M = S\,f = .2\,s\,G\,L\text{, and } f = \frac{.2\,s\,G\,L}{S} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (11)$$

For example, let us examine into the strains that will come upon a side rod 90 inches long, 5 inches deep, by $2\frac{1}{2}$ inches wide, on an engine having 28 inches stroke. We find from the tables at end of book that the area of such a rod will be 12.5 square inches, and $12.5 \times .28 \times 90 = 315$ pounds weight = G, .28 being weight per cubic inch of steel. We also find that the

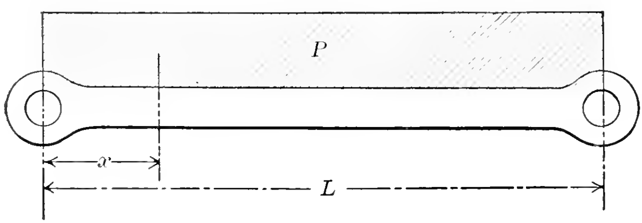


Fig. 8.

modulus of section, S = 10.42, and, as stated, L = 90 and s = 28, so that in equation 11 we substitute

$$f = \frac{.2\,s\,G\,L}{S} = \frac{.2 \times 28 \times 315 \times 90}{10.42} = 15{,}200 \text{ pounds}$$

per square inch maximum fiber strain at center of rod.

For the strain at any other point on the rod as at x, Fig. 8, we must consider that the moment at x is

$$M_x = \frac{P x}{2} - \frac{P x^2}{2 L} = \frac{P x (L - x)}{2 L}$$

If x = 0, $M_0 = 0$, or nothing at end.

$$\text{If } x = \tfrac{1}{2} L,\ M = \frac{\frac{1}{2} P L (\frac{1}{2} L)}{2 L} = \frac{P L}{8} \text{ as in equation 10.}$$

Inserting for P the value found for C in equation 9, we have

$$M_x = \frac{1.6 G s x (L - x)}{2 L} = \frac{.8 G s x (L - x)}{L}$$

$$\text{and } f_x = \frac{.8 G s x (L - x)}{L S} =$$

the extreme fiber strain at the point x. For this reason, parallel rods are sometimes made deeper at the center, as that is the point of greatest strain.

With the connecting or main rods the case is somewhat different, for, while the crankpin end travels in a circle the same as the parallel rods, the crosshead end moves in a hori-

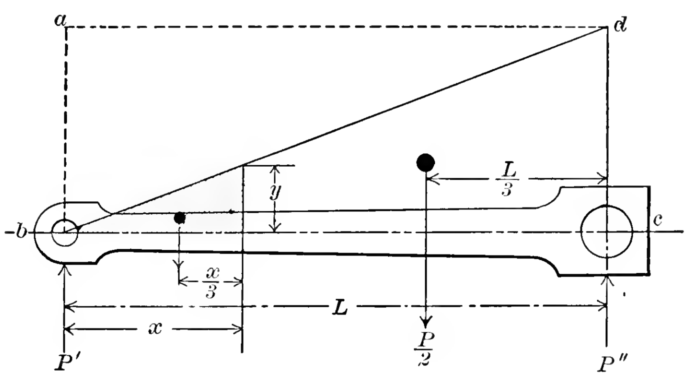


Fig. 9.

zontal direction only, and there can be no centrifugal force or rather vertical inertia at this end.

The load upon the rod due to vertical inertia may be represented by a triangle, as in Fig. 9. Consider P as the load of inertia (vertically) if the whole rod had a circular motion, same as crank end c, and let this load be represented by the rectangle a, b, c, d, uniformly distributed. As the effect is o at crosshead end b, and a maximum at crank end c, the effect of the variable load will be that of the triangle b, c, d, whose center of gravity is distant $\frac{L}{3}$ from c, and whose total value is $\frac{P}{2}$. The reactions at b, P′ and at c, P″ will therefore be $P' = \frac{P}{6}$ and $P'' = \frac{P}{3}$, and the corresponding moment at any point x will be $P' x = \frac{P x}{6}$.

The portion of the load between b and x will have the value $\frac{Px^2}{2 L^2}$; for y : x : : $\underline{d\ c}$: L; but $\underline{d\ c} = \frac{P}{L}$, so $y = \frac{Px}{L^2}$, and the shaded area $= \frac{1}{2} x\ y = \frac{Px^2}{2 L^2}$. As its center of gravity is $\frac{x}{3}$ from x, the moment about $x = \frac{Px^2}{2 L^2} \times \frac{x}{3} = \frac{Px^3}{6 L^2}$.

The difference of these moments will be the actual one, viz.:

$$M_x = \frac{Px}{6} - \frac{Px^3}{6 L^2} = P \frac{L^2 x - x^3}{6 L^2}.$$

If now we make $x = 0$, we have $M_0 = 0$, and also for $x = L$, $M_L = 0$, showing no moment at the ends. If we put $x = \frac{L}{2}$ and equate, we get the moment at the center, thus:

$$M_c = P \frac{\frac{L^3}{2} - \frac{L^3}{8}}{6 L^2} = P \frac{\frac{3}{8} L^3}{6 L^2} = \frac{3}{48} P L = .062 P L.$$

It is evident by inspection of Fig. 9 that the greatest moment will not be in the center of the rod, but nearer the end c. In order to find the point of maximum moment we must place the first differential coefficient $\frac{dm}{dx} = 0$.

$$\text{Now } dm = d\left[\frac{P}{6L^2}(L^2x - x^3)\right] = \frac{P}{6L^2}(L^2 - 3x^2)\,dx \text{ and}$$

$$\frac{dm}{dx} = 0 = \frac{P}{6L^2}(L^2 - 3x^2) \text{ or } L^2 \text{ must} = 3x^2 \text{ and } x^2 = \frac{L^2}{3}.$$

$x = \frac{L}{\sqrt{3}} = \frac{L}{1.73} = .58\,L$ for the distance of the point of greatest moment from end b. As connecting rods, however, are generally heavier in section at the crank end, we can assume this distance as .6 L, and calculate the maximum moment accordingly:

$$M_{max} = P\frac{.6\,L^3 - .216\,L^3}{6\,L^2} = .064\,P\,L.$$

Now if for P we substitute the value in equation 9, we have for the

Moment at center $= 1.6\,G\,s \times .062\,L = .099\,G\,s\,L.$
Moment at .6 L $= 1.6\,G\,s \times .064\,L = .102\,G\,s\,L.$

or, say, from $\frac{5}{10}$ to $\frac{6}{10}$ L, M = .1 G s L, and

$$f = \frac{.1\,s\,G\,L}{S} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (12)$$

which is one-half of value by formula 11 for parallel rods. Thus, if the rod cited above were to be used as a main rod, the fiber strain, due to vertical inertia, would be =

$\frac{.1 \times 28 \times 315 \times 90}{10.42} = 7,600$ pounds, when running at a speed in miles per hour equivalent to the diameter of the drivers in inches. (A further analysis of the forces affecting rods will be considered under the heading "Steam Action.")

RECIPROCATING PARTS.

As these parts have motion in a horizontal direction only, the effect of inertia upon them will be confined to this direction, alternating forward and backward with each stroke of the piston. The parts which are to be so considered should be strictly limited to the piston with rings, etc., complete; piston rod with keys and nuts, and crosshead with wristpin, etc. These parts never move out of a straight line, and all the forces of inertia brought into being by their momentum will be exclusively horizontal, as far as action upon themselves is concerned. With the main or connecting rod the case is different, one end of this moving in a horizontal line and the other end in a circle, as we have seen above. It will therefore be necessary to study separately the momentum or inertia effects of the reciprocating parts, pure and simple, and of the connecting rod. While the transverse inertia of this rod varies from 0 at crosshead end to a maximum at the crank end, it is at once evident that all parts of the connecting rod are subject to horizontal inertia, as every part of the rod partakes of a variable motion with a horizontal component.

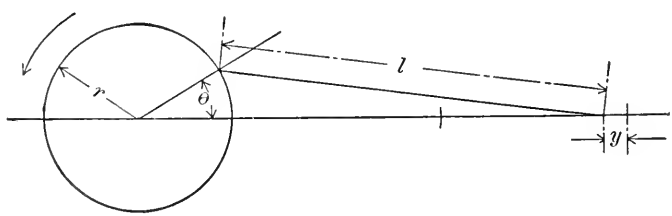


Fig. 10.

In Fig. 10, let the following letters designate the several parts mentioned, all values being in feet:

l = length of connecting rod,

r = radius of crank,

Θ = angle of crank at any instant,

y = distance of crosshead from commencement of stroke, then

$$y = r + l - \sqrt{l^2 - r^2 \sin^2 \Theta} - r \cos \Theta$$

and if we let $x = \frac{y}{r}$ represent the relation of crosshead position y to crank radius, we can write

$$x = \frac{y}{r} = 1 + \frac{l}{r} - \sqrt{\frac{l^2}{r^2} - \sin^2 \Theta} - \cos \Theta$$

The velocity of crosshead at any point y will be represented, relatively to the crank, by

$$\frac{dx}{d\Theta} = \frac{\sin \Theta \cos \Theta}{\sqrt{\frac{l^2}{r^2} - \sin^2 \Theta}} + \sin \Theta = \sin \Theta \left(\frac{\cos \Theta}{\sqrt{\frac{l^2}{r^2} - \sin^2 \Theta}} + 1 \right)$$

Now let V = velocity of crosshead,
v = velocity of crankpin,
t = time, then

$\frac{dV}{dt}$ = acceleration of crosshead.

The actual velocity of the crosshead will be

$$\frac{v\,dx}{d\Theta} = v \sin \Theta \left(\frac{\cos \Theta}{\sqrt{\frac{l^2}{r^2} - \sin^2 \Theta}} + 1 \right) = V \text{ and}$$

$$dV = v \left[\cos \Theta + \frac{\frac{r^2 \sin^4 \Theta}{d^2} + \cos 2\Theta}{\frac{l}{r}\left(1 - \frac{r^2 \sin^2 \Theta}{l^2}\right)^{\frac{3}{2}}} \right] d\Theta \quad *$$

---

*This reduction is as follows:

$$dV = v \left[d(\sin \Theta) \left(1 + \frac{\cos \Theta}{\sqrt{\frac{l^2}{r^2} - \sin^2 \Theta}} \right) + \sin \Theta \, d \left(1 + \frac{\cos \Theta}{\sqrt{\frac{l^2}{r^2} - \sin^2 \Theta}} \right) \right]$$

dividing both terms by dt, we have

$$\frac{d\,V}{d\,t} = v\left[\cos\Theta + \frac{\dfrac{r^2\sin^4\Theta}{l^2} + \cos 2\,\Theta}{\dfrac{1}{r}\left(1 - \dfrac{r^2\sin^2\Theta}{l^2}\right)^{\frac{3}{2}}}\right]\frac{d\,\Theta}{dt}$$

but $\dfrac{d\,\Theta}{d\,t} = \dfrac{v}{r}$, therefore

$$\frac{d\,V}{d\,t} = \frac{v^2}{r}\left[\cos\Theta + \frac{\dfrac{r^2\sin^4\Theta}{l^2} + \cos 2\,\Theta}{\dfrac{1}{r}\left(1 - \dfrac{r^2\sin^2\Theta}{l^2}\right)^{\frac{3}{2}}}\right]$$

$$= v\left[\cos\Theta\left(1 + \frac{\cos\Theta}{\sqrt{\dfrac{l^2}{r^2} - \sin^2\Theta}}\right) - \sin^2\Theta\,\frac{\dfrac{l^2}{r^2} - \sin^2\Theta - \cos^2\Theta}{\left(\dfrac{l^2}{r^2} - \sin^2\Theta\right)^{\frac{3}{2}}}\right]d\,\Theta$$

$$= v\left[\cos\Theta + \frac{\cos^2\Theta}{\left(\dfrac{l^2}{r^2} - \sin^2\Theta\right)^{\frac{1}{2}}} - \frac{\dfrac{l^2}{r^2}\sin^2\Theta - \sin^4\Theta - \sin^2\Theta + \sin^4\Theta}{\left(\dfrac{l^2}{r^2} - \sin^2\Theta\right)^{\frac{3}{2}}}\right]d\,\Theta$$

$$= v\left[\cos\Theta + \frac{\dfrac{l^2}{r^2}\cos^2\Theta - \sin^2\Theta\cos^2\Theta - \dfrac{l^2}{r^2}\sin^2\Theta + \sin^2\Theta}{\left(\dfrac{l^2}{r^2} - \sin^2\Theta\right)^{\frac{3}{2}}}\right]d\,\Theta$$

$$= v\left[\cos\Theta + \frac{\dfrac{l^2}{r^2} - \dfrac{l^2}{r^2}\sin^2\Theta + \sin^4\Theta - \dfrac{l^2}{r^2}\sin^2\Theta}{\left(\dfrac{l^2}{r^2} - \sin^2\Theta\right)^{\frac{3}{2}}}\right]d\,\Theta$$

and as the forces producing acceleration are proportional to the accelerations produced, we can write $P = \frac{G}{g} \times \frac{dV}{dt}$. (P being the force of acceleration and G the weight of the body) and

$$P = \frac{G v^2}{g r} \left[\cos \Theta + \frac{\frac{r^2 \sin^4 \Theta}{l^2} + \cos 2 \Theta}{\frac{l}{r} \left(1 - \frac{r^2 \sin^2 \Theta}{l^2} \right)^{\frac{3}{2}}} \right]$$

As this value of P depends largely upon the value of cos Θ, which is greatest at values of Θ = 0° and 180°, at which points sin Θ is small, we can omit the fractions

$$\frac{r^2 \sin^2 \Theta}{l^2} \text{ and } \frac{r^2 \sin^4 \Theta,}{l^2}$$

and write this equation more simply—

$$P' = \frac{G v^2}{g r} \left[\cos \Theta + \frac{r \cos 2 \Theta}{l} \right] \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (13)$$

where P′ = the effect of inertia for the reciprocating weights only.

If we consider the case of a connecting rod of infinite length, so that $\frac{r}{l} = 0$, we have

---

$$= v \left[\cos \Theta + \frac{\frac{r^2 \sin^4 \Theta}{l^2} - 2 \sin^2 \Theta + 1}{\frac{l}{r} \left(1 - \frac{r^2 \sin^2 \Theta}{l^2} \right)^{\frac{3}{2}}} \right] d\text{-}\Theta$$

$$= v \left[\cos \Theta + \frac{\frac{r^2 \sin^4 \Theta}{l^2} + \cos 2 \Theta}{\frac{l}{r} \left(1 - \frac{r^2 \sin^2 \Theta}{l^2} \right)^{\frac{3}{2}}} \right] d \Theta$$

as $1 - 2 \sin^2 \Theta = \cos 2 \Theta$

$$P'' = \frac{G v^2}{g r} \cos \Theta \quad \ldots\ldots\ldots \quad (14)$$

which we recognize as the horizontal inertia of a revolving weight, found in our study of Fig. 7, and replacing the sine by the cosine, as the force now under investigation is at right angles to the one then being considered.

We are now able to analyze the case of the connecting rod, which has a combined movement. The horizontal inertia of that portion at the crank end will equal P″ and at the crosshead end P′, if we let G represent the weight of the portion being considered. Any part or weight having a motion between the two, as, for instance, a part of the rod at a distance x from the crank end, will be subject to horizontal inertia forces =

$$P_x = \frac{G v^2}{g r} \left[\cos \Theta + \frac{x}{l} \frac{r \cos 2 \Theta}{l} \right]$$

If now we consider the whole weight of the rod acting at its center of gravity at a distance d from the crank end, we have for the horizontal component of the forces on the rod

$$P''' = \frac{G v^2}{g r} \left[\cos \Theta + \frac{d r}{l^2} \cos 2 \Theta \right] \quad \ldots\ldots\ldots \quad (15)$$

At the ends of the stroke, the force will be at its maximum, and can be determined by inserting the proper trigonometrical values; thus, for the front end of stroke $\Theta = 0°$, and for $\Theta = 0°$

$$P = \frac{G v^2}{g r} \left(1 + \frac{dr}{l^2} \right) \quad \ldots\ldots\ldots \quad (16)$$

and for the back or crank end of stroke, $\Theta = 180°$, and for $\Theta = 180°$,

$$P = \frac{G v^2}{g r} \left(1 - \frac{dr}{l^2} \right) \quad \ldots\ldots\ldots \quad (17)$$

Also in formula 15, if the load be located at the crank end entirely, or $d = 0$, we have for $d = 0$, $P = \frac{G v^2}{g r} \cos \Theta$, same as equation 14, and if it be concentrated at the crosshead end, d = l,

$$P = \frac{G v^2}{g r} \left\{ \cos \Theta + \frac{r}{l} \cos 2 \Theta, \right\}$$

same as equation 13.

From equations 16 and 17 we can also eliminate d by making it equal to o or l, whereby they become for

$$\left.\begin{array}{ll} \Theta = 0^\circ & \Theta = 180^\circ \\ d = 0,\ P = \dfrac{G v^2}{g r} & P = \dfrac{G v^2}{g r} \\ d = l,\ P = \dfrac{G v^2}{g r}\left(1 + \dfrac{r}{l}\right) & P = \dfrac{G v^2}{g r}\left(1 - \dfrac{r}{l}\right) \end{array}\right\} \ldots (18)$$

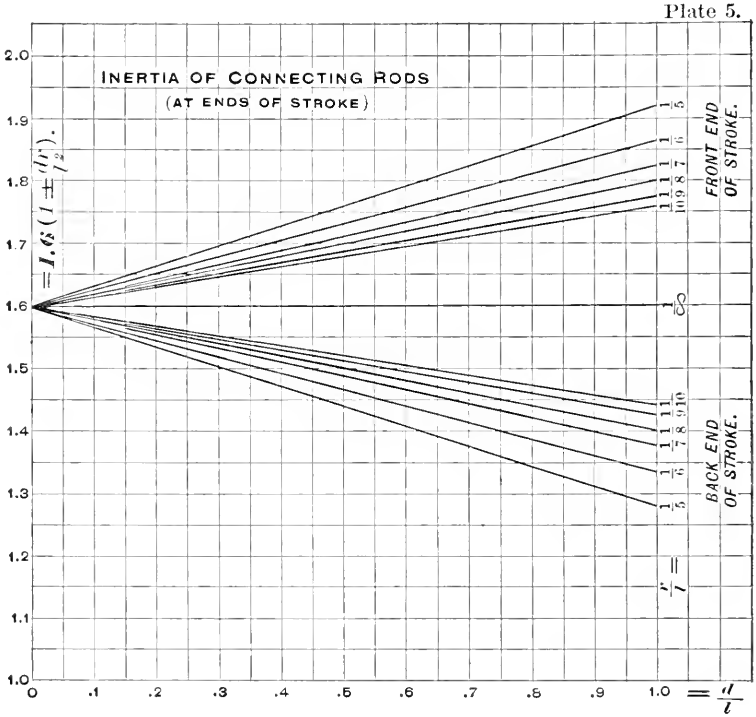
Now, we know that $\frac{G v^2}{g r}$ is the centrifugal force, and using the nomenclature in obtaining equation 9, we can write $\frac{G v^2}{g r} = 1.6\, G\, s \frac{V^2}{D^2}$, and substitute it in the above formulæ. If we consider the maximum case to be where V = D, we have for the several cases just discussed the following values for

P .... (19)

	Front end of stroke $\Theta = 0^\circ$	Back end of stroke $\Theta = 180^\circ$
Revolving weight d = o	1.6 G s	1.6 G s
Connecting rod d = d	$1.6\, G\, s \left\{1 + \frac{d\, r}{l^2}\right\}$	$1.6\, G\, s \left\{1 - \frac{d\, r}{l^2}\right\}$
Reciprocating weight d = l	$1.6\, G\, s \left\{1 + \frac{r}{l}\right\}$	$1.6\, G\, s \left\{1 - \frac{r}{l}\right\}$

We will now take some practical examples of these formulæ and try and realize what they mean. Plate 5 is a graphical representation of formula 19, and gives the values of 1.6 $\left(1 \pm \frac{d\, r}{l^2}\right)$, or what might be termed the coefficient of G s, so that in order to find the inertia at the end of stroke, it is simply necessary to multiply together the weight of the part or parts in question in pounds, the stroke in inches, and the value of the coefficient, as found from plate 5. The proper line or locus must be selected which corresponds to the ratio of crank radius to length of connecting rod. They are given for ratios 1-5 to 1-10, as these will cover the ordinary cases of modern prac-

Plate 5.



tice. The horizontal line is for a rod of infinite length, and marks the boundary between variations due to a finite length of rod.

In selecting the proper value of $\frac{d}{l}$, consideration must be given to the motion of the part being examined. As d is the distance of the center of gravity of the body, from the crank end, if the parts are all revolving, d will have no value, and the coefficient is simply 1.6. If they are purely reciprocating, such as crosshead, piston, etc., the value of d must be equal to l, or $\frac{d}{l} = 1$. This value must be taken for such parts as are just mentioned, even if a greater distance than l from the crank, as they have a motion identical with the crosshead wristpin, which point on the rod is distant l from the crankpin. The connecting rod itself will have an intermediate value, to obtain which we proceed as follows:

Weigh each end of the rod separately upon a platform scale, taking care that both ends are supported so that the

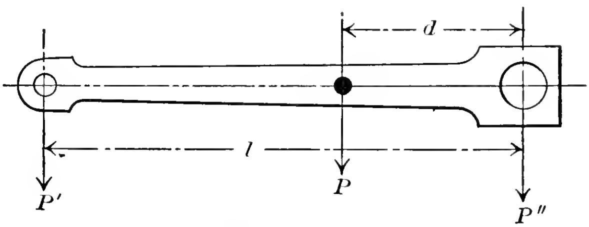


Fig. 11.

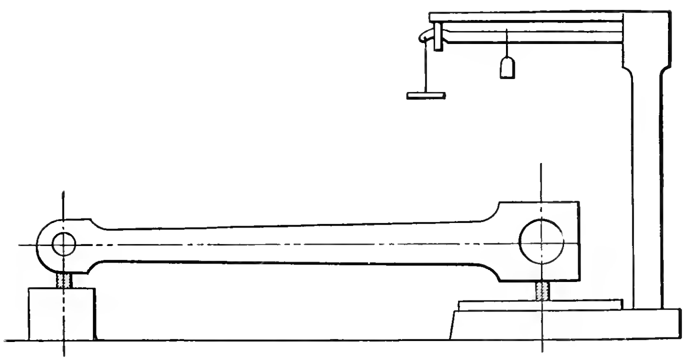


Fig. 12.

center line of rod is horizontal; also, that a narrow block or strip supports both ends directly under the pin center.

When balanced, as shown in Fig. 12, we will have the weight P'' shown in Fig 11. Reverse the rod, and obtain P'. Then the total weight $P = P' + P''$. Now, to obtain d, multiply the weight P' by the length of rod and divide by the total weight P. Expressed by an equation, we have

$$d = \frac{P' l}{P' + P''} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (20)$$

which gives the location of the center of gravity. The proper value of $\frac{d}{l}$ on the diagram must be taken, evidently to correspond with the distance just found, divided by the length of rod.

The connecting rod of a simple 10-wheel locomotive weighs 142 pounds at the crosshead end and 285 pounds at the crank end. The total weight is 427 pounds and the center of gravity is at $\frac{142 \times 9}{427} = 3$ feet from the crank end, the rod being 9 feet between centers. As the stroke is 24 inches, the crank radius is 1 foot, therefore we have $\frac{r}{l} = \frac{1}{9}$ and $\frac{d}{l} = .333$. The piston and rod weigh 308 pounds, and the crosshead 183 pounds, or a total of 491 pounds reciprocating weight, purely. Now follow the lines of $\frac{r}{l} = \frac{1}{9}$ to value .333 of $\frac{d}{l}$, and we find the coefficients = 1.66 at front end and 1.54 at back end; therefore, multiplying, we have

$1.66 \times 427 \times 24 = 17{,}011$ pounds at front end for rod
$1.54 \times 427 \times 24 = 15{,}781$ pounds at back end for rod.

For the reciprocating parts, where $\frac{d}{l} = 1$, we take the right-hand ends of the same lines, which we find at 1.78 and 1.42, therefore we have

$1.78 \times 491 \times 24 = 20{,}975$ lbs. at front end for recip. weights.
$1.42 \times 491 \times 24 = 16{,}733$ lbs. at back end for recip. weights.

The total thrust (or pull) on crankpin at each end of stroke, due to inertia, will be the sum of these, or

$$17{,}011 + 20{,}975 = 37{,}986 \text{ at front end.}$$
$$15{,}781 + 16{,}733 = 32{,}514 \text{ at back end.}$$

(This does not allow for compression, of which we will treat later, but this would be correct for drifting at the speed considered to be the regular maximum.)

Thus there is found to be a difference of 5,472 pounds between the two ends of the stroke, and this difference is due entirely to the angularity of the connecting rod, which augments the inertia at the front end and reduces it at the back end.

By the formula 19 we get the same result. For the rod

$$1 \pm \frac{d\,r}{l^2} = 1 \pm \frac{3 \times 1}{9^2} = 1 \pm \frac{3}{81} = 1 \pm .037 =$$

1.037 and .963 and 1.6 times these values $= 1.037 \times 1.6 = 1.66$, and $.963 \times 1.6 = 1.54$, these, multiplied with 427 and 24 (the values of G and s), giving 17,011 and 15,781, as already found.

So, for the reciprocating weights,

$$1 \pm \frac{r}{l} = 1 \pm \frac{1}{9} = 1 \pm .11 = 1.11 \text{ and } .89,$$

and multiplied by $1.6 = 1.78$ and 1.42, and further by 491 and 24, give 20,975 and 16,733. This shows the importance of keeping down the weight of rod, piston, crosshead, etc., to a minimum, as the inertia is directly proportional to these weights; it also shows that the longer the connecting rod, the smaller will be the maximum thrust, and the less the difference between the imposed load at the two ends of the stroke. Even with the light parts of the case just assumed, there is a load of 16 tons applied at each end of the stroke, when the speed equals the diameter of the drivers.

In another case, we have a compound (4-cylinder) engine, otherwise the same as that above quoted; but the reciprocating weights (not including any part of main rod) amount to 1,209 pounds. Treating this in the same way, we find a total inertia of 68,500 pounds at front end of stroke and 57,100 pounds at back end. These figures are nearly double those for the simple engine, and the increase is due entirely to the heavy crosshead, double pistons, etc. The effect of this on the counterbalance will be considered under the proper heading.

Let us now consider the balanced engine, with four cylinders, two of which drive through an outside crankpin and two through a crank axle, the pin being opposite or 180° from the contiguous axle crank. In an example of this kind we have the weights:

	Inside cylinder.	Outside cylinder.
Piston and rod	356 lbs.	463 lbs.
Crosshead, etc.	310 lbs.	310 lbs.
Reciprocating parts	666 lbs.	773 lbs.
Connecting rods	552 lbs.	579 lbs.

The stroke is 26 inches, or crank radius 1.08 feet, the length of main rod 7 feet, with the center of gravity 1.9 feet from crank end, therefore $\frac{r}{l} = .155$ and $\frac{d}{l} = .27$. Proceeding with the calculations as already explained (and which have been made by a slide rule in this and the last example, which accounts for the cyphers in the last two places), we find for the inertia:

	Inside cylinder.		Outside cylinder.	
	Front end.	Back end.	Front end.	Back end.
Connecting rod	23,900	21,900	25,200	23,100
Reciprocating parts	31,900	23,400	37,100	27,100
Total	55,800	45,300	62,300	50,200

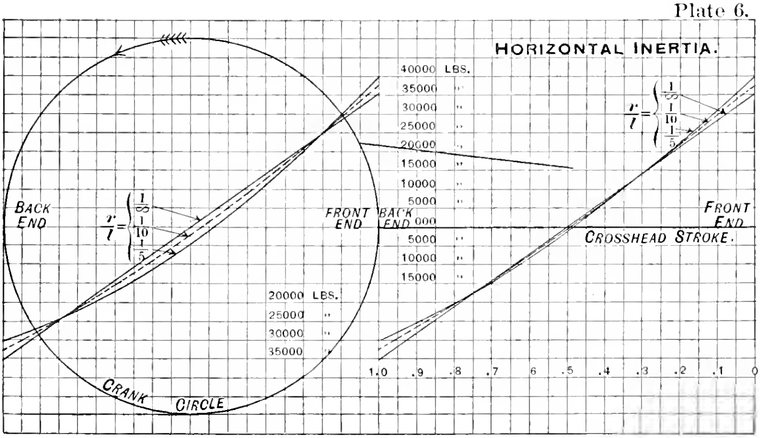
Now, in this balanced engine, when the inside crank is at front end, the outside pin is at back end, and vice versa, so we will have the total effect on the engine itself for

Inside front and outside back = 55,800 — 50,200 = 5,600 lbs.
Outside front and inside back = 62,300 — 45,300 = 17,000 lbs.

Of course, the effect on the pin and crank is as given next above but one. The effect upon the locomotive as a whole, however, in the several cases quoted, will be about twice as much for the simple as for the balanced engine and about twice as much for the 4-cylinder compound as for the simple locomotive. It is also apparent that, while the sum of the effects of the main rod and the reciprocating parts will come upon the crankpin, the crosshead wristpin will have to take that due to the reciprocating parts only, and as these forces have large values at high speeds, we can readily understand the pounding of these parts, when running down hill at great velocity, if there be any lost motion in the bearings on the pins, and if the compression be insufficient to take up this force of inertia at the end of the stroke, when the direction of motion reverses.

We have given most of our attention to the inertia at the end of stroke, as it is a maximum at that point, but by making

Plate 6.



a graphical representation of formula 15 we can see clearly its effect throughout the stroke. This has been done in plate 6 for a simple engine, same as above considered; that is, with a connecting rod weighing 427 pounds, reciprocating parts weighing 491 pounds, and with a stroke of 24 inches, the center of gravity being $\frac{1}{3}$ from crank end. In order to show the effect of length of connecting rod upon the inertia, we have given three curves; one (a straight line) for rod of infinite length; one for a rod 10 times the crank radius (dotted line), and one for a rod five times the crank radius (solid line). The left-hand diagram shows the inertia laid off vertically upon the projected crank circle as a base line, the crank revolving as shown by the arrow, and the front or crosshead end, supposed to be at the right. The right-hand diagram shows the inertia forces laid off on the crosshead stroke as a base, the actual positions of the crosshead, as determined by the length of connecting rod, being used to lay off the vertical values of inertia. For a rod of infinite length, the forces are the same as in the first or left-hand figure, and are identical at the two ends of the stroke. The force above the center or base line is supposed to be acting ahead, that is, opposed to the motion of the crosshead, and the force below, to assist or act with the crosshead, when the crosshead is moving on its backward stroke, as indicated by the arrow. As the motion reverses at the end of the stroke, and the inertia forces reverse their direction at or near mid-stroke, it is evident that the assistance of inertia is changed to resistance, the instant the crosshead reverses its direction of motion. Inertia therefore assists the piston during the last half of the stroke, and opposes it during the first half.

The shorter the main rod, the greater are the forces at the front end and the less at the back end of stroke; and the angularity of the rod throws the crosshead back, so that all the loci or curves in the right-hand figure cross the neutral line almost at the center of stroke. For practical purposes we can therefore assume that the inertia of connecting rod and reciprocating parts is 0 at the center of the stroke, and increases uniformly to the end of the stroke, at which point it reaches the value given in equations 19, and that it may be approximately represented by a straight line from the center of the stroke to an ordinate representing its maximum value at the end of the stroke. The force is in no sense a "hammer blow," but increases from nothing to its maximum twice in each revolution, and, as the speed assumed requires 336 revolutions, there will be nearly 700 such variations in a minute, and in alternate directions. In some instances tubular piston rods have been used in order to reduce the weight, and much study has been given to the production of a light but strong piston. All piston attachments must be securely made, such as nuts, bolts, keys, etc., as at each end of the stroke, at the maximum speed, every piece tends to continue its motion beyond that imposed by the crank, with a force about 40 times its weight. This demonstrates the necessity for tightly securing the different parts of the mechanism which we have been considering.

We have seen in equation 18 that the inertia of these parts varies with the square of the velocity, so that if we wish to consider the effects at half of the speed, we must take one-quarter of the forces at the original speed. Thus, in the figures of plate 6, the velocity is assumed to be equal to the diameter of the drivers in inches; if the velocity were but one-half of this, the end forces would be only about 9,000 pounds instead of 36,000 pounds, as illustrated.

There is still another force that comes into play in a horizontal direction, but while it is a force due to inertia, it is not due to reciprocating action—this is the centrifugal force due to the swinging action of the main rod, and which constantly acts along the axis of this rod and in a backward direction, that is, toward the rear of the engine. Let us refer to formula 5, $C = \frac{G v^2}{g r}$, which gives the centrifugal force of a body, and also to Figs. 10 and 11. As centrifugal force is measured by the velocity and radius of action of the center of gravity of the body, we will have to determine these for the rod oscillating about the crosshead pin. It is evident that the rod vibrates at its greatest angular speed at the ends of the stroke, and that at this moment the crank end of rod has the same velocity as the crankpin, v. The center of gravity of rod being at d from crank end or $l - d$ from crosshead end, will have a reduced velocity $= v \frac{l - d}{l}$; also the radius of action will be $l - d$.

Substituting these values, we have $C = \frac{G v^2 (l - d)^2}{g (l - d) l^2} = \frac{G v^2 (l - d)}{g l^2}$, where v still represents the velocity of crankpin.

This force always acts backwardly or negatively. We saw in equation 16 and 17 that the inertia of the main rod (horizontally) was increased at the front end of stroke and diminished at back end by a quantity $\frac{G\,v^2}{g\,r} \times \frac{d\,r}{l^2}$, and equation 15 showed that this acted in both cases ahead, or positively, it appearing under the negative sign in equation 17 as the main force of inertia in this case is really negative, as it acts toward the rear.

From this it is evident that the centrifugal force of the main rod will tend to neutralize the irregular effect just referred to. In order to discuss this action, we write

$$\frac{G\,v^2}{g\,r} \cdot \frac{d\,r}{l^2} - \frac{G\,v^2}{g\,r} \cdot \frac{r\,(l-d)}{l^2} = \frac{G\,v^2}{g\,r}\left(\frac{d\,r}{l^2} - \frac{r\,(l-d)}{l^2}\right) =$$

$$\frac{G\,v^2}{g\,r}\left(\frac{d\,r - r\,l + d\,r}{l^2}\right) = \frac{G\,v^2}{g\,r} \cdot \frac{2\,d\,r - r\,l}{l^2},$$

which is the difference between the two forces, or the amount that is not neutralized. It will be seen at once, that this value depends entirely upon the second portion of the formula, as the first is constant for our present purpose. If we make this equal to zero, we have the condition of complete neutralization, or $r\,\frac{2\,d - l}{l^2} = 0$, for which we see that 2 d must equal l or $d = \frac{1}{2}\,l$, that is, the center of gravity should lie at the middle of the rod. If it be nearer the crank end, as is usual, 2 d will be less than l, and the result will be negative, that is, the centrifugal force more than neutralizes the irregularity of the horizontal inertia of the rod, but of course not of the reciprocating parts. If we take the case of the main rod of the simple engine, we have d = 3, l = 9 and r = 1, so that $-\frac{r\,(l-d)}{l^2} = -\frac{6}{81} = -.074$. This will change the coefficients of $\frac{G\,v^2}{g\,r}$ or

1.6 G s, as follows:

Front end, 1 + .037 — .074 = .963
Back end, 1 — .037 + .074 = 1.037

so that the total force in the rod will be at

Front end, 1.6 G s × .963 = 15,781 pounds.
Back end, 1.6 G s × 1.037 = 17,011 pounds.

This will therefore overbalance or neutralize a portion of the reciprocating parts. The total effect, including those parts, would be

Front end, 15,781 + 20,975 = 36,756
Back end, 17,011 + 16,733 = 33,744

a difference of only 3,012 pounds, instead of 5,472 pounds when this centrifugal force is not considered.

The balanced engine would give as a total

Inside cylinder—Front, 53,235; back, 47,865.
Outside cylinder—Front, 59,600; back, 52,900.

and for

Inside front and outside back = 53,235 — 52,900 = 335 lbs.
Outside front and inside back = 59,600 — 47,865 = 11,735 lbs.

COUNTERBALANCE.

We will now consider the question of counterbalance, and this must include both vertical and horizontal forces, as the counterbalance moves in a path which is circular in relation to the engine, and therefore exerts inertia forces in various directions, but, of course, in a vertical plane. A great deal of study has been given to this subject, and to good purpose, as its effects at modern high speeds are much too important to be neglected.

The counterbalance may be justly considered a "makeshift," for the reason that by it we attempt to neutralize certain forces, acting in a horizontal direction, by means of a revolving body. It is unnecessary to state that a perfect balance by this means is impossible, and at the best we must be satisfied with an arrangement that will give us only approximately what we desire.

In the preceding sections, we have studied the vertical effect of the motion of the rods upon themselves, and the horizontal effect of the reciprocating parts upon the crank and its connections. We must now determine the effect of these parts upon the engine as a whole, and upon the track and bridges which support it. In addition, the purely revolving bodies, such as crank, crankpin, hub, etc., must be investigated as far as their general influence on this question is concerned.

In formula 19 and plate 5, we have found that the reciprocating parts, as well as the connecting rod, exert large strains upon the various parts of the engine at the ends of the stroke, and plate 6 illustrated calculations made upon an actual locomotive. The general effect upon the engine is to cause what is commonly called "nosing," or occasionally "one side trying to get ahead of the other side." If the inertia of these forces acted at the center line of the engine, there would be no "nosing," but a tendency to leap ahead or lag behind at alternate strokes, but as the cylinders are from 40 to 45 inches from the center line of the engine, there is exerted a moment equal to the force of inertia by the lever arm mentioned. If the engine be working steam, the lead may be so arranged as to cushion the force of the rods and take up the lost motion on the pins before the end of the stroke, when the force is at its maximum. This cannot entirely dispense with the nosing, although it may destroy the bad effect upon the pins. It is also a fact that sometimes the worst cases of nosing occur when a large cylinder engine is working slowly with a full throttle, but this is due to the action of steam against the cylinder heads.

For every revolution, then, we will have four maximum forces, first, ahead, right side, then ahead, left side; back, right side, and last, back, left side, assuming that the right crank leads. This sequence of forces follows at every revolution, and keeps up a continual disturbance of the locomotive in a horizontal plane. In addition to the weight of the rods, etc., the crankpin and hub assist in the nosing action, to an amount 1.6 G s, as shown in equation 19 for revolving weights.

One means of effecting a balance has been illustrated, that is, by using four cylinders, like the Shaw locomotive of a few years ago, or, more recently, the Vauclain balanced compound, but the wisdom of thus increasing the complication is questioned. Even then, it is impossible to obtain a perfect balance on account of the angularity of the connecting rod, as has been demonstrated by formula 19. If, however, the vertical effect upon the track and structures did not have to be considered, we could obtain a very satisfactory horizontal balance by using a weight directly opposite to the crank, and if considered at the same radius as the crank, having a value equal to the sum of the reciprocating parts, the connecting rod and the crankpin and hub. This, however, would give a large excess balance when considered vertically, as to oppose this counterweight we have only the crankpin and hub, and the vertical effect on the crankpin due to the oscillation and angle of the connecting rod. (In the case of a coupled engine, the parallel or side rods constitute revolving weights in addition to the crankpin and its hub.)

In the demonstration under Fig. 7 we have seen that the vertical inertia of a revolving body is greatest when the body is 90 degrees or 270 degrees from the horizontal zero; in other words, at the top or bottom, and that it is then equal to the centrifugal force, which at the speed assumed to be our maximum, is about 40 times the weight. This would mean a downward pressure, when the balance was at the bottom, of 40 times the excess weight, which would be in addition to the static weight of the engine passing normally through the wheel, and when at the top of its path, the upward tendency would be the same, resisted by the weight on the rail. If this normal load were 20,000 pounds, then divided by 40 we have 500 pounds of reciprocating weight, which could be balanced without actually causing the wheel to leave the track, and this would also momentarily double the load on the rail when the balance was down—both constituting a very serious condition of affairs.

Having made this preliminary study of cause and effect, let us ascertain what the most recent investigations in the matter suggest as the proper course to pursue, and what results we obtain by following these recommendations.

In 1896 the author presented to the Association of Engineers of Virginia a paper on "Locomotive Counterbalancing," making certain recommendations which were later indorsed by a committee of the Master Mechanics' Association and embodied in their final report. Three cardinal principles were announced, as follows:

(1) The amount of reciprocating weight that can be left unbalanced may be a definite function of the total weight of the engine.

(2) The total pressure of wheel upon the rail must not exceed a certain definite amount, depending upon the construction of bridges, weight of rail, etc.

(3) The vertical influence of the excess balance must never be sufficient to lift the wheel from the rail.

Let us consider principle number one. We have just seen that, to reduce as much as possible the vertical influence of the counterbalance upon the rail, the excess balance must be the minimum allowable consistent with good practice. By excess balance is meant so much of the counterbalance over and above that necessary to balance the revolving weights or those producing vertical forces of acceleration and retardation. Revolving weights can be perfectly balanced by other revolving weights, so that if we have a partial balance the force resulting from revolution will be the difference in weight by the proper function. Thus, if G_1 be the counterbalance at radius r_1 and G_2 the revolving weights at radius r_2, the centrifugal force of each will be (from formula 5), $C_1 = \frac{G_1 v_1^2}{g r_1}$ and $C_2 = \frac{G_2 v_2^2}{g r_2}$, and the resultant pressure in one direction $C_1 - C_2$. The angular velocities $\frac{v_1}{r_1}$ and $\frac{v_2}{r_2}$ must be the same, as they are on the same wheel, so that by denoting the angular velocity by ω, we can write these equations

$$C_1 = \frac{G_1}{g} \omega^2 r_1 \text{ and } C_2 = \frac{G_2}{g} \omega^2 r_2,$$

and by reducing to radius r_2, C_1 becomes

$$\frac{G_1}{g} \omega^2 r_1 \frac{r_2}{r^2} = \frac{G_1 \frac{r_1}{r_2} \omega^2 r_2}{g}$$

now $G_1 \frac{r_1}{r_2}$ is the counterbalance reduced to the crank radius r_2,

and $C_1 - C_2 =$

$$\frac{G_1 \frac{r_1}{r_2}}{g} \omega^2 r_2 - \frac{G_2}{g} \omega^2 r_2 = \left(G_1 \frac{r_1}{r_2} - G_2 \right) \frac{\omega^2 r_2}{g},$$

and as $G_1 \frac{r_1}{r_2} - G_2$ is the difference of the weights reduced to the crank radius, we find that the resultant effect is the difference of the weights (reduced to crank radius) multiplied by $\frac{\omega^2 r_2}{g} = \frac{V_2^2}{g r_2}$, or the centrifugal force of the difference in the weights, reduced to the crank radius. As we have seen, the excess balance is for the purpose of neutralizing the effect of the reciprocating parts, so that by reducing these weights to a minimum, and by balancing as small a proportion as possible, we obtain the smallest excess balance, and therefore the corresponding lowest rail pressures.

We should make every allowable effort to reduce the weights of piston, piston rod, crosshead and main rod. The tables of areas and moments of inertia show the advantage of the I section over the rectangular for the main rod, and the tubular over the solid for the piston rod. The piston and crosshead may be constructed largely of cast steel, so as to obtain the lowest possible weight consistent with the necessary strength. The importance of giving special attention to these points cannot be overestimated.

As a given weight of reciprocatng parts will produce a certain force at a given speed, it is evident that the greater the weight of the locomotive as a whole, the smaller will be the effect of the disturbing forces—and, therefore, the larger amount of reciprocating weight that may be left unbalanced. The author considered that $\frac{1}{360}$ of the weight of the locomotive as a whole could be left unbalanced on a side without seriously affecting the engine or causing undue nosing. (The weight of the tender is, of course, not included in the weight of the engine.) The Master Mechanics' committee saw fit to reduce this unbalanced weight to $\frac{1}{400}$ of the weight of the locomotive, and this value has been largely used since their report was made.

Principle number two hardly needs any elaboration, as it is self-evident. However, attention should be called to the fact that, if a total allowable pressure be established, and we are able by some means or other to reduce the excess balance, we can permit a greater static weight upon the drivers, without injury to the track and bridges. This is the principal claim for superiority made for the Shaw and Vauclain balanced locomotives, and it is worth proper consideration.

When this subject was discussed with the engineering department of the Norfolk & Western Railway, at the time the paper was written, it was considered that the total rail pressure per wheel, including the static weight and the centrifugal force, should not exceed

28,000 pounds for 4—4—0 engines
26,000 pounds for 4—6—0 engines
25,000 pounds for 2—8—0 engines

the loads being per wheel and not per axle.

Another point to be considered is, that while the adhesive weight of a locomotive is generally intended to be divided equally between the different driving wheels, there are actually few cases where this has been accomplished. In case the combined static weight and centrifugal load upon the wheels which carry the greatest weight exceed the limit allowed, a portion of the balance may be carried by one of the wheels of lesser loading; this will result in a division of the balanced reciprocating weight which will not be uniform, but as it will not affect the riding of the engine, and will reduce the maximum rail pressure, it is perfectly allowable.

The third principle is also axiomatic, but in practice it is not wise to permit the upward centrifugal force to approach nearly to the weight on the wheel. In order to insure that this will not occur, it is well to limit this force at a speed in miles per hour which equals the diameter of the drivers to 75 per cent of the static load. There will then be sufficient weight to maintain the wheel solidly upon the rail. By this means we avoid the "hammer blow" so often erroneously referred to. Locomotives have been run in which the driving wheels have actually left the rails at high speeds, and in this case there was a veritable hammer blow. In 10-wheel engines, and those in which the main rod does not take hold of the front drivers, there will be a lifting tendency on the lower quarter, due to the steam action and the angularity of the rod. This can be readily estimated by calculations, and some experiments made by blocking an engine on a track scale showed a decrease of 5,000 pounds on the front drivers. Of course, this diminishes the load to be overcome by the centrifugal force of the counterbalance in order to lift the front wheels.

The revolving weights can be completely balanced, and this should always be done in each individual wheel. The reciprocating balance may be shifted from an overloaded to an underloaded wheel, but the revolving weights must be balanced in each wheel. On each wheel, except the main drivers, the revolving weight consists of the weight of the side rods bearing on that particular pin, the crankpin, with its washer and nuts, if used, and the boss in the wheel center or the crank cheeks and journal of a crank axle. The weight of the rods (parallel) can be found by supporting each end on a scale platform, in the same position as found on the engine, as indicated in Fig. 13.

The weight of the pin and collars may be obtained before forcing in the wheel center, or by calculation. The crank hub must be figured from its center of gravity, by taking the portion outside of the axle hub, and reducing it to the crank radius, by multiplying its weight by the distance of its center of gravity from the axle center, and dividing by the crank radius. This will have to be estimated from the drawing or obtained from the pattern of the crank. The sum of the rod weight (on the pin), the crankpin, and the crank hub (reduced

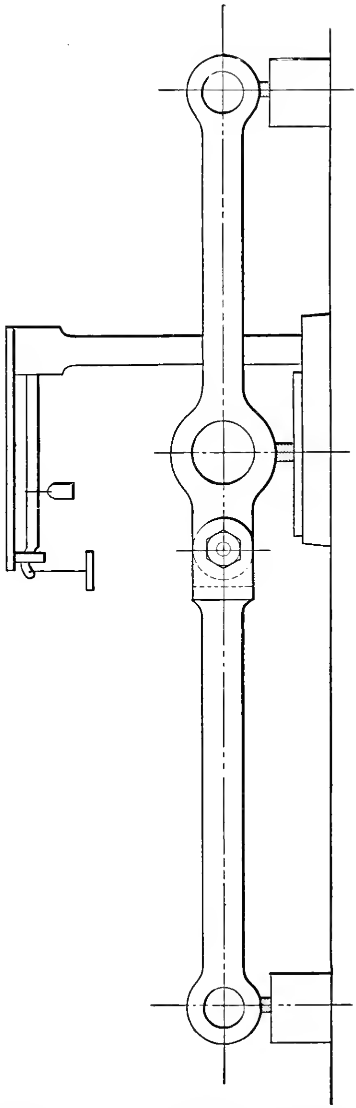


Fig. 13.

as explained), may be represented by G_2, as in our last formula, and the crank radius by r_2. If G_1 represents the weight of the balance acting at the radius r_1 (that is, the distance of its center of gravity from the axle center), and assuming an angular velocity for both $= \omega = \frac{v}{r}$, we have for equal centrifugal force (and therefore equal balance) $G_1 r_1 \frac{\omega^2}{g} = G_2 r_2 \frac{\omega^2}{g}$ and $G_1 r_1 = G_2 r_2$, or $G_1 = G_2 \frac{r_2}{r_1}$; that is, the moments of the weights and the balance about the axle center must be equal.

This gives the weight of counterbalance to take care of

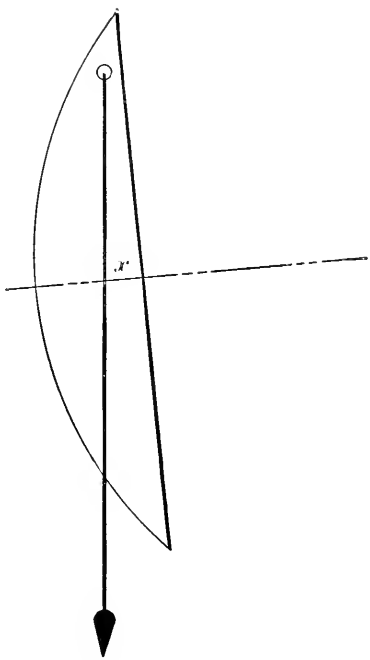


Fig. 14.

the revolving weights only, but more must be added to take care of a portion of the reciprocating parts. All over the amount just found will be termed "excess balance."

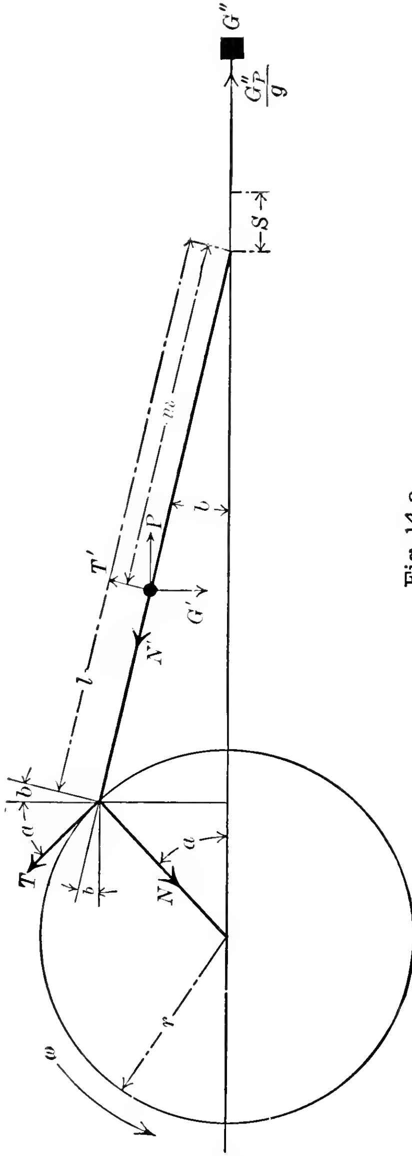


Fig. 14 a.

It will be convenient to consider the balance as reduced to the crank radius in what follows, or equal to $G_1 \frac{r_1}{r_2}$, but it must be borne in mind that the actual balance can always be found by multiplying by the crank radius and dividing by the distance of the center of gravity of the balance from the center of the axle.

Before the wheel is cast, the best way to determine the location of the center of gravity of the counterbalance is to make a template of the proposed balance, full size, in soft wood, about ¼ inch thick. By suspending one corner loosely on a nail and dropping a plumb line from the center of the nail, the intersection of the plumb line and a center line of the balance marks the center of gravity of the counterbalance. Thus, in Fig. 14, x denotes the center of gravity. The farther distant this is from the axle center, the smaller will be the required balance weight.

When the main wheels are to be considered, we must add a certain amount to balance the vertical effect of the connecting rod, which is a maximum at the top and bottom quarters.

Mr. R. A. Parke, in a paper before the New York Railroad Club, presented a mathematical discussion of this subject, which we will use as a partial guide in our analysis.

In connection with Fig. 14a we will use the following symbols:

G' = weight of connecting rod in pounds.
G'' = weight of piston, piston rod and crosshead in pounds.
r = radius of crank in feet.
l = length of rod in feet.
m = distance of center of gravity of rod from crosshead pin in feet.
k = radius of gyration of rod about crosshead pin in feet.
a = angle of crank with horizontal center line.
b = angle of rod with horizontal center line.
ω = angular velocity of crank.
Φ = angular velocity of rod about crosshead pin.
Θ = angular acceleration of rod.
p = lineal acceleration of crosshead.

The rod has two separate component motions—one of horizontal translation, and the other of oscillation about the crosshead pin. The crank revolves with a velocity which, at the instant of consideration, is uniform. As the rod is moved by the crank (it must be remembered that we are not considering steam action, but the acceleration of the rod as governed by the crank), there is always a force between the rod and the crankpin, and this force may be resolved into two components, a normal force N, acting in the line of the crank, and a tangential force T, at right angles to the crank.

At any instant, the external forces, acting at the two ends of the rod, must be in equilibrium with the forces generated by the inertia of the rod itself, in any plane under consideration. In addition to the normal and tangential forces acting at the crankpin, there is the force of inertia of the reciprocating parts acting at the crosshead pin, $\frac{G''}{g} p$, p being the variable acceleration of the crosshead, etc. The forces generated by the inertia of the rod itself, or the internal forces, as they may be called, are the oscillating inertia $T' = \frac{G'}{g} \Theta m$, Θ being the variable angular acceleration; the centrifugal force $N' = \frac{G'}{g} \Phi^2 m$, Φ being the variable angular velocity of the rod about the crosshead pin; and the horizontal inertia $P = \frac{G'}{g} p$. These, being reduced to a horizontal direction, can be placed equal to each other, thus:

$$T \sin a + N \cos a - \frac{G''}{g} p = \frac{G'}{g} \Theta m \sin b - \frac{G'}{g} \Phi^2 m \cos b + \frac{G'}{g} p,$$

the first member being the external, and the last the internal forces.

The moments of the external forces about the crosshead pin must also equal the moments of inertia and horizontal acceleration, which are $\frac{G'}{g} \Theta k^2$ and $\frac{G'}{g} p m \sin b$, respectively, and

$$T\,l\cos(a+b) - N\,l\sin(a+b) = \frac{G'}{g}\Theta k^2 + \frac{G'}{g} p\,m\sin b.$$

We can eliminate T by solving each equation for T and setting down one to equal the other, thus

$$T = \frac{\frac{G'}{g}\Theta\ m\sin b - \frac{G'}{g}\Phi^2 m\cos b + \frac{G'}{g}p + \frac{G''}{g}p - N\cos a}{\sin a} = \frac{\frac{G'}{g}\Theta k^2 + \frac{G'}{g}p\ m\sin b + N\,l\sin(a+b)}{l\cos(a+b)}$$

and clearing of fractions, we have

$$\frac{G'}{g}\Theta\ m\ l\sin b\cos(a+b) - \frac{G'}{g}\Phi^2 m\ l\cos b\cos(a+b) + \frac{G'+G''}{g}p\ l\cos(a+b) - N\ l\cos a\cos(a+b) = \frac{G'}{g}\Theta k^2 \sin a + \frac{G'}{g}p\ m\sin a\sin b + N\ l\sin a\sin(a+b)$$

and solving for N, we obtain

$$N\left[l\cos a\cos(a+b) + l\sin a\sin(a+b)\right] = \frac{G'}{g}\Theta\ m\ l\sin b\cos(a+b) - \frac{G'}{g}\Phi^2 m\ l\cos b\cos(a+b) + \frac{G'+G''}{g}p\ l\cos(a+b) - \frac{G'}{g}\Theta k^2\sin a - \frac{G'}{g}p\ m\sin a\sin b,$$

and

$$N = \frac{G'}{g}.\frac{\Theta\ m\ l\sin b\cos(a+b) - \Phi^2 m\ l\cos b\cos(a+b)}{l\cos b} - \frac{\Theta k^2\sin a + p\ m\sin a\sin b}{l\cos b} + \frac{G'+G''}{g}\ \frac{p\ l\cos(a+b)}{l\cos b}$$

as $\cos a\cos(a+b) + \sin a\sin(a+b) = \cos b$.

We found in connection with Fig. 7 that the vertical inertia was greatest for crank angle = 90 degrees, and for this position of crank and rod the trigonometrical functions in above equation become $\sin a = 1$; $\cos a = 0$; $\sin b = \frac{r}{l}$; $\cos b = \frac{\sqrt{l^2 - r^2}}{l}$; $\cos(a + b) = \sin b = \frac{r}{l}$; and substituting these values, we obtain

$$N = \frac{G'}{g}\left(\Theta\, m \frac{r^2}{l\sqrt{l^2 - r^2}} - \Phi^2\, m \frac{r}{l} - \frac{\Theta\, k^2}{\sqrt{l^2 - r^2}} - \frac{p\, m\, r}{l\sqrt{l^2 - r^2}}\right) + \frac{G' + G''}{g} \cdot \frac{p\, r}{\sqrt{l^2 - r^2}}. \quad *$$

Now when $a = 90°$, $\Theta = -\frac{r\,\omega^2}{\sqrt{l^2 - r^2}}$; $\Phi = 0$; and $p = \frac{r^2\,\omega^2}{\sqrt{l^2 - r^2}}$ or $-\frac{r^2\,\omega^2}{\sqrt{l^2 - r^2}}$. *

---

*The angular distance traveled by the crank is a, and the angular velocity $\omega = \frac{d\,a}{d\,t}$, and as ω is constant, $\frac{d\,\omega}{d\,t} = 0$. The angular distance traveled by the rod is b, and the angular velocity $\Phi = \frac{d\,b}{d\,t}$, the angular acceleration being $\Theta = \frac{d\,\Phi}{d\,t} = \frac{d^2 b}{d\,t^2}$. Also $r \sin a = l \sin b$, and $d\,(r \sin a) = r \cos a\, da$ and $d\,(l \sin b) = l \cos b\, db$, hence $r \cos a \frac{d\,a}{d\,t} = l \cos b \frac{d\,b}{d\,t}$, or $r\,\omega \cos a = l\,\Phi \cos b$, and $\Phi = \frac{r \cos a}{l \cos b}\,\omega$.

Now

$$\Theta = \frac{d\,\Phi}{d\,t} = \frac{r\,\omega}{l}\,\frac{-\sin a \frac{d\,a}{d\,t} \cos b + \sin b \frac{d\,b}{d\,t} \cos a}{\cos^2 b}$$

that is + for clock-wise rotation and — as in Fig 14a, and substituting these values, using the + for p,

$$\Theta\, m \frac{r^2}{l\sqrt{l^2 - r^2}} \text{ becomes } - \frac{r\,\omega^2}{\sqrt{l^2 - r^2}} \times \frac{m\, r^2}{l\sqrt{l^2 - r^2}} = - \frac{m\, r^3\, \omega^2}{l\,(l^2 - r^2)},$$

$$-\Phi^2\, m \frac{r}{l} \text{ becomes } o \times - m \frac{r}{l} = o,$$

$$- \frac{\Theta\, k^2}{\sqrt{l^2 - r^2}} \text{ becomes } - \frac{r\,\omega^2}{\sqrt{l^2 - r^2}} \times - \frac{k^2}{\sqrt{l^2 - r^2}} = + \frac{r\,\omega^2\, k^2}{l^2 - r^2},$$

$$- p \frac{m\, r}{l\sqrt{l^2 - r^2}} = \frac{r^2\,\omega^2}{\sqrt{l^2 - r^2}} \times - \frac{m\, r}{l\sqrt{l^2 - r^2}} = - \frac{m\, r^3\,\omega^2}{l\,(l^2 - r^2)},$$

and

$$p \frac{r}{\sqrt{l^2 - r^2}} = \frac{r^2\,\omega^2}{\sqrt{l^2 - r^2}} \times \frac{r}{\sqrt{l^2 - r^2}} = \frac{r^3\,\omega^2}{l^2 - r^2}, \text{ we obtain}$$

$$\Theta = \frac{r\,\omega}{l} \cdot \frac{-\,\omega \sin a \cos b + \Phi \sin b \cos a}{\cos^2 b}$$

$$= \frac{r\,\omega^2}{l} \left\{ \frac{r \sin b \cos^2 a}{l \cos^3 b} - \frac{\sin a}{\cos b} \right\}, \text{ and for } a = 90^\circ$$

$$= \frac{r\,\omega^2}{l} \left\{ - \frac{l}{\sqrt{l^2 - r^2}} \right\} = - \frac{r\,\omega^2}{\sqrt{l^2 - r^2}}.$$

The horizontal distance traveled by rod, cross-head, etc., is $s = l + r - l \cos b - r \cos a$, and its horizontal velocity at the instant is

$$V' = \frac{d\,s}{d\,t} = l \sin b \frac{d\,b}{d\,t} + r \sin a \frac{d\,a}{d\,t}$$

$$= r\,\omega \sin a + r\,\omega \frac{\sin b \cos a}{\cos b} \text{ and}$$

$$p = \frac{d\,V'}{d\,t} = r\,\omega \left\{ \cos a \frac{d\,a}{d\,t} + \frac{\cos^2 b \cos a \frac{d\,b}{d\,t} - \sin a \sin b \cos b \frac{d\,a}{d\,t} + \sin^2 b \cos a \frac{d\,b}{d\,t}}{\cos^2 b} \right\}$$

$$p = r\,\omega \left\{ \omega \cos a + \frac{\omega\, r \cos^2 a}{l \cos b} - \frac{\omega \sin a \sin b}{\cos b} + \frac{\omega\, r \cos^2 a \sin^2 b}{l \cos b} \right\}$$

$$N^{+} = \frac{G'}{g}\left(\frac{r\,\omega^2 k^2}{l^2 - r^2} + \frac{r^3\,\omega^2}{l^2 - r^2} - \frac{2\,m\,r^3\,\omega^2}{l\,(l^2 - r^2)}\right) + \frac{G''}{g} \cdot \frac{r^3\,\omega^2}{l^2 - r^2} = \frac{G'}{g}$$

$$\omega^2\, r \cdot \left[\frac{k^2 + r^2 - 2\,m\,\dfrac{r^2}{l}}{l^2 - r^2}\right] + \frac{G''}{g}\,\omega^2\, r\,\frac{r^2}{l^2 - r^2} = \frac{G'}{g}\,\omega^2\, r$$

$$\frac{k^2 + r^2\left(1 - \dfrac{l}{2\,m}\right)}{l^2 - r^2} + \frac{G''}{g}\,\omega^2\, r\,\frac{r^2}{l^2 - r^2}.$$

Using the negative value for p, we obtain

$$N^{-} = \frac{G'}{g}\left(\frac{r\,\omega^2 k^2}{l^2 - r^2} - \frac{r^3\,\omega^2}{l^2 - r^2}\right) - \frac{G''}{g} \cdot \frac{r^3\,\omega^2}{l^2 - r^2} = \frac{G'}{g}\,\omega^2\, r\,\frac{k^2 - r^2}{l^2 - r^2}$$

$$- \frac{G''}{g}\,\omega^2\, r\,\frac{r^2}{l^2 - r^2}.$$

and the average of the two values of N will be

$$N = \frac{G'}{g}\,\omega^2\, r\,\frac{k^2 - r^2\,\dfrac{m}{l}}{l^2 - r^2} \qquad \text{.......................... (21)}$$

Here it will be noticed that the term containing the reciprocating parts has disappeared entirely. A little thought will make this clear. When the rotation is clockwise, the angularity

---

$= r\,\omega^2 \left\{ \cos a - \dfrac{\sin a \sin b}{\cos b} + \dfrac{r \cos^2 a\,(1 + \sin^2 b)}{l \cos b} \right\}$ and for $a = 90°$

$p = - r\,\omega^2 \dfrac{\dfrac{r}{l}}{\dfrac{\sqrt{l^2 - r^2}}{l}} = - \dfrac{r^2\,\omega^2}{\sqrt{l^2 - r^2}}$, if rotation be as in Fig. 14a.

If opposite, $s = l \cos b - r \cos a + r - l$, and the signs of the development of $l \cos b$ are reversed, finally giving us

$p = \dfrac{r^2\,\omega^2}{\sqrt{l^2 - r^2}}$ for clockwise rotation.

of the rod will cause the reciprocating parts to drag, as they have not reached the center of the stroke (see plate 6) at the 90-degree crank angle, but when the rotation is contra-clockwise, the reciprocating parts have passed the center of the stroke, and tend to accelerate the motion. In actual fact, it is not really the direction of motion, but whether the crank is moving toward or from the crosshead, that should be considered as causing the different values of X, but the mean value given by equation 21 disposes of the direction of motion.

The force X in equation 21 can be balanced by a force produced by a weight solidly attached to the wheel, thereby causing it to rotate with the same angular velocity ω, the distance of such weight from the center of axle being =

$$\frac{G' r}{G'''} \; \frac{k^2 - r^2 \frac{m}{l}}{l^2 - r^2}$$

and opposite to the crank. If we assume this counterweight to be at the same distance from center of axle as the crank-pin, r, we have

$$G''' = G' \frac{k^2 - \frac{m}{l} r^2}{l^2 - r^2} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (22)$$

where G''' is the weight at radius of crank r, necessary to balance the vertical inertia of the connecting rod at quarter stroke.

In equation 22, all the factors except k and m are known. The determination of m can be made, if the rod be completed, as explained by Figs. 11 and 12. Here, however, $m = l - d$, as shown by the figures, or $m = \frac{P'' l}{P' + P''}$, instead of the form given in equation 20.

In order to determine the radius of gyration experimentally, we must first find the radius of oscillation. Suspend the rod from the crosshead pin in such a manner that it may swing freely, and swing it about this center a, Fig. 15, noting the exact time of the oscillations.

Remove the rod from the pin and reverse it, but suspend it by means of a yoke clamped on the rod, with two journals, as shown at b, and adjust the clamp longitudinally on the rod

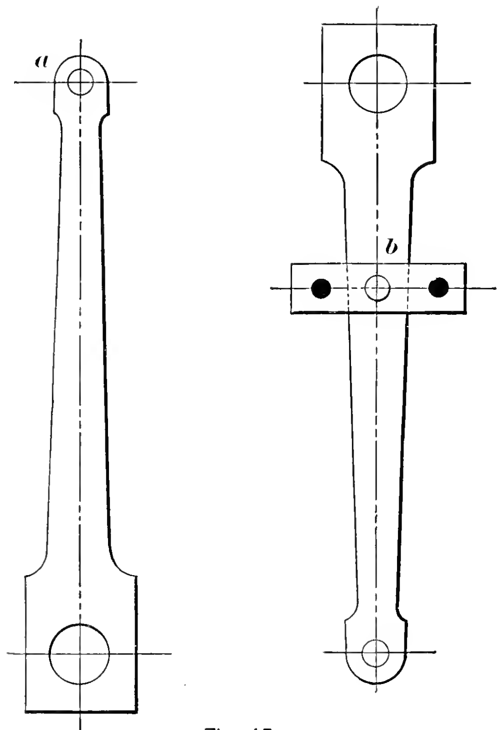


Fig. 15.

until the point is found where the time of oscillation is identical with the first test. The distance of this center so found from the crosshead pin center is the radius of oscillation, o. From this we can determine the radius of gyration, which is a mean proportional between the radius of oscillation and the distance m of center of gravity of rod from crosshead pin, by making

$$k = \sqrt{m\,o} \text{ or } k^2 = m\,o.$$

The value of "o" can be determined by noting the accurate time of oscillation about the crosshead pin, without using the reversed bearing, b. The pin "a" should have knife-edge bearings at the center, and the time taken (with a stop watch) of 100 single oscillations (or 50 complete vibrations), and $\frac{1}{100}$ of this time taken to represent one swing. The amount of oscillation should not be over 3 or 4 inches at the lower end.

If we let t = the time in seconds of a single oscillation, we have, from the law of pendulum motion,

$$t = \pi \sqrt{\frac{o}{g}}$$

and transposing,

$$\frac{t}{\pi} = \sqrt{\frac{o}{g}}, \frac{t^2}{\pi^2} = \frac{o}{g} \text{ and } o = \frac{t^2 g}{\pi^2}$$

and substituting the values of g and π, we have $o = 3.26\, t^2$, o being in feet, same as k and m. Then, as $k^2 = m\, o$, we can also write

$$k^2 = 3.26\, t^2\, m$$

and this value may be used in equation 22. A connecting rod 10 feet long, with the center of gravity 6 feet from crosshead pin, oscillated in 1.624 seconds, so that

$$k^2 = 3.26 \times 1.624^2 \times 6 = 51.6$$

and, as $l = 10$,

$$\frac{k^2}{l^2} = \frac{51.6}{100} = .516$$

If it be not convenient to obtain the values of m and k experimentally, as just described, we can calculate them closely enough for general purposes. Let us suppose a rod as in Fig. 16, having the dimensions given.

For convenience, divide it into three general portions, as shown by the cross-hatching at different angles, which we may designate as crosshead end, crank end and shank. By calculating the cubical contents of each part and multiplying the cubic inches by .28, we obtain the weight in pounds as marked, viz., 70, 146 and 150 pounds, respectively, or 366 pounds total.

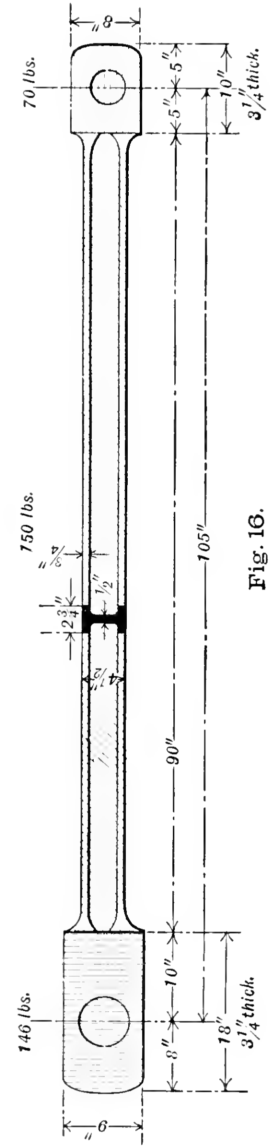

8"
70 lbs.
5"
5"
10"
3¼" thick.
150 lbs.
¾"
½"
2¾"
4½"
90"
105"
10"
146 lbs.
8"
18"
3¼" thick.
6"

Fig. 16.

Multiplying each section by the distance of its center from the crosshead pin, we have

$$\begin{aligned} 70 \times 0 &= \ldots\ldots \\ 150 \times 50 &= 7{,}500 \\ 146 \times 104 &= 15{,}184 \\ \hline & \quad 22{,}684 \end{aligned}$$

and dividing the total moments by the total weight ($22{,}684 \div 366 = 62$), we have the center of gravity 62 inches from the crosshead pin, or .60 of the length of rod between centers. This point is the center of the static moment, and will correspond with the distance m in the experimental method; in feet, m = 5 1-6, to be used in formula.

For the radius of gyration, we must remember that the moment of inertia of a bar of uniform cross-section, about an axis perpendicular to its length, but passing through its center of gravity $=$ weight $\dfrac{\text{length}^2}{12}$; also, that the moment of inertia about an axis not passing through the center of gravity, but parallel to such an axis, is equal to the moment of inertia about the axis through the center of gravity plus the product of the weight of the body multiplied by the square of the distance between the two parallel axes. These operations are represented below:

$$\begin{aligned} 70 \times \frac{10^2}{12} &= 560 \\ 150 \times \frac{90^2}{12} + 150 \times 50^2 &= 476{,}250 \\ 146 \times \frac{18^2}{12} + 146 \times 104^2 &= 1{,}583{,}078 \\ \hline & \quad 2{,}059{,}888 = \text{Mom. of inertia} \end{aligned}$$

and dividing by the weight, $2{,}059{,}888 \div 366 = 5{,}628$, which equals the square of radius of gyration, and the square root of this

$$\sqrt{5{,}628} = 75 \text{ inches}$$

or $6\frac{1}{4}$ feet, the radius of gyration, or the square in feet $= 39$, which would be used in the formula. We also see that

$$\frac{k^2}{l^2} = \frac{(6\frac{1}{4})^2}{(8\frac{3}{4})^2} = \frac{39}{76} = .51$$

or the square of the radius of gyration is about one-half the square of the length of rod. We found above, that in this case the center of gravity was .6 the length of the rod from the crosshead pin. Other rods examined give nearly the same values for $\frac{m}{l}$ and $\frac{k^2}{l^2}$, and Mr. Parke declares that if we put

$\frac{k^2}{l^2} = \frac{1}{2}$ and $\frac{m}{l} = \frac{5}{8}$, we will cause no material error.

If we place these values, $k^2 = \frac{1}{2} l^2$ and $m = \frac{5}{8} l$ in equation 22, we obtain.

$$G''' = G' \frac{\frac{1}{2} l^2 - \frac{5}{8} r^2}{l^2 - r^2} \quad \text{.................................} (23)$$

which must be taken as an approximate formula, as m and k have been given assumed values. If wanted accurately, equation 22 should be used.

A short analysis of equation 22 will give a better idea of its significance, and for this purpose we can divide it into two portions, of which the first (a) represents the effect of oscillation of the connecting rod, and the second (b) the effect of horizontal inertia, caused by and transferred longitudinally through the connecting rod and producing a load on the crankpin having a vertical component, which component it (b) represents. The annexed table gives the numerical values of these two portions or cases for $l = 5$ r, 8 r, 10 r and infinity, using the values $m = \frac{5}{8} l$ and $k^2 = \frac{1}{2} l^2$, as in equation 23.*

*In November, 1903, before the New England R. R. Club, Mr. Parke gave results of his tests upon three solid and three fluted main rods, from which he found the values of $\frac{m}{l}$ and $\frac{k^2}{l^2}$ to be respectively as follows:

	Solid Rods.			Fluted Rods.		
$\frac{m}{l} =$	.68	.61	.64	.60	.62	.65
$\frac{k^2}{l^2} =$	.577	.490	.547	.518	.525	.562

Case	Formula	Equivalent	Coefficients of G′ for $l = 5r$	$l = 8r$	$l = 10r$	$l = \infty$
a	$G' \frac{k^2}{l^2 - r^2}$	$G' \frac{1}{2} \frac{l^2}{l^2 - r^2}$	.520	.508	.505	.500
b	$G' - \frac{\frac{m}{l} r^2}{l^2 - r^2}$	$G' \frac{5}{8} \frac{-r^2}{l^2 - r^2}$	— .026	— .010	— .006	— 000
Total			.494	.498	.499	.500

From this we see that the oscillations of the connecting rod cause the principal disturbances, as might be expected, and that, for general purposes, one-half of the weight of the main rod is not far wrong for an approximation as to the balance needed to be added to the main wheel, considered as acting at the crank radius.

Having studied the balancing of revolving weights and also the vertical effect of the connecting rod, it remains to examine the proper methods of caring for the reciprocating parts. From formulæ 16 and 17 we have seen that these parts create their greatest effects at the ends of the stroke, the values at those points being =

$$\frac{G v^2}{g r}\left(1 + \frac{d r}{l^2}\right)$$

for the front end and =

$$\frac{G v^2}{g r}\left(1 - \frac{d r}{l^2}\right)$$

at the back end. It will evidently be impossible to balance both forces, so the mean will have to be taken, which $= \frac{G v^2}{g r}$, or considering the counterbalance to have the same radius as the crank, its weight should be equal to the parts which it is to balance. As the revolving parts have already been considered as fully balanced, this, for a total balance, should equal the combined weight of the connecting rod, crosshead, piston and piston rod. According to equations 22 and 23, we have already arranged to balance a portion of this weight for vertical disturbance, and, of course, this balance will be effective also in a horizontal direction. We have also decided that $\frac{1}{400}$ part of the weight of the engine should be left unbalanced. If we use G^{i}, G^{ii} and G^{iii} to represent the weights of the connecting rod, reciprocating parts, and the vertical balance for rod, as in equations 22 and 23, also the total weight of the engine by W, we have for the weights still to balance =

$$G^{iv} = G^{i} + G^{ii} - G^{iii} - \frac{W}{400} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (24)$$

That is to say, the weight of main rod, crosshead, piston and piston rod, less the amount placed in the main wheel to take care of the vertical forces of the main rod and less $\frac{1}{400}$ of the weight of the engine itself. The amount to be placed in each wheel will properly be equal to G^{iv} divided by the number of driving wheels on each side, but if, as explained before, the main wheel, or any wheel, should thereby be overloaded as to the resistance of the track, etc., a portion or all of this part of G^{iv} may be transferred to other drivers.

In order to exemplify these rules practically, we will consider the balancing of a fast passenger locomotive of the 4—4—2 type. The general features of this engine are as follows:

Weight of engine	158,000 lbs.
Weight on front drivers	43,000 lbs.
Weight on main drivers	48,000 lbs.
Cylinders	20 by 26 ins.
Diameter driving wheels	80 ins.
Connecting rod, length	130 ins. = 10.8 ft.
Connecting rod, front end weight	218 lbs.
Connecting rod, back end weight	332 lbs.
Parallel rod, front end weight	120 lbs.
Parallel rod, back end weight	160 lbs.
Piston and rod weight	410 lbs.
Crosshead weight	190 lbs.
Steam pressure	200 lbs.

As the stroke is 26 inches, the crank radius r = 1.08 feet, and, as above, l = 10.8 feet, a ratio of $\frac{l}{r} = 10$. We find the value of m to be

$$\frac{332 \times 10.8}{550} = 6.5 \text{ feet, and } \frac{6.5}{10.8} = .6 \text{ for } \frac{m}{l}$$

We can therefore use one-half of the weight of the main rod (= 218 + 332 = 550 pounds), or 275 pounds for the balance in main wheel (at crank radius) to take care of the vertical influence of the main rod.

The reciprocating parts weigh 410 + 190 = 600 pounds, and as $\frac{W}{400} = \frac{158{,}000}{400} = 395$ pounds, we can write, from formula 24,

$$G^{iv} = 550 + 600 - 275 - 395 = 480 \text{ pounds}$$

total balance on one side for reciprocating parts, and as there are two drivers on a side, one-half of this, or 240 pounds, should go to each wheel. What we have done by these calculations has been to subtract $\frac{1}{400}$ of the weight of the engine (= 395 pounds) and the balance necessary to take care of the vertical motion of the main rod (= 275 pounds) from the sum of the total weight of main rod (= 550 pounds) and the reciprocating parts (= 600 pounds), and one-half of this difference is the amount of balance (= 240 pounds) to place in each wheel, at the radius of the crank, to partly neutralize the effect of inertia of the reciprocating parts.

We can now state the balance required in each wheel thus (all at crank radius):

Part to balance—	Front wheel.	Main wheel.
Reciprocating parts	240 lbs.	240 lbs.
Connecting rod		275 lbs.
Parallel rod	120 lbs.	160 lbs.
Total loose from wheel	360 lbs.	675 lbs.
Crankpin	70 lbs	170 lbs.
Crank hub	100 lbs.	150 lbs.
Total	530 lbs.	995 lbs.

It was found that the counterbalances could be so arranged that their centers of gravity would be 30 inches from the center of axle, therefore the amount needed in each wheel was

$$\text{Front wheel, } \frac{530 \times 13}{30} = 230 \text{ lbs.}$$

$$\text{Main wheel, } \frac{995 \times 13}{30} = 431 \text{ lbs.}$$

That is, the weight to be balanced at crank radius, multiplied by that radius and divided by the distance of the center of gravity of the counterbalance from the center of axle. It is convenient to make both of these balances of the same shape and vary the thickness to suit the weight. We can construct a segment of 206 square inches area, with a center of gravity located as described, and it will need to have thickness as follows:

Front wheel, $230 \div 206 \times .28 = 4$ inches thick
Main wheel, $431 \div 206 \times .28 = 7\frac{1}{2}$ inches thick

That is, the desired weight divided by the area of counterbalance multiplied by the weight of a cubic inch of cast steel (= .28 pound), as these wheel centers were made of cast steel and had solid balances. (If hollow balances filled with lead had been used, the weight of lead would have been substituted, viz., .41 pound per cubic inch.)

After the wheels were mounted on the axle and the pins inserted, they were tested as shown in Fig. 17.

The wheels were set upon trestles provided with perfectly level straight edges, the journals of the axle resting on the straight edges. A pan suspended from the crankpin by wires or cords was filled with weights until the wheels balanced, with the side shown on a horizontal line through axle and pin, and the other side with a vertical line through axle and pin, the pin, of course, being up. The amount of weight applied to balance, including the weight of the pan and its hangings, gave the equivalent counterbalance at crank radius available for balancing the parts designated as loose from the wheel.

If the counterbalances are left with extra thickness, the metal in excess can be turned off after the just described test is made, and a very accurate adjustment effected.

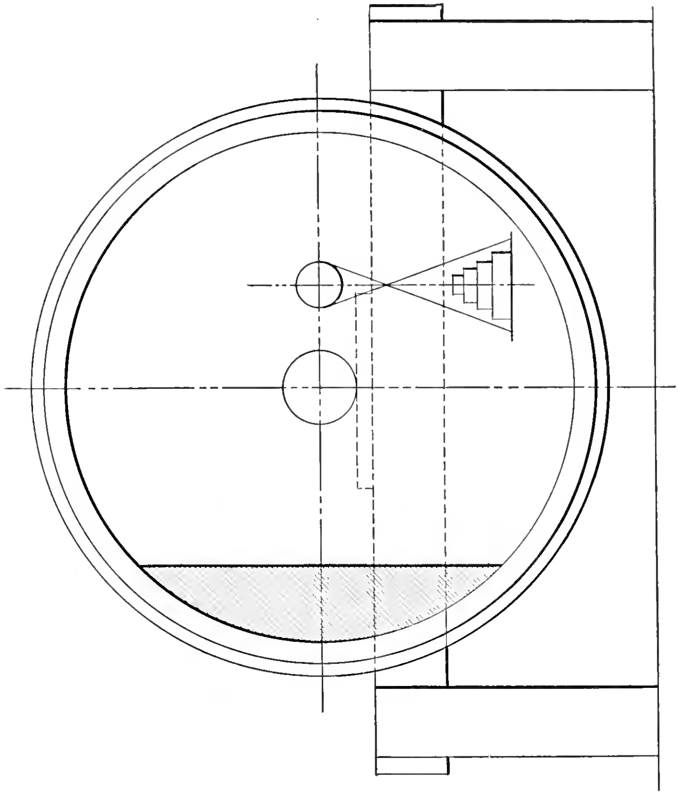


Fig. 17.

In the engine under consideration, the test showed as follows:

	Front wheel.	Main wheel.
Balance at pin	345 lbs.	670 lbs.
Balance desired for "loose parts"	360 lbs.	675 lbs.
Wheels short	15 lbs.	5 lbs.
Loose revolving parts	120 lbs.	435 lbs.
Excess balance	225 lbs.	235 lbs.

Thus it will be seen that the front and main wheels were short 15 and 5 pounds, respectively, at the crank radius. The "excess balance," or part for which there is no corresponding vertical force, is the difference between the balance at the pin and the "loose revolving" parts, and amounts to 225 and 235 pounds, respectively. It is now necessary to determine the effect of this excess balance upon the track, which we recognize as the "hammer blow" of certain authorities (?).

From equation 9, we can put for

Front wheel, $1.6 \times 225 \times 26 = 9,360$ lbs. centrifugal force

Main wheel, $1.6 \times 235 \times 26 = 9,776$ lbs. centrifugal force

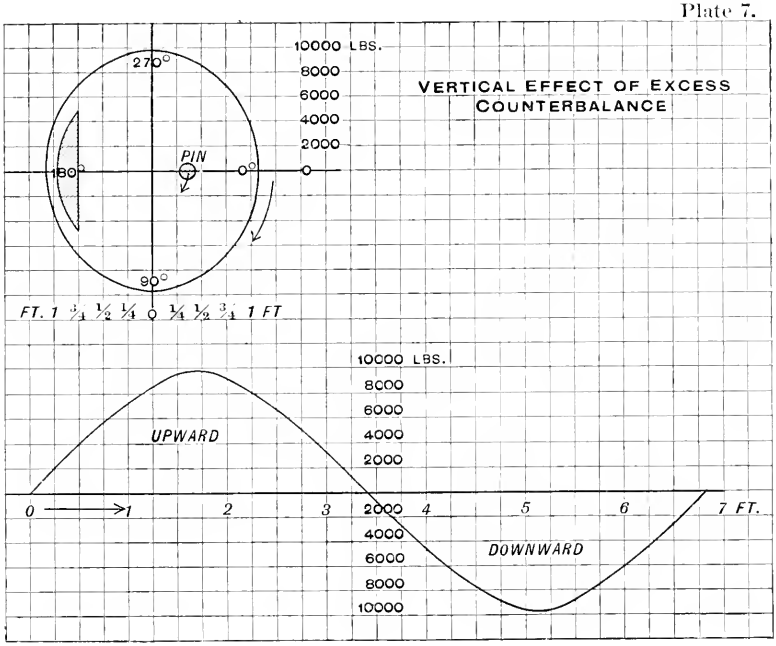
and the effect upon the track

	Front wheel.	Main wheel.
Static load	21,500 lbs.	24,000 lbs.
Centrifugal force of excess	9,360 lbs.	9,776 lbs.
Maximum rail pressure	30,860 lbs.	33,776 lbs.
Minimum rail pressure	12,140 lbs.	14,224 lbs.
75 per cent of static load	16,125 lbs.	18,000 lbs.

From this it appears that at 80 miles per hour (speed = driver diameter) the rail load is only about 34,000 pounds, at its maximum, and when the counterbalance is up the lifting tendency is less than 10,000 pounds—very considerably under the 75 per cent limit, so that there is no danger of the wheels leaving the rail—in fact, there is at least 12,000 to 14,000 pounds rail pressure. Plate 7 gives a graphical representation of the vertical effect of the excess balance on the main wheel, the upper diagram showing ordinates corresponding to this force laid out on the projection of the crank circle, and the lower one, on the development of the crank circle, approximately correct, the forces being $= \sin \Theta\ 9,776$, see demonstration of Fig. 7.

It is well known that track has often been damaged by hauling at high speeds engines without the rods upon the pins, and many roads refuse to accept for shipment engines without the side rods applied. In the engine which we have just discussed, we found that the main wheel balanced with 670 pounds on the pin, and if the rods were not applied, this would act as the excess balance, and at 80 miles an hour the centrifugal force would be $= 1.6 \times 670 \times 26 = 28,080$ pounds, and the maximum and minimum rail pressures $24,000 \pm 28,080$, or $+ 52,080$ pounds, and $- 4,080$ pounds; that is, the wheel would leap vertically from the rail each time the counterbalance was uppermost, and press downward with 52,000 pounds when down. Generally, when engines are hauled without the rods, orders are issued to the trainmen not to exceed 15 miles per

Plate 7.



hour. In such a case, with the engine which we have just studied, the effect would be as the squares of the speeds, or

$$\left(\frac{15}{80}\right)^2 = \frac{225}{6400} = .035 \text{ and } .035 \times 28{,}080 = 983 \text{ pounds.}$$

an amount which is only about one-tenth of that when the engine is in operation at high speed.

In testing the counterbalances of existing engines, as explained by Fig. 17, it is often found that they are excessive, and it is sometimes desirable to quickly reduce the balance of a number of engines which may be in everyday service. Such a case recently occurred on a western road, where, due to a number of derailments, it was thought advisable to reduce the amount of balance on the front drivers of 40 locomotives. As we have found that the centrifugal force is proportional to the difference in the weights, we can easily and quickly reduce the counterbalance by applying a counter-counterbalance, that is, blocks of metal may be cast to fill in the spaces between spokes surrounding the pin, the blocks being parted in the plane of the wheel, and being riveted together, clamping the spokes, and

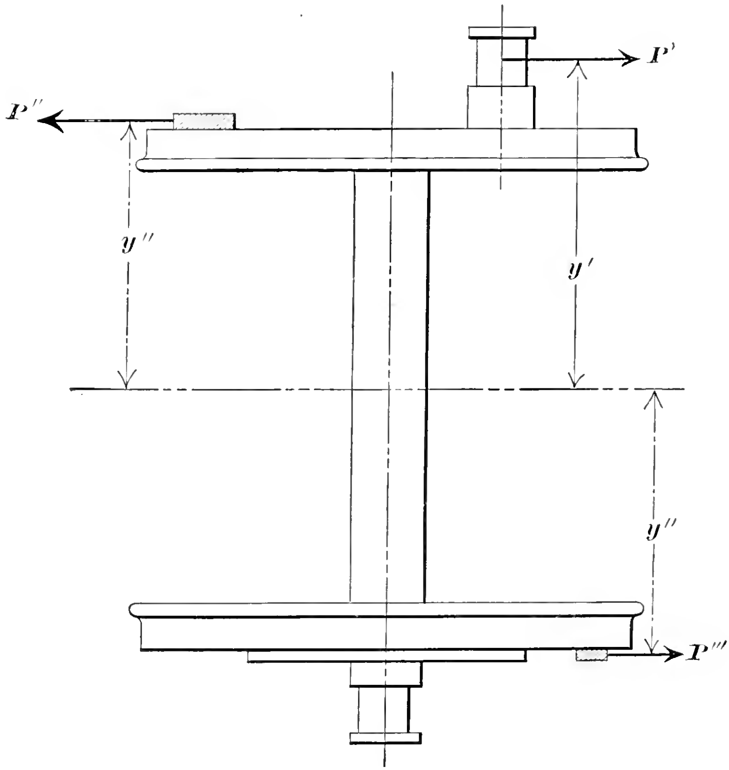


Fig. 18.

holding the blocks securely in place. This can be done in the roundhouse between trips, and in the case just quoted gave a quick and efficient remedy.

We have so far studied the action of the counterbalance in a vertical plane parallel with the longitudinal center line of the engine only. It will, however, be clear upon consideration that the balance will not be perfect in a horizontal plane, even if it could be made so in a vertical plane. The inertia of the rods, etc., acts at a certain distance from the center of the engine, and the centrifugal force of the counterbalance at another distance, and in outside cylinder engines, a shorter one. This causes a moment in a horizontal plane, which can be overcome if considered necessary, but which refinement is generally omitted, especially in engines with outside cylinders.

In Fig. 18 let us consider the balanced reciprocating parts

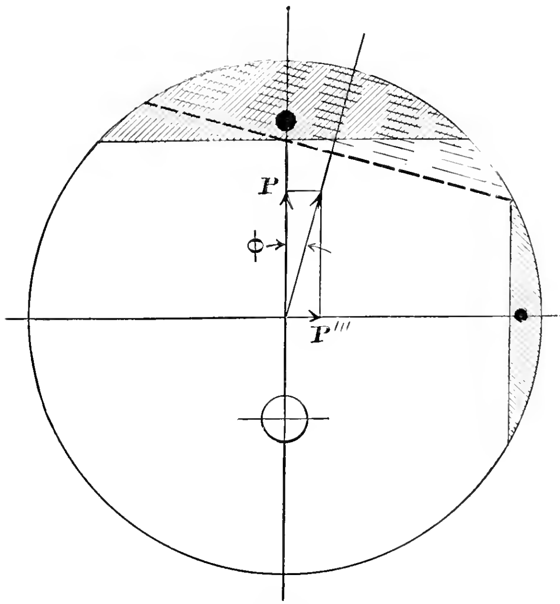


Fig. 19.

acting at P', and at a distance y' from the center line of the engine. The counterbalance acts at P'', a distance y'' from the center of the engine. There will then be a horizontal moment, which, if P'' = P', as is customary, will be equal to P' (y' — y''), on account of the opposing forces acting in different vertical planes. This can be corrected by placing on the opposite wheel, and 180 degrees from the balance of the first wheel, an amount (at crank radius) $= P' \frac{y' - y''}{2 y''}$, for, in order to produce equilibrium, the opposite moments must be equal, or $P' (y' - y'') = P''' 2 y''$, and this transposed gives us $P''' = P' \frac{y' - y''}{2 y''}$, and also $P'' = P' + P'''$.

This would produce a wheel with two counterbalances, 90 degrees apart, as shown in Fig. 19. The main balance, P, will, however, be larger than P'', as the revolving parts are also balanced by the main counterweight, and P'' was considered to apply only to the balance of the reciprocating parts and main rod. It is evident that by moving the main balance through the angle Φ toward the small balance, and increasing it the proper amount, one balance may be used in place of two. As the forces are proportional to the weights, the latter will be represented by the resultant of P and P''', laid off as shown in Fig. 19, and will equal

$$\sqrt{P^2 + P'''^2}$$

and the angle Φ will be that whose tangent is $\frac{P'''}{P}$.

Now let us examine the necessity for this refinement in the engine just quoted. In this case P' = the balanced reciprocating parts and revolving part of main rod $= 480 + 275 = 755$ pounds, $y' = 43''$ and $y'' = 33''$, therefore $P''' = 755 \times \frac{10}{66} = 115$. $P'' = 755 + 115 = 870$ pounds. (This considers all the balancing done by one wheel, as it simplifies the discussion, and as the latter deals with horizontal forces only, no error will affect the result.) The total balance, including that for revolving parts $= 995 + 530 = 1{,}525$, and to this should be added 115, or 1,640 pounds total balance in the two wheels on one side. Now $\tan \Phi = \frac{115}{1{,}640} = .07$, or $\Phi = 4$ degrees. As the wheel contained 18 spokes, the angle between contiguous spokes was 20 degrees, and it will be seen that 4 degrees would be an extremely small amount, and, generally, the practice in this country is to ignore it. In the case of inside cylinders, however, where $y' - y''$ is large, it is worth consideration.

In outside cylinder engines the moment should be made as small as possible by keeping the counterbalance as far out on the face of the wheel as the rods and guides will permit, thus reducing the value of $y' - y''$ to a minimum.

We can now lay down the following points to be observed in counterbalancing locomotives:

A. Each wheel should be fully balanced for all revolving weights attached to it, and an additional amount for reciprocating parts.

B. About one-half the weight of connecting rod is to be balanced in the main wheel as revolving weight, the exact amount to be obtained by the formula (22).

C. One four-hundredth of the weight of the engine can remain unbalanced from the reciprocating parts of one side, these parts to include the piston and rod, crosshead, and such part of the connecting rod as has not been balanced under section B.

D. The remainder from section C should be counterbalanced by dividing this amount equally between the driving wheels on one side, provided that the sum of the static load on any one wheel, plus the centrifugal force of the excess balance, does not exceed the maximum rail pressure allowed at the highest speed at which it will run. If some of the static wheel loads are too great when the proportion of centrifugal force is added, the lighter wheels may take more and the heavier wheels less than an equal division of the reciprocating balance, but rotating balances should not be transferred to another wheel.

E. The centrifugal force of the excess balance should not exceed 75 per cent of the static load on any wheel.

F. The center of gravity of the counterbalance must be opposite the crank, and the weight of the balance proportional to the parts to be balanced at the crankpin as the crank radius is to the distance of the center of gravity of the balance from the center of the axle.

G. The center of gravity of the counterbalance should be placed as near the rim as possible, and the weight of the balance correspondingly reduced. A segmental shape fills this requirement, whereas the old sector form did not.

H. The counterbalance should be brought out from the face of the wheel as far as clearance and good design will permit.

I. The spread of the cylinders should be kept as small as possible.

J. The reciprocating parts should be as light as possible. The pistons and crossheads may be of steel, of light ribbed construction, the piston rods may be tubes, and the main rod should be of I-section, all made of the strongest and lightest available material.

The point is often made, that as the excess counterbalance travels in a trochoidal path and not in a circular one, the centrifugal forces do not properly apply. This fallacy may easily be disproved by again referring to Fig. 7, and the explanation there given. It will be found that the vertical force depends entirely upon the vertical retardation and acceleration of the revolving body. While in the driving wheel of a locomotive, all parts have a movement of horizontal translation as well as rotation, it is evident that this movement of translation has no vertical component, and that the vertical acceleration or retardation of any point on the wheel cannot thereby be affected. This leaves the force resulting from such acceleration or retardation in a vertical direction also unaffected in quantity, although it will be applied at different points on the rail at each subsequent revolution, these points being more widely separated as the diameter of the wheel is increased.

CHAPTER II.

STEAM ACTION.

The action of the steam in operating the locomotive is, of course, the most important of the several functions to be considered in this treatise. The forces of inertia are properly secondary forces, as they are the result of motion caused by the steam action. The latter, is, however, a primary force. In fact, it creates all the other forces and functions which unite to complete the operation of the locomotive. It is therefore important that we should have a clear idea in the start of the various phases and changes which occur while the steam is passing from the boiler to the atmosphere, and performing its different functions. In this chapter we will discuss only the action of the steam, and will reserve for a later one the discussion of its formation, which will naturally include the question of capacity. A limit to the capacity will have to be assumed, in this chapter, for the derivation of certain facts, but this will be fully examined under the caption of "Steam Capacity."

Let us follow the steam briefly in its journey and see to what changes it is subject, and what results directly from its action. On opening the throttle valve the steam issues from the boiler where it is generated and maintained under pressure, and passes through the steam pipes into the valve chamber or steam chest. On this part of its journey it undergoes its first change—a loss in pressure, for we find that upon arrival at the steam chest there has been a drop in number of pounds per square inch, as indicated by gauges upon the boiler and steam chest, or by the steam chest indicator diagram, when taken in connection with the regular cylinder diagrams. We do not say that there would be this loss if the engine were not in motion, but as the locomotive naturally moves upon the opening of the throttle, there results motion to the steam, and as the friction through the various passages retards its flow, it is maintained (during the opening of the throttle) at a pressure less than that in the boiler. How much less, depends upon the throttle opening and the rate at which it is drawn off through the valve, and this again depends upon the speed of the engine, which is, in a measure, dependent upon the opening of the throttle. Under all circumstances, it will be less than the pressure in the boiler.

After reaching the steam chest it is admitted alternately to opposite ends of the cylinder, through the medium of the valve. This opening and closing by the valve is a process continually kept up, and as the amount of opening varies from zero to its maximum, there must be two periods of wiredrawing during each admission, even if the valve at its maximum opening does not really wiredraw the steam passing through an opening often much smaller than it should be. This causes another drop of pressure by the time that steam has found its way to the cylinder, and the greater the speed of the engine and the consequent flow of steam, the greater will be this loss.

After admission to the cylinder another loss confronts us, that of condensation due to the cooler cylinder walls and heads. The metal of which the cylinder is made freely conducts away the heat of the steam, and even non-conducting coverings cannot prevent this entirely. Even if there were no heat conducted to the outer atmosphere and other parts of the engine by the contact of the hot cylinder casting, the fact that the exhaust occurs at a lower pressure and temperature would be sufficient to cool the cylinder walls, at least to the average temperature of the steam during the stroke—a temperature considerably below that of the newly admitted live steam.

Expansion during the performance of its work constitutes another drop in pressure, and this depends upon the point of cutoff in operation at the time. As this can be varied at the will of the engineer within very wide limits, it may be that a very small or very great change in pressure will be occasioned thereby. During this portion of its travel the steam is doing useful work—the first that it has performed since it was generated in the boiler. When in the steam chest its pressure upon the valve caused friction, which destroyed a portion of the useful work generated by the piston, and which friction, by action through the link motion and eccentrics, formed a resistance to the rotation of the axle.

The steam, acting upon the piston, turns the crank (or main driver) through the medium of the connecting rod, with a pressure which varies throughout the stroke, due to the expansion of the steam, the exhaust and compression upon the opposite side of the piston, and the angularity of the connecting rod and the crank itself. The angularity of the rod also causes a pressure of the crosshead against the guides, the resultant friction reducing the piston pressure. As the angle between the connecting rod and the crank is continually changing, the tangential pressure upon the crank, which is really its cause of rotation, varies constantly, so that in order to find the turning moment or effect at any instant, all these contributing conditions must be considered. At high speeds, as we saw in the last chapter, the inertia of the reciprocating parts also affects the result by decreasing or increasing the thrust of the connecting rod. These are all vital points, and of the greatest importance, as upon them depends the power of the machine as a whole, and its capability of doing useful work of transportation.

After the steam has moved the piston to the end of the stroke it is permitted to leave, but always at more or less pressure—this constitutes a resistance to the next stroke of the piston in the opposite direction and produces back pressure. Moreover, the closing of the valve before the piston has fully accomplished its back stroke causes an additional resistance in the shape of compression, which, however, is not without advantage, as will be seen. From the exhaust cavity in the cylinder it escapes through the exhaust nozzle in the smoke box, having traversed the more or less tortuous passages in the casting. Here it is very much reduced in pressure, but still able to do work by entraining the hot gases and producing a small vacuum in the smoke box, which is due to the velocity of the exhausted steam and its ejector-like action. It finally escapes through the stack at about atmospheric pressure, accompanied by the products of combustion. Here, again, for the second time, it performs useful work—not in the way of moving the machine, but indirectly by exciting and urging the fire in the fire box to an exceedingly rapid rate of combustion—a rate which is unsurpassed in any other type of construction, unless it be the steam fire engine, which operates under similar conditions.

This sketch shows briefly what the steam does, how it does it, and how it changes its conditions while doing its work, and as all these operations are important they will be considered in detail.

STEAM CHEST PRESSURE.

We have stated in the preamble above that the steam chest pressure depends upon the throttle opening and the speed of the engine. By means of the throttle valve it can evidently be reduced from its maximum to zero, which is the case if the throttle be completely closed, and this can be done when running at any speed; for instance, when drifting down grade, the throttle may be entirely closed. This means of variation of the steam chest pressure is in the hands of the engineer exclusively, and therefore no rule can be laid down for its control, as it may have any possible opening from zero to its maximum at any rate of speed. Moreover, the proportion of opening is really a matter of small consequence, and as no means are given to determine the amount under ordinary working conditions, there can be no rules laid down for its manipulation, except the general one that under ordinary conditions it is best to run with a full throttle opening, and regulate the speed and other conditions by means of the reverse lever. (In compound engines it is generally not desirable to cut off closer than .4 of the stroke; in simple engines not less than .15 or .2 of the stroke, on account of excessive cylinder condensation, and in such cases where less power is required the proper proceeding is to effect the reduction by partly closing the throttle.) Ordinarily the engineer has little opportunity to learn what percentages of cut-off correspond to the various notches in the quadrant, and it is not desirable to stamp the quadrant itself with the cut-off, as future valve settings will disarrange the figures. The author has adopted the plan of numbering the notches from the front end, the forward corner being No. 1, and posting in the cab of the locomotive a card giving the cut-off for various notches of the quadrant. This card can be easily changed with subsequent adjustments of the link motion, and the quadrant can be stamped with large figures before being case hardened, which figures can remain permanently as located. While it is true that the engineer can at will reduce the steam chest pressure below the maximum any desired amount, yet he is powerless to increase the maximum pressure, which will be determined by the resistance in the steam pipes and the speed of the engine. It is therefore important to know what relation that maximum steam chest pressure bears to the boiler pressure, with a full throttle opening. This cannot well be determined by calculation, but is best found by examination of typical indicator diagrams. From a careful investigation in the manner just indicated, it is believed that the following table will fairly represent existing locomotive conditions:

Relation of steam chest pressure to boiler pressure, with full throttle opening:

Revolutions per minute.	Starting	50	100	150	200	250	300	350
Steam chest pressure	.99	.97	.95	.94	.93	.92	.91	.90
Per cent loss	1	3	5	6	7	8	9	10

Boiler pressure considered = 1.00.—all in gauge pressure.

These values represent the average steam chest pressure, for this pressure does not remain constant. Fig. 20 is a typical steam chest indicator diagram. The cut-off is at about half stroke and it will be seen that while the valve is open to the cylinder, the steam chest pressure is reduced by the draft of steam to the cylinder. As the valve is about to close the pressure at once rises, as it is supplied by the steam pipe without being drawn off to the cylinder, but as soon as the port is open at the end of the stroke it again falls, the drop increasing as the valve opening and speed of piston increase. With a large steam chest the drop would not be as great as in a small one, but nearly all chests are small relatively to the cylinder. This indicates the value of steam pipes of ample proportions, in order to reduce the friction, and also the importance of large steam chests. In most cases the valve is so big that it nearly fills even a large chest, leaving little room for steam. The steam pipes, being in continual communication with the chest, give it

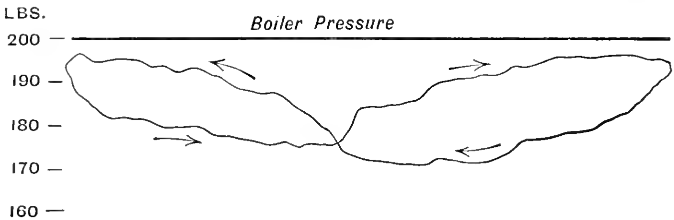


Fig. 20.

an increased volume, but often the passages leading to the steam chests are so tortuous and narrow that the supply does not come as freely as it should. In many cases the steam passages lie next to the outside wall of the cylinder casting, inducing condensation even before the steam has entered the valve chamber. This is the case very generally with piston valves having outside admission—a point in favor of the inside admission valves.

Another loss of pressure which sometimes occurs in the chest is that due to leaky valve stem packing. I admit that under these circumstances the packing is in a faulty condition, yet it is a fact that on a cold morning a large percentage of locomotives will show this leak about the valve stem gland. Locomotive practice consists largely in meeting conditions as they exist, not as they should exist, and if we can dispense with a packing we are reducing the chances of a leak. Where the valve stem is prolonged through the front head or end of chest an additional packing exists, not only requiring maintenance, but sometimes causing a blow. Often there is no real need of such an extension, and it should be dispensed with. The fact that the valve stem travels the most of the time in a short path causes the stem to wear hollow or thin at the center of the length of contact with the packing, and when it moves in full stroke, as in starting, a blow is very apt to occur, unless the packing be maintained in the best condition. An advantage enjoyed by the inside admission piston valve is that only exhaust pressure will come against the valve stem packing, and it has been found that it can be kept tight with the old hemp filling and plain gland. These points are well worth considering from a practical standpoint.

VALVE MOTION.

As the steam valve controls the admission and discharge of steam to and from the cylinder, its motion is of the greatest importance, and cannot be studied too carefully. There are ordinarily but two types of valve motion applied to locomotives—the Stephenson link motion, which is almost universally used in this country, and the Walschaert radial valve gear. The former consists of a pair of eccentrics located with the proper angular advance for forward and backward movement, respectively, the forward ends of the eccentric rods being connected by a link, the shifting of which changes the direction of motion of the engine, and, incidentally, varies the travel, lead and cut-off of the valve. The latter consists of a single eccentric with a rocking link, the shifting of the block in this link reversing the motion of the engine and changing the point of cut-off. As but one eccentric (on a side) is used, it is of necessity set at 90 degrees from the crank, consequently it can impart no lead to the valve, being without angular advance. An independent connection to the crosshead furnishes the requisite lead, and as the crosshead always has the same stroke, the amount of lead is constant for all points of cut-off.

The investigation of the Stephenson link motion mathematically is rather lengthy, but on account of the importance of the subject it will be given in detail. We are indebted to Professors Zeuner and Peabody for this analysis. Fig. 21 illustrates the ordinary Stephenson link motion with "open rods," that is, when the angular advance of both eccentrics throws them both ahead of a vertical line passing through the center of the axle, the eccentric rods will not cross each other, but the rod attached to the top of the link will be found on the uppermost eccentric, and the one secured to the bottom of the link will be connected with the lower eccentric, as they stand at that instant. This has nothing to do witn the position of the crank, as it may be either on the forward or back center, depending upon whether the link motion includes a rocker arm

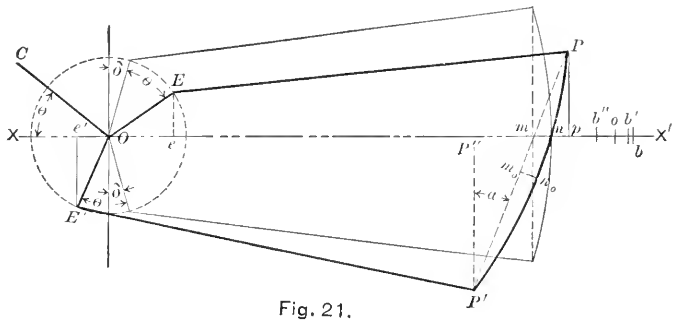


Fig. 21.

or whether it is direct. The definition given above makes no reference to the crank, however, and will enable one to determine at once whether the rods are "open" or "crossed." As the latter are hardly ever used on locomotives, only the "open rod" arrangement, as shown in Fig. 21, will be here considered.

In this diagram the thin lines give the relative positions when the crank is at the back dead point and the link central, the motion being a "direct" gear. The analysis is, however, identical, if a rocker be used, and no error will appear on that account. The angular advance of the eccentrics is denoted by δ, and is considered the same for both eccentrics. The heavy lines show the positions when the crank has moved through the angle Θ. The circle is the path traveled by the centers of the eccentrics, and their eccentricity is r. The link pins are shown upon the arc or center line of the link, whose radius is ρ, and the length of the eccentric rods from center of eccentric to center of link pin is l. The length of half the link arc is c, and the amount by which the die-block is displaced from the center of the link arc is d. It is assumed that the center of the die-block travels on the center line X — X'; also, that the arc and the chord of the link between the link pins are equal to each other, which we can do without sensible error, and which permits us to measure d either upon the arc or the chord.

The distance from the center of the axle to the middle of the valve, represented by b, is

$O\,b = O\,m + mn + nb = O\,p - mp + mn + nb$ ......(25)

Here nb represents the length of valve stem, and may be replaced by s.

The term mp can be evolved as follows:

$mp = (c - d) \sin a$, where a is the angle of the inclination of the chord of the link to the vertical.

$$\text{But } \sin a = \frac{p\,p'}{P\,P'} = \frac{O\,p - O\,p'}{2\,c} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots (26)$$

$O\,p = O\,e + e\,p = O\,e + [E\,P^2 - (P\,p - E\,e)^2]^{\frac{1}{2}}$ ......(27)

and $O\,e = r \sin(\Theta + \delta)$, $E\,e = r \cos(\Theta + \delta)$, $E\,P = l$ and $P\,p = (c - d) \cos a$, which, substituted in equation 27, give $O\,p = r \sin(\Theta + \delta) + [l^2 - \{(c - d) \cos a - r \cos(\Theta + \delta)\}^2]^{\frac{1}{2}}$

Expanding the term within the large brackets by the binomial theorem, and rearranging the terms with the higher powers of l in the denominator, we have

$$O\,p = r \sin(\Theta + \delta) + l - \frac{(c - d)^2 \cos^2 a}{2\,l} + \frac{(c - d)\, r \cos(\Theta + \delta) \cos a}{l} - \frac{r^2 \cos^2(\Theta + \delta)}{2\,l}$$

Now, as the angle a is small as compared with the denominator l, we may consider that $\cos a = 1$ with little error, from which we derive

$$O\,p = r \sin(\Theta + \delta) + l - \frac{c^2}{2\,l} + \frac{c\,d}{l} - \frac{d^2}{2\,l} + \frac{(c - d)\, r \cos(\Theta + \delta)}{l} - \frac{r^2 \cos^2(\Theta + \delta)}{2\,l} \quad \ldots\ldots (28)$$

Similarly we find that

$$O\,p' = -\, r \sin(\Theta - \delta) + l - \frac{c^2}{2\,l} - \frac{c\,d}{l} - \frac{d^2}{2\,l}$$

$$+ \frac{(c + d)\, r \cos(\Theta - \delta)}{l} - \frac{r^2 \cos^2(\Theta - \delta)}{2\, l} \quad \ldots\ldots (29)$$

Substituting these values of O p and O p′ in equation 26, we have

$$\sin a = \frac{r \sin(\Theta + \delta) + r \sin(\Theta - \delta)}{2\, c} + \frac{2\, c\, d}{2\, c\, l}$$

$$+ \frac{(c - d)\, r \cos(\Theta + \delta) - (c + d)\, r \cos(\Theta - \delta)}{2\, c\, l} -$$

$$\frac{r^2 \cos^2(\Theta + \delta) - r^2 \cos^2(\Theta - \delta)}{4\, c\, l}, \text{ therefore}$$

$$\sin a = \frac{r}{c} \cos \delta \sin \Theta - \frac{r}{l} \sin \delta \sin \Theta - \frac{dr}{cl} \cos \delta \cos \Theta + \frac{d}{l}$$

$$- \frac{r^2}{4\, c\, l} \left[\cos^2(\Theta + \delta) - \cos^2(\Theta - \delta)\right] \quad \ldots\ldots\ldots\ldots (30)$$

For the value of m n, reference to Fig. 22 will show that it

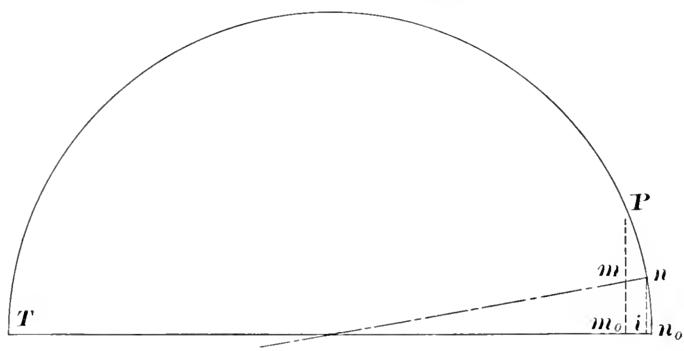


Fig. 22.

is very nearly equal to m_o i, so that we can write

$$m\, n = m_o\, i = m_o\, n_o - i\, n_o$$

and from the general properties of a semicircle

$$m\; n = \frac{\overline{P\, m_o}^2}{m_o\, T} - \frac{\overline{n\, i}^2}{i\, T} = \frac{c^2}{2\, \rho} - \frac{d^2}{2\, \rho} \quad \ldots\ldots\ldots\ldots\ldots (31)$$

We can now substitute in equation 25 the values of the terms as demonstrated in formulæ 26, 28, 30 and 31, viz.:

$$O\,b = r \sin\delta \cos\Theta + r \cos\delta \sin\Theta + l - \frac{c^2}{2\,l} + \frac{c\,d}{l} - \frac{d^2}{2\,l}$$

$$+ \frac{r\,(c - d)}{l}\,(\cos\Theta \cos\delta - \sin\Theta \sin\delta) - \frac{r^2 \cos^2(\Theta + \delta)}{2\,l}$$

$$- (c - d)\left\{\frac{r}{c}\cos\delta \sin\Theta - \frac{r}{l}\sin\delta \sin\Theta - \frac{d\,r}{c\,l}\cos\delta \cos\Theta\right.$$

$$\left.+ \frac{d}{l} - \frac{r^2}{4\,c\,l}\left[\cos^2(\Theta + \delta) - \cos^2(\Theta - \delta)\right]\right\} + \frac{c^2}{2\,\rho} -$$

$\dfrac{d^2}{2\,\rho}$ + s, and reducing, we obtain

$$O\,b = r\left(\sin\delta + \frac{c^2 - d^2}{c\,l}\cos\delta\right)\cos\Theta + r\,\frac{d}{c}\cos\delta \sin\Theta -$$

$$\frac{r^2}{4\,c\,l}\left[(c + d)\cos^2(\delta + \Theta) + (c - d)\cos^2(\Theta - \delta)\right]$$

$$+ (c^2 - d^2)\,\frac{l - \rho}{2\,l\,\rho} + l + s \;\dots\dots\dots\dots\dots\dots (32)$$

The third term in equation 32 will be at a maximum when d = c, in which case it =

$$\frac{r^2 \cos^2(\Theta + \delta)}{2\,l}$$

and for the ordinary length of eccentric rods the values will be very small, and as its omission will simplify the equation greatly, we will write

$$O\,b = r\left(\sin\delta + \frac{c^2 - d^2}{c\,l}\cos\delta\right)\cos\Theta + r\,\frac{d}{c}\cos\delta \sin\Theta +$$

$$(c^2 - d^2)\,\frac{l - \rho}{2\,l\,\rho} + l + s \;\dots\dots\dots\dots\dots\dots (33)$$

When the crank is at the dead points, $\Theta = 0$, or 180, and these values in equation 33 give

$$O\,b_{0 \text{ or } 180} = \pm\, r\,(\sin\delta + \frac{c^2 - d^2}{c\,l}\cos\delta) + (c^2 - d^2)\,\frac{l - \rho}{2\,l\,\rho}$$
$$+\, l + s \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(34)$$

and the central position of the valve will be the mean of these positions, or

$O\,o = (c^2 - d^2)\,\dfrac{l - \rho}{2\,l\,\rho} + l + s$, and as ρ should always be made equal to l, we have, as we should expect,

$$O\,o = l + s$$

Applying the value of $l = \rho$ to equation 33 and substracting the value of $O\,o$ just found, we have for the displacement of the valve from its central position,

$$c\,b = O\,b - O\,o = r\,(\sin\delta + \frac{c^2 - d^2}{c\,l}\cos\delta)\cos\Theta + r$$
$$\frac{d}{c}\cos\delta\,\sin\Theta \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(35)$$

This equation (35) appears complicated, but it may be very simply represented by a diagram first proposed by Dr. Zeuner,

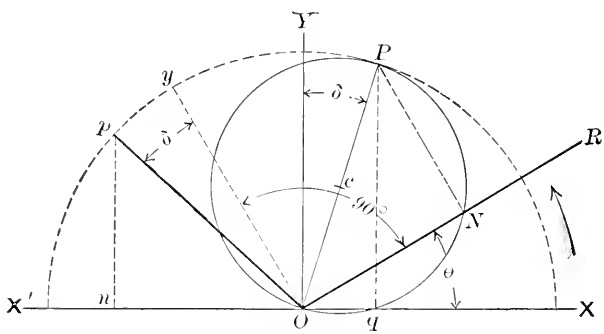


Fig. 23.

and commonly termed a Zeuner diagram. In Fig. 23 let X O X′ and O Y be a pair of rectangular axes, and assume that the crank has a left-handed rotation, as shown by the arrow. Lay off the angle Y O P = δ, toward the crank, making O P =

r, and draw on O P, as a diameter, the circle O P N, which is termed the valve circle. Then the displacement of the valve in a simple, non-reversing motion, for a given crank angle Θ, is equal to the chord O N caused by the crank line O R intersecting the valve circle.

For, if we lay off the line O p to represent the position of the eccentric corresponding to the crank position O R, we shall have the angle p O R = 90° + δ, and O n will be the valve displacement, O p being the eccentricity, and

$$O\,n = r \cos p\,O\,n = r \cos (180° - 90° - \Theta - \delta) = r \sin (\Theta + \delta) \quad \ldots\ldots (36)$$

But the triangles O p n and O P N are equal, since they both are right-angled triangles, with the sides O p and O P equal, and the angles P O N and p O n each equal to 180° — 90° — Θ — δ.

If we let v = the displacement of the valve from its central position, we can place it equal to o b in equation 35, and by representing the coefficients of the trigonometrical terms of Θ by A and B, we can write

$$v = A \cos \Theta + B \sin \Theta \quad \ldots\ldots (37)$$

$$\text{where } A = r \left(\sin \delta + \frac{c^2 - d^2}{c\,l} \cos \delta \right) \quad \ldots\ldots (38)$$

$$\text{and } B = r \frac{d}{c} \cos \delta \quad \ldots\ldots (39)$$

If we expand equation 36, we have a similar form.

$$O\,n = r \sin (\Theta + \delta) = r \cos \delta \sin \Theta + r \sin \delta \cos \Theta$$

which can also be written

$$O\,n = a \cos \Theta + b \sin \Theta \text{ if we let } a = r \sin \delta \text{ and } b = r \cos \delta.$$

Now by referring to Fig. 23 we find that r sin δ and r cos δ are the co-ordinates of the point P, which is the end of the valve circle diameter, and as the other end is at the origin O, the values of a and b definitely fix the size and location of the valve circle. We can therefore conclude that the values of A and B, as given by equations 38 and 39, will locate any number of circles, depending upon the variable d, which can be used to determine the elements of the valve motion for any values of d. We know that at full gear d = c, and at mid-gear d = o, so that we have for

Full Gear.	Mid-gear.
$A = r \sin \delta$	$r\left(\sin \delta + \frac{c}{l} \cos \delta\right)$
$B = r \cos \delta$	o . . . . . (40)

We are now in a position to construct a set of Zeuner diagrams for a Stephenson link motion, and as a practical example

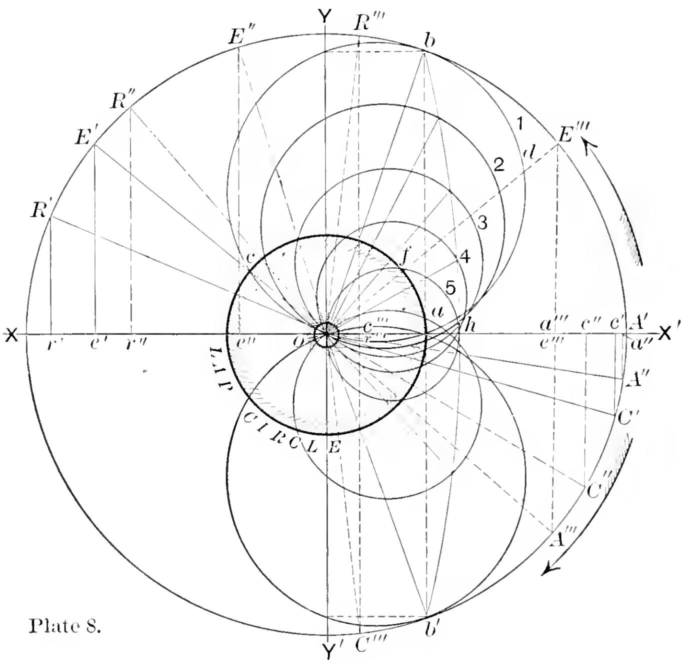


Plate 8.

ZEUNER DIAGRAM OF STEPHENSON LINK MOTION.

will take one of the N. Y. C. & H. R. Rd. 4—4—2 engines. The motion is a direct one, the eccentricity of the eccentrics being 2⅞ inches, the link motion arm of rocker 10¾ inches long and the valve arm 11½ inches, both arms hanging downward from the bearing. The steam lap is 1 inch, the exhaust clearance ⅛ inch and the lead zero in full gear. The radius of link is 60 inches and the link pins are 13 inches apart, and, of course, back of the link.

As the rocker augments the motion of the eccentrics, we must consider that the eccentrics have an eccentricity that would give the increased valve travel, or multiplying and dividing by the rocker arms, we have $2\frac{7}{8} \times 11\frac{1}{2} \div 10\frac{3}{4} = 3$ inches (approximately), which gives a total travel of valve 6 inches in full gear. Referring now to plate 8, after laying out the axes X — X′ and Y — Y′, we construct the large circle with the intersection of the axes at O as a center, using 3 inches virtual eccentricity as a radius. As the movement of the valve each side of its central position is to be represented by the distance from the center O, this circle will limit the travel of the valve.

We should next lay off the angular advance from the line Y — Y′ toward X′, which is considered as the dead point, the crank revolving in either direction as indicated by the arrows, but we do not know directly what angle it is. We can find it, however, from the lap and lead, as we know that at the dead center the valve displacement will always be equal to the lap plus the lead, and that this is equal to $r \sin \delta$, as it constitutes the reason for the angular advance. We therefore lay off the lap circle with a radius equal to the lap = 1 inch, and as the lead in full gear is zero, the lap is evidently $= r \sin \delta$. We now erect a perpendicular to X X′ at a and extend it to intersections of the large circle at b and b′, and connect these points with the center O. The lines O b and O b′ will be diameters of the full gear valve circles, and if we consider the upper circle to represent forward motion, in accordance with the upper arrow, the lower circle will represent backward motion, as shown by the lower arrow. (It is immaterial which circles are used to represent forward motion, provided that the direction of the arrow nearest to the valve circle being considered is taken to represent the motion of the crank.) Let us now consider what we have in the circle O a b c, erected upon the line O b as a diameter. The angle Y O b is the angular advance = δ and $O a = r \sin \delta = A$, and $a b = r \cos \delta = B$ in the formulæ No. 40. The valve movement from its central position at any angle of the crank can be found by measuring upon a radial line through O, making the angle with X X′ that the crank has been supposed to rotate from its dead center X′, from O to its intersection with the valve circle, O a b c. As the valve must move an amount equal to the lap (1 inch) before any opening of the port can be had, we can obtain the port opening directly by measuring on the radial line from its intersection with the lap circle to its intersection with the valve circle. For instance, if we consider that the crank has moved from X′ to E‴, or through the angle X′ O E‴ from the dead center, we will find the total valve displacement represented by the distance O d on the line O E‴ = 2 9-16 inches, and the port opening by the distance f d on the same line = 1 9-16, which is seen to be the distance on the line O E‴ between its intersections with the lap and the valve circles.

It will be instructive to follow the general features as shown by the valve circle O a b c. When the crank is at the dead end X′, the port is on the point of opening, or at "admission," as it is called, and is seen by the fact that the valve circle intersects the lap circle. The port opening increases as the crank revolves arrow-wise, until the angle X′ O b is reached, at which point the valve attains its maximum travel, 3 inches from the center. When the crank reaches E′, expansion begins, as the port closes, the valve and lap circles again intersecting. The small circle is drawn with a radius of ⅛ inch, equal to the exhaust clearance, and when the crank reaches R′ the exhaust cavity uncovers the port, and release occurs, as is determined by the intersection of the valve circle and the "clearance circle," as it may be termed. The crank has now nearly reached the opposite dead center, X, and on the return stroke, the valve maintains the exhaust opening, until the next intersection of the valve and clearance circles, which occurs when the crank reaches C′, just before returning to the dead center X′. This completes the cycle of operations—the opposite side or edge of the valve duplicates these events for the other end of the cylinder. It should be borne in mind that these several periods have been determined by the crank angle, and not by the position of the piston; if an accurate relation between the piston and valve positions be desired, the location of the crosshead for the several crank angles enumerated must be calculated or laid off to scale, to allow for the angularity of the connecting rod. If we consider a rod of infinite length, and that the large circle represents the crank circle to some arbitrary scale, we can find the piston positions by simply dropping perpendicular lines from the points E′, R′ and C′ to the axis X — X′, as shown at e′, r′ and c′.

We have so far studied only the full-gear valve circle, and must now determine the effect of shifting the link. From formulæ 40 we find that for mid-gear, B = o, so that the diameter of the valve circle will coincide with the axis X — X′

$$\text{A however} = r \left\{ \sin \delta + \frac{c}{l} \cos \delta \right\}$$

or the amount O a increased by $\frac{c}{l}$ times a b. By the specifications of the gear, we saw that l, the link radius = 60 inches, and that the link pins are 13 inches apart, and back of the link. This will be equivalent to about 14 inches at the link arc or center line, and c = half of this, or 7 inches. We must therefore multiply the length a b by half the projected link pin distance and divide by the link radius, or 2.83 × 7 ÷ 60 = .33 inch, and this amount .33 inch must be laid off on X X′ from a = a h : O h is then the diameter of the valve circle for mid-gear position of the link, and the cycle of admission, expansion (or cut-off), release and compression can be followed as shown at A‴, E‴, R‴ and C‴ for crank angles, and a‴, e‴, r‴ and c‴ for piston position, on the above basis, using the valve circle on diameter O h, designated as number 5. In order to determine the cycles for intermediate positions of the reverse lever between full and mid gear, we must construct the curve b h b′, which includes all possible positions for the ends of valve circle diameters. For this purpose, it is sufficiently accurate to draw the arc of a circle which will pass through these three points, the center of this circle being on X — X′ prolonged. We can then divide the arc b h into as many parts as we desire, equal or unequal, and construct valve circles upon the diameters connecting the curve b h with O. Three such intermediate circles are shown in plate 8, numbered, respectively, 2, 3 and 4, and a cycle of operations, like those described for circles 1 and 5, has been indicated for circle 3 at A″, E″, R″ and C″, as well as at a″, e″, r″ and c″.

A diagram can thus be prepared in a few moments which will give us a knowledge of the various operations caused by rotation of crank and shifting of link. The earlier cut-off and greater lead, as well as increased compression caused by "hooking-up," are clearly shown, as, by moving the reverse lever half way between "full" and "out" positions, we find that the cut-off has been shortened an amount e′ e″, the lead increased 1/4 inch and the compression increased by an amount of stroke c″ c′. Diagrams like this can quickly be constructed for various amounts of lap and valve travel, and form a ready solution of such problems. As stated above, the angularity of the connecting rod is uncorrected, but the correction can be readily made when desired, by laying off on X — X′ prolonged, the various points of the stroke corresponding to certain crank

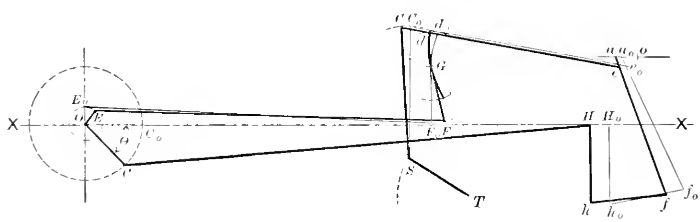


Fig. 24.

angles, using a tram equivalent to the length of the rod in the scale selected for the crank circle, and scribing from points on the crank circle on to the axis X X′ prolonged; or the piston locations may be calculated as described in connection with Fig. 10.

The mathematical consideration of the Walschaert valve gear is much simpler than that of the Stephenson link motion. The general arrangement is illustrated in diagram form by Fig. 24. H represents the crosshead and a the end of valve stem, which is moved through the radius rod C e, one end of which carries a block that may be set to any position in a curved rocking link d F, which is fulcrumed at G, receiving motion from an eccentric O E, through the eccentric rod E F, this eccentric having no angular advance, but being at right angles to the crank. The other end e of radius rod takes hold of a combining lever a f at e, the lower end of this lever being linked to the crosshead, so that the valve stem a receives a combined motion from both the eccentric and the crosshead. In Fig. 24, the thin lines represent the gear when at the front dead center, and the heavy lines when crank has moved through the angle $C_0 O C = \Theta$. The radius rod is raised and lowered by the reverse shaft arm T S so that d G can be any desired amount either above or below G within the limits of the link.

Professor Peabody, in his "Valve Gears," gives the following discussion: "If the motion of the crosshead be considered uniform at both ends of stroke, as it would with a connecting rod of infinite length, the motion which it imparts to the valve could also be given it by an eccentric with 90° angular advance, the total motion being equal to twice the lap plus twice the lead. If the block d is at the middle of the link, or at the fulcrum or trunnion G, the valve will derive motion from the crosshead only, and the mechanism will be at mid-gear. As the radius of the link is made equal to the length d e of the radius rod, the lead will be constant for all positions of d in the link, as at the dead points the link will be upright. If the point h of the link h f were fixed, the valve would receive motion from the eccentic O E only, which has no angular advance—by reducing the distance between G and d, as in 'hooking-up' the motion is reduced proportionately to the distance from G. If the block d be placed below G, the motion is reversed."

We have seen in equation 36 that the movement of a valve driven by a simple and single eccentric is $= r \sin(\Theta + \delta)$, and as the motion derived from the crosshead is equivalent to that from an eccentric having 90° angular advance (if the rod be infinitely long), we can write the displacement due to crosshead

$$e_1 = r_1 \sin(\Theta + 90^\circ) = r_1 \cos \Theta \quad \ldots\ldots (41)$$

From the proportions of the combining lever and the length of the crank $R = O C$, we have

$$r_1 = \frac{a\,e}{e\,f} R$$

The displacement of the valve due to the eccentric O E is

$e_2 = r_2 \sin \Theta$ .... (42)

in which

$$r_2 = O\,E \frac{d\,G}{G\,F} \times \frac{a\,f}{e\,f}$$

The total displacement therefore may be written

$e = e_1 + e_2 = r_1 \cos \Theta + r_2 \sin \Theta$ .... (43)

and if, as in equation 37, we let A and B be the coefficients of the trigonometrical values, we have

$$A = r_1 = \frac{a\,e}{e\,f} R \quad \text{.................................... (44)}$$

and

$$B = r_2 = O\,E \frac{d\,G}{G\,F} \times \frac{a\,f}{e\,f} \quad \text{.......................... (45)}$$

which will again be the co-ordinates of the several valve circles in constructing a Zeuner diagram for this gear. As the connecting rod is not infinitely long, the diagram will contain errors, but it will give us a fair idea of this valve motion as compared with the Stephenson.

Plate 9 gives a Zeuner diagram of a valve motion of the Walschaert type, having the same general proportions as the Stephenson motion shown on plate 8. In this the lap is 1 inch, with no lead in full gear, the crank radius 12 inches, and eccentricity of eccentric 3½ inches. The arm G F of link is 8 inches and the extreme distance of block from fulcrum d G, 6 inches. The combining lever is 26 inches total length, divided into a 2-inch portion and a 24-inch part. We lay off the rectangular axes X — X′ and Y — Y′, with the origin at O, as before, in plate 8.

From equation 44, we find that

$$A = \frac{a\,e}{e\,f} R = \frac{2}{24} \times 12 = 1,$$

and, as these values are all constant, the line b — b', erected on X — X' at point a, 1 inch from O, if made straight and per-

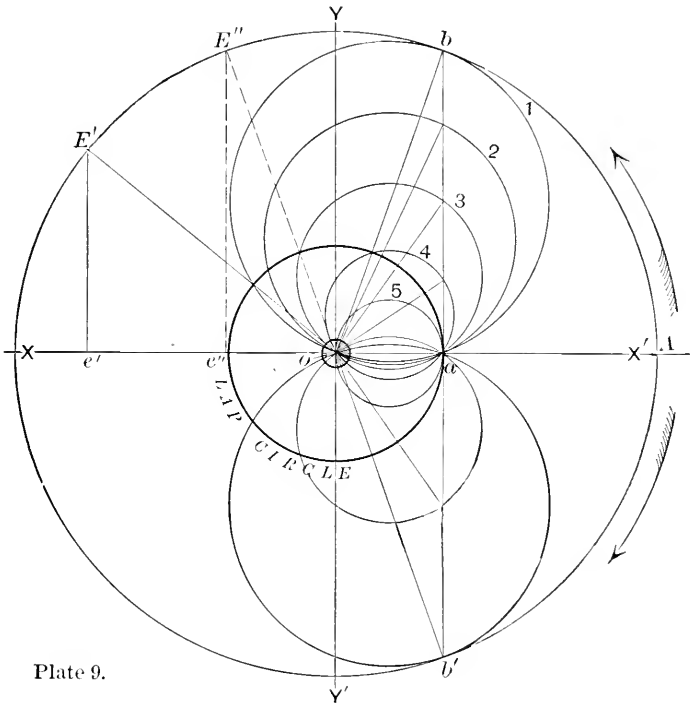


Plate 9.

ZEUNER DIAGRAM OF WALSCHAERT VALVE GEAR.

pendicular to X — X', will fix the ends of the diameters of all possible valve circles.

Equation 45 gives the value of

$$B = O\,E\,\frac{d\,G'}{G\,F} \times \frac{a\,f}{e\,f},$$

which for the greatest travel of valve becomes

$$B = 3\tfrac{1}{2} \times \frac{6}{8} \times \frac{26}{24} = 2.83'',$$

and which distance is laid off on the perpendicular above described from a to b and b'. The circles described upon the lines O b and O b' as diameters, are the valve circles for full travel of valve, which is found to be 6 inches. As seen in equation 45, the value of B depends directly upon the distance d G of the block d from the trunnion G, and by dividing a b into four equal parts, we obtain five valve circles, which correspond to those in plate 8. As neither of the motions in plates 8 or 9 have lead in full gear, we find that the points of admission and expansion are identical in both cases, for this position of the reverse lever. Valve circle number 2 shows that no lead has been gained, the admission being still at the dead center, but the travel is less and the cut-off anticipated by an amount $e' - e''$. In mid-gear, circle 5, there is still no lead, and as the dead point is the crank location at which the valve has its greatest travel, there will be no port opening, and this provides a means of stopping the locomotive. (It is customary, however, to give a small amount of lead, which, of course, will be constant for all positions of the reverse lever.) The constancy of the lead or admission point in plate 9 is due to the fact that at the ends of stroke, the crosshead alone is responsible for the position of the valve, and as this crosshead position is always the same (at the end of stroke), the valve will occupy the same position—"hooking-up" decreases the travel, but does not alter the lead. This constitutes the chief difference between the Walschaert and Stephenson motions; in the latter, the lead at mid-gear is increased by $\frac{c}{l}$ times $\overline{a\ b}$, which depends upon the length of link and its radius. If the link be longer, or if its radius be reduced, a still further increase in lead at mid-gear will be accomplished, and the formula explains why engines with short eccentric rods increase the lead so much when the lever is brought toward the center of the quadrant—this has an important effect upon the setting of the valves, especially if the engine be expected to work at high expansion ratios. Release and compression are somewhat different in the two gears, but not greatly so. While European locomotive designers largely favor the Walschaert gear, partly on account of the constant lead, American engineers prefer the increasing lead of the Stephenson motion, as, at high speeds, which must be accompanied by early cut-off, it is desirable to have greater preadmission, so that the lead will be sufficient to permit the steam to enter freely into the cylinder. An idea of the relative speed of the valve when opening and closing the port can also be gained by a further study of plates 8 and 9. We know that the angular speed of the crank is practically uniform during one revolution, and that the location of the valve is represented by the distance of the intersection of a crank line and valve circle from O. Therefore, in Fig. 25, if O a is the lap circle

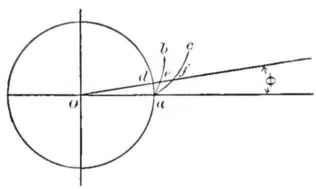


Fig. 25.

and a b a portion of a valve circle, we find that, by the time the crank has moved through the small angle Φ, the valve has changed its position by the amount d e. If, however, we consider another valve circle a c, which has this portion of its arc making a greater angle with the lap circle, or intersecting the lap circle at a greater angle than does the valve circle a b, we see that when the crank has rotated through the angle Φ, the valve has moved by an amount d f, which is greater than d e, in the first case. We therefore conclude that the greater the angle of intersection of the valve and lap circles, the faster the valve will move when the port is being opened or closed. Plates 8 and 9 both show that at early cut-off the valve circle intersects the lap circle at a much smaller angle than in full gear, and therefore the speed of the valve is greatly reduced under these conditions. This is what causes wire-drawing, and the fall in the admission line of indicator cards from the commencement of stroke to the point of cut-off—a fall sometimes so rapid that it is difficult to separate the admission or steam line from the expansion line.

Having shown how the motion of the valve may be readily studied, we must turn our attention to the valve itself, examining its peculiarities. But two general types of valves are ordinarily used on locomotives, the flat, or D valve, as it is sometimes called, and the piston valve. With the increase in cylinder dimensions and steam pressures, the flat valve became unduly large for a proper length of port, and even if partly balanced created an excessive amount of friction when moved over its seat. The piston valve has been largely introduced to overcome this difficulty, and has been very successfully operated, notwithstanding the fears expressed when it was first applied to a locomotive. The plain slide or flat valve and piston valve do not differ in the manner of steam distribution, except as to the size of port and opening, but various additional ports have been added to each type of valve, for the purpose of overcoming the slow movement of the valve at admission and cut-off referred to in connection with Fig. 25. Most of these improvements are on the basis of the Allen valve, which gives a double opening of the port when it is most needed. Valves are balanced in order to overcome or reduce the frictional resistance to their motion, but this only indirectly affects the steam admission, which is the part of the problem now under discussion.

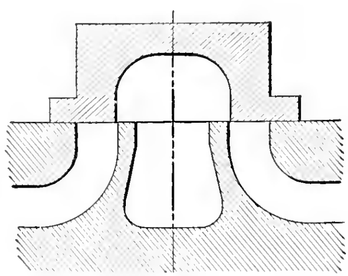


Fig. 26.

Fig. 26 shows a section of the ordinary flat valve and the controlled ports, and Fig. 27 the same for a piston valve having the same steam controlling elements. Fig. 28 illustrates a piston valve with inside admission. As the Stephenson link motion can be constructed with quite accurate adjustment for opposite ends of the stroke when a rocker arm is used with a valve having outside admission, on account of the angularity of the connecting rod, partially offsetting irregularities in the motion, if for any reason a direct motion be desired, it is advisable to use a valve with inside admission, as in both cases the eccentrics will occupy the same relative position to the crank. As already pointed out, the piston valve has a number of advantages when arranged with inside admission, as in Fig. 28, prominent among which are the absence of live steam at the ends of the valve chamber, where it is more readily cooled, and also absence of high pressure upon the valve stem packing. The motion of the valve is studied in the same way in either

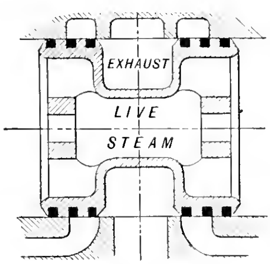


Fig. 27.

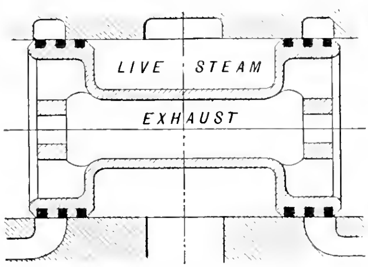


Fig. 28.

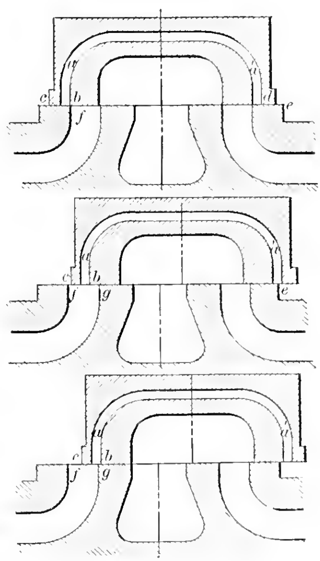


Fig. 29.

case. The valves shown in Figs. 26, 27 and 28 all have the same lap and clearance, and would all give the same steam distribution, except that, as the ports are generally longer in a piston valve (being the circumference of the valve minus the sum of the width of the bridges), the same amount of port opening by the valve gives a greater area for the steam to pass to the cylinder. The same applies to the exhaust, so it may be considered that ordinarily piston valves give a freer passage for the steam to and from the cylinder than the ordinary flat valve.

The Allen valve was designed to give a larger steam admission, and for a part of the valve travel the area of steam opening to the cylinder is actually doubled. This valve is shown in section in Fig. 29. The upper view represents the valve in its central position—the passage a a′ is termed the Allen port. The lap b c is duplicated by the distance d e, for, as shown in the middle view, when the edge c of the

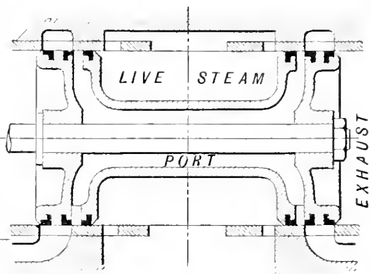


Fig. 31.

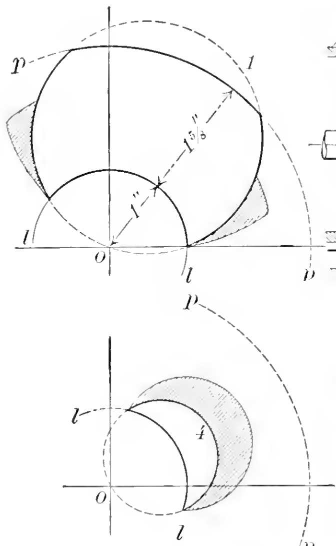


Fig. 30.

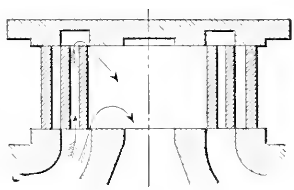


Fig. 32.

valve uncovers the port f, the passage a′ at the other end of the valve will be opened by the edge e of the seat, and steam will enter the cylinder through both openings. When the edge b reaches the bridge g, the port a will be closed by the same amount that the port f is opened by the edge c, until, as shown in the lower view, the port a is entirely closed at one end, and the port is inoperative, until the closing of the port f reverses the operation just described. This practically doubles the speed of the valve at opening and closing, by doubling the area by which steam is admitted to the cylinder, but the exhaust is unaffected. Fig. 30 illustrates, by a Zeuner diagram, the practical effect of this valve. If O is the center or origin, as before, the lap circle will be represented by l l and the outer edge of port by p p. The upper view shows a valve circle corresponding to circle 1 in plate 8, that is, in full gear. The lower view corresponds to circle 4 in same plate, or a cut-off of a little less than half stroke. The absolute movement of the valve is indicated by the circles numbers 1 and 4, in the two views, respectively, but an additional curve is drawn, which increases the distance from the lap circle l l at any crank angle in the same proportion that the Allen port gives an additional steam opening. The increase so given by this auxiliary port is shaded, and a glance is sufficient to demonstrate how much more important the results are at early cut-offs than when in the corner notches. The area lying between the valve circle and the lap circle represents the ordinary port opening, and is bounded by heavy lines.

These auxiliary ports have also been applied to piston valves, as illustrated by Fig. 31, which probably needs no further description.

It has been stated above that the Allen valve did not assist in the exhaust of the steam, as the auxiliary port is used only for the admission. The Wilson valve, however, provides a double exit, as well as a double entrance for the steam. This valve is shown in Fig. 32, and it will be seen that in connection with the vertical ports or openings in the valve, it is provided with a balance cover plate, which balances the valve and gives a double admission and exhaust for the steam.

The existence of these various devices demonstrates the recognized importance of giving the steam the greatest opportunity for rapidly entering and leaving the cylinder—the object in view being the raising of the steam line and the lowering of the exhaust or back pressure line of the indicator diagram, thus increasing its area and the work done by the locomotive. Authorities have set 100 feet a second or 6,000 feet per minute as the maximum desirable velocity of steam in its passage from boiler to cylinder, but the small port opening obtainable at high speeds and early cut-offs by locomotive link motion makes it impossible to keep within the limit above specified. In fact, it seems as though there was no way by which the opening of the valve could be made as great as it should be.

The different valves shown in Figs. 26 to 32 have all been drawn with the same steam lap and exhaust clearance, and are arranged for the same travel and width of steam port, so that the different types may be readily compared. The common points are as follows:

Steam lap	1 inch
Exhaust clearance	1/8 inch
Maximum valve travel	6 inches
Width of steam port	1 5/8 inches

The practical effect of these various forms will be studied in the following section. In order to exhibit the customary practice in this country, a table is given which shows the principal elements in current locomotive design. In compound locomotives, where two values are given, the higher figure represents the high-pressure cylinder and the lower value the low-pressure cylinder. As the lineal amount of port opening measured in a line with the valve stem will evidently not vary greatly, on account of the similarity of valve travel, it is interesting to compare the length of port with the area of cylinder which it must supply. In the table we find this value from .05 to .12, that is, for the length of port divided by cylinder area, dimensions in lineal and square inches.

STEAM DISTRIBUTION.

Having analyzed the principal locomotive valves and their peculiar features, also the motion imparted to them by the eccentrics, etc., we are ready to study the effect of these mechanisms upon the distribution of steam in the cylinder. This is the whole purpose of the valve gear, and the operation of the locomotive depends almost entirely upon the proper admission and discharge of steam from the cylinders—it is the vital feature of the machine, and as such is worthy of the most careful investigation.

In order to outline our method of examination, let us refer to Fig. 33, which represents a typical indicator diagram. The

ELEMENTS OF VALVE MOTIONS IN AMERICAN LOCOMOTIVES.

Type.	Service.	Railroad.	Diam. Drivers. (Inches)	Cylinder Diam. and Stroke. (Inches)	Type of Valve.	Diam. Valve. (Inches)	Port Length. (Inches)	Steam Port Width. (Inches)	Exhaust Port Width. (Inches)	Travel of Valve. (Inches)	Steam Lap.	Exhaust Clearance.	Full Gear Lead.	Ref. No.
4-4-0	Pass.	C. & A.	73	19x26	Piston	10	21½	2	....	6¼	1⅛	0	0	1
4-4-2	"	P. & L. E.	72	20x26	Allen	....	18	1⅝	3	6	1⅛	....	0	2
4-4-2	"	M. C.	79	21x26	Piston	....	....	....	....	6	1	⅜	0	3
4-4-2	"	Penna.	80	20½x26	Piston	....	....	....	....	7	1⅜	3-32	1-32	4
					Flat	....	....	....	....	7	1⅜	3-32	1-32	5
4-4-2	"	C. & N. W.	80	20x26	Piston	11	....	....	....	6	1¼	⅛	0	6
4-4-2	"	B. C. R. & N	75	19½x26	Piston	10	....	....	....	4¼	1	0	3-32 +F −B	7
4-4-2	"	B. & O.	78	15&25x28	Piston	15	34	1½	4½	5½	⅞ ⅝	⅛ ¼	0 ⅛	8
4-4-2	"	N. Y. C. & H. R.	79	21x26	Piston	12	....	....	....	6	1	⅛	0	9
4-4-2	"	C. R. I. & P	78½	20¼x26	Piston	11	25½	1⅞	....	5⅜	1⅛	⅛	1-16	10
4-4-2	"	C. M. & St. P.	84	15&25x28	Piston	....	....	....	....	5¼	¾ ⅝	⅛ ¼	0 ⅛	11
4-4-2	"	C. of N. J.	85	20½x26	Piston	....	25½	1¾	....	5¾	1¼	....	3-32	12
4-4-2	"	B. R. & P.	72	20¼x26	Piston	11	25½	1⅞	....	5⅜	1⅛	....	3-32	13
4-4-2	"	Penna.	80	20½x26	Flat	....	20	1½	3	7	1½	....	....	14
4-6-0	"	D. & H.	72	21x26	Wilson	....	....	....	....	5½	1	0	1-16	15
4-6-0	"	D. L. & W.	70	20x28	Piston	....	....	....	....	6	1 1-16	1-16	−1-16	16
4-6-0	"	U. P.	72	15½&26x28	Piston	....	30	1½	4½	5¾	¾ ⅝	⅛ ¼	0 ⅛	17
4-6-0	"	C. P.	69	22&35x26	Piston	14	....	....	....	6	1¼ 1	¼ ⅜	0 0	18
4-6-2	"	N. P.	69	22x26	Piston	....	....	....	....	6	1	⅛	0	19
4-6-2	"	C. R. I. & P.	69	21x26	Piston	....	....	....	....	5½	1⅛	0	1-32	20
4-6-2	"	Mo. Pac.	69	20x26	Piston	....	25½	1⅞	....	5 5-16	1⅛	0	0	21

ELEMENTS OF VALVE MOTIONS IN AMERICAN LOCOMOTIVES.—*Continued.*

Type.	Service.	Railroad.	Diam. Drivers. (Inches)	Cylinder Diam. and Stroke. (Inches)	Type of Valve.	Diam. Valve. (Inches)	Port Length. (Inches)	Steam Port Width. (Inches)	Exhaust Port Width. (Inches)	Travel of Valve (Inches)	Steam Lap.	Exhaust Clearance.	Full Gear Lead.	Ref. No.
4-8-0	Freight	B. R. & P.	55	20x26	Piston	....	....	....	....	4 9-16	$\frac{7}{8}$	0	3-32 —F +B	22
4-8-0	Freight	C. & E. I.	51	21½&33x30	Flat	....	18 21	$1\frac{3}{8}$ 2	3 3⅛	5 6	1 1	....	1-16 1-16	23
2-6-0	"	N. Y. C. & H. R.	57	20x28	Flat	....	20	1½	3	5½	$\frac{7}{8}$	0	—1-32	24
2-6-2	Pass.	L. S. & M. S.	80	20½x28	Piston	11	26½	2	....	5½	$1\frac{1}{8}$	0	$\frac{1}{8}$	25
2-6-2	"	A. T. & S. F.	79	17&28x28	Piston	15	34	1½	1½	5½	$\frac{7}{8}$ $\frac{3}{4}$	$\frac{1}{4}$ $\frac{3}{8}$	0 $\frac{1}{8}$	26
2-6-2	Freight	C. G. W.	63	16&28x28	Piston	12	....	....	....	5¼	1 1⅛	0 $\frac{1}{4}$	1-16 1-16	27
2-6-6	Pass.	N. Y. C. & H. R.	63	20x24	Piston	10	....	....	....	5½	$\frac{7}{8}$	0	0	28
2-8-0	Freight	N. Y. C. & H. R.	51	16&30x30	Piston	....	....	....	....	6	$\frac{7}{8}$	0 ¼	0	29
2-8-0	"	N. Y. C. & H. R.	63	15&28x31	Piston	....	....	....	....	6	1	0 ¼	0	30
2-8-0	"	P. B. & L. E.	51	24x32	Flat	....	....	....	....	8	1	0	1-10	31
2-8-0	"	I. C.	57	23x30	Allen	....	23	1 5-16	3¾	6	....	....	....	32
2-8-0	"	R. G. W.	56	22x28	Piston	....	....	....	....	6	1	0	1-32	33
2-8-0	"	N. Y. C. & H. R.	63	23&25x32	Piston Allen	....	 23	.... 2⅛	 3	6 6	1¼ 1	$\frac{1}{8}$ $\frac{1}{4}$	—1-32	34
2-8-0	"	A. T. & S. F.	57	21x32	Piston	....	....	....	....	$6\frac{1}{8}$	$\frac{7}{8}$	0	0	35
2-8-0	"	A. T. & S. F.	57	16&28x28	Piston	....	....	....	....	6	$\frac{7}{8}$	0 $\frac{1}{4}$	0	36
2-10-0	"	A. T. & S. F.	57	17½&30x34	Piston	....	....	....	....	6	$\frac{7}{8}$	$\frac{1}{8}$ $\frac{1}{4}$	0 $\frac{1}{4}$	37
2-10-0	"	A. T. & S. F.	57	19&32x32	Piston	13	29¾	....	....	6	$\frac{7}{8}$ $\frac{3}{4}$	$\frac{1}{4}$ $\frac{3}{8}$	0 $\frac{1}{8}$	38

line a a is the atmospheric line, and b b the boiler pressure. As we have found in connection with Fig. 20, the average steam chest pressure is always less than that in the boiler, and is here represented by the line c c. These may be termed lines of reference. In the diagram itself, starting at the beginning of the stroke d, we have the steam line or admission from d to e.

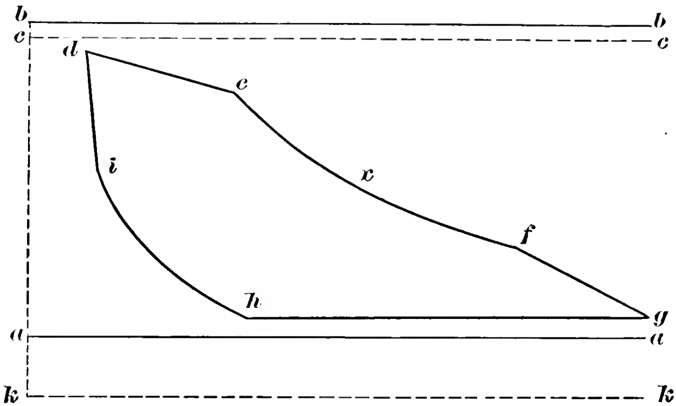


Fig. 33.

(The admission really commences at i on the back stroke, as will be considered later.) At e, the valve closes the port and expansion begins, and continues to f, where the valve opens the cylinder to the exhaust. This continues not only from f to the end of stroke g, but also on the return stroke to h, constituting back pressure, which reacts against the motion of the piston. At h the exhaust is closed by the valve, and compression begins, continuing to i, where the valve opens on account of lead, and admission takes place. This is one cycle of operations, and each portion must be studied separately.

Starting at the point d, we notice that it falls below the line c c. (Undue compression may raise it above this line, or even above b b, but we are not now considering such a case.)

This means that there must be a drop in pressure in passing through the port from the chest to the cylinder, occasioned by frictional resistance, and condensation. As explained in the preamble to this chapter, the latter is caused by the contact of the entering hot steam with the cooler cylinder heads and walls, and this is greater the earlier the cut-off. It is evident that, after a number of cycles have been performed and the cylinder has been subjected to a number of alternate heatings and coolings, it will assume a temperature somewhere between the maximum or steam temperature when entering from the boiler, and the outside or atmospheric temperature. The greater the portion of the time that hot steam is being admitted, the hotter will be the cylinder, and the smaller the time of admission, the cooler will be the cylinder. Now, if the cut-off be at one-half stroke, the admission will be one-fourth of the cycle, or one-fourth of the time of a revolution of the engine; if, however, the cut-off is at quarter stroke, the admission is only one-eighth of a revolution, consequently the cylinder is exposed to the heating action for a much less time, of the inflowing steam. It is true that the steam will still remain in the cylinder during expansion, but the terminal pressure will be lower, and so will the mean pressure and temperature; on account of this, the cylinder will actually be cooler, and of necessity the condensation will be greater.

As it is difficult to figure the amount of friction between the throttle and the steam chest and the loss in pressure which it occasions, so also is it difficult to make reliable calculations upon cylinder condensation. Naturally, the friction of the steam in the passages increases with the speed—this increase should be a benefit as far as condensation is concerned, as the period of time allowed for condensation is shorter.

From a number of locomotive indicator diagrams, after a careful study, we are able to deduce certain ratios between the initial cylinder and boiler pressures, which fairly represent the regular practice of to-day.

Relation of initial pressure to boiler pressure with full throttle opening:

Revolutions per minute	Starting	50	100	150	200	250	300	350
Initial pressure . .	.98	.95	.92	.90	.88	.87	.86	.85

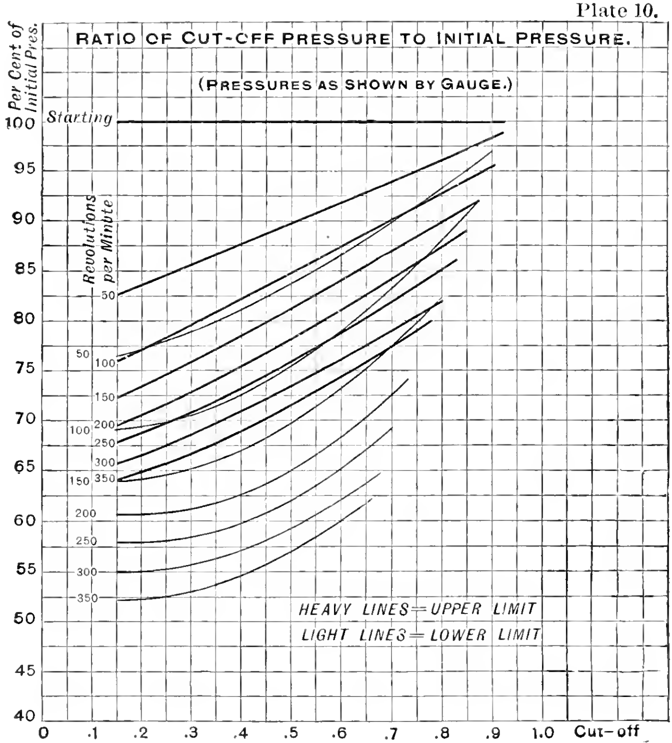
Boiler pressure considered = 1.00; all in gauge pressures.

When the speed of piston is small, or the revolutions few, the steam or admission line d e will be practically parallel with the line b b, and the cut-off pressure e will be the same as the initial pressure d. As the speed of the engine increases, however, the point e drops, the port opening not being sufficient for the steam to follow up the increasing speed of the piston. As a matter of fact, the speed of the piston increases from zero at the dead point to the middle of the stroke, while from plate 8 we see that in the ordinary working notches, the port opening begins to reduce soon after passing the dead center, and closes when the piston has its greatest speed. The drop in the line d e is therefore readily explained, and it is also evident that the shorter the steam port, the greater will be the drop. We found in the table of valve motions that the length of steam port in inches varied from .05 to .12 of the area of the cylinder in square inches. In the case of the Allen or Wilson valves, which give a double opening, the effective port length is really twice the actual length, and it should be so considered when analyzing the steam line d e.

In 1897 a committee of the Master Mechanics' Association, reporting on the "Ratios of Grate Area, Heating Surface and Cylinder Volume," gave information by which the point e may be determined; and it will be accurate enough for practical purposes to simply connect the points d and e when determined, by a straight line.

Plate 10 shows the ratio of cut-off pressure to initial cylinder pressure for various speeds of rotation and percentages of cut-off, and was worked up from the report just referred to. Two sets of lines will be noticed—the heavy lines show the upper limit, or the ratio generally obtained when the length of port is about .12 of the area of the cylinder, and the light lines when the ratio is about .05, the length of port being designated in inches and the cylinder area in square inches. The advantage of ample port length is very prominent in this series of curves. Intermediate values of port ratio may be interpolated between the heavy and light lines of similar designation. At first sight, it seems somewhat surprising to find a lower cut-off pressure at a given speed for early cut-offs, but a little thought will make clear that when the valve closes the port early in the stroke, the steam is much more wiredrawn, partly because the port is not fully opened at any time, and partly because the closing of the port begins almost as soon as the

Plate 10.



stroke has commenced and, in fact, closes very shortly after it has opened. It cannot be expected that the information given by plate 10 is absolutely accurate in all cases, but for general purposes it is thought to be quite reliable, as the information was checked against considerable data. The clearance allowed is 8 per cent and the pressures as indicated by gauge.

As an illustration of the use of plate 10, let us suppose a locomotive running 30 miles per hour, having 60-inch driving wheels, with a boiler pressure of 200 pounds per square inch and Allen valves cutting off at half stroke. It is necessary to know the revolutions per minute made at this speed, which are found to be 168. The table of ratio of initial pressure gives, for this speed (interpolate between 150 and 200) 89 per cent of boiler pressure, or $200 \times .89 = 178$ pounds at commencement of stroke. As the valves are of the double-port (Allen) type, we select the heavy lines on plate 10, marked 150 and 200, and at .5 cut-off, by interpolation, find the cut-off pressure to be 80 per cent of the initial, so that $178 \times .80 = 142$ pounds is the pressure in the cylinder at the cut-off point.

In order to facilitate the making of these calculations,

REVOLUTIONS PER MINUTE FOR VARIOUS DIAMETERS OF DRIVING WHEELS AND SPEEDS.

Diameter. of Wheel.	MILES PER HOUR.							
	10	20	30	40	50	60	70	80
50 inches	67	134	201	268	336	403	470	538
56 "	60	120	180	240	300	360	420	480
60 "	56	112	168	224	280	336	392	448
62 "	54	108	162	217	271	325	379	433
66 "	51	102	153	204	255	306	357	408
68 "	49	99	148	198	247	296	346	395
72 "	47	93	140	187	233	279	326	373
78 "	43	86	129	172	215	258	301	344
80 "	42	81	126	168	210	252	294	336
84 "	40	80	120	160	200	240	280	320
90 "	37	75	112	150	186	224	261	299

PERCENTAGES OF INITIAL PRESSURE FOR VARIOUS MEAN EFFECTIVE AND CUT-OFF PRESSURES.

M. E. or C. O. Pressure.	INITIAL PRESSURE.															
	100	110	120	130	140	150	160	170	180	190	200	210	220	230	240	250
20	20	18	17	15	14	13	12	12	11	11	10	10	9	9	8	8
30	30	27	25	23	21	20	19	18	17	16	15	14	14	13	12	12
40	40	36	33	31	29	27	25	24	22	21	20	19	18	17	17	16
50	50	45	42	38	36	33	31	29	28	26	25	24	23	22	21	20
60	60	54	50	46	43	40	37	35	33	32	30	29	27	26	25	24
70	70	64	58	54	50	47	44	41	39	37	35	33	32	30	29	28
80	80	73	67	62	57	53	50	47	44	42	40	38	36	35	33	32
90	90	82	75	69	64	60	56	53	50	47	45	43	41	39	37	36
100	100	91	83	77	71	67	62	59	56	53	50	48	45	43	42	40
110	....	100	92	85	79	73	69	65	61	58	55	52	50	48	46	44
120	....	....	100	92	86	80	75	71	67	63	60	57	55	52	50	48
130	....	...	....	100	93	87	81	76	72	68	65	62	59	57	54	52
140	....	....	....	....	100	93	87	82	78	71	70	67	64	61	58	56
150	....	....	....	....	....	100	94	88	83	79	75	71	68	65	63	60
160	....	....	....	....	....	....	100	91	89	84	80	76	73	70	67	64
170	....	....	....	....	....	....	..	100	94	89	85	81	77	74	71	68
180	....	....	....	....	....	....	....	....	100	95	90	86	82	78	75	72
190	....	....	....	....	...	....	....	....	....	100	95	90	86	83	79	76
200	....	....	....	....	....	....	....	....	....	..	100	95	91	87	83	80
210	....	....	....	....	....	....	....	....	....	....	....	100	95	91	88	84
220	....	....	....	....	....	....	....	...	....	....	....	....	100	96	92	88
230	....	....	....	....	..	....	....	....	....	....	....	....	....	100	96	92
240	....	....	....	....	....	....	....	....	....	...	....	....	....	....	100	96
250	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	100

tables giving the revolutions per minute for various diameters of wheels and speeds, and also the percentages of initial pressure for various mean effective and cut-off pressures are introduced.

As soon as the valve closes the steam port, expansion commences. No further supply can reach the cylinder from the boiler, but the steam confined in the cylinder exerts nearly its full pressure upon the piston, or, at least, the cut-off pressure. Under the influence of this confined pressure the piston continues its motion, but as the space in the cylinder, or the volume of the confined steam increases, the pressure falls, in accordance with the well-known law of the expansion of gases. Steam may be considered as expanding adiabatically or isothermally, although in general practice it does neither, nor does it expand in a curve of equal weights of steam enclosed. If we consider a piston that has moved, say, one-fourth of its stroke, and has confined back of it an amount of steam, and then permit it to complete its stroke, without the addition or subtraction of any heat whatever, we should have a case of adiabatic expansion; but as long as the steam is in contact (as it must be) with the metal walls of the cylinder, there will be heat conducted from or to it, so that pure adiabatic expansion is never realized in practice.

Again, if we consider the same piston and steam volume, but arrange to heat the steam as it expands, so as to maintain it at a uniform temperature throughout the stroke, we should have isothermal expansion, but it is apparent that we do not have such treatment in practice. Moreover, unless it be superheated, it is impossible to change the pressure of steam without changing its temperature.

It has been previously explained that upon entering the cylinder a portion of the steam is condensed; this is re-evaporated toward the end of the stroke, from which it appears that there is a greater weight of steam (as steam) in the cylinder at the point of release than at the point of cut-off. But as the valve has been closed, there has been no way by which it could reach the cylinder, and therefore it must have condensed when admitted, and later re-evaporated, when the temperature of the steam in the cylinder has become less than the cylinder itself.

We see from the above that none of the methods enumerated, though theoretically correct, does really correspond with practice, and if we use Marriott's law to express the relation between pressure and volume, it will be sufficiently accurate, and has the advantage of simplicity. This law is expressed as follows: With a constant temperature the volume of a gas varies inversely at its pressure, or, the product of pressure and volume is constant. If we have a volume of steam v at a pressure p, and then by expansion increase the volume to v′ and reduce the pressure to p′, we can, in accordance with the law of Marriott, write $p\,v = p'\,v' = \text{constant}$, and conversely,

$$\frac{p}{p'} = \frac{v'}{v} \qquad (46)$$

The definition of the law required a constant temperature to insure this proportion, but the loss in pressure due to cooling by expansion is offset by the re-evaporation during the latter part of the stroke.

From equation 46 we see that the curve of expansion is a hyperbola with rectangular asymptotes, one of which corresponds to zero volume and the other to zero pressure, in a system of rectangular coordinates and therefore one asymptote will correspond with the axis of abscissas and the other with the axis of ordinates. The values of p and v must be given from an absolute zero; that is, the pressure must be from a vacuum, and the volume must include the clearance. On locomotives a common figure for this clearance is .08 of the cylinder volume; that is, the area multiplied by the stroke. The clearance includes the volume between the valve and the piston, when the latter is at the end of its stroke, and takes in not only the space between the piston and cylinder head and the volume of the steam port, but also the contents of all pipes and cavities so connected to the port or cylinder end, that they would be filled with steam upon the opening of the valve. The clearance has an important effect upon the expansion and compression, as by it an apparent cut-off of a known ratio creates in reality a considerably longer actual cut-off. An example will best illustrate the meaning of this. Let us assume a cylinder of any given volume (area x stroke), in which the clearance is 10 per cent. If the valve cuts off apparently at 20 per cent, or one-fifth of the stroke, we should have an expansion of 5 ($1 \div 1/5$). As a matter of fact, the cut-off will really be at 30 per cent (20 per cent apparent and 10 per cent clearance), and the total volume will be 1.10 per cent (volume plus clearance), so that the ratio of expansion will be $1.10 \div .30 = 3.66$, instead of 5, as would be expected from the apparent cut-off. As we have already found a means of locating the point e in Fig. 33, we are enabled to determine how the diagram will continue to point f, by means of formula 46. We may construct a line a c at a distance from the point d, such that it bears the same ratio to the horizontal length of the diagram that the clearance bears to the cylinder volume; then the distance of any point on the card from the line a c will represent the steam volume at that point in the piston travel. The line a c is, of course, at right angles to the line a a.

At a distance below a a of 14.7 pounds to the scale of the diagram, draw a parallel line k k, and the distance from this line will represent the absolute pressure. These lines a c and k k are the coordinate axes for the hyperbola, of which the line e f will be a part. Now, if we let the pressure (absolute) at point e, which is the distance from the line k k, be represented by p, and the volume at point e, which is the distance from the line a c, be designated as v, we can, by equation 46, find what pressure we should have at any point x, distant from a c by an amount representing the volume at point x, by letting the volume at x be represented by v_x, and writing

$$p_x = \frac{p\,v}{v_x} \qquad \text{(47)}$$

That is, the pressure and volume at cut-off, multiplied together and divided by the desired volume, will give the desired pressure, pressures being absolute and volumes including clearance. So for the point f,

$$p_f = \frac{p\,v}{v_f}.$$

If the release (point f) does not occur until near the end of the stroke, it is often assumed that the expansion is continued clear to the point g. In this case the terminal pressure would be

$$p_t = \frac{p\,v}{v_t} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (48)$$

where v_t = the terminal volume. Now, in this equation, v = the total volume at point of cut-off e, including clearance, and v_t = the total cylinder volume, including clearance, so that

$$\frac{v_t}{v} = r = \text{the ratio of expansion.}$$

and therefore $p_t = \frac{p}{r}$ .... (49)

Therefore, the terminal pressure is the quotient obtained by dividing the cut-off pressure by the ratio of expansion, except when release occurs so early that the exhaust reduces the terminal pressure g still further.

Understanding as above that r is the ratio of actual expansion, including clearance, we have $\frac{1}{r}$ = the actual or real cut-off in terms of the stroke, as distinguished from the apparent cut-off d e, divided by the length of the diagram.

In the example last quoted, we found that the cut-off pressure would be 142 pounds (gauge) when cutting off at (apparently) half stroke. The actual cut-off is, however (with 8 per cent clearance) $= \frac{.58}{1.08} = \frac{1}{1.86}$, or the expansion ratio = 1.86. The terminal pressure can now be obtained from equation 49, by letting $p = 142 + 14.7 = 156.7$, absolute cut-off pressure, $p_t = \frac{156.7}{1.86} = 84.2$ pounds absolute, and $84.2 - 14.7 = 69.5$ pounds by the gauge, or above the atmosphere.

From equation 47 we can calculate any number of points on the curve c f. In the example just given, the dividend will be $p\ v = 156.7 \times .58 = 91$, remembering that the divisor v_x must be the apparent stroke + .08 for clearance, and that p_x will be in absolute pressures. These calculations can be quickly made with a slide rule, by inverting the slide and bringing 58 on the slide opposite 1567 on the rule (using the rider). Then we read directly off the rule as follows:

Apparent stroke ..	.50	.60	.70	.80	.90	1.00
Actual stroke	.58	.68	.78	.88	.98	1.08
Absolute pressure.	156.7	133.6	116.5	103.2	92.7	84.2
Gauge pressure. ..	142.0	118.9	101.8	88.5	78.0	69.5

If the exhaust opens or release occurs at .8 of the stroke, the real volume is .88 and the pressure f would be 88.5 pounds above the atmosphere.

Mr. D. L. Barnes, in his revision of Wood's Compound Locomotives, gives a ready method of constructing this curve graphically. In Fig. 34, let k k be the zero line of pressures or

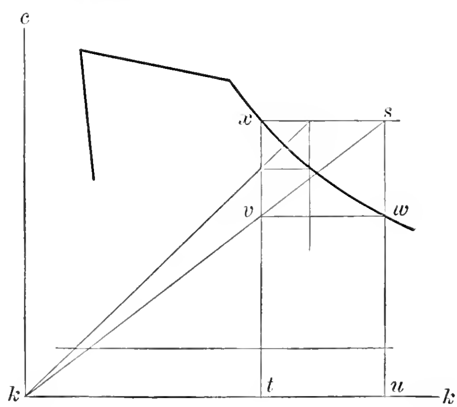


Fig. 34.

vacuum line, k c the zero line of volumes or clearance line, and x a known point on the hyperbola. Through x draw x s parallel to k k, and x t and s u perpendicular to k k, and also draw a line from k to s. Through the point v, where k s crosses x t, draw v w parallel to k k, and where this line cuts s u at w is a second point on the curve. Any number of such points can be found in a similar manner. Plate 11, however, gives a solution without calculations or construction. In it are a complete set of hyperbolic expansion curves, covering every cut-off and pressure up to 200 pounds above the atmosphere. As indicated, the clearance is assumed to be 8 per cent of the volume, and the curves are constructed on that basis, as this approximates to usual conditions. The pressure during expansion can be determined by inspection for any point of the stroke. As an example, let us consider the case, which we previously calculated, of 142 pounds (gauge) cut-off pressure, with the cut-off at half stroke (apparent). By examining plate 11 we find that 142 pounds and .5 cut-off intersect slightly below one of the hyperbolas, and by following this curve (or really an imaginary one slightly below it), it is seen that it intersects the .6, .7, .8, .9 and 1.0 stroke verticals at 119, 102, 88.5, 78 and 69.5 pounds, respectively, which results are the same as we have secured by calculation.

Plate 11 (at end of book) shows us that with the high pressures now existing there is not much danger of expanding down below the atmospheric line; that is, to a partial vacuum. If we consider a cut-off and pressure of 10 per cent and 50 pounds, however, we see that the expansion will reach atmospheric or zero (gauge) pressure at 70 per cent of the stroke, and beyond this there will be a partial vacuum formed in the cylinder, which will continue until release, when air will pass into the cylinder from the smokebox. As soon as a vacuum is formed, the work performed by the piston is negative; that is, it absorbs work instead of creating it. If, in formula 48, we let $p = 50 + 15$, $v = 10 + .08$ and $v_t = 1.00 + .08$, we can put the terminal pressure

$$p_t = \frac{65 + .18}{1.08} = 10.8 \text{ pounds absolute, or } 10.8 - 14.7 = -$$

3.9, or, say, 4 pounds of vacuum, and this, of course, tends to pull back or retard the movement of the piston.

Continuing our study of Fig. 33, the portion of the diagram from f, when release occurs, to g, the end of the stroke may be represented by a straight line. The point g itself cannot well be set in advance, and it depends greatly upon the design of the cylinder and exhaust pipe, as well as upon the speed of the engine and point of cut-off, pressure, etc. It is generally, however, at the same height above the atmospheric line a a as the portion of the curve g h, or, in other words, g h is normally parallel to a a, and if we determine the distance from a a to g h we will locate the point g. The back pressure line g h depends also upon the construction of the cylinder passages and exhaust pipes, the speed of the engine, the elements of the valve gear and pressure carried. Of the adjustable elements, the size of exhaust nozzle and the clearance of the valve probably control the back pressure more than any other factors. Fig. 35 illustrates this point. The full line was taken from an engine with 18½ by 24 inch cylinder, 23-inch ports, 4⅜-inch exhaust

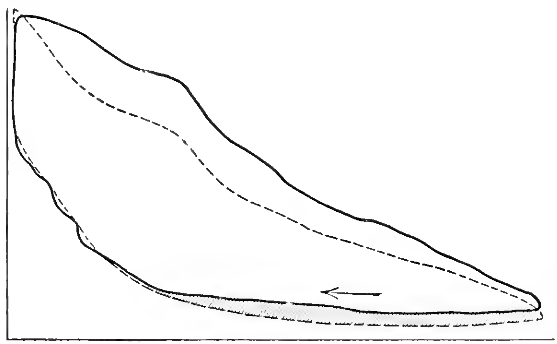


Fig. 35.

nozzle and no exhaust clearance, running at 50 miles an hour. The broken line was from a 19 by 24 inch cylinder locomotive, with only 17¼-inch ports, but with 5-inch exhaust nozzle and 1/16-inch exhaust clearance, the speed being the same in both cases. It is very important to reduce the back pressure to the lowest limit, and exhaust nozzles should be as large as possible without reducing the steaming qualities of the boiler below the needed capacity. Bridges and other temporary expedients should not be permitted; if more draft be needed, the nozzle must be reduced. The clearance of the valve should be large enough to take full advantage of the nozzle opening, and must be greater for high-speed than for slow-speed engines. Care should be taken when designing cylinders to see that the exhaust passages are ample. As the back pressure acts nearly throughout the stroke, it may cause a great reduction in the area of the diagram, and consequently in the work performed by the engine. This is shown in Fig. 35, where the difference in back pressure in the two cases is shaded, from which will be seen the important part which it plays in the engine cylinder.

The average back pressure in locomotives with single expansion cylinders is probably about eight pounds per square inch, although it is often much greater at high speeds, particularly if the ports and passages be restricted. One criticism often made upon the Allen valve is that, while it affords double opening for admission, it gives no particular advantage for the free exhaust of the steam. The exhaust always opens the port to a much greater extent, however, than the steam edge, as the latter has a considerable lap, while the exhaust edge of the valve is generally constructed with clearance. The Wilson valve gives a double opening for exhaust as well as for steam, and should make a very smart engine.

At slow speeds the back pressure may be almost zero; that is, the line g h may coincide with a a. This is generally found when starting with the lever in full gear. In such cases there will generally be a "hump" discoverable in the exhaust line, near the middle of the stroke, caused by the exhaust at high

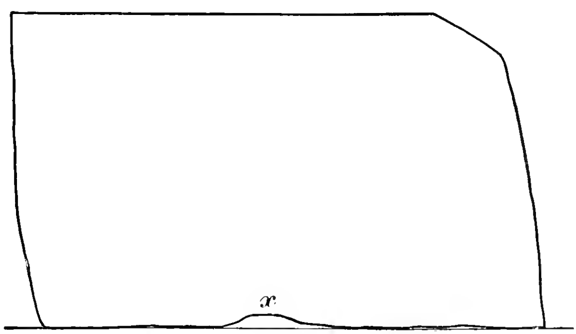


Fig. 36.

pressure from the opposite cylinder blowing back over the partition in the exhaust pipe. Some designers extend the bridge clear to the top of the nozzle, thereby having a "double exhaust pipe," in which case there would be no disturbance of the exhaust line. Fig. 36 shows this rise in the line g h at the point x, being taken from a single nozzle engine. A high-speed passenger locomotive, when making a particularly good run over a division 140 miles long with a 10-car train, weighing 440 tons back of the tender, showed by indicator diagrams back pressure as follows:

Speed.	Cut-off.	Back Pressure.
Starting	21"	0
20 miles per hour..................	17"	10 pounds
50 miles per hour..................	10"	15 pounds
70 miles per hour..................	10"	20 pounds

The stroke was 26 inches and the drivers were 80 inches in diameter. This engine has been remarkably successful, and, although fitted with piston valves, showed quite a high back pressure at high speed. The effect of the point of cut-off upon

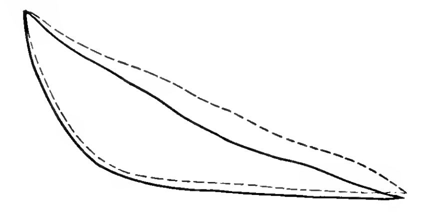


Fig. 37.

the back pressure is shown by Fig. 37, which is a copy of indicator cards, both taken from the same engine at 70 miles an hour, the full lines with a cut-off of 8½ inches and the broken lines at 10¾ inches. In this engine the valve, as mentioned above, was of the piston type, 11 inches in diameter, and had a clearance of ⅛ inch and a maximum travel of 6 inches. As the cylinder diameter was 20 inches, these dimensions seem quite liberal, but the back pressure was higher than wished for.

The next period that we have to consider is that of compression from h to i. Like the expansion curve e f it depends entirely upon the clearance volume and pressure at port closure. As soon as the valve covers the port and prevents the escape of the exhaust steam to the atmosphere, the pressure of the confined steam begins to rise, and continues to do so as long as its volume is diminishing. Most of our remarks about the expansion of steam apply with equal force to its compression, except that perhaps the rectangular hyperbolas do not fit the actual case quite as closely as they do in expansion. They will, however, answer our purpose sufficiently well, so we consider that the same law applies to compression as to expansion.

When compression begins we have a certain volume of steam ahead of the piston, which, including the clearance, we designate by v. The pressure at that point is the back pressure measured above a vacuum, which we will call p. The clearance is a fixed quantity $= v_t$; then the final pressure, due to compression at the end of the stroke, above a vacuum, will be from equation 48.

$$p_t = \frac{p\, v}{v_t}$$

That is, the final pressure (absolute) at end of stroke, due to compression, will be the quotient of the absolute back pressure multiplied by the volume ahead of the piston (including clearance) at the instant of port closure, divided by the clearance.

Steam economy demands a final compression pressure not far below the initial pressure; the proper cushioning of the reciprocating weights also requires a definite terminal pressure, so that we are often bound to consider equation 48 in a slightly different form. Thus, if p_t is desired to have a certain value, and p and v_t are known or fixed, then we must determine the volume v, which will cause the pressure p_t at the end of stroke. We therefore write

$$v = \frac{p_t\, v_t}{p} \quad \ldots\ldots\ldots\ldots \quad (50)$$

Or, the volume ahead of the piston at instant of port closure, including the clearance, is equal to the product of the desired terminal absolute pressure and the volume of clearance divided by the absolute back pressure.

Any point on this curve can be found by formula 47, starting either from the terminal end or the port closing end, depending upon whether we are considering an actual case or working backward through a hypothetical example. The compression curve may also be constructed in a similar manner to the method suggested for constructing the expansion curve, with modifications to suit the reverse operation.

In Fig. 38, let k c be the clearance line, constructed as in Fig. 33, also the vacuum line k k. Through point h, where it is desired, the curve shall start, draw the line h j, parallel to

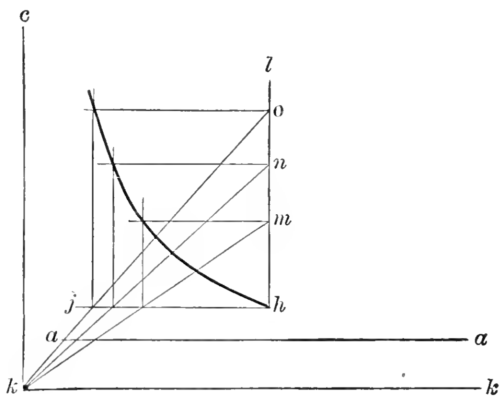


Fig. 38.

k k, and h l perpendicular to it, and from k draw any number of lines, k m, k n, k o intersecting h l and h j. From the points of intersection of these lines with h j, erect perpendiculars, and from the points m, n and o draw lines parallel to k k. The corresponding lines cross at points in the desired curve.

Plate 11 can also be used in the same manner as for expansion, if the 8 per cent clearance corresponds closely with the case under consideration. As an example, if we desire 95 pounds terminal pressure, and have a back pressure of eight pounds by the gauge, we find that the hyperbola intersecting the zero vertical at 95 pounds, crosses the eight-pound line at point .7 of the stroke, or .3 uncompleted. The same by equation 50 would be

$$v = \frac{(95 + 15) \times .08}{8 + 15} = \frac{110 \times .08}{23} = .38$$

and as the clearance = .08, the portion of stroke to be completed will be .38 — .08 = .30, as found by the plate. The amount or ratio of clearance exercises a great deal of influence upon the compression of the steam. For example, if we should have an engine with only one-half as much clearance, or 4 per cent of the volume, we would obtain, from formula 50:

$$v = \frac{110 \times .04}{23} = .19,$$

and subtracting the clearance, .19 — .04 = .15 of the stroke, or one-half the distance from completion of stroke found for the 8 per cent clearance. If the compression began at the same point in the stroke, the terminal pressure would be $p_t = \frac{23 \times .34}{.04} = 195$ pounds absolute or 180 pounds by gauge.

COMPRESSION.

Our study of the valve gears usually applied to locomotives by means of plates 8 and 9 showed us that as the rate of expansion increased by cutting off earlier, the exhaust closure was also hastened. At high speeds compression is more needed, partly to overcome the effect of inertia of the reciprocating parts, and partly to insure the proper initial pressure, and as these high speeds are ordinarily accompanied by an early cut-off, the valve motion automatically produces the greater compression. The terminal pressure may, however, reach a higher point than is considered desirable, and if the cylinder clearance be small, it will undoubtedly do so, unless the exhaust closure be unduly retarded by an abnormal amount of exhaust clearance in the valve. This is objectionable at low speeds, as the steam will blow through both ports when the valve is near its central position, and it will also release the steam too soon during expansion, reducing the work performed. With special valve movement mechanisms, the clearance may be reduced, but with the Stephenson and Walschaert, a smooth-running engine requires a moderate amount. With some of the special motions above referred to, such as the Allfree, we understand that the clearance can be as small as 2 or 3 per cent.

This valve gear has an attachment which gives the valve an exceedingly rapid motion at certain parts of the stroke, delaying it at others, which produces a late compression, thus permitting a very small clearance without unduly raising the final pressure. The Allfree valve gear consists of a rocker arm connected, as usual, to a Stephenson link motion, but having the valve rod pin or journal arranged as an eccentric shaft, which eccentric is rotated by mechanism from the crosshead, the effect being either to anticipate or delay the valve motion as controlled by the rocker at certain parts of its travel. The rocker arm has a toothed sector rotating about a center common to the rocker bearing, which sector meshes into a pinion forming part of the eccentric valve rod bearing. The sector is oscillated independently of the rocker by levers and rods connected with the crosshead, and the motion of the sector relative to the rocker produces a rotary displacement of the eccentric shaft through the pinion, and this eccentric practically advances or retards the valve rod bearing in a fore and aft direction. The resultant valve motion is quite complicated, but an idea of the distribution can be obtained as follows: In Fig. 38a a vertical

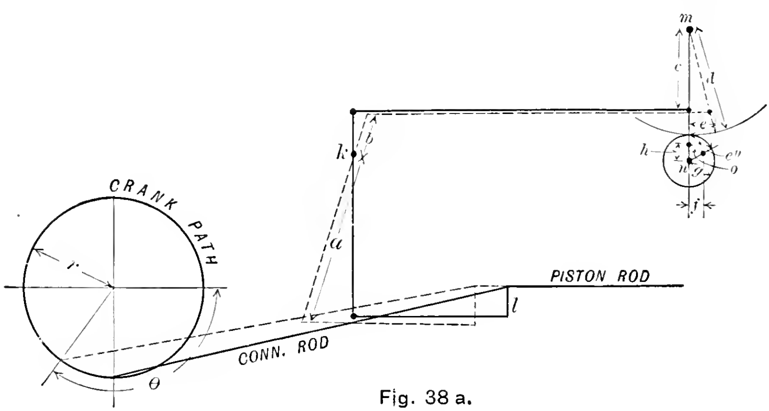


Fig. 38 a.

lever fulcrumed at k, the lower end being connected with the crosshead l, gives motion to the sector of radius (at pitch circle) d, whose center m coincides with that of the rocker, the rod from lever taking hold of sector at a distance c from the center. The lateral displacement of any point at the bottom of the sector will then be (supposing a connecting rod infinitely long)

$$e = \frac{r \cos \Theta \, b \, d}{a \, c}.$$

The valve rod pin center of rocker "n" will have a displacement which can be found by an ordinary Zeuner diagram, and which we will call e′. Then the rotation of the pinion of pitch radius g will depend upon the relative motion of the sector and the rocker, and if this is represented by e″ and is measured on the pitch line of the pinion, we shall have

$$e'' = e \pm e'$$

The valve rod is connected at o, the eccentric distance from n to o being represented by h, and the lateral displacement of o is $f = h \sin \frac{e'' \, 180}{g \, \pi}$, the fraction or sine being in degrees for convenient use of tables. As this displacement either increases

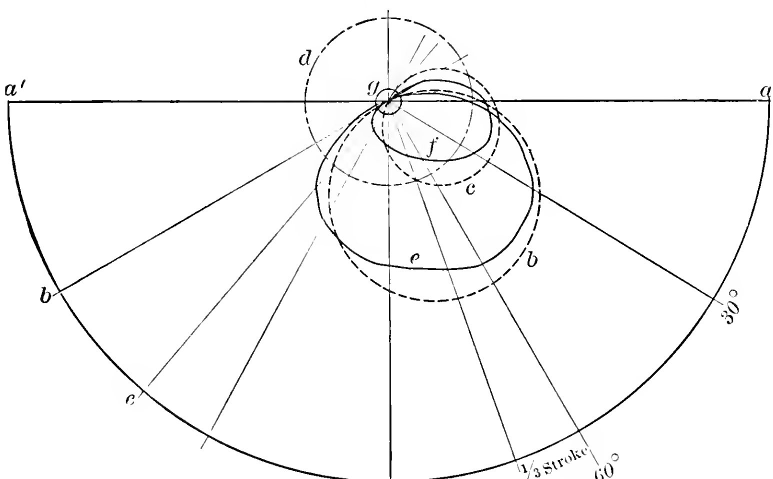


Fig. 38 b

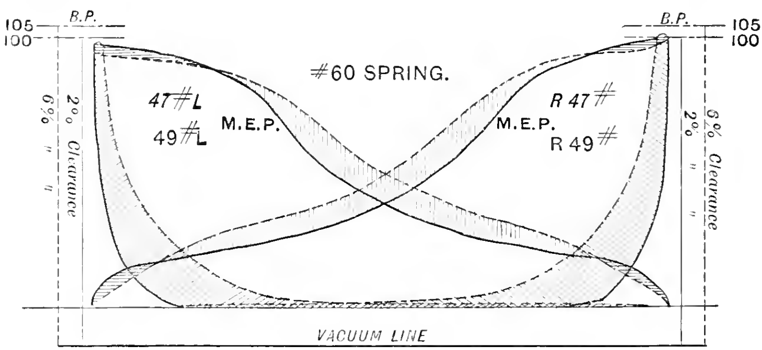
or diminishes the valve distance from center, we have the total valve displacement = e′ ± f.

From an actual case (Kansas City, Mexico and Orient Railway, locomotive No. 101), we have r = 13″; a = 26″; b =

$7\frac{1}{2}''$; $c = 7\frac{1}{2}''$; and $d = 9''$; then $e = \dfrac{13 \times 7\frac{1}{2} \times 9}{26 \times 7\frac{1}{2}} \cos \Theta$ $= 4\frac{1}{2} \cos \Theta$, and this is the cosine of a circle of $4\frac{1}{2}''$ radius, as a a′ in Fig. 38b. The distance e′ can be measured from the radii vectors of the valve circles b, c. The lap circle is marked d. Therefore e″ is the sum or difference of the cosines of a a′ and the radii vectors, and can be determined by measurement. From this we can find f, when we know that $h = \frac{3}{8}''$; $g = 2''$ and $\pi = \dfrac{22}{7}$, so that

$$f = \tfrac{3}{2} \sin \frac{e'' \times 180 \times 7}{2 \times 22} = \tfrac{3}{2} \sin (29\, e'')$$

and this amount must be added to or subtracted from e′, the regular motion of the rocker. This has been done and the



DATA.

CARDS	CYLINDERS	% CLEARANCE	B. PRESSURE	R.P.M.	I.H.P.	WATER H.P.H
47 L & R	14 x 15	6	98	172	93.5	28.35
49 L& R	14 x 15	2	105	172	97.0	23.72

Fig. 38 c.

lines e and f so produced, see Fig. 38b. We find here that when near mid gear, the cut-off occurs earlier for a given position of the reverse lever than with the Stephenson motion, as seen where the valve curves c and f cross the lap circle d, and that the release and compression are very much delayed as determined by the crank angle for the intersections of the valve curves c and f with the exhaust clearance circle g. This combination brings about the indicator card shown by Fig. 38c, where the later release and compression shown by the full lines permit a much smaller clearance than could be used with the ordinary motion, shown by the dotted lines, also increasing the effective work of the engine by reducing the back pressure.

The Allfree valve gear is not used on locomotives separately or apart from the Stephenson link motion or the Walschaert valve gear, but is used in conjunction therewith, so as to delay the exhaust opening and exhaust closure at all points of cut-off, and to increase the ratio of expansion and decrease the negative work of compression by the reduced volume in compression resulting from the later closure of the exhaust port.

When the terminal pressure is raised by compression above the initial pressure, a loop is formed in the diagram, which

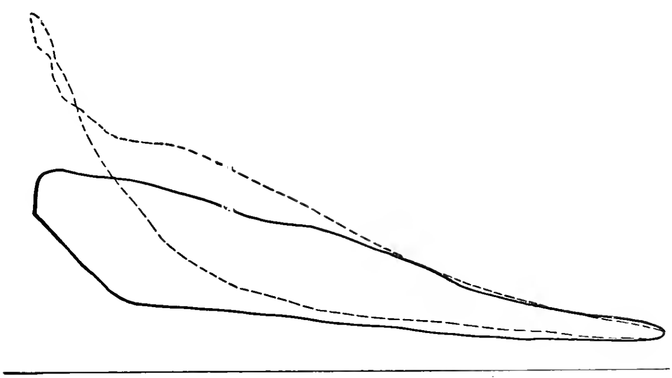


Fig. 39.

represents negative work. If the inside valve clearance be not already excessive, the loop can be overcome by cutting out more clearance from the exhaust edge of the valve. If this is not possible, a special cylinder head may be prepared which will contain additional clearance volume, and such a method is sometimes resorted to, particularly in compound engines, in which the low-pressure clearance ratio is often small. Fig 39 shows the result obtained by increasing the exhaust clearance of the valve from 7-32 inch to ⅜ inch on the low-pressure cylinder, the speed being 70 miles an hour; the change resulted in an increase from 12.24 to 16.01 pounds mean effective pressure—a gain in the work done of about 30 per cent. The broken line was taken with the smaller clearance.

The remaining portion of the diagram, from i to d, needs little discussion. It is at this point that the valve opens the port for pre-admission. The line rises suddenly, and almost in a straight line to d. At great speeds the indicator pencil will often overjump the admission line, due to the inertia of the moving parts of the instrument, and a series of up and down

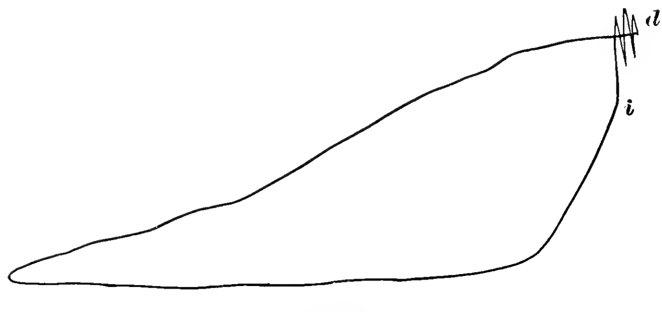


Fig. 40.

strokes will follow. This is indicated by Fig. 40. As the compression evidently stopped at i by the opening of the port, the zig-zag at d is due to inertia of the indicator parts, and should not be considered as a part of the real diagram.

It must not be expected that the various changes from admission to expansion, or from back pressure at exhaust to compression, will be always indicated by sharp angles or sudden changes of curvature on the indicator card. At very slow speeds the points will be plainly marked, but as the speed increases they blend together or pass from one portion to another by easy curves, so that it is often difficult to say by inspection of an indicator card, just where expansion or compression begins or ends. If we have a record of the valve motion, we can lay these points off quite definitely, but not entirely so, as the lost motion in the joints and the spring of the various parts which compose the valve gear cause a very irregular action at high speeds, as has been pointed out by Professor Goss. This gentleman has also called attention to the fact that long and bare indicator pipes prevent the formation of a diagram that actually shows the work going on in the cylinder. In order to demonstrate the effect of the pipe, he took simultaneous diagrams from the locomotive on the testing plant at Purdue University, with two indicators, one connected to the cylinder by a pipe 3½ feet long, and the other by simply a pipe ell. Fig. 41 shows these two diagrams superimposed, that taken from the long pipe in broken lines, and the one directly attached to the cylinder in full lines. The general effect of a

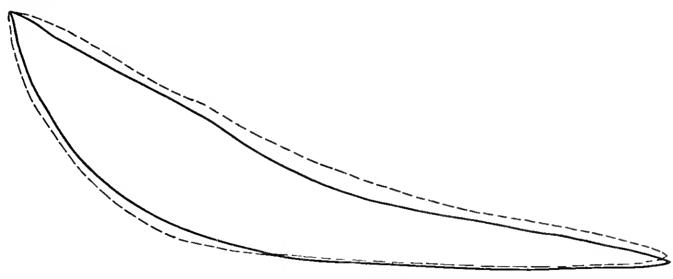


Fig. 41.

long indicator connection or pipe is to record the various events later than they actually take place in the cylinder, the transitions from one action to another are more gradual, and the area of the card is greater than it should be. Thus from a number of such tests he found the excess of power shown by the indicator at the end of the 3½-foot pipe over that shown by the one with the short connection to vary from 1.5 to 17.2 per cent, as illustrated by the following table:

Speed—	Excess power indicated.
25 miles per hour	1.5 per cent
30 miles per hour	2.1 per cent
35 miles per hour	2.9 per cent
40 miles per hour	4.9 per cent
45 miles per hour	8.4 per cent
50 miles per hour	14.0 per cent
55 miles per hour	17.2 per cent

As, from the nature of the work, the pipes are always unduly long on a locomotive indicated in actual service, it will appear that the error is likely to be considerable. As in the experiments to which we have just referred, the pipe was well wrapped and bent with easy curves, we can gather some idea of the errors that will creep in when the pipes are bare and the changes in direction made with ordinary elbows, as so often is the case in a road test.

The study which we have just given the indicator card is made not only that a clear idea may be had of the distribution of the steam during the various epochs of its stay in the cylinder, but that sufficient information might be placed conveniently before us, so that we can prophesy closely what kind of a card will be produced by an engine having certain peculiarities of valves and gears. It is frequently of great importance to be able to determine, without the delay incident to a test and indication, what the distribution of steam will be under certain conditions of speed, etc., and such determinations may be made by the use of the tables, plates and formulæ embodied in this section.

WORK OF STEAM.

In passing through the various changes and events just studied, the steam does useful work, and it is of the utmost importance to know what amount of work can be obtained from a locomotive cylinder (or piston) under assigned conditions. The various rules which have been given to enable us to prepare, in advance of a test, an indicator diagram, will also instruct us how to determine the work that can be performed by the locomotive. As the indicator card illustrates the work performed by a unit of piston surface, the determination of the work done by this unit of surface will fix the whole work of the machine. An indicator card represents, to a certain scale, the steam pressure in the cylinder at every part of the stroke, which in turn is represented by a scale which bears the ratio of the length of the card to the piston stroke. Now if we measure the actual area of such a card and divide it by the actual length of the card, we will have the mean effective pressure, as it is called, or the average effective pressure during the stroke, upon a unit of the piston surface, determined by the scale of pressure to which the diagram was constructed. For instance, if the scale of pressures be 100 pounds to the inch, the length of the card be 4 inches, and the area d e f g h i (in Fig. 33) be 6 inches, we should find $6 \div 4 = 1\frac{1}{2}$ inches average height, or 150 pounds as the mean effective pressure.

When we have actual indicator diagrams from the desired machine, the process is extremely simple, as by means of a planimeter we merely obtain the inclosed area, and divide it by the length of the card, which gives the average height, and knowing the scale of the indicator spring which was used, by multiplication obtain at once the M. E. P.

When we must determine this M. E. P. in the absence of actual diagrams, it is necessary to be guided by the rules stated in the last section. As it will be in this case a matter of calculation in place of measurement, we can disregard the scale of an indicator card and work directly with the pressure in pounds and the stroke or piston travel in inches.

The work done in the cylinder of an engine is both positive and negative. When the confined pressure assists the motion of the piston, the work is positive, and when it opposes this motion, it is negative. Thus, in Fig. 33, the pressure represented by the line d e f g assists the piston and performs positive work. The back pressure represented by the line g h i d resists the piston and performs negative work. As our calculations are based upon a reference line which is either the atmospheric pressure a a or the absolute zero k k, the effective work will be the difference between the positive and negative portions of the steam action.

The positive work is represented in Fig. 42 by the area d e f g b a, in which, for purposes of calculation, we can consider the actual initial pressure in pounds above the atmosphere to be the height a d and the piston travel in inches to be a b. Here the distance from k c to a d is the volume of clearance, divided by the area of cylinder; that is, it is represented as an addition to the stroke, which gives an equal clearance volume. the line k l is below a b the amount of atmospheric pressure, say, 15 pounds, and it will be better to reckon all our pressures from the absolute zero. For convenience of study, let us divide the stroke into three portions, admission, expansion and re-

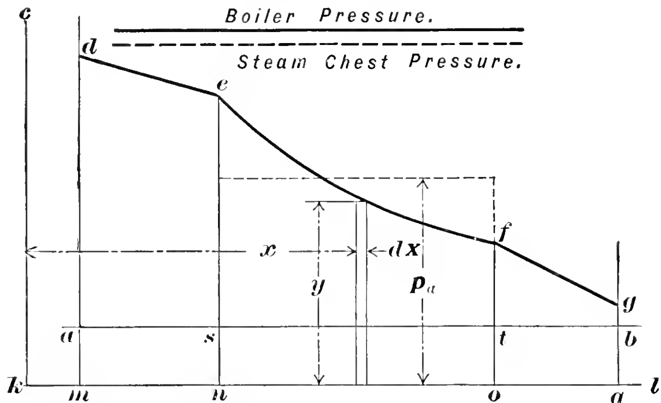


Fig. 42.

lease, and obtain the work in inch pounds performed by a square inch of the piston surface during each portion.

For the period d e.—From the table giving the relation of initial pressure to boiler pressure, we can determine the value a d for the speed desired, and from plate 10 the cut-off pressure s e. As these are both in gauge pressures, by adding 15 to each we obtain the absolute pressures m d and n e. As the line d e may be considered straight, the work done in inch pounds above a vacuum will be $\frac{\overline{md} + \overline{ne}}{2} \times \overline{as}$, m d and n e being in pounds and a s in inches.

For the period e f.—We found in formula 47 that the curve of expansion could be located at any point, if the pressure and volume of some definite point be known. We have already fixed the point e by our knowledge of n e and a s, and as we also know k m (the clearance stroke), we have n e to represent p and k n to represent v in this equation. In order to determine the area of work e f o n, let us consider k as the origin of a system of rectangular coördinates, and the equation of the expansion curve e f, as referred to this system, will be x y =

p v, x and y being any coördinate points on the curve. At f, we have $o\,f = p_f$ and $k\,o = v_f$ and $p_f\,v_f = p\,v = x\,y$. Now, if we let the unknown area e f o n be represented by A, we have for the area of the elementary strip at distance x from the axis, k e, $d\,A = y\,dx$, but we have just seen that $y = \frac{p\,v}{x}$.

so that $d\,A = y\,dx = p\,v\,\frac{dx}{x}$.

Now integrating between v and v_f (k n and k o), we have

$$A = \int_{v}^{v_f} p\,v\,\frac{d\,x}{x} = p\,v\,\log v_f - p\,v\,\log v =$$

$$p\,v\,(\log v_f - \log v) = p\,v\,\log\frac{v_f}{v},$$

but $\frac{v_f}{v}$ is the ratio of expansion for this portion of the stroke, which we may designate by r_f so that we can write simply

$$A = p\,v\,\log r_f$$

To obtain the average pressure p_a, or $\frac{A}{v_f - v}$, we put

$$p_a = \frac{p\,v\,\log r_f}{v_f - v} = p\,\frac{\log r_f}{r_f - 1} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (51)$$

$$\text{as } \frac{v}{v_f - v} = \frac{1}{\dfrac{v_f - v}{v}} = \frac{1}{\dfrac{v_f}{v} - \dfrac{v}{v}} = \frac{1}{r_f - 1}.$$

If the expansion continued without release to the end of the stroke, the ratio of expansion would be r, as before explained, and the average pressure, from n to q, would be

$$p_m = p\,\frac{\log r}{r - 1} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (52)$$

which is the general way in which the formula is written. It must be remembered that the pressures are absolute, the volumes include the clearance, and that the Hyperbolic or Napierian logarithms must be used.

Having now determined the average pressure, the work in inch pounds for the section e f o n $= p_a \times \overline{s\,t}$.

For the period f g.—We found from equation 47 that the pressure $\overline{o\,f} = p_f = \dfrac{p\,v}{v_f} = \dfrac{\overline{n\,e} \times \overline{k\,n}}{\overline{k\,o}}$. We will have to assume $\overline{q\,g}$, being guided by the data given near Fig. 36, which pressures are gauge, and to which we must add 15 to reduce them to absolute pressures. As the line is straight from f to g, we have the work done during this portion of the stroke $= \dfrac{\overline{o\,f} + \overline{q\,g}}{2} \times \overline{t\,b}$ in inch pounds. Now, by adding the work in inch pounds done during the three portions of the forward stroke, and dividing the sum by $\overline{a\,b}$, the stroke in inches, we obtain the average positive pressure above a vacuum.

The negative work is represented in Fig. 43 by the area g h i d a b, the various units of representation being the same

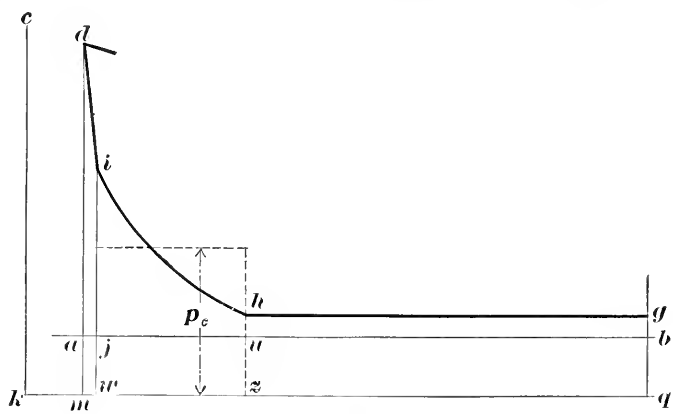


Fig. 43.

as for Fig. 42. The parts of stroke into which we will divide it are the exhaust, compression and pre-admission.

For the period g h.—The amount of back pressure b g must be estimated as just shown for determining the value of $\overline{q\,g}$, with 15 pounds added to give the pressure $\overline{q\,g}$ from a vacuum. The work done will be $\overline{q\,g} \times \overline{b\,u}$, in inch pounds, $\overline{b\,u}$ being, of course, in inches.

For the period h i.—We know from equation 48 that the pressure $\overline{w\,i} = p_t = \frac{p\,v}{v_t} = \frac{\overline{z\,h} \times \overline{k\,z}}{\overline{k\,w}}$, and if we let $r_c = \frac{\overline{k\,z}}{\overline{k\,w}}$ = the ratio of compression, we can write area h z w i = p v log r_c and the average pressure $p_c = \frac{p\,v \log r_c}{v - v_c}$, v_c being equal to k w, the clearance volume plus incompleted stroke, and by a treatment similar to that in reducing equation 51, we obtain

$$p_c = p\,r_c \frac{\log r_c}{r_c - 1} \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (53)$$

and the work in inch pounds or the section h z w i = $p_c \times \overline{u\,j}$.

For the period i d.—We have just seen that the pressure w i, at the end of compression, would be $\overline{w\,i} = \frac{\overline{z\,h} \times \overline{k\,z}}{\overline{k\,m}}$, and we have already determined the pressure m d in our discussion of the positive work. Now, as i d may be considered a straight line, we can put the work for this part of the stroke as equal to $\frac{\overline{w\,i} + \overline{m\,d}}{2} \times \overline{j\,a}$. As before, we must add the three values together and divide by the stroke in inches in order to obtain the average negative or back pressure, and this subtracted from the average positive pressure leaves the average effective pressure, or, as it is generally termed, the mean effective pressure, written briefly M. E. P.

(If a table of Hyperbolic Logarithms should not be available, a table of common logarithms may be used by multiplying the common logarithm of the number by 2.3026, which will give the hyperbolic logarithm of the same number.)

The above process for determining the M. E. P. is somewhat tedious, but if carefully done, it should give fairly accurate results. The actual steam distribution is dependent upon so many detail points, which affect it greatly, not to mention the condition of the engine, that it is impossible to foretell exactly how much each of the various items will modify the results.

Plate 12 (at end) gives the results of a series of tests made by the author in 1901 on the Chicago & Northwestern Railway, the work comprising dynamometer car readings on the road to supplement a complete set of working tests upon the locomotive testing plant in Chicago.

The various curves designated by per cent cut-off give the ratio of mean effective to boiler pressure for various revolutions per minute. The principal features of the engine tested were as follows:

Cylinders	20 in. by 26 in.
Diameter of drivers	63 in.
Steam pressure	190 lbs.
Boiler diameter	64 in.
Grate area	29 sq. ft.
Heating surface	2,332 sq. ft.
Weight on drivers	118,350 lbs.
Weight of engine and tender	260,000 lbs.
Steam ports	$1\frac{5}{8}$ by 16 in.
Exhaust ports	3 by 16 in.
Valve	Allen-American
Steam lap	$\frac{7}{8}$ in.
Exhaust clearance	None
Valve travel	$5\frac{1}{2}$ in.
Lead at 6-in. cut-off	$\frac{1}{4}$ in.

The curve marked "boiler capacity" shows the limit of speed at which steam should be continuously produced by the boiler and the pressure maintained. The broken line shows the maximum limit of M. E. P. at various speeds in accordance with the report of a committee to the Master Mechanics' Association in 1897. It will be recognized that the steam capacity constitutes a vital question, and this phase will be fully treated under the head of steam capacity; but it is well now to call attention to the fact of this limit and the bearing which it has upon M. E. P. at various speeds. For approximate information the data embodied in plate 12 may be used without great error for other simple engines which may be under consideration, though of course with larger boilers, in proportion to the cylinders, the limiting line would be shifted from the position here shown. In compound locomotives, where the boiler bears a much larger ratio to the high-pressure cylinders, the curves would be considerably different, as will be explained under "Hauling Capacity." The work done by the piston of any engine in making one complete stroke is the product of the total pressure by the length of the stroke, and this total pressure is the product of the M. E. P. by the area of the piston. As in an ordinary simple locomotive, there are four strokes to each revolution (two by each piston), the total indicated horsepower (I. H. P.) of both cylinders will be

$$\text{I. H. P.} = \frac{\text{M. E. P.} \times \text{area} \times \text{stroke} \times 4 \times \text{rev. per min.}}{12 \times 33000}$$

area and stroke both being in inches.

If we let d = diameter of cylinder in inches,
s = stroke of piston in inches,
n = revolutions per minute,
M. E. P. = mean effective pressure in pounds per square inch,

we can write

$$\text{I. H. P.} = \frac{\text{M. E. P.} \times \pi \times d^2 \times s \times 4 \times n}{4 \times 12 \times 33000} = \frac{\text{M. E. P. } d^2 s n}{126050} \quad \text{.................... (54)}$$

If, as before, we let V = speed in miles per hour,
D = diameter of drivers in inches,

$$n = \frac{V \times 5280 \times 12}{\pi \times D \times 60}$$

substituting the value of n in equation 54, we have

$$\text{I. H. P.} = \frac{\text{M. E. P. } d^2 s V}{375 D} \quad \text{............................ (55)}$$

From the information given in plate 12, we can construct a new one, which will show the indicated horsepower for the various combinations of cut-off and speed. This has been done in plate 13, at end of book. As we would expect from the curves of plate 12, the I. H. P. increases regularly with the speed up to about 15 miles per hour; from this point the curves begin to droop, as the steam cannot now get into and out of the cylinder fast enough to maintain the M. E. P. constant. The maximum I. H. P. for any of the higher expansion ratios is found at about 30 miles per hour, beyond which speed the drop in M. E. P. more than offsets the effect due to speed. The line marked maximum continuous I. H. P. shows the limits of the low-expansion ratio curves set by the capacity of the boiler—that is, the speed at which the cylinder uses the steam as fast as the boiler can supply it, and which in the engine tested was about 1,000 indicated horsepower. It is evident that, if the boiler supply were more abundant, this maximum I. H. P. line would be raised above its shown location. How many of the curves would reach it depends upon the valve and gear. It would apparently be no benefit in the case represented for cut-offs of 40 per cent and under, but it would increase the capacity of those above that point, as they are now limited by the capacity of the boiler. There is no doubt, however, that, even if the supply of steam were inexhaustible, the curves of all ratios of expansion would, at some point, reach the maximum power at which steam could be admitted and discharged, and that higher speeds would render a decreased I. H. P. Whether or not this limit would be in the neighborhood of 30 or 40 miles an hour is not known, but the indications are that it would not be far from those speeds. A passenger engine with the same size cylinder, but with piston valves, and having a much larger boiler, gave about 1,500 I. H. P. at 50 miles per hour, the number of revolutions corresponding to 40 miles an hour with the class of engine represented in the plate, the cut-off being a little less than 50 per cent. We do not believe that the 50 per cent cut-off curve in plate 13 would reach 1,500 I. H. P. even with an unlimited supply of steam, which demonstrates the necessity for proper valves and gears in order to obtain liberal horsepowers.

The gradual rise in the maximum I. H. P. line at speeds above 15 miles per hour may be accounted for by the fact that more work is gotten out of a fixed volume of steam at high expansive ratios than at low ones, so that, as the cut-off decreases, we obtain a greater I. H. P. for the steam supply delivered by the boiler.

QUANTITY OF STEAM.

The amount of steam used by the engine during a stroke is of great interest, and can be studied with advantage in connection with its distribution. The larger subject of water consumption will be taken up later, but while we are occupied with indicator diagrams, it will not be amiss to consider this feature.

If, in Fig. 33, we take any point x on the expansion curve e f, we can, from our knowledge of the dimensions of the engine, state the volume or cubic feet of steam back of the piston (including clearance), and from the diagram itself, we know the pressure and can determine from steam tables the weight of a cubic foot of steam, and consequently the weight of steam in the cylinder (and clearance) back of the piston. As the main valve keeps the port closed from e to f, we should naturally expect that the same amount of steam would be found in the cylinder, back of the piston, at any point between e and f. Such, however, is not the case, for the nearer we approach to the point of release f, the greater will be the amount of steam found in the cylinder. In fact, the amount of steam accounted for in this way may be 5, 10 or even more per cent greater at release than at cut-off. This is explained by the condensation of the steam upon its entrance to the cylinder, and its later re-evaporation, when the pressure in the cylinder falls by expansion to a point below that corresponding to the temperature of the cylinder walls. In calculating the volume of steam used in a stroke, it must be remembered that when the valve opens, there is a certain quantity of steam already in the clearance space—that retained and compressed from the previous back stroke. The consumption will therefore be the amount of steam in the cylinder at f less the amount at i, when the valve opens. This quantity is to be determined in precisely the same manner.

The rule for computing the amount of steam by indicator is therefore—multiply the weight of steam at release pressure by the volume of steam back of piston (including clearance) at that point and subtract from the product the weight of steam

VOLUMES OF CYLINDERS IN CUBIC FEET.

Diam. in Inches.	Length in Inches.														
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
10	0.04	0.09	0.13	0.18	0.23	0.27	0.31	0.36	0.40	0.45	0.49	0.54	0.58	0.63	0.67
11	0.05	0.11	0.16	0.22	0.27	0.33	0.38	0.44	0.49	0.55	0.60	0.66	0.71	0.77	0.82
12	0.06	0.13	0.19	0.26	0.32	0.39	0.45	0.52	0.58	0.65	0.71	0.78	0.84	0.91	0.97
13	0.08	0.15	0.23	0.31	0.38	0.46	0.54	0.62	0.69	0.77	0.85	0.92	1.00	1.08	1.15
14	0.09	0.18	0.27	0.36	0.44	0.53	0.62	0.71	0.80	0.89	0.98	1.07	1.16	1.25	1.33
15	0.10	0.20	0.31	0.41	0.51	0.61	0.71	0.82	0.92	1.02	1.12	1.22	1.33	1.43	1.53
16	0.12	0.23	0.35	0.46	0.58	0.70	0.81	0.93	1.04	1.16	1.28	1.39	1.51	1.62	1.74
17	0.13	0.26	0.40	0.53	0.66	0.79	0.92	1.06	1.19	1.32	1.45	1.58	1.72	1.85	1.98
18	0.15	0.29	0.44	0.59	0.73	0.88	1.03	1.18	1.32	1.47	1.62	1.76	1.91	2.06	2.20
19	0.16	0.33	0.49	0.66	0.82	0.98	1.15	1.31	1.48	1.64	1.80	1.97	2.13	2.30	2.46
20	0.18	0.36	0.55	0.73	0.91	1.09	1.27	1.46	1.64	1.82	2.00	2.18	2.37	2.55	2.73
21	0.20	0.40	0.60	0.80	1.00	1.20	1.40	1.60	1.80	2.00	2.20	2.40	2.60	2.80	3.00
22	0.22	0.44	0.66	0.88	1.10	1.32	1.54	1.76	1.98	2.20	2.42	2.64	2.86	3.08	3.30
23	0.24	0.48	0.72	0.96	1.20	1.44	1.68	1.92	2.16	2.40	2.64	2.88	3.12	3.32	3.60
24	0.26	0.52	0.78	1.04	1.30	1.57	1.83	2.09	2.35	2.61	2.87	3.13	3.39	3.65	3.91
25	0.28	0.57	0.85	1.14	1.42	1.70	1.99	2.27	2.56	2.84	3.12	3.41	3.69	3.98	4.26
26	0.31	0.61	0.92	1.23	1.53	1.84	2.15	2.46	2.76	3.07	3.38	3.68	3.99	4.30	4.60
27	0.33	0.66	0.99	1.32	1.65	1.99	2.32	2.65	2.98	3.31	3.64	3.97	4.30	4.63	4.96
28	0.36	0.71	1.07	1.42	1.78	2.14	2.49	2.85	3.20	3.56	3.92	4.27	4.63	4.98	5.34
29	0.38	0.76	1.15	1.53	1.91	2.29	2.67	3.06	3.44	3.82	4.20	4.58	4.97	5.35	5.73
30	0.41	0.82	1.23	1.64	2.04	2.45	2.86	3.27	3.68	4.09	4.50	4.91	5.32	5.73	6.13

VOLUME OF CYLINDERS IN CUBIC FEET.

Diam. in Inches.	Length in Inches.														
	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30
10	0.72	0.76	0.81	0.85	0.90	0.94	0.99	1.03	1.08	1.12	1.17	1.21	1.26	1.30	1.35
11	0.88	0.93	0.99	1.04	1.10	1.15	1.21	1.26	1.32	1.37	1.43	1.48	1.54	1.59	1.65
12	1.04	1.10	1.17	1.23	1.30	1.36	1.43	1.49	1.56	1.62	1.69	1.75	1.82	1.88	1.95
13	1.23	1.31	1.39	1.46	1.54	1.62	1.69	1.77	1.85	1.92	2.00	2.08	2.16	2.23	2.31
14	1.42	1.51	1.60	1.69	1.78	1.87	1.96	2.05	2.13	2.22	2.31	2.40	2.49	2.58	2.67
15	1.63	1.73	1.84	1.94	2.04	2.14	2.24	2.35	2.45	2.55	2.65	2.75	2.86	2.96	3.06
16	1.86	1.97	2.09	2.20	2.32	2.44	2.55	2.67	2.78	2.90	3.02	3.13	3.25	3.36	3.48
17	2.11	2.21	2.38	2.51	2.64	2.77	2.90	3.04	3.17	3.30	3.43	3.56	3.70	3.83	3.96
18	2.35	2.50	2.65	2.79	2.94	3.09	3.23	3.38	3.53	3.67	3.82	3.97	4.12	4.26	4.41
19	2.62	2.79	2.95	3.12	3.28	3.44	3.61	3.77	3.91	4.10	4.26	4.43	4.59	4.76	4.92
20	2.91	3.09	3.28	3.46	3.64	3.82	4.00	4.19	4.37	4.55	4.73	4.91	5.10	5.28	5.46
21	3.20	3.40	3.60	3.80	4.00	4.20	4.40	4.60	4.80	5.00	5.20	5.40	5.60	5.80	6.00
22	3.52	3.71	3.96	4.18	4.40	4.62	4.84	5.06	5.28	5.50	5.72	5.94	6.16	6.38	6.60
23	3.84	4.08	4.32	4.56	4.80	5.04	5.28	5.52	5.76	6.00	6.24	6.48	6.64	6.96	7.20
24	4.18	4.44	4.70	4.96	5.22	5.48	5.74	6.00	6.26	6.52	6.79	7.05	7.31	7.57	7.83
25	4.54	4.83	5.11	5.40	5.68	5.96	6.25	6.53	6.82	7.10	7.38	7.67	7.95	8.24	8.52
26	4.91	5.22	5.53	5.83	6.14	6.45	6.75	7.06	7.57	7.68	7.98	8.29	8.60	8.90	9.21
27	5.30	5.63	5.96	6.29	6.62	6.95	7.28	7.61	7.94	8.27	8.61	8.94	9.27	9.60	9.93
28	5.70	6.05	6.41	6.76	7.12	7.48	7.83	8.19	8.54	8.90	9.26	9.61	9.97	10.32	10.68
29	6.11	6.49	6.88	7.26	7.64	8.02	8.40	8.79	9.17	9.55	9.93	10.31	10.70	11.08	11.46
30	6.54	6.95	7.36	7.77	8.18	8.59	9.00	9.41	9.82	10.22	10.63	11.04	11.45	11.86	12.27

at pre-admission multiplied by the volume of steam ahead of the piston (including clearance).

If we do not have the indicator diagram from which to obtain our pressures, we must determine the cut-off pressure as per plate 10, and use the corresponding weight of steam in connection with the volume at cut-off. The amount of compression would have to be estimated in this case, but as at fairly high speeds and expansive ratios, the compression will generally be sufficient to fill the clearance nearly to initial pressure, we can approximately consider the apparent cut-off volume to represent the quantity of steam used in one stroke.

In order to facilitate these calculations, we insert a table giving the weight of a cubic foot of steam at different pressures above the atmosphere, and also one giving the volume in cubic feet for different lengths of various diameter cylinders. If clearance is to be allowed, it should be added to the apparent length of cut-off or release, and the new length taken.

Weight of a cubic foot of saturated steam in pounds, at pressures above the atmosphere:

Pressure.	Weight.	Pressure.	Weight.
0	0.038	110	0.284
10	0.063	120	0.306
20	0.086	130	0.327
30	0.109	140	0.348
40	0.131	150	0.370
50	0.154	160	0.390
60	0.176	170	0.411
70	0.198	180	0.432
80	0.219	190	0.453
90	0.241	200	0.474
100	0.263		

As an example, let us consider an engine with 19-inch cylinders and 24-inch stroke, with 155 pounds boiler pressure, and cutting off at half (apparent) stroke when running 30 miles per hour, the drivers being 60 inches in diameter. From the table previously given, we find that this speed corresponds to 168 revolutions per minute. As this engine had Allen valves, we may take the upper limit in plate 10, and interpolating between the 150 and 200 revolution lines, find that the cut-off pressure will be about 80 per cent of the initial pressure, which in turn will be 89 per cent of the boiler pressure, if the throttle be maintained wide open. Therefore, the cut-off pressure $= 155 \times .89 \times .80 = 110$ pounds, and the weight per cubic foot $= 0.284$ pounds. A space 19 inches diameter and 12 inches long has a volume of 1.97, or, say, 2 cubic feet. Clearance has not here been allowed, as it is assumed that the compression of the previous stroke has filled this space. Therefore, we obtain $.284 \times 2 = .568$ pounds of steam per stroke, accounted for by indicator. It will be shown that this does not by any means represent the actual steam consumption.

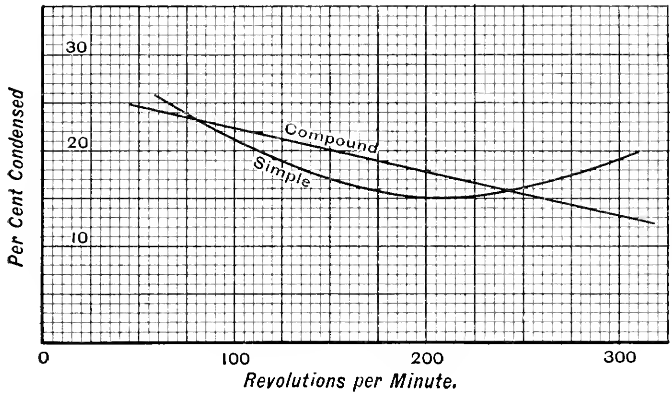
Reference has previously been made to cylinder condensation, which is caused by the steam entering the cylinder whose walls are cooler. There is also a percentage of water entrained. This latter should be small if the dry pipes are properly designed and located, but is probably never zero, and, if the boiler be foaming, the amount of water carried over will be very great. This water is not accounted for by the indicator. It varies within very wide limits, and must always be added to the amount determined by the indicator. It can be reduced by steam jackets, superheating and similar methods, but is probably never obliterated. The cooler the cylinder, and the longer time the steam has to remain in the cylinder, the greater will be the condensation. From this we conclude that the proportion of steam condensed will be greater at early cut-off, as the mean pressure and temperature of the steam in the cylinder will be lower, thus retaining the cylinder itself at a lower temperature. We should also expect to find less condensation at high speeds, unless the speed also wiredraws the entering steam and keeps the cylinder temperature abnormally low.

The principal advantage of the compound engine lies in the fact that there is less variation in the pressure, and therefore in the temperature of the steam and the cylinder in which it is used; in other words, there is much less difference between the admission and exhaust temperatures in any one cylinder, and therefore the condensation is proportionately reduced.

As would naturally be supposed, the laws affecting cylinder condensation are complex, when considered as affected by the various combinations of expansion, speed, temperature of air, jacketing, cylinder design, etc., and our information on the subject is anything but complete. It has been customary formerly to consider it dependent entirely upon the expansive ratio, but this hypothesis would not give a diminished condensation for increased speed. In the absence of more com-

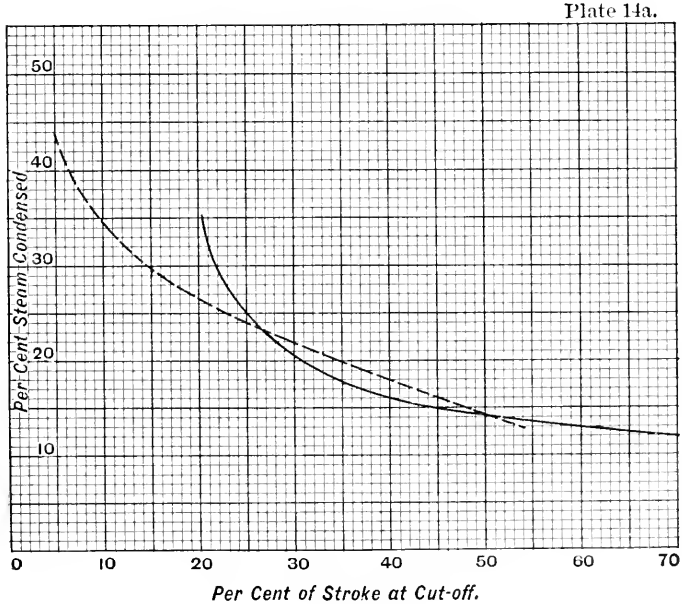
CONDENSATION IN CYLINDER.

Plate 14.



CONDENSATION IN CYLINDER.

Plate 14a.



plete and detailed information, plates 14 and 14a are introduced. Plate 14a gives a graphical representation of the percentage of steam passed through the cylinder, which is condensed, and therefore not accounted for by indicator. The full line was derived from tests made at Purdue University by Professor Goss, upon a simple locomotive. The broken line has been traced from information given in Barrus' pocket book on the Tabor indicator. Both of these loci show the effect of the ratio of expansion upon condensation.

Plate 14 shows the effect of changing speed with a constant rate of expansion. The curve marked "simple" was compiled from a simple locomotive at one-third cut-off, while the one marked "compound" was prepared from tests of a Vauclain compound with 61 per cent cut-off. The release point on the diagram was used to determine the steam "accounted for;" in the case of the compound, the release of the low-pressure cylinder was used.

These several curves demonstrate the fact that speed, as well as expansion, has an important bearing on cylinder condensation, and the plate 14 indicates a minimum limit at about 200 revolutions per minute, for the simple engine. The compound curve does not appear to have been extended sufficiently to discover a minimum limit, but it no doubt exists at some point. In the Purdue simple engine, Professor Goss found that the lowest consumption of steam per indicated horsepower hour was at a little less than 200 revolutions per minute, and this he termed the "critical speed." This seems also to be the speed of lowest condensation, but we have very few tests by which to demonstrate the universality of this important feature.

We must not consider these curves as denoting the actual proportions of condensation under all circumstances, as there will be wide variations from that shown under different conditions of design and operation. We must, however, always allow liberally, in addition to the amount shown by the indicator. Thus, in the last example, where we found .568 pound of steam per stroke, we should add a correction of at least 17 per cent, making the expected consumption .665 pound per stroke, as the diagram shows 14 per cent condensed at one-half cut-off and $14 \div (100 - 14) = 17$ per cent of the amount of steam shown by the indicator. In fact, as there is so much uncertainty about this whole subject of cylinder condensation, it would be safer, in making estimates for steam capacity needed in boilers, etc., to add 25 per cent to the amount accounted for by indicator. When it is possible to make a regular test, by measuring the water actually fed into the boiler, or by condensing the steam exhausted from the cylinders, it is in all cases preferable to using that deduced from indicator cards, either real or hypothetical, as the test method takes care of all the water sent to the cylinders, whether condensed or maintained in the form of steam. Even in some tests of compound locomotives, the results showed 22 per cent of water used unaccounted for by the indicator, but in a road test, there is more or less leakage, which, of course, always increases this quantity.

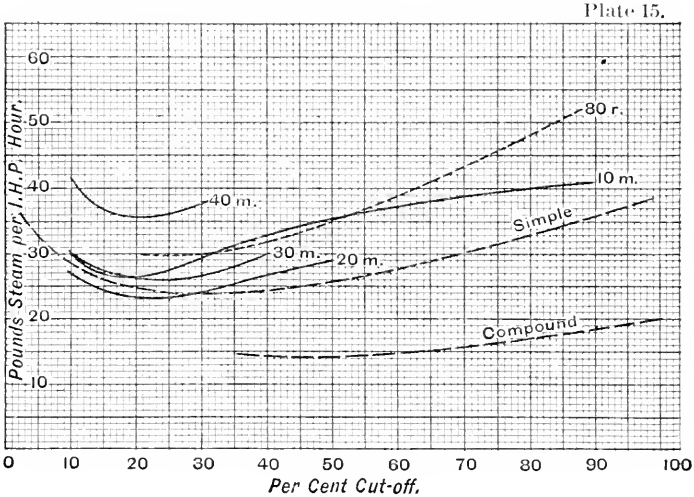
By combining the steam consumption and the power generated, we obtain a very useful quantity for comparing the commercial value of the work performed by different locomotives, that is, the weight of steam used per indicated horsepower hour. We can estimate this from a diagram either actual or hypothetical, as in the last example, by multiplying the quantity of steam used per stroke by the number of strokes that would be made in one hour at the speed assumed, and dividing this by the indicated horsepower generated. Allowing 17 per cent for condensation, we would have $.568 \times 1.17 = .665$ pound per stroke. As the wheels would make 168 revolutions a minute at the speed assumed, we should have $.665 \times 4 \times 168 \times 60 = 26{,}813$ pounds steam per hour. From plate 12, using the M. M. Assn. line, we should have, at 168 revolutions per minute, 44 per cent of the boiler pressure, or $155 \times .44 = 68$ pounds M. E. P. Now, from formula 54, we can state the I. H. P.

$$= \frac{68 \times 361 \times 24 \times 168}{126{,}050} = 652 \text{ I. H. P.}$$

and $\dfrac{26{,}813}{652} = 41$ pounds of steam per indicated horsepower hour.

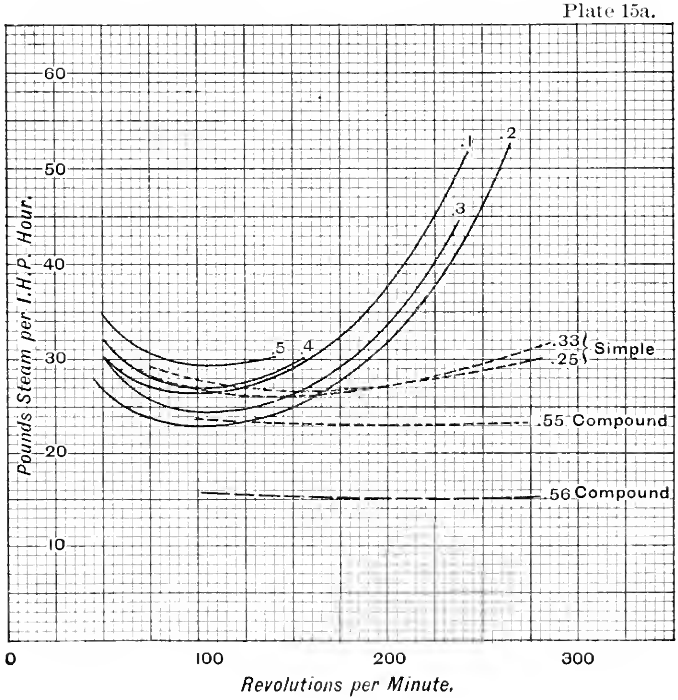
STEAM CONSUMPTION PER INDICATED HORSE POWER HOUR.

Plate 15.



STEAM CONSUMPTION PER INDICATED HORSE POWER HOUR.

Plate 15a.



Plates 15 and 15a show the amount of steam used per I. H. P. hour as determined by several different locomotive tests. The full lines are taken from the same tests which were used in constructing plates 12 and 13. Plate 15 shows the effect of varying expansions at constant speed, and the plate 15a the result of changing speed at constant cut-off. The designations at the curves 40 m, etc., give the speed in miles per hour which was constant for that locus; in plate 15a, the point of apparent cut-off is shown for each curve. The broken lines designated as "simple" and "compound" are from tests made by the Baldwin Locomotive Works, and appeared in No. 11 of their "Record of Recent Construction." The dotted lines are from tests made at Purdue University, upon their testing plant. The line marked "80 r" was from their simple locomotive, at 80 revolutions per minute. The curves in 15a were from simple and compound engines, the cut-off being designated in each case.

Two very interesting facts are brought out by these plates; the first, that there is a minimum steam consumption for a certain rate of expansion, and the second, that there is a minimum consumption for a certain speed of revolution.

Taking up the first proposition, an examination of the loci from simple engines, shows that the most economical cut-off, as far as steam consumption per indicated power is concerned, is about one-quarter of the stroke—all of these curves reaching their lowest point between 20 and 30 per cent cut-off. The compound engine, on the contrary, has its minimum at about half stroke, and the curve is much nearer to a straight line. This is no doubt due to the fact that, with the proportion between low and high pressure cylinders which ordinarily exists in compound locomotives, the 50 per cent cut-off is really about 20 or 25 per cent cut-off, considering the complete expansion of the steam, from the cut-off in high-pressure cylinder to the release in the low-pressure cylinder. The reduction of steam consumption by the compound as compared with the simple engines, is quite clearly shown. As demonstrated by these curves, the results are likely to be quite different for engines of various design, but it is clear that, if a simple engine be so designed that it can do the major portion of its work at one-quarter cut-off, it will be operating at the maximum steam efficiency. Whether it will be working at the maximum financial efficiency is a different problem.

The second proposition concerns the speed of maximum steam efficiency. This will probably vary with different types of engines. Thus we see, for the C. & N. W. engine which was tested, that the lowest consumption at all cut-offs was at about 100 revolutions per minute. The simple engine at Purdue has its minimum near 175 revolutions per minute. In the compound engines it is not well defined. It must be remembered that these were Vauclain compounds, and it cannot be stated that the same curves would apply to the 2-cylinder type. The general indication from this set of curves is, that efficient steam practice requires driving wheels of as large diameter as can be conveniently used for the work intended.

ROTATIVE FORCE.

The energy of the steam pressure on the piston is expended in causing the rotation of the driving wheels, by which the engine is propelled. While the greatest pressure of steam that comes upon the piston is, as we have seen, at the commencement of the stroke, yet in this position it cannot exert any turning influence upon the wheel, as the connecting rod and the crank are then in a direct line, and only a heavy thrust against the bearing results. As the wheel turns, and the line of thrust (or pull) of the rod passes above or below the center of the axle, a moment is produced, and the pressure along the rod helps to turn the crank. The greatest leverage is found near the middle of the stroke, but at early cut-off the steam pressure is here reduced, and this reduction increases to the end of the stroke; but in the last half of the stroke, the lever arm of the rod thrust is being continually reduced, which causes a double reduction in the rotative moment. In addition to this, the piston on the other side of the engine repeats the operation 90 degrees later (or earlier) and the total rotative force at any instant is the sum of the instantaneous rotative forces on each side. The more uniform in amount that we can maintain this total rotative force, the more uniform will be the action of the locomotive, and the less likelihood of undue slipping. As many important questions depend upon a clear understanding of the action of the rotative force for their solution, we shall examine this problem with considerable attention to detail.

It is first of all necessary to know what will be the piston pressure at any and all points of the stroke, and if we have an indicator card from the engine which we wish to examine, the information will be obtained by measuring the height of the steam line from the atmospheric line, and subtracting the height of the back-pressure line, at each and every point, the back-pressure line being taken from the opposite end of cylinder, as it acts against the piston. This, in effect, gives us the distance between the steam line of one end of cylinder and the back pressure or exhaust line of the other end.

In Fig. 44 we have the indicator cards from opposite ends of cylinder superimposed; if, now, we measure the distance ver-

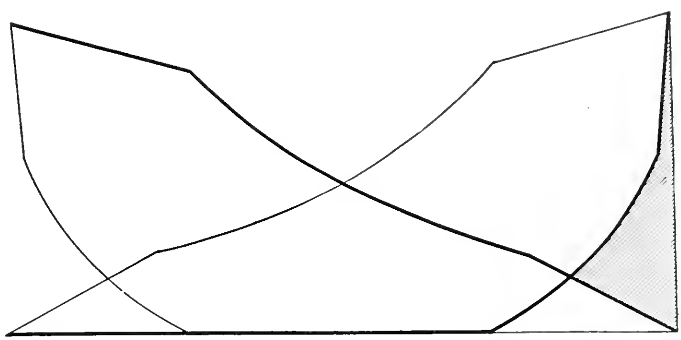


Fig. 44.

tically between the heavy lines, we shall have the effective or instantanous piston pressure at that point of the stroke, the distance measured being taken, of course, in connection with the scale of the indicator spring. At the shaded portion, the pressure will be against the piston, and will operate in the opposite direction to its motion.

It is frequently necessary, however, to study the rotative effect when we are without the actual indicator cards, and we must therefore prepare one from the best information at hand.

If we have actual cards from a similar type of engine taken under parallel conditions, we may use these, but if not we can proceed rapidly as follows: From the table giving the ratio of initial pressure to boiler pressure, we can locate the admission point above the atmospheric line, and from plate 10, the cut-off pressure, for the speed and rate of expansion being considered, and, as before explained, these may be connected by a straight line. This should be done on tracing or thin paper, and the scales of stroke and pressure should be the same as those on plate 16, at back of book. Now lay the sheet upon which we are working upon plate 16, being sure that the atmospheric line and the starting or initial point correspond with those of the plate. The expansion line can then be traced from the plate, commencing at the cut-off point previously determined, and continuing till we reach the portion of stroke at which release occurs, when a straight line may be drawn to the back-pressure point, located on the terminal line of the stroke. For the back-pressure line, draw one parallel to the atmospheric line, at the assumed back pressure, and continue it until compression occurs, when one of the hyperbolas at lower right side of diagram will act as a guide to trace our curve to the point of pre-admission or lead opening, when a straight line should connect to the initial point. The points for release, compression, etc., can be assumed or taken from a Zeuner valve motion diagram, as previously explained, and with plate 16 we can, in a few minutes, be in possession of a full set of indicator cards which will represent service conditions quite fairly. If sufficient care be used, the pressure at the different points of the stroke can be measured on plate 16 without the construction of the hypothetical card just described, but it will be found somewhat confusing on account of the many lines of the plate, and the method proposed is considered to be well worth the small amount of extra labor. The various pressures are, of course, to be multiplied by the area of the piston in order to obtain the full pressure upon the cross-head.

In order to determine the rotative force, we must know the amount of force that acts tangentially upon the crankpin, as that is the force which induces rotation. This tangential force multiplied by the crank radius will give the rotative moment; thus, if the stroke is 2 feet, the crank radius will be 1 foot, and the tangential forces will then also represent the rotative moment in foot-pounds; if the stroke be 30 inches, the tangential forces must be multiplied by $1\frac{1}{4}$ $\left(\frac{30}{2 \times 12}\right)$ to obtain the moment in foot-pounds.

In Fig. 45 let P be the total effective pressure on the piston at any point of the stroke, or the force acting along the piston rod.

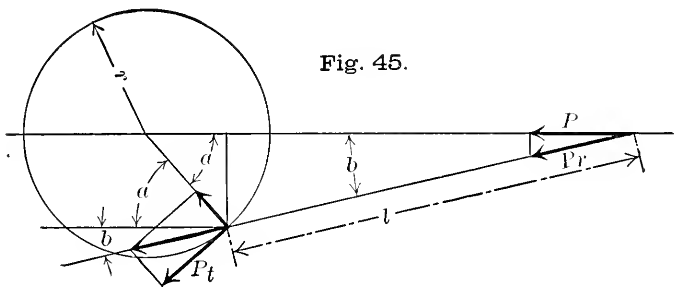


Fig. 45.

Then, neglecting the friction of the cross-head, the force acting along the axis of the connecting rod will be

$$P_r = \frac{P}{\cos b} \qquad (56)$$

At the crankpin this force P_r resolves itself into two components, one acting through the axis of the crank, toward (or from) the center of axle, and the other at right angles to the crank, or tangentially. This tangential force will be represented by P_t, and if "a" is the crank angle from the dead point, and "b" the angle of the connecting rod, we have

$$P_t = P_r \sin (a + b) = P \frac{\sin (a + b)}{\cos b} \qquad (57)$$

Expanding equation 57, we obtain

$$P_t = P \frac{\sin a \cos b + \sin b \cos a}{\cos b}$$

$$= P \left(\frac{\sin a \cos b}{\cos b} + \frac{\sin b \cos a}{\cos b} \right)$$

Now if r = radius of crank,
l = length of connecting rod,
both in the same units, we have, from Fig. 45,

$$r \sin a = l \sin b, \text{ and}$$

$$\sin b = \frac{r}{l} \sin a.$$

Also we can write

$$\cos b = \sqrt{1 - \sin^2 b}$$

but

$$\sin^2 b = \frac{r^2}{l^2} \sin^2 a$$

and

$$\cos b = \sqrt{1 - \frac{r^2}{l^2} \sin^2 a}$$

therefore, by substituting these values, we obtain

$$P_t = P \left\{ \sin a + \frac{r}{l} \frac{\sin a \cos a}{\sqrt{1 - \frac{r^2}{l^2} \sin^2 a}} \right\} . (58)$$

all angles being in terms of the crank angle.

But as $\frac{r^2}{l^2} \sin^2 a$ is generally very small, we can neglect this term and write equation 58 simply

$$P_t = P \left\{ \sin a + \frac{r}{l} \sin a \cos a \right\} \ldots\ldots\ldots (59)$$

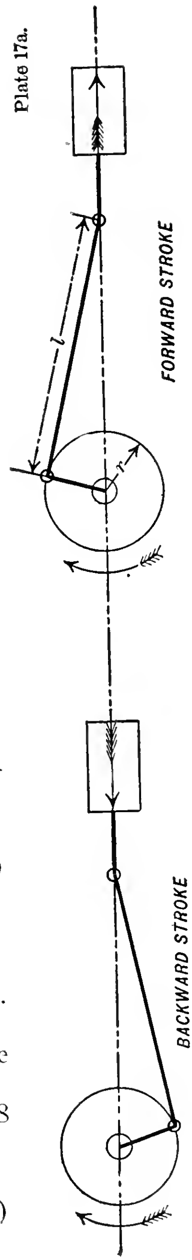
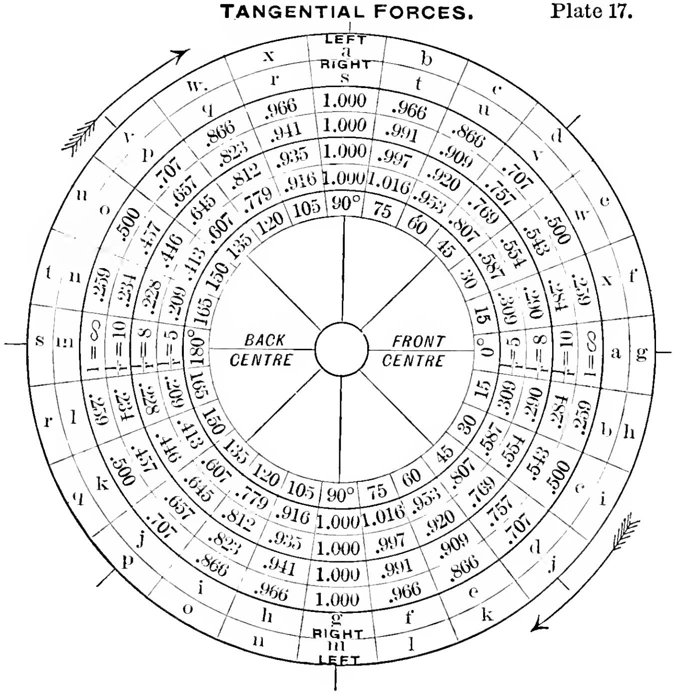


Plate 17a.

At the two dead points, where a = 0° and 180° we have sin a = 0 and there is no tangential or rotative force. Plate 17 gives the value of $(\sin a + \frac{r}{l} \sin a \cos a)$ for each 15 degrees of crank motion, and for ratios of main rod length to crank

TANGENTIAL FORCES. Plate 17.



radius of 5, 8 and 10 and for a connecting rod of infinite length. These values multiplied by P, the force acting at the instant upon the piston, give us the tangential force. As the position of the piston in its stroke is different for various ratios of $\frac{l}{r}$, even at the same crank angle a, there is shown, at the bottom of plate 16, the positions of the piston for the several crank angles and ratios of $\frac{1}{r}$. The upper marks of each set are for the backward stroke (as designated on plate 17a), and the lower marks are for the forward stroke; these are indicated by B. S. and F. S. The angles marked identify the piston positions with the crank angles. With a rod of infinite length, there is, of course, no difference between the forward and backward strokes. The points should be laid off on the thin paper diagram, and the vertical lines drawn through these on the card give the points of measurement for the piston forces corresponding to the respective crank angles.

The outer circles on plate 17 containing the letters are for ease of reference in combining the rotative force of both sides of the engine for the total force. It will be noticed that the same letters in both circles are 90 degrees apart, therefore, if we obtain the tangential force, say, for 30 degrees backward stroke or "c" on right pin, we should add to it the tangential force for "c" or 60 degrees forward stroke on the left pin, in order to obtain the total tangential or rotative force. This assumes that the right crank leads. If the left crank is ahead, we can take the opposite letters and add the values: thus, for "c" on right pin, the letter opposite being "i," we simply add to the rotative or tangential force at "c," the similar force at "i." This will be illustrated by example. Let us consider the New York Central 4—4—2 type passenger engine, with 21-inch by 26-inch cylinders, piston valves, drivers 79 inches in diameter, and carrying 200 pounds steam pressure. We will first study the rotative forces when running at 40 miles an hour, cutting off at 37 per cent of the stroke, and with a connecting rod about 10 times the crank radius. The valve motion will be represented by valve circle number 4 on plate 8, from which we learn that the elements will be about as follows:

Cut-off at 37 per cent of stroke passed.
Release at 72 per cent of stroke passed.
Compression at 24 per cent of stroke to go.
Admission at 2 per cent of stroke to go.

As the speed is to be 40 miles an hour, the revolutions per minute will be 170, and from the table we find the initial pressure will be 200 $\times$ .89 = 178 pounds By plate 10, the cut-off pressure should be 178 $\times$ .77 = 137 pounds above the atmosphere.

We can now start the construction of our hypothetical diagram, as shown on plate 18 at back of book. Lay out the atmospheric line and the vertical terminals of the diagram, and then with plate 16 for a scale locate the initial and cut-off points d and e, e being, of course, at the intersection of .37 stroke and the 137-pound steam line, and connect with a straight line. It will be found that the point e does not fall directly over a line on plate 16, but so closely to one that a curve of expansion can readily be traced to f, which will be upon the line representing .72 of the stroke. Assume that the back pressure will be 13 pounds and connect the point f by a straight line to another point 13 pounds above atmosphere on the limit line. This finishes our steam line. For the back pressure, draw a line parallel to the atmospheric line at a height of 13 pounds, and from the left hand side unto .76 of the stroke, this leaving 24 per cent to be completed. Trace from the compression curve coinciding with the point h unto within .02 of the end of stroke, and connect this point i and 178 pounds on the limit line by a straight line; this completes our diagram.

As the rod is 10 times the crank radius, trace from plate 16 the piston positions corresponding to the several crank angles for $\frac{l}{r} = 10$, and draw verticals from these points through the diagram as shown. The distance between the steam and back-pressure lines, to the scale of pressures, measured on these lines, will be the instantaneous effective pressures upon a unit of piston surface, and multiplied by the piston area = 346 square inches, will give the total piston pressures. It is convenient to tabulate the data as here shown. The letters refer to the crank angles of the right side, as shown on plate 17, and the angles are measured from the front center. The pressures per square inch are those measured from our diagram just prepared, and the total pressures are these values multiplied by 346. The tangential factor is taken from plate 17, and corresponds to the crank angles in the circles marked $\frac{l}{r} = 10$, and the tangential force is the product of the two values immediately above.

We are now prepared to lay out a diagram of rotative or tangential forces. Referring to plate 19 (see end of book) (the upper figure), we lay off our base line, divided into 24 equal spaces corresponding to the various crank angles, and, commencing at the front center, perpendiculars on these points are marked 30°, 60°, etc., to correspond to the crank positions. On the perpendiculars, lay off, to a convenient scale, the values obtained as tangential forces. This line is drawn solid in the plate and is marked "steam force." It will be noticed that from 140° to 180° on the backward stroke, and from 35° to 0° on the forward stroke, that it passes below the base line. This is occasioned by the negative pressures in columns k, l, w and x, brought about by the compression line exceeding the steam line at those crank angles, and means that the piston absorbs work from the momentum of the engine, instead of performing it. The table shows the difference in force due to the angularity of the connecting rod, as otherwise the piston positions and the tangential factors for the same columns in the upper and lower portions of the table would be the same, that is, with a connecting rod infinitely long, the upper values representing the backward stroke, and the lower the forward stroke.

We have seen under the heading of inertia that the reciprocating weights absorb power the first part of the stroke and emit it the last part. This will have evidently an effect upon the tangential force brought upon the crankpin. In order to determine the amount of this inertia, let us suppose the piston and rod and crosshead to weigh 580 pounds, and the main rod 600 pounds, which cannot be far from right. For the speed equal to the drivers = 79 inches we have from plate 5, for

$\frac{r}{l} = \frac{1}{10}$, the coefficient 1.76 for front end and 1.44 for back end of stroke, therefore from formula 19 we have for the inertia force at end of stroke

$$580 \times 26 \times 1.76 = 26{,}500 \text{ pounds at front end.}$$
$$580 \times 26 \times 1.44 = 21{,}700 \text{ pounds at back end.}$$

for the reciprocating parts.

The connecting rod has its center of gravity about .4 the length from the crank end, therefore from plate 5 we have

$$600 \times 26 \times 1.66 = 25{,}900 \text{ pounds at front end.}$$
$$600 \times 26 \times 1.54 = 24{,}000 \text{ pounds at back end.}$$

This gives us a total of 52,400 pounds at front end and 45,700 pounds at back end, at 79 miles an hour. At 40 miles, the effect would be about $\frac{1}{4} = (\frac{1}{2}^2)$ as much, or 13,100 pounds front end and 11,425 pounds at back end.

We found in connection with our study of plate 6 that we could represent the action between the end and center of stroke by a right line connecting the force at end to scale with zero at the center, so we draw an inertia base line at any point on our new diagram (see plate 18), and to a determined scale, lay off the values found, selecting the left side as front end and the right side as back end, as at k and l. The force of inertia at any of the specified crank angles will therefore be scaled off on the verticals through the crosshead positions corresponding to those angles, from the selected base line. In this case we must use the upper scale of crosshead positions, not only for the backward stroke, but also for the forward one; otherwise we should have two values for each end of the stroke, which would not be proper. We should also recollect that whether above or below the base line shall be considered negative depends upon the direction in which the crosshead is moving—always negative during the first half of stroke and positive during the last half.

With these explanations, we can now fill in the line "Inertia" in the table, and it will be seen that the 90-degree angle of crank is positive in the backward stroke, in which the crosshead has passed the center of stroke and negative in the forward stroke, where it has not yet reached the center. The values multiplied by the tangential factor give us the tangential inertia force. (The multiplications were made by slide rule, and are not accurate to the last figures.) This force is now laid off upon our rotative force diagram as shown in plate 19, marked "inertia." By performing the algebraic addition of the inertia and the steam force curves, we obtain a new one, designated as effective force. It should be noticed that these forces help to rectify each other, for where either one is negative, the other is always positive, and the effective force line does not show as great variations between the maxima and minima as the steam line.

By duplicating this effective force right side curve 90 degrees later, we produce the effective force acting upon the left side (assuming here that it follows the right side), and by taking the algebraic sum of these two curves, we can construct a total rotative force curve, designated as "both sides."* This curve partakes of the characteristics of both of the component curves, but its maxima and minima points fall between the others. We find in the total curve that the maximum points fall about 30 degrees behind each of the four dead centers, and the minimum points correspond to the dead centers; that is, every time a dead center is passed on either side, the rotative force is at a minimum. No two of the maximum points have the same value, although the minimum points are paired in value. The greatest rotative force is found at 60 degrees on the forward stroke, and the smallest when the crankpins pass the front center. The angularity of the connecting rod introduces irregularities, not only in the steam force, but also in the forces of inertia, which must be examined in detail for a full understanding of them.

In the middle diagram of plate 19, the total rotative force is drawn without the individual force curves, the average rotative force for the whole revolution is also shown. This was obtained by planimetering the area between the curve and

---

*The counterbalance and driving wheel itself will help to counteract the effect of the reciprocating forces of acceleration and retardation, due to its "fly-wheel action," but the rotative force as imposed upon the crank pin by the connecting rod will evidently be as determined above.

ROTATIVE FORCES AT STARTING.

Letters	b	c	d	e	f	g	h	i	j	k	l
Angles	15	30	45	60	75	90	105	120	135	150	165
Pressure, square inch	196	196	196	196	196	196	196	196	196	180	80
Total	67800	67800	67800	67800	67800	67800	67800	67800	67800	64300	27700
Tang. Factor	.284	.513	.757	.909	.991	1.000	.911	.823	.657	.457	.234
Tang. Force	19200	36800	51300	61700	67300	67800	63800	55800	44500	30000	6480
Letters	n	o	p	q	r	s	t	u	v	w	x
Angles	165	150	135	120	105	90	75	60	45	30	15
Pressure, square inch	196	196	196	196	196	196	196	196	196	188	80
Total	67800	67800	67800	67800	67800	67800	67800	67800	67800	65000	27700
Tang. Factor	.234	.457	.657	.823	.941	1.000	.991	.909	.757	.543	.284
Tang. Force	15800	31000	44500	55800	63800	67800	67300	61700	51300	35300	7860

ROTATIVE FORCES AT 40 MILES AN HOUR.

Letters	a	b	c	d	e	f	g	h	i	j	k	l
Angles	0	15	30	45	60	75	90	105	120	135	150	165
Pressure, square inch	165	164	157	147	135	115	88	66	50	14	—17	—54
Total pressure	57100	56700	54300	50900	46600	39800	30500	22800	17300	4850	—5880	—18700
Tang. Factor	0	.284	.543	.757	.909	.991	1.000	.941	.823	.657	.457	.234
Tang. Force	0	16100	29500	38500	42400	39600	30500	21400	11200	3180	—2680	4380
Inertia	—13100	—12900	—11000	—9500	—6500	—3000	500	3000	6000	8000	10000	11000
Tang. Iner.	0	—3670	—5980	—7180	—5900	2970	500	2820	4940	5250	4570	2580
Letters	m	n	o	p	q	r	s	t	u	v	w	x
Angles	180	165	150	135	120	105	90	75	60	45	30	15
Pressure, square inch	165	164	158	151	138	126	96	72	57	22	13	54
Total pressure	57100	56700	54600	52200	47700	43600	33200	24900	19700	7610	4500	18700
Tang. Factor	0	.234	.457	.657	.823	.941	1.000	.991	.909	.757	.543	.284
Tang. Force	0	13200	24900	34200	39200	41000	33200	24600	17900	5760	2440	5310
Inertia	—11100	11000	—10000	8000	—6000	3000	—500	3000	6500	9500	11000	12900
Tang. Iner	0	2580	4570	5250	4940	2820	—500	2970	5900	7180	5980	3670

ROTATIVE FORCES AT 80 MILES AN HOUR.

Letters	b	c	d	e	f	g	h	i	j	k	l
Angles	15	30	45	60	75	90	,105	120	135	150	165
Pressure, square inch	116	127	101	64	28	23	16	2	—21	—72	—134
Total	50500	43900	35000	22100	13100	7960	5530	692	—7270	—24900	—46400
Inertia	—51000	—45000	—36000	—24000	—11000	3000	14000	24000	33000	40000	44000
Effective	—500	—1100	—1000	—1900	2100	10960	19530	24692	25730	15100	—2400
Tang. Factor	.284	.543	.757	.909	.991	1.000	.941	.823	.657	.457	.234
Tang. Force	—142	—600	—757	—1720	2080	10960	18300	20300	16850	6900	—562

Letters	n	o	p	q	r	s	t	u	v	w	x
Angles	165	150	135	120	105	90	75	60	45	30	15
Pressure, square inch	116	134	109	74	47	29	18	8	—16	—62	—134
Total	50500	46300	37700	25600	16200	10010	6230	2776	—5540	—21400	—46300
Inertia	—44000	—40000	—33000	—24000	—14000	—3000	11000	24000	36000	45000	51000
Effective	6500	6300	4700	1600	2200	7010	17230	26770	30460	23600	4700
Tang. Factor	.234	.457	.657	.823	.911	1.000	.991	.909	.757	.543	.284
Tang. Force	1520	2870	3090	1310	2070	7010	17050	26100	23000	12800	1330

the base line, and dividing by the length of the diagram. Thus we see that the average rotative force is about 38,000 pounds, while the maximum runs up to 46,000 pounds, and the minimum drops to 30,000 pounds, and that the variation in force will amount to 21 per cent above and below the average, at this speed and cut-off.

In order to study the same force at 80 miles an hour we prepare a table as shown herewith. This is arranged somewhat differently from the 40-mile table, so as to save constructing so many curves and enable us to lay out the total rotative forces for both sides together, without the intermediate processes resorted to in the 40-mile curve. Immediately below the line of total piston pressures, we enter the amount of inertia, which is scaled from the lines m, o, n, laid off on plate 18 from 52,400 and 45,700 pounds at m and n, respectively, as found by calculation to represent the horizontal inertia forces at ends of stroke at 79 miles an hour. The algebraic sum of these inertia and piston pressures is entered in the line marked "effective," and these values are in turn multiplied by the tangential factor to produce the last line, or tangential force.

Examination of the table demonstrates the importance of the reciprocating weights. From 0 to 60 degrees from the forward center, the inertia is actually greater than the total steam pressure upon the piston, or, in other words, the steam is not powerful enough to move the reciprocating parts at the necessary velocity, and they are actually carried to this point by dragging upon the crankpin. At 75 degrees, however, the acceleration has diminished so that the steam pushes the piston, and continues to do so until at 135 degrees the compression acts to retard the backward motion. Now, however, the inertia of the parts is rapidly increasing,. and pushes them backward despite the back pressure of compression, and continues to perform useful work, until within about 15 degrees of the end of stroke, when compression and preadmission raise the back pressure so high that it is actually greater than the force of inertia, and so acts as a cushion taking up the slack of the connections and preventing their pounding. The forward stroke is similar in result, except that, as the effect of inertia is greater at the front end of stroke than the back end, there is a forward pressure exerted upon the pin during the first portion of this stroke, and at the end or front center, the inertia forces completely overbalance the compression. Thus, while we found several crank angles in the backward stroke where the rod was dragging back upon the crankpin, namely, at b, c, d, e and l, or absorbing work, in the forward stroke the rod always pulls the pin ahead, as all the "effective" values are positive.

Let us suppose, in this case, that the left crank leads, and combine the tangential forces to enable us to lay out our rotative forces as in the middle diagram of plate 19, marked 80 miles per hour. As a and m of the right side, corresponding to 0 and 180 degrees, are always zero, the only rotative force at that angle will be produced by the left side of the engine. Now, if the left side be ahead, the left crank will be at 90 degrees, corresponding to g on the right side circle, and so we find g opposite to a at the front center on the plate. Then all that is necessary is to add together the values under a and g in our table. But a = 0, so g = 10,960 must be set down on the ordinate passing through 0 degrees.

In plate 17, we find h opposite b, so, adding — 142 and 18,300, we obtain 18,158 as our total force to lay off on 15-degree line. So, for 30 degrees, add c and i = — 600 + 20,300 = 19,700, and continuing, add the values under the columns headed by the letters which come opposite in plate 17.

(If the right side leads, simply take the values corresponding to the angles located by the same letters; thus, add a (in right circle) and a (in left circle): that is, values under 0 and 90 degrees (on the forward stroke), or 0 and 7,010 for the first point; h and b or 15 (B) and 75 (F) = — 142 + 17,050 for the second point, and so on around the circle.)

By this process we can construct the 80-mile-an-hour curve, and with the planimeter as before, determine the average rotative force. Here we find it to be 13,300 pounds; the maximum 29,000 pounds and the minimum only 1,500 pounds, or 118 per cent above and 89 per cent below the average. The greatest force is again exerted when the leading pin is 60 degrees from the front center and is approaching that center on its forward stroke. This was also the case at 40 miles an hour, where we considered the right pin as leading.

We should also develop the rotative force curve for starting. The opposite page gives these values. As the speed is slow, we do not have to consider the effects of inertia. We notice that there are no negative values, and the resultant curve is more uniform in value than any of the others. It is shown on plate 19, the bottom diagram, the scale being only one-half as large as the former curves. The average rotative force is 84,000 pounds, the maximum 102,000 and the minimum 66,000 pounds, or 21 per cent above and below the average. Here the maximum force occurs 45 degrees after the leading crank has passed the front center or when the two cranks are both 45 degrees from the front center.

It will be of interest to compare the locations of the maxima and minima of the three curves, and their relation to the leading pin. We find them to be as follows:

	Maxima				Minima			
	Backward		Forward		Backward		Forward	
Starting	45°	135°	135°	45°	90°	180°	95°	5°
40 miles	40°	130°	145°	60°	90°	180°	90°	0°
80 miles	35°	120°	150°	60°	75°	165°	105°	10°

The maximum points are thus seen to fall when the leading crank is at or approaching the 45-degree points, and the minimum points at or approaching a dead center, but not more than 15 degrees from these points. If we wish to know the maximum rotative force only, then it is not necessary to calculate all the points as we have done, but only those immediately in the neighborhood of the 45-degree points, viz., 30°, 45°, 120°, 135°, backward stroke, and 150°, 135°, 60°, 45°, forward stroke.

This study can, with advantage, be extended to compound engines of various types, and with various kinds of balancing. Fig. 46 shows the rotative force due to steam alone, that is, not considering the effects of inertia, for a simple engine and a Baldwin balanced and ordinary compound, the compound engines having cylinders 15 and 25 by 26 inches, and the simple engine having 20 by 26 inch cylinders, or of approximately equal power, and all when the wheels are revolving at 336

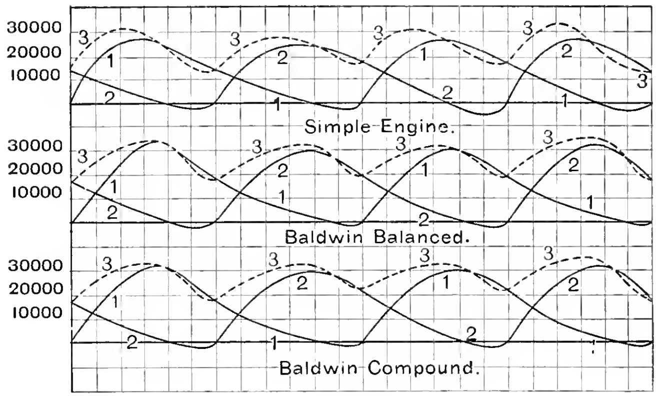


Fig. 46.

revolutions per minute. Line number 1 represents the right pin; line 2, the left pin, and line 3, both sides.

STRAINS INDUCED.

In accomplishing the rotation of the crank and axle, the steam induces certain strains in the various members of the mechanism, and we are now in a position to investigate these strains. Not only the moving parts are subject to these strains, but also the quiescent parts, such as cylinders, cylinder heads, frames, etc., as they naturally react against the forces producing motion. When steam enters the cylinder and pushes the piston backwards, there is a similar pressure upon the front cylinder head, which passes through the head studs, the cylinder itself, and, by means of the bolts, to the frame, where it meets a counter pressure in the opposite direction, caused by the thrust of the driving box against the pedestal. These pressures, or strains which they produce, are great (if the parts *are* quiescent), as the load is equal to the area of piston multiplied by the steam pressure. Thus, in the engine which we have been considering, we find that it is in round numbers 34 tons, and for most parts of the structure it is constantly reversing in direction. The cylinder heads and their studs are strained only one way, but the strain is released during the alternate stroke, and becomes practically zero. The cylinder connections and frames are subject to stresses in the reverse direction, and of equal magnitude, consequently the strain is more severe. Modern specifications for railway bridges take cognizance of this reversal of strain by assuming a variable unit of allowable stress, generally in some such form as follows:

Where the variable strains are of the same kind but different intensities,

$$a = b\left(1 + \frac{\text{min.}}{\text{max.}}\right)$$

and where they are of opposite kinds

$$a = b\left(1 - \frac{\text{max. lesser intensity}}{2 \times \text{max. greater intensity}}\right)$$

a being the working stress allowed and b a constant for the material of which the parts are made. In the parts of locomotives now under consideration, these formulæ can be grouped into practically three varieties: First, where the stress is uniform, as in axle and crankpin fits, and where min. = max.; second, where the stress varies from zero to a maximum, as in cylinder heads and studs, where min. = 0; third, where the stress reverses in kind but to practically same intensity, as in piston rods, where max. lesser intensity = max. greater intensity. Therefore we can confine our consideration to the following forms:

1. When min. = max., a = 2 b.
2. When min. = 0, a = b.
3. When — lesser = + greater, a = ½ b.

If we consider a quiescent load to require a factor of safety of 3 or 4, based upon the ultimate strength, these formulæ would become, letting U = ultimate strength.

1. $a = 2\,b = \frac{1}{3}$ U or $\frac{1}{4}$ U.
2. $a = \,b = \frac{1}{6}$ U or $\frac{1}{8}$ U.
3. $a = \frac{1}{2}\,b = \frac{1}{12}$ U or $\frac{1}{16}$ U.

(It would be desirable to work with the elastic limit, instead of the ultimate strength, as this would give the advantage due to nickel steel, but we are not considering the question of design entirely, but of results from certain designs.)

This would require very low strains per unit of section, or very liberal sectional areas in parts subjected to reversing stresses; for instance, with 80,000-pound steel. the unit strain would be only 5,000 to 6,000 pounds per square inch, which would generally be considered abnormally small, but these formulæ are introduced to call especial attention to the fact that the cylinder fastenings and frames, being subjected to this heavy and sudden reversal of strain, are especially liable to cause trouble, unless the proportions are ample. This manifests itself by broken frames, sheared or worn bolts in cylinder and frame connections, and occasionally in a broken cylinder. The expense of replacing these parts is great, and as they are

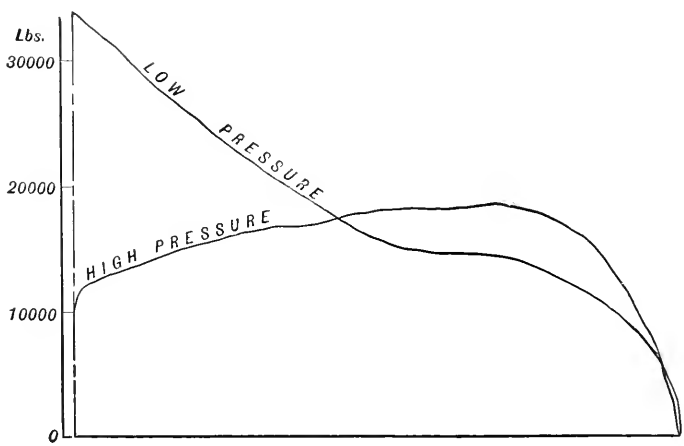


Fig. 47.

not in actual motion, relatively to the engine, this feature is sometimes overlooked, or some previous practice followed without due regard to the strains imposed. As the methods of fastening frames, etc., are so numerous, it will be impossible here to figure out actual strains, but any particular cases of interest can be studied in the light of what precedes.

PISTON RODS.

The piston, piston rod, main and side rods, crankpins and axles appeal more strongly to us, as they are in continuous motion. The piston itself is of more or less complicated form, and precludes a close analysis of its strains, but the piston rod is a comparatively simple subject. But two strains, tensile and compressive, are ordinarily induced in piston rods, but the unequal pressure upon the high and low pressure pistons of the Baldwin compound, resulting in a rapid wear of the crosshead upon the guides, causes a transverse strain, especially if the guides be neglected. The writer's instructions were to take up this crosshead wear by closing the guides, as soon as the play (vertically) amounted to 1-32 inch, in order to prevent this transverse strain so destructive to the piston rods of these engines. This unequal load is illustrated by Fig. 47, which shows the variation of total piston pressure throughout the stroke, the engine from which this diagram was worked up being a 10-wheel locomotive, with 14 and 24 by 24 inch cylinders, and when indicated was cutting off at about ¾ stroke at a speed of 21 miles an hour and with a boiler pressure of 188 pounds.

From our recent calculations, which were tabulated, we have seen that the maximum pressure which ordinarily comes upon a piston rod can be taken as the product of the boiler pressure and the area of the cylinder. As the rod is not of uniform section throughout, however, the stress will vary at the changes of cross-section. The tensile stress is likely to be greatest at a in Fig. 48, where the section is smallest at the bottom of the

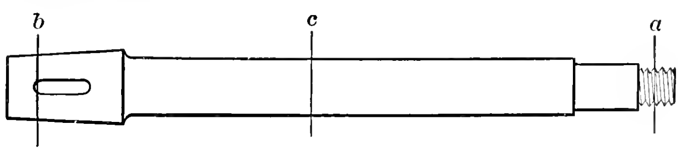


Fig. 48.

thread, although there is a possibility of a still smaller sectional area at b, especially if the crosshead fit be not enlarged. Both points should be examined; at a, the area must be taken as of a circle whose diameter is the same as the bottom of the thread. This diameter can be found by subtracting double the depth of the thread from its outside diameter—the table gives this amount to be subtracted:

Threads per inch.	Double depth in inches.
3	.433
4	.325
5	.260
6	.216
7	.185
8	.163
9	.144
10	.130

At b, the width of key-way multiplied by the diameter at that point, must be subtracted from the area at same point. The area of the straight portion should be checked also, if the thread at a be larger than c. From 8,000 to 9,000 pounds per square inch for steel rods of 80,000 pounds ultimate strength seems to correspond with current practice, and while the factor of safety is not as large as indicated by our recent discussion for reversed strains, it seems to be quite liberal.

Piston rods should have the "long column" treatment when considering the compressive strength. In order to facilitate the operation plates 20, 20a, 20b, 20c are introduced, being transcripts of those prepared by Prof. M. Merriman, and they are a graphical representation of the formula,

$$C = \frac{B}{1 - \frac{n B}{10 E} \cdot \frac{l^2}{r^2}} \qquad (60)$$

Where

C = maximum compressive unit stress on concave side at middle of column.

B = load per square unit of area.

E = modulus of elasticity of the material = 25,000,000 for iron and 30,000,000 for steel.

l = length of column.

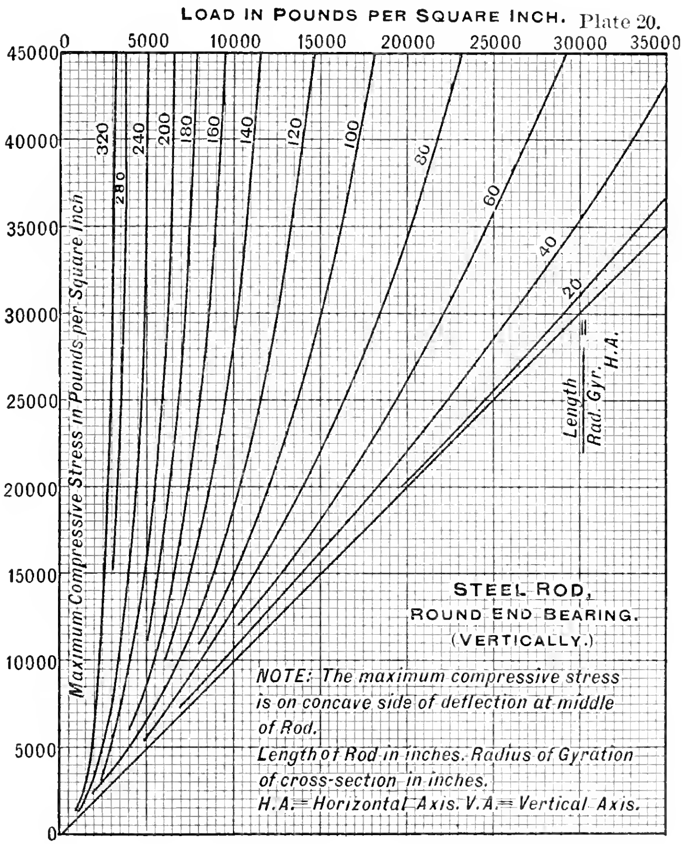
r = radius of gyration of cross-section.

n = 1 for round end bearing and ¼ for square end bearing.

All in pounds and inches. The plates give the values of C vertically for values of B horizontally. (The radius of gyration can be obtained from the tables at the end.)

As an example, let us consider the piston rod of an engine with 20-inch cylinders, carrying 200 pounds of steam, the rod

COMPRESSIVE STRESSES IN RODS.



being $3\frac{1}{4}$ inches diameter of thread (6 per inch) at a, 3 inches at c and $3\frac{3}{8}$ inches at b, with a keyway $\frac{3}{4}$ inch wide, and the length between piston and crosshead fits 42 inches, the material being steel of 80,000 pounds tensile strength.

The total pressure $= 314 \times 200 = 62{,}800$ pounds.

The area at a $= (3\frac{1}{4} - .216)^2 \times .7854 = 7.21$ square inches.

$b = (3\frac{3}{8})^2 \times .7854 - 3\frac{3}{8} \times \frac{3}{4} = 6.42$ square inches,
$c = 3^2 \times .7854 = 7.07$ square inches.

Therefore, the greatest tensile strain will be found at b and equals $\frac{62,800}{6.42} = 9,800$ pounds per square inch. For compres-

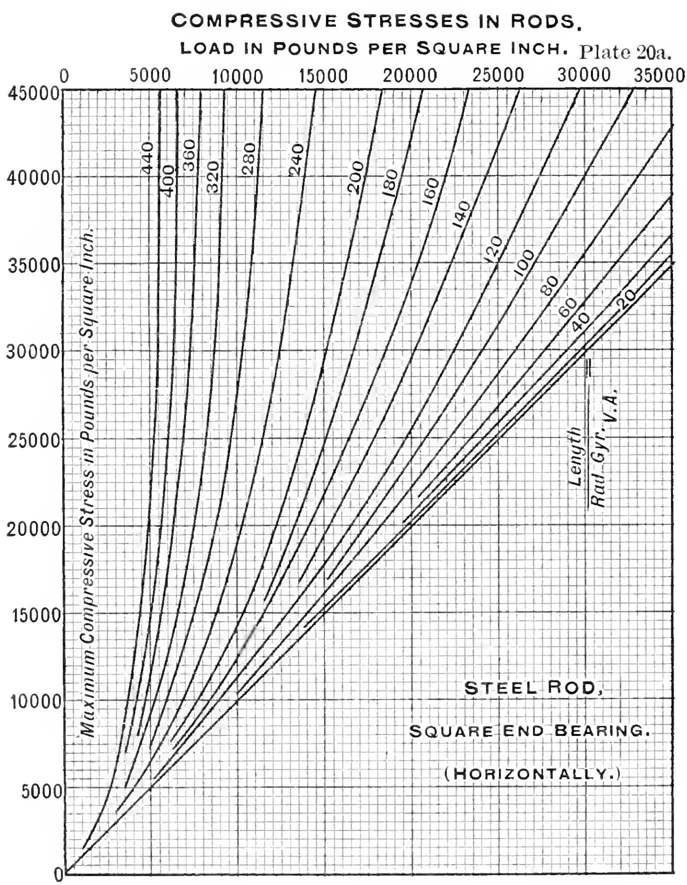


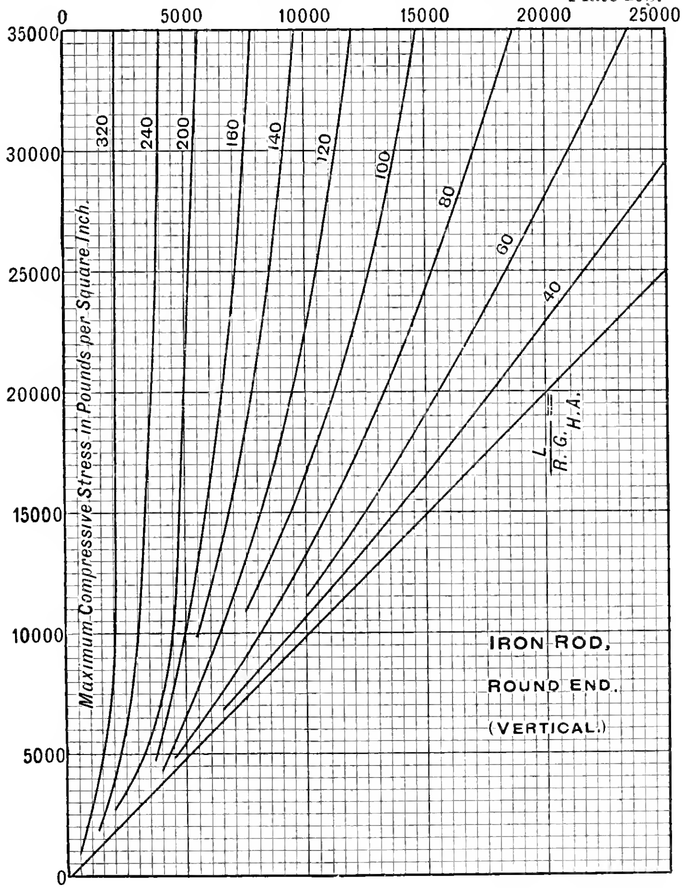
Plate 20a.

sion, we find from the tables that the radius of gyration of a 3-inch circle is .75 and as the length between fits is 42 inches, and these fits support it so securely that we can consider them to constitute a square bearing, we can use the diagram of plate

20a. For length ÷ radius of gyration, we have 42 ÷ .75 = 56, so on the nearest line, marked "60," we follow down until we find the intersection with the unit load = 62,800 ÷ 7.07 = 8,900 pounds per square inch (using the area at center of rod

COMPRESSIVE STRESSES IN RODS.

LOAD IN POUNDS PER SQUARE INCH. Plate 20b.



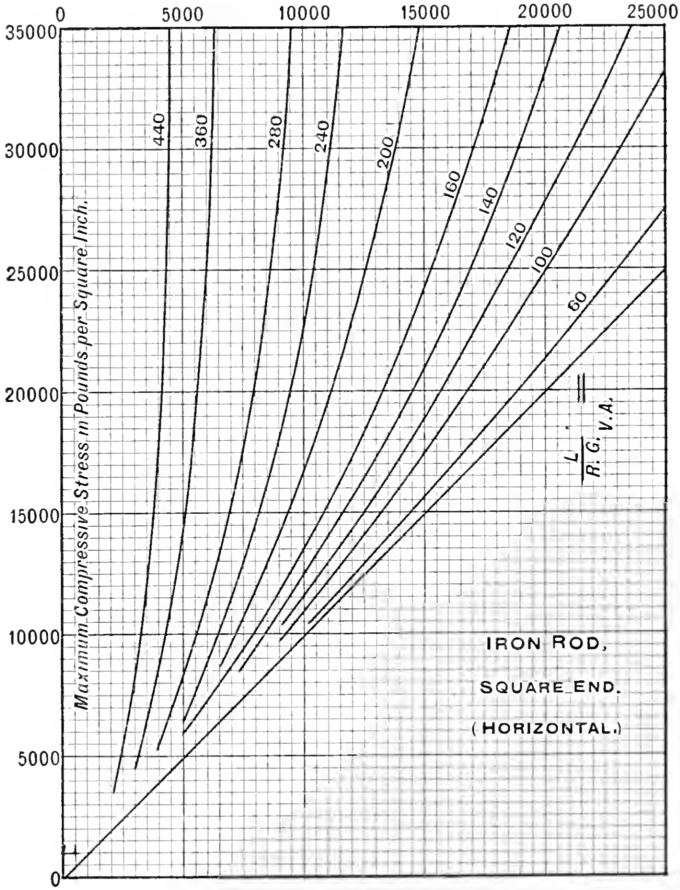
or c) and find the maximum compressive stress to be about 9,100 pounds per square inch.

Hollow rods, or tubes, are treated in the same way, and from the tables it will be seen that such forms have a higher value for radius of gyration than the solid rod, but the area of metal to stand the strain is also much less, and the unit stress is increased on this account.

The key retaining the piston rod in the crosshead is in

COMPRESSIVE STRESSES IN RODS.

LOAD IN POUNDS PER SQUARE INCH. Plate 20c.



double shear. In our example it was ¾-inch thick and, let us assume 3 inches wide. The area of section in shear will be $\frac{3}{4} \times 3 = 2.25$ square inches, or 4.5 square inches total. The shearing stress is therefore $62{,}800 \div 4.5 = 14{,}000$ pounds per square inch. The strain on the key can never reverse in direction, so that a much higher unit stress can be safely used. In fact, as these keys are driven in with a sledge, the strain is probably a constant one, as the tension or stretch of the rod (if it comes to a shoulder) will not be greatly increased, if at all, by the pull of the piston on its forward stroke. We must remember, however, that the shearing resistance of a metal to rupture is only about 4-5 of its tensile strength. If the latter be 80,000 pounds, the shearing strength should be taken as 64,000 pounds, and the working stress (without variation) may be 1-3 or ¼ of this or say from 16,000 to 21,000 pounds per square inch. This part of the connections has received little attention, as by figuring the shearing stress in a number of cases, a variation from 13,000 to 32,000 pounds per square inch was found. These keys "shoulder" or "set" very frequently in practice, and at times, shear completely in two, and an investigation into the shearing stress actually produced in the key, will often reveal the causes of trouble. It is a small item, but of considerable consequence.

GUIDES.

The crosshead causes a varying pressure upon the guides, which can be analyzed as follows: In Fig. 49, let P be the

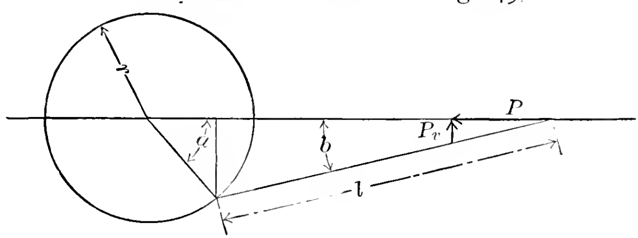


Fig. 49.

total effective pressure upon the piston at the instant of consideration. Then the pressure against the guides will be $P_v = P \tan b$. But as we found in producing formula 58,

$$\sin b = \frac{r}{l} \sin a \text{ and } \cos b = \sqrt{1 - \frac{r^2}{l^2} \sin^2 a}$$

and as $\tan b = \dfrac{\sin b}{\cos b}$ we can write

$$P_v = P\frac{r}{l}\frac{\sin a}{\sqrt{1 - \frac{r^2}{l^2}\sin^2 a}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (61)$$

We have seen that P generally varies throughout the stroke, but in starting, P not only has its greatest value, but this value is constant through all but a small portion at the end of the stroke. Examination of equation 61 shows that it will be a maximum when sin a is a maximum, or when a = 90 degrees, in which case we have

$$P_{v\ max} = P\frac{r}{l}\frac{1}{\sqrt{1 - \frac{r^2}{l^2}}},$$

but as $\frac{r^2}{l^2}$ is small we can write more simply

$$P_{v\ max} = P\frac{r}{l} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (62)$$

and this occurs when the crank is at the 90 degree point on either the backward or the forward stroke. At the dead centers sin a = 0 and $P_v = 0$, gradually increasing to the maximum value nearly in proportion to the sine of the angles. In running forward, the pressure is upwards on both the forward and backward strokes. This accounts for all the wear coming upon the top guides of road engines; the lower guide is worn only when backing. The guides are subject, not only to a bending strain caused by the crosshead, but also to a deflection, which, if great, will throw the piston and rod out of line. If the guide be rectangular in section, the section modulus will be $\frac{bh^2}{6}$, where b = the breadth and h = the height (or thickness measured vertically). Consider the length in inches = L, that is, between the supporting lugs or bolts, assuming that the guide is not braced at any point between the ends, then the bending moment will be $\frac{P_v L}{4}$, and this must equal $\frac{b h^2}{6} f$, where f = the maximum fiber strain in the guides. Substituting the value given in equation 62, we have

$$\frac{P r L}{4 l} = \frac{b h^2}{6} f \text{ and } f = \frac{3 P r L}{2 b h^2 l} \quad \ldots\ldots\ldots\ldots \quad (63)$$

P being as before the steam pressure multiplied by the area of piston. If each top and bottom guide consists of two bars, b must represent the total width; that is, of both bars. The thickness h is to be taken at the center, where the maximum pressure is applied. As guides are usually of uniform height or thickness (or nearly so) throughout their length, it will be unnecessary to figure the strain at points other than the center.

The deflection of the guide can be determined by the formula,

$$d = \frac{P_v L^3}{4 E b h^3} = \frac{P r L^3}{4 E l b h^3} \quad \ldots\ldots\ldots\ldots \quad (64)$$

Where d = deflection in inches at center.

E = modulus of elasticity of material, 25,000,000 for wrought iron and 30,000,000 for steel, the other values as before.

Let us consider the case of the 20-inch cylinder discussed regarding the piston rod strains, where P = 62,800; L = 50 inches; b = 5 inches; h = 2¾ inches; $\frac{l}{r} = 10$; and E = 30,000,000 pounds, the guides being of 80,000-pound steel. Then from equation 63 we have

$$f = \frac{3 \times 62{,}800 \times 50}{2 \times 5 \times 7\frac{1}{2} \times 10} = 12{,}560 \text{ pounds per square inch,}$$

remembering that $(2\frac{3}{4})^2 = 7\frac{1}{2}$. As this strain varies from zero to the maximum just calculated, $\frac{80{,}000}{6} = 13{,}000$ pounds should provide ample safety, so that we are within the safe limit for strength.

From equation 64 we have for the deflection at center of guide

$$d = \frac{62,800 \times 125,000}{4 \times 30,000,000 \times 10 \times 5 \times 21} = .06, \text{ or } 1\text{-}16 \text{ inch},$$

as $50^3 = 125,000$ and $(2\frac{3}{4})^3 = 21$. This amount is rather excessive; it should not exceed 1-32 inch, so we find that while the guide is strong enough, it is not stiff enough.

By clamping the top and bottom guides together at the center, as is sometimes done, we make the bottom guide take half of the strain (but not the wear) when running forward. Under these conditions we practically double the width, b, and so halve the strain and the deflection.

When we have a case like the Baldwin compound, with two piston rods acting on a single crosshead, but at quite a distance apart vertically, we obtain an additional strain upon the guides. Figure 47 shows that in the case then being considered, there was a difference of about 20,000 pounds between the high and low pressure pistons (the low pressure being that amount in excess), at the beginning of the stroke, but that it gradually reduced to almost equality near the middle, and continued approximately so to the end of stroke. The greatest strain will therefore be at the commencement of stroke. If the distance between centers of piston rods is 20 inches, and the length of crosshead 24 inches, we have $\frac{20,000 \times 10}{24} = 8.333$ pounds as the pressure of the toe of crosshead against the guide. If the low pressure piston be below, as in the case in hand, the pressure will come against the middle of top guide and the end of bottom guide. The upward pressure due to angularity of connecting rod at center of stroke will be from formula 62, with the combined crosshead pressure of 40,000 pounds; $\frac{40,000}{10} = 4,000$ pounds, from which it is apparent that the uneven pressures upon the pistons will produce twice as much strain upon the guides as the angularity of the connecting rod. This causes the point or toe of the crosshead to wear rapidly, though this is reduced by lengthening the crosshead and using a hard metal lining at the ends. As the crossheads wear more at the ends than in the center, the rocking causes a bending strain upon the piston rods. When the low pressure cylinder is above, the tendency is to lift the pistons, causing the cylinders to wear most at the top. The piston rods must be made very heavy to stand this bending as well as the longitudinal strains.

The guide yoke must take about one-half of the upward thrust $= P\frac{r}{l}$, if the guides are supported at the end; if near the middle, then the yoke takes practically the full thrust, $P\frac{r}{l}$. The bolts connecting the yoke to the frame brackets are subject to constant wear and need renewal quite frequently. As the part is not in motion, these wearing points are not as carefully examined as those actually moving, but the varying load causes the bolts and connections to become loose.

The crosshead wristpin is in shearing when it has a good bearing on the rod brass, but as this may not always occur, it is safest to consider it in bending, and its length equal to the brass bearing, with a central load. We found in equation 56 that the load upon the connecting rod was $P_r = \frac{P}{\cos b}$, and substituting for cos b, its value in terms of a, we get

$$P_r = \frac{P}{\sqrt{1 - \frac{r^2}{l^2}\sin^2 a}} \quad \text{. . . . . (65)}$$

This will evidently be a maximum when a = 90 degrees, when

$$P_r = \frac{P}{\sqrt{1 - \frac{r^2}{l^2}}}$$

Even for the proportion of $l = 5\ r$, we find that P_r will be only 2 per cent greater than P, and with longer rods, the increase will be less, so that we may safely assume $P_r = P$. If l = the length of bearing of pin, between crosshead fits, we have the bending moment $= \frac{P\,l}{4}$, and if S = the section modulus (see the tables for this function), the strain per square inch will be

$$f = \frac{P\,l}{4\,S} \quad \text{.......... (66)}$$

If in the case which we have been considering, the crosshead pin was 3 inches in diameter, with a 3-inch bearing, we should have (the section modulus being 2.66 for a 3-inch circle)

$$\frac{62{,}800 \times 3}{4 \times 2.66} = 18{,}000 \text{ pounds strain.}$$

This is somewhat high, as we should be limited to 15,000 pounds for steel and 12,000 pounds for wrought iron.

RODS.

The connecting rod is strained in a number of ways. First, by a direct tensile stress; second, by a direct compressive stress; third, by compression as in a long column, with neutral axis vertical and square end bearings; fourth, by compression as above, but with neutral axis horizontal and round end bearings; fifth, by compression as a long column, with round end bearings and neutral axis horizontal, and also the bending action due to inertia vertically at high speeds.

First—The tensile strain at any point is obtained by dividing the total piston pressure P by the net area at the point. This area is generally smallest either at the eye containing the front end brass, or in the neck immediately back of the "end." When the end is being considered, the proper deductions must be made for bolt holes, oil holes, keyways, etc., in order to obtain the net area.

Second—This compressive strain is figured in the same way as the first, and is generally greatest at the "neck" immediately back of the end.

Third—For "long compression" horizontally (that is, against horizontal springing of the rod), we should take the area at the center for determining the unit load, and also the radius of gyration about a vertical axis of the section at center of rod. The area and radius of gyration (axis vertical) may be obtained from the tables under columns A and r. If L = length of rod between centers in inches, we must select a curve on plate 20a or 20c having the marked value of $\frac{L}{r}$, or interpolate if necessary, as the ends are practically "square bearing" for horizontal deflection. At the intersection of the proper curve and the abscissa corresponding to $\frac{P}{A}$ (using the values at the top of diagram) we read off on the left side the maximum compressive stress on the concave side when deflection occurs or would occur.

Fourth—For this case we proceed as indicated in the last clause, except that we use R = radius of gyration about a horizontal axis, and use plate 20 or 20b, as the deflection is now being considered vertically, and the pins constitute round end bearings.

Fifth—Here we must combine the effect of vertical inertia and the stresses due to vertical deflection under long compression. By equation 12 we found that the maximum fiber strain due to vertical inertia at a speed in miles per hour equal to the diameter of drivers in inches was $f = \frac{.1 \, s \, G \, L}{S}$ at center of rod, s being the stroke in inches, G weight of rod in pounds, L the length, center to center, in inches, and S the modulus of section around a horizontal axis. To this f must be added the value of C in formula 60, as the combined stress is $f + C = \frac{.1 \, s \, G \, L}{S} + \frac{B}{1 - \frac{n \, B}{10 \, E} \cdot \frac{l^2}{r^2}}$. We have shown that plates 20 to 20c can be quickly used to determine the value of

C, also at middle of rod. At such high speeds, we found when discussing the rotative force, that it is impossible to obtain anything like full boiler pressure at the middle of stroke, which is the point at which f reaches a maximum, and if we take the longitudinal force on the rod at this point and speed as equal to ½ P we will be well on the side of safety. This will give us a unit load of $\frac{P}{2A}$ to use in connection with plate 20 or 20b.

The first and second cases are so simple that they hardly need any example, but the third to fifth can be illustrated with advantage. The connecting rod of a 4-cylinder compound locomotive was 10 feet long and had a section at the middle as shown in Fig. 50, the cylinders being 14 and 24 inches in diameter, with 200 pounds boiler pressure. The full pressure could be turned into the low-pressure cylinder when starting, the high-pressure piston being almost balanced on each side. Thus we can take P = 425 × 180 = 80,000 pounds, that is, the area of the low-pressure cylinder by 90 per cent of the boiler pressure. In the table of I-sections, 5½ inches high, as our flange is ¾ inch thick, we must interpolate between the ½ and 1 inch flanges, and so obtain for our section (Fig. 50) A = 5.37, S = 8.00, R = 2.00, r = .53.

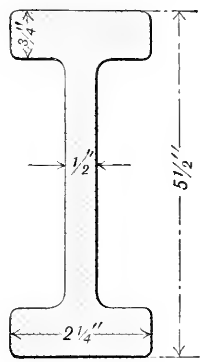


Fig. 50.

As L = 120 inches, we have $\frac{L}{r} = \frac{120}{.53} = 226$, and $\frac{P}{A} = \frac{80,000}{5.37} = 15,000$ pounds per square inch as the unit load. As the rod was of steel, we use plate 20a, marked "Steel Rod—Horizontally" in order to find the maximum stress due to horizontal deflection, as in case 3. Now, as 226 is slightly more than half way between 200 and 240 (for which we have curves), we find that a line for $\frac{L}{r} = 226$ would cross the 15,000-pound $\frac{P}{A}$ line at about 39,000 pounds, as shown on the left, and this would be the maximum compressive stress on the concave side, should the rod spring horizontally due to the piston pressure. It will be seen at once that this is a very high strain—almost the elastic limit, and as a matter of fact, the rod actually sprung or bulged sidewise (horizontally) when starting with the engine operated simple.

If we consider the same rod on a simple engine the equivalent cylinders, say 20 inches in diameter, and take the maximum piston pressure $= 314 \times 200 = 62{,}800$ pounds, we have for case 3, $\frac{P}{A} = \frac{62{,}800}{5.37} = 11{,}700$ pounds, and by the diagram the maximum compressive stress would be 22,000 pounds per square inch. This is still higher than good practice would dictate, as it should not exceed 13,000 pounds. For the same conditions, case 4 would give us for vertical springing, where $\frac{L}{R} = \frac{120}{2} = 60$ (using plate 20), 18,000 pounds maximum compressive stress for the compound engine and 13,700 pounds for the simple engine, these values being found on the left-hand side of diagram where the $\frac{L}{R} = 60$ curve intersects the 15,000 and the 11,700 lines of $\frac{P}{A}$, or loads per square inch.

For case 5 we must combine the effects of inertia at maximum speed. The rod weighs about 200 pounds between centers of pins, the stroke of piston is 24 inches, and we found by the tables that S = 8. Therefore, the strain per square inch due to inertia at this speed $= f = \frac{.1 s G L}{S} =$

$$\frac{.1 \times 24 \times 200 \times 120}{8} = 7{,}200$$ pounds. If we take the piston pressure at one-half that at slow speeds, we have $\frac{P}{2\,A} = \frac{62{,}800}{2 \times 5.37} = 5{,}850$ pounds. This corresponds to about 6,500 pounds maximum strain due to compression. The sum of these two $= 7{,}200 + 6{,}500 = 13{,}700$ pounds per square inch, is the extreme fiber stress due to both causes. If the pressure upon the piston at the maximum speed is found to be only one-fourth that at starting, we can use $\frac{P}{4\,A} = 2{,}925$ pounds unit load, which gives about 3,000 pounds maximum compression, and $7{,}200 + 3{,}000 = 10{,}200$ pounds for the total strain. Thus, while this rod was weak horizontally, it was quite strong enough vertically.

In 2-cylinder compounds, the total piston force is often taken as that necessary to slip the wheels, as the low-pressure cylinder, if provided with high-pressure steam, will have a greater force than can be held down by the drivers. In this case, the coefficient of adhesion between the wheels and the rail should be taken quite high, say 30 per cent, or $\frac{1}{3\frac{1}{2}}$. In such cases we should have

$$P = \frac{\text{Weight on drivers} \times \text{Diameter of drivers}}{3\frac{1}{2} \times \text{Stroke}} \quad \ldots\ldots\ (67)$$

where the total weight on all drivers must be taken, as if the high-pressure piston be at a dead center, the low-pressure piston would be able to slip all drivers, if given sufficient pressure.

The parallel rods are to be treated in a manner similar to the connecting rod, but the longitudinal force must be considered from a different standpoint; in fact, two different values must be used for the different cases. In cases 1 to 4, inclusive, which cover the starting of the engine, and when we have full pressure in the cylinder, it is evident that if the main wheels happen to rest upon a part of the track more slippery than the other wheels, that sufficient force must pass through the parallel rod to slip the coupled wheels. In this case the force will be as in equation 67, except that only the weight of drivers operated by the rod under consideration (but including both sides of the engine) should be taken. Thus, if an engine of the 4—4—2 type be under consideration (see

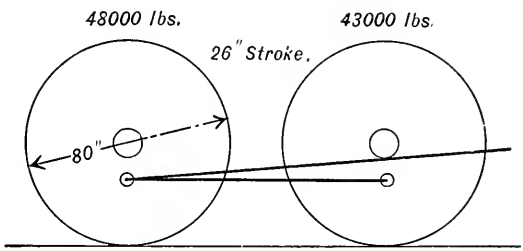


Fig. 51.

Fig. 51), the weights and sizes being as shown, it is evident that the side rod might be called upon to slip the forward wheels, which would result in a force $P = \frac{43{,}000 \times 80}{3\frac{1}{2} \times 26} =$ 37,800 pounds passing through the rod. As the engine has 20-inch cylinders and 200 pounds boiler pressure, it would ordinarily be considered that but $\frac{314 \times 200}{2} = 31{,}400$ pounds, one-half of the total piston pressure, would be the maximum force transmitted. In the case of a consolidation type, Fig. 52, the rods a and c will only have to slip the wheels D and G, respectively, but the rod B must be strong enough to slip the wheels F and G, and the weight upon these two pairs of wheels must be used in finding the maximum load for the rod B.

In case 5, at high speeds, it will be correct to take the proportion of the piston pressure represented by the proportion of wheels operated by the rod. For instance, in Fig. 51, the pressure transmitted by the side rod would be one-half of that upon the piston, and in Fig. 52, one-half for rod B and one-quarter each for rods A and C, remembering that the piston pressure at high speeds is much below that when starting, as we have seen by our calculations in connection with the rotative force.

With these differences in the longitudinal force explained, the consideration of cases 1 to 4 will be the same as for con-

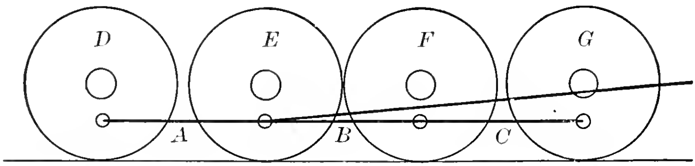


Fig. 52.

necting rods, and plates 20, a, b and c will give the maximum compressive stress.

For the case at high speeds, including the effect of inertia, we must combine the stress found by equation 11 and the compressive stress found by plates 20 and 20b.

As an example we will discuss for cases 3, 4 and 5, the rods of rectangular section used to illustrate formula 11, and of I-section shown in Fig. 50 as applied to locomotive of Fig. 51. The cylinders are considered as being 20 inches diameter by 26-inch stroke, boiler pressure 200 pounds, drivers 80 inches diameter, with 43,000 pounds on the front pair, the rod being 90 inches between centers. The maximum force that can come in this rod is that which would slip the front wheels, and is, as we have already seen, $P = \frac{43{,}000 \times 80}{3\frac{1}{2} \times 26} = 37{,}800$ pounds. This value of P will be used in third and fourth cases. The steam pressure at high speed in case 5 will cause a strain which we may take as $P' = \frac{314 \times 200}{2 \times 2} = 15{,}700$ pounds, this being half of the total piston pressure at high speed, which, in turn, is taken as half of the "slow" pressure. In order to show the results of both rods, the different calculations will be paralleled.

Rectangular Section.

Height, 5 inches; width, 2½ inches; weight, 315 pounds.

$A = 12.5$, $S = 10.42$, $R = 1.44$, $r = .72$ — From tables.

Case III—Horizontal Deflection.

$$\frac{P}{A} = \frac{37{,}800}{12.5} = 3{,}000$$

$$\frac{L}{r} = \frac{90}{.72} = 125$$

So max. comp. stress = 3,000 by plate 20a.

Case IV—Vertical Deflection.

$$\frac{L}{R} = \frac{90}{1.44} = 62,$$ and by plate 20 max. comp. stress = 3,000.

Case V—High Speed.

By eq. 11:

$$f = \frac{.2\, s\, G\, L}{S} = \frac{.2 \times 26 \times 315 \times 90}{10.42} = 14{,}200$$

$$\frac{P'}{A} = \frac{15{,}700}{12.5} = 1{,}250,$$ and as $\frac{L}{R} = 62$, max. comp. stress = 1,250, and 14,200 + 1,250 = 15,450 pounds total fiber strain.

I Section.

Height, 5½ inches; width, 2¼ inches; weight, 150 pounds.

$A = 5.37$, $S = 8.00$, $R = 2.00$, $r = .53$ — From tables.

Case III—Horizontal Deflection.

$$\frac{P}{A} = \frac{37{,}800}{5.37} = 7{,}000$$

$$\frac{L}{r} = \frac{90}{.53} = 170$$

So max. comp. stress = 8,500 by plate 20a.

Case IV—Vertical Deflection.

$$\frac{L}{R} = \frac{90}{2} = 45,$$ and by plate 20 max. comp. stress = 7,500.

Case V—High Speed.

By eq. 11:

$$f = \frac{.2\, s\, G\, L}{S} = \frac{.2 \times 26 \times 150 \times 90}{8} = 8{,}800.$$

$$\frac{P'}{A} = \frac{15{,}700}{5.37} = 2{,}900,$$ and as $\frac{L}{R} = 45$ max. comp. stress = 3,000. 8,800 + 3,000 = 11,800 pounds, total fiber strain.

Thus it is evident, that while the I-section is much lighter, it is strained less per square inch than the rectangular section. The above calculations are made with the assumed maximum piston pressure, and the highest speed at which the engine is supposed to work, but any other speed or pressure can readily be used if so desired. It is believed that this study of the rods will make the analysis of the working strains easy of solution for any set of conditions.

CRANKPINS.

The crankpin, in receiving the thrust of the rod and transmitting this force to the wheel, plays a very important part.

Crankpin breakages are rather frequent, even when there is apparently a good factor of safety employed in the design. It is probable that many fractures are started by water in the cylinder, in which case the mechanism acts like a toggle joint at the end of the stroke, and the fracture begun, it continues to increase until the end of the pin drops off the wheel. Many such fractures show evidence of considerable antiquity by the smooth way in which the surface has been worn by the working of one part upon the other, indicating the severe strain to which the part giving way last has been subjected. It is often found that the pin has been running for some time with very much less than one-half of the original section intact, and as the strength of the pin varies as the cube of its diameter, it is evident that when it gave way the unit stress was at least eight times what was originally intended.

Such breakages are often serious, not on account of the value of the pin, but owing to the consequential damage resulting. In many cases the main rod is badly bent and the cylinder head or even the cylinder itself is destroyed; it sometimes happens, however, that, beyond delaying the train, the consequences are not serious. Partial fractures can often be discovered before they give entirely away, and whenever the rods are removed from an engine in service, and opportunity is thus afforded, a careful inspection should be made. If the pin is reduced in diameter in the wheel fit, the fracture is almost certain to start back of the collar, where it cannot be seen by any kind of inspection. It is current practice, however, to use enlarged wheel fits, in which case fracture is likely to start in the fillet of the inside bearing, at which point its discovery is comparatively easy. As pins are so easily replaced, they should never be allowed to run in a condition that is the least questionable, as a little water in the cylinder, or a sudden slipping, may complete the rupture.

Crankpins are subject to bending strains only, but, as in large modern engines, these strains are great, they are preferably made of a high grade of steel—sometimes nickel steel. The main pins, having two bearings, receive a partial support from the side rods, but it is best not to consider this support when determining the working stress in the pin. In Fig. 53, P represents the force of the main rod upon the pin, which, as we have seen in connection with equation 65, may ordinarily be taken as the product of the piston area and the boiler pressure, but for 2-cylinder compounds (and perhaps 4-cylinder compounds also), the value should be determined by formula 67. The side rod reacts in the opposite direction, and with a force P′. The bending moment at the face of the wheel hub would be $P\,l - P'\,l'$, and at any point outside of the side rod, simply $P'x$. It will be apparent, with a little study, that the side rod

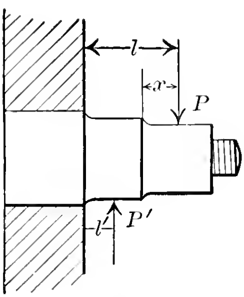


Fig. 53.

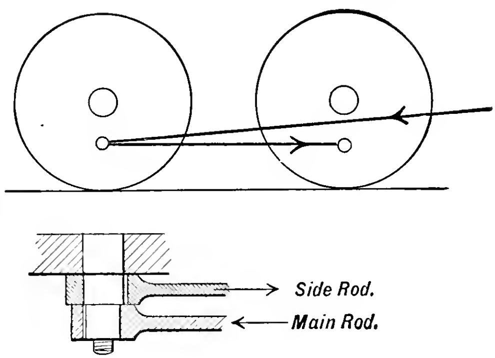


Fig. 54.

does not always exert an opposing force, P′. We all know that the rods are never tight on the pins, and the side rods especially are liable to run with 1-16 or ⅛ inch wear oblong in the brass, that is, in the direction of the axis of the rod.

If, now, in Fig. 54, we consider the crank as being on the quarter, and the rods acting in the direction of the arrows, it is evident that the play in the brass is taken up in the main rod on the front of the pin, and in the side rod on the back of the pin; in this position the side rod assists the pin in opposing the force of the main rod. If, however, the engine is on a center at the instant which we are considering, the play in the side rod will naturally be back of the pin, as the main pin has been pushing through the side rod. But steam pressure pushes the main rod, so that the contact is also on the front of the pin, as shown in Fig. 55. Under these circumstances it is evident that the pin obtains no support from the side rod, and we have seen that, at this position, the piston pressure is greatest—we will, therefore, in our analysis, neglect any support from the side rod, and write the bending moment upon

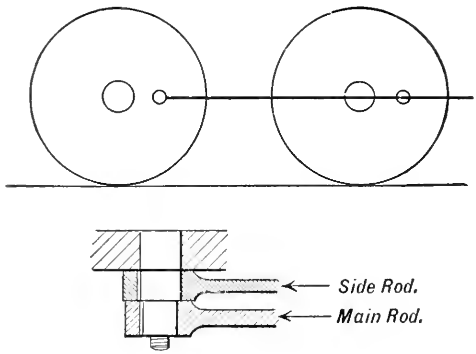


Fig. 55.

the pin at the face of the crank hub = P l. The modulus of section of circular form is $\frac{\pi d^3}{32}$, d being the diameter of the circle in inches, and if f = fiber strain, as before, we have

$$P\,l = \frac{\pi d^3}{32} f \text{ and } f = \frac{32\,P\,l}{\pi d^3} \text{ or } \frac{10\,P\,l}{d^3} \text{ (nearly)} \quad \ldots\ldots\ldots (68)$$

l being the distance from center of main rod bearing to face of crank hub, in inches. The strain at any other point distant x from center of main rod bearing is $f = \frac{10\,P\,x}{d_x^{\,3}}$, d_x being the diameter of the pin at the distance x.

Pins other than main should be examined in a similar manner, but the force in this case should be the amount necessary to slip the pair of wheels upon which they act, in accordance with equation 67.

The proper limit of stress in crankpins has been much discussed. Records of breakages do not always afford a satisfactory method of settling the question. For instance, on a large western road, operating a thousand locomotives, there were reported broken in the year 1899 five pins which, by formula 68, were strained between 8,000 and 9,000 pounds per square inch; eight between 16,000 and 20,000, five between 20,000 and 23,000, and three over 25,000 pounds, the pins being both iron and steel Probably the safe maximum stress would be

Iron	12,000 to 14,000 pounds per square inch
Steel	15,000 to 17,000 pounds per square inch
Nickel steel	18,000 to 20,000 pounds per square inch

although we have seen that some broke under these strains, and others ran without trouble with a much higher stress.

As an example, let us consider a main pin on a locomotive with 20-inch cylinders, 175 pounds boiler pressure, $7\frac{3}{4}$ inches from center of main rod bearing to face of crank hub, and $6\frac{3}{4}$ inches diameter of wheel fit. Then, from equation 68,

$$f = \frac{10\ P\ l}{d^3} = \frac{10 \times 314 \times 175 \times 7\frac{3}{4}}{(6\frac{3}{4})^3} = 13{,}850 \text{ pounds.}$$

If we consider the front pin in the example of side rod strains taken above, the wheel fit being $4\frac{1}{2}$ inches in diameter and the center of side rod bearing 3 inches from face of hub, we have, as before, for the load upon the pin necessary to slip the wheel $P = \frac{43{,}000 \times 80}{3\frac{1}{2} \times 26} = 37{,}800$ pounds, and for the stress $f = \frac{10\ P\ l}{d^3} = \frac{10 \times 37{,}800 \times 3}{(4\frac{1}{2})^3} = 12{,}450$ pounds per square inch.

DRIVING AXLES.

The driving axle is one of the most important parts of the engine, and has a great variety of strains imposed upon it, that of carrying the load being the least. The whole power of the engine is transferred to the train through these axles, and besides, the continual jar and pound in running at high speeds over track more or less rough constitute a severe punishment in addition to the strains imposed by the action of the steam. Driving axles generally break close to the wheel fit, though sometimes closer to the center of the journal. The practice (now generally abandoned) of making the wheel fit smaller than the journal, produced a sudden change of cross-section, in itself liable to start a fracture, and when so started, the neck being hidden from inspection by the hub of the wheel, no warning cracks could be discovered before breakage on the road. The enlarged wheel fit, now almost universally adopted, throws the point of rupture into the journal itself, and when the wheels are removed, as at shopping periods, there is full opportunity afforded for a close inspection. Some of these axle breakages are quite curious, the author having in mind one case where the axle broke off close to the right and also the left wheel at the same instant. For a long time hammered iron was the favorite material for driving axles, as it was thought that the fibrous material would permit a crack to announce itself before total fracture occurred, but the reduced wheel fit formerly used prevented the warning being made apparent. The axles of modern locomotives are so massive that it is practically impossible to obtain them of iron properly worked under the hammer clear to the center, and as the rolling process by which steel billets are finished consolidates the metal much more perfectly, besides having a greater modulus of strength, it is the generally accepted material. Even steel axles are not used without failure, but in many locomotives the proportions are not what they should be, too much importance being attached to the use of a standard box or wheel center. While failures will probably never be entirely eliminated, they can be greatly reduced by proper design and careful workmanship and inspection. The importance of the latter should not be underestimated. One large transcontinental line has adopted the following practice: When driving wheels (on axles) are brought to the wheel lathe for tire turning, the journals are carefully cleaned with naphtha, which removes all the grease; they then being coated thinly with white paint, in which only turpentine is used as a vehicle. This is allowed to dry before the wheels are put into the lathe. When the tires are being turned, the stress in the axle due to the pressure of the tool, etc., opens any crack that may exist, and the grease exudes from it, and discolors the white paint on the journal, thereby giving notice of the presence of the defect.

The normal strains in a driving axle may be analyzed by using Fig. 56 as a guide. The dimensions represented in the sketch by a, b, c, d and e are all in inches, and the forces are in pounds. W is the weight upon the two journals, and for this investigation may be taken without large error as equal to the weight of the pair of drivers upon the rails. The force $\frac{W}{2}$ is supposed to act at the center of the journal, and the forces

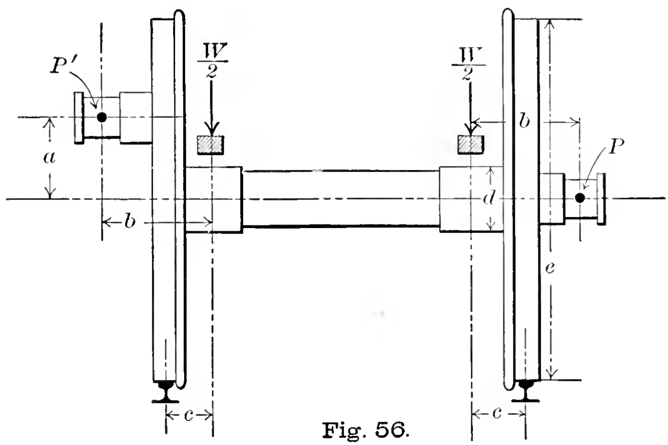


Fig. 56.

P and P′ at the center of the main rod bearing. The force P is that produced by the steam pressure upon the piston transmitted by the main rod, and is the product of the area and boiler pressure, as in the investigations of the main rod strains. The force P′ is that which, applied to the pin normal to the crank line, would slip the wheels, and in accordance with equation 67 is $P' = \frac{W e}{3.5 \times 2 a}$. S is the section modulus, approximately $\frac{d^3}{10}$ where d is the diameter of the journal, and n is the number of drivers on one side of the engine.

In the main driving axle, we have two cases to consider—one when on the dead center, with steam admitted to start a stroke, and the other on the top quarter, these being the maximum points for the two cases, respectively.

At dead center, or commencement of stroke, there will be a horizontal bending moment caused by force P and equal to = P b. The weight of the engine acting vertically will produce a bending moment in that direction = ½ W c. The resultant of these moments will be in a diagonal direction, and will be equal to the hypothenuse of a right-handed triangle, of which the right-angled sides are respectively the horizontal and vertical moments, or =

$$\sqrt{(P\,b)^2 + (\tfrac{1}{2}\,W\,c)^2}$$

and this equals the section modulus by the fiber strain = S f $= \frac{d^3}{10} f$. Equating and transposing, we have the stress resulting from the steam and weight of engine =

$$f = \frac{10}{d^3}\sqrt{(P\,b)^2 + (\tfrac{1}{2}\,W\,c)^2} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (69)$$

The full piston pressure P is here taken, as it has been shown in our study of crankpins, that the pin is likely to be deprived of the support of the side rods when at the end of the stroke. On the quarter, however, the case is different, as the wheels would slip until the side rods took their share of the load. This is the reason for using P′ instead of $\frac{P}{n}$ in the paragraph following.

At the top quarter or half stroke, we still have the vertical bending moment = ½ W c as at the end of the stroke, but the horizontal force is dependent upon the slipping of the wheels in question, for it is evident that the other wheels may slip, causing these to follow. The force P′ causes a horizontal bending moment P′ b; but the resistance of the near wheel against slipping also causes a horizontal bending moment in the same direction, whose value is $\frac{W\,c}{2 \times 3.5}$, therefore the horizontal bending becomes

$$P' b + \frac{W c}{7} = \frac{W e b}{7 a} + \frac{W c}{7} = \frac{W e b + W c a}{7 a} = W \frac{e b + c a}{7 a}.$$

In addition to this, there is a twisting force in the axle, caused by the resistance of the far wheel to slipping, which is equal to

$$\tfrac{1}{2} P' a = \frac{W e a}{2 \times 3.5 \times 2 a} = \frac{W}{2 \times 3.5} \times \frac{e}{2} = \frac{W e}{14}.$$

We can combine the bending moments as before, and obtain the resultant bending moment

$$= \sqrt{(\tfrac{1}{2} W c)^2 + \left(W \frac{e b + c a}{7 a}\right)^2}$$

Rankine has shown that bending and twisting moments may be combined to produce equivalent bending moments by the formula

$$M' = \tfrac{1}{2} (M + \sqrt{M^2 + T^2}) \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (70)$$

Where M = bending moment,

T = twisting moment,

M' = equivalent bending moment.

Now by substituting $\sqrt{(\frac{1}{2} W c)^2 + \left(W \frac{e b + c a}{7 a}\right)^2}$ for M and $\frac{W e}{14}$ for T, we obtain the equivalent bending moment =

$$\tfrac{1}{2} \left[\sqrt{(\tfrac{1}{2} W c)^2 + \left(W \frac{e b + c a}{7 a}\right)^2} + \sqrt{(\tfrac{1}{2} W c)^2 + \left(W \frac{e b + c a}{7 a}\right)^2 + \left(\frac{W e}{14}\right)^2} \right]$$

$$= \frac{W}{2} \left[\sqrt{\frac{c^2}{4} + \frac{(e b + c a)^2}{49 a^2}} + \sqrt{\frac{c^2}{4} + \frac{(e b + c a)^2}{49 a^2} + \frac{e^2}{196}} \right] = \frac{d^3}{10} f,$$

and transposing, we have the fiber stress

$$f = \frac{5 W}{d^3} \left[\sqrt{\frac{c^2}{4} + \frac{(e b + c a)^2}{49 a^2}} + \right.$$

$$\sqrt{\frac{c^2}{4} + \frac{(e\,b + c\,a)^2}{49\,a^2} + \frac{e^2}{196}}\Bigg] \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (71)$$

The axle should be large enough at the journal to stand the strains shown by equations 69 and 71.

For the driving axles other than the main, it will be sufficient to determine the strain by formula 71, as they are not likely to receive much load at the beginning of the stroke. The distance b will be much smaller in the other axles, but this will reduce the strain by formula 71 a very small amount. The strain by equation 71 will generally be much less than by equation 69, but it is not usually considered advisable to have too great a difference in the diameter of the different axles.

The main axle is often ½ inch or 1 inch larger in diameter than the remaining axles, and is sometimes made of a stronger material—nickel steel, for instance. The formulæ for axle strains look rather formidable, but the strains are complicated, as was seen by the explanation given with Fig. 56.

As an example, we will examine the main axle of a 4-4-0 type engine, with 19 by 24 inch cylinders, 190 pounds of steam, 40,000 pounds on the main driving wheels (both). The several values are $a = 12$; $b = 21\frac{1}{2}$; $c = 7\frac{3}{4}$; $d = 8$; $d^3 = 512$; $e = 75$; $W = 40{,}000$; $P = 284 \times 190 = 54{,}000$, and $P' = \frac{W\,e}{7\,a}$

$$= \frac{40{,}000 \times 75}{7 \times 12} = 35{,}700.$$

For the fiber strain at the end of stroke, we have from equation 69

$$f = \frac{10}{8^3}\sqrt{1{,}160{,}000^2 + 155{,}000^2}$$

$$= \frac{10}{512}\sqrt{1{,}345{,}600{,}000{,}000 + 24{,}025{,}000{,}000}$$

$$= \frac{10}{512}\,1{,}170{,}000 = 22{,}900 \text{ pounds per square inch.}$$

For the top quarter, by equation 71, we have

$$f = \frac{5 \times 40{,}000}{8^3}\left[\sqrt{\frac{7.75^2}{4} + \frac{(75 \times 21.5 + 7.75 \times 12)^2}{49 \times 12^2}} + \sqrt{\frac{7.75^2}{4} + \frac{(75 \times 21.5 + 7.75 \times 12)^2}{49 \times 12^2} + \frac{75^2}{196}}\right]$$

$$= \frac{5 \times 40{,}000}{512}\left[\sqrt{\frac{60}{4} + \frac{2{,}907{,}025}{7{,}056}} + \sqrt{\frac{60}{4} + \frac{2{,}907{,}025}{7{,}056} + \frac{5{,}625}{196}}\right]$$

$$= 390\,[\sqrt{15 + 412} + \sqrt{15 + 412 + 29}] = 390\,(20.64 + 21.35)$$

$$= 390 \times 42 = 16{,}380 \text{ pounds per square inch.}$$

There is considerable difference of opinion as to the proper stress that should be permitted in an axle, also there are a number of methods for computing the stress, but it is believed that the above methods allow for the various strains as closely as possible; many of the other rules give only approximate results. Mr. L. R. Pomeroy, of the Cambria Steel Company, suggested 18,000 pounds per square inch for iron and 21,000 pounds for steel axles, while Mr. F. J. Cole recommends 10,000 pounds for iron and 13,000 for steel. There does not seem to be any good reason why the safe stresses suggested for crankpins should not be applicable to axles, viz.:

Iron	12,000 pounds per square inch
Steel	15,000 pounds per square inch
Nickel steel	18,000 pounds per square inch

It is always well to consult current practice by making comparisons with modern locomotives which have given success as to the particular detail under consideration.

A comparison of formulæ 69 and 71 shows that for ordinary types of locomotives, the strain in main axle will be greater at end of stroke, or by equation 69, than by equation 71. When the weight on main drivers exceeds ¾ of the total weight on all drivers (as in locomotives with a single pair of drivers only), the strain will be greater by equation 71; but if the weight on main drivers be less than 75 per cent of the total adhesive weight, equation 69 will give the maximum stress.* This does not apply to driving axles other than main, as formula 71 must be used on account of the uncertain load taken by the side rods at the commencement of stroke.

The calculations may be reduced, by solving part of the problem graphically. We will illustrate this method by the last example.

For strain at end of stroke: Multiply together the boiler pressure, piston area and distance of center of main rod bearing from center of journal $= P\,b = 190 \times 284 \times 21\frac{1}{2} =$

---

*The demonstration is as follows: The difference between the bending moments in the development of equations 69 and 71 is between the terms P b and $W\frac{e\,b + c\,a}{7\,a}$, and as we have seen by our example that the strain due to slipping the opposite wheel is small in comparison to $W\frac{e\,b + c\,a}{7\,a}$, it will here be neglected, especially as it appears under only one of the radicals in eq. 71. The maximum ratio of adhesive weight to theoretical tractive force in current engines is probably 3.5, or $\frac{p\,d^2\,s}{D} = \frac{W_t}{3.5}$, using the ordinary symbols. But in eq. 69, $P = .7854\,d^2\,p$, and, substituting in the above, also 2 a for s and e for D, our symbols in eqs. 69 and 71, we obtain $\frac{2\,P\,a}{.7854\,e} = \frac{W_t}{3.5}$ and $P\,b = .112\,W_t\frac{e\,b}{a}$. In the quantity $W\frac{e\,b + c\,a}{7\,a}$, the smallest practical ratio $\frac{e}{a} = \frac{48}{12} = 4$, and as ordinarily $c = \frac{b}{3}$, the second term when written $\frac{W\,e\,b}{7\,a} + \frac{W\,c}{7}$, will bear the largest proportion to the first term when these values are substitued or when the quantity becomes $\frac{4\,W\,b}{7} + \frac{W\,b}{3 \times 7}$, or $\frac{13}{12} \times \frac{4\,W\,b}{7}$, so that the quantity will be maximum at $\frac{13}{12} \times \frac{W\,e\,b}{7\,a} = .155\frac{W\,e\,b}{a}$. Now eqs. 69 and 71 will be equal (neglecting the twisting moment) when $.112\,W_t\frac{e\,b}{a} = .155\,W\frac{e\,b}{a}$, or when $\frac{W}{W_t} = \frac{.112}{.155} = .72$, or, say, when W, the weight on main wheels $= \frac{3}{4}\,W_t$, the total adhesive weight. If the ratio be greater, equation 71 will give the largest stress, but if less, equation 69 will be larger.

1,160,000, and lay it off on a straight line to a suitable scale, say, 100,000 inch pounds to the inch, or 11.6 inches. At right angles to this line, and at one end, lay off one-half the product of the weight on the main drivers (both sides) and the distance from center of rail tread to center of journal, horizontally $= \frac{1}{2}$ W c $= \frac{1}{2} \times 40{,}000 \times 7\frac{3}{4} = 155{,}000$, or 1.55 inches. Measuring the diagonal of these lines, as in Fig. 57, we find it to be 11.7 inches, or 1,170,000 inch pounds. This is to be multiplied by 10 and divided by the diameter of the journal cubed $= 8^3 = 512$, or $1{,}170{,}000 \times 10 \div 512 = 22{,}900$ pounds fiber strain, same as equation 69.

For the top quarter: Add together the product of driver diameter by distance of center of main rod bearing from center of journal $=$ e b $= 75 \times 21\frac{1}{2} = 1{,}612$, and product of distance of center of rail head from center of journal, horizontally,

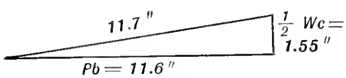


Fig. 57

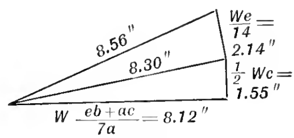


Fig. 58.

by crank radius $=$ c a $= 7\frac{3}{4} \times 12 = 93$; and divide the sum $= 1{,}612 + 93 = 1{,}705$ by 7 times the crank radius $= 7$ a $= 7 \times 12 = 84$, or $1{,}705 \div 84 = 20.3$, and multiply the quotient by the weight on main drivers $= 20.3 \times 40{,}000 = 812{,}000$ inch pounds. Now lay this off on a straight line to same scale as before, or 8.12 inches, and at right angles at one end, lay off the same distance as in Fig. 57, or 1.55 inches. Draw the diagonal, and at one end erect a perpendicular to diagonal equal to the weight on drivers multiplied by the wheel diameter and divided by $14 = \frac{\text{W e}}{14} = 40{,}000 \times 75 \div 14 = 214{,}300$, or, to scale, 2.14 inches. Now measure the two diagonals, add them together to the scale and multiply the sum by 5 and divide by the cube of the diameter of journal, or $8.30 + 8.56 = 16.86$ inches, or, to the scale, 1,686,000 inch pounds, and $1{,}686{,}000 \times 5 \div 512 = 16{,}400$ pounds, as against 16,380 pounds by calculation, the difference being due to scaling the diagonals in sketch 58.

DRIFTING.

We have been studying the effect of steam action while the locomotive is in operation, but the engine often runs with a closed throttle, as when dropping down grades—this is termed "drifting." While we do not admit steam to the cylinders, they are still full of air or vapor, and the effects of expansion and compression will be present. The inertia of the reciprocating parts will also make itself manifest.

It is customary to place the link motion or reverse lever in full gear, when drifting, as this causes less compression, or resistance to the piston, and also diminishes the suction of cinders, etc., from the smokebox into the cylinder. The explanation of this feature is as follows: If the lever be set to give three-quarters (apparent) cut-off, and the pressure in cylinder at commencement be 10 pounds above the atmosphere, we find by formula 48 that the pressure at end of stroke will be

$$p_t = \frac{p\ v}{v_t} = \frac{(10 + 15) \times (.75 + .08)}{1.00 + .08} = 19.2 \text{ pounds above}$$

a vacuum, or say $4\frac{1}{2}$ pounds above the atmosphere. (The expansion of air has a somewhat different coefficient from that of steam, but the results will be close enough for our purpose if we follow Mariotte's law.) Thus, there will be positive pressure in the cylinder at the end of stroke when the valve opens to release, and air (or vapor) will pass from the cylinder into the exhaust pipe. If, however, the lever be near the center of the quadrant, so that the cut-off occurs at quarter stroke, we will have the terminal pressure

$$p_t = \frac{(10 + 15) \times (.25 + .08)}{1.00 + .08} = 7.7 \text{ pounds absolute or about}$$

seven pounds of vacuum, and gases will be "sucked" into the cylinder, when the valve opens for release, from the smokebox. These two examples assume that the cylinder back of the moving piston is replenished with air (or vapor) at 10 pounds pressure, as the piston advances, as long as the valve remains open. This is not exactly a possible condition, as, in advancing, the piston draws in air from the steam chest, of a very limited capacity. If provision were not made to prevent, a very heavy vacuum would quickly be produced in the steam chest, by the exhausting action of the pistons, so that relief valves are applied to open inwardly, and admit air from the atmosphere whenever the pressure in the chest is less than that of the atmosphere. These valves should be so applied that they will fall open by gravity, which prevents their dancing and beating themselves to pieces when the engine is drifting, and as soon as the throttle is opened, steam closes these valves and holds them shut. It is important that they be made large enough to admit air freely, otherwise at high speeds they may not prevent the sucking in of smokebox gases. If the air were admitted freely, so that the piston was followed by atmospheric pressure clear to the point of cut-off, we should have for the terminal pressure in the two cases just considered

$$p_t = \frac{15 \times .83}{1.08} = 11.5 \text{ pounds absolute at } \tfrac{3}{4} \text{ cut-off, and}$$

$$p_t = \frac{15 \times .33}{1.08} = 4.6 \text{ pounds absolute at } \tfrac{1}{4} \text{ cut-off.}$$

Fig. 59 is a diagram taken from the low pressure cylinder of a compound locomotive while drifting, and the drop of the admission line below the atmospheric line shows that even with

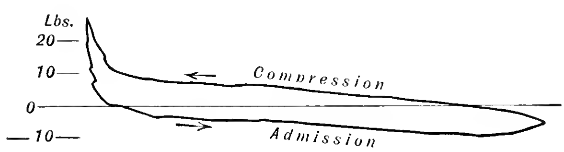


Fig. 59

relief valves nearly 3 inches in diameter, there was considerable "suction." This diagram was taken during a test made to discover the cause of soot from the smokebox being found in the low pressure cylinder. The compression is also considerable, although starting from a partial vacuum, as is evident in

Fig. 59. This is also reduced by keeping the lever near or at the full gear position, and the amount of compression can be determined by equation 48 or plate 11, if we consider it to start at atmospheric pressure. For instance, in the Stephenson valve gear, which we previously considered, it was found that when the reverse lever was in the corner, the port closed when the piston had but 2 or 3 per cent of its stroke to complete, whereas, when cut back to a cut-off of 20 per cent, the compression began with one-third of the stroke incompleted, the port opening for admission when 9 per cent was still incompleted. Plate 11 demonstrates that while, in the first case, the final pressure due to compression would amount to but four or six pounds by the gauge, in the second case, it would be 20 pounds when the valve uncovered the port, and this would then be forced into the steam chest. Without this pre-opening, it would run up to 60 pounds per square inch. If the clearance were less than 8 per cent, on which amount plate 11 is based, the pressure at end of stroke would be still greater. This is especially true in compound locomotives, where the clearance is sometimes 5 or 6 per cent of the low pressure cylinder, and, the piston being large, the total back pressure is quite great. In ordinary slide valves, the valve will lift and allow the pressure to pass into the steam chest, whence it will be withdrawn on the next stroke. With piston valves this is not possible, and recourse to special by-pass valves is had. These are generally designed so as to reduce the vacuum on one side of the piston as well as the compression on the other side. The committee of the Master Mechanics' Association on piston valves described a device used on the Southern Pacific designed by Mr. P. Sheedy, and a description will be found in the proceedings of that association for 1903, on page 309. Ordinary "safety valves" have been placed in the cylinder heads, to relieve any water or excess pressure that might appear in the cylinder, but the committee states that it has been the experience of some that these valves, after being in service for a short time, from corrosion or other causes, fail to lift at the pressure for which they are set, thus becoming useless. In any event, they cannot open until a pressure greater than the normal steam pressure

is produced, otherwise they would blow continually when using steam. The Sheedy arrangement is designed to overcome these difficulties. The device consists of a circulating pipe connecting the opposite ends of the cylinder, governed by valves which seat by steam pressure when the throttle is open and close communication through the pipe. When the throttle is closed, a spring lifts the valves and establishes communication between the two ends of the cylinder through the pipe, allowing the air or vapor to pass backward and forward through it, without undergoing expansion or compression. A safety valve

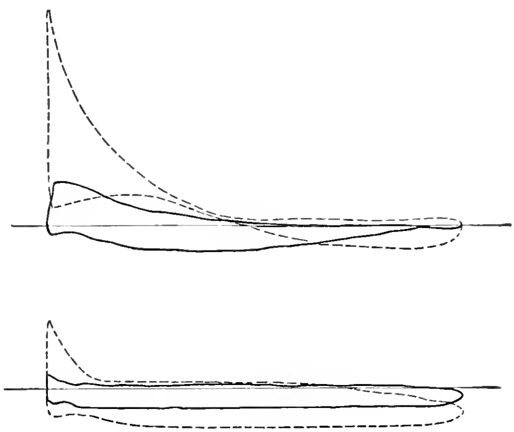


Fig. 60.

in addition provides relief for water or excess pressure while using steam. Fig. 60 presents diagrams taken from an engine with the circulating device in operation and also when cut out. The dotted lines show the card taken when drifting with the valves closed, and the solid line with the apparatus in operation. The upper cards were taken with 10 inches of cut-off, and the lower with 22 inches of cut-off, and all at about 42 miles per hour. They show a relief of about 80 per cent of terminal pressure by the action of the circulating pipe.

A simpler form of valve devised by the author, and used by many of the important roads in this country, is illustrated in Fig. 61. This shows the latest arrangement as improved by the American Locomotive Company and as applied to piston-valve engines of their manufacture. The opening "a" connects with the steam passage, between the valve chamber and the saddle, and "b" connects with the steam port, between the valve chamber and the cylinder. When using steam, the valve is closed by the pressure from "a," unless an excessive pressure or water appears in the cylinder, when this pressure, passing through "b," forces the valve open and gives relief into the steam passage. When drifting, the valve opens by gravity,

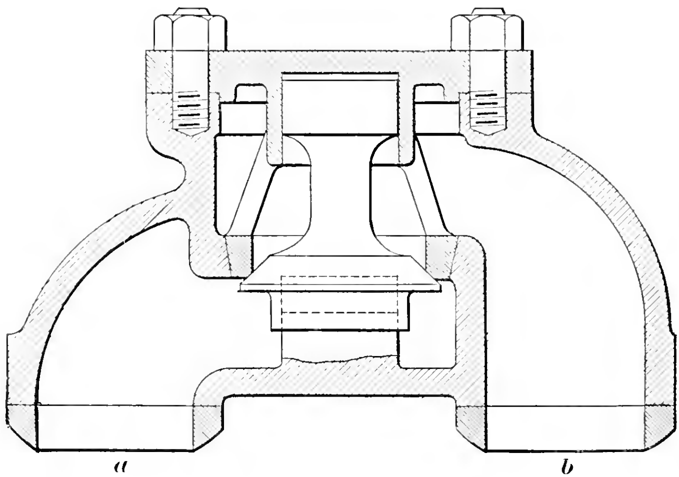


Fig. 61.

permitting free communication between the end of the cylinder and the steam passage, and hence between both ends of the cylinder. It has been noticed that engines having these valves draw in much less air through the relief valves in the steam passage than those without them, demonstrating the relief action of the by-pass valves. When by-pass valves are used, it is sometimes considered preferable to place the lever near the mid-gear position in drifting down long hills, as this reduces the valve travel and consequently the work done by the eccentrics, and also equalizes the work done between the forward and backward eccentrics, thus relieving the eccentrics and straps of a portion of the strain and wear. As the air or vapor circulates from end to end of cylinder, the chilling effect of cold air drawn in through the relief valve will be obviated, also the fanning of the fire by the exhaust.

Both the expansion and compression done in drifting absorb work, and retard the motion of the engine. In descending a grade this assists the brake, and even constitutes a very efficient brake, if the details are properly arranged for this purpose. This will be discussed under the head of "braking." Often, however, the grade is slight, and the compression and expansion are a detriment to the speed which is desired. This is especially true of compound engines, in which the great area of the low pressure cylinder offers so much resistance to the vacuum formed that it is found necessary often to open the throttle, and supply enough steam to overcome the vacuum which would be formed, in spite of a number of large and various kinds of relief valves. In fact, the braking action is such that the Westinghouse Air Brake Company recommend and use a lower coefficient of brake power for certain compounds than for simple locomotives. On some very long grades which the author has in mind, upon the great continental divide, it was the general opinion that as much steam was consumed by the compounds in descending, as was saved by them in ascending. Sometimes an undue amount of compression causes the piston and rod to be heated to a very high temperature, by the compression of the air, which, as we know, liberates quite a large amount of heat. It is advisable to take indicator cards from engines when they are drifting, in order to determine whether there is an improper amount of useless work done in the cylinders.

In the tables calculated in connection with the determination of the rotative force, we have seen that, at the commencement of the stroke, when the speed in miles per hour equals the diameter of the drivers in inches, the inertia of the reciprocating parts may be great enough to wholly neutralize the steam pressure, and inversely at the end of stroke, the compression may be high enough to balance the inertia of those parts, which causes the lost motion to be taken up gradually as the piston approaches the end of the stroke, and the cushion so formed prevents the pounding of these parts. Too much compression will, on the other hand, cause a pound of itself, which is extremely hard on the rods and allied parts. This has often been blamed for the breakage of crankpins. We found from the tables to which reference is made that in the engine there considered the force of inertia at 80 miles an hour would amount to about 140 pounds per square inch of piston area at the end of stroke. We could never realize this amount when drifting, unless actually using the cylinders as air pumps by reversing the link motion, but such compression as was developed would help to restrain the pounding of the rods. The amount of compression depends, as we have seen, upon the position of the reverse lever, which also commands the cut-off, and more compression means also more expansion, and the consequent suction of the smokebox gases into the cylinders.

At 40 miles an hour, the forces of inertia are one-fourth of the amount at 80 miles, or equivalent to about 35 pounds per square inch of piston area, but even to obtain this amount, the reverse lever must be maintained quite near the middle of the quadrant.

We see, therefore, that, under ordinary conditions, while the compression during drifting will assist in softening or reducing the pound of the rods, it cannot possibly be expected to completely overcome it, but that when using steam it will generally take up the lost motion by the time the piston has reached the end of its stroke.

CHAPTER III.

RESISTANCE.

Opposing the force of steam, which we have just considered, are the resistance of the locomotive and the resistance of the train which it draws. The power is always equal to the resistance as a total, and although some forms of resistance may be obscure and difficult of analysis, nevertheless, they all go to make up the total against which the engine is working. The main resistance is usually that caused by the train which is being drawn—we say usually, as times and conditions may occur in which more than one-half of the indicated power of the cylinders is utilized in moving the engine and its tender. Most of the resistances against which the locomotive operates are caused by friction, but a few, such as those of gravity and head wind, are independent of it. Frictional resistances manifest themselves in every part of the engine and train that moves —at the flange of the wheel, in the journal bearing, upon the guides, in the cylinders and steam chests and throughout the link motion, as well as upon the center plates and side bearings in curving. The friction of the driving wheels upon the rails is the only form in which the resistance is positively useful—so useful, that different mechanical means have been introduced in order to increase this friction, which constitutes a resistance to slipping. The friction of the brakeshoe against the wheel is also useful in stopping the train, but is of no benefit while running. As these different resistances are of great importance in studying the subject of locomotive operation, they will be discussed separately. Inertia (which we have already examined) acts as a resistance in so far as it absorbs power at increasing speeds, but as an equal amount of work is performed by inertia when the speed is being reduced, it cannot be considered strictly as a resistance.

RAIL FRICTION OR ADHESION.

The friction of the wheel upon the rail, which limits the power of a locomotive of a given weight on drivers, has been the subject of much investigation. Perhaps the most complete experiments were made by Capt. Douglas Galton, in 1878, upon the London Brighton & South Coast Railway. The coefficient of friction was found to be very different, if the wheel ceased to revolve, and slid upon the rail. In fact, the friction changed from static to dynamic, that is, the friction of surfaces which are relatively at rest, or moving at very slow speeds is much greater than when sliding upon each other at a considerable rate. This must not be confounded with the speed of revolution of the wheel, for as long as it revolves without slipping, the friction on the rail is static, because the surfaces roll, but do not slide upon each other. But let the wheel commence to slide upon the rail, or slip, that is, revolve without advancing, and the friction at once reduces. This is familiarly exhibited when starting a highly powered locomotive with a wide throttle opening. As soon as the wheels start to slip, they spin with great violence, until the throttle be closed, or until sand be delivered to them. Even a reduction of throttle opening is not always sufficient to stop the slippage; it must be completely closed. This is forcible evidence that, when once started, the slipping of the wheels reduces the friction which normally holds them to the rail.

The following table gives these coefficients as determined by Captain Galton:

DYNAMIC FRICTION BETWEEN WHEEL AND RAIL

(Both Being Steel).

Miles per Hour.	Feet per Second.	Coefficient of Friction.
Just coming to rest	..	.242
6.8	10	.088
13.6	20	.072
27.3	40	.070
34.1	50	.065
40.9	60	.057
47.7	70	.040
54.5	80	.038

While the friction is reduced by slipping, it is greatly increased by the judicious use of sand, in fact, it was found in some tests, to approach 40 per cent. In the case of damp or greasy rails, it is reduced. When the rails are thoroughly wet, as in a heavy rain, the adhesion is usually considered as good as when dry.

Mr. A. M. Wellington, in his "Railway Location," sums up the situation thus:

"The coefficient of static friction between rail and wheel is not sensibly affected by the velocity of motion (that is, rolling motion).

"It is very greatly affected by the insistent weight, increasing rapidly therewith.

"It is very greatly affected by the condition of the surfaces as respects moisture or other equivalent for a lubricant, and the effect is rarely twice alike.

"It is greatest when the rails are either very dry or very wet, moisture or frost having the most injurious effect.

"The coefficient of dynamic or sliding friction is very greatly less than static friction, and very greatly affected by velocity, in inverse ratio thereto."

He then gives a statement, which is briefly as follows:

Ultimate coefficient of friction under very favorable conditions, and with loads exceeding 10,000 pounds per wheel	.35
Working coefficient, with sand	.33
Working coefficient in summer, and maximum limit with loads of less than 10,000 pounds per wheel	.25
Working coefficient in winter (damp or frosty rail)	.20

Judging from the experiments made in obtaining the friction of brakeshoes, it is probable that the coefficient will also vary with the diameter of the wheel and the load upon the wheel, or the pressure upon the rail. If, with the same weight, the diameter of the wheels be enlarged, the area of contact of the wheel with the rail will be increased, thus reducing the pressure per unit of surface and we should expect that the coefficient would be greater, thus increasing the total friction. Ordinarily, also, an increase in load would mean that the pressure per unit of contact surfaces would be increased, and the coefficient of friction would naturally be expected to drop. There are many points in connection with this subject which are still undetermined, so that we must be governed by the dictations of current practice. Wellington states that the ratio of adhesion or the coefficient of friction assumed by foreign railroad officials, is considerably less than what is allowed in this country. Thus, for the ultimate coefficient at slow speeds, they would use .25, where we would adopt .33, and for the working coefficient .20, instead of .25, as with us.

In 1887 a committee of the Master Mechanics' Association recommended the following ratios of tractive force to weight on drivers:

Passenger engines	.25
Freight engines	.23½
Switching engines	.22

The tractive force was to be figured on tires half worn.

In 1898, in a report to the same association, another committee stated that in view of the excellence of pneumatic sanding arrangements now placed upon locomotives, the friction between wheel and rail could be considered at 25 per cent, and when such sanders were not used, 21 per cent. In winter it is likely that these values should be reduced about 10 per cent. This coefficient .25 is taken as the actual working value of the friction, against which is opposed the tractive force of the engine, with the internal resistance or friction of the machinery deducted. This also corresponds to the average actual rotative force, and not the maximum rotative force, which we have seen, is about 20 per cent greater than the average in full gear. This would assume a coefficient of $1.20 \times .25 = .30$ to prevent slipping at the points of maximum rotative force. If we consider 25 per cent as the normal frictional resistance or adhesion of road engines, and 22 per cent for switching engines, we will be within the safe limits of current railroad practice. It must not be assumed that slipping will never occur with these ratios, as there are so many varying track conditions that the most stable engines will sometimes slip, and particularly if not properly handled, but with a careful man at the throttle, good results can be obtained. Large cylinders are generally desirable in order to obtain a great tractive effort at high speeds, and even if the engine must be started carefully, and with sand, it is advisable to have the cylinders large. Taking all the information together, it seems as if Wellington's values of .35 for ultimate adhesion, under most favorable conditions, and .25 for the ordinary working coefficient will be quite fair figures upon which to base our conclusions.

TIRE WEAR.

Tire wear is produced by four distinct causes:

Rolling abrasion and flow of the metal, due to the continuous cold rolling process which the tire undergoes.

Slipping abrasion, which is caused by the spinning of the wheels in starting, and which may not only wear, but loosen the tires, if carried to excess.

Flange wear, which is caused by the pressure of the flange of the tire against the rail in curving.

Brake wear, which is caused by the rubbing of the brake-shoes against the tire, when producing a stop, and sometimes, when released.

All these unite to wear the tires of the locomotive, and as worn treads are very hard on frogs and crossings, and worn flanges are dangerous to the operation of the engine, it is highly important that the wear be kept to or near the minimum limit, even regardless of the question of cost and delay to the engine while returning or replacing the worn tires.

There are differences in the amount of tire wear, even with the same make of tires and upon engines of the same class, and the manner in which the locomotive is handled is of the greatest importance. A committee of the Master Mechanics' Association, reporting on the wear of tires in 1887, said that "the locomotive engineer has a great deal to do with the wear of tires by judicious manipulation of the sand and exercising proper care in starting, in avoiding slipping." This committee also stated that in their opinion the slipping was worse than sand, in wearing the tire. A case, in fact, was cited wherein one engine, operated by careless men, who slipped the wheels and also used sand freely, made 12,000 miles to 1-16-inch wear of tire, whereas a second engine of the same type, in the hands of a careful man who avoided sand and slipping, made 23,000 miles to 1-16-inch wear, in both cases the tires being made of Krupp steel.

Another report, made in 1888, showed mileage varying from 5,000 to 25,000 miles per 1-16-inch wear for road engines, and for switching locomotives from 3,000 to 6,000 miles. Of course, the reported mileage of switch engines is practically never accurate, but recent investigations indicate that the generally allowed speed of six miles per hour in service is much too great. Switching locomotives usually are roughly handled and are often subjected to considerable slippage and treated to generous doses of sand, besides working on greasy and poorly maintained tracks. Even taking these facts into consideration, we are hardly prepared for such a great reduction in the mileage per 1-16-inch wear.

In 1889 the results of 252 sets of tires used on the Illinois Central were reported as giving 11,500 miles per 1-16-inch wear, and another batch of 33 sets as giving only 8,000 miles for the same wear. The first lot cost 7¾ cents per pound and the latter 5¼ cents. Of course, these prices are now very different, but the statement is quoted as indicating the variation in results depending upon the quality of the material used in making the tire.

Again, in 1895, a number of 4—4—0 engines were mentioned as in use on the Northern Pacific Railroad, all having driving wheels 69 inches in diameter, part of the locomotives having 10,000 pounds load per wheel and part 18,000 pounds. The mileage to 1-16-inch wear amounted to 15,000 for the average of the lighter engines, and only 6,700 for the heavier machines.

In view of the greater wear of tires upon switching engines, the steel is generally made harder for this service: thus, while in passenger engine tires the tensile strength of the steel is about 100,000 pounds per square inch, with an elongation of 12 per cent in 2 inches, in switchers it should be 120,000 pounds per square inch and 8 per cent elongation.

All these facts which we have mentioned indicate that the greatest wear of the tire is caused by slipping abrasion, although we know, as in the case with truck and car wheels, that there is some tread wear, due to ordinary rolling. We do not here consider the case of slid wheels, as these are the result of extremely careless manipulation of the engine, and are no more a factor of proper locomotive operation than are overheated crown sheets. The merest tyro must know that this should never be permitted. Sometimes, however, capable engineers are caught, particularly when using the water brake or reversing the engine, when the air brake is holding.

The ordinary motion of the engine produces a "cold rolling" of the tires, and when the mileage is large, may in time cause them to become loose upon the wheel center. As the tires wear they must be re-turned, which gradually reduces the thickness until a point is reached beyond which it is unsafe to go. One large transcontinental line has adopted the following minimum limits for thickness of driving wheel tires:

Service.	Shop Limit.	Road Limit.
Passenger (20,000 pounds per wheel or over)	1¾"	1½"
Passenger (less than 20,000 pounds per wheel)	1½"	1¼"
Freight	1½"	1¼"
Switching	1¼"	1"

The "shop limit" is the minimum thickness to which they should be turned, and the "road limit" the minimum thickness that should be allowed to continue in service. There are also wear limits assigned, for hollow or worn treads and for flanges. If the tread exceeds ¼ inch at any point (ordinarily near the flange) lower than the outer part of tread, the wheel is considered destructive to frogs and crossings, and turning is necessary—so also if the flange become unduly high by wear of the tread, say, to 1½ inches, it is necessary to reduce it in order to prevent its riding upon the filling blocks in frogs and crossings. In order to prevent these contingencies and increase the time or mileage between turnings, brakeshoes are provided which bear only upon the flange and outside of tire, not wearing the part normally running upon the rail. These are sometimes called "tire dressing shoes," and for this purpose frequently contain hard metal inserts, the object being to reduce the tire uniformly, the shoe wearing away the parts not touched by the rail. While perhaps there is not a great deal of increase in the mileage per 1-16-inch wear, yet there is an increase in mileage between turnings, which keeps the engine out of the shop and reduces the cost of repairs. These shoes are seldom applied to truck or tender wheels, as the tread wear is ordinarily small, flange wear being the chief destructive agent, and which would be increased by such shoes. Thus it will appear that the wear of tires by the brakeshoes can be made use of to increase the mileage between turnings, and as some good metal is removed every time a tire is turned, the mileage per unit of tire thickness may even be enlarged, which at first sight seems like a paradox.

Flange wear, the third on our list, and the only one not yet considered, is more dreaded and carefully watched (or should be) than either of the other three forms. As intimated above, it is the chief cause of trouble in truck and tender wheels. This is readily accounted for when it is remembered that no power is applied to these wheels (except in braking) and the treads have only to encounter rolling abrasion, with the exception of slipping of one or both wheels on an axle when traversing a curve. The flange, however, comes in contact with the side of rail head when passing through a curve, the force depending largely upon the degree of curvature, and the rotation of the wheel at the same time, produces a very heavy friction and wear. If we wish to gather some idea of this friction and abrasion, we have only to examine the inside head of the outside rail, where it has been in use for some time, and compare it with that on straight track. A. M. Wellington gives the result of examination of rails on curves on the New York Pennsylvania & Ohio Railroad. The outside rail on a ½-degree curve lost 2.78 pounds per yard after being traversed by 24,000,000 tons, whereas on a 16-degree curve, the loss was 7.80 pounds per yard with a traffic of only 6,000,000 tons—one-fourth as much. Nearly all of this wear was on the side of the head, which was ground down to approximately the shape of the wheel flanges. The wear on the inside rail was 1.88 and 1.45 pounds per yard, respectively, and was due to the slipping of the wheel, as the tendency is to throw more friction upon the outer wheel, so that the inner wheel must slip by an amount equal to the difference in the lengths of the outside and inside rails. The inside rails were worn entirely upon the top. This will explain why, on crooked roads, the flange wear of truck wheels is so much greater than the tread wear, even where brakes are applied to these wheels, for while the tread wears, the flange wears faster.

Flange wear is serious from two points: If the flange wears vertical, the wheel may climb a defective joint or open a split switch; when turning the tire to obtain a normal flange, it is

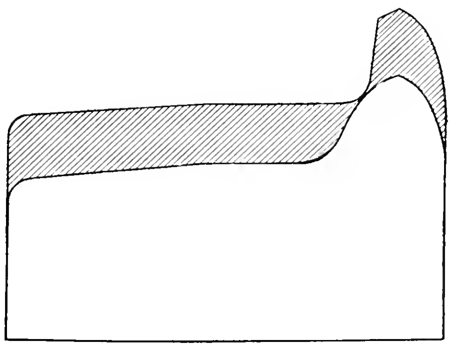


Fig. 62.

necessary to remove a large quantity of valuable wearing metal. These two counts mean danger and expense. The Master Car Builders' rules of interchange reject a wheel whose flange has a flat vertical surface extending more than 1 inch from the tread for cars of 80,000 pounds capacity or less, and 7/8 inch for larger cars. In the leading truck wheels of a locomotive it is even more important, as the security of the whole train depends upon these flanges. The great waste of metal is shown in Fig. 62. The upper line indicates a bad case of sharp or worn flange, and the lower line the standard tread. The shaded portion shows the amount of good metal that must be turned off (and wasted) in order to produce the standard flange.

Sometimes this flange cutting can be reduced by turning the truck end for end, or at least it may by this means be distributed among the other wheels. Often, however, it is due to a variation in the diameter of the two wheels upon the axle, which causes a continual tendency to roll in a circular path, crowding the flanges hard against the rails. In such cases, the obvious remedy is to put the wheels in a lathe and reduce the larger to the size of the smaller wheel. If wheels are properly mated for size, but the chemical composition and the wearing qualities are different, the softer wheel will wear the most rapidly, and as soon as it becomes smaller, the diminished moment will cause it to slip continually, thus increasing the discrepancy very rapidly. Steel tired wheel manufacturers ordinarily mark the wheels so that they can be mated, not only for size, but also for similarity of metal; that is, from the same heat. The importance of observing this in mounting should not be underestimated. In connection with Fig. 62, when it is remembered that the mate wheel must also be turned to the same diameter, even though its flange may not be worn, the necessity for inspection and action is made clear.

If the flange wear on truck wheels is important, it is more so on driving wheels. Even on a question of expense alone, we can recognize this fact when we consider that all the drivers must be turned to the size of the smallest, whether their flanges be all worn or only one. The drivers being so much larger, the tendency to climb a bad joint or a switch or frog point is greater in proportion, and troublesome derailments can be traced to driving wheels with sharp flanges, particularly when on the forward wheels. In order to save the drivers, the truck must do the principal part of the guiding, and if these wheels wear, they are much more easily and cheaply replaced than the drivers. The latter, however, can do a portion of the work without much detriment, especially if properly arranged. A few years ago it was customary with long rigid wheel base engines, such as those of the 2—8—0 type, to use flanges on the front and rear drivers, and supply the second and third wheels with bald or plain tires. This practice has now been quite generally abandoned, and modern locomotives are provided with flanged tires on all wheels, those of the front and rear being slightly closer together than the middle ones, say ⅛ or ¼ inch.

The guiding action of the front truck constituted a subject of experimental inquiry by the "Big 4" railroad in 1897 (see Proceedings of Master Mechanics' Association). A mogul or 2—6—0 engine with rigid wheel base 15 feet 6 inches long and total wheel base 23 feet 2 inches, the radius of truck being 5 feet 4½ inches, was used for this purpose. The adhesive weight was 73,000 pounds, with 16,500 pounds on the truck. The swing hangers were made adjustable and two lengths (6⅝ and 8 inches) and a number of angles were experimented with. The piece of track selected for the work contained a 3-degree curve with 4 inches elevation, and an average speed of 32 miles per hour was maintained. An apparatus for measuring the flange stress of the truck wheels was mounted upon the engine. The hangers were suspended with inclined angles, outclined angles and parallel. The 6⅝-inch hangers gave the lowest flange pressure when inclined 18 degrees, the top centers being closer than the bottom centers. The pressure horizontally in this case was 1,560 pounds. With the same hangers vertical or parallel, the pressure was 2,550 pounds, and when they were outclined by 12 degrees, or closer at the bottom than at the top, the pressure was 2,320 pounds. With the 8-inch hanger, the lowest stress was at 28 degrees inclined angles, or 2,850 pounds; 3,640 vertical and 3,150 outclined 12 degrees. A rigid truck gave 3,230 pounds. From this it appeared that the shorter hangers, inclined about 18 degrees, caused less truck flange friction, and guided the engine more easily through the curve. What friction was caused by the driving wheel flanges is not known.

At the present time, a large number of roads are using the 3-center or "heart shaped" hanger, as it is sometimes called. This permits a parallel motion, and at the same time produces quite a horizontal pull to return the engine (or truck) to a central position. In Fig. 63 is shown the four common arrangements, a being with hangers parallel, b inclined, c outclined and d the 3-center hanger. The hangers are all laid off 8 inches long, and the broken lines show the effect of 2 inches horizontal displacement. The horizontal pull can be judged by the angle which the center line of the hanger makes with the

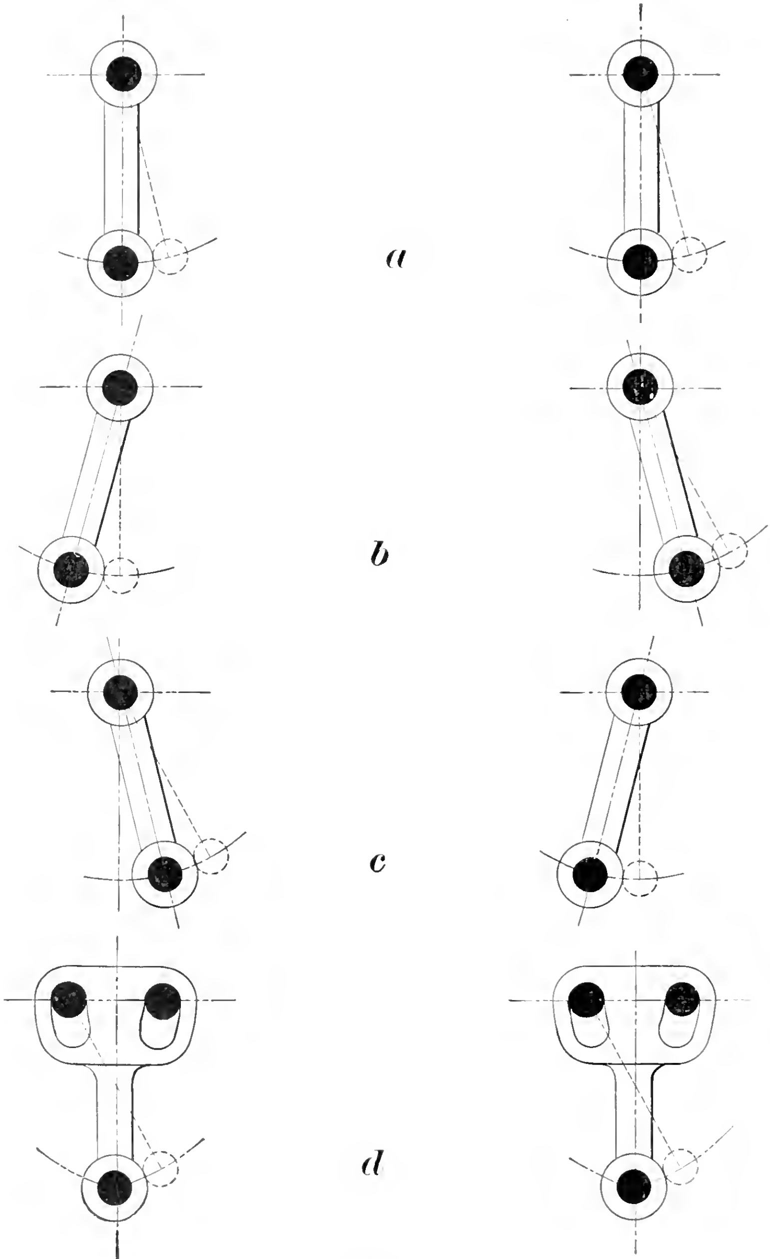


Fig. 63.

vertical. In "a" this pull is the same in both hangers, but is quite small, showing that the guiding power of the truck is insufficient; in b and c, it is confined almost entirely to one of the hangers; in d, however, both hangers exert a similar force, which can be made quite large by spreading the top centers, yet the motion is at all times parallel. As any swing raises the front of the engine at a rapid rate, this arrangement is much more stable than either of the others, and prevents an undue amount of "nosing" or side swinging when running at high speeds. The Chicago Burlington & Quincy Railroad, which operates a large number of mogul or 2—6—0 type engines in high speed passenger service, uses a hanger 8 inches vertically between centers of pins, and 5 inches horizontally between centers of the top pins. The hangers themselves are 23 inches apart center to center, crosswise of the track. In other cases, the central distance of the top pins is only 3 or 3½ inches.

Opinions differ as to the best particular adjustment that can be obtained. If the leading driving wheels wear their flanges sharp, it is evident that the truck is not doing its share of the work in curving; if the truck wheels sharpen their flanges, they do too much guiding, but this is easier to care for than the wear of flanges on the drivers. Sometimes we find the center pin of the front 4-wheel truck 8 or 10 inches back of the center of the truck wheel base, in order to relieve the front truck wheels of some of the wear. Often the flange wear of front drivers can be decreased, by compelling the second pair to do a portion of the guiding. This is done by setting the front driver tires ⅛ to ¼ inch closer together, as has been already referred to. As an example, some 2—8—0 engines on the Lake Shore & Michigan Southern had the tires of the first and last drivers placed 53¼ inches between the backs of flanges, and the second and third pairs of tires were 53½ inches between flanges. The Santa Fe, which has a great many curves, has used spacing of 53⅛ for the first drivers, 53¼ for the second, 53⅜ for the third and 53¼ for the rear wheels. The Lackawanna Road, which is very crooked where it passes over the mountains in Pennsylvania, has used a somewhat different application of this principle; that is, the flanges of the front driving wheels are turned thinner than the other wheels. This does not seem like as good a method as that of placing the tires closer together just described, as it removes useful metal from the tires, but it does give a standard guard rail clearance. The tires actually wear thicker at the flange throat, as was demonstrated by a 10-wheel engine that made 72,000 miles after having tires turned so as to allow ⅞-inch lateral play on front wheels, 11-16 inch on main and ½ inch on rear wheels. When finally measured, after the mileage stated, the play of the front and main wheels was found to have decreased 1-16 inch, with no change on the rear wheel. This was brought about by the wear on the tread actually adding to the stock of the flange. The play of the driving boxes was 3-16 inch total, and in some cases a mileage of 150,000 has been made between turnings. The standard allowance of flange play is as follows:

Type of engine.	Truck.	Drivers. 1st.	2d.	3d.	4th.
4—4—0	½"	⅞"	½"	...	...
2—6—0	⅜"	⅞"	⅝"	½"	...
2—8—0	⅜"	⅞"	⅝"	⅝"	⅞"

Total lateral play at driving box hubs = 3-16 inch.

ROLLING FRICTION.

That rolling friction exerts an appreciable resistance to the action of steam in the cylinders there can be no doubt, but it must be very small in comparison with the other resistances which we have to consider. By rolling friction, we mean the power which it would require to maintain a uniform speed on a straight and level track, with a pair of wheels and axle weighing as much as the total load which they transmit to the rails. This may be considered a strange hypothesis, but it is the only one which can be made in order to convey the strict meaning of rolling friction, for in such a case there would be no journals or moving machinery to create frictional resistance, but simply the rolling of the treads of the wheels upon the rails. If we place a cylindrical body upon a plane surface and gradually elevate one end until the body commences to roll, we can express the static coefficient of rolling friction by the tangent of the angle which the plane makes with the horizontal. Thus in Fig. 64, if the weight of the body be represented by the line a b, the force tending to roll it down the incline will be represented by b c, and the normal pressure upon the plane by a c. The coefficient of rolling will therefore be $\frac{b\ c}{a\ c}$, and as the angle b a c $= \Theta$, the coefficient $= \tan \Theta$. This Θ would be commonly called the angle of static rolling friction, and the simplest experiment will demonstrate to us its minute value.

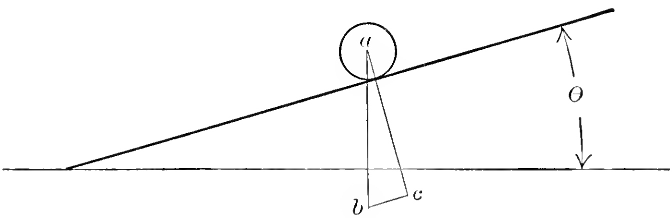


Fig. 64.

Still less will be the angle of elevation which is only necessary to maintain the rolling when once started, and it is the latter that would be the angle of dynamic rolling friction.

In railway equipments, however, we always encounter journal friction (if nothing greater), in addition to the rolling friction, and it is not customary to attempt to separate the two —nor is it necessary to do so. The various resistances are generally taken together, and given a value which represents the entire resistance to motion on a straight, level track, at a uniform velocity. Some experimenters have separated the wind resistance, both as to its action upon the head and also the side of the train, and the flange friction due to oscillation, etc., from the journal friction, but the latter is considered to cover the simple rolling friction in each case which has come to the attention of the author.

JOURNAL FRICTION.

The friction of journals depends upon the pressure per unit of surface, the composition and condition of the surfaces in contact, and the lubricant and its application. Many are the varieties of bearing metals placed upon the market, each laying claim to reduced friction, and consequent wear on journals and bearings. Many roads have for years, and still consider phosphor bronze the most suitable metal for driving box shells. This metal is composed of about 79½ per cent copper, 10 per cent tin, 9½ per cent lead and 1 per cent phosphorus, the latter ingredient being introduced principally in order to make the metal flow readily and produce sound castings. As a general proposition, the harder the metals in contact, the lower is the coefficient of friction; at the same time it should be remembered that the hard metals are more rigid, and thus likely to cause a concentration of load upon a small amount of surface, excluding the lubricant, where a bearing that has a certain amount of plasticity will distribute the pressure over a larger area. No matter how carefully the surfaces of the journal and bearing are finished, there will still be irregularities, which, with the lateral wear, are liable to increase, and the shifting continually taking place while running, often produces a hollow bearing, with high ridges or parts upon which the load concentrates. Under such circumstances, a bearing which adjusts itself to the journal may produce better results. The driving box brasses are subjected to very heavy and variable loads, due to the thrust of the main rod at the commencement of its stroke, and yet even these have given excellent results with a lining of magnolia metal or other lead-antimony mixtures. Recently the Ajax Metal Company have introduced a bronze composed of 64 per cent copper, 5 per cent tin, 30 per cent lead and 1 per cent nickel. While this mixture does not show any marked diminution of friction over the hard bronze, yet the wear is found by test to be only about one-third as great, and the journal runs at a lower temperature.

A very common practice is to place white metal spots or strips in the brass shell of a driving box, or as a complete lining in truck boxes. Often this metal is composed of one part antimony to four lead. The "spots" have the advantage that the brass shell is not weakened as it is by the strip of white metal, which requires a longitudinal pocket, frequently causing breakage.

The friction of the various bronzes, if thoroughly treated, would form a volume of itself, and is not the object of this treatise. The condition of the weather affects the result, the coefficient being higher in winter than in summer. The kind of lubricant also causes a difference in the friction, and especially the method of applying the lubricant. Wellington refers to numerous tests made which indicated that a bath of oil was much superior to a syphon or pad—the friction being only about one-sixth as great. The ordinary friction obtained by experimenting with cars was found at slow speeds to be .09 to .12 of the pressure for loads of from 30 to 280 pounds per square inch. These speeds were very small, and as they increased, the coefficient dropped to .02 or .03 as the velocity reached 5 miles an hour, with wheels nine or ten times as large

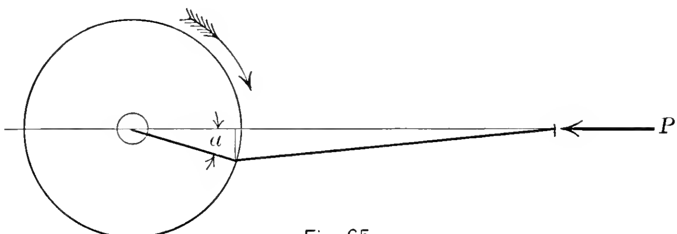


Fig. 65.

as the diameter of the journal, the lower value obtaining with loads of about 300 pounds per square inch of projected journal area, and the higher with 150 pounds. Mr. J. A. F. Aspinall, in a recent paper on "Train Resistance," gives the coefficient of friction of oil lubricated axle boxes at .018 and grease lubricated boxes at .032; also engine axle boxes, with oil lubrication at speeds from 15 to 20 miles per hour at .052. Other reports show still lower values for car journal friction. As stated above, there are so many existing conditions to affect the coefficient of friction, that we cannot attempt to assign a definite value for any case. If we assume .02 for truck journals and .05 for driving axles, we will probably not be far from the actual values.

The friction upon driving axle journals plays quite an important part upon the rotative moment exerted by the connecting rod. It is evident that for a small angular displacement of the crank from the dead center, the rotative moment will be absorbed in overcoming the friction of the journal.

In Fig. 65 consider the crank to have just passed the dead center by the angle a, moving in the direction of the arrow. As the angle of the connecting rod with the center line of the engine is small, it may be neglected, and we can write the rotative moment = P r sin a,

Where P = the total piston pressure,

r = the crank radius.

If we let s = the stroke in inches,

d = the diameter of journal in inches,

f = the coefficient of friction,

the frictional moment of resistance will be

$$P f \frac{d}{2}$$

and equating these values, we can determine the angle at which the rotative moment just equals the friction moment.

$$P r \sin a = \frac{P s \sin a}{2} = \frac{P f d}{2} \text{ and}$$

$$s \sin a = f d \text{ or } \sin a = \frac{f d}{s} \quad \text{.........................} \quad (72)$$

For smaller angles, the friction will give the greatest moment, and the cylinder on the other side will pull the wheels around until the angle of the crank reaches the value given by equation 72, after which the moment of the steam pressure will be in excess. If we assume a stroke of 30 inches with a 10-inch journal, we have

$$\sin a = \frac{.05 \times 10}{30} = .017 \text{ or "a"} = 1 \text{ degree.}$$

As the crank pins would also create frictional moments of approximately the same value, or slightly greater, the actual angle would be about 2 degrees.

The unit pressure is of much importance, as if it be excessive, the lubricant is not able to get between the surfaces in contact, and imperfect lubrication results. A number of modern locomotives had this feature examined, and it was found that a very considerable discrepancy existed. In each case, the projected area of the journal (diameter by length) was divided into the weight carried by the wheel, and the quotient taken as the unit load, being in pounds per square inch. As a matter of fact, the weight of the wheel and one-half of the axle should be deducted for accuracy, but the method adopted admits of ready comparisons.

For the front truck, the unit loads varied from 105 to 165 pounds per square inch, and occasionally (though seldom) reached 250 pounds. A safe figure seems to be 150 pounds per square inch. There did not seem to be any greater unit pressure with freight than upon passenger locomotives.

For the tender trucks, the loads (considering the tender full of water and fuel) showed less variation—from 290 to 330 pounds per square inch. A conservative figure would be probably 300 pounds. This load, of course, is subject to great variations, as with the water and fuel at a low point, the pressure would be only about one-half as much.

In driving axles, considering only the weight of the engine, the pressure varied from 185 to 230 pounds, a fair value being 200 pounds per square inch. The pressure of the piston will create a horizontal load perhaps twice as great, but it must be remembered that this is continually changing from one side to the other, with every stroke of the piston, thus affording the oil an opportunity to enter between the journal and the bearing, and by the mere fact of its reversal, a much higher pressure per unit of surface is permissible. This change of load does not occur with the static vertical pressure due to the dead weight of the engine, which necessitates the use of a smaller unit load than on the crank pin.

The wear of both journal and bearing resulting from the friction is a matter of much importance, but it is not as well understood as would be desirable. Mr. Van Alstine of the Chicago Great Western gave some data before the Northwest Railway Club in 1902, which indicated the following mileage for 1-16 inch wear of the journal; that is, a reduction in diameter of ⅛ inch.

Locomotive driving axles........................120,000 miles
Locomotive truck axles.......................... 60,000 miles
Locomotive tender axles........................120,000 miles

He concluded that it was not the quality of the bearing metal or the packing that was responsible for the wear, as the

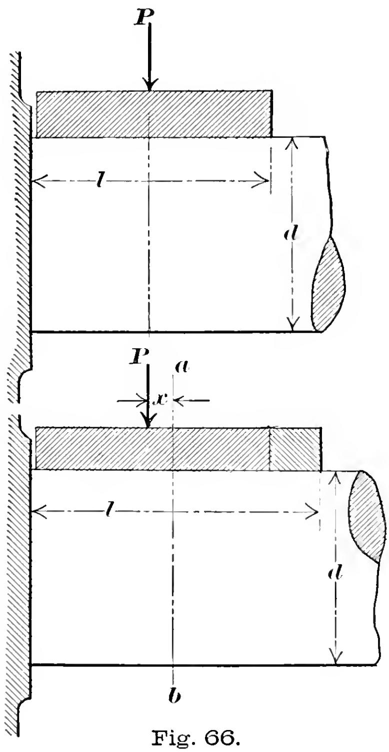


Fig. 66.

exclusion of dirt was the principal object to be attained, and that the grit was the chief cause of the wearing taper of driving axle journals. An eccentric load will also cause this trouble, as well as local heating. It has been rather a common practice to obtain a greater length of driving axle journal, and so reduced unit pressure, by adding a couple of inches to the thickness of the box on the inside. The spring saddle still maintaining its original position, the load due to the weight of the engine was not now applied centrally as to the length of the journal. This caused heating and uneven wear, and better results were obtained by shortening the box so that the center of load would coincide with the center of length of journal. The average unit load was thereby increased, but a concentration of load was avoided. This can be explained by referring to Fig. 66. In the upper view, the load P is applied centrally as to the length l, and the unit pressure is

$$p = \frac{P}{d\,l}, \quad \ldots\ldots (73)$$

d being the diameter, and this pressure p will be uniform throughout the length l. If now we add to the inside of the bearing, as shown in the lower view, so that the load P is away from the center of the new length by the distance x, the line a b being the center line of the bearing, the unit load or pressure at the edges of the bearing will be

$$p' = \frac{P}{d\,l} \pm \frac{P\,x}{\frac{1}{6}\,d\,l^2} = \frac{P}{d\,l}\left(1 \pm \frac{6\,x}{l}\right) \quad \ldots\ldots (74)$$

the positive sign referring to the edge nearest to P and the negative to the farthest edge. As an example, let us take a case in which the load P = 16,000 pounds, the diameter d = 8 inches, and the length l = 10 inches. If the load be central, equation 73 gives us for the uniformly distributed pressure

$$p = \frac{16{,}000}{8 \times 10} = 200 \text{ pounds per square inch.}$$

If now we add 2 inches to the inside of the box, leaving the application of the load as before, we have l = 12 and x = 1, and from equation 74 we obtain

$$p' = \frac{16{,}000}{8 \times 12}\left(1 \pm \frac{6}{12}\right) = 166\,(1 \pm \tfrac{1}{2}) = 249 \text{ and } 83,$$

the larger value being the unit pressure at the outside edge and the lower value the unit pressure at the inside edge of the bearing. While the average pressure is only 166 pounds, the concentration increases the maximum pressure to 25 per cent more than with the 10-inch journal and the central load.

The soft metals are often considered to cause more rapid wear of the journal than the harder mixtures, probably caused by the imbedded grit which they secrete, which forms minute cutting edges. The wear of the bearings themselves can often be reduced by the addition of lead to the bronze or brass.

Several years ago the Pennsylvania Railroad made exhaustive service tests with various combinations of copper, tin and lead, and the conclusions drawn from these experiments were as follows:

Copper-tin alloy showed about 50 per cent more wear than phosphor bronze.

Wear increases with the proportion of lead and tin.

Alloys containing more than 15 per cent of lead, or less than 8 per cent of tin, could not be produced because of segregation, but if this could be accomplished, it is believed a better metal would result.

As before mentioned, the plastic 30 per cent lead bearings of the Ajax Company show about one-third the amount of wear that phosphor bronze bearings do, this, no doubt being caused by a uniform pressure, which reduces the maximum pressure at different points brought about by the bearing not being a true fit upon the journal. Lead lining is also very common, the principle being that the lead is soft and will squeeze out from under the high spots, so distributing the pressure until it gradually wears off, leaving the brass with a fairly uniform bearing all over the surface. In such cases the lead is about 1-16 inch thick. In filled brasses the yellow metal is generally left rough, and a considerable thickness of white metal is used, say 1/4 to 3/8 inch. It is never intended in these brasses that the journal shall touch the yellow metal, but in cases of emergency, such as the melting of the white filling, the brass will be present to protect the journal. Upon Mexican railroads malleable iron shells have been used with a white filling, in order to reduce the value of the bearing, and thus diminish its saleability when stolen.

In 1900 a committee of the Master Mechanics' Association stated that the miles run to a pound of bearing metal worn away had risen from 800 in 1891 to 2,000 in 1897, in passenger and freight car service. There are so many factors entering into the question of wear of bearings, however, that it is very difficult to obtain reliable information on the subject.

The heating of journals, which constitutes such a serious obstacle to high speeds and sandy countries, depends entirely upon the friction of the bearing upon the journal. Mr. Robert Job of the Philadelphia & Reading Railway has made a study of the character of bearing metals as determined by microscopic investigations, and states that the principal causes of heating are—

Segregation of the metals.

Coarse crystalline structure.

Dross or oxidation products, and an excessive amount of enclosed gas in the metal.

(Lack of lubrication must ever be considered a source of trouble, and cannot be too carefully watched.)

These three conditions are all due to careless or improper foundry work, and Mr. Job lays great stress upon the importance of having this branch of the work properly supervised. The pressure of oil between the bearing surfaces was investigated by Mr. Josef Grossmann of the Northwestern Railroad of Austria, who finally recommended a bearing with a narrow area of contact—not nearly as wide as the diameter of the journal. Several holes are drilled through the crown of the bearing, not for the passage of oil downward to the journal, as the lubricant was supplied by a bath or oily waste, but for the purpose of taking the oil up from the journal and allowing it to trickle down the sides of the bearing, and drop on the unloaded part of the journal.

In some other tests it was found that with a load of 100 pounds per square inch of horizontal projection of the journal, and an oil bath below, the oil rose in a hole drilled at the center of the bearing with a force that registered 200 pounds per square inch by a gauge. When the gauge was applied half way between the top center and the edge, a reduced pressure was obtained. This pressure was less on the side which the surface of the journal approached in its revolution than the side which it left, demonstrating that the greatest hydrostatic force of the lubricating oil was near the top of the bearing, dropping off to nothing at the edges.

In line with these results, a master mechanics' committee in 1900 recommended an oil slot in the shells of driving boxes a little below a point 45 degrees from the top of the bearing, in order to deliver the oil into a zone of lighter pressure, but even a small pressure would prevent the entrance of oil unless forced into the cavity. The writer has known of cases where the side oiling failed absolutely in service. What is satisfactory in some cases seems to produce the opposite results at other times. It is probably true, however, that a method of forcing the oil into the cavities and onto the bearings will in time be considered the only practical method of lubrication under high surface pressures. In all cases of underneath lubrication, whether by bath or waste, care should be taken that the edge of the bearing does not scrape off the oil and prevent its getting between the journal and bearing.

Some very terse rules for the care of journal boxes were laid down by the New York Central a few years ago, and while they refer principally to car journals, they can be considered with advantage in connection with locomotives. The following is a portion treating of the method of packing and the preparation of the waste:

"In packing boxes, the first portion of waste applied is to be wrung moderately dry, and it is to be packed moderately tight at the rear of the box, so as to make a guard for the purpose, not only of retaining the oil, but excluding the dust as well. Care is to be taken to keep the waste at the side of the box down below the bottom of the journal bearing about an inch, and also to have that portion of the waste in the front end of the box separate and distinct from that which extends from the front end of the journal to the back of the box. This will avoid derangement of the packing in the rear of the box.

"The roll of packing which is placed in the front of the box is not to extend above the opening in the front.

"At terminals or yards when journal boxes require special attention to the packing, the following practice is to be adopted:

"A packing knife or spoon of standard style should be used. This packing knife or spoon is to be used to ascertain whether the packing is in the proper place at the back of the box, and to loosen up the waste at the rear and side of the journal. This particular treatment is given to prevent glazing of the packing (which occurs when it is too long in contact with the journal) and, at the same time, to put the packing in the proper place at the rear of the box. It is desirable to give this treatment at intervals of 500 miles' run for cars and tenders if possible.

"A small quantity of packing is to be removed from the sides of the journals when found not in a good condition, and this replaced by similar quantity of well-soaked packing. No box is ever to have oil applied before the packing is properly loosened up on the sides and back of the box with the packing iron. Before applying a bearing to a journal the surface of the bearing is to be examined to insure that it is free from imperfections of any kind that will cause heating. The surface of the bearing is then to be oiled or greased before it is placed on the journal. When applying wheels or axles the journals are to be examined to insure their being free from any imperfections which would cause heating. When wheels or axles are carried in stock, the journals should be protected with a good material suited to protect the surface, without hardening, and one which is not difficult to remove.

"When the journal is found heated and there is a good supply of packing in the box, it is evidence of some imperfection of the journal, journal bearing, box or wedge, and the bearing is to be removed, provided the box is heated to such an extent as to require repacking of the box. Boxes which have warmed up slightly will in most cases, by partially replacing with freshly soaked packing, give better results than by entire removal of the packing from the box. When it is necessary and permissible to oil boxes, it shall be as short a time before leaving time of the train as possible.

"When preparing packing, the dry waste is to be pulled apart in small bunches and any hard particles in it removed. Each bunch is to be loosely formed to facilitate soaking and packing, as in this form boxes can be packed in a more satisfactory manner, and with less waste of oil. This loose, dry packing is to be put in soaking cans or tanks provided for that purpose, pressed down moderately tight, then covered with oil and allowed to remain at least 48 hours. After being saturated for this length of time the surplus oil is to be drained off, leaving it then in proper condition for use in packing boxes. Standard equipment for saturating and draining packing is to be provided at all points where packing is to be kept for use, unless suitable equivalent equipment is already in use."

In taking care of locomotive driving boxes, it is generally preferable not to disturb the packing too often, unless the box is giving trouble. A thorough packing two or three times a month is ordinarily sufficient. A packing knife can be run in between cellar and journal, however, a couple of times a week to be sure that the waste is up against the journal. Wool waste in the cellar and cotton waste on top of the box gives the best results.

Graphite would no doubt make a good lubricant, but it is very difficult to apply to the bearing without choking oil holes and grooves.

A system of driving box lubrication by grease instead of oil has been used quite successfully on the Lackawanna Railroad. The grease in the form of a block is pressed against a curved, perforated shield by a spring in the cellar, and is forced through the openings in the shield and wiped off by the journal. The amount of grease used per 1,000 miles per driving box is about 2½ ounces, and about 1½ ounces for truck boxes, the cost said to be about 1-10 of that with oil, and with reduced trouble from heating.

PIN BEARINGS.

What has been said about journal friction applies very largely to crank and crosshead pins; the unit pressures are, however, much greater in current practice, largely due to the fact that the maximum pressures are never maintained for any length of time at high speeds, and that they are continually reversing in direction. We have seen under the head of steam action that the duration of the heavy piston rod loads is extremely short, but the maximum is useful as a unit of comparison. For main crank pins the pressure may be considered as the product of the boiler pressure and the cylinder area; for side rod bearings on pins, it will be the above product, divided by the number of driving axles, multiplied by the number of wheels rotated by the bearing in question. Thus in a 20-inch cylinder engine with 200 pounds boiler pressure, the load will be $314 \times 200 = 62{,}800$ pounds for the main rod bearing. If the engine be a 2—8—0 type, the main wheel side rod bearing must rotate the other three driving wheels, so that its load will be ¾ of 62,800, or 47,100 pounds, and the middle connection front and back pins will each receive ¼ of 62,800, or 15,700 pounds, as a maximum. The ordinary unit pressure allowed on crank pins, with the load figured as above, is 1,600 or 1,700 pounds per square inch. Thus, if in the case just quoted the pin were to have its main bearing 6 inches in diameter on account of considerations of strength, its length should be $\frac{62{,}800}{1{,}700 \times 6} = 6\frac{1}{6}$ inches; that is, its projected area multiplied by 1,700 must equal the piston load, 62,800 pounds.

With crosshead pins, the amount of motion due to the vibration of the main rod is so slight that the pressure per square inch of projected area allowed runs about 4,800 pounds. It is probably safe to permit the wearing of these two pins so that the unit pressure will not exceed 2,000 and 5,000 pounds, respectively, these considerations, of course, referring to the freedom from heating, and not to the strength of the parts.

The value of the friction will probably be about the same as for journals, but as the load is constantly varying, the resistance will be continually changing.

Grease is, in many cases, much more satisfactory as a lubricant than oil for pin bearings. It possibly wears the metal faster, but the reduction of hot pins by its use is so general that the saving in replacements is likely to overcome the increased wear. It is common for locomotives to run 500 miles or more with one filling of the rod cups with grease.

The use of white metal strips or filling plugs in rod brasses is not to be recommended, especially with oil lubrication. The soft metal holds grit and induces cutting of the journal, and should the pin become heated, the molten metal fills the oil hole, preventing further supply reaching the journal. Asbestos has given very good results when packed in the cavity ordinarily used for babbitt, as it keeps the journal wiped off and acts like a swab.

The feeding of grease is partly automatic, for if the journal heat to any extent, the grease is melted, and runs to the bearing surface, thus reducing the friction and cooling the brass.

Solid bushings are now almost universally used in side rods, and require little attention except the ordinary lubrication. When worn sufficiently, they must be replaced. Hard bronze is the metal chiefly used, but some white metals, like "Lumen," give excellent results.

GUIDE FRICTION.

We have seen in formula 61 that the pressure of the cross-head upon the guide is a very variable quantity, being expressed by the equation

$$P_v = P\frac{r}{l}.\frac{\sin a}{\sqrt{1 - \frac{r^2}{l^2}\sin^2 a}}$$

and which may be written simply $P_v = P\frac{r}{l}\sin a$, as the radical in the denominator is in practice very nearly equal to unity. Not only does the pressure P_v depend upon the angle of the crank, which is continually changing, but the piston pressure P, as we have seen, is subject to very great fluctuations, except when starting, when it retains its maximum value almost throughout the stroke. As $\sin a = 1$ at 90 degrees, the maximum pressure will be $P\frac{r}{l}$, but the average pressure will be

$.785 \frac{r}{l} P$, as the average of the sines for the half circle corresponding to one stroke equals $\frac{2^2 \times \pi}{2 \times 4 \times 2} = \frac{\pi}{4} = .785$. The average friction F will then be

$$F = .785 f \frac{r}{l} P \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (75)$$

Where f = coefficient of friction,
r = radius of crank,
l = length of main rod,
P = total piston pressure.

Now if we let $\frac{l}{r} = 10$ and $f = .05$ we have the average friction

$= \frac{.785 \times .05}{10} P = .004 P$ or the frictional resistance of the crosshead absorbs $\frac{4}{10}$ per cent of the work done by the piston.

Very low values are generally given to the unit average surface pressure of the crosshead against the guides, in fast passenger locomotives sometimes not much over 30 pounds per square inch, and in freight engines reaching 50 pounds, the pressure being determined by equation 75. The exposed location of the guides makes it difficult to exclude sand and grit, which probably accounts for the existing low unit pressures; fortunately the upper guide carries all the wear on road engines, and the dirt and sand are not so liable to cling to its under surface.

Tin or some form of babbitt is much used for a wearing surface, although some roads still cling to brass or bronze. Cast iron is a very good wearing metal, but in the laudable desire to reduce the weight of reciprocating parts, cast steel is much used, which necessitates a white or yellow metal wearing surface.

Oil lubrication is the best for guides, but there being no regular motion, a needle or similar feed cup must be used.

STUFFING BOX FRICTION.

Upon this subject we have very little reliable information. Mr. C. H. Benjamin, in 1899 presented the results of some tests to the American Society of Mechanical Engineers, but unfortunately, no metallic packings were tested. The experiments were made upon a rod 2 inches in diameter, passing through a cylinder containing steam, with a stuffing box at each end, the rod having a travel of 4½ inches, and making 200 revolutions or about 140 feet per minute. He concluded that probably one per cent of the work done by the steam in the cylinder was used in overcoming the friction of the piston rod packing, the tests covering only soft packings. He also stated that the friction depends upon the kind of packing; that it increases directly with the steam pressure; that injudicious use of a wrench upon the gland stud nuts causes undue friction; and that oiling the rod always reduces it—sometimes by one-half. The tests only covered the friction of the rod, as it was moved in a straight line. In a locomotive, the lost motion between the crosshead and guides continually causes vertical vibrations, which are extremely hard on packing. The principal metallic packings allow for this vibration, but some of them do not give sufficient room. In the Baldwin compounds the unequal pressure upon the high and low pressure pistons causes a great disturbance of the packing, making it very difficult to prevent leaks, and in some cases a double packing is used for these engines.

The quality of metal in the packing rings is of prime importance. The United States Metallic Packing Company use a composition composed of 83 1-3 per cent lead; 8 1-3 per cent tin, and 8 1-3 per cent antimony. It is of the utmost necessity to exclude grit and dust, and in a locomotive this is extremely difficult to accomplish. The maintenance of a good swab cup, kept filled with greasy packing, is essential, its province being to wipe off the dirt from the rod before it is drawn in between the rod and rings. The packing must be kept well oiled, which reduces the friction and prevents cutting, and probably nothing is more important to the success of metallic packing than lubrication. Often we see oil cups which would feed upon the rod in an engine room, but upon the road the drops are carried away by the wind before reaching the rod; or the oil is fed through a pipe that freezes up in cold weather. It is also important that the rods are round and true, and in first-class shape when the packing is applied; the white metal rings should be carefully finished in order to make steam tight joints —if very soft metal is used with the idea that it will squeeze to a bearing, it is apt to blow or drag through the gland.

One of the greatest enemies to metallic packing is water in the cylinders, whether produced by priming or condensation. It is almost sure to ruin the packing in one trip—this accounts for the greater difficulty of maintaining rod packings in satisfactory condition in territories of foaming water.

CYLINDER FRICTION.

While the importance of this subject is great, and many inventors have been working to reduce this friction, we know little about its actual amount. As the weight of the piston must come upon the lower part of the bore (unless an extension rod be used), we know that it is desirable to make it as light as possible consistent with strength. With the old style packings that were set out with a wrench, and which were very wide, the friction was no doubt very great, especially when carelessly adjusted, but the modern narrow rings depending upon the steam pressure for their tightness against leaking, are without doubt easier upon the cylinders. We should normally look for the greatest wear upon the bottom of the piston and cylinder, but in Baldwin compound locomotives the unequal pressure upon the two pistons causes the greatest wear upon the top of the bore, if the low pressure cylinder be above, due to the tilting of the crosshead. Large pistons are often provided with extension rods to relieve the cylinder of this weight, but the stuffing box or brass sleeve upon which this extended rod passes through, causes considerable trouble. Cylinders should be provided with bushings, and pistons with wearing rings, which can readily be renewed when worn, without scrapping too much good material. Care should be given to the proper dressing off of dowels which are used to prevent the rings from turning, as cylinders have been ruined by steel dowels coming in contact with the bore, cutting deep V-shaped grooves the full length of the stroke.

The best wearing surface is cast iron, with a small amount of steel, say 15 per cent, both for cylinder bushing and packing rings. This material takes a glaze, greatly reducing the friction and wear.

The lubrication of the cylinders is of the utmost importance, but, as a rule, is very imperfectly accomplished. Engineers often have extremely vague ideas of the quantity of oil necessary, and insist on using more than is needed. Again, some lubricators feed irregularly, and occasionally we find the pipes so connected to the cylinder or valve chamber that the oil never reaches the piston. A good grade of high fire test valve oil must be used, and sometimes flake graphite is introduced with excellent results. There are devices now upon the market for feeding graphite into the cylinders of locomotives. It is often advisable to place a double set of lubricators, or one with four outlets, upon tandem and other four-cylinder compounds, as the low pressure cylinder is liable to heat when drifting if the oil is taken to the high pressure cylinder only.

As stated above, the performance of condensing or displacement lubricators is often uncertain. The amount of steam which is condensed in the dome of the lubricator displaces an equal volume of oil, which runs down the "tallow" pipes. But if the throttle be closed, the pressure at the cylinder end is much less than that upon the lubricator, and an increased flow of oil will result, unless special arrangements are introduced to maintain a steady supply.

Several years ago one of the large western roads made tests of a number of displacement lubricators, with results as briefly stated below:

Lubricator A: Performance unsteady and lubrication very poor. Test had to be stopped on account of valves squealing. Closing the throttle caused the rate of feed to double.

Lubricator B: Delivery of oil was fair, but somewhat unsteady, coming in gushes. One valve squealed half of the time. On closing the throttle the rate of feed quadrupled.

Lubricator C: Lubrication was fair, but test was finally stopped on account of the valves running dry. The closing of the throttle was followed by a slight increase in rate of feed.

Lubricator D: Lubrication was excellent and flow of oil steady. The rate of feed was only slightly increased upon closing the throttle.

Several varieties of "force pump" lubricators are now upon the market, and it seems probable that these will eventually take the place of the condensing lubricator at high steam pressures.

The information following was obtained from the Galena Oil Company:

LUBRICATOR FEEDS.

(One pint of valve oil contains about 6,500 drops.)

Drops Per Minute Each Cylinder	Air Pump.	Total Per Minute.	Total Per Hour.	Time Required to Consume 1 Pint.	Miles Per Hour.	Miles Per Pint.
8	2	18	1,080	6 hours	10	60
7	2	16	960	6¾ "	15	102
6	2	14	840	7¾ "	20	155
5	1	11	660	9½ "	25	237
4	1	9	540	12 "	30	360
3	1	7	420	15 "	35	525
Passenger						
8	2	18	1,080	6 "	30	180
8	2	18	1,080	6 "	40	240
Switch						
4	1	9	540	12 "	5	60

The following allowance of miles to a pint of cylinder oil should ordinarily be made without difficulty:

Type of Engine.	Service.	Diameter Cylinder.	Miles Per Pint.
4-4-0 Simple.	Passenger.	17 and 18 inch.	150
4-4-0 Simple.	Freight.	17 and 18 inch.	100
4-6-0 Simple.	Passenger.	19 and 20 inch.	115
4-6-0 Simple.	Freight.	19 and 20 inch.	90
2-6-0 Simple.	Freight.	20 inch.	80
2-6-0 Compound.	Freight.	15½ and 26 inch.	75
2-6-2 Compound.	Passenger.	17 and 28 inch.	90
2-6-2 Compound.	Freight.	17 and 28 inch.	75
2-8-0 Compound	Freight.	17 and 28 inch.	75
2-8-0 Simple.	Freight.	20 and 21 inch.	60

In many individual cases, even better results may be obtained than here stated, but the average will always fall considerably below the best performances. If all waste of valve oil is eliminated, and the men are prevented from putting it on driving boxes and crankpins, where it should not be used, a material gain can be made.

VALVE FRICTION.

Few parts of the locomotive have received as much attention from designers as the main valve, with a view to reducing its friction upon the seat. When valves were small and steam pressures were light, the resistance to the motion of the valve was not great, but with present pressures and sizes of ports, the removal of valve friction becomes of the highest importance. We are glad to state that the efforts to reduce this friction have been eminently successful, as has been demonstrated by a number of tests.

In 1896 a committee of the Master Mechanics' Association conducted a series of tests upon the experimental plant of the Purdue University, at Lafayette, Ind. The locomotive with which the tests were made has cylinders 17 inches in diameter by 24 inches stroke. The ports are 16 inches long, the steam port being 1¼ and the exhaust port 2½ inches wide. The bridges are 1⅛ inches wide. The valve had a maximum travel of 5½ inches, ¾-inch steam lap, 1-32-inch exhaust lap and was set with 1-16-inch lead in full gear forward and 7-32-inch blind in full gear backward. The radius of link is 63 inches, the valve stem 1¾ inches and the piston rod 3 inches in diameter, the driving wheels 62¾ inches in diameter and the boiler pressure 145 pounds.

Four different valves were tested, unbalanced D valve, Richardson balanced and American with single and double balance rings. A fluid dynamometer was placed in the connection between the valve stem and rocker arm in order to measure the friction. The valves weighed 78, 85½, 79¼ and 84 pounds respectively, in the order given above; the dynamometer 105, and the yoke 37 pounds. The Richardson valve had 56 per cent of the area of the valve balanced, this being effected by flat strips, held against the balance plate by springs.

The American valves had 61½ and 66 per cent balanced by single and double rings respectively, the rings fitting over a coned surface.

A steam engine indicator was arranged to draw a diagram, in which the length corresponded to the stroke of the valve, and the pressure of the fluid in the dynamometer was shown by the height. The tests were made at different cut-offs, and at 10, 20 and 40 miles an hour. A few of the results are tabulated below:

VALVE FRICTION TESTS.

Data.	Miles Per Hour.	22-inch Cut-off.			18½-inch Cut-off.			9½-inch Cut-off.		
		10	20	40	10	20	40	10	20	40
	Valve.									
Mean pull at 100 pounds steam chest pressure....	Richardson....	382	396	772	362	370	694	361	442	468
	Amer. single..	......	522	872	430	484	765	394	535	591
	Amer. double..	......	488	762	467	373	576	412	500	568
	Unbalanced....	1118	1062	1207	1187	1060	924	1322	1240	1180
Per Cent of I.H.P. of one cylinder required to move 1 valve	Richardson....	.43	.49	1.54	.24	.27	.73	.32	.34	.61
	Amer. single..	.48	.65	1.91	.30	.37	.85	.27	.40	.67
	Amer double..	......	.61	1.66	.31	.28	.68	.27	.45	.63
	Unbalanced....	1.20	1.30	2.42	.80	.76	1.11	.82	.....	1.62

The committee summed up the results by stating that the average friction or resistance of unbalanced valves was about twice as great as that of balanced valves, and they recommended that the area of balance = area of exhaust port + area of two bridges + area of one steam port.

The cards taken in these tests are quite interesting. Fig. 67 is a reproduction of those taken with the unbalanced valve with lever in first and thirteenth notches (22 and 9½ inch cut-off respectively), and at 10, 20 and 40 miles per hour, the highest speed being at the bottom. The left side of the card is the front end of the valve travel. The angular lines intersecting the base line, show the force due to inertia, so that the friction should really be measured from these lines.

We will study the card taken in the first notch at 20 miles an hour a little closer. In Fig. 68 the upper view is a reproduction from the middle left-hand card of Fig. 67, and in this test the steam chest pressure was only 59 pounds. On the forward stroke the average pull was 763 pounds, and on the back stroke 486 pounds, the mean of the two being 625 pounds,

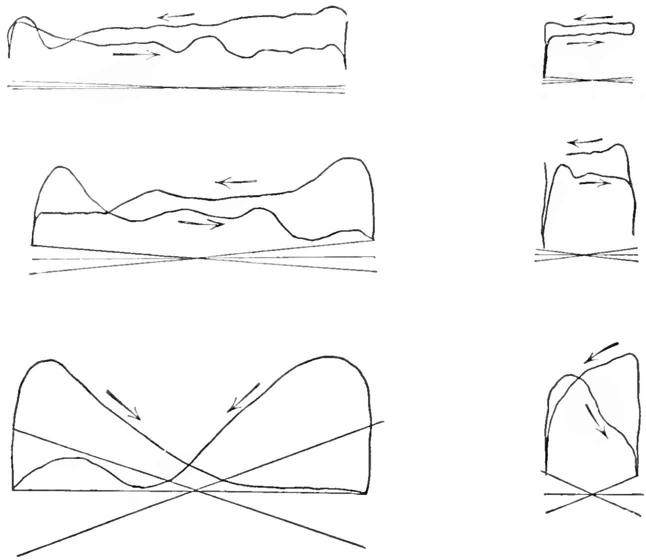


Fig. 67.

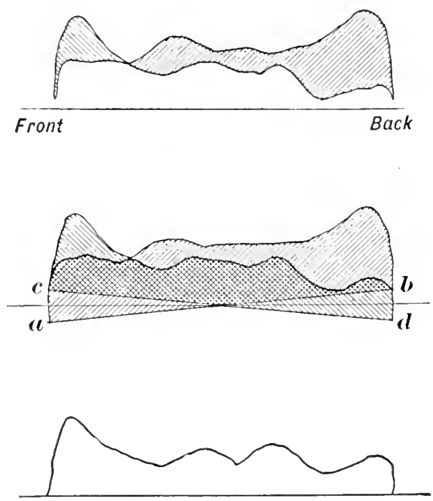


Fig. 68.

and at 100 pounds steam chest pressure the mean pull or friction of the valve would have been 1,062 pounds, at the same ratio. As the area of the valve was 8¾ × 17¾ = 155 square inches, the total pressure would be 155 × 59 = 9,145 pounds, assuming that none was balanced by the steam in the ports, 625 ÷ 9,147 = .07, approximately, or the coefficient of friction was about 7 per cent. The difference between the forward and backward strokes of 277 pounds on the average, as shown by the shaded area in the upper view, was caused by the area of the 1¾-inch valve stem, or 2.4 × 59 × 2 = 283 pounds difference, on account of the steam assisting upon the backward and resisting the forward stroke. The calculated difference is only 6 pounds more than shown by the card.

In the middle sketch the lines a b and c d represent the forces of inertia to the same scale as the diagram. These are figured by using formula 8, in connection with a Zeuner diagram, as shown in plate 8, the latter being used to determine the equivalent eccentricity of the valve motion. The circles in the plate, numbered 1, 2, 3, etc., represent by their diameter an eccentric that would give a similar motion to the valve as that produced by the shifting link, and by such a diagram we find for the 22-inch cut-off an eccentricity of 2.77 inches, or .23 feet; for the 18½-inch cut-off, 1.62 inches, or .135 feet, and for the 9½-inch cut-off, .97 inches, or .08 feet. This eccentricity is, of course, one-half of the valve travel. For the revolutions at 10, 20 and 40 miles an hour we have 54, 107 and 214 per minute, respectively. Now, using these values in formula 8, as for example, the 22-inch cut-off at 20 miles per hour, we have for the inertia at end of stroke, which equals the centrifugal force (the eccentric rod being very long relatively to the valve travel) $= .00034\,G\,r\,n^2 = .00034 \times .23 \times 107^2\,G = .92\,G$, and as the total weight of parts ahead of the dynamometer was 220 pounds, the inertia at end of stroke was 202 pounds. The coefficients of G for the three speeds and cut-offs are:

	—Miles per hour—		
Cut-off.	10.	20.	40.
22 inches	.22	.92	3.58
18½ inches	.13	.53	2.10
9½ inches	.077	.31	1.25

and the inclined lines in Fig. 67 were drawn from these figures, multiplied by the weight. This value, divided by the area of the dynamometer piston, was laid off vertically at both ends of the card from the base line, and connected by a straight line, as we have already found this to be approximately correct. The diagonal hatching shows the friction areas as corrected for inertia. The lower diagram in Fig. 68 shows the friction alone, with the inertia and steam pressure on valve stem area deducted. The variation in pull is due to the change of steam pressure in the ports under the valve, as they are opened and closed by its motion, thus partly balancing the pressure on top of the valve. No allowance was made for the stuffing box friction, as its value was unknown.

When we recollect that the balanced valves had from 56 to 66 per cent of their area balanced, we readily understand why the tabulated records show from one-third to one-half as much friction as the unbalanced valve.

Some tests of the same kind made by the Chicago, Burlington & Quincy showed an average friction of 905 pounds for the unbalanced and 330 pounds for the balanced valves, which had 42 per cent of the horizontal area balanced. The coefficient of friction of the unbalanced valve in this case was .04, and the balanced valves offered but 36 per cent as much resistance as the plain valve.

The Master Mechanics' committee reported a wear of from 1-32 to 1-16 inch per 100,000 miles with balanced valves and two or three times as much with plain valves.

Some later tests made on the Chicago, Burlington & Quincy, to demonstrate the relative frictional resistance of balanced slide valves and piston valves, indicated that the latter required only about half as much force as the former.

The lubrication of the valve is very important, especially when drifting, and the pipes should deliver the oil to the place where it is needed. Some complaint has been made regarding the breakage of rings in the piston valve, when dropping the reverse lever to the low notches, and the fact that the engineer is likely to go out of the cab door. This was discussed at the Master Mechanics' meeting of 1903, when Mr.

John Player claimed that this was due to improper handling of the engine, caused by the men dropping the lever too quickly. His explanation was that the exhaust surface of the bushing, which is not traveled by the valve at short cut-off, becomes encrusted with a scum, and if the lever is suddenly dropped, this scum must be cut off at one stroke of the valves, and this is liable to break the rings, or throw the reverse lever violently forward. In a slide valve it will simply lift and ride over this crust, but a piston valve cannot lift. If the valve is handled in a proper manner, he claimed, not dropped down suddenly, but gradually, there will be no difficulty experienced. It is a well established fact that when using steam the reverse lever can be moved all over the quadrant with one hand. This was a difficult feat for even a strong man prior to the use of piston valves, and gives at once a forcible demonstration of the reduction in friction accomplished by their use.

LINK MOTION FRICTION.

As the link motion moves the valve, the friction of the various rotating and sliding connections will depend primarily upon the friction of the valve. As there is always some slip in the link, there will be friction developed there, depending largely upon the relative angle of the link during the cycle of operations. The large diameter of the eccentrics gives a comparatively great frictional moment and work. The various pins in the motion, being of small diameter, cause little loss of energy, the principal amount being with the eccentrics. Ordinarily in full gear, the surface of the eccentrics will move five times as far in a revolution as the valve, and when cutting off early the ratio will be 10 or 12. If the friction of the eccentrics and straps is taken at .05, we shall have the frictional work done by them $= .05 \times 5 = .25$, or $.05 \times 10 = .50$ of the work done in moving the valve. The other losses in the motion are difficult to estimate, but if we assume that they are nearly as great as those in the eccentrics, we can call the total resistance of the link motion equal to that of moving the valve; that is, the whole amount of power absorbed in moving the valve through the medium of the link motion will be double that consumed in moving the valve alone.

It is of great importance that the various rubbing and wearing surfaces be kept properly lubricated, especially the eccentrics. These latter are more or less difficult of access and exposed to a constant cyclone of dust and grit, and very frequently give trouble by heating. Oil cups should be used that can be readily filled from the outside of the engine—in fact, open pieces of gas pipe, 4 or 6 inches long, containing some wool waste for a strainer, give excellent results and encourage the engineer to provide sufficient lubrication. The motion pins, if not attended to, frequently stick and twist completely off, even if the surface velocity is small. The force necessary to reverse the engine is quite a good index to the lubrication of the motion, and is of value in this respect. With the power reversing gears this was absent, and no indication was given until trouble actually occurred. This was a large factor in determining their abandonment.

INTERNAL RESISTANCE.

The several causes of frictional resistance which we have just considered go to make what is generally known as the "internal resistance" of the locomotive. As might be expected, this varies considerably among different engines. Some authorities consider it a constant quantity, regardless of the speed or cut-off, others as a function of the cylinder or indicated power. It probably falls between the two hypotheses.

Several years ago, Prof. Goss, in a paper read before the New York Railroad Club, presented the formula for internal friction =

$$3.8 \frac{d^2 s}{D} \qquad (76)$$

where d = diameter of cylinder,

s = stroke of piston,

D = diameter of drivers, all in inches.

This value obtains at the circumference of the drivers and is constant, regardless of cut-off or speed, and was deduced from a large number of tests with the locomotive in the Purdue Laboratory, for which engine the resistance of internal friction was about 400 pounds. Tests made on the Chicago, Burlington & Quincy Railroad, reported in the Western Railway Club Proceedings of 1893 by Mr. William Forsyth, indicated an internal resistance of about 450 pounds at speeds from rest up to 60 miles an hour. As this engine had 18 by 24 inch cylinders and 69-inch drivers, the resistance by equation 76 would be $\frac{3.8 \times 324 \times 24}{69} = 430$ pounds, showing a marked agreement between the test and Prof. Goss' formula. A Baldwin compound, with cylinders 14 and 24 by 24 inches and 72-inch drivers gave an internal resistance of about 1,300 pounds, being slightly greater at long cut-off. As these cylinders are equivalent to one 20 inches in diameter, the effect of the four pistons, etc., is seen by figuring the resistance for a simple engine of same power, viz., $\frac{3.8 \times 400 \times 24}{72} = 507$ pounds. If we add the squares of the 14 and 24 inch cylinders together, we obtain

$$\frac{3.8 \times (196 + 576) \times 24}{72} = 977 \text{ pounds.}$$

These values were 3 and 10 per cent of the indicated tractive power of the simple and compound engines respectively at 10 miles an hour, and 9 and 20 per cent at 50 miles an hour. The Purdue locomotive above referred to gave about 3 per cent at slow speeds and long cut-offs and 10 per cent at 50 miles an hour. This also agrees with the Chicago, Burlington & Quincy simple engine.

Wellington, in his "Railway Location," gives the internal friction as ranging from 5 to 8 per cent, and the Master Mechanics' committee of 1898 on tonnage rating used 8 per cent in their calculations. Tests made upon the plant of the Chicago & Northwestern Railway with a 4—6—0 freight locomotive, and also with a dynamometer car in road service indicated an internal resistance of 9 per cent, with long cut-off and slow speed, and 15 per cent with one-quarter cut-off and a speed of 50 miles an hour. In the latter case the actual resistance was apparently but one-fourth that in the former.

If we sum up the several resistances which we have studied in detail, we obtain results as follows, which must, however, only be considered as approximations:

The journal friction here should be taken only so far as the piston pressure is concerned, the weight of the engine going into the general rolling friction. The circumference of the driving axle is generally about equal to the stroke, and the relation of the work of friction to that of the steam will be

$\frac{P f s}{2 P s} = \frac{.05 P s}{2 P s} = .025$, or 2.5 per cent, P being the total piston pressure, and f the coefficient of journal friction, assumed to be .05. The main pin bearing is usually about two-thirds the size of the driving axle, so that the relation would be two-thirds of 2.5 per cent, or 1.7 per cent. As the side rods transmit from one-half to three-fourths the power of the main pin, and as there are two bearings for each load, we can take the side rod frictional resistance also equal to 1.7 per cent.

In our study of crosshead friction, we have seen that it is equal to about .4 per cent. The piston and rod packing is uncertain—probably at least 1 per cent. We found that balanced valves absorbed about .6 per cent, and assumed that the link motion used the same amount. Now, adding these together we have:

Driving axle journals	2.5 per cent
Main pin bearings	1.7 per cent
Side rod bearings	1.7 per cent
Crosshead	.4 per cent
Piston and rod	1.0 per cent
Valves	.6 per cent
Link motion	.6 per cent
Total internal resistance	8.5 per cent

of the piston pressure, or the indicated power of the engine. At high speeds and early cut-off, compression causes frictional resistance and reduces the M. E. P. in the cylinder, thus accounting for the greater percentage of friction.

From the different tests reported above, we can prepare a formula that will fairly represent the variation in percentage of indicated power that is consumed by internal resistance.

Let V = speed in miles per hour,

c = a constant, whose value may vary from 2 to 8, the latter figure being the safest to use for heavy and slow work.

Then the percentage of indicated power consumed in friction =

$$.15\ V + c \quad \ldots\ldots (77)$$

Until more extended experiments are made, it will probably be difficult to decide between equations 76 and 77. Under certain circumstances they will both give the same result, as in the test of the Chicago, Burlington & Quincy simple engine above referred to, where the smaller value of c in equation 77 will give figures corresponding very closely with the test.

BRAKESHOE FRICTION.

The first experiments of importance which were made to determine the friction of brakeshoes were probably those of Messrs. Galton and Westinghouse, in the year 1878, and reported to the Institution of Mechanical Engineers the same year. These results were published by the Westinghouse Air Brake Company in 1894, and in the preface the publisher states that "the striking characteristic of the tests is that the friction is greatest when the wheels are just revolving, and that at consecutively increased speeds the friction becomes constantly diminished, but at a less rapid rate as the speeds become greater." This variation in the coefficient of friction f is approximately represented by the formula

$$f = \frac{.326}{1 + .03532\ V} \quad \ldots\ldots (78)$$

where V is the speed in miles per hour. This equation gives the following values:

V =	0	10	20	30	40	50	60
f =	.326	.241	.191	.158	.135	.118	.105

It must be borne in mind that the tests here represented were conducted with cast-iron brakeshoes and steel tired wheels.

It was also found that for constant speed the friction diminished as the time of rubbing of the brakeshoe upon the wheel is extended. If f = the coefficient of friction for a given speed at the time when the brakeshoe is first applied to the wheel, as found by equation 78, then at t seconds afterward the coefficient of friction will be

$$f^1 = \frac{f}{1 + .0022 \sqrt{t}} \qquad \text{.............................} (79)$$

These figures illustrate the great variation between static and dynamic friction; that is, the friction of rest and that of motion. The effect of sand upon the rail was found to largely increase the friction, both of the shoes upon the wheels and the wheels upon the rails. The shoe friction, just before skidding, ran up to about 45 per cent of the pressure applied, when sand was used. The condition of the weather will also influence the friction of the brakeshoes, just as it will the friction of adhesion on the rails.

By far the most valuable tests made upon this subject were those organized by the Master Car Builders' Association in 1895. A special machine was designed and built for this purpose, and a long series of experiments were made upon various kinds of metal and shoes. (These reports may be found in full in the 1895 and 1896 volumes of the M. C. B. Proceedings.) The apparatus was so arranged that a diagram could be taken of each test, in which the horizontal distance represented the space traveled by the surface after the shoe was applied to the wheel, and the vertical the friction generated or "pull" of the brakeshoe upon the surface of the wheel. Fig. 69 shows the general appearance of these cards. The pressure was applied at a, and continued as designated by the solid line, until the wheel came to rest at b, the distance a b being the length of the stop due to the friction of the shoe. The broken line shows the speed of the wheel at each instant during the stop, being greatest at a, the commencement of the test. This speed drops uniformly until near the point of stopping, as would be expected by the nearly uniform friction, but as the wheel approaches a low rate of speed, the friction suddenly increases, and the wheel is quickly

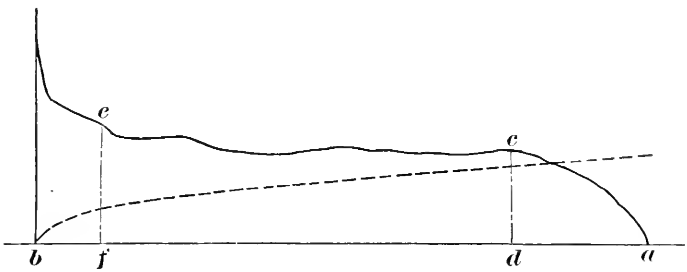


Fig. 69.

brought to a standstill. All the diagrams bore a marked similarity to each other, though, of course, the length of stop and the amount of friction were different, but they uniformly showed a great increase in the coefficient just before stopping. Some gave evidence of a reduction of friction about the middle of the "stop," while others gradually increased, as would be expected from formula 78, although it will be remembered that equation 79 demonstrated a falling coefficient during the application. In making a study of these cards, the coefficient of friction was determined and stated for three conditions:

The average coefficient throughout the length of the stop, which would be the area of the card divided by the length a b.

The initial coefficient, which was taken to be the highest value obtained at a point shortly after the shoe was applied, which would be represented by the height c d.

The final coefficient, which was taken at a point 15 feet from the end of stop, designated in Fig. 69 by the height e f, located at a distance b f, equivalent on the scale of length to 15 feet.

The tests were made starting from different speeds, and with various shoe pressures, on steel tired and chilled cast-iron wheels.

The following table is a part reproduction of the 1895 report. The character of the shoe is given in the first column, and next the reference or text letter, then the pressure of the shoe upon the wheel, the kind of wheel, chilled or tired, the speed in miles per hour when the shoe was applied, and the three coefficients of friction as explained above, in per cents. The wear of the shoes relatively to the soft cast-iron or A shoe, was determined by service tests under passenger cars, and is also given, both upon chilled iron and steel tired wheels. It was found that the soft pressed steel and wrought-iron shoes were very hard on steel tired wheels, and had a tendency to ruin the tires; so much so that in some cases the test had to be discontinued in order to save the wheels:

TEST OF VARIOUS BRAKE SHOES BY M. C. B. COMMITTEE IN 1895.

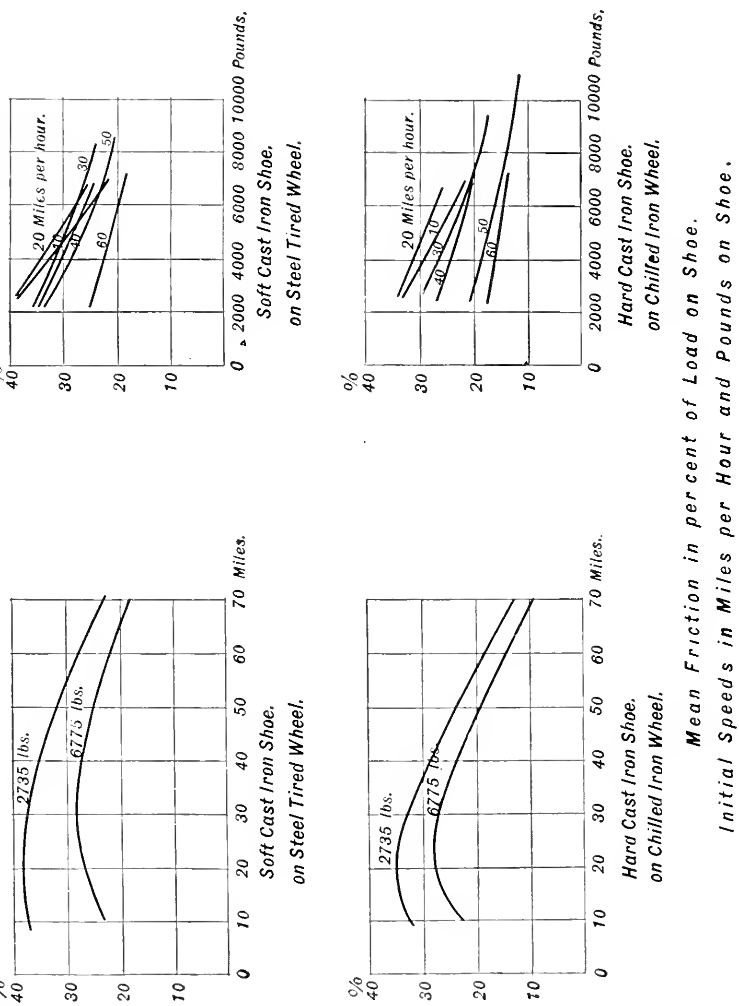
Kind of Shoe.	Letter.	Pressure,	Wheel.	Speed.	Avg. f.	Initial f.	Final f.
Soft cast iron.....	A	2,798	Chil.	40.0	31.3	31.8	42.1
Hard cast iron...	B	"	"	40.4	20.3	26.1	36.6
Soft O. H. steel...	C	"	"	40.2	17.1	20.5	29.5
Hard O. H. steel..	D	"	"	39.7	16.3	20.0	29.2
Malleable iron....	E	"	"	39.9	19.6	21.0	35.5
Congdon..........	H	"	"	40.4	20.3	27.5	31.3
Meehan...........	I	"	"	40.4	16.0	22.8	25.8
Lappin...........	J	"	"	40.4	16.9	18.4	32.3
Safety...........	K	"	"	40.0	29.4	33.0	37.3
Soft steel (pres'd)	L	"	"	40.0	19.8	22.4	32.6
Wrought iron "..	M	"	"	40.5	20.1	22.1	31.2
Sargent special...	N	"	"	40.5	19.5	21.2	28.4
Soft cast iron....	A	10,733	"	63.9	14.9	16.1	21.3
Hard cast iron....	B	"	"	64.7	10.9	12.3	16.6
Soft O. H. steel...	C	"	"	64.7	9.8	10.3	18.6
Hard O. H. steel..	D	"	"	64.8	9.1	9.8	16.2
Malleable iron....	E	"	"	64.4	9.1	9.3	15.2
Congdon..........	H	"	"	63.7	10.3	10.1	18.1
Meehan...........	I	"	"	64.6	8.9	9.9	16.2
Lappin...........	J	"	"	64.3	8.5	8.5	14.3
Safety...........	K	"	"	64.4	16.8	16.3	24.9
Soft steel (pres'd)	L	"	"	61.4	9.5	8.9	18.6
Wrought iron "	M	"	"	64.7	11.7	11.6	20.8
Sargent special .	N	"	"	64.5	11.2	13.1	17.4
Soft cast iron.....	A	"	Tired.	64.8	9.9	12.4	18.7
Hard cast iron....	B	"	"	61.9	8.5	9.5	16.2
Soft O. H. steel...	C	"	"	65.0	11.0	10.4	22.0
Hard O. H. steel..	D	"	"	65.3	11.0	10.2	21.6
Malleable iron....	E	"	"	65.0	8.5	9.4	15.5
Congdon..........	H	"	"	64.6	8.8	9.6	15.5
Meehan...........	I	"	"	64.9	8.4	9.1	16.4
Lappin.	J	"	"	64.3	8.1	8.6	15.8
Safety............	K	"	"	64.7	10.8	11.5	17.4

WEAR OF BRAKESHOES RELATIVELY TO "A" SHOE.

Kind of Shoe.	Let.	Chil.	Tired	Kind of Shoe.	Let.	Chil.	Tired
Soft cast iron	A	1.00	1.00	Meehan.................	I	.31	.21
Hard cast iron	B	.86	1.06	Lappin..................	J	.55	.27
Soft O. H. steel.........	C	.17	.12	Safety..................	K	.70	.77
Hard O. H. steel.	D	.10	.31	Soft steel (pressed)....	L	.11	.29
Malleable iron.........	E	.53	.51	Wrought iron (pressed)	M	.11	.29
Congdon	H	.31	.30	Sargent special.........	N	.22	.33

In 1896 a further report was rendered, in which the influence of unit pressure upon the coefficient of friction was shown. A large number of diagrams was presented illustrating the decreased average coefficient which attended an increase

Plate 21.



in pressure upon the shoe, the pressures ranging from 2,800 to 10,730 pounds, corresponding to from about 60 to 240 pounds per square inch of projected bearing area. An approximate ratio of the drop in the coefficient, due to increase in unit pressures is expressed by the following equation, in which

f'' = coefficient of friction (in per cents) with increased pressure.

f = coefficient at 60 pounds per square inch load.

p = pressure per square inch of surface.

b = a constant, of value about .04 for 65 miles an hour and .06 for 30 miles an hour.

then

$$f'' = f - b\,(p - 60) \qquad \text{(80)}$$

In 1900 Mr. R. A. Smart presented a paper to the Western Railway Club on the brakeshoe tests at Purdue University, and plate 21 is a reproduction of the diagrams, showing graphically the change in the coefficient of friction due to variation in speed and pressure, for soft cast-iron shoes upon steel tired wheels and hard cast-iron shoes upon chilled wheels.

In 1901 the results of further tests made by the standing committee were reported, the principal figures being as below:

PER CENT FRICTION OF BRAKESHOES FROM 1901 M. C. B. REPORT.

Name of Shoe.	A	A′	B	B′	C	C′	D	D′	E	E′
Lappin	18.68	29.02	12.90	20.13	26.95	31.01	25.16	31.85	22.51	27.92
Streeter	18.88	28.85	13.41	20.32	17.39	21.22	16.45	22.00	15.69	19.05
Corning	12.05	25.47	10.65	19.67	16.60	27.10	12.65	20.63	11.83	18.27
Herron	17.64	28.85	11.79	19.49	19.10	29.35	18.17	26.13	15.83	21.49
Cardwell	17.05	29.77	11.68	20.69	25.56	33.95	25.81	31.39	21.89	26.57
Ideal	11.38	20.20	10.93	15.83	20.08	28.87	17.01	25.30	15.29	18.90
Sargent C	15.31	31.61	12.02	19.74	17.68	29.68	15.94	31.84	14.85	25.04
Composite	16.73	27.78	12.79	19.40	28.31	33.03	25.83	30.61	23.44	27.58

In this table A is the coefficient of friction obtained with the steel tired wheel at 65 miles per hour and with a shoe pressure of 2,808 pounds, and B with 6,840 pounds. C is for 40 miles an hour initial speed upon chilled iron wheels, with a pressure of 2,808 pounds; D, 4,152 pounds, and E, 6,840 pounds upon the shoe. The simple letters indicate the mean or average friction for the whole stop, and the prime letters (A′, B′, etc.) the final friction, as before explained. Some tests of the "Diamond S" shoes, put up in soft and hard cast iron, were made against the same metal, without the expanded steel strips, the hard iron shoes being thought of the same degree of hardness as the B shoes of the 1895 tests. The columns mean the same as in the last table, except that all speeds are at 40 miles an hour.

COMPARATIVE TESTS OF SHOES WITH AND WITHOUT EXPANDED METAL STRIPS, IN PER CENTS.

Shoe.	A	A'	B	B	C	C	D	D	E	E'
Soft iron, with strips	25.1	34.1	20.3	23.1	24.1	33.2	....	....	20.9	30.2
Soft iron, without strips	23.4	35.6	18.1	27.0	25.0	39.7	....	....	23.1	31.8
Hard iron, with strips	17.2	29.3	13.2	22.1	21.7	34.8	....	....	17.7	28.7
Hard iron, without strips	15.8	30.4	12.1	22.6	16.5	30.0	....	....	13.8	26.8

The mean coefficients of friction of the hard "Diamond S" brakeshoes were higher than those of the plain shoes from the same metal, but in the case of the soft metal shoes the expanded strips do not appear to materially affect the friction one way or other.

In 1901 the Master Car Builders' Association adopted the following specifications for brakeshoes:

"Shoes when tested on the Master Car Builders' Association testing machine, in effecting stops from an initial speed of 40 miles per hour for chilled iron wheels and 65 miles per hour for steel-tired wheels, shall develop upon the test wheel a mean coefficient of friction of not less than

22 per cent when brake shoe pressure is 2.808 lbs.		
20 per cent when brake shoe pressure is 4.152 lbs.	}	For chilled iron wheels.
16 per cent when brake shoe pressure is 6.840 lbs.		
16 per cent when brake shoe pressure is 2.808 lbs.		
14 per cent when brake shoe pressure is 4.152 lbs.	}	For steel tired wheels.
12 per cent when brake shoe pressure is 6.840 lbs.		

In a paper read before the New England Air Brake Club in August, 1902, Mr. F. W. Sargent drew the following conclusions from the Master Car Builders' tests:

"First—The softer cast-iron shoes show a greater retarding power on the chilled wheel than on the steel tire.

"Second—The composite shoes (hard and soft metal and inserts) show a rise or fall on the two kinds of wheels, depending upon the character of the inserts, those having well-defined cutting edges taking a high stand on the steel tire by reason of tire dressing, and those which do not cut taking a lower place, depending upon their relative hardness.

"Third—The very hard and heavily chilled shoes occupy practically the same position on both the steel tire and chilled wheel."

He further on states that the important point in selecting a brakeshoe is to draw the line between friction and durability. Work and wear go together, and to stop the wheel something must be worn, either the shoe or the tread of the wheel, or both. In locomotive driving wheels the action of the piston is to slip and wear away the tire where it bears upon the rail. This necessitates a shoe that will rub the tire where the rail does not, and that will cut away the flange and outer part of tread, thus preventing the hollow tread so destructive to frogs and crossings. This is the function of the hard steel inserts—to turn or dress the tire with each stop in about the same rate that the rail wears it, maintaining the original profile. This cutting causes great resistance and gives a splendid hold upon the drivers in making a stop. But tire metal should not be removed unnecessarily—that is, more than sufficient to keep the treads in good condition. Cast-steel shoes are excellent for this purpose, as well as the dressing inserts.

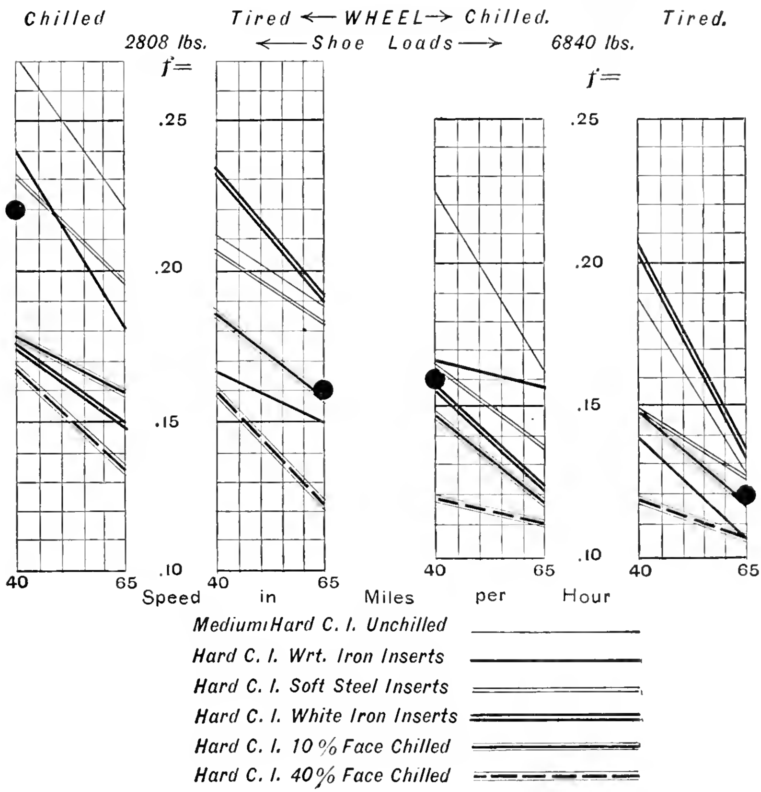
For the truck and tender wheels, which are never slipped, but are slid only by the action of the brakeshoes, tire dressing shoes are out of the question, as the rail wear is generally small, the flanges causing the most of the trouble. The shoes should be selected to suit the kind of wheel employed. Soft steel or iron inserts are usually very hard upon steel-tired wheels in truck service, but give good results upon chilled wheels. It is necessary also to obtain a good proportion of wear out of the shoes before scrapping, and a reinforced shoe with a steel back to insure strength even when worn thin is essential. Driver brakeshoes should give considerably more than 50 per cent wear, and car or truck shoes over 75 per cent. A shoe has even been produced which can be absolutely worn entirely out, it being so designed that when thin it is backed by a new shoe, and allowed to wear itself into the new shoe without being removed.

Plate 22 is taken from Mr. Sargent's paper and produced in a slightly different form. It shows the effect of speed upon the coefficients of six different types of shoes. The Congdon is an example of hard cast iron with wrought iron insert; the Diamond S with soft steel insert; the Streeter with white iron insert; the Sargent U with 10 per cent of face chilled, and the Lappin and Corning with 40 per cent of face chilled. The Master Car Builders' limits are shown by the solid black circle.

CENTER PLATE AND SIDE BEARING FRICTION.

The importance of this subject has been brought out only within the last few years by means of laboratory tests and

Plate 22.



trials upon the road. The old style of center plates and side bearings were perhaps as crude as anything could well be. The desire to haul greater tonnage has caused the different resistances to be more thoroughly investigated, and as center plate and side bearing friction reduces the load that can be hauled around curves, it is worthy of careful consideration. If railroads were straight, there would be no friction of center plates, as there would be no curves to traverse. If side bearings maintained the clearances given them when cars are built, this friction would be avoided, but the deflection of the body and truck bolsters will often allow the side bearings to come together when the car is heavily loaded, and even if this does not occur on straight track, the centrifugal force will throw the bearings on the outside of the curve in contact if the speed be at all great. The latter condition shows us that no matter how stiff the bolsters may be, the side bearings will at times take a load, and that time will be when it is most desirable to reduce the side bearing friction. If a constant load be permitted to rest upon the side bearings, there will not be so great a bending load upon the bolsters, and the swaying motion of cars with high center of gravity when passing through curves will be eliminated. This, however, means a greater frictional moment, or resistance to curving, because the lever arm of the friction will be greater. If, however, we reduce the coefficient of friction proportionately, we can permit the longer arm without apprehension.

The ordinary method of reducing friction is by lubrication, but the difficulty of access and inspection almost prevents the continuous lubrication when in service. Many seem to be satisfied with a lump of grease placed upon the bottom plate and side bearings when the car is built, never expecting to renew it. The interchange of cars is also not encouraging to add expense of construction for the benefit of other lines.

In 1890 the Dayton Malleable Iron Company produced a center plate with an oil cavity, and a series of tests showed that the lubricated plates required about one-fourth as much power to cause rotation as the dry plates. When loaded with 10,000 pounds, the dry plates required a moment of 1,488 foot pounds and the lubricated plates 350 foot pounds to start rotation; that is, 350 pounds at a lever arm one foot long.

Perhaps the first comprehensive service test of roller center plates and side bearings was made by the Pittsburg & Lake

Erie Railroad with the Hartman devices on freight cars These have been in use for five or six years on the road mentioned with evident satisfaction, the practically absolute absence of flange wear upon the wheels during this time bearing testimony of the elimination of curve friction. The showing in this way is quite remarkable, and, in fact, is almost incredible.

A rotating test was made upon a car that had been in service for over three years, in comparison with a flat center plate and side bearings. The power required to start rotation in percentages of the plain plate is shown below:

Flat center plate and side bearings, with ⅛-inch deflection of bolster on side bearing	100 per cent
Flat center plate without side bearings	34 per cent
Hartman ball bearing center plate and side bearings, bolster load as above.................	9 per cent
Hartman center plate without side bearings.....	9 per cent

In 1900 a committee reported to the Master Car Builders' Association certain tests which they had made with roller side bearings in comparison with plain bearings. A car was dropped down a 4 per cent grade for 125 feet, when it entered a short 15-degree curve, followed by a level tangent. The average distances run on the tangent in the several tests were as follows:

Side bearings not touching......................	345 feet
Side bearings sustaining weight...................	197 feet
Roller bearings sustaining weight.................	345 feet
Roller bearings not touching by ⅛ inch............	311 feet

As might be expected, the roller bearings reduced the friction over the plain bearing—in fact, the resistance was the same as when the bearings were not in contact, the resistance evidently coming from the center plates. These were of the ordinary type, so it is not known how far the car would have run had it been equipped with a ball center plate.

The Master Car Builders' committee in 1903 made an elaborate report on the subject, which gives us the most complete information to date. In these tests the committee endeavored to determine the best material, the best condition for a given metal, the effect of lubrication, the best shape, the best size and the relative values of special designs. The metals tested were grey iron, chilled iron, malleable iron, cast steel and pressed steel, and the conditions of finish were rough, smoothed and roughly fitted on the emery wheel, and machined in a lathe. The lubricant used was a thick brown grease made by the Galena Oil Company. Flat and spherical plates were tested for shape, and also ball bearing plates and ball and roller side bearings. For the best size, plates of five different areas of bearing were used.

From the results of the tests the committee concluded that the best shape had a flat bearing surface 11¾ inches outside diameter and 3¼ inches diameter at center, the bottom plate having a center tube 3⅛ inches in diameter extending upward into the central hole of the upper plate 1½ inches. The outer flange of the bottom plate was ⅛ inch larger in diameter than the upper plate, or 11⅞ inches, and projected upward 1½ inches also, permitting ribs of this depth to support the horizontal portion. The area of contact faces was 100 square inches. This plate was later adopted as the standard of the association. The ball bearing center plates and side bearings gave such remarkable results that the committee thought there was no doubt of the reduction of flange friction by their use if found durable. As above noted, the Pittsburg & Lake Erie Railroad has used them for half a dozen years, and reports entire satisfaction; there is certainly a great field of usefulness for them.

The following table gives the data from the principal tests, the flat center plate referring to the Master Car Builders' plate explained above.

Friction of center plates: pull in pounds on lever arm 33 inches long = wheel flange leverage:

Center Plate.	Metal.	Finish.	Lubrication.	Tons Load on Plate. 10	20	30
M. C. B. flat	Mal. iron	Rough	None	600	1,300	2,100
			Lub.	150	350	600
		Smooth	None	300	850	1,400
			Lub.	150	400	700
		Machined	None	500	1,800	3,000
			Lub.	300	600	1,000
M. C. B. flat	Cast iron	Rough	None	1,300	1,900	2,600
			Lub.	300	550	800

Center Plate.	Metal.	Finish.	Lubrication.	Tons Load on Plate.		
				10	20	30
M. C. B. flat	Cast iron	Smooth	None	200	500	900
			Lub.	200	400	600
		Machined	None	200	500	700
			Lub.	200	500	700
M. C. B. flat	Chilled iron	Rough	None	800	2.200	4,000
			Lub.	100	200	500
		Smooth	None	150	400	700
			Lub.	150	300	500
M. C. B flat	Cast steel	Rough	None	1.200	2.900	.....
			Lub.	300	700	1.300
		Smooth	None	200	600	1.100
			Lub.	200	600	900
		Machined	None	300	800	1.400
			Lub.	500	800	1.200
Flat	Pressed steel	Rough	None	800	1.900	3.400
			Lub.	200	400	700
Hartman ball bearing	....	....	None	100	400	450
Baltimore ball bearing	....	....	None	50	100	200

The value of lubrication is very evident from these tests, the friction being at times only one-tenth that of the dry plate. The balls in the Hartman plate roll in a pocket which is deeper at the center, causing a slight lifting of the car as the balls roll "up hill," which probably accounts for the higher values than shown by the Baltimore plate, where the path is level. The value of ball or roller plates is evident from the figures.

When the side bearings were tested they were set 25 inches from a pivot center provided to eliminate center plate friction from the side bearing tests.

Friction of side bearings: pull in pounds on lever arm 33 inches long = wheel flange leverage.

Side Bearing.	Metal.	Finish.	Lubrication.	Tons Load.	
				10	20
Flat	Cast iron	Rough	None	4.900	....
			Lub.	1,300	3.500
"Frictionless" roller	....	....	None	300	800
"Baltimore" ball	....	....	None	200	1.500

The value of an anti-friction device is very clear. The loads of 10, 20 and 30 tons correspond approximately to those found in the center plate of a loaded car of 30, 40 and 50 tons capacity, with the side bearings standing clear.

TRAIN RESISTANCE.

This constitutes the total work of the engine, as all its power is utilized in overcoming the resistance of itself and the train to which it is attached. In order therefore to be able to state the equation between power developed and work accomplished, we must know the values of the various components of train resistance. We have discussed the internal resistance of the engine itself, as an engine, but we must still consider the power needed to move it as a car, as well as the tender, and the train to which it is attached. Under this caption, therefore, we will study the resistance due to journal friction, wind resistance (these two are generally taken together and classed as "resistance due to speed"), force necessary to overcome gravity in ascending grades, force necessary to pass around curves, effect of maximum, intermediate and minimum loading, and also that due to weather, temperature, etc.

JOURNAL RESISTANCE.

In our study of journal resistance we have seen that the coefficient of friction at very low speeds was from .09 to .12, or, say, .1 of the load, whereas it was taken at .02 for speeds of 5 miles an hour and upward.

The average diameter of car journals is probably about 4 inches, and as 33-inch wheels are very common under freight equipment, the ratio of wheel to journal diameter is approximately 8. The force necessary to start a ton of load would be therefore

$$\frac{.1 \times 2000}{8} = 25 \text{ pounds}$$

and to maintain a speed of 5 miles an hour

$$\frac{.02 \times 2000}{8} = 5 \text{ pounds}$$

Mr. J. A. F. Aspinall, in his paper read before the Institution of Civil Engineers in 1901, described some tests which he had conducted to determine the starting resistance of trains. These were made by finding upon what incline the cars would start themselves. He found that on a ¾ per cent grade, with a very slight wind blowing against the train, there was no movement. When the wind was blowing at right angles to the train with a velocity of 6 miles an hour, it just moved. With a 9½-mile breeze nearly with the train, it started more readily. From this it appears that the train tested required about 15 pounds per ton (the tractive effort of gravity on a ¾ per cent grade) to put it in motion.

In the discussion of this paper Mr. P. V. McMahon mentioned some tests in which the resistance in starting had been found to be 20 or 25 pounds per ton. A similar experiment with a locomotive gave a resistance of 25 to 30 pounds per ton. The Master Mechanics' committees have shown in their diagram of train resistance a starting force necessary of about 18 pounds per ton. Wellington gives this value at from 14 to 18 pounds, with considerable fluctuations. He states that most of this initial resistance is almost wholly instantaneous, and consumes little power, but that the normal axle friction is probably increased 2 pounds per ton for the first few car lengths.

Mr. B. A. Worthington in a paper recently indicated 17 pounds per ton. These figures are not intended to cover inertia, as they represent slow starts. It is apparent that the resistance reduces immediately the train moves, and also that the slack in couplers or compression in draft springs tends to reduce the pull on the engine by permitting the train to start one car at a time, as it might be termed, because locomotives will start trains on grades that are given them considering the low friction of 5 miles an hour, and which could never be started if all the cars were rigidly coupled together. On account of this fact, the starting resistance is seldom used in making up engine ratings, but questions at times arise which make it an important consideration. The profiles of a number of "Hump" freight yards, whose cars are sorted by gravity, show from 1¾ to 3 per cent grades at the summit, tapering off to much lower rates after the point has been reached where the car will be in motion, some of these lower inclines being only .3 per cent, which corresponds to 6 pounds per ton.

WIND RESISTANCE.

Many experiments and theories have been pursued in the endeavor to effect a satisfactory solution of this problem, but there is considerable discrepancy between the results. One of the earliest formulæ for train resistance was that of D. K. Clark, which was written $R = 8 + \frac{V^2}{171}$, in which V = velocity in miles per hour, and R = resistance per long ton (2,240 pounds) in pounds. Reduced to the 2,000-pound ton, as generally observed in American practice, this would read

$$R = 7.2 + \frac{V^2}{188} = 7.2 + .0053\ V^2$$

We observe that this is composed of a constant factor, regardless of the speed, and one that depends upon the square of the velocity. However, it does not include any factor that would be governed by the end surface exposed, as we should ordinarily expect a train of box cars to cause a greater wind resistance than a train of flat cars.

The pressure of air in motion or wind against a vertical surface is not clearly settled, as different experiments have produced widely different results. If we apply the rules of hydraulics to the problem, we get a result that agrees with some of the experiments. The well-known formula $v = \sqrt{2gh}$ can be used, where v = velocity in feet per second and h = head in feet, by substituting for "h" the value $\frac{P}{G}$, where P = pressure in pounds per square foot of surface, and G = weight of a cubic foot of air = .076 pounds, for we readily see that P = G h, and if $v^2 = 2.15\ V^2$ (as we found in the development of equation 1), we have

$$v^2 = 2gh = 2g\frac{P}{G} \text{ and}$$

$$P = \frac{G v^2}{2g} = \frac{.076 \times 2.15\ V^2}{2 \times 32.2} = .0025\ V^2 \quad \ldots\ldots\ldots\ldots \quad (81)$$

If we consider that the end area of a box or passenger car is about 100 square feet, and averages 45 tons loaded, we have

$$\frac{.0025\ V^2 \times 100}{45} = .0055\ V^2$$

for the resistance per ton due to wind pressure, which is not far from Clark's formula.

Mr. Aspinall, in the paper referred to above, estimated the pressure in pounds per square foot at

$P = .003\ V^2$ . . . . (82)

although he stated that it appeared rather high, and referred to the value found by Prof. Nipher where $P = .0025\ V^2$, the same as equation 81.

Prof. Goss, after a series of elaborate experiments at Purdue University with model cars in an air duct evolved the following values for wind resistance for the different members of a passenger train:

Locomotive	$.11\ V^2$
First coach	$.001\ V^2$
Second coach	$.00008\ V^2$
Intermediate coaches, each	$.0001\ V^2$
Last coach	$.00026\ V^2$

In France, on the Paris, Lyons & Mediterranean Railway, some of the fast trains have been fitted with "wind cutters" shaped in a measure like a modern snowplow, and placed on the engine, and the openings between the cars were reduced by close vestibules. These are said to have saved 10 to 15 per cent of fuel after six months' operation, as compared with similar engines in the same work, but not fitted with the shields. Tests of atmospheric resistance on bodies of various shapes indicated the following relative resistance:

Flat surface	100 per cent
Cone, apex foremost	42 per cent
Double cone, base to base	25 per cent

Side winds are, as a rule, much harder on the pulling power of the locomotive than head winds. If a train running 40 miles an hour is at the same time pushing against a 40-mile wind in the opposite direction, the resultant wind pressure is equivalent to a speed of 80 miles an hour. However, side winds, while not susceptible of close computation, cause flange friction on the leeward side which greatly increases the work to be performed by the engine; in fact, on the western prairies it is not unusual for a long freight train to be almost, if not entirely, stalled by a strong gust of wind on the quarter. Several years ago a passenger train on a meter gauge railway in India, with 14 cars, had quite a disastrous experience in a gale blowing 60 or 70 miles an hour. It had been running at fairly good speed under shelter, but as it came to an open plain it entered a curve, and, meeting the wind "end on," its speed became continuously slower. By the time the train reached the following tangent, with the wind on the quarter, it came to a dead stop, and was rolled over bodily.

These facts illustrate the importance of carefully considering the effects of wind resistance.

MISCELLANEOUS RESISTANCES.

The journal friction and wind resistance do not constitute the entire obstruction to the speed of the locomotive. Flange friction, oscillation, concussion, etc., cause the expenditure of a considerable amount of power. Wellington considers that these items are responsible for two-thirds of the velocity resistance. Aspinall figures them at about 55 per cent of the resistance due to speed or 50 per cent of the total resistance of the train at 80 miles an hour. In a 20-car train, these resistances figure about

$$\frac{V}{4.84} - 2$$

and for a five-car train

$$\frac{V}{5.4} - 1.$$

As these resistances which are not well understood, are fully as great as those due to wind and journal friction, it seems illogical to attempt to set an accurate figure on the latter, when we must guess at the former. On this account, the great majority of formulæ in current use are framed to cover all these resistances, and ordinarily consist of a constant, representing a uniform journal resistance, and a variable, which takes care of the atmospheric and miscellaneous resistances. These will now be taken up.

SPEED RESISTANCE.

As previously stated, the formula of D. K. Clark was advanced many years ago, and until recently was very generally observed. Reduced to American tons of 2,000 pounds, it stands

$$R = 7.2 + .0053\ V^2$$

V being speed in miles per hour and R being resistance in pounds per ton (2,000 pounds).

A. M. Wellington gives several values which he deduced for trains of various kinds, thus:

For 20 loaded box cars, $R = 4 + \frac{V^2}{130}$,

For 40 empty box cars, $R = 6 + \frac{V^2}{106}$,

For 20 loaded flat cars, $R = \frac{V^2}{113}$,

For 40 empty flat cars, $R = \frac{V^2}{81}$.

J. A. F. Aspinall, as the result of his experiments, which were carefully conducted, proposes

$R = 2.5 + \frac{V^{\frac{5}{3}}}{50.8 + .0278\ L}$, or, in tons of 2,000 pounds,

$R = 2.25 + \frac{V^{\frac{5}{3}}}{56 + .03\ L}$, where L = length of train in feet over coach bodies.

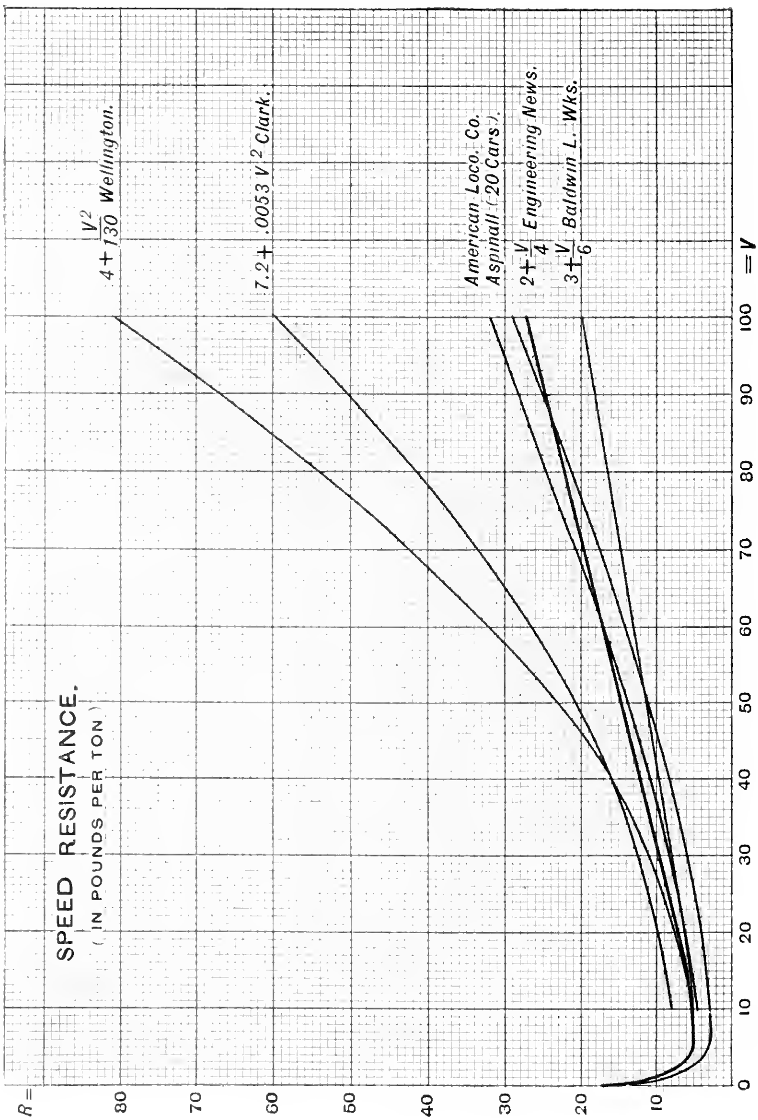
Prof. R. H. Smith recommends (in American tons)

$$R = 2.25 + \left\{1.8 + .0032\ L - \frac{180}{100 + 1.1\ W}\right\} V^{\frac{5}{3}}$$

W being the weight of trains in tons.

The rules used in this country are much simpler, and as the case is such that exact values are impossible, it seems perfectly rational to follow the easier formulæ. Several years ago the Engineering News suggested

Plate 23.


SPEED RESISTANCE.
(IN POUNDS PER TON)
$4 + \frac{V^2}{130}$ Wellington.
$7.2 + .0053\ V^2$ Clark.
American Loco. Co.
Aspinall (20 Cars).
$2 + \frac{V}{4}$ Engineering News.
$3 + \frac{V}{6}$ Baldwin L. Wks.
R =
80
70
60
50
40
30
20
10
0
10
20
30
40
50
60
70
80
90
100
= V

$$R = 2 + \frac{V}{4},$$

which corresponded with tests made by Mr. Angus Sinclair on the New York Central & Hudson River Railroad. The Baldwin Locomotive Works are guided by

$$R = 3 + \frac{V}{6}.$$

The most important of those given above are represented graphically on plate 23. These can be used for the locomotive and tender, as well as for cars, allowing, of course, for the internal friction of the engine, in addition to the resistance for speed, as determined from the plate. The Southern Pacific uses the Wellington formula or curve, in rating freight engines, up to 35 miles an hour.

The author has preferred to follow the Engineering News formula

$$R = 2 + \frac{V}{4} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (83)$$

and the corresponding curve will be used in this treatise. It starts with about 16 pounds per ton, drops to five pounds at 6 miles an hour and then increases uniformly in accordance with equation 83. This curve is indicated by a heavier line than the others.

INERTIA RESISTANCE.

This has been studied in the first chapter, and will only be referred to here. Equations 1 and 2 can be used for cars as well as locomotives, and plates 1 and 3 give the various values without calculation. These are in pounds per ton, and can be directly added to the speed resistance in order to produce the total. If a change in velocity is to be considered equation 3 should be used. If the speed is to be increased, the force due to inertia must be overcome by the power of the locomotive, but if the speed is to be reduced, the locomotive will be assisted, or the brakes will have to be brought into use. In the first case, the force of inertia is to be added to the speed resistance, and in the second case, subtracted from it, in order to obtain the total train resistance.

As an example, a train of 1,000 tons weight, on a straight, level track, running at 30 miles an hour, will have a resistance or pull on the engine of $1{,}000 \times 9\frac{1}{2} = 9{,}500$ pounds ($9\frac{1}{2}$ being $2 + \frac{30}{4}$ or as found from plate 23). If, however, the same train is brought from rest to 30 miles an hour in two minutes or 120 seconds, the pull due to inertia alone will be $1{,}000 \times 24 = 24{,}000$ pounds, as found by plate 3, or equation 2.

GRADE RESISTANCE.

If we except the effects of inertia, we may say that grade resistance is the only one which is susceptible of accurate computation. There can be but one value for the force necessary

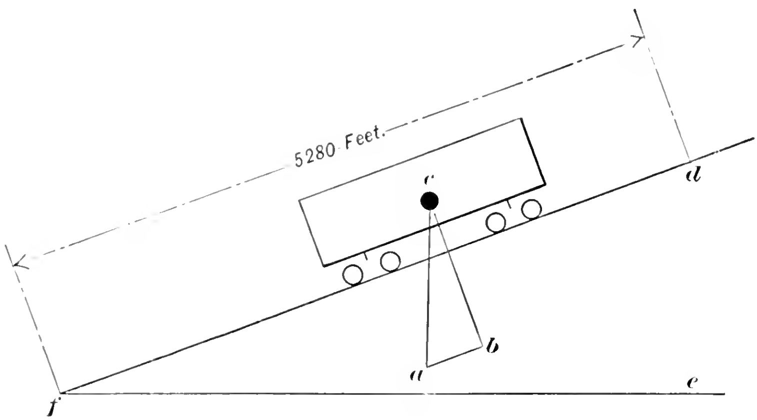


Fig. 70.

to overcome a given grade, although the effects of inertia, due to a reduction in the speed, will apparently decrease the normal resistance. It is due to this certainty that calculations for engine rating on heavy grades are always made with less chance of error than when the work is to be performed on light grades or levels—the latter is extremely uncertain, as plate 23 has demonstrated. In Fig. 70 let d f represent a mile of track, in which distance the rise is d e in feet. Consider that the car shown weighs 1 ton = 2,000 pounds, represented by the vertical force line a c. Resolving this force into two rectangular forces a b parallel with the track, and b c normal to it, we have the force a b as the amount necessary to move the car up the grade, considering gravity only. Now the numerical value of a b is 2,000 sin $\overline{a\ c\ b}$, but the angle $\overline{a\ c\ b}$ is equal to the angle $\overline{d\ f\ e}$, so that a b = 2,000 sin $\overline{d\ f\ e}$.

But from the figure we see that sin $\overline{d\ f\ e} = \frac{d\ e}{d\ f}$ or the rise in feet in one mile = m, divided by 5,280, number of feet in a mile, so that a b, which is the resistance to gravitation, and will be designated by R_g, becomes

$$\overline{a\ b} = R_g = \frac{2{,}000\ m}{5{,}280} = .38\ m \quad \ldots\ldots\ldots\ldots (84)$$

If the grade be expressed in percentages, as is common in this country, then, when m_{pc} = the grade in per cent, we have

$$m_{pc} = \frac{m}{5{,}280} \times 100 \text{ and } m = \frac{5{,}280}{100} m_{pc}.$$

Now, substituting this value in equation 84, we obtain the resistance

$$R_g = \frac{2{,}000}{5{,}280} \times \frac{5{,}280}{100} m_{pc} = 20\ m_{pc} \quad \ldots\ldots\ldots\ldots (85)$$

Where R_g = grade resistance in pounds per ton.

When the benefits of momentum or inertia are available, we have what is termed a "virtual grade," and which is less or greater than the actual grade, depending upon whether the speed of train is being retarded or accelerated. This can best be illustrated by an example.

Suppose a train approaches an up grade at a speed of 40 miles an hour; that is, at "a." The grade is 1½ per cent, and is 7,000 feet long; then in Fig. 71 a b will be 7,000 feet, and the height ascended b c will be 7,000 × .01½ = 105 feet. If the train be permitted to reduce its speed gradually to 5 miles an hour at the summit "b," we find from formula 3 that the force developed to assist the engine by the reduction of speed will amount to $70 \frac{1{,}600 - 25}{7{,}000} = 15.75$ pounds per ton. The same value is found in plate 2, at the intersection of the 40-mile curve and the 7,000-foot line. From equation 85, the normal resistance due to a $1\frac{1}{2}$ per cent grade is $= 20 \times 1\frac{1}{2} = 30$ pounds, and $30 - 15.75 = 14.25$ pounds, which will be the pull required of the engine, per ton of train; this is the normal pull on

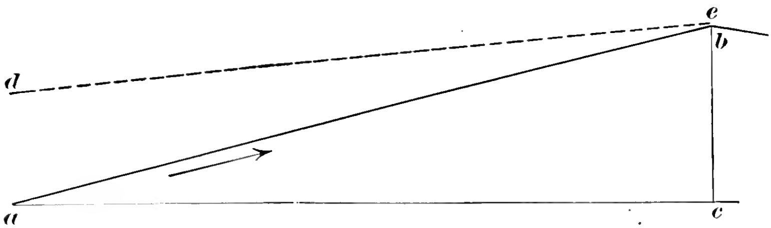


Fig. 71.

a .712 per cent grade, and the virtual grade is, therefore, .712 per cent.

This may be explained in another way. From the table given in connection with equation 4, we find the velocity head at 40 miles an hour to be 56 feet, and at 5 miles, .875 feet. The head represented by the drop in speed is $56 - .875 = 55.13$ feet. This can be deducted from the actual rise b c, or 105 feet, so that the engine must perform the equivalent of lifting the train $105 - 55.13 = 49.87$ feet. But as it travels 7,000 feet in so doing, the virtual grade is $49.87 \div 7{,}000 = .00712$, or .712 per cent. This is shown graphically in Fig. 71, by laying off to the scale of the figure the height a d = 56 feet velocity head at "a," and .875 feet at b e, the velocity head at b, and connecting them by a straight line; thus, while the line a b represents the actual grade, the line d e represents the virtual grade under the conditions of speed assumed in our proposition.

CURVE RESISTANCE.

There is considerable uncertainty about the actual resistance of trains in passing around curves. Wellington states that it may vary between .33 pound per ton per degree of curvature, with track and equipment in perfect order and 1 or even 1.5 pounds per ton per degree of curvature, when rails are worn and track is rough. The velocity of the train also causes a change in resistance. Thus it was found that on a 1-degree curve, the resistance at 12 miles an hour was over 1 pound per ton, and at 22 miles an hour, only about .5 pound per ton. (If the curvature should be stated by giving the radius in feet, the degree of curvature can be approximately found by dividing the radius (in feet) into 5,730; thus a curve of 2,865 feet radius would also be a curve of $\frac{5{,}730}{2{,}865} = 2$ degrees. This is very nearly correct up to 10 degrees of curvature.)

In 1897 a committee of the Master Mechanics' Association recommended the resistance on curves be taken as .7 pound per ton per degree of curvature for cars, and double that, or 1.4 pounds, for locomotives. In 1899 tests were made on the Lehigh Valley Railroad, by dragging a consolidation locomotive of about 78 tons weight (exclusive of tender) through a 14-degree curve, and measuring the force required by a dynamometer. Three arrangements of flanged wheels were used; first, on the first and fourth wheels, spaced 4 feet 5¼ inches between flanges of tires; second, on the first, third and fourth wheels, spaced the same distance; third, on all wheels, the first and fourth being spaced 4 feet 5⅛ inches between flanges and the second and third 4 feet 5¼ inches. All arrangements showed practically the same power required to pull the engine at a speed of 28 miles an hour, or, after deducting the resistance due to grade and speed, about 25 pounds per ton, or 1.8 pounds per ton per degree of curvature. At 5 miles an hour the resistance ran up to about 3 pounds per ton per degree. It is thought, however, that the values assigned by the Master Mechanics' committee for locomotives, viz., 1.4 pounds per ton per degree, will ordinarily be sufficient, and in making calculations with heavy trains, it is often permissible to use the factor .7 for the total train.

When curves occur on heavy grades, as they generally do, it is considered good practice to compensate the grades for the curves; that is, to reduce the grade at the curve by such an amount that the total train resistance due to grade and curve will be no greater than the maximum grade on a tangent. If the resistance be taken at .7 pound per degree, the grade should be reduced .035 per cent per degree of curvature, or 1.84 feet per mile, per degree. That is to say, if we have a mountain grade of 1 per cent, or 53 feet to the mile, and on this grade a 10-degree curve, the grade should be reduced at the curve to 1 — .35 = .65 per cent, or 53 — 18.4 = 34.6 feet per mile. Then the train resistance in passing around the curve would be no greater than on the tangent. Of course, the average grade on the tangents, if uniform, would have to be somewhat greater on account of the loss of rise at the curve, but the advantage in train haul would still be with the compensated grade. If stopping places occur on curves, Wellington recommends a reduction of .1 per cent per degree of curvature.

If curves are short, a slight drop in the speed of the train allows them to be traversed without difficulty by the inertia, and the extra resistance is often omitted from tonnage calculations for this reason.

RESISTANCE AFFECTED BY LOADING.

We found in our discussion of speed resistance that formula 83 gave us a ready means of determining the force necessary to maintain a speed V. This equation, however, applies only to loaded cars; that is, of about 33 tons weight, and where there is a great variation, such as in freight traffic, a modification is necessary. It is true that there are passenger equipment cars weighing 50 tons or more, but these are usually provided with six-wheel trucks, which arrangement probably maintains a fairly uniform resistance. With freight cars, however, the gross weight on eight wheels may be 10 tons or 70 tons, according to the type and load, as, for instance, empty flat cars in the one case, and loaded 100,000-pound capacity coal or ore cars in the other. Such discrepancies are never met in passenger trains. As freight trains usually travel at from 5 to 15 miles an hour (unless they be stock or fast freights), especially upon heavy grades, where the question of resistance is of most importance, and where the closest figures are made for hauling capacity, it will probably be sufficient to study the effect of loading at low speeds. From plate 23 we find the resistance at this speed from 5 to 6 pounds per ton. As stated above, this is only correct for cars of about 33 tons weight, and on level track. It has long been recognized that a train of empties, pulled much harder than a train of the same weight of loads, and dispatchers who gave an engine its full rating in empties often found that it was necessary to double controlling grades. From experiments made by the late A. M. Wellington on the Lake Shore & Michigan Southern Railway in 1878, he concluded that there was an increase of 2 pounds per ton in the resistance of empty freight cars over loaded cars. Other prominent engineers consider an allowance of 30 per cent on a level to be sufficient. At 15 miles an hour, with normal resistance at 6 pounds per ton, this would make an allowance of 1.8 pounds additional for each ton of empty cars. It must be remembered that this is true only on the level. Some roads allow 25 per cent additional weight on empties uniformly, regardless of the amount of grade. This would be too liberal on a hill, and not sufficient on the level. An allowance of 1.8 or 2 pounds per ton extra on the weight of all empties would be about right, regardless of the grade, but when trains are made up the instructions state the tonnage to be given a certain class of locomotives. Under these conditions, it is more convenient to give the percentage of allowance for empties. As train resistance as a whole is composed principally of that due to speed and grade, we see at once that the allowance of 1.8 pounds will be a very much smaller proportionate increase when the grade is high than when it is low. The percentage of increase for any grade is found by dividing 1.8 by the sum of the grade resistance and the speed resistance, which we may take as 6, thus for a level, $1.8 \div 6 = 30$ per cent, and for a 1 per cent grade, $1.8 \div (20 + 6) = 7$ per cent. The following table gives the percentage of allowance on this basis.

Percentages to be added to empty weights:

Grade in feet per mile..	0	10	20	30	40	50	60
Excess percentage	30	18	13½	10½	8½	7	6½
Grade in feet per mile......	70	80	90	100	120	140	
Excess percentage	5½	5	4½	4	3½	3	

With grades steeper than 140 feet per mile the effect is so slight that it may be neglected. If the district under consideration utilizes momentum grades, the virtual grade should be used instead of the actual in selecting the percentage of increase.

The above rule makes no distinction for cars that are partially loaded—a 60,000-pound capacity car might have a 15,000-pound load—it would not be an empty, and yet if an engine were given a train of such cars with full tonnage, the additional resistance would soon manifest itself. In order to include all conditions of loading, the author has devised the following formula:

Let R_e = resistance of train in pounds, or the pull at the tender draw bar, on straight, level track, and at a speed of 10 miles an hour.

T = weight of train in tons (of 2,000 pounds).

C = number of cars in train: then

$R_e = 3.5\ T + 50\ C$ . . . . (86)

or the pull of the train at tender draw bar, on straight, level track in pounds will be the sum of 3.5 times the weight of train in tons + 50 times the number of cars in train. Thus, if there were 60 cars in a train of a total weight 1,200 tons, the average weight per car would be only 20 tons, and the resistance on a level would be

$$R_e = 3.5 \times 1{,}200 + 50 \times 60 = 7{,}200 \text{ pounds.}$$

For the same speed on a grade, we have simply to add to the coefficient of T the pounds that are needed to pull one ton up the grade in question, which can be obtained from equations 84 and 85. Thus, if the grade be ½ per cent, we have by equation 85, $R_g = 20 \times \frac{1}{2} = 10$, and $3.5 + 10 = 13.5$, so that for this grade, equation 86 becomes

$$R_{e(\frac{1}{2}\%)} = 13.5\ T + 50\ C$$

or for the same train, $13.5 \times 1{,}200 + 50 \times 60 = 19{,}200$ pounds.

It will be interesting to discover what numerical values formula 86 gives us for 16 2-3, 33 1-3 and 50 ton cars. Let us take a train of each, of 100 tons weight, and we will have 6, 3 and 2 car trains, respectively.

Then for 16 2-3-ton cars, $3.5 \times 100 + 50 \times 6 = 650$ pounds,
for 33 1-3-ton cars, $3.5 \times 100 + 50 \times 3 = 500$ pounds,
and for 50-ton cars, $3.5 \times 100 + 50 \times 2 = 450$ pounds,
or 6.5, 5 and 4.5 pounds per ton, or the empties give a resistance $\frac{6.5 - 5}{5} = 30$ per cent more than the 33 1-3-ton cars, and the 50-ton cars give 10 per cent less, which appear to be about the generally accepted proportions.

Mr. D. F. Crawford, in a paper presented at the December, 1901, meeting of the Western Railway Club, gave a diagram showing the resistance of cars of various total weights. The following statement compares the values obtained from this source and those derived by means of equation 86:

RESISTANCE OF CARS.

Gross weight of car in tons	= 10	$16\frac{2}{3}$	20	$33\frac{1}{3}$	50	60	80
Crawford's R	= 7.70	5.75	5.10	3.50	2.75	2.50	2.10
Formula 86	= 8.50	6.50	6.00	5.00	4.50	4.25	4.10

Mr. Crawford's figures were based on some tests with a dynamometer car, but we think it safer to use equation 86, as the first values are very low, and were probably obtained on first-class track and with favorable conditions. For a speed of 12 miles an hour, both formulæ 83 and that of the Baldwin Locomotive Works give a value of 5, viz.:

$$2 + \frac{12}{4} = 5 \text{ and } 3 + \frac{V}{6} = 5,$$

and as seen above, this is the value by equation 86 for 33 1-3-ton cars, which probably represented the average weight of loaded cars in trains until within the last few years, when the large capacity cars have in many cases raised the average. If we desire to use formula 86 at speeds above 12 miles an hour, we can increase the coefficient of T, as was explained in order to apply the rule on grades, by adding to it one-fourth of the amount by which the speed under consideration exceeds 12 miles an hour. This reduced to a formula is simply $3.5 + \frac{V - 12}{4}$, so that equation 86 would become

$$R_e = \left(3.5 + \frac{V - 12}{4}\right) T + 50\,C$$

For 33 1-3-ton cars at 12 miles an hour (or less) we have R = 5, as in the table presented above, and also obtained by formula 83. For greater values of V, as, say, 40 miles an hour, we find by equation 83, $R = 2 + \frac{40}{4} = 12$, and by the above form, $R_e = \left(3.5 + \frac{40 - 12}{4}\right) \times 100 + 50 \times 3 = 1{,}200$, for 100 cars of 33 1-3 tons each, or 12 pounds per ton. This formula will give the same values for 33 1-3-ton cars as are obtained by equation 83, and can, therefore, be used for high-speed calculations with confidence. It is only necessary so to select the coefficient of T, that the effects of grade and speed will be covered by its value. The following table will give these values for various combinations of grade and speed:

(If necessary to allow for curvature, the coefficient of T should be correspondingly increased.)

Values of the coefficient of T in formula 86, or, coef. $\times$ T + 50 $\times$ C = R_e.

Per Cent of Grade.	Speed in Miles Per Hour.						
	10	15	20	25	30	35	40
0.0	3.50	4.25	5.50	6.75	8.00	9.25	10.50
0.2	7.50	8.25	9.50	10.75	12.00	13.25	14.50
0.4	11.50	12.25	13.50	14.75	16.00	17.25	18.50
0.6	15.50	16.25	17.50	18.75	20.00	21.25	22.50
0.8	19.50	20.25	21.50	22.75	24.00	25.25	26.50
1.0	23.50	24.25	25.50	26.75	28.00	29.25	30.50
1.2	27.50	28.25	29.50	30.75	32.00	33.25	34.50
1.4	31.50	32.25	33.50	34.75	36.00	37.25	38.50
1.6	35.50	36.25	37.50	38.75	40.00	41.25	42.50
1.8	39.50	40.25	41.50	42.75	44.00	45.25	46.50
2.0	43.50	44.25	45.50	46.75	48.00	49.25	50.50
2.2	47.50	48.25	49.50	50.75	52.00	53.25	54.50
2.4	51.50	52.25	53.50	54.75	56.00	57.25	58.50
2.6	55.50	56.25	57.50	58.75	60.00	61.25	62.50
2.8	59.50	60.25	61.50	62.75	64.00	65.25	66.50
3.0	63.50	64.25	65.50	66.75	68.00	69.25	70.50

RESISTANCE AFFECTED BY WEATHER.

That the weather has a considerable effect upon train resistance is self-evident. Tests made by C. J. H. Woodbury in 1884 indicated a coefficient of friction at a temperature of 40 degrees Fahrenheit, two or three times as great as at 100 degrees, and in railroad service we have very much greater variations in temperature: from 40 below in Dakota in winter to 130 in Arizona in summer—nearly the difference between ice and steam! In a paper before the Western Railway Club in 1903, M. H. Wickhorst referred to some winter dynamometer car tests made on the Burlington Road, and from which they established three factors of train resistance, shown by the following table:

RESISTANCE IN POUNDS PER TON.

Tons weight of car	20	30	40	50	60
Weather above 30° F.	5.0	3.8	3.2	2.8	2.4
Between 10° and 30° F.	8.3	6.2	5.5	5.2	5.0
Below 10° F. above zero	10.3	8.2	7.5	7.2	7.0

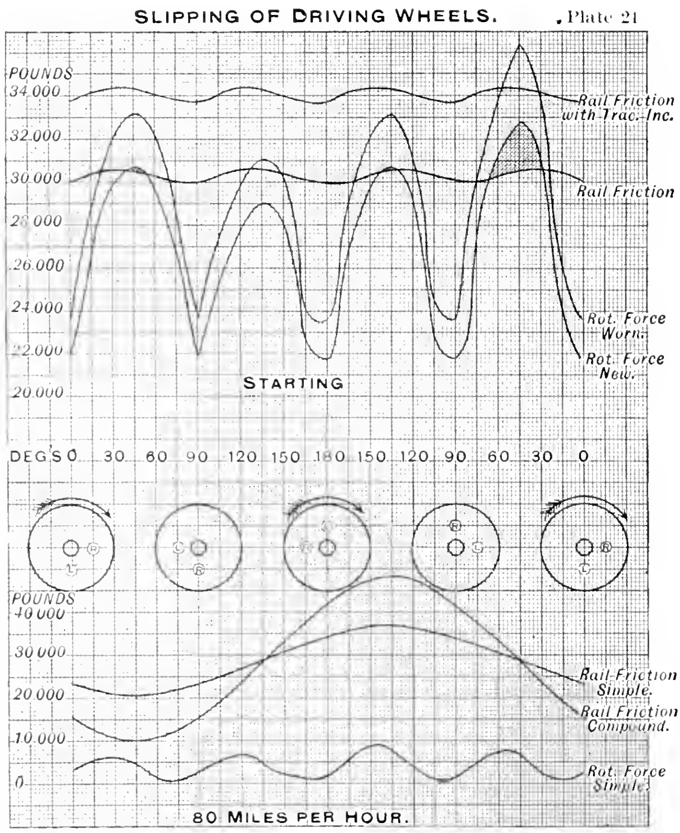
Some roads make use of quite an elaborate system of deductions from the standard loading in cold weather, which will be explained under "Hauling Capacity." In northern latitudes a deduction of 7 per cent is sometimes made for wet or frosty rail—some arbitrarily reduce loads from 10 to 15 per cent in winter. Another northern line reduces 10 per cent for inferior rail and unfavorable weather and 20 per cent for inferior rail and stormy weather. Often, however, there is no fixed rule, but a declaration that "during inclement or windy weather, a reduction from the established rating shall be made, at the discretion of the superintendent or his representative," and it is well known that such reductions are seldom authorized, until the superintendent finds that the trains are not getting over the road, and then a reluctant order is issued to cut 10 or 15 per cent off the rating. The actual resistance during various kinds of bad weather is, of course, not definitely known, but it is often sufficient to completely tie up traffic. This will be examined more closely when we take up "Tonnage Rating."

CHAPTER IV.

SLIPPING.

Whenever, during the action of a locomotive, the rotative force at the circumference of the driving wheels is in excess of the friction between the wheels and rails, slipping ensues. This must not be confounded with "sliding or skidding," when the wheel ceases to revolve and the engine still moves. Under the head of "Steam Action" we found that the rotative force continually varies, not only with changes of speed and cut-off, but during a single revolution of the drivers. In the first chapter we saw that at high speed the "excess balance" caused a continual change in the wheel pressure on the rail, this also varying between very wide limits in a single revolution. The coefficient of friction of the wheel upon the rail has been investigated and found that it may, under favorable conditions, be as high as 35 per cent, or with sand, possibly, 40 per cent. On the other hand, it may be very much lower, if the track be frosty or greasy. It appears from the above that the conditions are rather difficult of accurate analysis; in fact, even that which seems the simplest, the coefficient of friction, cannot be positively determined for any special case in advance. It was stated that if the maximum available tractive force at the circumference of the drivers was not over one-fourth of the adhesive weight, there would be little danger of slipping—that is, to any great amount; but even the most stable engine will at times "fly up" (as it is termed in railroad parlance), especially when starting a heavy train on a muddy road crossing.

We saw in plate 19 that the maximum rotative force exceeded the average by a considerable percentage, and this must also be reckoned with. We have presented rules and formulæ covering the points just mentioned, so that it requires merely a combination of the proper data to determine the details of slipping. The downward pressure of the connecting rod upon the main wheel increases its adhesion, although it may, in some types, reduce the weight on the front driver. In tests recently made upon a track scale, with a 2—6—2 type engine, the weight on the forward drivers was 5,000 pounds less than normal, with the crank on the lower quarter and steam pressure on the pistons. This was evidently caused by the upward pressure upon the guides tending to raise the front of the engine. If the guides are opposite the front drivers, as in 2—8—0 engines, the downward excess on the main wheel may be nearly overcome by the reduction of weight on the front wheels. If the main wheel is being considered independently of the other wheels, the full downward pressure may be allowed. If the engine has no truck, it is evident that the total load on the drivers cannot change, whatever the angle of the crank. Perhaps the best way to study this subject will be to construct a diagram, giving the rotative force and the friction for a complete revolution of the drivers when the rotative force is greatest, as at starting, and also when the effect of the counterbalance is greatest, as at maximum speed. Let us take the New York Central's 4—4—2 engine, and as we have already constructed the rotative force curves, as shown on plate 19, we can work directly from these diagrams. The determination of rotative force has been fully explained in the chapter on steam action, and a repetition here would be useless; suffice it to say that at high speed the inertia of reciprocating parts must be included, as in plate 19. The curves given in that plate show the rotative force at the crank pin, and in order to reduce it to the tread of the wheel it must be multiplied by the stroke and divided by the diameter of the driver, or in this case multiplied by $\frac{26}{79}$. We should also make an allowance for the internal friction, or, according to Professor Goss (formula 76), a uniform deduction of 550 pounds. Treating the lower or "starting" diagram of plate 19 in this way, we produce the curve marked "Rotative Force New." See upper or "starting" diagram, on plate 24. This, it should be remembered, includes all the drivers and both sides of the engine. As the speed is so low (starting) we will not have to consider the inertia of the excess balance, but the vertical thrust of the connecting rod



should be included. Equation 61 gave the vertical component of the main rod thrust as

$$P_v = \frac{P\,r}{l} \cdot \frac{\sin a}{\sqrt{1 - \frac{r^2}{l^2}\sin^2 a}}$$

and as the radical is nearly equal to unity, we can write more simply,

$$P_v = P \frac{r}{l} \sin a \qquad (87)$$

As the center of the guides is about one-fourth of the distance from the truck center to the center of the equalized weights, it will be proper to consider that the crosshead will take three-fourths of this thrust off the truck and one-fourth off the drivers, so we shall use three-fourths of the values obtained by equation 87. This must be worked up for the variation in steam pressure and (at high speeds) the inertia of the reciprocating parts must be added, and which can be obtained from plate 18, and the tables produced therefrom.

These values are to be multiplied by $\frac{3\,r}{4\,l} \sin a$, and the corresponding results from the two sides of the engine, 90 degrees apart, must be added together for the total effect. When running ahead, the vertical thrust should be added to the normal adhesive weight, but when running backwards, it should be subtracted, as the effect is to reduce the weight on drivers in back motion. In order to make this clear we will figure half a revolution at starting. From the table constructed already and referred to above, and from plate 17, we can obtain the piston pressures and calculate rail pressures, thus:

RAIL FRICTION AT STARTING.

Letters	a	b	c	d	e	f	g
Degrees	0	15	30	45	60	75	90
Total P	67,800	67,800	67,800	67,800	67,800	67,800	67,800
$\frac{3r}{4l} \sin a$	0	.022	.037	.052	.065	.073	.075
P_v one side	0	1.192	2.540	3.526	4.547	4.949	5.085
P_v both sides	5,085	6.441	7.087	7.052	6.186	5.558	5.085
Ad. weight	95,000	95,000	95,000	95,000	95,000	95,000	95,000
Total rail pres	100,085	101,441	102,087	102,052	101,186	100,558	100,085
Friction at .30	30.025	30,432	30.626	30.615	30.355	30,167	30,025

Letters	h	i	j	k	l	m
Degrees	105	120	135	150	165	180
Total P	67,800	67,800	67,800	64,300	27,700	67,800
$\frac{3r}{4l} \sin a$	.073	.065	052	037	.022	0
P_v one side	4.949	4.547	3.526	1.639	609	0
P_v both sides	6.441	7.087	7.052	6.186	5.558	5.085
Ad. weight	95.000	95,000	95,000	95,000	95,000	95,000
Total rail pres	101,441	102,087	102,052	101,186	100,558	100,085
Friction at .30	30.432	30.626	30.615	30.355	30.167	30,025

These figures repeat themselves throughout the return stroke, and in the same order, so that we can construct the curve of rail friction in plate 24, assuming that the coefficient of rail friction is .30. The ordinates show the force and resistance at the rail, and when the curve of force lies above the curve of friction, slipping will follow. Thus we see that at 45 degrees (back stroke) and 135 degrees (forward stroke, measured from front center) the force exceeds the resistance slightly, as shown by the shaded areas. But at 45 degrees on the forward stroke, there is a large excess. As the cylinders wear and are rebored, and the tires wear down, the rotative force increases, and if we allow the cylinders to reach $21\frac{1}{2}$ inches in diameter and the tires 76 inches, we shall have a relative rotative force worn to new of $\frac{462}{441} \times \frac{79}{76} = 1.08$, or 8 per cent increase. This is illustrated by the curve marked "Rot. Force Worn," and it is apparent that the slipping will be greatly aggravated. This engine is, however, equipped with a "traction increaser," and when in operation it gives about 12,000 pounds additional adhesive weight, and under these circumstances the rail friction will take the higher curve shown in plate 24. In 1895 a committee of the Master Mechanics' Association submitted a report on tire wear, and gave measurements for a 4—4—0 and a 4—6—0 engine, showing that generally the wear was greatest at the points indicated by our diagram where the rotative force exceeds the rail friction, the wear being as great as .04 or .05 inches below regular wear line at these points. Owing to the lost motion of the driving boxes in the pedestals, there was an increased wear at the reversal of stroke on the respective sides, closely following the dead points. In these tests the counterbalance was later removed, but no difference in the wear was discovered, demonstrating that the slipping was evidently caused at low speeds, as these were simple engines, with small cylinders (16 by 24 and 19 by 26); it must not be concluded that the counterbalance does not at times affect the slipping and tire wear. In fact, in some mogul or 2—6—0 engines, identical except that part were simple and part were 4-cylinder compound, it was found that the tires wear twice as fast on the compound as on the simple engines. In order to study the effect of large counterbalances (necessitated by heavy reciprocating parts), the lower diagram on plate 24 has been constructed. The rotative rail force has been produced from the 80-mile-an-hour curve of plate 19 in the same manner as the starting curve, allowance having been made for internal resistance. The curves of rail friction have been calculated by summing the static load, the rod thrust $(P \frac{3\,r}{4\,l} \sin a)$, and the vertical effect of the excess balance, the algebraic sum being taken, as the counterbalance, when above the center line, reduces the weight on the drivers, by the amount (at maximum speed) $1.6 \times \text{stroke} \times \sin a$ times the excess balance. In the simple engine from formula 24, we should have as the proper excess balance on each side, $600 + 580 - 300 - \frac{176{,}000}{400} = 440$ pounds, and for 90 degrees, the vertical effect is $1.6 \times 440 \times 26 = 18{,}304$ pounds plus or minus, depending upon its position below or above the axle. For other angles we have simply 18,304 sin a, "a" being the angle of crank from dead center. The two sides are, of course, taken together for the total rail friction, and for the simple engine it is seen that there is no danger of slipping at high speeds, as the curves of force and friction do not approach near to each other. The tabulated values are here given.

With the 4-cylinder compound, however, the case is different. If we take the reciprocating parts as weighing 1,200 pounds on each side, the excess balance will be $600 + 1{,}200 - 300 - \frac{176{,}000}{400} = 1{,}060$ pounds, and the centrifugal effect of this will be $= 1.6 \times 1{,}060 \times 26 = 44{,}500$ pounds. With the proper additions, etc., this curve has been produced and marked "Rail Friction Compound," 30 per cent being considered the coefficient of friction, as before. Now the curves of force and resistance approach each other very closely at one point, and

RAIL PRESSURE AT 80 MILES AN HOUR.

Letter	a	b	c	d	e	f	g	h	i	j	k	l
Degrees	0	15	30	45	60	75	90	105	120	135	150	165
Total P. 3r/11	....	500	1,100	-1,000	1,900	2,100	10,960	19,530	21,692	25,730	15,100	-2,400
sin a	....	.022	.037	.052	.065	.073	.075	.073	.065	.052	.037	.022
P_v one side	....	-11	-41	-52	124	153	820	1,125	1,610	1,335	559	-53
P_v both sides	820	1,111	1,569	1,283	435	100	820	1,568	1,843	1,579	663	108
C. B. right side	....	4,760	9,150	13,950	15,800	17,700	-18,300	-17,700	15,800	-13,950	-9,150	-4,760
C. B. left side	18,300	-17,700	15,800	13,950	-9,150	1,760	....	+4,760	+9,150	+13,950	+15,800	+17,700
Ad. weight	95,000	95,000	95,000	95,000	95,000	95,000	95,000	95,000	95,000	95,000	95,000	95,000
Total rail press.	77,520	73,954	71,619	68,383	70,485	72,640	77,520	83,628	90,193	96,579	102,313	108,018
Letter	m	n	o	p	q	r	s	t	u	v	w	x
Degrees	180	165	150	135	120	105	90	75	60	45	30	15
Total P. 3r/11	....	6,500	6,300	1,700	1,600	2,200	7,010	17,230	26,770	30,460	23,600	4,700
sin a	....	.022	.037	.052	.065	.073	.075	.073	.065	.052	.037	.022
P_v one side	....	113	233	244	104	161	526	1,260	1,715	1,585	874	103
P_v both sides	526	1,103	1,978	1,829	978	264	526	1,219	1,704	1,533	750	256
C. B. right side	....	+4,760	+9,150	+13,950	+15,800	+17,700	+18,300	+17,700	+15,800	+13,950	+9,150	+1,760
C. B. left side	+18,300	+17,700	+15,800	+13,950	+9,150	+1,760	....	1,760	-9,150	-13,950	-15,800	-17,700
Ad. weight	95,000	95,000	95,000	95,000	95,000	95,000	95,000	95,000	95,000	95,000	95,000	95,000
Total rail press.	113,826	118,863	121,928	124,729	120,928	117,721	113,826	109,189	103,354	96,533	89,100	82,316

a slippery portion of track, whereby the coefficient became reduced, would cause slippage of the wheels when the counterbalances were uppermost. This gives another argument in favor of reducing the weight of the reciprocating parts to the lowest limit consistent with strength. We saw in the chapter on "Resistance" that when the motion of one surface upon another has started, the coefficient of friction at once drops, so that, in the diagrams just presented, the slipping will continue beyond the shaded portion. If it should not stop before the next maximum period, 90 degrees of revolution, the slipping will be continuous, but, as we see, the rotative force drops quickly after leaving the maximum, so that the slippage is only likely to be spasmodic. The 4-cylinder balanced compound will be more stable at high speeds—there being no counterbalance excess, so the load upon the rails will be practically constant, but there would be little difference at slow speeds. Here we must depend upon the judgment of the engineer, as, by the proper use of sand and manipulation of the throttle, most engines can be kept down to the rail; of course, as we saw in the last chapter, sufficient adhesive weight must be given for proper working.

The Master Mechanics' committee of 1898 gave, as the available tractive force at the circumference of the drivers, with the reverse lever in full gear, $\frac{.8 \, P \, d^2 \, s}{D}$, where P = boiler pressure, d = diameter of cylinder, s = stroke and D = diameter of drivers, and stated that if this were not over .25 of the adhesive weight, slippage would not ordinarily occur. As the maximum rotative force at starting may be 20 per cent greater than the average, this would require a coefficient of 30 per cent at the high point of the rotative force curves, and if we figure the New York Central engine just considered, we have

$$\frac{.8 \times 200 \times 441 \times 26}{79 \times 95{,}000} = .244$$

for the ratio of available tractive force to adhesive weight, and our diagram shows us that the peaks of the force curve reached the friction curve. As the cylinders and wheels become worn, the engine will become more slippery, and require careful handling; however, it is generally desirable to have cylinders of large size, as at early cut-off the average effective pressure is so much reduced that the engine will be hampered by weak tractive force at high speeds if the cylinders are not given liberal dimensions. The slipping in starting can be controlled by proper care. At the same time, it does not seem advisable for road engines to have a maximum available tractive force in excess of 25 per cent of the adhesive weight, and switch engines, which never run at great speeds, should have cylinders proportionately smaller.

In a paper presented to the New York Railroad Club in September, 1903, Mr. Lawford H. Fry gave a statement of the relation between tractive force and adhesive weight of a large number of engines.

If we take the maximum available tractive force at the circumference of the drivers at $\frac{.8 P d^2 s}{D}$, as given above, and divide this by the adhesive weight, we can compare the single expansion locomotives with the .25 recommended by the Master Mechanics' Association committee of 1898. The values are represented by the following table:

RATIO MAXIMUM AVAILABLE TRACTIVE FORCE TO ADHESIVE WEIGHT.

Type.	No. Engines.	Maximum.	Average.	Minimum.
4-8-0	6	.267	.242	.225
4-6-0	35	.267	.200	.172
4-4-0	16	.269	.231	.182
4-4-2	21	.275	.229	.190
4-6-2	5	.216	.220	.202
2-6-0	15	.222	.210	.184
2-6-2	6	.215	.190	.167
2-8-0	48	.283	.233	.186

Compound locomotives will have, as a rule, lower ratio values, when figured running compound, one reason being that they must be stable, when starting, under which conditions there is an excess of piston pressure. In 2-cylinder compounds, the effective pressure in the high-pressure cylinder will be reduced by the back pressure or intermediate receiver pressure, when operating compound, which back pressure will be approximately

$$\frac{.8 \text{ boiler pressure}}{\text{Cylinder ratio} + 1}$$

When starting, there is no back pressure, and the piston has consequently an effective pressure equal to .8 boiler pressure. If the ratio of the cylinder areas be 2, as is about the ordinary proportion, we have in the first case an average available piston pressure of 2-3 × .8 boiler pressure, and in the second case simply .8 boiler pressure, so that the piston effect will be 50 per cent greater. If, therefore, the ratio of tractive force to adhesive weight were .20 compound, it would be .30 starting simple, and to prevent undue slipping during this operation, it is necessary to use a lower factor when compounded, as is usually the case. In 4-cylinder compounds the action is different, but an increased effort is available from starting, and must be regarded to avoid undue slipping.

TRACTION INCREASERS.

When desirable to use cylinders of such size that the engine would be ordinarily "overcylindered," recourse can be had to some form of traction increaser, as above mentioned in connection with the New York Central engine. The idea is not a new one by any means, but recently they have been more generally used than heretofore. All types of engines will not admit their application; in general, it must be an engine that has truck or trailing wheels (not drivers) at each end; and furthermore, at least one of these sets of "carrying wheels" must be equalized with the drivers. Fig. 72 shows the three varieties which are most commonly used. The style marked "A" is advocated by the American Locomotive Company, and is the kind applied to the New York Central engine quoted above. A pair of air cylinders "a" force downward a lever b, which pushes the equalizer c free of its normal fulcrum d and substitutes for it a new fulcrum e, forward of the normal fulcrum, thus throwing a portion of the weight normally upon the trailing wheel, upon the drivers, the amount so transferred depending upon the relative location of the two fulcrums.

In the style B, devised by Mr. John Player, formerly superintendent of motive power of the Santa Fe System, the air cylinder f pulls a bell crank g, which, by lifting up the spring h over the trailer, removes a portion of the weight from this axle and transfers it to the main frames.

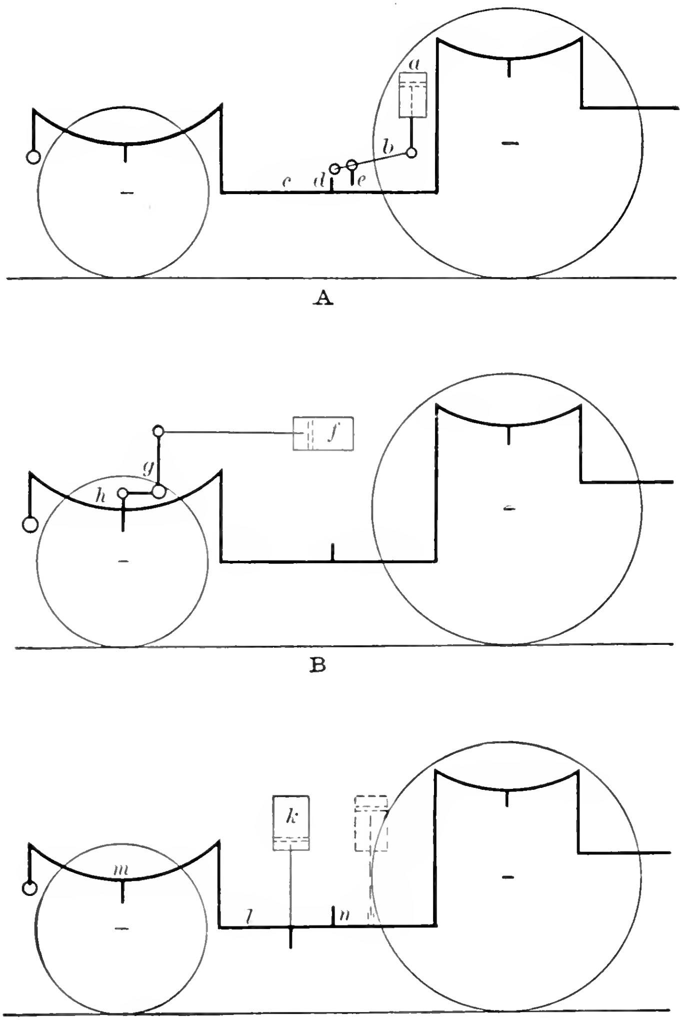


Fig. 72

The C variety was designed by the author, and in this type the air cylinders k pull up the equalizer l on the trailing wheel side, thus reducing the load upon the trailing spring m. The air cylinders could be placed ahead of the fulcrum n of the equalizer 1 (as shown in dotted lines), and be made to push down instead of pulling up, thereby avoiding the difficulty of keeping the packing tight around the piston rod; this seems like a small matter, but the intermittent use of the device renders the proper care of this rod packing quite a difficult matter. Proper arrangements must be made to take care of the vibration or lateral motion of the piston rod, otherwise there will be constant leakage. If the front truck be equalized with the drivers, as in engines with pony or 2-wheel trucks, a similar arrangement will be needed at the front end. This can often be made to pull upward, through a lever, upon the truck pin or plunger, thereby reducing the weight on the truck. In all cases, the attachments should be rigid and ample to withstand the strain imposed by the air pressure.

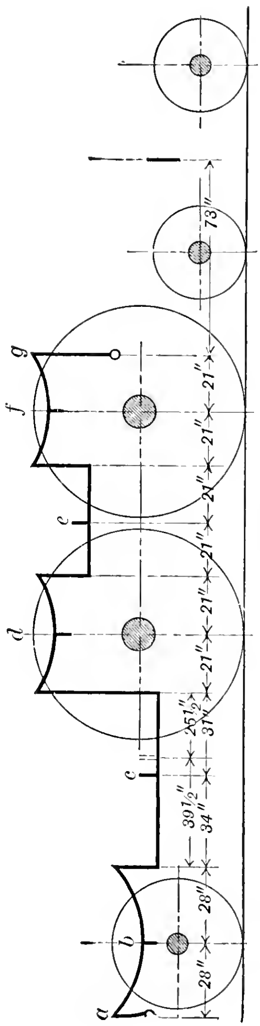


Fig. 73

If the front truck be of the regular 4-wheeled type, simply supporting the front of the engine, and not connected in any way with the spring rigging of the drivers, no apparatus will be needed at the front end, either at the drivers or truck, as the removal of weight from the trailers will also effect a removal of load from the truck. As this seems, in a measure, paradoxical, we will analyze the effect of the traction increaser on the New York Central engine.

Fig. 73 gives the necessary dimensions for making this study. There are two 8-inch cylinders connected to the lever, which, when air is admitted, inserts a new fulcrum $5\frac{1}{2}$ inches ahead of the normal fulcrum of the equalizer, increasing the weight upon the drivers 12,000 pounds, and diminishing that upon the truck and trailers a like amount.

The normal weights upon the track are as follows:

Front truck ....42,500 pounds
Drivers (four) ....95,000 pounds
Trailer ....38,500 pounds

If we deduct the weight of wheels, axles, boxes, etc., from the drivers and trailer, we will have the following normal loads on the boxes:

At b (trailer box) = 38,500 — 2,500 = 36,000 pounds.
At d (rear driver) = 47,500 — 7,500 = 40,000 pounds.
At f (front driver) = 47,500 — 7,500 = 40,000 pounds.

These values representing the total of both sides of the engine.

The load on the spring rigging attachments will be

At a (both sides, included) ....18,000 pounds
At c (normal fulcrum both sides) ....38,000 pounds
At e (both sides included) ....40,000 pounds
At g (both sides included) ....20,000 pounds

The rear equalizer has unequal arms and the moments of each end about the fulcrum should be equal, or $\frac{36,000}{2} \times 34$ should equal $\frac{40,000}{2} \times 31$, which is nearly true. It must be remembered that the friction of the gibs and pins in the spring rigging is always considerable, and we cannot expect more than approximate balances. In order to replace the normal fulcrum with a new one, we see, then, that at least 38,000 pounds will be required to force down the equalizer, and if we consider a pressure of 80 pounds per square inch in the two 8-inch cylinders, each having 50 square inches of area, we have $\frac{80 \times 100 \times 21.75}{4.25} = 41,000$ pounds, which will be sufficient to move the rear equalizer. When the temporary fulcrum is in place the load coming upon the back end or rear driving spring will be $\frac{38,000 \times 39\frac{1}{2}}{65} = 23,000$ pounds; and upon the front end of trailer spring $\frac{38,000 \times 25\frac{1}{2}}{65} = 15,000$ pounds. This will diminish the load on b and increase it on d and f, thus:

Temporary load on b.........(15,000 × 2) = 30,000 pounds
Temporary load on d.........(23,000 × 2) = 46,000 pounds
Temporary load on f.........(23,000 × 2) = 46,000 pounds

And the loads on the rigging will be (both sides):

Temporary load at a........................15,000 pounds
Temporary load at c........................38,000 pounds
Temporary load at e........................46,000 pounds
Temporary load at g........................23,000 pounds

We see that the load on driving boxes has increased from 40,000 pounds at d and f to 46,000 pounds, or 12,000 pounds for both axles, making the total load on drivers (46,000 + 7,500) × 2 = 95,000 + 12,000 = 107,000 pounds. The trailer has lost at b 36,000 — 30,000 = 6,000 pounds, but the drivers have gained 12,000 pounds so evidently 6,000 pounds have been taken from the front truck. If we take the sum of the weights at the several points normally and with the traction increaser in operation we find that the 6,000 pounds cannot be found in the spring rigging or the trailer.

	Normal.	With Trac. Inc.
Weight at a............	18,000 pounds	15,000 pounds
Weight at c............	38,000 pounds	38,000 pounds
Weight at e............	40,000 pounds	46,000 pounds
Weight at g............	20,000 pounds	23,000 pounds
Total	116,000 pounds	122,000 pounds
		116,000 pounds
Increase on spring rigging..............		6,000 pounds

For the wheels we have:

	Normal.	With Trac. Inc.
Weight at b............	36,000 pounds	30,000 pounds
Weight at d............	40,000 pounds	46,000 pounds
Weight at f............	40,000 pounds	46,000 pounds
Total	116,000 pounds	122,000 pounds
		116,000 pounds
Increase on wheels......................		6,000 pounds

So, again, the 6,000 pounds must come from the truck. We know that the center of gravity of the engine cannot

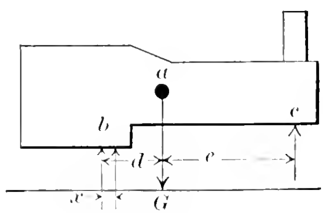


Fig. 73 a.

change, and that the moments about the truck center in both cases should be the same. We have for these conditions:

	Normal Moments.—		Temporary Moments.—	
Load at a......................	18,000 × 320	= 5,760,000	15,000 × 320	= 1,800,000
Load at c......................	38,000 × 230	= 8,710,000	38,000 × 224½"	= 8,531,000
Load at e......................	10,000 × 136"	= 5,110,000	16,000 × 136"	= 6,256,000
Load at g......................	20,000 × 73	= 1,460,000	23,000 × 73"	= 1,679,000
Totals......................	116,000 lbs.	21,400,000	122,000 lbs.	21,266,000

nearly the same total moments. But if we divide the sum of the moments by the sum of the loads we get $\frac{21,400,000}{116,000}$ $= 184''$ and $\frac{21,266,000}{122,000} = 173''$ as the distance from the truck center of the resultant reactions, and as the latter is closer to the truck, it evidently has reduced the load on it temporarily, for, in Fig. 73a, let the center of gravity be at a, then the load c on truck will be $\frac{G\,d}{d + e}$. If, however, the point of reaction b be moved ahead by an amount x, we have for the load on truck $\frac{G(d-x)}{d+e-x}$, and as x is negative and bears a greater ratio to d than it does to d + e, the reaction at c will be less in the latter case than in the former one. If we substitute in these equations the values G = 146,000 (the assumed weight of engine above wheels and truck); $e = \frac{21,400,000}{146,000} = 146''$; $d = 184 - 146 = 38''$, and $x = 184 - 173 = 11''$, we obtain for the first value 30,000 pounds and for the second value 23,000 pounds, a difference of 7,000 pounds, a little more than our first calculations indicated would be taken from the truck. We see that this style of traction increaser depends upon the distance which the fulcrum is changed, and not upon the pressure in the cylinders, except that it must be sufficient to dislodge the equalizer from the permanent fulcrum. If the pressure be greater, it will cause the piston to move downward until it strikes the bottom head or some other point, but no matter how much pressure is applied, there will be no more load transferred to the drivers than indicated by our calculations. If the C style (Fig. 72) is used, however, the case is different, and the amount of load transferred depends entirely upon the pressure. If this kind of traction increaser were used, with the same size cylinders, in order to increase the adhesive weight 12,000 pounds, it would have to be applied about 13 inches back of the fulcrum; then we should have

$\frac{80 \times 100 \times 13}{34} = 3,000$ pounds release on the front end of trailer spring, or 6,000 pounds at box, and $\frac{80 \times 100 \times 13}{31} =$ 3,300 pounds on rear end of driver springs, or 6,600 on each pair, a total of 13,200 pounds increase in adhesive weight. If the cylinders were larger, or the pressure increased, in this case, the load transferred would be correspondingly greater.

The connection of the traction increaser to the air system is important, and requires careful study. The air must not apply the brakes when used in the traction increaser cylinders, nor must it reduce the main reservoir pressure beyond the safe limit, for prompt release and control of trains. For these reasons a separate reservoir should be provided for this purpose, and a special check valve between it and the main reservoir should insure no withdrawal of air from the main reservoir until the pressure therein exceeds 90 pounds per square inch. The valve should also close when the pressure in the special reservoir reaches the proper limit, in order that the traction increaser pistons may have only the intended pressure imposed upon them.

The application of the traction increaser may be made to depend automatically upon the position of the reverse lever, or it may be operated at the will of the engineer by a cock in the cab. If the first method be adopted, the increaser should be in action only when the lever is in the low notches of the quadrant, and a cock should be placed in the cab, so that it may be cut out when drifting. If the application is independent of the lever, there is danger that it will be left on at high speeds, when it is not needed and when the excess driver weight on the rail is a disadvantage to the track. In either case, proper instructions should be placed in the cab designating exactly how it is to be handled. Very good results may be obtained when starting heavy trains or when ascending steep grades, but the apparatus requires constant and careful attention. The enginemen as a rule seem opposed to their use, often because they are not properly maintained and then become a source of continual leakage and annoyance.

CHAPTER V.

BRAKING.

It is one of the laws of physics that a body once set in motion will continually maintain this motion until some opposing force neutralizes or reduces the momentum with which it has been endowed. So also a railroad train, to which a high velocity has been imparted, will continue to proceed until the resistance of wind and journal friction absorb the energy which was put into the train in bringing it up to speed. For even ordinary stops this gradual reduction of speed would be too slow and uncertain, and when, from danger in front, or other causes, a quick stop is desirable, it is absolutely essential that means be provided to arrest the motion of the train in the shortest possible distance. For this purpose brakes are applied to railway equipment, which, by a large amount of friction generated at the rubbing surfaces, are able to quickly absorb the energy of the moving body.

Naturally, there are several ways of producing this friction, one of the most direct by forcing a shoe or sliding piece against the rail, thereby causing a resistance to the movement of the car. Ordinarily there are objections to this method from a mechanical point of view, such as the resulting damage to frogs and switches, and railway equipment is universally provided with brakeshoes, which are brought against the circumference of the wheels, and by retarding their revolution, bring the cars to rest.

There is a type of brake in limited use which does not depend upon the friction of surfaces for its retarding power, but upon the performance of mechanical work, such as the compression of air by pistons attached to the revolving wheels by means of cranks and rods, and this principle is made use of in the Le Chatelier or the Sweeney brakes, as adapted to locomotives operating upon heavy grades.

In this country power brakes are required by law, and as practically no form is used but the air brake (except in special cases on locomotives, as above mentioned) it will not be necessary to consider vacuum or hand train brakes. Three varieties of air brake are, however, in common use on American railroads, viz., straight air, automatic, and high speed brakes. The first was the original form or prototype of the present brake, and is now used quite extensively for the driving wheels and tenders of locomotives engaged in switching service. Under such conditions the train pipe is usually not connected to the locomotive, and the quicker action, and especially release, of the straight air brake results in a much more efficient switching service. It is applied in addition to the automatic brake, so that if a train is to be taken any great distance the automatic brake can be "cut in" quickly and operated under the usual conditions.

The high speed brake is rapidly coming into use in passenger service, and, like the "straight air," is arranged so that the ordinary quick acting automatic can be operated by simply turning a cock on the engine. The mechanical details by which these different systems do their work are fully explained in treatises on the air brake, and it would be out of place to describe them here, but the results of their action can be studied, as they form an important part of "Locomotive Operation."

In the three systems mentioned, many functions are identical—the air is compressed by the same kind of a steam-driven pump, discharging into a main reservoir, in which the compressed air is stored, and from which it is admitted to or released from the train pipe by the engineer's brake valve, applying or releasing the brakes, in accordance with the system in use. In the straight air no auxiliary reservoir is needed, but in the other two this intervenes between the train pipe and the brake cylinder of each vehicle, storing air for its particular car, and being regulated and operated by the triple valve. In all three systems, a piston in a cylinder is forced outwardly by air pressure back of it, thus pushing the brakeshoes against the wheels, through a system of rods and levers. The mechanisms referred to, complicated as they may seem, are designed for the very simple purpose of compelling the brakeshoes to rub the circumference of the wheels, and it seems like a great deal of apparatus to accomplish a very simple result, but the proper pressure of the shoe against the wheel is a more complicated matter than would appear at first sight. We have seen, in the chapter on Resistance, that the friction of the shoe against the wheel depends upon the speed of rotation and the unit pressure upon the surfaces in contact, and that at the outset this introduces a complication; we have also seen that the friction of the wheel upon the rail is greatly reduced if there is the least tendency to skid, and this point is of the greatest importance in the action of brake gears.

The reprint of the Galton report to the Institution of Mechanical Engineers in 1878, by the Westinghouse Air Brake Company gives the conclusions drawn from the tests made on the Northeastern Railway, as to what appeared to be the essential conditions of a good brake, in the following language:

"First. The skidding of the wheel, so that it slides on the rail, is altogether a mistake, so far as rapid stopping is concerned.

"Second. The pressure with which the brakeshoes are applied to the wheels should be as high as possible, short of the point which would cause the wheels to be skidded and to slide on the rails.

"Third. The rotation of the wheel is arrested as soon as the friction between the brakeshoe and the wheel exceeds the adhesion between the wheel and the rail; and therefore the amount of pressure which should be applied to the wheel is a function of the weight which the wheel brings upon the rail.

"Fourth. In practice and as a question of safety it is of the greatest importance that, in the case of a train traveling at high speed, that speed should be reduced as rapidly as possible on the first application of the brakes.

"Fifth. The friction produced by the pressure of the brakeshoe on the wheel is less as the speed of the train is greater; to produce the maximum retardation, as far as speed is concerned, the pressure should be greatest on first application, and should be diminished as the speed decreases, in order to prevent the wheels being skidded.

"Sixth. The maximum pressure should be applied to the wheels as rapidly as possible and uniformly in all parts of the train.

"Seventh. To prevent retardation from the dragging of the brakeshoes against the wheels when the brakes are not in use, care should be taken that the shoes are kept well clear of the wheels (say ½ inch) when in a state of inaction."

These conclusions of Captain Galton, stated 25 years ago, are quite remarkable, from the fact that the science of braking at the present day confirms every point taken as the proper logic upon which to design a brake system, and we cannot do better than to take up the different points, and study their effect in everyday practice.

The first deduction, that skidding is detrimental to rapid stopping, may cause some surprise. If we refer back to the chapter on Resistance we will find under the caption of "Rail Friction" that the coefficient of friction between wheel and rail is given as 24 per cent when just coming to rest, or for static friction. At 7 miles an hour the coefficient drops to 8.8 per cent, about one-third the static friction. As an illustration, let us consider a car weighing 100,000 pounds, with brakes applied to all wheels. If, in making a stop, sufficient pressure be applied to the brakeshoes, so that there is produced a frictional resistance to the wheel's rotation of 23,000 pounds, we shall evidently have a braking force or resistance of this amount to produce retardation, neglecting journal and wind resistance. If, however, from any cause, the wheels should stop revolving, and "skid" upon the rails, if the speed be at the rate of 6 or 7 miles an hour, the resistance caused by the brakeshoes ceases (because the relative motion between the shoe and wheel has ceased) and the wheels slide along the rails with a resistance of only 8,800 pounds, or about one-third as much as when they were revolving. If the speed be higher the resistance would be still less.

Fig. 74 illustrates this point, and is reproduced from one of Captain Galton's tests. The height of the curve shows the retarding effect of the shoes upon the wheels, but at x they commenced to slide upon the rails, when the retarding effect was immediately reduced, as shown in the figure. In certain tests made on the Brighton Railway a car was detached from the engine at a definite speed and allowed to come to rest. In one case, when the brake pressure was not sufficient to stop the rotation of the wheels, the car came to rest in a distance of 189 yards. In the other case, where enough pressure was

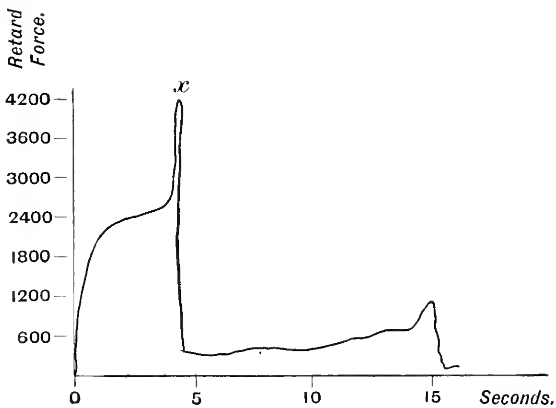


Fig. 74.

applied to arrest the rotation of the wheels, so that they slid upon the rails, the car ran over 400 yards before coming to rest. Besides the reduction in stopping power caused by skidding or sliding, the result is disastrous to the wheels, and particularly in the case of locomotive driving wheels. The Master Car Builders' interchange rules permit renewal of wheels which are slid flat for 2½ inches or more. Besides the damage to wheels, the jar and pound caused by a flat spot is very hard on axles and other parts of trucks. Sometimes a flat spot, if not too large, can be worn out, especially in driving wheels, where the wheel is forced to rotate by the power of the cylinders. If this cannot be done, all the wheels must be re-turned before the engine is fit for duty, causing loss of valuable metal and time from service. Some engineers object to driving wheel brakes on account of the danger of sliding the wheels, but if the engine be properly handled, and the brakes carefully proportioned, there is no reason for such apprehension, and as the drivers represent a large carrying load, the application of brakes to such wheels is of great value.

The second statement, that the pressure of the shoes against the wheels should be as high as possible, without skidding the wheels, needs no argument, but an amplification might be made by stating that to insure the maximum total pressure of the shoes against the wheels, every wheel in the train must be braked. A few years ago it was customary to omit brakeshoes on the middle wheels of six-wheel trucks, also on locomotive driving and truck wheels. The high speeds of the present day require advantage to be taken of every opportunity offered to increase the braking power, and even the trucks of locomotives are now quite commonly fitted with brakes. Formula 1, when inverted, or, as $S = 70 \frac{V^2}{P}$, shows us that the distance in which a train will be stopped is inversely as the force of retardation applied, and this is evidently a maximum, when each wheel is braked as high as permissible for the load which it carries. If the equipment is carried on six-wheel trucks, and brakes are omitted from the middle wheels, we must expect our train to run half as far again after brakes are applied as it would if all wheels were equipped with them. If the driving wheels of the locomotive carry one-tenth of the total weight of the train, our stopping distance will be one-ninth greater if the driver brakes are cut out, than when in action. This has also been demonstrated experimentally. Captain Galton estimated the following stops made from 50 miles an hour, with various proportions of retarding force to weight of train:

RETARDING FORCES AND STOPS.

Per Cent of Retarding Force.	Length of Stop in Yards.	Per Cent of Retarding Force.	Length of Stop in Yards.
5	555⅔	16	173⅔
6	463	18	154⅓
7	369⅔	20	139
8	347⅓	22	129⅔
9	308⅔	24	115⅔
10	277⅔	26	107
12	231⅓	28	99⅓
14	198⅓	30	92⅔

The momentum of the wheels requires an increase of braking power, but equation number 1 includes the effect of wheel inertia.

The third point treats of the friction between the brakeshoe and the wheel exceeding the friction between the wheel and the rail, causing locking of the former and skidding of the latter. It has been shown that the friction upon the rail is a function of the pressure of the wheel upon the rail, as long as the former is revolving, and no slipping occurs, and it may be considered about 25 per cent of the load upon the wheel.

In Fig. 69 we saw that at the moment of stopping the coefficient of friction of the brakeshoe upon the wheel suddenly

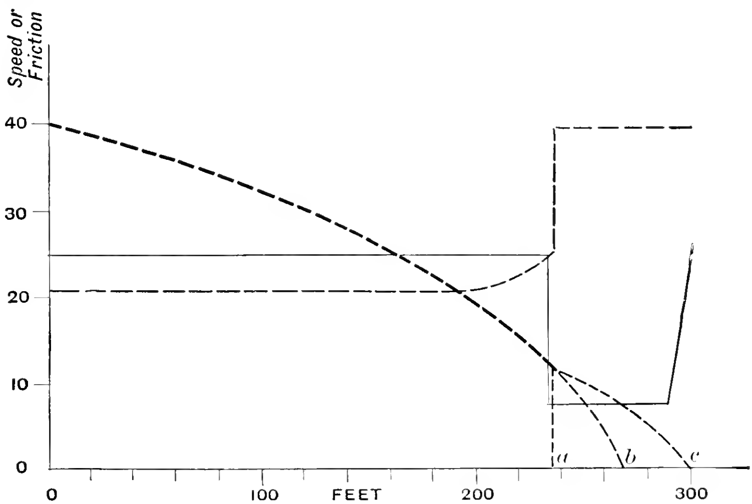


Fig. 75.

increased. In some cases the final friction was double the average—a 50 per cent increase would be a conservative value. The average friction of the Congdon shoe on chilled iron wheels from a speed of 40 miles an hour was found from the table to be 20.3 per cent, whereas the final friction was 31.3 per cent, and this value was taken 15 feet from the dead stop, at which point it was still higher. Now, let us consider what happens when a wheel of chilled iron has a Congdon shoe forced against it with a force equal to the load upon the wheel.

Fig. 75 represents the various factors graphically. The length of stop is shown by the dotted line on the scale in feet. The application of brake is made at a speed of 40 miles per hour, with a retarding force of 21 per cent = 20.3 friction of Congdon shoe and .7 speed resistance. The velocity gradually decreases as shown by the height of the dotted line above the base line, until the coefficient of brakeshoe friction, shown by the broken line, rises, owing to the reduced speed of rotation, to equal the coefficient of adhesion, shown by the solid line. At this instant the wheel stops revolving, the brake shoe friction jumps to double the normal amount, locking the wheel, and the coefficient of adhesion between wheel and rail drops from static to dynamic, or about 8 per cent. Under this reduced retarding force the car travels further than it would have done if the wheel had not skidded, as shown by the distance b c, the rotation of the wheel having stopped at "a." Just as the car comes to a stop, the friction of adhesion rises, as seen by the solid line, returning to that of rest. The retarding force throughout the stop is represented by the broken and solid lines, the lower one to be taken at each or any point during the stop. In other words, as long as the wheels continue to revolve, the retardation is measured by the friction between the brakeshoes and the wheels, but as soon as the wheel begins to slide on the rail, the retardation is measured by the friction between the wheel and the rail.

We have already seen that the coefficient of friction between the shoe and the wheel may vary between very wide limits, by the use of different kinds of shoes. The Master Car Builders' committee report of 1895 showed values from an initial friction of 8.5 per cent to a final friction of 42.1 per cent. Under these conditions it would seem only logical to take into consideration the kind of shoe that was to be used when designing brake gears, but this is seldom, if ever, done. The difference in the friction at varying speeds, and the uncertainty of the amount of rail friction, tend to render any such refinement unnecessary. Still, there is no reason why a shoe of low friction and holding power should not be applied with more force than one which takes a "good grip on the wheel."

After the Burlington brake tests of 1886-7, it was decided by the Master Car Builders' Association to limit the braking power of freight cars to 70 per cent of the light weight of the car (considering brakes applied to all the wheels), as this low limit was found by experience to be necessary in order to avoid sliding the wheels on the rails.

The generally accepted ratios of braking power to the "light weight" on wheels to which brakes are applied are as follows:

Passenger cars	90 per cent.
Freight cars	70 per cent.
Tenders	100 per cent.

For locomotives, the ratio is figured upon the weight in "working order."

Simple locomotive driver brakes.........	65 to 75 per cent.
Compound locomotive driver brakes......	60 to 65 per cent
Locomotive truck brakes.................	75 per cent.

These rules make no allowance for the material of which the shoes or wheels are composed, nor do they give any correction for "double brakes;" that is, shoes on both sides of the wheel. The Norfolk & Western Railway conducted tests a few years ago which demonstrated that with double brakes a lower proportion must be used, if we wished to insure absence of skidding.

If we reverse equation 80 so that it reads $f = f'' + b\,(p - 60)$, and let $f'' = 20$; $b = .06$, and $p = 100$, we have for $f = 20 + .06 \times 40 = 22.4$, or 12 per cent increase, and the tests referred to indicated that with double brakes, the shoe pressure (total) must be reduced about 10 per cent to avoid sliding.

The fourth item, viz., the rapid reduction of speed upon the application of the brakes, has been the principal burden of the brake companies. The old straight air brake worked quite well with a short train, but when a large number of cars were to be handled, the air could not pass quickly enough through the train pipe to reach the rear end in time to effect a satisfactory stop; and besides, the cars at front end would apply first and harder than those at the rear, causing heavy impact from the rear end of the train. The automatic brake, and later, the quick acting device were introduced to bring about this very point. The recommended practice of the Master Car Builders' Association specifies that when tested in 50-car trains (or on equivalent testing rack) the brakes must apply on the fiftieth car with at least 45 pounds pressure and 6 inches piston travel in the brake cylinder in 3 seconds from the first movement of the handle of the engineer's brake valve, and that in 3½ seconds the pressure in the brake cylinder should indicate at least 55 pounds. The conditions are readily fulfilled by the apparatus manufactured by at least two brake companies.

If the brakeshoes do not develop a large amount of friction when brought against the wheel, much time will be lost in making a quick stop. At some places a shoe that lasts the longest time between renewals is favored, regardless of the amount of friction developed. As the brakes are for the express purpose of creating a frictional resistance, it is important that they be of such material as will generate friction. It is true that the same retarding force may be obtained with a shoe having a lower coefficient of friction by increasing proportionately the pressure against the wheel, but this change is seldom made, and, even then, the additional side pressure is a positive disadvantage to the journal bearings, and this was one of the points sought to be overcome by the double brake. At the same time some shoes are very severe on steel-tired wheels, and this factor must also be borne in mind.

The fifth point regarding the greatest braking pressure being needed at the first application and diminished as the speed decreases has been accomplished by the high-speed brake. It can be approximately imitated by a careful engineer by making two applications—the first a heavy one of 15 or 20 pounds, and when the speed of the train has been checked, releasing and applying with from 5 to 10 pounds reduction.

In the regular high-speed brake the train-pipe pressure is increased from 70 to 110 pounds, and the brake cylinder pressure will rise at the start to about 85 pounds, gradually reducing (in about 20 seconds) to 60 pounds, which is the normal emergency cylinder pressure for the quick-action brake. Fig.

76 reproduces brake cylinder cards for the high speed and the quick-action brakes, the former in solid lines and the latter in broken lines. The abscissa is the length of stop, and the ordinate the pressure. Now, as we know that the coefficient of friction gradually rises, we see at once that the high-speed ar-

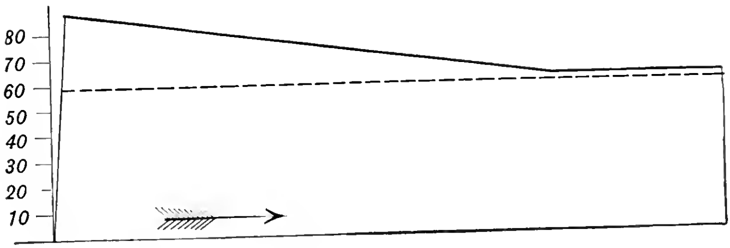


Fig. 76.

rangement will give us a more uniform retarding action than the quick-action brake, and will especially increase the brake resistance at the first part of the application, where it is very desirable, dropping to the normal (60 pounds) pressure by the time the speed has reduced, so as to avoid sliding the wheels. Fig. 77 shows how the retarding force at the circumference of the wheels would be affected by the two styles of brakes, the

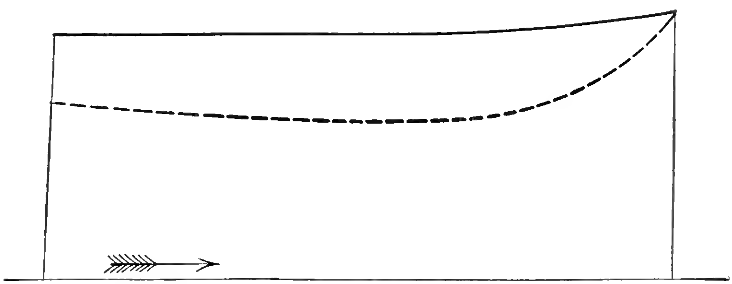
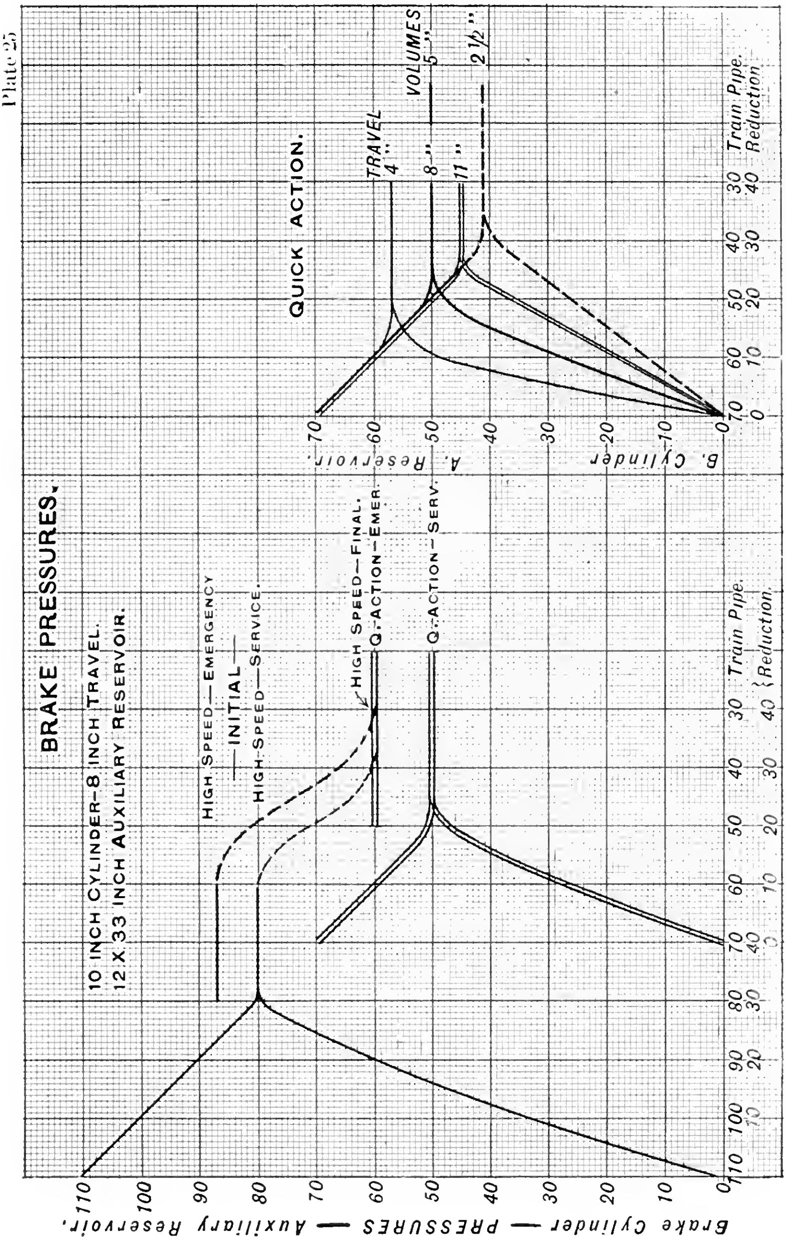


Fig. 77.

solid line representing the high speed and the dotted the quick action, as before.

The left-hand diagrams of plate 25 show the action of the air in the train pipe, auxiliary reservoir and brake cylinder of both the high speed and quick-action brakes, with service and

Plate 25


BRAKE PRESSURES.
10 INCH CYLINDER—8 INCH TRAVEL.
12 X 33 INCH AUXILIARY RESERVOIR.
HIGH SPEED—EMERGENCY
—INITIAL—
HIGH-SPEED—SERVICE.
HIGH SPEED—FINAL.
Q. ACTION—EMER.
Q. ACTION—SERV.
Train Pipe.
Reduction.
QUICK ACTION.
TRAVEL
4"
8"
11"
VOLUMES
5"
2½"
A. Reservoir.
B. Cylinder
Brake Cylinder — PRESSURES — Auxiliary Reservoir.

emergency stops. The single lines show the high-speed brake, the pressure in brake cylinder and auxiliary reservoir being shown by the ordinate, and that in the train pipe by the abscissa. For instance, starting with a train pipe and auxiliary reservoir pressure of 110 pounds, a reduction of 20 pounds, or 90 in the train pipe (as indicated by the abscissa), is accompanied by 90 pounds in the auxiliary reservoir (see upper line) and by 60 pounds in the brake cylinder, assuming that the brake cylinder is 10 inches in diameter, that the piston has 8 inches travel and that the auxiliary reservoir is 12 by 33 inches. A reduction of 25 pounds in train pipe gives 85 pounds in auxiliary reservoir and 70 pounds in brake cylinder. Equalization occurs at 80 pounds in train line, when the auxiliary reservoir and brake cylinder also assume this same pressure. Any further reduction in train-pipe pressure produces no further effect either on auxiliary reservoir or brake cylinder, as they have equalized with service application. The automatic reducing valve, however, commences to operate whenever the brake cylinder pressure is over 60 pounds, and gradually reduces the pressure to this limit, as shown by the broken line in the diagram, the time required for this drop varying from a few seconds to 20 for a brake cylinder pressure of 80 or 85 pounds. If an emergency application be made, the admission of air from the train pipe to the brake cylinder direct raises the pressure in the latter to about 87 pounds, which later falls to 60 pounds, as shown.

In the quick-action brake a similar condition exists, as shown by the double lines, except that when equalization is effected, there is no further drop in the brake cylinder pressure, unless that due to a leak. The point of equalization is seen to be at about 22 pounds reduction, or 48 pounds actual pressure in train pipe, the auxiliary reservoir and brake cylinder equalizing at 50 pounds. With an emergency application, the Westinghouse brake vents air from the train pipe to the brake cylinder, causing a pressure of 60 pounds; the New York brake merely vents this train-pipe pressure to the air, and the emergency application will thereby cause equalization at 50 pounds, as in service reductions.

The sixth requirement is one of practical importance, viz., that the pressure should be applied uniformly to all parts of the train. Design and maintenance are both responsible for the proper fulfilment of this condition. When the full service power of the brake is effective, the auxiliary reservoir and the brake cylinder have equalized; that is, their volumes have been thrown together, causing a uniform pressure in both the reservoir and the cylinder. If the pressure in the auxiliary reservoir is 70 pounds gauge or 85 pounds absolute, and when equalized should be 50 pounds gauge or 65 pounds absolute, the ratio of volumes of the reservoir and the reservoir + the piston displacement, pipes, etc., should be as 65 is to 85, or $\frac{85}{65} = 1.31$; that is, the volume should increase 31 per cent. This must include the volume back of piston in cylinder, the pipes leading to same, and the space in the triple valve in connection with the brake cylinder. If we take the nominal dimensions of the reservoir as representing its volume (that is, a 10×24 to equal $78 \times 24 = 1,872$ cubic inches) and the stroke and area of brake piston as indicating its displacement volume, the ratio should be about 5 to 1, the volume of pipes, etc., reducing the actual expansion ratio to about 3 to 1. In the right-hand diagram of plate 25 the heavy solid line shows the service equalization and graduating points of a brake having the reservoir volume about five times the volume of piston displacement, and the heavy broken line that of a brake whose reservoir is but 2½ times the piston displacement. (This case was found upon a locomotive in service, the small ratio reservoir supplying the driver brake cylinders, while the large ratio reservoir furnished air for the tender brake cylinders, and the report that the driver brakes did not hold properly led to the investigation which revealed the above facts.) By examining the diagram, it will be found that while the tender brake (solid line) equalized with 8 inches travel at 50 pounds, and with a reduction in train-pipe pressure of about 22 pounds, the driver brakes (broken line) equalized at 41 pounds, and even this necessitated a reduction in train-pipe pressure of over 30 pounds. There was not only an absence of desirable braking power, but a great loss in train-pipe pressure to produce that power. The engine would not do its share of the braking, and the tender would be compelled to assist in checking the speed of the engine. We can readily imagine the unsatisfactory results, if such inequality were permitted throughout the train. For graduated points, such as a 10-pound reduction, we see that the solid line gives twice the brake cylinder pressure that the broken line does. Locomotives are fitted usually with "plain triple valves," which produce only 50 pounds in the brake cylinder (if brake be properly proportioned) either with emergency or full service applications, whereas the "quick action" triples on the cars give 60 pounds in emergency and 50 pounds in full service applications, and brake rigging must be proportioned accordingly.

The same effect is produced by variation in piston travel under diffierent cars or engines.

The plate (25) shows the operation with 4, 8 and 11 inch piston travel, with the standard proportion of cylinder and reservoir, and for graduated and full service applications. The equalizations take effect at 57, 50 and 45 pounds, respectively, and require train-pipe reductions of 16, 22 and 26 pounds to produce them. With a 10-pound reduction in the train line, the brake pressures will be 48, 26 and 17 pounds. Thus, if three cars in a train, with brakes properly proportioned, had 4, 8 and 11 inches piston travel, the brakes would hold about three times as much for the first one, and twice as much for the second as for the third one. Not only would a good stop be impossible, but the couplings would be strained unnecessarily. In releasing, the increasing train-pipe pressure will force the triple valve piston on the car with 11-inch travel to release position first; then the 8-inch, and last the 4-inch; thus the brake with the greatest power will be the last to release, causing heavy strains upon the couplings, also danger of sliding wheels.

In addition, a greater quantity of compressed air is required in applying and releasing the brake with long piston travel, entailing greater demands on the air pump, with correspondingly increased wear and tear.

The seventh and last conclusion called attention to the necessity of having the brakeshoes clear the wheels when released. Ordinarily $\frac{3}{8}$ or $\frac{1}{2}$ inch is considered desirable for this purpose. This feature depends principally upon the piston travel, although it is partly controlled by the manner in which the brakes are supported or hung. If the brakebeams are so supported that they are lower when the car is loaded and the springs compressed, then if the car be unloaded, there will be less clearance between the shoes and wheels, unless the former are at or near the horizontal center line of the latter. In case that they are hung low, and the load is removed from the car, the brakebeams may rise so high that, if they do not actually rub the wheels, the piston travel, when applied, will be so short that there will be danger of sliding the wheels by the excessive cylinder pressure.

If the travel be too short, the wheels are apt to be dragged by the shoes even when released; and, conversely with our last proposition, if the car rises upon its springs, the braking power will be increased and still greater tendency to drag the brakes. When brakes are applied while the train is in motion, the piston travel is likely to be about $1\frac{1}{2}$ inches greater than when standing, and as tests and adjustments for travel are usually made with the train at rest, this should be remembered. The hanging of the brakes, and the fact of there being much or little loading in the car, should also be considered in making adjustments.

In the case quoted above, where the driver brake auxiliary reservoir was too small for the brake cylinders, an approximation to the results of the tender brake might have been obtained by shortening the travel of the driver brake pistons to 3 inches, but this would be too small to allow the brakeshoes to clear the wheels properly when the engine rolled in going around curves—in fact, cases have been known where the cylinder head has been broken and pulled off by the shoes catching the wheels when the engine lurched heavily, making it difficult and expensive to maintain the brakes.

Summing up the several points just discussed, we find the following diagnosis:

1. The wheels must not be skidded.
2. All wheels should have brakes, and applied hard.
3. The brakeshoe friction must be proportional to the load.
4. The shoes must apply quickly and hold well.
5. The pressure should be greatest when first applied.
6. The pressure should be uniform throughout the train.
7. The shoes must clear the wheels when released.

We are now ready to study the results accomplished in the way of retardation of trains.

RETARDATION BY BRAKES.

In the chapter on "Inertia" we discussed the effect of accelerating and retarding forces, and the formulæ there introduced will apply here, although some transpositions will be advantageous. Equation 1, which was written

$$P_t = 70 \frac{V^2}{S}$$

Where P_t = the accelerating or retarding force in pounds per ton, including the rotative energy or momentum of the wheels,
V = the velocity in miles per hour,
S = the distance in feet in which the acceleration or retardation takes place

can be used here to better advantage by substituting 20 $P_{\%}$ for P_t, where $P_{\%}$ = the retarding force in per cent of the total weight of the train being considered, whence we have

$$P_t = 20\,P_{\%} = 70 \frac{V^2}{S} \text{ and } P_{\%} = 3.5 \frac{V^2}{S} \text{ and}$$

$$S = 3.5 \frac{V^2}{P_{\%}} \quad \ldots\ldots \quad (88)$$

Equation 2, $P_t = 95.6 \frac{V}{t}$ can also be written

$$t = 95.6 \frac{V}{20\,P_{\%}} = 4.78 \frac{V}{P_{\%}} \quad \ldots\ldots \quad (89)$$

where t = time in seconds during retardation.

Also when V_2 and V_1 represent the initial and reduced velocities during a stop or retardation, we obtain from equation 3

$$S = 70 \frac{V_2^2 - V_1^2}{20\ P\%} = 3.5 \frac{V_2^2 - V_1^2}{P\%} \quad \ldots\ldots\ldots\ldots\ldots (90)$$

$$\text{and } V_2^2 - V_1^2 = \frac{S\ P\%}{3.5}, \text{ also } V_1^2 = V_2^2 - \frac{S\ P\%}{3.5} \quad \ldots\ldots (91)$$

These may be termed the fundamental formulæ for brake retardation, and apply no matter what the style or kind of brake in use. The best brake generally is that which will give the smallest values of S, which, of course, means the greatest values of P%. This latter depends upon the

Maximum limit to be observed in order to avoid skidding;

Proportion or number of wheels equipped with brakes;

Pressure on brakeshoes throughout the stop, and throughout the train;

Friction of brakeshoes against the wheels, which, in turn, depend upon the construction, maintenance and operation of the brake, and P% also depends upon the train resistance due to

Speed,

Grade,

Curvature,

which are independent of the brake apparatus, but which are important factors in determining the length of a stop.

For steel-tired wheels the Master Car Builders' standard requires a brakeshoe that will develop an average coefficient of friction of at least 12 per cent. when applied with a pressure of 6,840 pounds, the wheel running at 65 miles an hour at the commencement of the test. From plate 22 this would probably give 15 per cent friction from a 40-mile-an-hour stop. At the end of the stop, that is, just before coming to a rest, this may increase 80 per cent, or a coefficient of 27 per cent. As passenger cars are braked up to 90 per cent of their light weight, it is possible to have the retarding effect of the brakes arranged as above described, amount to $27 \times .90 = 24.3$ per cent of the load upon the wheels. This is nearly at the sliding point. If shoes are used having a higher coefficient of friction, there will be still more likelihood of skidding. If a lower unit or shoe pressure exists than 6,840 pounds, or the speed when applied be less than 40 miles an hour, the friction will be higher, as we have already found. As stated before, a shoe of any definite amount of friction may be used with good results, if we adjust the pressure accordingly, but this is seldom done, and the pressures for application heretofore mentioned are usually followed, regardless of the kind of brakeshoe.

No one pair of wheels can be braked above the sliding point, and one wheel cannot be overbraked to make up for the deficient braking of another wheel. In order, then, to obtain the value of P% we should figure on the braking power applied to each individual wheel in the train. The sum of all these powers will be the retarding force due to the action of the brakes.

We must also know the pressure on each brakeshoe in the train, and this is determined by the action and arrangement of the brakes. If we follow the generally accepted ratios, we can determine this from the weight of the equipment.

The coefficient of friction of the shoe on the wheel should be known, as this multiplied by the pressure on the shoe gives the braking power. In this case, we use the average friction for the conditions which we are studying.

Plate 26 (at end of book) will be found useful in computing the value of P%. The right-hand diagrams give the values of the coefficient of friction of the Master Car Builders' standard specifications for brakeshoes, the upper curves for variations in speed, and the lower for variations in pressure. If some definite shoe is selected, the value can be obtained from the tables and plates in the section on Brakeshoe Friction. If none is specified, it will be safer to use the Master Car Builders' limit as shown. This value is followed on the corresponding line of the lower left-hand diagram to intersection with vertical passing through the per cent of brakeshoe pressure, when the per cent of friction to weight is read on the left side figures. This must be done for each car or engine, and multiplied by the respective weights, added together, and divided by the total weight of train. To this quotient we must add the value obtained under the head "Physical," and this total constitutes the function $P_{\%}$. The locus marked "average for speed" gives the average speed resistance for the whole retardation from the initial speed to zero, and should be used in selecting the different constituents of $P_{\%}$.

When this value ($P_{\%}$) has been determined we can solve equations 88 to 91. The first will be most generally used, $S = 3.5 \frac{V^2}{P_{\%}}$. We perceive that the loci will be equilateral hyperbolas, when the coördinates are $P_{\%}$ and S, as their product is always constant for any given speed. Plate 27 (at end of book) gives a graphical solution of this formula for the different values of the several factors which are likely to be needed in the computations of train braking. By selecting the locus for the speed in question, the intersection gives the length of stop for different percentages of braking power. It must be remembered that this is the theoretical stop, and delay in making the application, unequal piston travel, or other defects will reduce the efficiency and increase the length of the stop. The plate shows at a glance the influence which velocity plays in the stopping of trains. For instance, with braking friction at 10 per cent of the weight of the train, the best possible stops on a straight, level track would be at

20	40	60	80	miles an hour,
140	560	1,260	2,240	feet.

The percentage of friction gives stops as follows, at a velocity 40 miles an hour: with

5	10	15	20	per cent friction,
1,120	560	370	280	feet.

In order to make clear the use of plates 26 and 27, let us consider a passenger train composed of the following equipment:

A locomotive of the 4—4—2 type, having drivers carrying 91,000 pounds braked with 75 per cent pressure at 50 pounds in brake cylinders, and with 67,000 pounds on truck and trailer, without brakes.

A tender with a light weight of 50,000 pounds, braked to 100 per cent on all wheels, and having a loaded weight of 110,000 pounds, the brake cylinder pressure being 50 pounds.

Ten passenger cars with 80,000 pounds light weight, each braked to 90 per cent, with 60 pounds in cylinder, and weighing loaded 85,000 pounds each, with brakes on all wheels.

The speeds will be assumed at 40 and 60 miles per hour. The brakeshoes will evidently have high unit pressures, so for Master Car Builders' specifications, we find from lower right-hand diagram of plate 26, steel-tired wheels at 40 miles per hour, a coefficient of friction of .12, and for 60 miles an hour, .11

We will tabulate the computations below:

		Braked at 40.	60 Miles Per Hour.
Loco. drivers...	91,000 lbs.	$.75 \times .12 = 9\%$, or 8,190 lbs.	$.75 \times .11 = 8\%$, or 7,280 lbs.
Tender.........	50,000 lbs.	$1.00 \times .12 = 12\%$, or 6,000 lbs.	$1.00 \times .11 = 11\%$, or 5,500 lbs.
10 cars.........	800,000. lbs.	$.90 \times .12 = 11\%$, or 88,000 lbs.	$.90 \times .11 = 10\%$, or 80,000 lbs.
Total brake friction		102,190 lbs.	92,780 lbs.

Total weight of train is:

Locomotive drivers..........	91,000 lbs
Truck and trailer..........	67,000 lbs.
Tender, loaded..........	110,000 lbs.
10 cars, loaded..........	850,000 lbs
Total..........	1,118,000 lbs.

We then have for the braking force on the train at 40 miles an hour, $\frac{102{,}190}{1{,}118{,}000} = 9.14$ per cent, and at 60 miles an hour,

$$\frac{92{,}780}{1{,}118{,}000} = 8.30 \text{ per cent.}$$

To these we must add the physical resisting force, which we obtain from the "Average for Speed" in the upper left-hand diagram, or .35 and .45 per cent, respectively, so that we have for 40 miles an hour, $P_{\%} = 9.14 + .35 = 9.49\%$ and for 60 miles an hour, $P_{\%} = 8.30 + .45 = 8.75\%$ Now, for the length of stop, we have from equation 88,

$$S = 3.5 \frac{V^2}{P_{\%}} = 3.5 \frac{1{,}600}{9.49} = \frac{5{,}600}{9.49} = 590 \text{ feet,}$$

or from plate 27 we find that the $9\frac{1}{2}$ value of $P_{\%}$ crosses the 40-mile-an-hour curve at 590 feet. In the same way, we see that the length of stop at 60 miles per hour will be 1,450 feet.

If the high-speed brake be used in the latter case, we will obtain an increase in shoe pressure which will average from 20 to 30 per cent of that with the quick-action brake. If we call this increase 30 per cent, we have for the braking friction $8.3 \times 1.3 = 10.79$, and adding the physical force .45, we obtain a total resisting force of $10.79 + .45 = 11.19$ per cent. From plate 27 we find the length of stop will be 1,120 feet.

As freight cars carry a load two or more times as great as their light weight, and as brakes must be proportioned so that the wheels will not slide when cars are empty, we will examine the same weight of train in loaded coal hoppers. If we take seven cars of 80,000 pounds capacity and 41,000 pounds light weight, we have for braking power at 70 per cent and at 40 miles an hour, $7 \times 41{,}000 \times .085 = 24{,}395$
and adding that for engine, 8,190
and tender, 6,000

we obtain a total braking force of 38,585

The weight of the train will be 1,115,000 pounds, so $\dfrac{38{,}585}{1{,}115{,}000}$ = 3.50 per cent, and adding .35 for speed resistance, we have a retarding force of 3.85 per cent, and stopping distance of 1,460 feet. If we use the high-pressure control, carrying 90 pounds in train pipe instead of 70 pounds, we obtain about 30 per cent increase in the braking force on the cars, but safety valves on the locomotive and tender brake cylinders prevent a rise above 50 pounds. Under these circumstances our braking force will be increased by $.30 \times 24{,}395 = 7{,}318$, a total of $38{,}585 + 7{,}318 = 45{,}903$, and this divided by $1{,}115{,}000 = 4.12$ per cent, to which must be added the speed resistance .35, or 4.47 total per cent resistance, which means a stopping distance 1,260 feet, a reduction of 200 feet from the stop made with 70 pounds train line pressure. It would not be safe to use this heavy train-pipe pressure with empty cars, as slid flat wheels would surely result, and discretion is necessary in its operation.

Fig. 78 shows these several stops graphically, as determined from the assumptions stated. The abscissa represents the length or distance run after the brake has been applied, and the ordinate the speed at each point during the retardation. For instance, the passenger quick-action 40-mile-an-hour stop shows a reduction in speed to 30 miles an hour in 260 feet, to 20 miles in 440 feet, 10 miles in 550 feet and full stop in 590 feet. The velocities at any point may be determined by equation 91; for instance, in the above curve the speed at 440 feet from the point of application of the brakes is found

$$= \sqrt{V_2^2 - \frac{S P_G}{3.5}} = \sqrt{1{,}600 - \frac{440 \times 9.49}{3.5}} = \sqrt{400} = 20$$

miles an hour. This curve can readily be constructed by means of plate 27. Find the distance to dead stop where each velocity

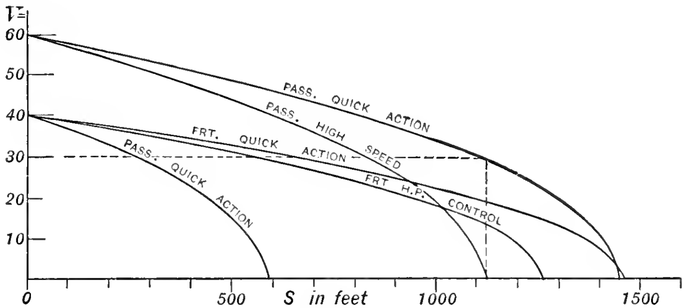


Fig. 78.

curve crosses the proper line representing P_G, as for the case being discussed, we see that the 9.5 line is crossed by the

10	20	30	40 mile speed curves at
40	150	320	590 feet distant from the

stop, therefore it is only necessary to lay off these distances back from the stopping point in order to determine where the train will attain the corresponding speeds. In this way diagrams like Fig. 78 can be quickly prepared, which will answer the various questions regarding retardation.

Another interesting point is brought out by comparing the quick action and high speed passenger brakes at 60 miles an hour. The latter brake produces a stop 330 feet, or 23 per cent shorter than the former, but the difference in speed is more convincing. By erecting a vertical at the 1,120-foot point, or the high-speed stop point, as shown by the dotted line, until it intersects the quick-action curve, we see that while at this distance the train fitted with the high-speed brake would be brought to a standstill, the train with the quick-action brake only would still be moving at a speed of 30 miles an hour—rather a dangerous one at which to strike a heavy obstacle, such as a rock or a derailed car.

We have, in the above cases, considered the effect on level track only. If the braking be done upon a grade, it will either increase or diminish the value of $P_{\%}$, depending upon whether the grade be positive or negative; that is, up or down hill. Plate 26 indicates the amount to allow for the grade. Thus, if it be 53 feet per mile, or 1 per cent, we see that 1 should be added or subtracted from the previously found value of $P_{\%}$. In the passenger quick-action brakes the value of $P_{\%}$ was found to be 9.49 and 8.75 at 40 and 60 miles, respectively, therefore the values on a 1 per cent grade will be $9.49 \pm 1 = 10.49$ and 8.49, and $8.75 \pm 1 = 9.75$ and 7.75, the first values referring to an up grade and the last to a down grade. Fig. 79 shows the re-

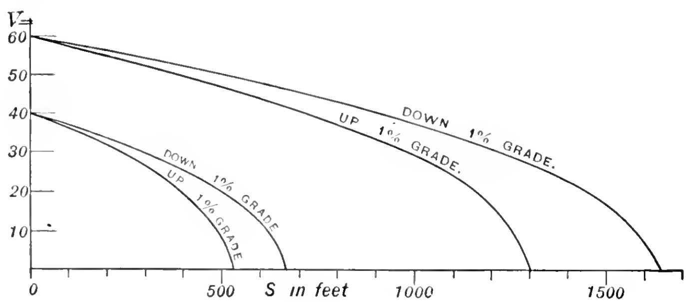


Fig. 79.

tardation curves for the passenger quick-action brake at 40 and 60 miles an hour on 1 per cent grades both ascending and descending, and while there is a considerable difference in the length of stop, it is not as great as would ordinarily be imagined. With a freight train, where the value of $P_{\%}$ derived from the brakes is only one-half or one-third that of passenger trains, the influence of grade would be much greater, as the additional allowance would be a larger proportion of the retarding force. Fig. 80 represents a test made on the

Central Railroad of New Jersey to determine the value of a brake on the engine truck wheels. From a speed of 78 miles an hour, with the brake on truck operative, the stop was made in 2,450 feet, and with this brake cut out the distance run was about 200 feet greater. At 2,450 feet in the second case, the train was moving about 24 miles an hour, whereas, in the first

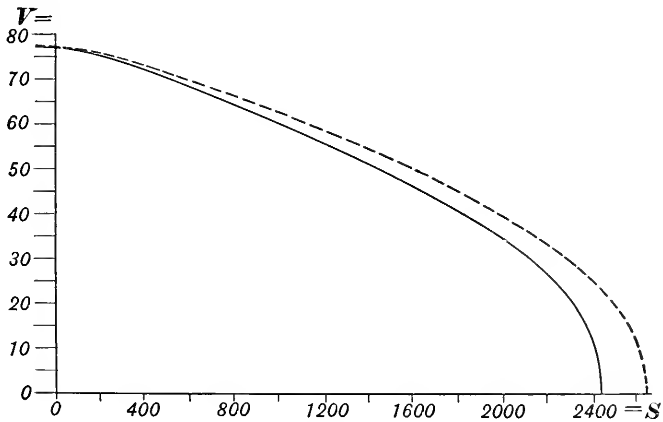


Fig. 80.

case, it had come to rest. By equation 88 we find the value of $P\%$ for both cases, thus:

$$P\% = 3.5 \frac{V^2}{S} = \frac{3.5 \times 78^2}{2{,}450} = 8.69 \text{ and } \frac{3.5 \times 78^2}{2{,}660} = 7.92,$$

or a gain of 9.7 per cent in the power and in the quickness of stop. This train was composed of three cars, besides the engine and tender, a total weight of 459,000 pounds, and the gain of 9.7 per cent in braking power indicates the advantage of the engine truck brake, and how important it is to maintain these in operation.

Through the courtesy of Mr. Burton, general air brake inspector of the road, we are able to compare the details of the brake apparatus with our calculated results. The percentage of weight braked for one pound of cylinder pressure was determined by figuring the brake power due to the cylinder and leverage with one pound per square inch pressure on the piston, and dividing this by the weight of the vehicle, and is given in column 3 of the table. The engine had plain triple valves—the rest of the equipment quick-action triples, which accounts for the 50 and 58 pounds given in column 4. Column 5 is the product of columns 3 and 4. Column 6 gives the coefficient of brakeshoe friction at .11, as the shoes were "Diamond S" and the wheels steel tired. Column 7 is the product of columns 5 and 6, and column 8 the product of 2 and 7.

Vehicle.	Weight.	Per Cent Weight Braked Per Pound Cylinder Pres.	Cylinder Pressure.	Per Cent Weight Braked.	Coefficient of Shoe Friction.	Per Cent Weight in Friction.	Braking Friction.
1	2	3	4	5	6	7	8
Locomotive	151,000	.930	50	46.5	.11	5.1	7,700
Tender	52,800	1.620	58	94.0	.11	10.3	5,450
Car 121..........	72,700	1.470	58	85.3	.11	9.1	6,830
Car 612	74,600	1.507	58	87.5	.11	9.6	7,160
Car 616..........	75,200	1.460	58	84.7	.11	9.3	6,980
Total........	426,300						34,120

Then the braking force of the train as a whole is 34,120 ÷ 426,300 = 8.00 per cent, and the physical resistance due to speed is .60 per cent, a total for $P_{\%}$ of 8.60. Our calculations based upon the actual results of the test gave $P_{\%} = 8.69$ — only 1 per cent difference between the actual and figured results.

The following is a statement of a number of tests made on the same railroad in the spring of 1903, and reported in the October number of Railway and Locomotive Engineering. The last two columns give the proportion of retarding force to the total weight of train, in percentages. The values in column 9 were calculated from the conditions of the stop, by means of formula 88. Those in column 10 were figured from the braking rigging, cylinder pressure and shoe friction, as just shown for the three-car train. It will be noticed that the force was greater at lower speeds, and in the same proportion as the coefficient of friction is also greater, as shown in plate 22. Also it is seen that the high-speed brake gave holding power 25 per cent in excess of the quick-action brake, and about 21 per cent shorter stops. In calculating column 10 consideration was given to the fact that in the shorter stops the cylinder pressure was greater, due to the action of the automatic reducing valve, which requires about 20 seconds to act, and as explained in the small table below:

TESTS OF WESTINGHOUSE HIGH-SPEED PASSENGER BRAKES MADE ON THE CENTRAL RAILROAD OF NEW JERSEY, 1903.

1	2	3	4	5	6	7	8	9	10
Number of Cars.	Actual Speed.	Actual T. Pipe Pressure.	Actual Length of Stop.	Speed for Calculated Stop.	T. Pipe Press. Calculated.	Length of Calculated Stops.	Average of Calculated Stops.	Value of $P_{\%}$.	Calculated $P_{\%}$.
6	70.03	109.75	1527.25	70	110	1522.88			
6	69.56	109.80	1533.66	70	110	1541.75	1509.55	11.25	11.44
6	70.31	111.80	1456.08	70	110	1464.04			
6	61.43	109.66	1029.50	60	110	979.11			
6	58.44	110.25	935.75	60	110	988.31	986.24	12.75	12.93
6	59.60	110.50	973.92	60	110	990.95			
6	51.12	110.25	670.11	50	110	635.15			
6	49.58	110.30	622.00	50	110	634.08	634.61	13.80	14.02
3	76.27	100.30	1915.50	80	110	1974.37	1974.37	11.35	11.00
6	70.03	70.08	1972.58	70	70	1972.73			
6	70.58	70.00	1893.50	70	70	1862.50	1899.42	9.00	8.92
6	70.31	70.00	1879.58	70	70	1863.01			

DETAILS OF HIGH-SPEED STOPS, FOR COLUMN 10.

Speed.	Time of Stop.	Initial Cylinder Pres.	Final Cylinder Pres.	Average Cylinder Pres.	Coefficient of Friction.
70 miles..........	30 seconds.	87 pounds.	60 pounds.	69 pounds.	11 per cent.
60 miles..........	22½ seconds.	87 pounds.	60 pounds.	72 pounds.	11½ per cent.
50 miles..........	17½ seconds.	87 pounds.	63½ pounds.	75¼ pounds.	12 per cent.

ARRANGEMENT OF BRAKES.

We have studied the action of the air in the brake cylinders, and the retarding action of the shoes upon the wheels. The mechanisms by which the pressure reaches the shoes from the brake cylinder piston are various, and a full knowledge of the apparatus is necessary in order to completely investigate any particular case. The cylinders themselves are ordinarily of standard sizes, and, as we have seen, may have 50 or 60 pounds per square inch air pressure in an emergency application with the plain or quick-action brake. The high speed increases this pressure from 20 to 30 per cent, depending upon the time of action, but braking powers are usually figured on the plain or quick action brake, and the allowance above mentioned may be made under the particular circumstances requiring it.

The Westinghouse Air Brake Company give the following standard sizes of cylinders and the emergency pressures obtained therewith in pounds:

Piston Pressures of Brake Cylinders.

Diameter.	6"	8"	10"	12"	14"	16"
50 pounds ..	1,400	2,500	4,000	5,650	7,700	10,050
60 pounds ..	1,700	3,000	4,700	6,700	9,200	12,050

Of course, the auxiliary reservoir must be of the proper size to give this pressure, as already explained. As a guide in examining brakes, the Westinghouse standard dimensions of corresponding cylinders and reservoirs is here given:

Brake Cylinders and Their Reservoirs.

Diam. of cylinder..	8"*	10"	12"	14"	16"
Size of reservoir ..	10x33"	12x33"	14x33"	16x33"	16x42"

The size of cylinder used for different weights on driving wheels is as follows:

8" cylinders up to 40,000 pounds on drivers.
10" cylinders from 40,000 to 85,000 pounds on drivers.
12" cylinders from 70,000 to 115,000 pounds on drivers.
14" cylinders from 110,000 to 170,000 pounds on drivers.
16" cylinders from 145,000 to 225,000 pounds on drivers.

The piston travel in tenders and cars is expected to be kept close to 8 inches in a full service application while running, which produces about 50 pounds in the cylinder, when equalization takes place. This corresponds to about 6½ inches "standing travel." In driver brakes, there is usually only one reservoir provided for the two brake cylinders—the brakes should be so adjusted, however, that equalization shall take effect in both cylinders at 50 pounds, to insure uniform action of all the brakes in the train. This should be tested with a gauge, when adjusting.

The connection from the brake cylinder to the brakeshoes is made by beams and levers, or their equivalents. Usually the

*Tender and truck cylinders 8 inches in diameter use a 10 by 24 inch auxiliary reservoir.

piston rod takes hold of a lever of the first and second order combined, which lever by means of rods, operates levers of the third order. The forces in the different members, and the shoe pressures are obtained by following through the series of rods and levers. A tender brake used as an illustration will make this clear. Fig. 81 shows the brake recommended by

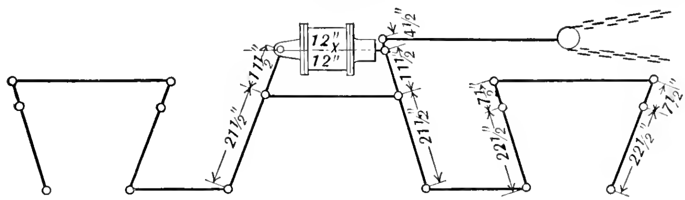


Fig. 81.

the M. C. B. committee for a tender having a light weight of 59,000 pounds. The brake cylinder is 12 inches by 12 inches, giving at 60 pounds air pressure, 6,700 pounds upon the piston, and as there are four brakebeams we have the total shoe pressure for the eight wheels

$$= \frac{6{,}700 \times 11\tfrac{1}{2} \times 30 \times 4}{21\tfrac{1}{2} \times 7\tfrac{1}{2}} = 57{,}340 \text{ pounds,}$$

nearly 100 per cent of the light weight. The same committee recommended certain standard levers, rods, pins, etc., and in designing these, worked to the following limiting unit strains in the various parts. These unit strains are here given for the benefit of those who wish to examine existing brakes, as it is thought that the strains should not exceed those recommended by the committee. When figuring these maximum strains, the greatest pressure that can come upon the piston must be used as a base. If the high speed device is used in passenger service, the pressure may reach 86 pounds per square inch, or if the high pressure control in freight service, 77 pounds per square inch. With these pressures, the maximum stress in the different members comprising the brake rigging should be:

Levers ....23,000 pounds per square inch
Rods (except jaws) ....15,000 pounds per square inch
(No rod to be less than ⅞ inch in diameter.)

Jaws ....10,000 pounds per square inch
Pins (shearing) ....10,000 pounds per square inch
Pins (bearing) ....23,000 pounds per square inch

All parts are supposed to be of wrought iron.

The reduction of stresses and forces in the parts, due to friction of the rigging, action of release springs, etc., was not considered by the committee, on account of its uncertainty. The vibration and jar of the car in service will naturally overcome a large part of such friction, and permit the parts to adjust themselves largely, as if friction did not exist.

(The full details of brake rigging as recommended by the committee can be found in the Master Car Builders' proceedings for 1903.)

The location of the shoes against the wheels is of considerable importance. The Master Car Builders' Association has prescribed 13 inches from the top of rail to the center of the face of new shoes for inside hung beams and 14½ inches for those which are outside hung. As wheels are generally in the neighborhood of 36 inches diameter, this places the shoe a little below the center of the wheel. It is desirable, when possible, that the beams and shoes should always be the same distance above the rail, regardless of whether the car be empty or loaded. In passenger cars the live load is a small percentage of the light weight, and the additional deflection of the springs due to this load is so small that the matter is of little importance. In freight cars, however, the live load is two or more times the light weight, and it is very necessary that the shoes do not move up and down with this variation in loading. This reduces the liability of slid flat wheels, since the piston travel is not affected by changes of load. When this method of supporting the beams is not permissible, they should be so arranged that the center of shoes coincides closely with the horizontal center line of the wheels, as in this position a vertical displacement affects the piston travel less than any other.

Besides the question of variable or non-variable height of the support for brakebeams, the angle of the hanger produces a marked effect upon the braking of the vehicle, and this point was not considered in our calculations upon brake rigging. In fact, it is customary to omit all specific considerations of the brake hanger angle, although its effect is always present. In Fig. 82 let P be the pressure in pounds by which the shoe is pressed against the wheel by the air pressure in the brake cylinder and the system of levers and rods connecting the two. Then an amount of friction F is generated, and if the wheel revolve in the direction shown by the arrow, this force F will act upwards, through the axis of the hanger, as indicated, and its value will be simply P f. If now the point of support be

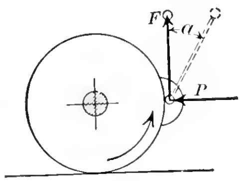


Fig. 82.

moved away from the wheel so that the hanger makes an angle "a" with the vertical, the compressive strain in the hanger due to the friction generated will be F sec a, and its reaction against the wheel in a horizontal direction will be F sec a sin a = F tan a. This is in addition to the horizontal force P produced by the lever, and as the tendency of the hanger is to force the shoe against the wheel, it will be positive; if the wheel revolve in the opposite direction, it will tend to draw the shoe away from the wheel, and in this case be considered negative. The proportional increase in horizontal pressure due to the angle of the hanger will, therefore, be $= I = \dfrac{F \tan a}{P}$ and the total force will be $P + F \tan a$, from which it is apparent that $F = (P + F \tan a)\, f$, where f is the coefficient of friction of the shoe on the wheel. Expanding, we have $F = P f + F f \tan a$ or $F - F f \tan a = P f = F (1 - f \tan a)$, and resolving for F, we obtain $F = \dfrac{P f}{1 - f \tan a}$. Now, substituting this value in the formula $I = \dfrac{F \tan a}{P}$, we have

$$I = \frac{P\ f \tan a}{P\ (1 - f \tan a)} = \frac{f \tan a}{1 - f \tan a} \quad \ldots\ldots\ldots\ldots\ldots \quad (92)$$

and by transposition*

$$\tan a = \frac{I}{f\ (1 + I)} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (93)$$

These equations fix the relation between the angle of the hanger and the proportional increase or decrease of the applied force P. If the truck be moving to the left, as shown by the arrow in Fig. 83, the wheels will rotate as indicated, and the hanger "a" will push the brakeshoe harder against the wheel by an amount I P, so that the total shoe pressure will be

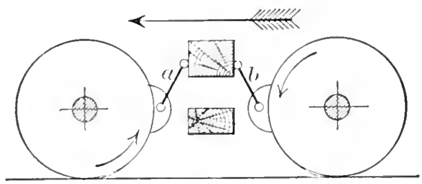


Fig. 83.

P + I P = P (1 + I). The hanger "b," on the contrary, will pull the shoe away from the wheel by a similar force I P, and the total pressure of the shoe on the wheel will be P (1 — I). If the motion be reversed, it is evident that the hanger "b" will increase and the hanger "a" diminish the shoe pressure; in other words, the shoe pressure will be increased on the leading wheels and decreased on the rear wheels, whatever be the direction of motion of the truck. The sum of both shoe pressures will be P (1 + I) + P (1 — I) = 2 P, or the average simply P, and if there were no danger of sliding the wheels, and the rail

---

*Let $x = f \tan a$, then $I = \frac{x}{1 - x}$, but $x = I\ (1 - x) = I - I\,x$, and $x + I\,x = I = x\ (1 + I)$, so that $x = \frac{I}{1 + I}$, or $f \tan a = \frac{I}{1 + I}$ and $\tan a = \frac{I}{f\ (1 + I)}$.

friction were proportionally great, the total braking power of the car would not be altered.

If the brakes are hung from the outside of the wheels, the results are reversed, and hanger "c" diminishes the pressure on the forward wheel, and hanger "d" increases that upon the rear wheel; see Fig. 84.

In order to find out what bearing this question of the angle of hanger has upon the retardation of a train (as the average

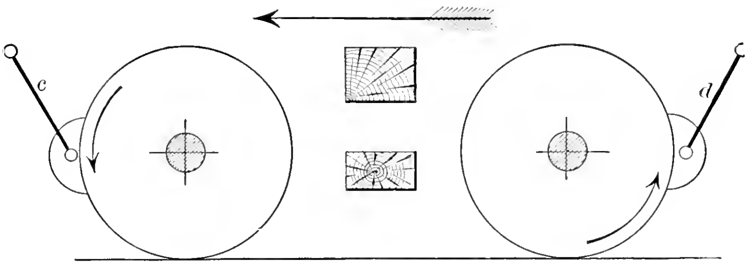


Fig. 84.

or total braking pressures are not thereby changed) it will be necessary to study what happens when brakes are applied to a moving car or other vehicle. In Fig. 85, suppose a passenger car moving in the direction of the large arrow. When the brakes are applied, the adhesion of the wheels upon the rails

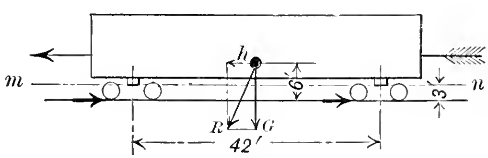


Fig. 85.

produces a retarding force, opposing the inertia of the car which acts ahead at its center of gravity, as indicated at "h." The weight acting vertically, a resultant force "R" is formed, and as this passes through the line of support of the center plates m n closer to the front truck than the rear, the load upon the former will be increased, and that upon the latter diminished. The difference in weight upon the two trucks will not be great, as is seen if we take a passenger car of 80,000 pounds total weight, the body weighing 56,000 pounds and each truck 12,000 pounds. The distance between truck centers is 42 feet, the height of center of gravity of body 6 feet and of the center plates 3 feet above top of rail. If we assume a retarding force of 15 per cent, which is probably not far from the highest average in regular service, we find the moment of this force of the body of the car about the center plates $= .15 \times 56{,}000 \times 3$ and the increase in load upon the front truck or the decrease in load upon the rear truck is $= \frac{.15 \times 56{,}000 \times 3}{42} = 600$ pounds, so small that it can practically be neglected.

In making brake tests of passenger trains, it is found that the rear wheels of each truck will slide or skid before the other wheels of a car, and more especially the rear wheels of the rear truck. This was demonstrated repeatedly in tests made in regular passenger service on the Norfolk & Western Railway some years ago—the rear wheel of the rear truck would slide perhaps 8 or 10 feet, the rear wheel of the front truck (on each car) perhaps half this distance, and the front wheels of both trucks would not slide at all, showing that there was either an unequal distribution of braking power, or of adhesive weight, or both. As the levers connected to the brake beams were alike, this could not be traced to the brake rigging, nor yet to the unequal weight of car on trucks, as the center plates were in the center of the truck wheel base.

We have seen how the inertia of the car body would relieve the rear truck of a portion of its load, although to a very limited extent, but it remains to examine the effect upon the two pairs of wheels of the truck. In Fig. 86 the vehicle is supposed to be moving in the direction of the arrow, and the retardation of the brakes, through the rail friction, causes an opposing force at the rail. If, as before, our car weighs 80,000 pounds, each truck will carry 40,000 pounds, and at 15 per cent retarding force the rail friction will be $= .15 \times 40{,}000 = 6{,}000$ pounds, or 3,000 for each pair of wheels. The car body will produce $.15 \times 28{,}000 = 4{,}200$ pounds at 3 feet above the rail, and the truck itself $.15 \times 12{,}000 = 1{,}800$ pounds at, say, $1\frac{1}{2}$ feet above the rail, a total of 6,000 pounds. The moment acting horizontally upon the truck to overturn it in a forwardly direction is, therefore, $4{,}200 \times 3 + 1{,}800 \times 1\frac{1}{2} = 15{,}300$ foot pounds, which is resisted by a 7-foot wheel base, so that $\frac{15{,}300}{7} = 2{,}200$ pounds (approximately) is the amount by which the front wheels will increase their rail pressure and the back wheels diminish theirs, or $20{,}000 \pm 2{,}200 = 22{,}200$ pounds for front and 17,800 pounds for back wheels, a change of 11 per

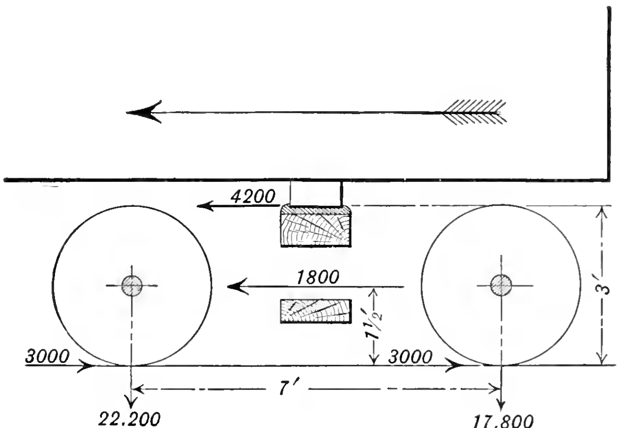


Fig. 86.

cent from the normal weight. We see from this analysis that the rear wheels of any truck will lose rail pressure and the front wheels gain pressure when the brakes are in operation. We saw above that while inside hung brakes increased the braking pressure on the front wheels of a truck and decreased it on the rear wheels, when the hangers were inclined, the reverse was true of outside hung brakes, and, therefore, if for no other reason, the inside hanging should be followed. But there are other reasons. Inside brakes diminish the tilting of the truck frames during a stop, while outside brakes augment it. It is this tilting which causes the sudden and disagreeable jolt at the stop. The proper inclination of the hanger causes the brakes to fall away from the wheels when released, dispensing with release springs. It is somewhat more difficult to replace worn shoes on an inside hung brake, but this is of small consequence compared to the other advantages.

We must now determine how to make the angle of the hanger neutralize the effects of inertia in shifting the wheel pressure during retardation. It must be understood that the angle a in Fig. 82 is not necessarily to be measured from the vertical, but from the tangent to the tread of the wheel at the

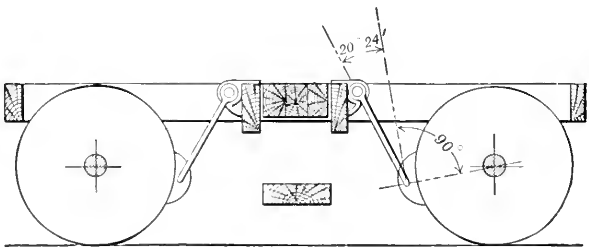


Fig. 87.

center of the brakeshoe. In the car which we have considered it was found that 11 per cent of the normal weight was removed from the rear axle and added to the front axle in stopping. In equation 93, this will represent the value of $I = .11$. The coefficient of friction f, we have taken at .15, so we write

$\tan a = \dfrac{.11}{.15 \times 1.11} = .66$, or an angle of $33\frac{1}{2}$ degrees. This angle is rather large to make a good mechanical arrangement, but if possible it means that each wheel of the car could be braked to the full safe limit against skidding, whereas if the hangers are parallel to the tangent, either the rear wheels will skid or the pressure throughout must be reduced accordingly.

In the case just considered, the braking power would have to be kept 11 per cent below that required for the average wheel load, in order to protect the rear wheels, which lose 11 per cent of their load. With the inclined hangers the correction would be made automatically, and the full power could be used throughout, giving an increase of 12 1-3 per cent efficiency.

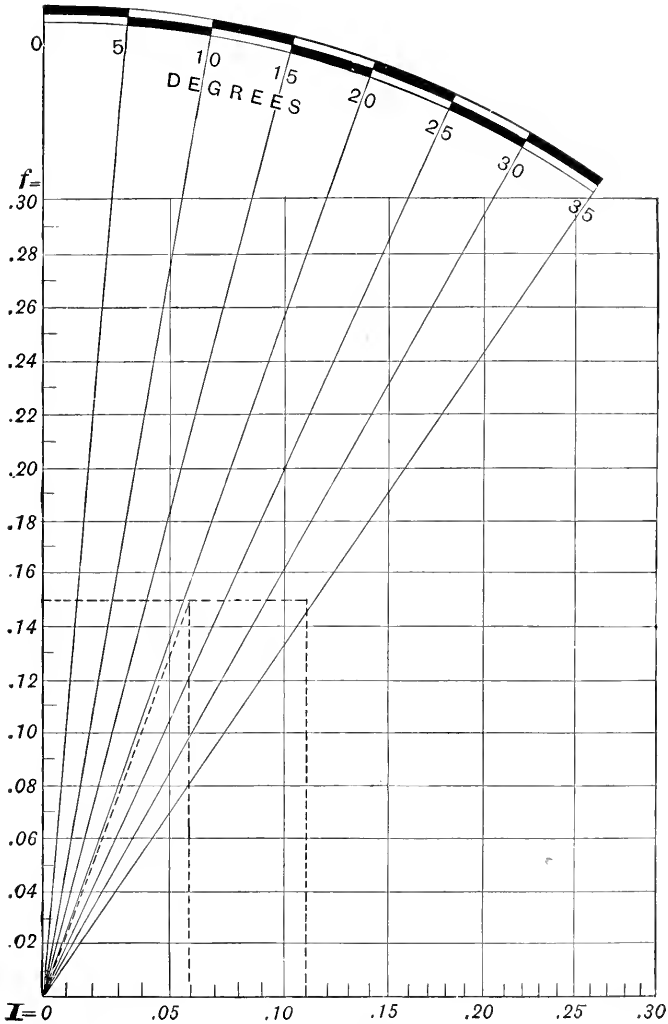

0
5
10
15
20
25
30
35
DEGREES
f=
.30
.28
.26
.24
.22
.20
.18
.16
.14
.12
.10
.08
.06
.04
.02
I=0
.05
.10
.15
.20
.25
.30

Fig. 88.

Fig. 87 shows a practical application of this principle to a passenger truck of the Erie Railroad. The angle of inclination of the brake hanger to the tangent is 20 degrees 24 minutes, and with the same coefficient of friction .15, the proportional change of pressure is, according to formula 92, remembering that tan 20° 24′ = .37

$$I = \frac{.15 \times .37}{1 - (.15 \times .37)} = \frac{.0555}{.9445} = .059;$$

an increase in braking power of about 6 per cent can, therefore, be utilized.

Fig. 88 gives graphical solutions of equations 92 and 93. The angle which the hanger makes with the tangent to the wheel at the center of brakeshoe is found upon the sector at the top; the value of f, the coefficient of friction between the shoe and wheel is read at the left side, and the proportional increase in shoe pressure (or decrease) due to the angle of hanger, I, is taken at the bottom. Thus, in the case of the Erie Railroad truck shown in Fig. 87, we lay off the angle 20° 24′, as shown by the broken line in Fig. 88, and its intersection with .15 friction is found at a value for I = .059. Likewise the attempt to balance the shifting of the weight caused by inertia, by giving the hanger sufficient deflection from the tangent to throw the braking power the same amount, is solved by noting the angle caused by the intersection of f = .15 and I = .11. As in the calculations, we find 33½ degrees would be necessary for a complete neutralization.

It is important that the beam have sufficient stiffness, as well as strength, or undue piston travel ensues. The Master Car Builders' standards require that the deflection at center shall not be over 1-16 inch, with 7,500 pounds at center for the light beam or 15,000 pounds for the heavy beam. Some tests made by the University of Illinois in 1900 demonstrated that with outside hung brakes the strain upon the beam was, at times, very much greater than when an emergency application is made. The brakes were applied with the car at rest; when it was moved the shoe rising as the rear wheel of the truck rotated increased the distance between the beams that were tied together by the bottom connector, in some cases producing a load upon the beams double that which it sustained at rest, or as much as 14,000 pounds.

Driver brakes, as generally applied at the present time, are systems of levers, and the force or pressure upon any shoe can be figured in a similar manner to car or tender brakes; that is, by starting at the brake cylinder and multiplying or dividing by the various lever arms introduced. As a rule, the hangers are nearly parallel to the tangent to the wheel at center of shoe contact, so that the angle does not enter into the computation.

If only the total braking power be wanted, it can quickly be obtained by multiplying the total piston pressure (tabulated previously) by the length of the long lever arm of the bell-crank (or the arm to which the piston is attached) and dividing by the length of the short arm (or one to which the pull rod is attached), the result being the total braking pressure for one side of the engine, when there is a cylinder for each side, as is usually the case. Twice this amount is the total braking power on the driving wheels of both sides of the engine. This rule applies only to the original form of outside equalized brakes, in which the pull on the rods went directly to the shoes, through equalizing levers only, and without the vertical levers often provided at this time, whereby the pressure on the shoe is further increased. When such multiplying levers are used, if all have the same ratio, the value found as above should be multiplied by the mechanical advantage of these levers, in order to obtain the combined shoe pressures on all drivers.

As driving wheel loads are heavy, the strains in the rods and levers are great and require massive sections to prevent stretching and bending. This point was formerly very generally overlooked, and frequent failures of the brake rigging ensued. The strains should not exceed the limits specified above for car and tender brakes.

As road engines run forward most of the time, it follows that when shoes are applied to the front side of the wheels there is an increased load thrown suddenly upon the spring rigging every time the brakes are used. This produces broken driving springs, and also is detrimental to the bearings in the axle boxes. The shoes, coming down with the frame, allow the pistons to travel farther and reduce the equalizing pressure. For these reasons it is considered preferable to place the brakes back of the wheels; then the springs and boxes are relieved by the friction of the shoes, and the upward motion of the frames has a tendency to reduce the piston travel and maintain a higher pressure in the brake cylinders. In addition the boxes are forced ahead in the pedestals against the solid shoe instead of against the adjustable wedge at the rear of the driving box, which is a further advantage. Then the brake cylinders are placed ahead, away from the heat of the firebox, which was so destructive to the packing, and the "push down" arrangement, dispensing with the piston rod stuffing box, adapts itself particularly well to this location.

Cam, or spread brakes, are seldom used now. The spreading action was hard on the parallel rods, and besides only two wheels were braked with one cylinder. The cam was really a wedge and two rollers, as the eccentricity of the cam surface was, mechanically, a wedge wrapped around a cylinder. When the cam was maintained at the length to which it was designed, the braking power was normal, but as the cam was being continually extended by the cam screw and nut, to take up the wear of the brakeshoe, the power was not definitely maintained. The Westinghouse Air Brake Instruction Book gives a ready and simple means of determining the braking power of cam driver brakes under any adjustment, normal or otherwise. To ascertain this power, apply the brake and measure the piston travel, using a full equalization in the cylinder, which pressure should be obtained by a gauge. Then release the brake, insert pieces of 1/4-inch steel wire crosswise between the tire and shoe at the upper and lower ends, and again apply the brake, measuring the piston travel as before. Divide the difference in the piston travels by the thickness of the steel wire inserts, and multiply the result by the total piston pressure. The result is the pressure of one shoe against the wheel, and four times this is the total braking power.

The brakes may be used for the purpose of overcoming the effect of gravity, as well as for overcoming the effect of inertia.

The retaining valve is especially designed for this purpose, by holding 15 pounds pressure in the brake cylinder during release and recharging of the auxiliaries. This is about one-quarter of the pressure with an emergency application, and in the first case considered under retardation would amount to $\frac{9.14}{4} = 2.3$ per cent, and with the .35 added for speed resistance, a retarding force of 2.65 per eent of the weight of train.

From the "Physical" diagram of plate 26 we find that this force corresponds to an up grade of 140 feet per mile, so that if the retainers be held when running down a 185-foot grade, the train would accelerate its speed due to a grade of 185 — 140 = 45 feet to the mile only, and if the grade were 140 feet or less, there would be no acceleration while the retaining valves acted. A similar effect is caused by making a 5 to 8 pound reduction, which will give a pressure of about 15 pounds in the brake cylinders, and holding the valve on lap. A greater or less continuous resistance may thus be afforded by regulating the air pressure from the engineers' valve.

POWER CONSUMED.

That there is a great deal of power consumed in stopping a train is evident by the temperature assumed by the brake-shoes and wheels, when a stop from a high speed has been made. An apt illustration of this was brought out by F. W. Sargent in a paper read before the New England Air Brake Club in August, 1902. He said, in part: "We burn coal in the firebox of the locomotive, a portion of the heat is taken up by the water in the boiler, converting it into steam, and this steam, through the medium of the cylinders and pistons, is transformed into motion. It may take from 5 to 15 minutes to get the train under way and up to speed, during which time much coal has been consumed and heat generated. There has been some loss due to friction and air resistance, but the greater part of the heat from the coal has imparted motion to the train and is measured by the energy stored therein; to stop this train by the brake, means a reversal of the process described—that is, the transformation of energy into heat, and all the heat must be generated at the face of the brakeshoe, not in 10 to 15 minutes, but in perhaps as many seconds, if our train is equipped with modern brakes. This means a very high rate of heat generation, neatly described as follows:

"The highest rate of conductivity can be but slow in comparison with the speed at which the immediate rubbing surface of the brakeshoe acquires temperature, when such a quantity of heat is generated in so short a time."

The work of braking falls ultimately upon the air pump. This has grown by increments from 6 inches in diameter to 8, 9½ and 11 inches. If the main reservoir be large—not less than 20,000 cubic inches on passenger and double that on freight engines—the pump is benefited, as it permits a sufficient volume of air to be compressed, while the brakes are applied to release and recharge without running the pump at a high rate of speed. The 8-inch pump supplies about 25 cubic feet of free air per minute and raises it to 90 pounds pressure, and the 9½-inch pump furnishes about 45 cubic feet of free air, but even the larger pumps are often taxed to their limit in an effort (sometimes fruitless) to supply the demand caused by a leaky train line and poorly adjusted brakes. This is not only wasteful, but sometimes results in the "cutting out" of air braked cars by the trainmen in order to get a fair service. It is seldom that we find a train of freight cars absolutely tight, and a few small leaks waste a large amount of air. On a heavy grade the power of the boiler is taxed to the uttermost, and steam wasted by pumping air to supply leaky train pipes means just that much less work performed by the engine in hauling the train. Long piston travel wastes air in two ways—first, in making it necessary to exhaust more air from the train pipe in order to obtain a given force in the brake cylinder, which air must be supplied at release, and as illustrated by plate 25; and, secondly, by consuming an extra volume of air in the brake cylinder, represented by the unnecessary piston displacement, and which must be made up by the pump when recharging. This may mean a waste of one-third of the air compressed. A slack adjuster will save the waste in the latter case, besides providing uniform braking action throughout the train.

CYLINDER BRAKES.

The air brake, which we have just studied, is strictly a friction brake; that is, the rotation of the wheels is impeded by the friction of the brakeshoes, and the work done during retardation is expended in heating the brakeshoes and wheels. In a locomotive we can, however, perform work of another kind by means of the pistons and valve gear with which it is provided, and this work will also have a retarding effect. In fact, we can convert our locomotive into a temporary air compressor, in which the inertia of the train will be the power, and the work performed in compressing air will effect the retardation of the train. This arrangement we have termed a "cylinder brake," and there are several varieties existent.

In order to make the action of such brakes clear, the Zeuner diagram of plate 8 is reproduced in Fig. 89, to a slightly smaller scale. Three valve circles are drawn, full gear forward in fine line, full gear backward in heavy line and mid-gear position, in fine line. The rotation throughout this explanation is assumed in the direction of the arrow; that is, as running ahead. If, now, the engine be drifting, with closed throttle, and reverse lever in front corner, and the speed be moderate, a diagram taken from the cylinder would have very little area, and would be practically as shown by the fine line indicator card at the bottom of the figure, and there would be no resistance offered by the cylinders. Suppose now that we bring the lever to mid-gear. Starting at the forward stroke of the piston, and considering pressure ahead of it only, we see that air would be forced out of the exhaust pipe until the valve closed at b. Having no escape, the air would compress ahead of the piston until c is reached, when the opening to the steam chest and steam pipes allows the air confined ahead of the piston to escape thereto, and as the volume is large, the pressure during the rest of the stroke to d will be practically constant. On the return, or back stroke, the reverse is true, the pressure in the pipes, etc., follows the piston to e, when closure of the port allows the air to expand, so that by the time release occurs at f, there is little pressure back of the piston. In this case as much work as was performed in compressing air is given out again during the next stroke, and the total work is

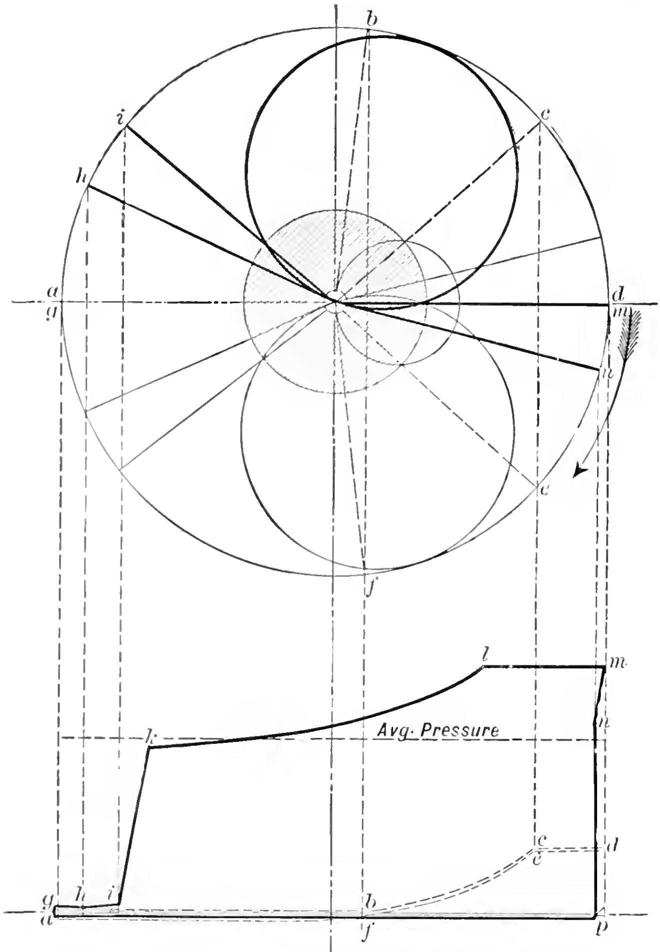


Fig. 89

nothing. (At high speeds the friction of the air entering the passages would produce a card showing considerable work, but we are not considering that condition of speed at this time.)

Now let us put the lever in full gear back motion, while the engine is still running ahead, and observe the action. Commencing again at the back end, g, we find by referring to the heavy valve circle and angular references that air will be driven out of the exhaust pipe until compression begins at h. The valve has now closed the exhaust port, and the air ahead of the piston is compressed slightly to i, at which point the valve opens to the steam chest.

In connection with the latter are the cylinder passages, steam pipes and dry pipe in boiler, the total volume of which is about twice the volume of one cylinder. Let us assume that connected with these pipes and steam chest there is a safety valve set at 90 pounds. Then when the valve opens at i, there will be 90 pounds in the chest and pipes, and this will rush into the cylinder, raising the pressure ahead of the piston, say, to 60 pounds, as at k. As the piston advances the pressure now rises slowly, as the air is being compressed in the large volume of the steam pipes as well as the cylinder, until 90 pounds is reached at l, when the safety valve opens, preventing a further rise in pressure to m, the end of the stroke. Here the valve closes the chest, and as the piston advances on its return stroke the air back of it expands until the exhaust port opens at n. The small volume of air soon equalizes at p with the atmospheric pressure in the exhaust pipe, and to the end of the stroke g, air is "sucked in" by the receding piston. Thus, for each stroke of the piston nearly a full cylinder volume of free air is compressed and pumped into the steam chest, and the work done is represented by the area g, i, k, l, m, n, p, g. In the case considered, the mean pressure is about 65 pounds per square inch, and for the engine which we considered in our calculations upon train braking, and whose valve motion may be represented by Fig. 89, we would have a resisting force at the circumference of the drivers of $\frac{65 \times 400 \times 26}{80} = 8{,}450$ pounds. In the calculations referred to, we found a frictional resistance due to the brakeshoes of 8,190 pounds, so we see that the compression of air by the pistons of the locomotive would be as powerful for braking purposes as the regular air brakes, as far as the drivers are concerned. In addition to the retarding effect upon the drivers of the engine, a supply of compressed air is furnished, which may be utilized for operating the brakes upon the rest of the train. This brings us to one form of cylinder brake, known as the "Sweeney," which has been used on a number of western roads having heavy grades. It is not intended to displace the regular air pump, but to act as an auxiliary, or if the pump fail, a train can be brought down a grade with the "Sweeney." It is this that limits the compression to 90 pounds, the pressure carried in the main reservoir. A pipe is tapped into the top of the steam chest, and thence leads to the main reservoir. A stop cock is placed in the pipe, arranged with control in the cab. There is also a check valve in the pipe to prevent discharge of air back from the main reservoir, and a safety valve which stops the overcharging of the main reservoir by allowing any excess above 90 pounds to escape to the atmosphere. In operating the Sweeney brake, the throttle is, of course, closed, and the stopcock in the connecting pipe just described is opened. The reverse lever is then brought gradually into the back part of the quadrant (assuming that the engine is running forward) and the pistons compress the air drawn in through the exhaust pipe, and force it into the steam chest and pipes, and also through the connection into the main reservoir. When the latter is fully charged, the excess air escapes through the special safety valve. The amount of resistance which the compression of air offers to the movement of the pistons depends upon the position of the reverse lever, as an intermediate position between full and midgear will give a reduced resistance.

While this seems like quite an attractive proposition, there are several drawbacks to its use. The air drawn in by the pistons is "sucked in" the exhaust pipe, and this means that the hot smokebox gases, soot, cinders, etc., will pass through the valves and cylinders of the engine and be deposited in the main reservoir, from which they will work their way through the train line and triple valves, gumming them up and leaving deposits of soot, cinders and such materials to interfere with the working of the air brake. Besides, the cylinders and valves themselves will be badly cut and damaged by the cinders passing through them. The heat of compression will also affect the lubrication of the cylinders and valves. If the air drawn in be at a temperature of 100 degrees Fahrenheit, compression to 90 pounds will raise it to nearly 500 degrees, a temperature greater than that of steam at 500 pounds per square inch pressure, which will unfavorably affect the packing, as well as the lubrication.

In order to obviate the troubles of dirt in cylinders and brake system, as well as the high temperature due to compression of air already heated in the smokebox, the Le Chatelier brake has been devised. The operation is similar to that of the Sweeney brake, as far as its retarding action upon the drivers is concerned, but wet steam is admitted in the exhaust cavity of the cylinders, thereby excluding the hot gases, and reducing the temperature of compression. The compressed vapor is not carried to the air brake system, but will lift the throttle and find its way back into the boiler, and the amount of resistance is regulated by the position of the reverse lever. A globe valve is set in the boiler, below the water level, and within easy reach of the engineer, and is connected to a pipe which branches and enters the exhaust passage of each cylinder. When it is desired to use this brake (the throttle, of course, being closed) water is admitted by the globe valve through the pipes into the exhaust cavity of the cylinders. As the temperature in the boiler is high, due to the pressure carried, as soon as this water enters the exhaust cavity at atmospheric pressure, a portion is converted into steam, and this prevents the suction of the smokebox gases, the steam and wet vapor being drawn into the cylinders instead. The compression and temperature due to same is sufficient to re-evaporate moisture, which finds its way into the cylinder, thus also affording lubrication to the pistons; the moisture also prevents superheating, as if there be enough admitted, the steam will be saturated throughout the stroke. Suppose, for instance, that compression raised the vapor to 200 pounds pressure (which would be necessary in order to overcome that boiler pressure and force its way past the throttle valve), then the temperature of the steam, if normally saturated, would be 388 degrees Fahrenheit, against 500 degrees for air with adiabatic compression to 90 pounds.

After admitting the water, the reverse lever is brought one or two notches back of the center of the quadrant. The proper amount of water to be supplied can be determined by the discharge from the cylinder cocks. If the steam so escaping is densely white, the supply is sufficient. If too much water is given, the excess will come out of the stack. The regulation of speed is obtained by moving the reverse lever, toward the corner for increased braking power, and toward the center for less The water valve need not be changed after once properly adjusted.

Both of these forms of brakes are intended more particularly for the control of trains on heavy grades, and not for ordinary stops. When train brakes are used in unison, the air driver brake must be cut out or slid driving wheels will result. It is customary to provide a cock for the purpose of releasing the air from the driving brake cylinders. This should be at least ½ inch diameter, and it is necessary that it be opened whenever the cylinder brakes are tested or used for continuous braking. They should always be used with great caution, as injudicious handling may break cylinder heads, and destroy the power of the brake when most needed; the speed should not be high, as they are difficult to operate at a velocity greater than 20 miles an hour.

The Baldwin Locomotive Works have introduced a "back pressure brake," which embodies some of the features of both the Sweeney and Le Chatelier brakes. The arrangement is shown in Fig. 90. A pipe connects the globe valve in the cab with the exhaust passages in the cylinder, as shown at A, same as in the Le Chatelier arrangement. There is, however, an auxiliary air inlet in each cylinder permitting air to enter directly into the exhaust passage, the opening being controlled by a valve, C, which valve closes when the engine is working steam. This can also be done from the cab, by means of the levers and rods illustrated. A flap cover B for the exhaust pipe is operated in unison with the valves C, and prevents cin-

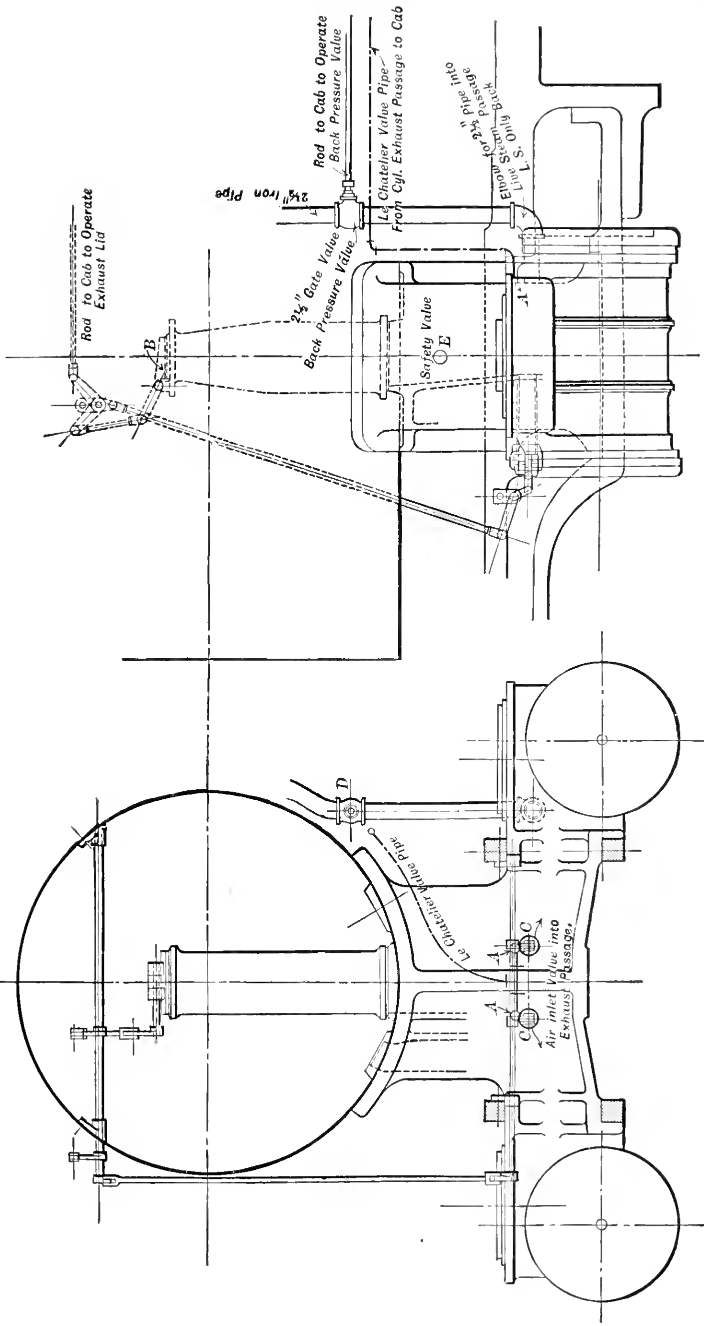

Rod to Cab to Operate Exhaust Lid
Rod to Cab to Operate Back Pressure Valve
Le Chatelier Valve Pipe
From Cyl. Exhaust Passage to Cab
2½" Iron Pipe
2½" Gate Valve
Back Pressure Valve
Safety Valve
E
B
C
A
Elbow for 2½" pipe into Live Steam Passage Back L. S. Only
D
Le Chatelier Valve Pipe
Air inlet Valve into Exhaust Passage.

Fig. 90.

ders and hot gases from the smokebox entering the cylinder. A pipe of liberal size is screwed into the steam passage, and controlled by a gate valve D, which is operated from the cab. A steam gauge, also in the cab, indicates to the engineer the amount of compression obtained. There is a safety valve connected with the steam passage, in order to prevent excessive pressure, seen at E. The operation is as follows: When the engine is drifting forward, the reverse lever is placed in full gear back, and the globe valve in cab is opened, supplying steam to the exhaust passages; the lever connected with the valves C and damper B is also thrown so as to open the former and close the latter. The mixture of steam and air is compressed by the pistons, and is relieved as desired by the gate valve, thus controlling the pressure against which the pistons work, and thereby regulating the speed of the engine. By fully opening the gate valve D, the air and steam will pass freely out of the pipe, and little retardation will be effected, but when the valve is nearly closed the back pressure will soon reach a high figure, and may even completely lock the drivers. The gauge assists the engineer in making the proper regulation of the valve. Trains may be brought down a long grade in this way without applying brakes to the car wheels, thus obviating the serious heating of tires, wheels and brakeshoes, as is experienced with the friction brake. By using both steam and air in the control, several advantages are obtained. The quantity of steam used is less than where it is the only medium employed, and the compressed air will not suffer condensation from passing through the pipe, as would be the case with steam at slow speeds. The steam (generated from the water admitted at the temperature of the boiler) assists in the lubrication of the cylinder and valve, and also provides a means of reducing the high temperature which would otherwise result from the adiabatic compression of the air, as in re-evaporating this water it will absorb heat units to the amount of its weight by its latent heat of evaporation. The reverse lever is allowed to remain in full gear back, the speed being easily adjusted by the gate valve. The driver brakes must be cut out, or slid flat tires may result, but the other brakes will be operative by the air brake as usual. Care must be taken not to allow too much water to enter the cylinders, or broken cylinder heads and pistons will probably occur.

The braking power will be as determined in connection with our study of Fig. 89, but the final pressure of compression will depend upon the adjustment of the gate valve, and as indicated by the gauge in the cab. In order not to overstrain the working parts, it should never much exceed the boiler pressure, but as the safety valve E must retain boiler pressure when working, it is likely to reach that figure. As we saw from our hypothetical diagram in Fig. 89, the mean average pressure will be always considerably below the maximum or limiting pressure, so that even if the latter reached boiler pressure, there should be little danger of skidding the wheels, provided that the air brake on drivers is cut out. The maximum available tractive force with steam is about $\frac{.8 \text{ P } d^2 \text{ s}}{\text{D}}$, as previously explained, and with compression to boiler pressure, we are not likely to have over $\frac{.7 \text{ P } d^2 \text{ s}}{\text{D}}$ for our resistance due to compression, which should prevent skidding the wheels, unless it were started by a slippery place in the track, when it would be likely to continue until the valve released the pressure. It may be of interest here to note that engines with this brake are brought down the Pike's Peak Rack Railroad every trip, having one passenger car behind them, the grade at the steepest part being about 25 per cent; the engineer simply regulates the speed by the gate valve, and can bring the train to a dead stop by closing it.

If the boiler pressure be 200 pounds, and we allow the compression to reach this figure, we will have approximately, as stated above, $\frac{.7 \times 200 \times 400 \times 26}{80} = 18{,}200$ pounds retarding force, over twice as much as we obtained with the Sweeney or the friction driver brake, and the engine should, with this brake, be able to hold back on a down grade as heavy a train as it can pull up the same gradient.

CHAPTER VI.

STEAM CAPACITY.

The capacity of a locomotive, or, more strictly, a locomotive boiler for generating steam, has come to be looked upon as the most vital feature connected with a locomotive. In the early days, comparatively little regard was paid to this part of the problem, and if the cylinders were large enough to pull a good sized train up the maximum grade, and the driving wheels were sufficiently loaded to enable the cylinders to do their work, the results were considered satisfactory. As the demand for greater speed was made, and at the same time an increased load desired, it was soon found that the speed of the engine with a given train was limited by the capacity of the boiler, and complaints were made of the engine "not steaming," which, while it was a logical reason for not being able to make a fast schedule with a heavy train, did not give the full burden of the trouble. The lack of steam might be the result of poor fuel, improper firing, bad adjustment of the front end, flues choked up, or a number of ailments, all of which would reflect upon the local management as showing negligence in the care or manipulation of the engine; but it soon came to be recognized that no matter if the machine was in the best condition and operated by skillful men, it was a physical impossibility to make a small boiler generate sufficient steam for large cylinders operating at a good rate of speed. So the boiler grew—not in a very rational manner at first, as the best proportions were not well known, nor are they now, but it was largely a guess how much heating surface and grate area were necessary to maintain a definite tractive force at a specified speed.

While, as we have stated, this is one of the principal problems connected with locomotive design, we are still considerably in the dark as to exact values; the great variety of fuels, and, we may even say, of a definite fuel, combining such different proportions of fixed carbon, volatile matter and ash, renders it extremely difficult to obtain absolute figures.

HEATING SURFACE.

While the various dimensions of the several parts of a boiler are all more or less important in their bearing upon the generation of steam, the amount of heating surface is, as a rule, paramount. The size of grate fixes the limit to the amount of coal that can be burned in a given time, as an hour, for instance, but the proportion of coal burned per unit of heating surface governs the rate of evaporation greatly. It is true that we have so many heat units in a pound of coal, and even if our heating surface be extremely meager, we shall get considerable evaporation—more per unit of surface, in fact, than if the surface be liberal, but not by any means in the inverse ratio of the surface, as our economy will be diminished. Locomotives have been operated practically without firebox heating surface: that is, covered with fire brick, and still there has been sufficient steam produced to supply the cylinders. The lower the rate of combustion per square foot of heating surface, however, the more water will be evaporated per pound of coal. The ratio of tube heating surface to firebox heating surface varies between quite wide limits; in some cases it is only nine times as great, and in others 18 times as large. There must be some certain proportion that is better than all others, but we do not know at this time just what that "best" proportion is. Modern engines have increased the ratio enormously.

The length of tubes has an effect upon the capacity of the boiler. More than 10 years ago M. Henry, of the Paris Lyons & Mediterranean Railway, conducted a series of elaborate experiments to determine the most economical length of tube. The vacuums used were low—not over 3 inches of water—much less than what we are accustomed to in this country, but while the best steaming length with his vacuums was about 14 feet, it is no doubt true that American practice can and does use a longer tube satisfactorily, not only as far as rate of evaporation per pound of fuel is concerned, but also per pound of boiler, which is equally important. Twenty-foot flues are as common to-day as 16-foot flues were 10 years ago. As long as the temperature of the smokebox gases is greater than the temperature of the steam in the boiler, we have some transfer of heat at the coolest end of the flues, even if not great, but the draft on the fire may be reduced by the extra length, so that there is not as much fuel burned on the grate, and this is what determined M. Henry's 14-foot length—our vacuums being greater, the length would naturally be longer for maximum capacity. This is a question that cannot be satisfactorily treated without actual tests, and it is hoped that such experiments will be made in the near future to determine this important question.

As a rule, locomotive boilers are made as large as possible, in order not to exceed the total weight desired for the engine. This is not a scientific proceeding, but in order to obtain all the steam possible for a given engine, the weights of the various parts are kept to the minimum, so that the boiler may have the benefit. No locomotive has ever been spoiled by being too free a steamer, and the greater the boiler power, the higher the speed that can be maintained. It is essential, however, to be able to say what performance can be expected from a given design, which is the converse of building an engine to perform a stated amount of work.

The condition of the flues and firebox heating surface has an effect upon the steam-making qualities of the boiler—if the surfaces are coated with a heavy scale, the heat transmission will not be as rapid as if they are clean. In 1898 some experiments were made on the Illinois Central Railroad in connection with the University of Illinois (see Railroad Gazette of January 27, 1899), to demonstrate the effect of scale upon boiler efficiency. A locomotive which had been in service 21 months was tested in the roundhouse. The engine then went to the shops, received new tubes and a thorough cleaning, after which the test was repeated. The average thickness of scale on the principal heating surfaces was 3-64 inch, a total of 485 pounds being removed from the boiler, showing an average analysis as follows:

Silica	15 per cent
Iron and alumina	6 per cent
Calcium carbonate	44 per cent
Magnesium carbonate	3 per cent
Calcium sulphate	14 per cent
Magnesia	10 per cent
Undetermined, etc.	8 per cent
	100 per cent

Before cleaning, there was an evaporation per square foot of water heating surface per hour from and at 212 degrees Fahrenheit of 5.89 pounds for one test and 6.09 pounds in the other, averaging 5.99 pounds of water for both tests, the rate of combustion being .94 pounds of coal per square foot of heating surface per hour. After removing the scale, with a rate of combustion of .97 pounds of coal per square foot of heating surface per hour, the evaporation was 6.81 and 6.76 pounds, or an average of 6.78 pounds of water per square foot of heating surface per hour, an increase of 13 per cent in the steam-making capacity of the boiler.

While the covering of a boiler does not increase the heat units transmitted to the water, yet the reduction in radiation prevents the loss by condensation of a part of the steam, and so increases the output of the boiler, and the better the covering, the greater will be this output. This is the more necessary with the high pressures and correspondingly high temperatures of the present day. The results of some tests of boiler coverings were reported at the Master Mechanics' convention of 1898, which indicated that .34 British thermal units were radiated from each square foot of external surface per hour per degree difference of temperature between the steam and outside air, when the boiler was lagged with mineral wool, and 1.10 units when covered with wood and sheet iron. We can readily determine what this difference in the covering means to steam capacity. The difference is $1.10 - .34 = .76$ British thermal units per degree. Now steam at 200 pounds pressure (absolute) has a temperature of 382 degrees, and if we take the air at 50, the difference in temperatures is 332 degrees. Modern boilers have 500 square feet of outside surface, and the latent heat of 200-pound steam is 845, so that we have for the amount of steam condensed per hour,

$$\frac{.76 \times 332 \times 500}{845} = 150$$

pounds more with wood lagging than with mineral wool, or some equally efficient covering. This represents about 5 horsepower. The number of British thermal units per square foot per hour per degree difference between inside and outside temperatures which are transmitted by different kinds of jackets, is given below:

Rock wool	.255 British thermal units
Mineral wool	.340 British thermal units
Magnesia	.384 British thermal units
Hair felt	.422 British thermal units
Fire felt	.502 British thermal units
Wood and sheet iron..	1.100 British thermal units
Bare (no covering)...	2.076 British thermal units

Of course, a difference in thickness of these several coverings will cause a variation in the amount of heat transmitted. The above represent average values.

The speed of the engine also affects the result. In 1899 some tests made on the Chicago & Northwestern Railway with a locomotive, having 219 square feet of covered boiler surface and 139 square feet uncovered, showed a condensation representing 4.5 horsepower when at rest, and 9 horsepower when pushed at 28 miles an hour speed, the temperature of the feed water being 80 degrees and 26 pounds of steam per hour being considered as equal to a horsepower.

Leaks, choked-up flues and the dozen or more ailments to which locomotive boilers are subject, all reduce their capacity, if only temporarily, and are referred to here merely to indicate what an inaccurate solution will probably result from an attempt to prognosticate accurately the work which a boiler is capable of performing. From the many tests which have been made and reported, we can obtain data that will enable us to foretell close enough for practical purposes, however, about what can be expected under stated conditions, and this we will now endeavor to present in such form that it may be used with a fair approximation to road service.

It is necessary that we distinguish carefully between the maximum capacity and the ordinary working capacity. If we take the results for a whole trip over a division of, say, 150 or 200 miles, and average them, as is usually done, we find that comparatively low values are obtained, both for coal consumption and steam capacity. Thus, while the results from a trip might show an average coal consumption of 100 pounds per square foot of grate per hour, and an evaporation of 5 pounds of water per square foot of heating surface per hour, yet the actual amounts at the critical points of the road may easily have been double this. When we consider the undulating nature of most railroads, and the fact that the engine may be coasting at least one-third, if not one-half of the time, it will be readily understood that the amount of work done while ascending the grades is greatly in excess of the average. The only satisfactory way in which to obtain data as to maximum quantities is to test the engine upon an especially prepared plant, where the conditions of heavy work can be maintained uniform for a considerable period of time, and not be dependent upon a varied profile or unexpected orders. Fortunately, there are a number of such outfits now in this country, and the results obtained from them have been of very great value.

GRATE AREA.

In coal burning engines the grate area is of prime importance, as it regulates, not only the economy, but also the amount of coal which can be burned. It is therefore necessary to consider the proportion of heating surface to grate area, which we will designate as $R = \frac{\text{Heating Surface}}{\text{Grate Area}}$, both, of course, in the same unit, which is ordinarily taken in square feet.

At the 1902 convention of the Master Mechanics' Association a committee, reporting upon the improvements in boiler design and proportions of heating surface and grate area for burning different kinds of coal, gave a statement of the values of R as found upon locomotives built within recent years:

RATIO OF HEATING SURFACE TO GRATE AREA.

Fuel.	Passenger.		Freight.	
	Simple.	Compound.	Simple.	Compound.
Free burning bituminous	65 to 90	75 to 95	70 to 85	65 to 85
Average bituminous	50 to 65	60 to 75	45 to 70	50 to 65
Slow burning bituminous	40 to 50	35 to 60	35 to 45	45 to 50
Bituminous slack and free burning anthracite	35 to 40	30 to 35	30 to 35	40 to 45
Low grade bituminous, lignite and slow burning anthracite	28 to 35	21 to 30	25 to 30	30 to 40

Bituminous coal, being quick burning and of good heating value, does not require such a large grate, therefore the ratio R is higher. The low grades, such as lignite, need a large grate, like anthracite, but for an entirely different reason. The latter is efficient from a standpoint of heat units developed per pound of fuel, but the rate of combustion is slow, and in order to burn enough to generate the desired quantity of steam, the grate must be large. On the other hand, lignite is a quick-burning fuel, but has a comparatively small heating value, as it is low in fixed carbon, and a large grate is needed in order to burn enough of it.

As bituminous coal is the chief locomotive fuel in this country, our information regarding its use is more complete than for the other grades. Prof. Goss made extended experiments with the locomotive testing plant at Purdue University, and has "served" as much as 240 pounds of Brazil block coal per square foot of grate per hour. We say served, because it was not all burned, as was evident from the quantity and quality of the "sparks" or cinders expelled from the stack. The author made a series of tests with a 10-wheel freight locomotive on the Chicago & Northwestern Railway's plant in 1900, and maintained a rate of combustion or "coal serving" of 205 pounds per square foot of grate per hour for a continuous period of 50 minutes. It is probably safe to say that 200 pounds per square foot per hour is the maximum which can be maintained for any length of time, and ordinarily in average service, the rate will not be over one-half that amount, or 100 pounds per square foot of grate. Of course, with a high rate of combustion, the economy will not be great, but we are now examining the question of capacity only—economy will be taken up under "Coal Consumption."

In the Purdue experiments, when coal was burned at the rate of 180 pounds per square foot of grate, the evaporation was 12 pounds of water per square foot of heating surface per hour, or 14½ pounds from and at 212 degrees Fahrenheit. The ratio R was 70, which gives us a definite idea of the relationship between "R" and "w," if by "w" we understand the pounds of water that can be evaporated per square foot of heating surface from and at 212 degrees per hour as a maximum. In a similar manner, the Chicago & Northwestern tests gave 13½ pounds from and at 212 degrees as a fair maximum value, the ratio R being 80 in this case.

By the aid of the Master Mechanics' committee report on grate area ratio, etc., of 1897, we can calculate the corresponding values of w for other values of R, when we consider 200 pounds per square foot of grate area as the maximum hourly rate of combustion. For the western lignites, the rate of combustion is greater, and will probably offset the smaller heating value, so that for given ratios R, the evaporation w can be taken about the same as for Indiana and Illinois coal.

On the other hand, anthracite coal is slow burning, and we must take a different figure for our maximum. From available records it appears that 100 pounds of coal burned per square foot of grate area per hour is about the most that can be expected for the large sizes of anthracite coal, and with the small sizes we cannot obtain much over 60 pounds, as the coal packs so closely on the grate that it is difficult for the proper supply of air to force its way through the fuel bed.

Fig. 91 illustrates the maximum evaporation in pounds of water per square foot of heating surface per hour from and at 212 degrees Fahrenheit that we can expect to obtain under ordinary conditions. These curves will generally apply only when ascending the limiting or heaviest grades on a division, or in passenger service at particularly high speed, and will perhaps never be reached as the average of a whole trip. As we explained above, they must not be considered as absolutely correct for any particular case, as the contingencies will affect the result one way or the other. They do, however, represent average practice and results as obtained in this country. In order to find the pounds of steam formed at any given pressure, the proper factors of evaporation must be used as divisors of the values obtained from Fig. 91, as those are "from and at" 212 degrees Fahrenheit.

The table below gives the factors of evaporation for different steam pressures and temperatures of feed water, or the ratio of water evaporated from and at 212 degrees to that evaporated under the conditions existing, for a given amount of heat generated:

FACTORS OF EVAPORATION.

Temperature of Feed Water.	Steam Pressure by Gauge.									
	150	160	170	180	190	200	210	220	230	240
32	1.236	1.237	1.239	1.240	1.241	1.243	1.211	1.215	1.216	1.247
10	1.227	1.229	1.230	1.232	1.233	1.234	1.236	1.237	1.238	1.239
50	1.217	1.218	1.220	1.221	1.223	1.224	1.225	1.226	1.228	1.229
60	1.207	1.208	1.210	1.211	1.212	1.214	1.215	1.216	1.217	1.218
70	1.196	1.197	1.199	1.200	1.202	1.203	1.205	1.206	1.207	1.208
80	1.186	1.187	1.189	1.190	1.192	1.193	1.194	1.195	1.196	1.198
90	1.176	1.177	1.179	1.180	1.181	1.183	1.184	1.185	1.186	1.187
100	1.165	1.167	1.168	1.170	[.171	1.172	1.174	1.175	1.176	1.177
110	1.155	1.156	1.158	1.159	1.160	1.162	1.163	1.161	1.166	1.167
120	1.145	1.146	1 117	1.149	1.150	1.151	1.153	1.151	1.155	1.156
130	1.134	1.136	1.137	1.138	1.140	1.141	1.112	1.114	1.145	1.146
140	1.124	1.125	1.127	1.128	1.129	1.131	1.132	1.133	1.134	1.135
150	1.113	1.115	1.116	1.118	1.119	1.120	1.121	1.123	1.124	1.125
160	1.103	1.101	1.106	1.107	1.108	1.110	1.111	1.112	1.113	1.115
170	1.092	1.091	1.095	1.097	1.098	1.099	1.101	1.102	1.103	1.101
180	1.082	1.083	1.085	1.086	1.088	1.089	1.090	1.091	1.093	1.094
190	1.071	1.073	1.071	1.076	1.077	1.078	1.080	1.081	1.082	1.083
200	1.061	1.063	1.061	1.065	1.067	1.068	1.069	1.071	1.072	1.073
210	1.051	1.052	1.053	1.055	1.056	1.057	1.059	1.060	1.061	1.062

Referring to the figure, the abscissæ are the ratio of heating surfaces to grate areas R, and the ordinates the rate of evaporation per square foot of heating surface, w. The several curves represent fuel as follows:

Curve "a," large sizes of anthracite coal.

Curve "b," small sizes of anthracite coal.

Curve "c," Pennsylvania and Virginia semi-bituminous coal.

Curve "d," Indiana and Illinois bituminous coal.

Curve "e," fuel oil.

The last curve, e, is seen to be straight, and to have the uniform value for w of 18 pounds. This was obtained from tests made on the Southern California Railway in freight service ascending a 3 per cent grade. In these tests the average evaporation for a 2½-hour run reached 14½ pounds per square foot per hour with steam at 180 pounds and feed water at 70 degrees, and multiplying by 1.2, the factor of evaporation, we have $14.5 \times 1.2 = 17.4$ from and at 212 degrees, or a probable limit of 18 pounds as a maximum. As the size of grate is of

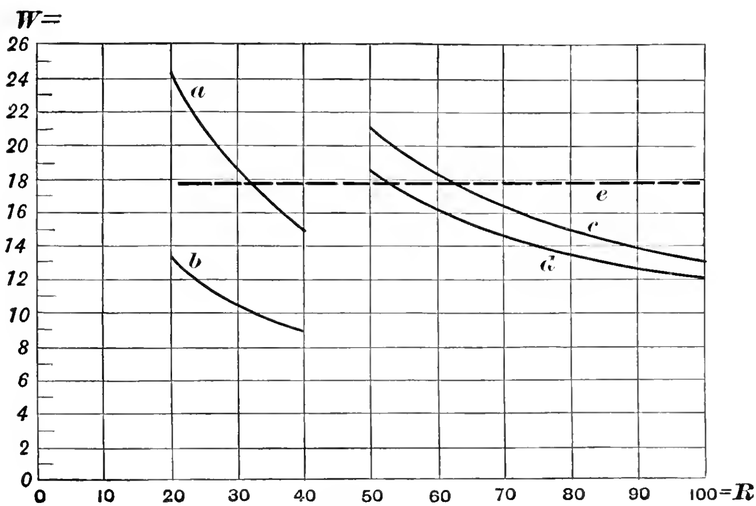


Fig. 91.

no importance in oil burning, in fact, there is no grate, the rate of evaporation has been taken as depending entirely upon the amount of heating surface. It is noticed that the locus lies generally above those for bituminous coal, and it has been clearly demonstrated that the same boiler will generate considerably more steam when burning oil than when using coal as a fuel.

The method of using Fig. 91 hardly needs explanation, but an example will suffice to make it perfectly clear. A locomotive boiler having 3,000 square feet of heating surface and 33 square feet of grate area has a ratio $R = \frac{3{,}000}{33} = 90$. If using Virginia semi-bituminous coal, it would be possible to evaporate 14 pounds of water per square foot of heating surface from and at 212 degrees. If the feed water be at 70 degrees, and the pressure in boiler 180 pounds, we must allow

for the factor of evaporation; thus, $\frac{14}{1.2} = 11.7$, and the total

amount of steam generated per hour at the maximum possible rate will be $3{,}000 \times 11.7 = 35{,}100$ pounds.

DRAFT ACTION.

We have seen that in order to obtain the maximum delivery from a boiler, the fuel must be burned at the greatest possible rate—a rate which has no equal elsewhere, unless in the steam fire engine, as before mentioned. As the stack is so low, we are forced to depend upon the draft produced by the blast of the exhaust. The most elaborate experiments upon this subject have been made at Purdue University, under the direction of Prof. W. F. M. Goss, and the results have been reported at various times in the journals of the different engineering societies.

In 1900 Professor Goss announced a formula for the relation between smokebox vacuum and rate of combustion, which he limited in application to Brazil block coal. Comparisons made with other bituminous coals, however, indicate a rather general application of this formula to bituminous coal at large. If we let

d = the negative pressure in smokebox in inches of water, or the "draft;"

c = the pounds of coal that can be burned per square foot of grate surface per hour, or the rate of combustion,

we have the relation expressed by the equation

$$d = .037\ c,$$

so that if we wish to burn 200 pounds of bituminous coal per square foot of grate per hour, we need a draft $= .037 \times 200 = 7.4$ inches of water at the smokebox.

Our information regarding anthracite coal combustion and front end pressures is very limited, but we believe that by using for the coefficient of c the values .123 and .074 for the small and large sizes, respectively, we will have a fair approximation to actuality. This would make our formulæ stand as follows:

$$\left.\begin{array}{ll}\text{For bituminous coal,} & d = .037\,c \\ \text{For large anthracite coal,} & d = .074\,c \\ \text{For small anthracite coal,} & d = .123\,c\end{array}\right\} \ldots\ldots\ldots\ldots\ldots (94)$$

By applying these equations to the maximum rates of combustion of the two varieties of anthracite coal, 100 and 60, respectively, we find the same vacuum in smokebox, viz., 7.4 inches of water, which is about the ordinary limit in American practice. This vacuum is employed in overcoming the resistance of air passing through the bed of fuel, in drawing the products of combustion through the tubes, and in pulling the gases around the obstructions in the front end, and approximately one third of the vacuum present in the smokebox is absorbed at each place. Thus, if we had 7½ inches in the main body of the smokebox, we should find 5 inches back of the diaphragm and 2½ inches in the firebox. This emphasizes the fact that to obtain the best combustion with the smallest draft we should keep down these resistances by carrying as light a fire as is consistent with keeping fuel on the grate, by keeping the flues well bored out, and by running with as high a diaphragm as possible and yet burn the fire evenly.

In the preamble to the second chapter it was explained that the last work performed by the steam before it left the locomotive was the production of draft whereby the enormous quantities of fuel needed to generate sufficient steam could be burned. It was also explained that this blast was caused by back pressure in the cylinders, which obstructed the motion of the pistons. As the blast of the exhaust steam produces the vacuum in the smokebox, there must be some general relation between the two. The action of the exhaust jet upon the smokebox gases is principally one ot induction, by frictional contact, though it also enfolds and entrains them. The draft seems to be independent of the impulses resulting from individual exhausts and to be nearly proportional to the weight of steam exhausted per unit of time. The vacuum is also different when the front end arrangement is altered; that is, the size and length of the stack and the size and height of the exhaust pipe.

The Master Mechanics' committee of 1896 demonstrated that there was little difference in the efficiency of double and single exhaust nozzles when each is properly proportioned to its work, but for incidental advantages, the committee recommended the single nozzle for general use. The later experiments above referred to showed conclusively that the lower the tip of the exhaust pipe, the better will be the draft generated by a given weight of steam in the unit of time and correspondingly by the same back pressure in the cylinders. The highest stacks, and, when straight, the largest diameters tested, gave the strongest drafts. With the taper stack, however, the diameter did not affect the draft as much as it did with the straight stacks. The diameters tested were 9¾, 11¾, 13¾ and 15¾ inches, and the heights 26½, 36½, 46½ and 56½ inches above the top of the smokebox, which was 54 inches in diameter, the "choke," or smallest part of the taper stack, being 16½ inches above the smokebox. The exhaust tips were all 4¼ inches in diameter, and varied in height from 10 inches below the center of the smokebox to 20 inches above, in steps of 5 inches. While the lowest exhaust pipes and the highest stacks gave the greatest smokebox vacuum for a given back pressure and quantity of steam discharged, so the lowest stacks and the highest nozzles gave the smallest vacuum. The variables in the study are great in number, and even elaborate as were the Purdue tests, we are still without real knowledge regarding smokeboxes of other sizes and with different diameters of exhaust nozzle.

It has been stated that the draft (or smokebox vacuum) was nearly proportional to the weight of steam exhausted, and in comparing the results of the tests we can make an approximate equation for the relationship between the two, bearing in mind that it represents the Purdue tests, and that it is more or less problematical how far its application can be extended to other sizes and conditions. Let $d =$ the draft in inches of water as before, and $q =$ the weight of steam in pounds per second passing through the exhaust pipe. Then the most efficient arrangement, as explained above, will be represented by

$d = 1.35\,q$
and the least efficient by .... (95)
$d = .63\,q$

and by efficiency we mean the greatest draft for the least back pressure or quantity of steam used.

If we consider the back pressure instead of the quantity of steam used, and let p represent this in pounds per square inch, we have for most efficient arrangement

$d = 1.88\,p$
and the least efficient .... (96)
$d = .9\,p$

Thus it appears that the best arrangement of front end is twice as efficient as the poorest, and this emphasizes the importance of using a low nozzle and a large and high stack. In modern engines of great power the latter is impossible, but the diameter and location of exhaust nozzle can largely make up for the lack of height. In all cases the nozzle should be kept low, and with straight stacks the preference should be given to liberal diameters.

It also appears from the tests that a good arrangement of front end and stack is equally efficient under whatever conditions the engine may run; what is good for one speed or cut-off is equally advantageous for all speeds and cut-offs. Of course, an increase in speed or cut-off increases the draft, as the values of p and q are both increased thereby, as whatever increases the volume of steam used increases the draft, but the arrangement will be equally efficient.

If it be desired to increase the draft, and the arrangement is already in accordance with the plans for maximum efficiency, it is not necessary to change its design—the only thing required is to reduce the diameter of the exhaust nozzle, not by bridges, but by a circular bushing. Thus, if a soft coal burner is to be changed to hard coal, and more draft is desired, a reduction in the diameter of the tip should produce the desired result. The back pressure will be increased, and so will the draft, in accordance with formula 96.

MAXIMUM HORSEPOWER.

We have considered the determination of the greatest quantity of steam which can be produced by a locomotive boiler in a unit of time, and have fairly established the rules which govern the same, and it is evident that the capacity of the boiler limits the amount of work that can be done by the engine. It is often more convenient, however, to express the boiler capacity directly in terms of work possible, as, for instance, of so many horsepower. While this gives a very fair idea of work possible of accomplishment—more so, in fact, than the mere statement of the number of pounds of steam generated per hour—yet it is very ambiguous as far as the boiler itself is concerned. A given boiler will produce more horsepower with a compound than with a simple engine, or more with an early and economical cut-off than with a later and more wasteful one, so that the resulting power must be dependent upon the boiler and the design of the cylinders, etc., as well as the method of operation. Thus, we see that while the statement of the horsepower of a boiler gives us an expression which appeals to our minds more readily than the amount of steam generated, it is not an accurate indication of the capacity of the boiler, and if exact computations are desired, they should be based upon the possible steam production from and at 212 degrees.

The American Society of Mechanical Engineers considers the production of 30 pounds of steam at 70 pounds pressure evaporated from feed water at a temperature of 100 degrees Fahrenheit as equivalent to one horsepower. This requires the same amount of heat that is needed to evaporate 34½ pounds of water at 212 degrees into steam at atmospheric pressure, or as it is commonly termed, "34½ pounds of steam from and at 212 degrees." In order to obtain the equivalent evaporation from any temperature to any pressure, we must use the table of "Factors of Evaporation" previously given. Thus we find, from 100 degrees of water to 70 pounds of steam, the factor is 1.15, and $\frac{34\frac{1}{2}}{1.15} = 30$, or the Association of Mechanical Engineers' standard given above. This unit is seldom used in connection with locomotives, however, as the output of the machine as a whole is considered, and, as explained, this depends upon the efficiency of the cylinders.

Professor Goss estimates the indicated (or cylinder) horsepower (= I. H. P.) of a simple locomotive at .43 times the heating surface in square feet; as an evaporation of 12 pounds of water per square foot of surface per hour was obtained at Purdue and as 28 pounds of steam is considered to be a representative figure for a horsepower in modern simple locomotives, then $\frac{12}{28} = .43$ times the heating surface, or 2 1-3 square feet per horsepower. Using the coefficient of evaporation, usually about 1.2 for current conditions, we have $28 \times 1.2 = 33\frac{1}{2}$ pounds of steam from and at 212 degrees for a horsepower.

The water rate for the Chicago & Northwestern tests averaged about 28 pounds per horsepower hour; the maximum I. H. P. was about 1,000, and as the heating surface was 2,332 square feet, we again find .43 I. H. P. per square foot of heating surface, or one I. H. P. for every 2 1-3 square feet.

It should be remembered that 28 pounds per horsepower is merely an average rate, for simple engines, under the conditions of the test. If a late cut-off is used, reducing the benefits of expansion, the water rate will be greater, reducing the horsepower per square foot of boiler heating surface; then the rate may be lower at certain speeds and cut-off, increasing the horsepower rate. Thus in the Chicago & Northwestern tests, a water rate of 24.3 pounds per I. H. P. per hour was obtained at 20 miles an hour and with 19 per cent cut-off, and a 32.6 rate at 16 miles per hour and with 70 per cent cut-off. This will be treated at length further on, but is referred to here in order to demonstrate that the values given as convenient units are only approximate. Then Fig. 91 shows that much depends upon the grate ratio, R.

With compound engines the saving in steam consumption as compared with simple engines is a very uncertain quantity. When the cylinders, pistons and valves are all in good condition there may be from 10 to 20 per cent less steam used per I. H. P. hour in a compound than in a simple engine, but when not in prime order there is little difference. If we allow .9 as much steam used for a compound as for a simple engine, we will probably not be far from the truth. This would make an allowance of about 2 square feet of heating surface for a horsepower. Then, again, in a simple engine with a late cut-off, say, at .9 stroke, the steam consumption would probably reach 35 pounds per I. H. P. hour, or, say, 3 square feet of surface per horsepower.

It seems from the above discussion that we may use, for approximate ratios, the following allowance of heating surface per indicated horsepower:

Compound locomotives	2	square feet
Simple locomotives (early cut-off)	$2\frac{1}{3}$	square feet
Simple locomotives (late cut-off)	$2\frac{2}{3}$	square feet
Simple locomotives (full stroke)	3	square feet

By early cut-off is meant under half stroke; late cut-off, ½ to ¾ stroke, and full stroke with lever at or near the corner, as when ascending heavy grades.

The manner in which speed and cut-off affect the maximum horsepower of a locomotive is worth study. Fig. 92 is intro-

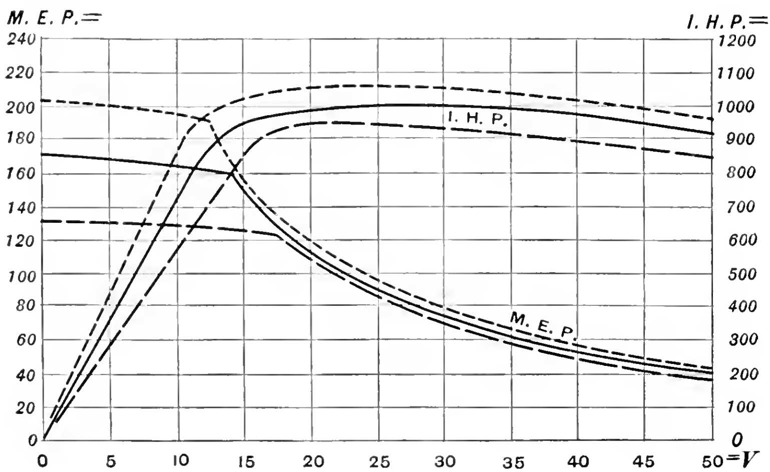


Fig. 92.

duced to make this clear. The abscissæ represent speed in miles per hour = V. The scale of ordinates on the left indicates mean effective pressures back of piston, and is used for the M. E. P. curves. The indicated horsepower is scaled at the right, and is to be used for curves marked I. H. P. The solid lines show the M. E. P. and the I. H. P. obtained in the Chicago & North-western tests. The M. E. P. line is taken from plate 12, and shows the greatest mean effective pressure obtained at different speeds with the normal boiler pressure, 190 pounds. The reverse lever was kept in the corner notch with increasing speeds until the boiler would no longer supply the necessary amount of steam (about 15 miles an hour), when it was moved back, and this had to be done with increasing speeds, until at about 30 miles an hour the cut-off was only 40 per cent, and for higher speeds it was not necessary to reduce the period of admission for the speed of the engine and valve travel continuously reduced the amount of steam admitted at each stroke with increases of speed. Now, having the greatest M. E. P. at each speed, the I. H. P. is readily obtained from equation 55,

$$\text{I. H. P.} = \frac{\text{M. E. P.} \times d^2 s V}{375 D}$$

and for the engine being discussed, having 20-inch diameter by 26-inch stroke cylinders and 63-inch driving wheels, this becomes

$$\text{I. H. P.} = \text{M. E. P.} \times V \times \frac{20^2 \times 26}{375 \times 63} = \text{M. E. P.} \times V \times .44$$

that is, we simply multiply together the M. E. P. as obtained from the curve, the speed at the point selected and the constant factor .44. Thus, at 30 miles an hour, we find the M. E. P. is 76 pounds, and, therefore

$$76 \times 30 \times .44 = 1{,}003 \text{ I. H. P.}$$

When the different speeds are so treated and connected together, we have the maximum horsepower curve, or "characteristic" of the engine in question. It is seen that this characteristic follows a straight course from starting to about 12 miles an hour. This far the I. H. P. is directly proportional to the speed, as the boiler is able to supply steam as fast as it is used, and the speed is so slow that the valve motion does not materially obstruct its passage to the cylinders, and a nearly uniform M. E. P. is maintained. As the speed rises above 12 miles an hour, the capacity of the boiler is soon reached, and it becomes necessary to cut off earlier, reducing the M. E. P.

and also the draft on the boiler. We have now reached the limit of steam generation, but an increase in I. H. P. continues with increasing speed and lower cut-offs because expansion uses the steam more economically--not that there is more of it, but it goes further—gives more power for a certain quantity. At 30 miles an hour the I. H. P. reaches its maximum, and at higher speeds the valve gear will not admit steam fast enough even to maintain the horsepower, but the M. E. P. falls faster than the speed increases, which causes a droop in the characteristic. Thus it is apparent that this curve gives us complete information regarding the power that can be obtained from the locomotive.

In figuring on the maximum horsepower of locomotives, we have so far practically neglected the question of boiler pressure. The Chicago & Northwestern engine carried 190 pounds, but it is evident if this were increased or reduced the horsepower would also change. There would be little difference in the weight of steam made in the boiler, provided that it maintained its original dimensions, as with moderate variations, the total heat in a pound of steam changes but slightly. Thus, at 150 pounds, the total heat is 1193.5; at 190 pounds, 1199, and at 230 pounds only 1203.7, a difference of only about .4 per cent below or above 190-pound steam; this quantity is so small that it could not be found in a steam boiler test, however carefully conducted. This steam would do less or more work, as compared with 190 pounds, and in Fig. 92 we have laid off the M. E. P. curves corresponding to these pressures, the broken line representing that which would be obtained with 150 pounds boiler pressure, and the dotted line at 230 pounds. As the volume for the same weight of steam will be greater or less as the pressure is lower or higher, the point where it is necessary to shorten the cut-off will be changed in proportion to the relative volumes of one pound of steam. For instance, the relative volume of one pound at 171 pounds (the cut-off pressure of 190 pounds boiler pressure with lever in the corner notch) is 156, and for 135 pounds (the cut-off for 150 pounds boiler pressure), 187, or $\frac{187 - 156}{156} = 20$ per cent greater; therefore, the speed can run 20 per cent higher ($14 \times 1.2 = 16.8$), or, say, 17 miles an hour before it will be necessary to reduce the cut-off. This is seen in the broken line. For the higher pressure the decrease in volume is about 7 per cent, or 13 miles an hour. That is, with the lever in the corner notch, these speeds would take the same weight of steam in a unit of time, as its density increases with the pressure.

Again, using formula 55, we are able to construct new characteristics corresponding to the assumed boiler pressures, the broken line giving the maximum horsepowers for 150 pounds boiler pressure, and the dotted line for 230 pounds. Here we find a variation of about 6 per cent above and below the 190-pound characteristic, which indicates approximately what could be gained (or lost) with the same heating surface, by changing the pressure. By this is meant gain or loss in capacity of the boiler, not in fuel economy, which is another question, and which will be considered later.

This does not make a very radical change in the value assigned by Professor Goss, viz., .43 of the heating surface, as the values would be about .405 and .455 with the lower and higher pressures, respectively, and from the nature of the problem, it is evident that we cannot expect in any case to obtain values which may be considered absolutely invariable.

CHAPTER VII.

HAULING CAPACITY.

Our study heretofore has been of a preliminary nature, developing the various forces and resistances connected with locomotive operation, but we now come to the work for which the engine is designed and built, that of moving traffic. The hauling capacity of a locomotive can only be determined when its tractive force is known, and also the resistances which tend to prevent the movement of the train. The tractive force is created by the action of the steam against the pistons, which, through the media of rods, crossheads, etc., cause the wheels to revolve and the engine to advance.

TRACTIVE FORCE, AT SLOW SPEED.

In order to analyze this force, let

M. E. P. = mean effective pressure in pounds per square inch;

d = diameter of cylinder in inches;

s = stroke of pistons in inches;

D = diameter of driving wheels in inches;

then the work performed by one piston in a single stroke in inch pounds is

$$\frac{\text{M. E. P.}\ \pi\ d^2\ s}{4}$$

and in one revolution (if the engine be a simple, two-cylinder machine) the work will be

$$\frac{4\ \text{M. E. P.}\ \pi\ d^2\ s}{4} = \text{M. E. P.}\ \pi\ d^2\ s,$$

as there are 4 strokes to each revolution.

The distance which the engine will traverse in one revolution in inches is π D, and as it is a well-known law of physics that equal amounts of work are accomplished when the products of the force and distance are equal, we can write

$$\text{I. T. F.}\ \pi D = \text{M. E. P.}\ \pi d^2 s$$

where I. T. F. = indicated tractive force in pounds, or solving for I. T. F.,

$$\text{I. T. F.} = \frac{\text{M. E. P.}\ \pi d^2 s}{\pi D} = \frac{\text{M. E. P.}\ d^2 s}{D} \quad \ldots\ldots\ldots\ldots (97)$$

This formula gives the indicated tractive force, because it is derived directly from the mean effective pressure in the cylinder, or the pressure that would be obtained from an indicator diagram.

If we let P = boiler pressure in pounds per square inch and substitute it for M. E. P. in equation 97, we obtain what is termed the theoretical tractive force, written T. T. F., or

$$\text{T. T. F.} = \frac{P d^2 s}{D} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (98)$$

and which is often used for the purpose of comparing different engines, though its full value is never realized in practice.

If we refer to plate 12 we see that at slow speeds the mean effective pressure is about 87 per cent of the boiler pressure, when the reverse lever is in the corner. When discussing internal resistance, we found that at full stroke it was approximately 8 per cent, so that if we subtract this amount from the mean effective pressure, we obtain $.92 \times .87 = .80$, or .8 of the boiler pressure for the maximum available pressure at the circumference of the drivers. Substituting now this value in equation 97 for M. E. P. we obtain

$$\text{T. F.} = \frac{.8 P d^2 s}{D} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (99)$$

where T. F. = maximum available tractive force at the circumference of the drivers, or the point of contact with the rail, except that it does not include or care for the rolling and journal friction of the engine, which must be considered separately, and as a function of the speed.

Some authorities consider the maximum available tractive force as equal to $\frac{.85 P d^2 s}{D}$, but as this makes no allowance for the internal resistance of the engine, we prefer to use the lower value, as exhibited in equation 99, and we have found this to be a very safe figure to use when rating locomotives in service, and one which there is little difficulty in attaining. Thus, in some tests made on the Chicago & Northwestern Railway with their class R locomotive and a dynamometer car, it was found that at slow speeds the full calculated T. F. of 25,000 pounds could be easily realized.

We saw in the chapter on Resistance that the weight on the drivers should be at least four times as great as the tractive force (maximum available) in order to prevent slipping, and it must always be remembered that to obtain the full value of T. F., the adhesive weight must be sufficiently great. As this maximum value T. F. is only realized at slow speeds, the capacity of the boiler has to be considered only when the speed is increased, in general, above 8 or 10 miles an hour.

Some of the points brought out in our study of rotative force apply equally to the tractive force; that is, the wear of cylinders and tires increases the T. F. and may bring about slipping, as explained in the chapter on that subject. It is customary, however, to base calculations on the tractive force of an engine when all parts have new dimensions, as the engine grows stronger, instead of weaker, as it wears. It is also expected that the full boiler pressure will be maintained.

When we consider the tractive force of compound locomotives, the formula becomes a little more complicated. We have two sizes of cylinders, whose area to each other bears the ratio R, that is, the low pressure cylinder has R times the area of the high pressure cylinder. In two cylinder compounds it is quite important that the work done, or the M. E. P. in each cylinder, be equal. The exhaust or back pressure of the high pressure cylinder is (generally speaking) the initial pressure of the low pressure cylinder. Now, if the high pressure cylinder obtained steam at full boiler pressure P, and exhausted at a lower pressure p, which would also be the initial pressure for the low pressure cylinder, we should have, for equal total pressure (or work), letting d_h = diameter of high pressure cylinder and d_l = diameter of low pressure cylinder,

$$(P - p)\frac{\pi d_h^2}{4} = p\frac{\pi d_l^2}{4};$$ but from our definition, $R = \frac{\pi d_l^2}{4} \div \frac{\pi d_h^2}{4}$, and therefore we can write $\frac{P - p}{p} = \frac{\pi d_l^2}{4} \div \frac{\pi d_h^2}{4} = R$, and $\frac{P}{p} - 1 = R$, $\frac{P}{p} = R + 1$, or finally,

$$p = \frac{P}{R + 1} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (100)$$

that is to say, the initial pressure in low pressure cylinder should be equal to the boiler pressure, divided by the ratio of cylinder plus one. The stroke is considered the same for both cylinders, as is usually the case.

We found, however, that the maximum available pressure was about .8 P, so that equation 100 becomes $p_a = \frac{.8\,P}{R + 1}$, where p_a = the mean available pressure on low pressure piston. Substituting this value for .8 P in equation 99, we obtain T. F. $= \frac{.8\,P\,d_l^2\,s}{(R + 1)\,D}$ for two-cylinder compounds, when the work is assumed equal in both cylinders, while operating compound.

For four-cylinder compounds, as the number of cylinders is doubled, we must double the fraction, or T. F. $= \frac{1.6\,P\,d_l^2\,s}{(R + 1)\,D}$, still assuming equal work in both high and low pressure cylinders, which, we must remember, is not always the case.

Compound locomotives, however, are arranged to start as simple engines, and also can be thrown simple when ascending heavy grades. In two-cylinder compounds, the general arrangement is to allow the high pressure cylinder to exhaust directly into the atmosphere, the steam from the boiler to the low pressure cylinder being reduced in pressure by passing through a special valve. This causes the tractive force, when operated simple, to be represented by $T.\ F. = \frac{.8\ P\ d_h^2\ s}{D}$.

If four-cylinder compounds are arranged in the same way, their simple or starting tractive force would be

$$T.\ F. = \frac{1.6\ P\ d_h^2\ s}{D}.$$

This type of engine, however, is generally "simpled" by means of a "by-pass" arrangement, whereby the opposite ends of the high pressure cylinder are put into communication with each other, effecting a partial balancing of the high pressure piston, and allowing steam from the boiler, reduced in pressure by passing through the tortuous passages of the by-pass mechanism, to act upon the low pressure piston. If the balancing of the high pressure piston were complete, and full boiler pressure were obtained in the low pressure cylinder, the tractive force would be $T.\ F. = \frac{.8\ P\ d_l^2\ s}{D}$, as only the two low pressure cylinders would be performing useful work. As explained above, however, this state of affairs seldom, if ever, actually exists.

Let us now see what values are ordinarily secured in daily practice. From information obtained from the American Locomotive Company, indicator cards taken from two-cylinder compounds, gave coefficients ranging from .875 to .905 for the indicated tractive force; deducting 8 per cent for internal resistance, we obtain coefficients ranging from .805 to .83, so that we can safely write, as before suggested,

$$T.\ F. = \frac{.8\ P\ d_l^2\ s}{(R + 1)\ D} \qquad (101)$$

for two-cylinder compounds, when working compound at slow speeds. For those engines that open the high pressure exhaust to the atmosphere when simple we can write, as heretofore,

$$T.\ F. = \frac{.8\ P\ d_h^2\ s}{D} \qquad (102)$$

which gives the starting force of a two-cylinder compound, working simple.

As stated above, four-cylinder engines are not usually so carefully balanced as regards the work performed in the different cylinders, or, in other words, the total piston pressures are not identical. Thus, in some tests recently made on the Santa Fe with Baldwin and tandem four-cylinder compounds, the percentage of work done by the different cylinders when working compound was as follows:

Engine.	High pressure cylinders.	Low pressure cylinders.
Baldwin	47.1 per cent	52.9 per cent
Tandem	39.3 per cent	60.7 per cent

With the tandem type, the difference in the total piston pressures is unimportant, as far as the working of the engines is concerned, but in the Baldwin compound this produces a rocking motion in the crosshead, which is severe on the guides and piston rods. This irregularity in pressure shows that we cannot safely use the formulæ above given for four-cylinder compounds, as they assumed equal work.

The formulæ given by the Baldwin Locomotive Works and by the American Locomotive Company for the Baldwin and tandem compounds, respectively, practically agree, and may be stated as

$$T. F. = \frac{P s}{D} (.66\, d_h^2 + .25\, d_l^2) \qquad (103)$$

The engines built by the Baldwin Locomotive Works, ranging from 10 and 17 inches to 18½ and 31 inches high and low pressure cylinders, respectively, have a ratio of from 2.68 to 2.90, with an average ratio of 2.81; the tandem engines of the American Locomotive Company vary from 2.42 to 3.52. In two-cylinder compounds the ratio is much smaller—generally from 2 to 2.3.

In order to bring equation 103 to the form used in equation 101, we can write it $T. F. = \frac{P\, d_l^2\, s}{D} \left(\frac{.66}{R} + .25\right)$; now, by comparing this with equation 101, we find that the only difference is in the coefficients, $\frac{.66}{R} + .25$ and $\frac{.8}{R+1}$. If in the numerator of the latter coefficient we substitute x for .8, and for R we use the average ratio of the Baldwin compounds, 2.81, we have, by equating them, $\frac{.66}{2.81} + .25 = \frac{x}{3.81} = .485$ and $x = .485 \times 3.81 = 1.85$, and by taking the 8 per cent allowance for internal resistance from this figure, we obtain for the coefficient, $1.85 \times .92 = 1.7$, and this gives us

$$T.\ F. = \frac{1.7\ P\ dl^2\ s}{(R + 1) D} \qquad (104)$$

as the maximum available tractive force for Baldwin compounds. The coefficient 1.7 instead of 1.6, based on equal work, is brought about by the larger amount of work done by the low pressure cylinder as above explained. When we consider that the tandems may perform 60 per cent of their work in the low pressure cylinders, as shown above, it is evident that the coefficient would be still greater. The tandem tested on the Santa Fe gave a maximum available tractive force about equal to equation 103, without making any deduction for friction, as was the case with the Baldwin compound. From this it is apparent that it is well-nigh impossible to determine the tractive force when the work is not balanced in the different cylinders, unless we know about what mean effective pressures to expect, and which must be obtained by consulting indicator cards for existing engines of similar proportions. We can say, however, that it will generally be greater than that indicated by the formula for equal work. Thus, in two engines identical except the cylinders, both carrying 210 pounds steam pressure and having 57-inch drivers, the Baldwin cylinders being 17 and 28 by 32 inches and the tandem 16 and 28 by 32 inches, the Baldwin engine gave a tractive force of about 42,000 pounds, and the tandem about 43,000 pounds. In the first case the cylinder ratio was 2.7 and in the latter 3.06, and from formula 103 we should naturally expect the tandem to be the weaker, but the larger proportion of work done in the low pressure cylinder actually made it the stronger.

For the tandem engine, equation 103 gives $T.F. = \frac{210 \times 32}{57} \times (.66 \times 16^2 + .25 \times 28^2) = 43{,}000$ pounds; and for the Baldwin compound equation 104, $T.F. = \frac{1.7 \times 210 \times 28^2 \times 32}{3.7 \times 57} = 42{,}500$ pounds.

In an article in the American Engineer of October, 1902, Mr. E. L. Coster states that the Lehigh Valley Railroad uses, for Baldwin compounds, the formula

$$T.F. = \frac{P\,s}{D}\,(.71\,d_h^2 + .265\,d_l^2)$$

which, it will be noticed, gives greater values than equation 103. It is possible that a different valve setting would produce this increased force, but we consider equation 104 more conservative.

In the same article Mr. Coster gives the Baldwin formula for compounds, starting, or simple, as it is generally termed. This is stated as follows: $T.F. = \frac{P\,s}{D}\,(.56\,d_h^2 + .34\,d_l^2)$ and reduced to the same form as equation 104 becomes $T.F. = \frac{1.88\,P\,d_l^2\,s}{(R + 1)\,D}$, or about ten per cent greater than when operating compound.

The tests above mentioned, however, showed only about 4 per cent increase in the tractive force for this engine (Baldwin compound), whereas the tandem gave about 18 per cent increase. The proportion of work done by the different cylinders, with the starting valve open, was as follows:

Engine.	High pressure cylinders.	Low pressure cylinders.
Baldwin	36.7 per cent	63.3 per cent
Tandem	21.4 per cent	78.6 per cent

From the above it is seen that in the tandem engine the high pressure piston is more nearly balanced by the steam on both sides, this being due to the fact that the equalizing passage is more direct than in the Baldwin, but while this is no disadvantage to the tandem, it would produce a serious crosshead disturbance if permitted in the Baldwin engine, and the larger proportion of work done in the high pressure cylinder is a positive advantage to that type of locomotive.

This greater tractive force at starting makes necessary a greater adhesive weight for a compound than for a simple engine of the same hauling capacity, in order to avoid slipping when operated simple.

We now present these several formulæ in a table for convenient use:

FORMULÆ FOR MAXIMUM AVAILABLE TRACTIVE FORCE = T. F.

Type	Work in the different cylinders	Operated simple,	Operated compound.
Single expansion..............	....	$\frac{.8\ P\ d^2\ s}{D}$	....
Two-cylinder compound.....	equal	$\frac{.8\ P\ d_h^2\ s}{D}$	$\frac{.8\ P\ d_l^2\ s}{(R+1)\ D}$
Tamdem compound......	unequal	$\frac{1.15\ P\ s}{D}\ (.66\ d_h^2 + .25\ d_l^2)$	$\frac{P\ s}{D}\ (.66\ d_h^2 + .25\ d_l^2)$
Baldwin compound..........	unequal	$\frac{1.8\ P\ d_l^2\ s}{(R+1)\ D}$	$\frac{1.7\ P\ d_l^2\ s}{(R+1)\ D}$
Four-cylinder compound,....	equal	$\frac{1.6\ P\ d_h^2\ s}{D}$	$\frac{1.6\ P\ d_l^2\ s}{(R+1)\ D}$

It is often desirable to compare simple and compound engines, or to state briefly that such a compound engine is equivalent in tractive force to a certain simple engine. If we consider, in making these comparisons, that the boiler pressure, diameter of drivers and stroke of pistons are the same, we can derive the relation of cylinder diameters in the following manner. In the formulæ given above, we observe that $\frac{P\ s}{D}$ is common to all, so that to find the diameter of cylinders in a simple engine which shall be equivalent to any stated diameters of cylinders in a compound engine, it is only necessary to equate the balance or remainder of the formulæ.

For two-cylinder compounds, operating compound,

$$.8\,d^2 = \frac{.8\,d_l^2}{R+1} \text{ and } d^2\,(R+1) = d_l^2 = d^2 + \frac{d^2\,d_l^2}{d_h^2};$$

$$d_l^2 - d^2 = \frac{d^2\,d_l^2}{d_h^2}, \text{ and } d_l^2\,d_h^2 - d^2\,d_h^2 = d^2\,d_l^2; \text{ also}$$

$$d_l^2\,d_h^2 = d^2\,d_l^2 + d^2\,d_h^2 = d^2\,(d_l^2 + d_h^2) \text{ and } d^2 = \frac{d_l^2\,d_h^2}{d_l^2 + d_h^2}$$

$$\text{or } d = \sqrt{\frac{d_l^2\,d_h^2}{d_l^2 + d_h^2}} \quad \ldots\ldots\ldots\ldots (105)$$

For four-cylinder compounds, performing equal work in all cylinders, we have $.8\,d^2 = \frac{1.6\,d_l^2}{R+1}$, and expanding and reducing as above, we obtain,

$$d = 1.41\sqrt{\frac{d_l^2\,d_h^2}{d_l^2 + d_h^2}} \quad \ldots\ldots\ldots\ldots (106)$$

For Baldwin compounds, $.8\,d^2 = \frac{1.7\,d_l^2}{R+1}$, and likewise

$$d = 1.45\sqrt{\frac{d_l^2\,d_h^2}{d_l^2 + d_h^2}} \quad \ldots\ldots\ldots\ldots (107)$$

For tandem compounds, $.8\,d^2 = .66\,d_h^2 + .25\,d_l^2$ or $d^2 = .82\,d_h^2 + .31\,d_l^2$ and $d = \sqrt{.82\,d_h^2 + .31\,d_l^2}$ .... (108)

In order to obviate the necessity for making these calculations, plate 28 is introduced, which shows at a glance the equivalent simple and compound cylinders, thus permitting a ready comparison, not only of compounds and simple engines, but of different kinds of compounds, one with another. The vertical lines represent diameters of simple cylinders, the horizontal lines diameters of low pressure cylinders, and the angular lines diameters of high pressure cylinders. For instance, a Baldwin four-cylinder compound having cylinders 17 and 28 inches in diameter, will pull the same load at slow speeds, in full gear, as a simple engine whose cylinders are 21¼ inches, the boiler pressure, stroke, drivers, weight, etc., being equal.

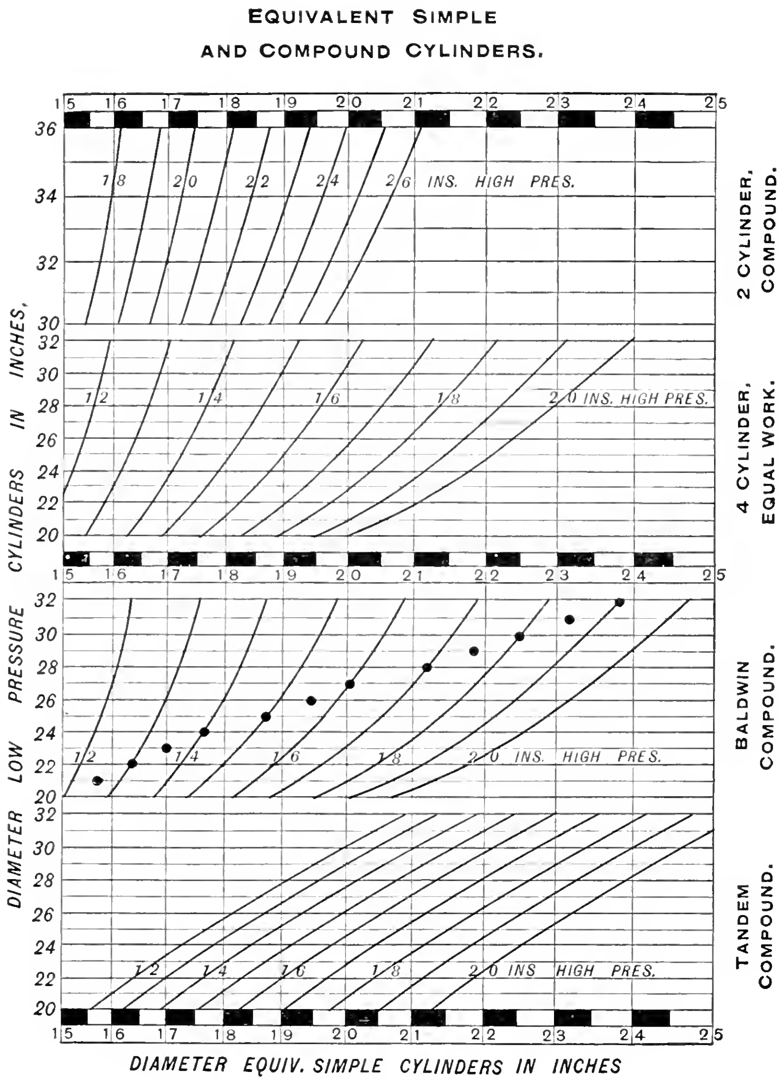

EQUIVALENT SIMPLE
AND COMPOUND CYLINDERS.
2 CYLINDER, COMPOUND.
18 20 22 24 26 INS. HIGH PRES.
4 CYLINDER, EQUAL WORK.
12 14 16 18 20 INS. HIGH PRES.
BALDWIN COMPOUND.
12 14 16 18 20 INS. HIGH PRES.
TANDEM COMPOUND.
12 14 16 18 20 INS HIGH PRES.
DIAMETER LOW PRESSURE CYLINDERS, IN INCHES
DIAMETER EQUIV. SIMPLE CYLINDERS IN INCHES

Plate 28

The dots indicate the combinations of cylinders usually employed by the Baldwin Locomotive Works.

TRACTIVE FORCE, AT HIGH SPEED.

We have just discussed tractive force at slow speed, where there was no question as to the capability of the boiler to supply the necessary quantity of steam to fill the cylinders at every stroke, but as the speed of the engine increases, the cylinders demand more steam in a unit of time, and it is not long before we reach the limit; that is, the gauge commences to fall, and we must shorten the cut-off in order that the cylinders shall not draw off more steam in a unit of time than can be generated by the boiler.

In the last chapter the capacity of the boiler was thoroughly discussed, and by the aid of Fig. 91 the approximate capacity of a locomotive boiler can be determined. The method of finding the maximum speed at which the boiler will supply steam at full stroke, is best explained by an example. Let us take the Chicago & Northwestern class R 10-wheel locomotive, with 20 by 26 inch cylinders, 190 pounds boiler pressure, 2,332 square feet of heating surface and 29 square feet of grate area, and burning Illinois coal. The volume of one cylinder is 4.73 cubic feet, and as locomotives seldom cut-off later than about 90 per cent of the stroke, we can assume that the port clearance is about equal to the uncompleted stroke at cut-off.

In our study of steam distribution we found that the initial pressure is about 94 per cent of the boiler pressure when the revolutions per minute are between 50 and 100, and from plate 10 know that the cut-off pressure will be about 96 per cent of the initial pressure, so that we have $.94 \times .96 = .90$ for the ratio of cut-off to boiler pressure, or $190 \times .90 = 171$ pounds at cut-off, which steam will weigh .411 pound per cubic foot. The ratio of heating surface and grate area is

$\frac{2,332}{29} = 80$ and from Fig. 91 we expect that the boiler will

produce 13½ pounds of steam from and at 212 degrees per square foot of heating surface per hour, with Illinois coal. The engine actually did a little better, producing about 14¼ pounds, or say 12 pounds of steam at boiler pressure, so that the total weight of steam generated per hour would be $2,332 \times 12 = 27,984$ pounds. Now the weight at cut-off was found to be .411 pound per cubic foot, so that the supply amounts to $\frac{27,984}{.411} = 68,090$ cubic feet an hour, or $\frac{68,090}{60} = 1,135$ cubic feet per minute. As there are four strokes in one revolution, and as each cylinder has a volume of 4.73 cubic feet, the draft on the boiler for each revolution will be $4.73 \times 4 = 18.92$ cubic feet, so that $\frac{1,135}{18.92} = 60$ revolutions per minute, which is the greatest speed at which the boiler will supply the cylinders when running with the reverse lever in the corner notch.

This process could be repeated for various degrees of expansion, in order to determine the maximum speed at which the boiler would supply that particular cut-off, but this would be laborious, and besides we have satisfactory data taken from various experiments and tests to give us this information empirically. As there is ample steam at speeds below the limiting one, the tractive force may be considered uniform, and at its maximum value for all such speeds. Plate 29 illustrates the manner in which the maximum available tractive force is dependent upon the capacity of the boiler. It is seen to be uniform for a short distance; in the plate, this corresponds to 60 revolutions per minute, which is the limit which we just found for the Chicago & Northwestern locomotive. From this point, or speed, the locus falls rapidly, indicating that the lever must be brought back, giving earlier cut-offs and lower tractive values. The plate gives ratios of T. F. or available tractive force to T. T. F. or theoretical tractive force, and thus the values taken from the curve may be used directly as coefficients of formula 98, $\frac{P\,d^2\,s}{D}$. Thus, from 0 to 60 revolutions per minute, the line A B indicates 80 per cent, so that equation 98

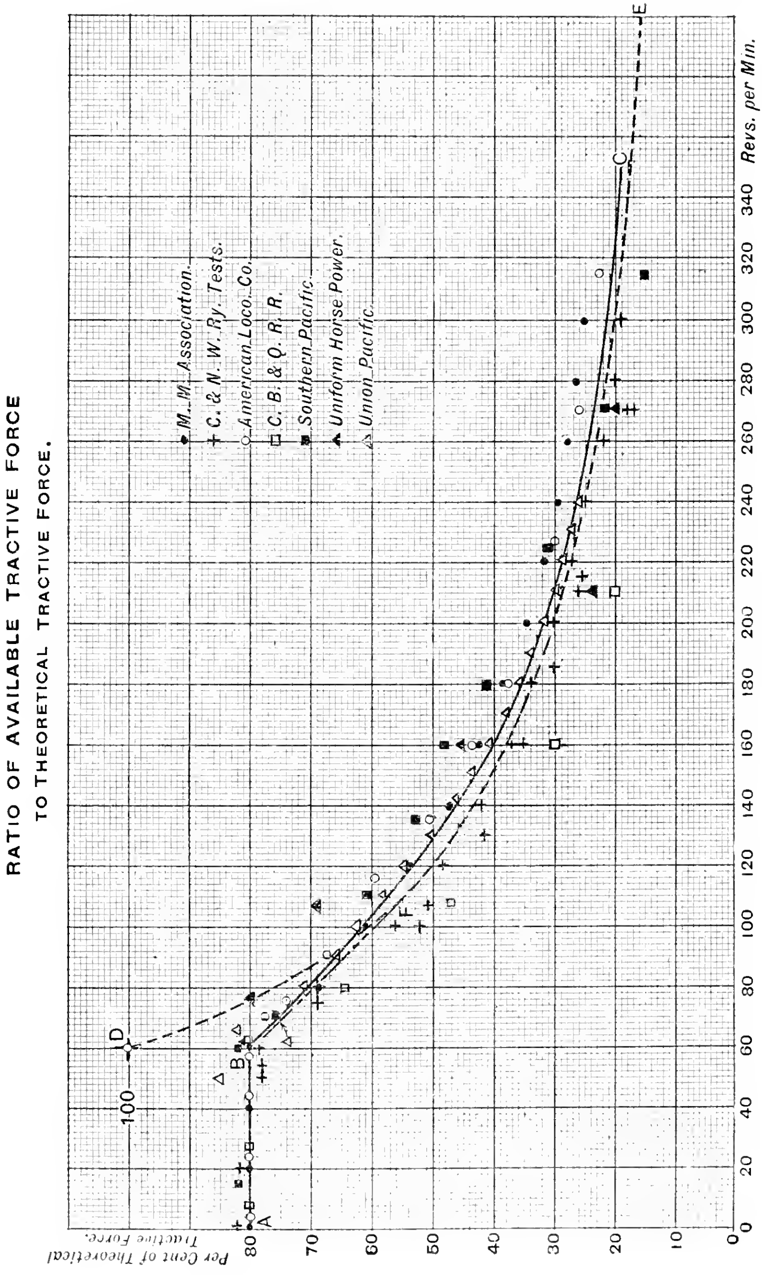

RATIO OF AVAILABLE TRACTIVE FORCE
TO THEORETICAL TRACTIVE FORCE.
M. M. Association.
C. & N. W. Ry. Tests.
American Loco. Co.
C. B. & Q. R. R.
Southern Pacific.
Uniform Horse Power.
Union Pacific.
Per Cent of Theoretical Tractive Force.
Revs. per Min.
0
20
40
60
80
100
120
140
160
180
200
220
240
260
280
300
320
340
10
30
50
70
A
B
C
D
E
F

Plate 29.

becomes, for these speeds, $\frac{.8 \text{ P d}^2 \text{ s}}{\text{D}}$; the same as equation 99. For higher speeds we must follow the line B C; thus, at 160 revolutions, the ordinate is 40, and the available tractive force at this speed will be $\frac{.4 \text{ P d}^2 \text{ s}}{\text{D}}$. (We are here considering single expansion engines only.)

The line A B C in plate 29 has been prepared with much care and study, and we believe that it fairly represents the ordinary simple locomotive practice of this period. The dots denote the curve which would be produced by following the one presented to the Master Mechanics' Association in 1898, and shown already on our plate 12: the crosses are from tests made by the Chicago & Northwestern Railway: the open circles were taken from a pamphlet published by the American Locomotive Company; the open squares from tests made on the Burlington Road; the solid squares show the values adopted by the Southern Pacific for purposes of rating trains; the open triangles give values used by Chief Engineer Berry, of the Union Pacific, in his grade reduction work; and the solid triangles from a curve of uniform horsepower (indicated) with internal resistances deducted. We therefore feel that the line A B C has what might be termed a "good pedigree."

We saw above that the point B, where the line begins to drop, depends upon the capacity of the boiler to supply the cylinders at full stroke, and that when this limit has been reached, the curve at once falls. If the boiler were smaller in proportion to the size of the cylinders, the drop would commence earlier; if larger, it would occur later. Thus, if it were half the size, only 30 revolutions could be made per minute before shortening the cut-off. While plate 29 represents the average, well-designed, modern simple locomotive, cases may occur where it is desirable to quickly construct a curve for special conditions. Let us see how this can be done. In the plate a broken line D E will be noticed. This starts from the point D, which is located on the ordinate corresponding to the speed at which the boiler ceases to supply steam at full stroke, and at a height which is equal to 100 per cent, or the full theoretical ratio. The line D E is a rectangular hyperbola starting from this point D, and as the co-ordinates of the point D are 60 and 100, the equation of the hyperbola is $x\,y = 6{,}000$, the product of 60 by 100. At any point in the curve D E, the product of the co-ordinates of the point will equal 6,000. Thus, at 100 revolutions, the percentage is 60, and 100 by 60 gives 6,000. So, for 200 and 300 revolutions the ordinates are 30 and 20 respectively, and we have $200 \times 30 = 6{,}000$ and $300 \times 20 = 6{,}000$. It will be observed that the two lines B C and D E lie quite close together—the hyperbola being slightly lower "or safer" from F to E. To the left of F its values exceed those of B C, but by drawing a straight line from the point B tangent to D E, we find that the combination tangent and curve B F E is a very close approximation to the line B C.

The general rule can, therefore, be given as follows: Locate the point D as already explained; from D construct an equilateral hyperbola such that the product of the co-ordinates of any point in the curve equals the product of the co-ordinates of the point D. Draw a straight line on the 80 per cent horizontal from the vertical axis to a point directly below the point D, and connect this point (on the 80 per cent line) with the hyperbola by means of a tangent to the hyperbola, passing through the said point. The locus consisting of the horizontal portion (on the 80 per cent line), the tangent, and the part of the hyperbola beyond the tangent will then give approximately the ratio of the available tractive force at different speeds of rotation to the theoretical tractive force, or the proper coefficient to use with equation 98.

It may be thought at first sight that the hyperbola should be constructed from the point B, but upon reflection it will be remembered that as the rate of expansion increases, we get more work out of the steam than we do at full gear, consequently more work out of the engine, and more tractive force at the circumference of the drivers than we would expect, considering the speed, or, in other words, the product of speed

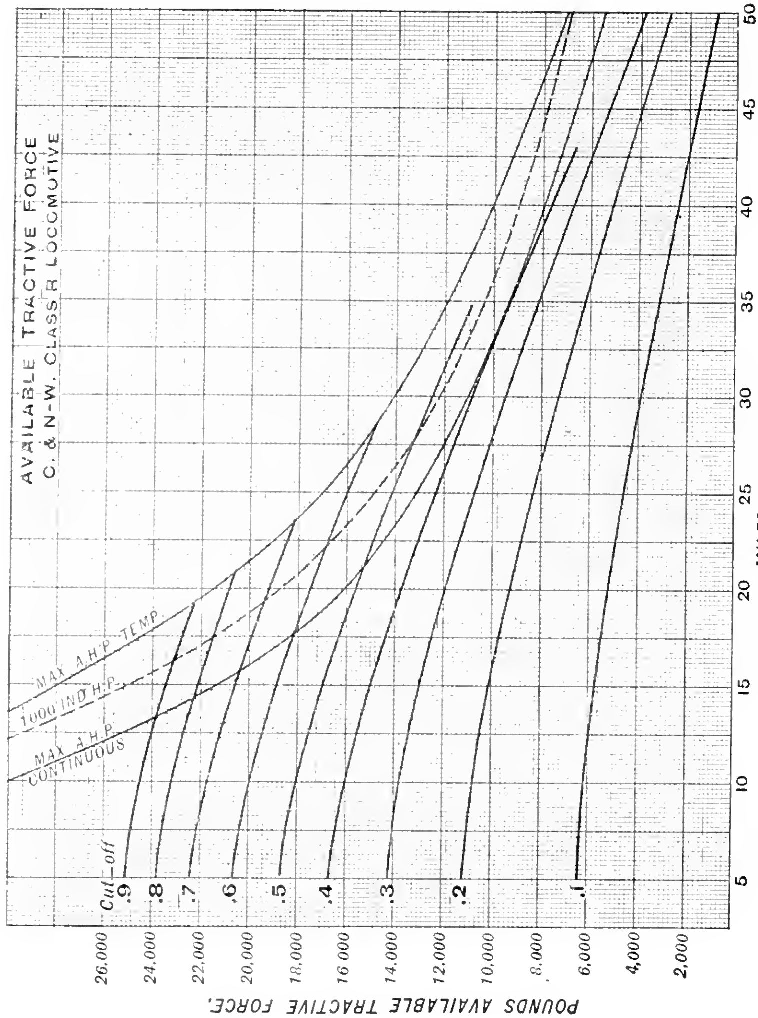

AVAILABLE TRACTIVE FORCE
C. & N-W. CLASS R LOCOMOTIVE
MAX. A.H.P. TEMP.
1000 IND. H.P.
MAX. A.H.P. CONTINUOUS
Cut-off
.9
.8
.7
.6
.5
.4
.3
.2
.1
26,000
24,000
22,000
20,000
18,000
16,000
14,000
12,000
10,000
8,000
6,000
4,000
2,000
POUNDS AVAILABLE TRACTIVE FORCE.
5
10
15
20
25
30
35
40
45
50
MILES PER HOUR

Plate 30.

and tractive force is greater at high than at low speeds, on account of the earlier cut-off.

Plate 30 shows the available tractive force at the circumference of the drivers, for the Chicago & Northwestern class R locomotive before described, and the curves were derived from tests made with a dynamometer car, by allowing for the resistance of the engine and tender on the grade and at the speed when the record was taken. These resistances were added to the recorded drawbar pull and the sum was considered to be the available force at the point of contact with the rail. The line marked "Max. Avail. H. P. cont." shows the limit of horsepower at circumference of drivers which can be continuously delivered, and the "Max. A. H. P. temp." is what could be developed for a short time only by closing the injectors and taking advantage of the supply of heated water in the boiler, and as the amount of heat used in raising water from 60 degrees to the temperature of 190-pound steam (384°) is about one-fourth of the total heat of evaporation $(1{,}236° - 60° = 1{,}176°) = \frac{384 - 60}{1{,}176} = .27$, this line is 25 per cent in excess of the continuous horsepower line. The curve for 1,000 horsepower is also given, as this is the power which we would expect from the 2,332 square feet of heating surface in the boiler, no deductions being made in this curve for internal engine resistance. The droop of the different lines representing the various proportions of cut-off at increasing speeds is due to inefficiency of the valve motion at high speeds. They may be taken as fairly representative, however, of a modern simple engine with the Stephenson link motion.

Plate 31 (at back of book) affords a quick method of obtaining the tractive force available at the rails for any engine at any speed, where approximate results are sufficiently close; where accurate figures are desired, they must be calculated as explained heretofore. The lines used for compound locomotives presuppose equal work in all cylinders, which, as we have seen, is not always the case. The operation of the plate can best be explained by an example. Consider a simple locomotive with

21 by 30 inch cylinders, 50-inch drivers and 190 pounds boiler pressure. Start at the intersection of the 30-inch stroke line with the 21-inch cylinder diameter line (marked by a dot) and move upwards parallel to the vertical lines until reaching the 50-inch driver diameter line; then continue parallel to the sloping lines to the dividing line between driver diameter and boiler pressure, and upwards, but to the right, parallel to the next set of angular lines till the 190-pound boiler pressure line is reached; now proceed upwards, again parallel to the vertical lines to the top of sheet, where we read off 40,300 pounds as the T. F. at slow speeds. If we wish the T. F. at 30 miles an hour, notice that the intersection of the 50-inch radial line in the upper right-hand corner and the 30-mile vertical line occurs at the line corresponding to 200 revolutions per minute, and by following the curved lines to the left, as shown by the dotted line, until this 200-revolution line is encountered, we find that the A. T. F. is 15,700 pounds. Conversely, if we wish to pick out the leading dimensions of an engine to produce a certain T. F. we proceed in the reverse order. If the engine has compound cylinders, we select our starting point at the left: for instance, for a two-cylinder compound, with diameters 26 and 33 inches, and 32 inches of stroke, we should follow the intersection of 26 and 33 horizontally till intersecting the 32-inch stroke line; then upwards as before. For a four-cylinder compound, 18 and 30 by 28 inch stroke, proceed as shown by the dotted line also.

For the tractive force of compounds at high speeds, plate 29 may be used for approximate results, by assuming an equivalent simple engine, as per plate 28, but the most accurate manner will be to construct the hyperbola suited to the existing conditions in the same manner as the maximum speed at which the boiler will supply steam at full stroke was determined. In the simple engine discussed, we found that a speed of 60 revolutions per minute was the limit. In compound engines, the high pressure cylinder alone is supplied by the boiler, and as it is comparatively small if of the four-cylinder type (or consists of only one cylinder, if of the two-cylinder type) the limiting speed for full stroke is much greater. As an example, the 2—6—2 type Baldwin compounds of the Santa Fe have cylinders 17 and 28 by 28 inches, 79-inch drivers, 200 pounds boiler pressure, heating surface 3,738 square feet and grate area 53.5 square feet. Thus the ratio was $\frac{3{,}738}{53.5} = 70$ and the evaporation 14.5 pounds of water from and at 212 degrees, or 12 pounds at working pressure per square foot of heating surface per hour, and for the total, $3{,}738 \times 12 = 44{,}856$ pounds. As at cut-off, the steam at $200 \times .9 = 180$ pounds pressure would weigh .432 pound per cubic foot, the volume supplied per minute would be $\frac{44{,}856}{.432 \times 60} = 1{,}730$ cubic feet, and as each high pressure cylinder has a volume of 3.7 cubic feet, the number of revolutions at which the boiler will supply steam for full stroke will be $= \frac{1{,}730}{4 \times 3.7} = 117$ per minute, or 28 miles an hour. At 70 miles an hour, or 300 revolutions per minute, the cut-off pressure would be about 132 pounds ($200 \times .86 \times .77$) and would weigh .33 pound per cubic foot. The volume supplied by the boiler would therefore be, $\frac{44{,}856}{.33 \times 60} = 2{,}300$ cubic feet a minute, and the total high pressure cylinder volume at 300 revolutions, $3.7 \times 4 \times 300 = 4{,}440$ cubic feet; therefore, $\frac{2{,}300}{4{,}440} = 52$ per cent of stroke for point of cut-off. As compression would undoubtedly fill the clearance, it need not be considered, but a correction of about 15 per cent should be made for condensation in the cylinder, as per plate 14, this being a compound engine, so that the actual cut-off would be $.52 \times .85 = 44$ per cent. Now, proceeding to find the M. E. P. by using plate 11, figuring upon an initial pressure of $200 \times .86 = 172$ pounds, a cut-off pressure of $172 \times .77 = 132$ pounds at 44 per cent of stroke, and a back pressure of $\frac{172}{2.71 + 1} = 46$ pounds (2.71 being the cylinder ratio), with compression to initial pressure, we determine the area with a planimeter (or by counting the included squares), and so compute the mean effective pressure at 60 pounds per square inch. If we consider equal work in both cylinders, the tractive force will be =

$$\frac{2 \times .92 \times 60 \times 289 \times 28}{79} = 11{,}350 \text{ pounds,}$$

in which we have allowed 8 per cent for internal friction, as usual. The theoretical tractive force of such an engine would be $\frac{2 \times 200 \times 784 \times 28}{(2.71 + 1) \times 79} = 30{,}000$ pounds, and therefore the tractive force available at 70 miles an hour or 300 revolutions per minute will be $\frac{11{,}350}{30{,}000} = 38$ per cent of the theoretical tractive force.

If we were to construct a hyperbola, as explained for simple engines, we find that the starting point would be at 117 revolutions and 100 per cent for the ordinate, thus $xy = 11{,}700$, x being the speed and y the per cent of T. T. F. At 300 revolutions, $y = \frac{11{,}700}{300} = 39$ per cent, or 1 per cent more than our value of 38 per cent found above by working out a hypothetical diagram. As there are so many variables in solving these questions for compound locomotives, no regular diagram is presented, as each case should be worked out as just explained above.

TRACTIVE FORCE AT VARIABLE SPEED.

Two kinds of variable tractive force may be considered—one due to the variation in cylinder effort throughout the period of a revolution, and the other due to the change of speed when accelerating or retarding the train. The first is so minute as to be inappreciable at fairly high speeds. It has been found that the records of even extremely sensitive dynamometer cars fail to indicate a variation in the pull of the drawbar, although the rotative force during a revolution of the drivers may reach a value 50 per cent greater than the minimum, or perhaps still more. In order to explain this, let us refer to plate 19, which gave the rotative forces of an engine weighing 176,000 pounds, or 88 tons, and select the 40-mile-an-hour curve. The minimum, average and maximum values are 30,000, 38,000 and 46,000 pounds at crank radius, respectively, and as the stroke was 26 inches and the diameter of drivers 79 inches, the tangential force at the rail will be $\frac{26}{79}$ times these amounts, or 9,900, 12,500 and 15,100 pounds, respectively, and if we further deduct 8 per cent for internal resistance, we obtain 9,100, 11,500 and 13,900 pounds for the minimum, average and maximum forces at the rail during one revolution. As the average force must be in equilibrium with the resistance overcome, any force of the drivers less than the average will act as a retarding force by the amount which it falls below the average, or the resistance will be that much greater, and for a force greater than the average it will act as an accelerating force; therefore during the period that the force line falls below the average line in plate 19, the speed must decrease, and when it is above the average line, the speed will increase. If we assume that at 0 degrees on the diagram the speed is just 170 revolutions per minute, or 40 miles an hour, then at 15 degrees rotation, when the force reaches the average, the speed will have been reduced, because the average force (tangential) during that time has been only about $\frac{11,500 + 9,100}{2} = 10,300$ pounds, or $11,500 - 10,300 = 1,200$ pounds less than the resistance, and this has acted as a retarding force, at the rate of $\frac{1,200}{88} = 14$ pounds per ton weight of engine, and the distance through which this force acted during 15 degrees rotation is $= \frac{15 \times 79 \times \pi}{360 \times 12} =$ .86 feet.

Equation 3 can be transposed to the form

$$V_1^2 = V_2^2 - \frac{P_t S}{70}$$

in which V_2 = the velocity at 0 degrees = 40 miles an hour
P_t = 14 pounds per ton, and
S = .86 feet distance through which the force P_t acts; then substituting these values, we have $V_1^2 = 1,600 - \frac{14 \times .86}{70} =$ $1,600 - 0.17 = 1,599.83$ and $V_1 = \sqrt{1,599.83} = 39.997$ miles per hour, or in feet per second, 58.40 at 0 degrees and 58.395 at 15 degrees, or in 15 degrees of rotation or .86 feet of translation, the velocity has diminished .005 feet per second—an amount too small to be appreciated in practice, actually about 1-16 of an inch, out of a total velocity of over 58 feet. From 15 to 30 degrees the force is greater than the average, so that by the time the wheel has rotated 30 to 40 degrees, it has regained its normal speed.

The second variation in tractive force is of much importance, however, and is due to the fact that as the speed increases, the tractive force diminishes, as illustrated by plates 29 and 30. The converse of this is also true, that as the speed decreases, the tractive force can be increased. This has its special application upon momentum grades, at the foot of which the engine is running at a high speed, and as the increased resistance of the grade reduces the speed, the engineer can gradually drop his lever, thus increasing the power of the engine.

Fig. 93 explains this more fully, being a reproduction of a portion of the dynamometer car and Boyer speed records for two important momentum grades on the Chicago & Northwestern Railway; also a profile of the parts of the road upon which the records were taken. The profiles are marked with the grades in feet per mile, the lengths being determined by the mile post designations at the foot of the diagram. The upper curves show the speed in miles per hour throughout the ascent, and it is seen that from 25 and 35 miles an hour at the foot of the hill the speeds drop to a very slow rate at the summit, showing that momentum has been used to the limit in assisting the engine over the hill. The lower curve is the available tractive force, and as the speed diminished the reverse lever was gradually dropped, lengthening the period of admission, and thus increasing the tractive force. The maximum T. F. for this engine was 25,000 pounds, and it will be noticed that this was reached shortly before attaining the top of the grade. The values of T. F. shown have been computed by adding to the

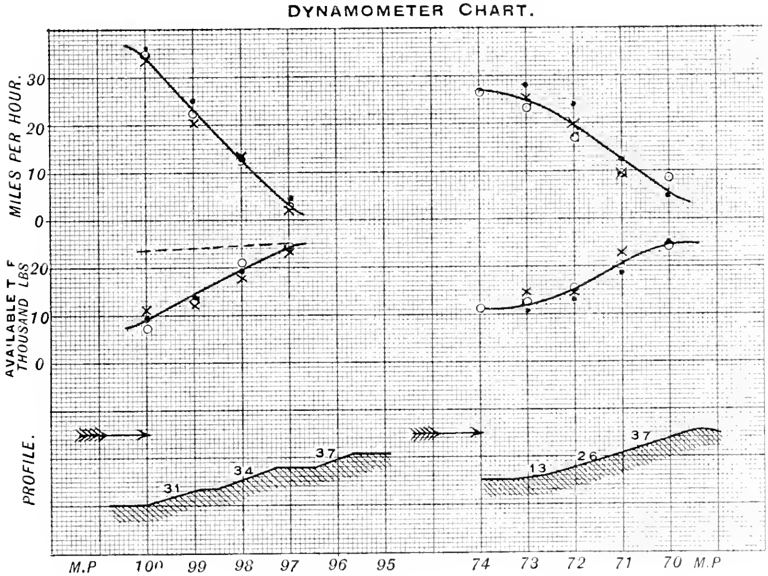


Fig. 93.

actual dynamometer record the resistance of the engine and tender due to the speed and the virtual grade of ascent. The dots, circles and crosses represent different runs with various engines of the same class, and the loci drawn locate the average for the three runs or tests. The broken line shows the total forces propelling the train, having been constructed by laying off above the locus of tractive force, the value of inertia due to the retardation shown by the speed curve; this force is seen to be nearly uniform, as would be expected.

The tractive force thus varies regularly (if the engine be properly manipulated) throughout the ascent, and as the effect of inertia is reduced, the power of the engine is increased, producing approximately an even balance between the forces urging the train onward and upward, and the resistances operating in the opposite direction, and it is this that enables the engine to take up a short grade a train considerably heavier than its absolute rating, provided that it can make a run for the hill.

When the speed of the engine varies between small limits only, the average tractive force for the period during the change can be taken from plate 29. Thus, a drop from 260 to 240 revolutions per minute may be accompanied by an average force of 25 per cent of the theoretical tractive force, while 24 and 26 per cent are the rates at the two limits chosen. When, however, the range is large, as from 260 to 60 revolutions, the average cannot be determined by inspection, as the line B C is a curve. In such cases a table, like that annexed, will facilitate the computations, as the average ratio between various limits can be obtained at a glance. Even this will not be correct unless the speed be varied uniformly and regularly.

AVERAGE RATIO OF AVAILABLE TRACTIVE FORCE TO THEORETICAL TRACTIVE FORCE FOR VARIABLE SPEEDS, IN PER CENTS

Revolutions Per Minute.	Between Revolutions Per Minute Given at Top and Side.														
	300	280	260	240	220	200	180	160	140	120	100	80	60	40	20
0	48.5	50.5	52.6	55.0	57.3	60.2	63.1	66.3	70.0	73.0	76.2	80.0	80.0	80.0	80.0
20	46.6	48.2	50.3	52.6	55.1	57.9	61.1	64.3	68.0	71.6	75.2	78.3	80.0	80.0	80.0
40	43.8	45.6	47.6	49.9	52.4	55.2	58.3	61.7	65.5	69.5	73.7	77.5	80.0	80.0	....
60	40.7	42.4	43.4	46.5	49.0	51.7	55.7	58.1	61.9	66.0	70.5	75.0	80.0	....	....
80	37.6	39.2	40.9	43.0	45.3	47.8	50.7	53.8	57.5	61.5	66.0	70.0	....	....	....
100	34.8	36.2	37.8	39.7	42.4	44.2	46.8	49.8	53.3	57.0	61.0	....	....	....	....
120	32.3	33.6	35.1	36.8	38.8	40.9	43.5	46.2	49.6	53.0	....	....	....	....	....
140	30.1	31.4	32.7	34.2	36.1	38.1	40.4	42.8	45.8	....	....	....	....	....	....
160	28.3	29.4	30.7	32.1	33.8	35.7	38.0	39.7	....	....	....	....	....	....	....
180	26.7	27.7	28.8	30.1	31.7	33.4	35.2	....	....	....	....	....	....	....	....
200	25.4	26.3	27.3	28.5	30.0	31.1	....	....	....	....	....	....	....	....	....
220	24.2	25.1	26.0	27.0	28.4	....	....	....	....	....	....	....	....	....	....
240	23.3	24.1	25.0	26.2	....	....	....	....	....	....	....	....	....	....	....
260	22.5	23.3	24.2	....	....	....	....	....	....	....	....	....	....	....	....
280	21.7	22.5	....	....	....	....	....	....	....	....	....	....	....	....	....

LOCOMOTIVE RATING.

In any machine, it is important to obtain from such device the greatest amount of work that can be produced, within consistent limits of wear and tear, if an economical operation is desired, as it always should be. This applies, not only to the locomotive, but also to the railroad as a whole, which really constitutes a machine of great complexity. To this end, the subject of tonnage rating in a scientific and modern manner has been given a great deal of investigation by prominent authorities; and not more than it deserves, as it is largely the keynote of successful railway operation. While locomotives should be loaded as heavily as they can be and still make the desired running time, it is a very serious error to overload them so that engine failures and delay to traffic are thereby caused.

The proper rating of a locomotive is simply the question of stating an equation between the power of the engine and the resistance of the train which it is to draw; this is a plain statement, and sounds like one of easy solution, but the various factors that go to make up each side of the equation are complex and require a thorough knowledge of the action of the locomotive and its train, both of which in great measure depend upon the physical condition and construction of the road and the schedule which is to be followed. A few years back, when the principles referred to were not as well understood as now, locomotives were rated by trial, a practical determination being made to see just what could be taken up a controlling grade; later, but one engine was tested on each division, and the other engines rated by their proportionate adhesive weight or cylinder power, the latter being preferable. Even at the present time, it is always desirable to check up the computed results by actual tests, and if a dynamometer car can be obtained for this purpose, it is doubly satisfactory, as if the engine be not able to do as expected, the dynamometer record at once locates the difficulty.

In the first place, it is necessary to collect all the data on the subject obtainable, and to know that it is correct. The writer calls to mind a case where an engine would not haul the load which he had specified for it; investigation finally showed that the grade was considerably heavier than stated by the chief engineer; evidently the fault was not with the engine. If the problem be correctly stated, there is little difficulty in rating locomotives in the office. Mr. Tweedy, when chief engineer of the Wisconsin Central Lines, wrote the author as follows: "I am convinced that if someone would take sufficient time and pay enough attention to the matter, it would not be very hard to get up a table that would be so accurate that every part of a road could be rated theoretically in the office from the track profile, and in such a manner that the results would be practically satisfactory."

In the foregoing part of this chapter, and in that on Resistance, we have given all the information of a fundamental character that is needed for the solution of the problem at hand, but it must be arranged for use in a proper manner. Let us begin with the simplest case—that of a slow freight train on a long controlling grade.

RATING OF SLOW FREIGHTS.

By the designation "slow freight" we mean a train that is not expected to make over 5 or 10 miles an hour up the controlling grade. If this grade be over 2 miles in length, momentum will be of little use, as from plate 2 we see that at ordinary freight speeds the effect of inertia is almost completely exhausted in a run of 10,000 feet. Formula 99 gives us the maximum available tractive force at the circumference of the drivers, but the weight of the engine and tender must be taken care of before we obtain the net pull back of tender. Plate 23 indicates that five pounds per ton will cover the rolling and journal resistance at slow speeds, formulæ 84 and 85 give the allowance for grade, and under "Curve Resistance" we find what is necessary to cover curvature. By adding together the coefficients for speed, grade and curvature, and multiplying this sum by the weight of the engine and tender in tons, we obtain a product which is to be subtracted from the tractive force (equation 99) in order to determine the net pull back of tender. This value constitutes one side of the equation —the other is to be obtained by using formula 86 with the modifications of the coefficient of T, as fully explained. The following example will illustrate this method:

A 4—6—0 type locomotive, weighing 132 tons with tender, has cylinders 20 by 26 inches, 63-inch drivers and 190 pounds boiler pressure. Formula 99, $T. F. = \frac{.8 P d^2 s}{D}$, gives us for the available (maximum) tractive force of this engine $\frac{.8 \times 190 \times 400 \times 26}{63} = 25{,}000$ pounds. Suppose that it be desired to rate this locomotive for a long .7 per cent grade in slow freight, then for the resistance of engine and tender we have, per ton:

By equation 85, resistance for grade = $20 \times .7$ . . = 14 pounds
By plate 23, resistance for slow speeds. . . . . = 5 pounds

Total resistance per ton . . . . = 19 pounds

and the weight of engine multiplied by this resistance per ton $= 132 \times 19 = 2{,}508$ pounds. Deducting this from the tractive force, we have $25{,}000 - 2{,}508 = 22{,}492$ pounds pull back of tender. This must be equal to or greater than the resistance of the train in order to pull it up the grade.

Equation 86 enables us to determine the train resistance back of tender. We will figure this for empty and loaded cars of, say 50 tons gross weight each, that is, car and lading. If we consider that the empty cars weigh 16 2-3 tons each, which will not be far from the actual average light weights, we find by the formula that the resistance on a .7 per cent grade for 100 tons will be $17.5 \times 100 + 50 \times 6 = 1{,}750 + 300 = 2{,}050$ pounds, and for the 50-ton cars, $17.5 \times 100 + 50 \times 2 = 1{,}750 + 100 = 1{,}850$ pounds. The empties will, of course, run six cars to the 100 tons, and the loads two cars; the coefficient 17.5 is made up of 14 for the grade and 3.5 for speed resistance, as explained under Train Resistance. Then for the total train, we have:

For empty cars, $\frac{22{,}492}{2{,}050} = 11$ approximately, or, say, 1,100 tons, and at six cars to 100 tons, there will be 66 cars in the train.

For 50-ton loads, $\frac{22{,}492}{1{,}850} = 12.16$, say, 1,220 tons, and as there are two cars to 100 tons, the train will consist of 24 cars. Both of these trains will give the same resistance ascending the

.7 per cent grade. We can also take intermediate cases of loading: that is, for cars which will weigh more than 16 2-3 and less than 50 tons gross each by interpolating the number of cars and tons, as shown below, and also demonstrate that the total train resistance is practically equal in all cases.

EQUIVALENT TRAINS ON A .7 PER CENT GRADE AT SLOW SPEEDS.

Cars........	21	30	36	42	48	54	60	66
Tons..........	1,220	1,202	1,185	1,168	1,151	1,131	1,117	1,100
Average weight....	51	40	33	28	24	21	$18\frac{1}{2}$	$16\frac{2}{3}$
17.5 T........	21,350	21,000	20,700	20,100	20,150	19,850	19,530	19,250
50 C..........	1,200	1,500	1,800	2,100	2,400	2,700	3,000	3,300
Resistance ..	22,550	22,500	22,500	22,500	22,550	22,550	22,530	22,550

Thus it will be seen that all these trains give approximately the same resistance or pull back of the tender, and this is what is known as "adjusted or equated tonnage rating." This resistance is, in some cases, about 60 pounds greater than our calculated force at the back of the tender, but this is close enough for practical purposes in all cases.

The subdivision or combinations of cars and loads between the maximum and minimum tonnage allowances (in this case 1,220 and 1,100 tons) has been the subject of many papers by various authorities and of many tables and formulæ to effect a ready solution. Formula 86 is a simple one, and has the advantage of requiring the knowledge of but two items, the tons and cars in a train, and these are always known factors. Thus, the force back of tender for the various classes of engines upon the controlling grade could be given to the conductors and yardmasters, and enough cars picked up to make the resistance equal the engine power. While this is a simple mathematical operation, it is desirable to eliminate such work as much as possible from the outside forces, and confine it to the engineering offices. It is therefore considered most satisfactory to publish the rating for the various classes of engines, both with loaded and empty cars, in some such shape as illustrated by the following table:

In the example chosen, the loads are figured on the basis of weighing 50 tons per car (car and lading) on the average;

CHICAGO DIVISION.

Tonnage Rating for Full Trains of Loads and for Full Trains of Empties.

Class of engine	158		389		451		566 and 606		1000		1050	
	Loads.	Empties.	Loads.	Empties.	Loads.	Empties.	Loads.	Empties.	Loads.	Empties.	Loads.	Empties.
WESTBOUND.												
Chicago to Streator	1,240	970	1,610	1,260	1,420	1,100	2,000	1,600	1,650	1,330	2,350	1,900
Streator to Chillicothe	1,080	860	1,110	1,120	1,210	990	1,780	1,125	1,400	1,120	2,000	1,600
Chillicothe to Shopton (pusher to Edelstein)	960	780	1,230	1,000	1,080	870	1,550	1,275	1,275	1,050	1,820	1,500
Shopton to Marceline	750	600	970	790	850	690	1,230	1,000	1,050	900	1,500	1,300
Marceline to Argentine	810	650	1,050	850	920	750	1,330	1,075	1,100	950	1,550	1,350
EASTBOUND.												
Argentine to Marceline	810	650	1,050	850	920	750	1,330	1,075	1,100	950	1,550	1,350
Marceline to Shopton	750	600	970	790	850	690	1,230	1,000	1,050	900	1,500	1,300
Shopton to Chillicothe	960	780	1,230	1,000	1,080	870	1,550	1,350	1,275	1,050	1,820	1,500
Chillicothe to Streator	1,080	860	1,410	1,120	1,210	990	1,780	1,125	1,400	1,120	1,500	1,300
Streator to Chicago	1,290	1,000	1,680	1,320	1,175	1,175	2,100	1,650	1,720	1,360	2,470	1,940

NOTE: Ratings for loads are calculated for loads of 50 tons per car. When a train consists of loads and empties use a proportionate amount of each as found by "Diagram for making up tonnage trains."

that is, a train of 1,000 tons should consist of 20 cars, even if some cars weigh over and other less than 50 tons apiece. The empties are figured at 16 2-3 tons each car. Of course, any other figures could be assumed and applied by formula 86.

The table, on its face, only applies to full trains of 50-ton loads or empties, and mixed trains of loads and empties must be separately determined. Several years ago Mr. L. R. Pomeroy presented a method which he had devised to instantly solve

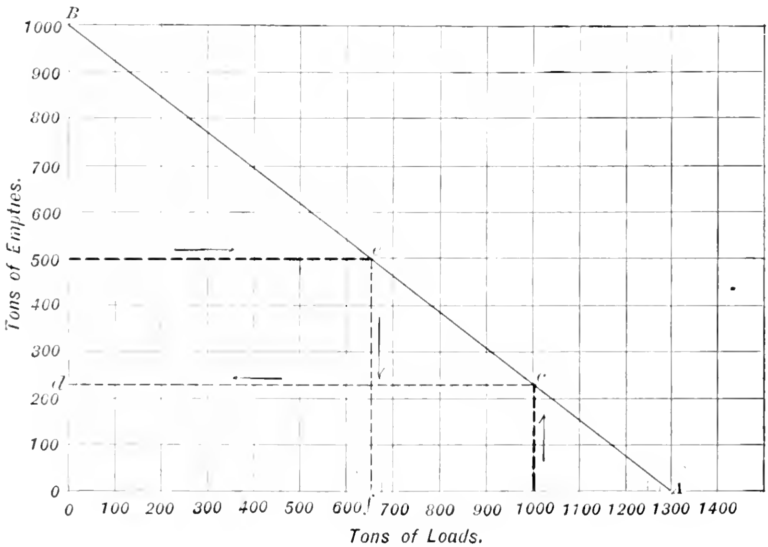


Fig. 94.

this question of mixed trains without requiring any computations. This is shown in Fig. 94, and it is explained as follows: In the Chicago Division table just presented, let us consider 158 class engine from Streator to Chicago, where the rating is given as 1,290 tons of loads and 1,000 tons of empties. Take a piece of cross-section paper and graduate the side in tons of empties and the bottom in tons of loads, as shown up to and including the limits of the train in question. Establish the point A at 1,290 tons on the loaded car line at bottom, and the point B at 1,000 tons on the empty car line at side. Draw a straight line between them, and the diagram will show, without calculation, how to fill up the train. Suppose we have 1,000 tons of loaded cars to haul and we desire to know what tonnage in empties the engine can take in addition, in order to complete the rated load: From the point opposite 1,000 tons at the bottom of the diagram follow the vertical line to the diagonal at c; then follow the horizontal line from c to d at the left-hand side of the diagram, where we find 230 tons as the amount of empties to be taken. Or, if we had 500 tons of empties and desired to know how many tons of loads would be required to make up a full train, start at 500 on the left side and follow horizontally to the diagonal at e, then downward to the base line at f, where we read 650 tons as the desired amount in loaded cars needed to complete the train rating.

This, however, does not take care of a train of cars which are not empty, but have light loads, such as 20, 25 or 30 tons to the car, total weight. Plate 32 (at end of book) embodies a diagram for this purpose, and is intended to be mounted upon a board and to have tacks placed in the dots of the left column and the bottom line. The loaded cars are based upon 50 tons each, as will be seen by the bottom lines, and the empties upon 16 2-3 tons each, as shown by the left-hand columns. Each square contains two numbers, the upper designating the number of cars, and the lower the number of tons in a train. Each number is the sum of the corresponding numbers in the end squares on the same vertical and horizontal lines as found in the left-hand column and the bottom line. The instructions for use are embodied in the plate—the rules for empty and fully loaded cars (50 tons gross) will be found the same as with Pomeroy's diagram, and upon which this was based. Any intermediate combinations, however, can also be quickly determined. Let us take, by way of an example, the train which we previously figured for the 20 by 26 inch locomotive. The line drawn from the 66-car, 1,100-ton "empty" square to half way between 1,200 and 1,250 tons (24 or 25 cars) squares in the loaded car line represents the string referred to in the instructions for using the diagram, and extends to a point about half way between the dot for 1,200 tons and that for 1,250 tons, because the calculated tonnage is 1,220 tons. This line passes through dots of combinations of cars and tons of equal resistance. For instance, commencing at the loaded train of 24 cars and 1,220 tons, we find the following "dots" crossed by the line:

Cars	24	27	32	37	42	47	56	61	66
Tons	1,220	1,217	1,200	1,183	1,167	1,150	1,133	1,117	1,100

These, it will be noticed, agree quite closely with our calculated trains—plenty close enough for ordinary practice. Any train combination to the left of the line will be less than a full loading, as, for instance, 1,100 tons in 26 cars, in which case six empties could be added, as we count three squares from the $\frac{26}{1,100}$ square to our line, and each square vertically represents 33 1-3 tons, or two empty cars. So, for 1,200 tons in 44 cars, the load is too great by one loaded car, there being one square horizontal distance between this train combination and our line. This chart is applicable to any grade or combination, as it depends only upon the line or cord drawn through the terminal squares representing the proper empty and loaded tonnage, and the combinations given will be in accordance with equation 86. The chart can be used equally well by the train conductor—when he drops off some cars (and tons) at a station he will know just how many to pick up at the next in order to fill out his tonnage.

The methods just described give the full tonnage or rating of the engine, but under certain circumstances this rating must be reduced, as in stormy weather, or if the engine be in poor condition. A few roads make no attempt to allow for stress of weather, and in a country like Mexico, where the climate is uniform, there is little need for such allowances. On the other hand, in northern climates, where the winters are particularly severe, it seems very necessary that the weather be considered. This is too often left to the discretion of some sub-official, who either has no desire to reduce the rating or neglects to do so, in the false notion that the heavy loading will improve the operating record of the division; and the engine pays the penalty, Some of the northern roads in this country make the following deductions from the established rating:

In winter,	10 to 15 per cent reduction.
For wet rails,	5 to 10 per cent reduction.

For frosty or wet rail,	7 per cent reduction.
From 32° to zero, Fahrenheit,	15 per cent reduction.
From 0° to -20° Fahrenheit	20 per cent reduction.

For inferior rail and unfavorable weather, 10 per cent reduction.

For inferior rail and stormy weather, 20 per cent reduction.

The dispatcher to determine, according to weather and condition of rail as evidenced by telegraphic reports from all points received twice a day; in the absence of special instructions, full loading is to be taken.

For temperature 40° and above,	no deduction.
For temperature 40° to 30°,	6 per cent deduction.
For temperature 30° to 20°,	12 per cent deduction.
For temperature 20° to 10°,	18 per cent deduction.
For temperature 10° to 0°,	24 per cent deduction.
For temperature 0° to -10°,	35 per cent deduction.
For temperature -10° to -20°,	41 per cent deduction.
For engines out of shop 9 to 12 mos.,	6 per cent deduction.
For engines out of shop 12 to 15 mos.,	12 per cent deduction.
For engines out of shop over 15 mos.,	18 per cent deduction.

These latter deductions seem to be unnecessarily heavy, but the nature of the traffic and the physical characteristics of the country passed through must guide in deciding when and how much to reduce the rating—it would be impossible to give general rules to cover these cases.

Another question that enters upon the rating problem when compound engines are used is the additional power that can be obtained at critical points by running the engine simple. We saw above that the tractive force could be increased from 5 to 15 per cent by this means, and it is generally made use of. If the controlling grades are of short duration, it is perfectly proper to so load the engine that to make the summit it must be operated simple for a short distance, but if the hill is long and continuous, we lose the results for which the compound engine was designed, and an extra sum paid for its construction, by operating it simple for a long distance.

It is this reserve power that enables compounds often to haul greater loads than simple engines of the same cylinder power, and also why some types of compounds are able to take heavier trains than others with the same size cylinders, as will be understood by referring to the general formulæ for tractive force.

RATING OF FAST FREIGHTS.

Fast or time freight trains generally cover stock, fruit or other perishable merchandise which it is desirable to transport with dispatch, and these trains are often called upon to average 30 to 40 miles an hour. This speed cannot be maintained up heavy grades, but it may be desirable to make 20 miles an hour up the controlling grades of the division. The method sometimes adopted is simply to take off 10 per cent (or some other arbitrary amount) from the "slow" rating, but the proper allowance can be determined in much the same manner as for slow freights. Let us consider the same engine and grade as previously, only so proportion the train that 20 miles an hour can be made ascending the .7 per cent grade. If the boiler be sufficiently large, so that we can use plate 29, we can determine the ratio of available tractive force to theoretical tractive force at once, knowing that with the 63-inch wheel, there will be 106 revolutions made per minute, and at this speed the ratio will be .585. (If the boiler be not large enough, we must construct the modified hyperbola as before explained.) The tractive force at rail will then be $\frac{.585 \times 190 \times 400 \times 26}{63} =$ 18,300 pounds. The grade resistance will be as before, or by equation 85, $20 \times .7 =$ 14 pounds,
and for 20 miles an hour from plate 23, 7 pounds,

a total resistance per ton of 21 pounds.

The weight of engine and tender being 132 tons, the resistance will be $= 132 \times 21 = 2{,}772$ pounds, and this subtracted from the tractive force at rail gives $18{,}300 - 2{,}772 = 15{,}528$ pounds back of tender. Equation 86 modified to suit the speed and grade becomes $R_c = 19.50\,T + 50\,C$, and for 100 tons of empties, $19.50 \times 100 + 50 \times 6 = 1{,}950 + 300 = 2{,}250$ pounds and for loaded cars, $19.50 \times 100 + 50 \times 2 = 1{,}950 + 100 = 2{,}050$ pounds. Then, as before, dividing the tractive force by these values, we obtain—

$$\frac{15{,}528}{2{,}250} = 690 \text{ tons of empties in 41 cars, and}$$

$$\frac{15{,}528}{2{,}050} = 757, \text{ say, 750 tons of loads in 15 cars.}$$

The interpolation for mixed trains and loads can be made precisely as before for the slow freight trains.

The speed which can be maintained on any lower grade with the same train can also be determined, as, for instance, 750 tons in 15 cars on a 0.2 per cent grade. Let us try 30 miles an hour, or 159 revolutions per minute for the drivers. Then we have for

$$\text{A. T. F} = \frac{.4 \times 190 \times 400 \times 26}{63} = \qquad 12{,}500 \text{ lbs.}$$

$$\text{Car resist.} = 12 \times 750 + 50 \times 15 = 9{,}750$$

$$\text{Loco. resist.} = 13.5 \times 132 = 1{,}780 \qquad 11{,}530 \text{ lbs.}$$

$$11{,}530 \qquad 970 \text{ lbs.}$$

or 970 pounds tractive force to spare at 30 miles an hour.

Fig. 95 illustrates a very convenient diagram for solving such questions rapidly, when there is much of this to be done. The curved lines are calculated for several different points in their length and then connected, showing the combinations of load and grade for each speed. Of course, there should be two such charts—one for loads and one for empties. In Fig. 95, we find that the engine for which it was constructed can take 850 tons of loads up a 50-foot grade at 10 miles an hour, or, on the same grade, 725 tons at 15 miles an hour; also that

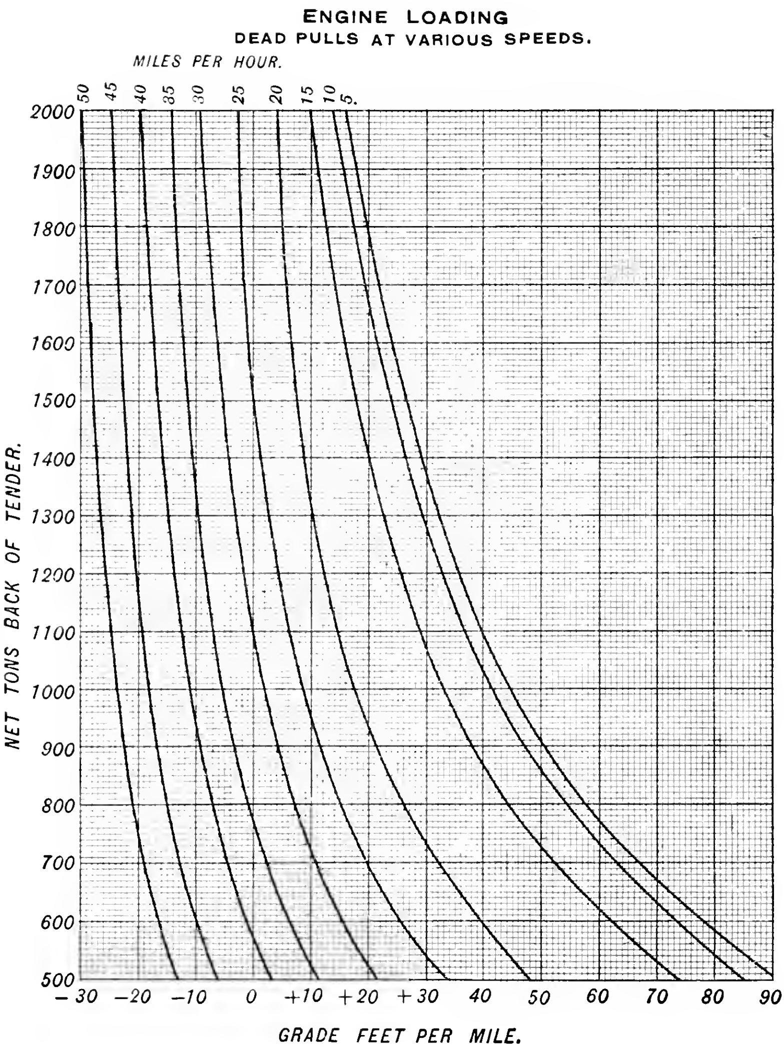

ENGINE LOADING
DEAD PULLS AT VARIOUS SPEEDS.
MILES PER HOUR.
50
45
40
35
30
25
20
15
10
5.
NET TONS BACK OF TENDER.
2000
1900
1800
1700
1600
1500
1400
1300
1200
1100
1000
900
800
700
600
500
– 30
–20
–10
0
+10
+20
+30
40
50
60
70
80
90
GRADE FEET PER MILE.

Fig. 95.

it can pull 850 tons at 15 miles an hour up a 41-foot grade. Such problems can be solved very quickly by the use of a chart, constructed as described above, and in this way the running time of a train over the various up and down grades of a division can be scheduled.

The importance of proper consideration being given to the speed which is desired, is clearly exhibited in this discussion—too often it is passed over in a manner anything but logical or technical, and when engines fail to make the schedule expected, it is a cause for criticism of the locomotive, whereas the real fault lies in not knowing or not caring that the proper reduction in lading is made to adapt the power of the engine to the required velocity. For compound locomotives, the curves for ratio of available tractive force to theoretical tractive force at different speeds should, of course, be constructed and used in place of plate 29, as a basis for the computations.

RATING FOR MOMENTUM RUNS.

If the grades which a train has to surmount be long, that is over 2 miles, the tonnage or rating which can be taken will be determined as just explained. When, however, the grades are short (2 miles or less), the engine can take more than the normal rating for such grades, provided that it be feasible to approach the foot of the grade at a high speed, and to allow this velocity to gradually diminish until the summit has been reached. This has been explained under "Grade Resistance," the loss in velocity acting to produce a grade of less slope than the actual one, and termed a "virtual grade." Formula 3 and plate 2 provide us with the means of determining how much inertia will assist the engine when the velocity is allowed to diminish.

There are a number of variables in this study, however, that require attention—the tractive force of the engine will or may increase as the speed decreases, and the resistance to motion will diminish, so that the effect or force of momentum may not be called upon to act uniformly during the retardation. Plate 33 has been prepared to show exactly what happens when an engine strikes a momentum grade with a heavier train than it

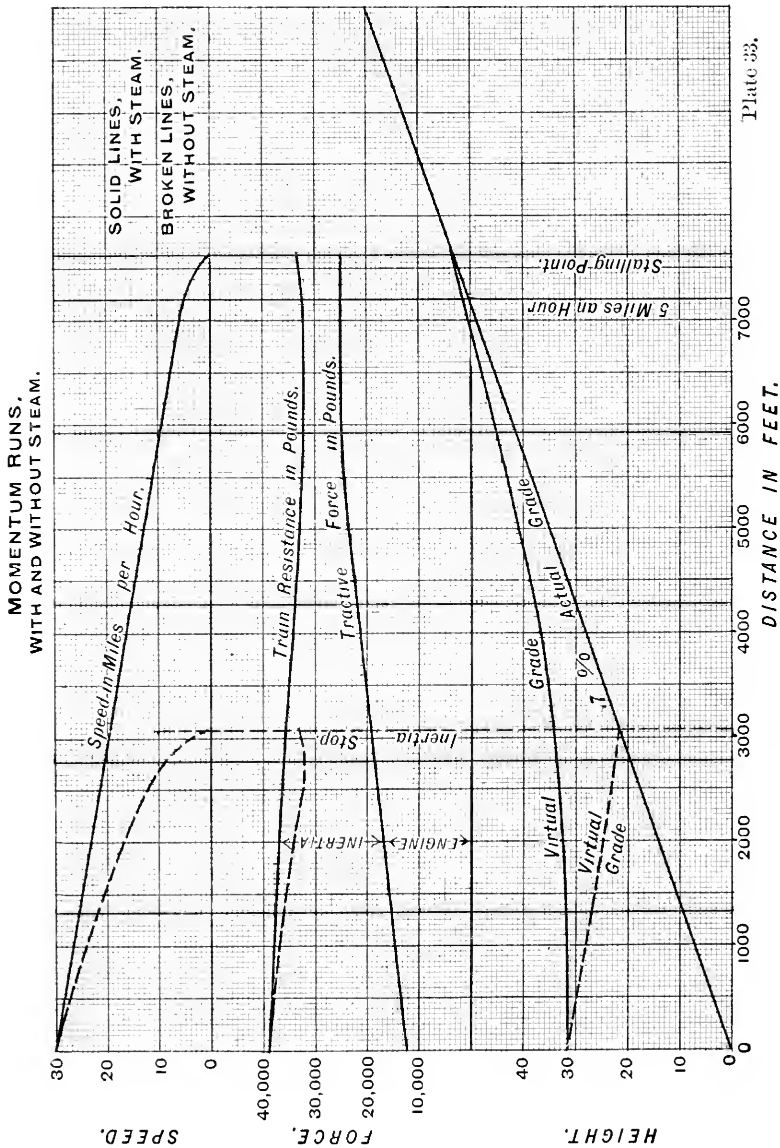

MOMENTUM RUNS.
WITH AND WITHOUT STEAM.
SOLID LINES, WITH STEAM.
BROKEN LINES, WITHOUT STEAM.
Speed in Miles per Hour.
Train Resistance in Pounds.
Tractive Force in Pounds.
Stalling Point.
5 Miles an Hour
Inertia Stop.
INERTIA
ENGINE
Virtual Grade
Actual Grade
.7 %
Virtual Grade
SPEED.
FORCE.
HEIGHT.
DISTANCE IN FEET.
30
20
10
0
40,000
30,000
20,000
10,000
40
30
20
10
0
0
1000
2000
3000
4000
5000
6000
7000

Plate 33.

could haul at a constant velocity. The same locomotive that was considered in the "slow freight" rating will be used, and upon the same grade, .7 per cent, but instead of 1,220 tons, a train of 1,600 tons in 32 cars will be attached to the tender, and a speed of approach at the foot of the grade of 30 miles an hour be assumed as possible; this, of course, precludes the possibility of a grade crossing, water tank, or other stop at the foot of the hill. If, for some unexpected reason, the train was compelled to stop at the foot of the grade, they would have to "double" up the hill. Plate 33 shows a profile of the actual grade, a rise of 7 feet in 1,000 feet; the virtual grades for the assumed conditions of speed, etc.; the train resistance and tractive force of the engine, and the speed at each point of its travel. With the weight of the engine and tender at 132 tons and the theoretical tractive force, that is, $\frac{190 \times 400 \times 26}{63}$ = 31,300 pounds, approximately, we are able, with plates 23 and 29 and equation 86, with its modifications, to figure the various elements of the run as follows:

At 30 miles an hour.

	Pounds.
Locomotive resistance = 9.5 for speed + 14 for grade = 23.5 pounds per ton, or 23.5 × 132 tons =......	3,100
Car resistance = 8 pounds for speed + 14 for grade = 22 pounds per ton = 22 × 1,600 + 50 × 32 =....	36,800
Total train resistance =........................	39,900
Available tractive force = .397 × 31,300 =...........	12,400
Leaving for the force of inertia =........	27,500

These amounts are laid off on the zero ordinate, or axis, with the speed of 30 miles an hour in the upper part of diagram.

From 30 to 25 miles an hour.
(160 to 133 revolutions per minute.)

	Pounds.
Average locomotive resistance = 22.87 × 132 =.......	3,020
Average car resistance = 21.37 × 1,600 + 50 × 32 =..	35,800
Total average train resistance =..................	38,820
Average available tractive force = .44 × 31,300 =.....	13,800
Needed average inertia =......................	25,020

and as train (total) weighs 1,600 + 132 = 1,732 tons, we have

$$\frac{25{,}020}{1{,}732} = 14.45$$ pounds per ton needed as the average force of inertia in dropping from 30 to 25 miles an hour. We can transpose equation 3 to the form, $S = 70 \frac{V_2^2 - V_1^2}{P_t}$, and for this case we have $S = 70 \frac{900 - 625}{14.45} = 1{,}330$ feet; that is, if we drop from 30 to 25 miles an hour in a distance of 1,330 feet, the average force of inertia or momentum will be 14.45 pounds per ton of train.

In the plate a vertical line has been drawn through the 1,330-foot distance on the scale, and the speed is shown as 25 miles an hour. The resistance and tractive force will be

At 25 miles an hour.

	Pounds.
Locomotive resistance = 22.25 × 132 = ...	2,940
Car resistance = 20.75 × 1,600 + 50 × 32 = ...	34,800
Train resistance = ...	37,740
Tractive force = .482 × 31,300 = ...	15,100
Force of inertia = ...	22,640

and these are laid off on the vertical line. This process is repeated for each drop in speed of 5 miles an hour, the calculated values being as follows:

AVERAGE VALUES FOR SPEED CHANGES OF

Data.	25 to 20 Miles.	20 to 15 Miles.	15 to 10 Miles.	10 to 5 Miles.	5 Miles to Stop.
Average locomotive resistance................	2,855	2,650	2,600	2,540	2,640
Average car resistance........................	33,800	31,800	30,200	29,600	29,600
Average train resistance......................	36,655	34,450	32,800	32,140	32,240
Average tractive force........................	17,750	20,200	23,700	25,000	25,000
Average inertia force.........................	18,905	14,250	9,100	7,140	7,240
Inertia, pounds per ton.......................	10.90	8.22	5.25	4.12	4.18
Distance in feet..............................	1,440	1,490	1,660	1,275.	420

VALUES AT DEFINITE SPEEDS OF

Data.	20	15	10	5	Stop.
Locomotive resistance.........................	2,770	2,640	2,575	2,510	3,820
Car resistance................................	32,800	30,800	29,600	29,600	29,600
Train resistance..............................	35,570	33,440	32,175	32,110	33,420
Tractive force................................	18,300	21,900	25,000	25,000	25,000

As the speed diminishes, the increase in tractive force is clearly shown—the difference between the tractive force line and the train resistance line being the amount supplied by inertia.

The broken lines represent a stop on the same grade with the train running while the engine throttle is closed, the run being due to inertia only. In this case, the train will come to a stop in 3,060 feet, while with steam it will run 7,615 feet before stalling. The inertia stops resemble the braking stops shown in Figs. 78 and 79, as far as the parabolic shape of the speed curve is considered. The several points were figured as in the stop with steam, and the virtual grade located by laying off above the actual grade line, the velocity heads at the several points calculated. Here the virtual grade is a straight line with a slope of about 3 feet in a thousand or .3 per cent grade, equivalent to six pounds per ton, which is approximately the rolling resistance of the train, and this signifies that the inertia force has been uniformly distributed throughout the run, and that the drop in velocity has been just sufficient to overcome the train resistance at each instant, resulting in the parabolic curve, as the force is proportional to the difference of the squares of the velocities. With the "steam run," however, this is different—the speed line is straight from 30 to 5 miles an hour, and the increasing tractive force calls for a lessened demand upon inertia. This was seen in the tabulated data, where the amount of inertia needed per ton decreased with the speed, although the distances through which it acted were nearly the same. These peculiarities are borne out by Boyer speed records—the stops, with steam shut off, produce a parabolic curve, while the slowdowns on momentum grades draw a nearly straight line; compare Figs. 80 and 93, reproduced from speed records. In this way the virtual grade is a gradually increasing one, as the power of the engine becomes greater. At a distance of 7,615 feet from the foot, the engine will stall—in fact, in figuring on momentum rating, we should not count on less than 5 miles an hour at the summit. This speed was reached in 7,195 feet, and if the hill had not been over 7,000 feet in length, the train would be able to pass the summit.

The method of figuring each 5 miles drop is too laborious for ordinary rating work; while it is the most accurate way, we can calculate from 30 to 5 miles an hour in one operation, thus:

30 to 5 miles an hour (160 to 27 revolutions).

	Pounds.
Average locomotive resistance = 21.25 × 132 = ...	2,800
Average car resistance = 19.75 × 1,600 + 50 × 32 = ..	33,200
Average train resistance = ...	36,000
Average tractive force = .634 × 31,300 = ...	19,850
Average inertia force = ...	16,150

and $\frac{16,150}{1,732} = 9.3$ pounds per ton, therefore $S = 70 \times \frac{900 - 25}{9.3}$ = 6,580 feet run to 5 miles an hour speed of train. This distance is 615 feet less than obtained by the long method, and is due to the fact that in the short method, we assumed that the average force of inertia would be 9.3 pounds per ton. In plate 33 it is seen that the distances through which the varying inertia forces act are not proportional to the inertia effect—in fact, when we take the average inertia force, as shown by a planimeter, we find that it is only 8.5 pounds per ton, instead of 9.3 and $70 \times \frac{900 - 25}{8.5} = 7,200$ feet, practically the same as by the long method. This discrepancy is due to the varying tractive force of the engine, as in our inertia stop the short and long methods give practically the same results. As the short method errs on the safe side, it will be sufficiently close for practical use.

Having studied the combined action of steam, inertia and resistance on a momentum run, we must look for the practical application of this to engine rating. In reality, we do not want to know how far up a grade an engine will take a train before dropping to 5 miles an hour speed, but, what train can be taken up a grade of certain amount and length, and just reach the summit at 5 miles an hour, which is the converse proposition. Let us figure our previous case in this way: Given a .7 per cent grade, 6,580 feet long, how many tons in 50-ton carloads can be taken up with a speed of approach of 30 miles an hour, the summit to be reached at a speed of 5 miles an hour.

For 50-ton cars, the rate will be (coefficient of $T = \frac{8 + 3.50}{2} = 5.75$ pounds average for speed + 14 pounds for grade, or total = 19.75) $19.75 \times 100 + 50 \times 2 = 2{,}075$ pounds per 100 tons, or 20.75 pounds per ton of train (including engine and tender at same rate for simplicity). Then from equation 3, $P_t = 70 \times \frac{900 - 25}{6{,}580} = 9.3$, and $20.75 - 9.3 = 11.45$ pounds per ton to be furnished by the locomotive, and as the average tractive force is 19,850 pounds, we have $\frac{19{,}850}{11.45} = 1{,}734$ tons, and subtracting the weight of engine and tender, we have $1{,}734 - 132 = 1{,}602$ tons, or say 1,600 tons in 32 cars back of tender. In the same manner we find for empties: $19.75 \times 100 + 50 \times 6 = 22.75$ pounds per ton, $22.75 - 9.3 = 13.45$, and $\frac{19{,}850}{13.45} = 1{,}475$ tons total, or $1{,}475 - 132 = 1{,}343$ tons in 80 cars. The inertia factor 9.3 can be taken without calculation, from plate 2, at the intersection of the 30-mile curve with the 6,580 foot distance ordinate, so in the short method, with 9.3 found by our figures, the distance can be obtained at once from plate 2.

If there be much momentum rating to be done, it will economize time to lay out a diagram as shown in Fig. 96. Each curve intersects a grade line at a load line that can be taken up a hill of the length designated by the line when the approaching speed is as calculated. The "dead pull" line is for long grades where momentum cannot apply, the "empty" being constructed accordingly. This can also be used for empty trains on momentum grades, as will be explained. These curves can be located by points, using the short method and plate 2, and a great amount of subsequent work saved. One

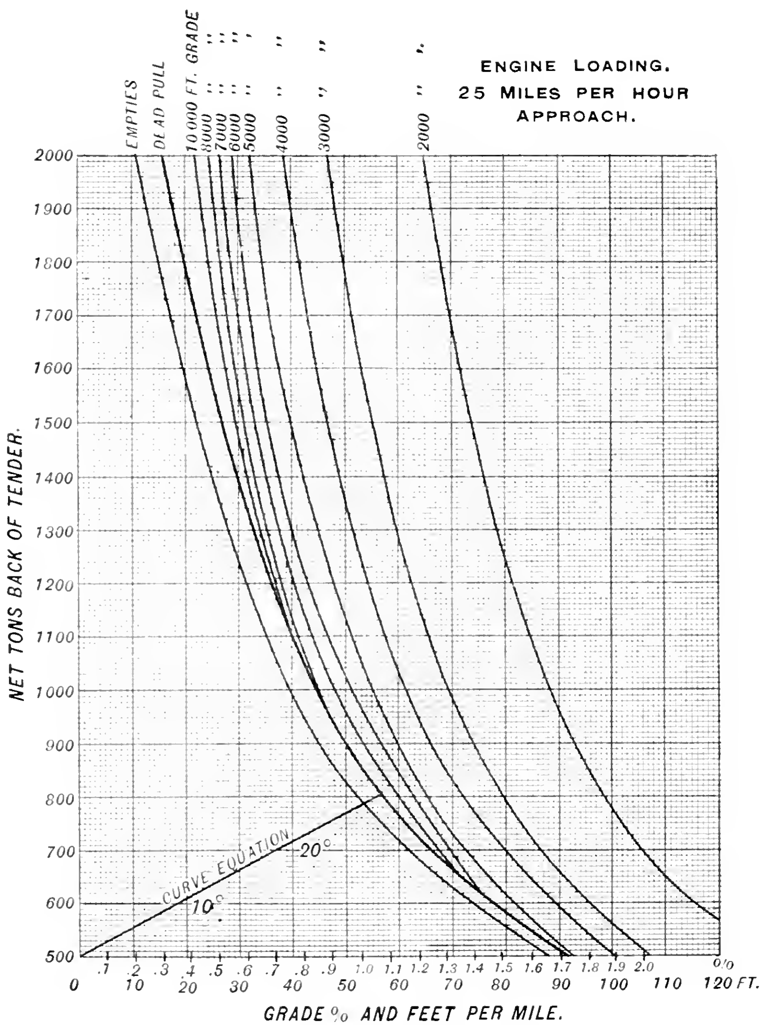

ENGINE LOADING.
25 MILES PER HOUR
APPROACH.
EMPTIES
DEAD PULL
10 000 FT. GRADE
8000
7000
6000
5000
4000
3000
2000
NET TONS BACK OF TENDER.
2000
1900
1800
1700
1600
1500
1400
1300
1200
1100
1000
900
800
700
600
500
CURVE EQUATION
10°
20°
.1 .2 .3 .4 .5 .6 .7 .8 .9 1.0 1.1 1.2 1.3 1.4 1.5 1.6 1.7 1.8 1.9 2.0 %
0 10 20 30 40 50 60 70 80 90 100 110 120 FT.
GRADE % AND FEET PER MILE.

Fig. 96.

such diagram will be needed for each class of engine, for accurate results. In Fig. 96, we find that the engine selected can take 1,100 tons of loads back of tender up a 40-foot grade of any length, whereas only 1,000 tons of empties could be hauled. Mixed trains should be determined by means of plate 32. If, however, the grade be only 5,000 feet in length, 1,500 tons of loads could be hauled. By running across on the 1,500-ton horizontal line to its intersection with the "dead pull" line, we find the virtual grade is 27 feet per mile, as the engine could

RELATIVE ENGINE LOADING.

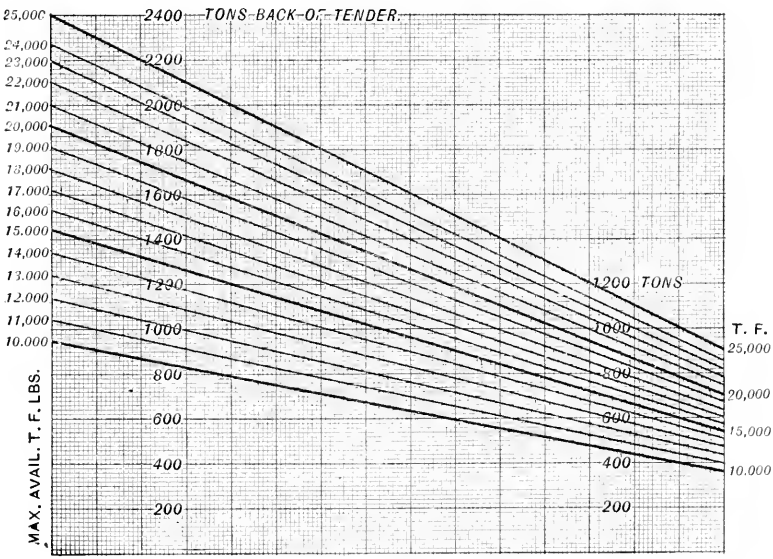


Fig. 97.

take 1,500 tons up a 27-foot grade for an unlimited distance. Therefore, the empties must be governed by this virtual grade, and following down the 27-foot line, we find that it intersects the "empty" line at 1,330 tons, which is the empty load back of tender, mixed trains to be computed by means of plate 32. It is believed that these explanations will be sufficient to enable the proper charts to be constructed for individual cases, so that the whole problem of engine rating, at slow and fast speed, and also with momentum, can be expeditiously handled. The charts for each engine can be used on any grades that may occur, and are not confined to one division or rate of gradient.

The line marked "curve equation" is used to convert curvature into grade of equivalent resistance, when necessary. Fig. 97 illustrates an approximate method of obtaining the rating for other classes of engines, without constructing additional charts, like Fig. 96. This gives the load suitable for an engine having a different tractive force when the load for the "standard" engine is known. Thus, if the charts like Fig. 96 be gotten up for the "standard" engine, having 25,000 pounds maximum available tractive force, and from such chart a load of 1,200 tons be found as proper for the grade and conditions being considered, then under similar conditions, an engine with 20,000 pounds maximum available tractive force would take 940 tons; this is determined by following down on the vertical lines from the intersection of the 25,000 tractive force line and the 1,200-ton line, to the crossing of the 20,000 tractive force line, which is found to be at 940 tons.

These lines are not radial, as would seem at first sight, but are parallel to radial lines whose tangents are proportional to the tractive forces of the different engines, the parallel being drawn below the radial lines by an amount corresponding to the weights of engine and tender. In this way, the loads back of the tender are comparable by the diagram, whereas it would be the power at the drivers, were the radial lines used. For accurate results, however, individual charts like Fig. 96 should be prepared for each important class of engine.

STARTING AND STOPPING.

The principal resistances to be overcome in starting a train, especially where quick acceleration is desired, are those of inertia. Plates 1 and 3 gave us a graphical idea of the large forces needed for rapidly putting a train in motion, showing that it required nearly 100 pounds horizontal force per ton of weight to bring a train from rest to a speed of 30 miles per hour in 30 seconds. This is about 5 per cent of the weight of the train, including engine and tender. But this force should be applied uniformly and regularly to produce a constant acceleration, and the power of the locomotive, as we have seen, cannot be maintained constant at increasing speeds. We shall therefore obtain a higher rate of acceleration at the instant of starting, and as the speed increases and the tractive force diminishes, the rate of acceleration will also fall off, the action of the locomotive being just the reverse here of what occurs on a momentum grade.

In connection with the preparation of a report on the "power required to operate the trains of the New York Central & Hudson River Railroad and the relative cost of operation by steam and electricity," by Bion J. Arnold, in 1902, a series of tests were made to determine the accelerating power of a heavy suburban locomotive, near Schenectady. As these tests are the best adapted for the purpose of our discussion that we know of, we will compare the actual results with those which might have been expected from the character of the tests. The locomotive used was New York Central 1407, of the 2—6—6 type, with a total weight of 214,000 pounds, including the tank, which was built on same frame as the engine, over the 6-wheel truck. The cylinders were 20 by 24 inches, the drivers 63 inches diameter, and the boiler pressure 200 pounds, the heating surface being 2,437 and the grate area 62 square feet. The theoretical tractive force was thus $= \dfrac{200 \times 400 \times 24}{63} =$ 30,300 pounds, approximately. The runs were made against an up grade of 1 per cent, and the trains selected for study were composed of: A, six cars, total weight 264 tons, including engine; B, three cars, total 184 tons, and C, one car, total 130 tons. In starting, the throttle was opened wide and steam used full stroke, the lever being brought back as the speed increased. The velocities attained are shown in the central portion of Fig. 98, by the broken lines A′, B′ and C′, respectively, the abscissæ giving time in seconds, and the ordinates speeds in miles per hour. The solid lines are the "calculated starts," and were produced as follows: The velocity obtained during 10-second intervals was determined, and added to that at the commencement of the 10-second period, thus giving the final velocity.

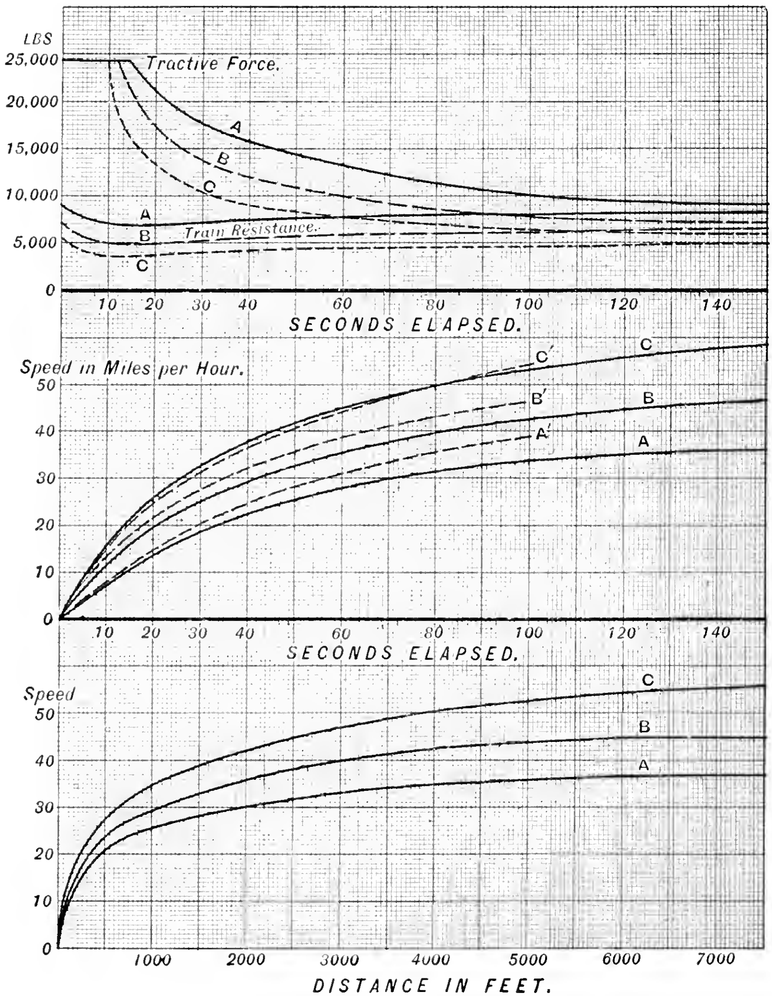

LBS
25,000
20,000
15,000
10,000
5,000
0
Tractive Force.
A
B
C
Train Resistance.
10 20 30 40 60 80 100 120 140
SECONDS ELAPSED.
Speed in Miles per Hour.
50
40
30
20
10
0
C′ C
B′ B
A′ A
10 20 30 40 60 80 100 120 140
SECONDS ELAPSED.
Speed
50
40
30
20
10
0
C
B
A
1000 2000 3000 4000 5000 6000 7000
DISTANCE IN FEET.

Fig. 98.

Thus, at starting, we have for the tractive force, train A = .8 × 30,300 = 24,300 pounds. The grade resistance is 20 pounds (for 1 per cent), and from rest take 10 pounds for friction, etc., so that the total is 30 × 264, or 7,920 pounds. The difference, 24,300 — 7,920 = 16,380, is the accelerating force and amounts to $\frac{16,380}{264} = 62$ pounds per ton. By transposing equation 2 to the form $V = \frac{P_t\, t}{95.6}$, we obtain the speed at the end of the time interval, or $V = \frac{62 \times 10}{95.6} = 6.5$ miles an hour in 10 seconds of time. This is one point in our curve marked "A." For the second point, the increasing speed will cause a decrease in the tractive force, but as the train has been gotten under way, the resistance has also reduced, and we have—

	Pounds.
Tractive force = .78 × 30,300 = ...	23,700
Resistance of train = 25½ × 264 = ...	6,750
To overcome inertia = ...	16,950

and $\frac{16,950}{264} = 64$ pounds per ton or 6.7 miles an hour added, or a total speed in 20 seconds of 6.5 + 6.7 = 13.2, which gives us a second point in curve A. This has been repeated up to 150 seconds of time elapsed from start, and trains B and C, treated in the same way. The particularly close correspondence of the theoretical and actual curves of train C is especially interesting. In the upper diagram of Fig. 98 the tractive forces and train resistances have been laid off—the distance between these two lines of a set is the amount available for overcoming inertia, and this value is so small at high speeds that the average is greatly reduced; therefore, it is not practicable to use an average value for the tractive force, as the lower powers work over so much longer distances, that we cannot say accurately what this average will be, without laying it out for short intervals, as we have just done.

The lower portion of the figure shows the speed curves laid off on distances run as abscissæ, and gives a better idea of what length of runs are necessary to obtain high speeds. It must not be forgotten that the computations for this series consider an adverse grade of 1 per cent. On a level, the acceleration would be much more rapid, as the resistances would be only about one-fourth or one-third as great. The locus for any combination of locomotive train, grade, etc., may be produced in the manner just described. The distances run were determined by formula 3 transposed to the form $S = 70 \frac{V_2^2 - V_1^2}{P_t}$.

As every start implies of necessity a stop, it is interesting to compare the acceleration and the retardation of a train. In our study of braking and stopping, we found that the high-speed brake could produce retarding forces equal to about one-eighth of the weight of the train, and in addition to this we have the regular train friction, which at 10 pounds per ton for an average, would be .5 of 1 per cent of the load. Using plate 27 to determine the average force which would bring the trains just discussed to a stop in 4,000 feet from the velocity attained in the same distance, we find for train A, from 35 miles an hour, 1.1 per cent; train B, 42 miles speed, 1.6 per cent, and for train C, 50 miles, 2.2 per cent. These figures cover inertia only. As the engine was operating upon an adverse grade, in making a comparison with braking power, we should consider a negative grade—one opposed to the actions of the brakes as the up grade is opposed to the action of the locomotive. To make the comparison still more accurate, the resistance should also be considered as opposed to the brake action, instead of assisting as ordinarily. Then we have for the three trains—

Reference letter	A	B	C
Per cent of load to overcome inertia	1.1	1.6	2.2
Per cent of load to overcome grade	1.	1.	1.
Per cent of load to overcome friction	.5	.5	.5
Total force in per cent of load	2.6	3.1	3.7

As compared with braking powers of 12 and 14 per cent, these figures are very low, so it is evident that it will take much longer to bring a train up to speed, than to stop from a corresponding speed. As an actual fact, with friction assisting the brakes and opposing the engine, this is still more marked. Take, for instance, the very light 3-car train B, weighing but 184 tons, including the engine, and which required a distance of 7,500 feet to attain a speed of 45 miles an hour; with brakes applied producing a resistance of 12.5 per cent, speed friction at .5 per cent, and grade (rising) at 1 per cent, the total retarding force would be 14 per cent of the weight of the train, and a stop would be secured in about 500 feet from a speed of 45 miles an hour—just one-fifteenth of the distance required to attain the speed. If the grade were negative, the distance would be about 600 feet. This explains the well-known feature in Boyer speed records, that the drop of the pencil in coming to a stop is very much quicker than its rise when pulling out of a station.

The enormous power required for high rates of acceleration is clearly demonstrated by the above analysis, particularly when the upper diagrams of Fig. 98 are considered, showing the quick and great reduction in power available for this purpose as the velocity increases.

The time or distance lost in making a stop, as well as the extra energy expended, can be computed from the information used in this study; take, for example, train B. From a speed of 45 miles an hour, on a 1 per cent up grade, the distance to a stop, as just found, would be 500 feet; to regain this speed requires 7,500 feet, or a total distance from 45 to 45 miles an hour again, of 8,000 feet. From equation 2 transposed, we have, $t = 95.6 \frac{V}{P_t} = 95.6 \frac{45}{280} = 16$ seconds (14 per cent, being equal to 280 pounds per ton), and as the acceleration to 45 miles requires 125 seconds, the time from speed to speed $= 125 + 16 = 141$ seconds. If the speed had been maintained regularly at 45 miles an hour, in this time the distance traveled would have been $\frac{45 \times 5{,}280 \times 141}{3{,}600} = 9{,}306$ feet, instead of 8,000 feet, as with the stop, thus the loss in distance owing to the stop is 9,306 — 8,000 = 1,306 feet, about 1/4 mile, or 20 seconds of time, not allowing for any time at rest after motion has ceased. For the difference in work done by the locomotive, if the speed had remained constant, we should have (13 + 20) × 184 × 8,000 = 48,576,000 foot-pounds exerted. With the stop, and not counting the work done in compressing air for the brakes, we find from plate 1 an average force of 18.5 pounds per ton to accelerate to 45 miles an hour in 7,500 feet, so that the work done by the engine will be approximately (18.5 + 20 + 10) × 184 × 7,500 = 66,930,000 foot-pounds, an increase of 18,354,000 foot-pounds over the continuous run.

HORSEPOWER CHARACTERISTICS.

While the tractive force of a locomotive gives a value that conveys to the mind an idea of what it can pull, it does not convey any impression of "work accomplished," except that from habit, we are accustomed to think of this tractive force being maintained at speeds of from 5 to 10 miles an hour. The term "horsepower" includes both tractive force and speed, so that a suggestion of work accomplished is contained in such an expression.

Formula 55 gave us for the indicated horsepower = I. H. P. $= \dfrac{\text{M. E. P.} \times d^2 s V}{375 D}$; in this, however, we recognize the indicated tractive force of equation 97, where I. T. F. $= \dfrac{\text{M. E. P.}\, d^2 s}{D}$, so that we can write more simply

$$\text{I. H. P.} = \frac{\text{I. T. F.} \times V}{375} \quad \text{.........................(109)}$$

and if we substitute for I. T. F. the available tractive force at the circumference of the drivers = T. F. (as explained with equation 99), we have the available horsepower at the point of contact with the rail =

$$\text{A. H. P.} = \frac{\text{T. F.} \times V}{375} \quad \text{.........................(110)}$$

We have seen that the total train resistance must not exceed T. F. (including engine and tender) if the locomotive is to be able to haul the train; we can therefore state this rule simply as follows: The available horsepower of a locomotive, at the circumference of its drivers, is equal to the available tractive force (at the drivers) or the total resistance of the train (including engine and tender) multiplied by the speed in miles per hour at which the train is moved continuously, and divided by the constant 375. From this we see that if we know the tractive force of a locomotive and the speed at which it can maintain this force, we can determine its horsepower; or if we have the resistance of a train and the speed at which it is to run, we can tell how much horsepower it will require to do the work. Under these circumstances, the horsepower of a locomotive becomes a quantity of great value. While the boiler limits the maximum horsepower, as was seen in the last chapter, the valve gear exercises restrictions at high speeds, so that it cannot be considered a constant quantity; we may also study the horsepower at the back of the tender, where the useful work is actually performed, and this necessitates deductions of the rolling, grade and curve resistances of the engine and tender. We can construct characteristics for these several points, viz., at circumferences of drivers and back of tender, as well as for the indicated cylinder power, and thus obtain a clear conception of the variation of the power of the engine due to changes in speed. This can best be illustrated by an example. Let us take the New York Central suburban engine used in the acceleration tests, and construct for it the desired characteristics. The ratio of heating surface to grate area is $\frac{2,437}{62} = 40$ (approximately) and as the fuel is anthracite coal, presumably in large sizes, we find from Fig. 91 that 15 pounds of steam per square foot of heating surface per hour from and at 212 degrees is the maximum that could be expected, or, allowing for evaporation, $\frac{15}{1.2} = 12.5$ pounds steam at boiler pressure, and for the total steam per hour,

$2,437 \times 12.5 = 30,400$ pounds. As the cut-off pressure at full stroke will be about $200 \times .9 = 180$ pounds, whose weight is .432 pound per cubic foot, the volume of steam per minute available will be $\dfrac{30,400}{.432 \times 60} = 1,165$ cubic feet. The volume of one cylinder is 4.37 cubic feet, so that the number of revolutions per minute which can be supplied with steam at full stroke is $\dfrac{1,165}{4 \times 4.37} = 67$, or with a 63-inch wheel, 13 miles an hour.

With our remarks upon plate 29 as a basis, we can construct our curve of tractive force ratios, as has been done in the upper diagram of Fig. 99, the hyperbola starting at 13 miles an hour, from the line marked 1.0 at the left. As the theoretical tractive force $= \dfrac{200 \times 400 \times 24}{63} = 30,300$, the available tractive force will be 30,300 times the ordinate of the curve at any point, and from equation 110, the available horsepower will be A. H. P. $= \dfrac{30,300 \, y \, V}{375}$ if by "y" we mean the ordinate of the curve at the velocity V. This gives us A. H. P. $= 80.6 \, y \, V$, so that the horsepower can be at once obtained by multiplying together the constant 80.6, the ordinate of the broken line, and the speed. (The constant 80.6 is for this size of engine only.) Locus "a" has been constructed in this way, and becomes the characteristic for this engine at the circumference of the drivers. The line "b" deducts the horsepower necessary to propel the engine itself (containing the tender) on a level, and so gives the horsepower at drawbar of tender. Lines c and d have the power deducted necessary to ascend .5 and 1 per cent grades, respectively, and give the power back of tender for those conditions. The point of greatest interest to the transportation department of a railroad is the power available at this locality—the tender drawbar, as it determines the commercial value of the engine, and the quantity of work done by a locomotive is of the greatest importance—often much more important than the question of fuel economy. Under these conditions, it is necessary to know at what speed we can obtain the greatest horsepower, and this is clearly shown by the characteristics. We see that all the lines, a, b, c and d, rise rapidly

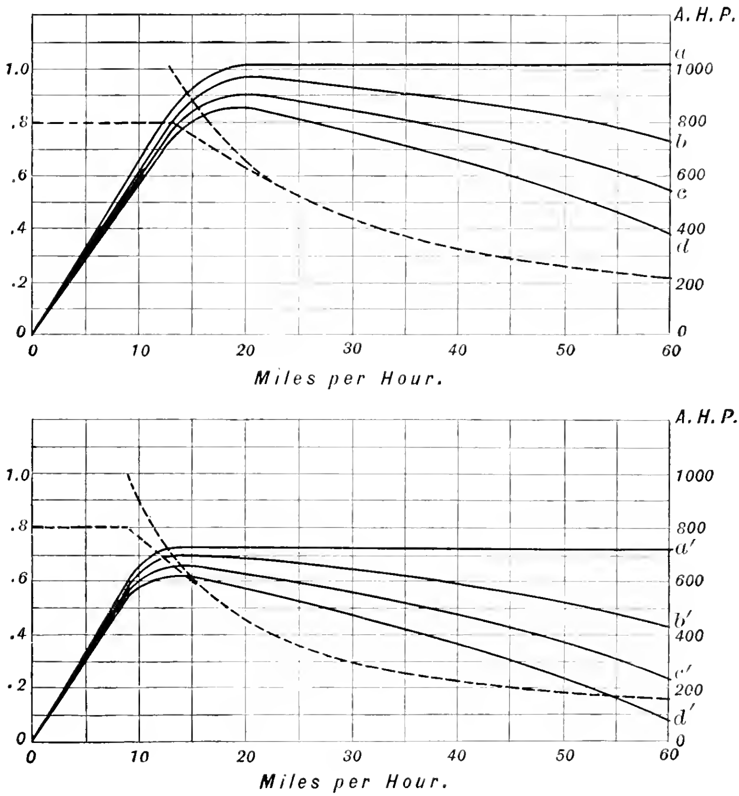


Fig. 99.

until the capacity of the boiler is reached; from this point the locus a assumes a horizontal direction, which means that the horsepower at the rail is constant for speeds above 20 miles an hour. The line b, however, drops from this point, as the resistance of the engine entails a larger amount of power to overcome it at high speeds. The same is true of gravitation, in even a more marked degree, as is shown by the curves c and d. As the characteristics b, c and d are the commercial ones, that is, of useful power, they are of the greatest interest. In each case we see that the maximum amount of power is obtained when the engine is running at a speed of 20 miles an hour, and that to obtain the greatest quantity of work from it, that speed should be maintained. The greatest tractive force is at velocities below 13 miles an hour, but the speed is so slow that less work is accomplished, and this means less "ton miles" hauled where the resistance is uniform over the division. Of course, other points of issue may make a heavier and slower train desirable, or a lighter and faster one, but in either case there will be less work obtained from the engine.

If the dimensions of the locomotive are changed, the characteristics will also be modified. Suppose, for instance, the boiler were about 30 per cent smaller, or would supply steam for full stroke up to 9 miles an hour only, then we should have characteristics like a′, b′, c′ and d′ in the lower diagram. Here the available horsepower is not only less, but the maximum for the "commercial" curves is at about 14 miles an hour, which shows that the capacity of the boiler not only limits the maximum power of the engine, but also the speed at which the maximum power can be developed. Thus it is apparent that the characteristic of a locomotive presents an exceedingly valuable method of analyzing its performance, and as such few elements are necessary for its computation, it can readily be constructed in advance of the design of the locomotive, by assuming the leading dimensions, and modifying them so as to produce the desired results.

As a rule, the maximum horsepower is not maintained for any great length of time, as the undulating profiles of most roads continually change the conditions of operation. A 4—4—2 passenger engine on the Chicago & Northwestern Railway took 11 cars from Chicago to Clinton, 138 miles, in three hours and 14 minutes. While at some places the indicated horsepower reached 1,500, yet the average for the trip was about 1,250, and as there were 3,015 square feet of heating surface, the average rate was about 2 1-3 square feet per indicated horsepower; the maximum rate was 2 square feet to a horsepower. If rough approximations only be desired, we may base our line a or a′ directly upon the horsepower as represented by the heating surface, using the proportions per indicated horsepower given in the last chapter, and deducting an allowance for internal friction, say 8 per cent. This will be fairly accurate for speeds above that at which the boiler will supply the cylinders at full stroke, but not for speeds below this point. In this way compound engines may be quickly treated, the necessary heating surface for such locomotives being about 15 per cent less than for simple engines working with an early cut-off.

CHAPTER VIII.

WATER CONSUMPTION.

The question of water supply was, for many years, considered to be one of quantity only, and wells were dug and water tanks located at points where a liberal supply could be secured, regardless of the kind of water furnished to the locomotives. In most waters, there is much that is not water, but material that either causes trouble in operating the engines, or expense in maintenance. While it is no doubt true that quantity is of greater importance than quality, and also that an engine low in water is justified in taking any kind that can be secured, it is also a fact that much expense in operation and maintenance can be saved by the proper and scientific handling of the question of water supply. We will, therefore, study this question under two general captions—quantity and quality—with such subdivisions as seem logical and necessary to bring out the different points under investigation.

MAXIMUM QUANTITY OF WATER.

In our study of the steam capacity of locomotive boilers there was presented a diagram (Fig. 91) showing approximately what output could be expected as a maximum, when various proportions of grate area and heating surface existed, and various kinds of fuel were used. While the loci there shown extended to rather large values for the weight of steam from and at 212 degrees per square foot of heating surface per hour, under ordinary conditions we do not obtain more than 15 or 16 pounds, or, say, from 12 to 13 pounds at the working pressure of the boiler, and even this is seldom maintained for any length of time, unless the engine be undergoing a test upon a specially prepared plant, or under a special road schedule. However, such rates of evaporation can be obtained —and possibly higher; but even 10 pounds would be ordinarily a high limit to maintain for any great length of time, as, for instance, an hour or longer. In making a series of tests upon the Chicago & Northwestern plant, a rate of evaporation as high as 13.4 pounds of water per square foot of heating surface per hour at boiler pressure (170 pounds) was maintained for 35 minutes, but for runs of one hour or more, the rate did not exceed 10 pounds. In some road tests recently conducted in England upon the Furness Railroad, the highest evaporative rate in 15 tests was 9.46 pounds per square foot of heating surface per hour for a run of 120 miles, lasting 5½ hours.

As may be expected, the maximum rate of evaporation is practically independent of the speed at which the engine is

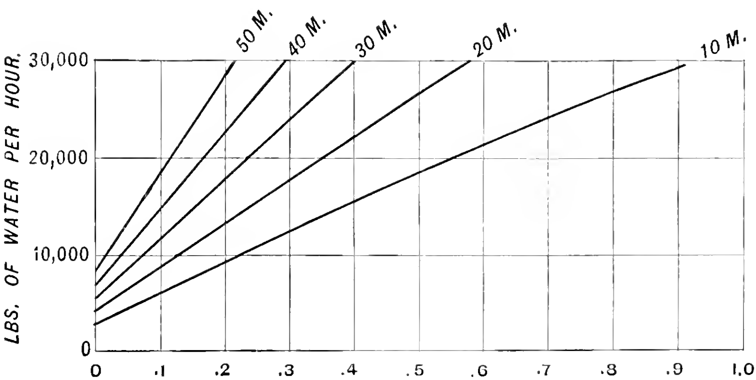


Fig. 100.

running, that is to say, the maximum cut-off that can be used (and maintain boiler pressure) at any and all speeds, except very low ones, will consume the same amount of steam, or all that the boiler can furnish. Fig. 100 gives a graphical representation of the results of the Chicago & Northwestern tests referred to above. The abscissæ give the proportion of cut-off and the ordinates the pounds of water evaporated per hour at the working pressure. Of course, the actual origin of the curves is at the clearance distance, .08 of the stroke to the left of the zero cut-off. At 12 pounds per square foot, the quantity of steam would be $2,332 \times 12 = 28,000$ pounds per hour, but as 30,000 pounds were reached in short runs, this figure is shown as the limit for the curves. It will be noticed that the cut-offs range as follows:

Speed in miles per hour....	10	20	30	40	50
Maximum cut-off	.94(?)	.58	.41	.30	.22

These are not inversely as the speeds, that is, .58 is more than half of .94 and .30 is more than one-fourth of .94, but if we add .08 for clearance, and then make correction for the variation in density of steam due to the difference in cut-off and speed, and allow for condensation, we find that the weight of steam used under these conditions in a unit of time is nearly uniform.

From plate 30 we can obtain the available tractive force at the above speeds and cut-offs. These values are:

Speed in miles per hour,	10	20	30	40	50
A. T. F. at maximum cut-off,	25,000	17,000	11,000	6,800	3,000
Ton miles work per hour,	125	170	165	136	75

The last item is the product of the tractive force in tons and the speed in miles per hour, and gives the work performed per hour at the circumference of the drivers. (It must not be confused with ton miles of freight hauled, as it is a quantity of work, pure and simple.) Now, as the amount of water used per hour is the same in each case, viz., 30,000 pounds, it is seen at once that very different quantities of work are performed by the same quantity of steam, or the economy of water consumption varies greatly with the conditions of operation. The "ton miles" item can be converted into available horsepower by multiplying the values given in the table by $\frac{2,000 \times 5,280}{60 \times 33,000}$, and then will show a close resemblance to plate 13, by comparing with the indicated horsepower for similar speeds and notches. It is probable that the curves of Fig. 100 are not entirely correct for speeds above 30 miles an hour, as they were not constructed from actual results of the test, but were prepared hypothetically—the loci from 10 to 30 miles were, however, taken direct from the tests.

In our description of plate 29 it was shown how to determine the speed at which the boiler would cease to supply steam to the cylinders at full stroke—this point gives the maximum speed for the corner notch, and conversely, the full water consumption will occur at this point. For other speeds, the maximum cut-off is obtained by calculating the combination of point of cut-off (or volume) and density due to cut-off pressure (obtained by means of plate 10 and the preceding table of initial pressures at various speeds), which, with the speed in question, will consume the same (total) weight of steam that the boiler can supply. This will give points which correspond to the upper limits of the loci in Fig. 100, and which constitute the maximum cut-offs possible with the different speeds. In making these computations, it must not be forgotten to allow for the cylinder clearance and condensation, the latter as illustrated by plate 14; in other words, the weight of steam drawn off by the cylinders in one hour must equal the capacity of the boiler.

The maximum quantity of water above considered has been based upon a unit of time, viz., one hour. It is often advisable to know the consumption in a unit of distance, as the mile. This will evidently be the maximum hourly consumption divided by the speed, except where it is quite low, generally less than 10 miles an hour, or, in fact, less than the speed at which the boiler will supply the cylinders at full stroke. It is evident that if this speed be 10 miles an hour, and the engine be working at 5 miles, the steam consumption will only be half as great, per hour, because the cylinders can take only their volume at each stroke. Above this limit, where the cylinder draft equals the output of the boiler, however, the full capacity can be made use of. In the locomotive just considered, where the maximum steam capacity was 30,000 pounds per hour, we see from Fig. 100 that at 5 miles an hour the consumption could not possibly exceed 15,000 pounds an hour, or 3,000 pounds per mile; at 10 miles an hour 30,000 pounds, or also 3,000 pounds per mile. At 20 miles an hour, the rate would be 1,500 pounds per mile, and at 30 miles, 1,000 pounds per mile. These are the maximum quantities that could possibly be used by the engine under consideration, as the limits are fixed by the volume of the cylinders at low speeds, and by the capacity of the boiler at high speeds.

INTERMEDIATE QUANTITIES OF WATER.

As stated before, it is seldom that a locomotive is called upon to operate at its maximum capacity for long periods of time—the controlling grade is usually of small length, compared to the operating division, and then the boiler will not be called upon to supply so much steam. It is desirable frequently to be able to figure the water consumption for these periods of lesser activity, and this is the burden of the present section. If we examine the 10-mile curve in Fig. 100, we find that as the cut-off is shortened, the quantity of water is decreased, but not in strict proportion to the former. One thing that accounts for this is the 8 per cent clearance from which the curve starts. Cylinder condensation and change in density of the steam due to variation in cut-off pressure, also help to throw the locus out of a straight line. The available tractive power of the engine does not decrease regularly, as might be inferred from plate 30, so it will be of interest to note how these functions vary with the cut-off. If we take the maximum tractive force at each speed as 100 per cent, and the maximum capacity of the boiler as 100 per cent, and examine for each cut-off under these speeds, the percentage which the tractive force and the water consumption bears to the maximum, we shall obtain a definite idea as to how these quantities change due to the cut-off. These values are given in the following table, the upper quantity being the per cent of maximum tractive force for the speed, and the lower quantity the per cent of maximum steam used per hour. For speeds of less than 10 miles an hour, which was the limit at which the boiler could supply the cylinders at full stroke in this engine, the values (of percentage) will be the same as at 10 miles, because in each case the tractive force would remain the same, but the quantity of water used at any cut-off would be directly proportional to the speed.

RATIOS IN PERCENTAGE OF TRACTIVE FORCE AT RAIL AND WATER CONSUMPTION.

Apparent Cut-off.	Speed in Miles Per Hour.				
	10	20	30	40	50
.1	25 21	30 29	36 39	35 48	27 61
.2	43 32	54 45	67 60	75 75	93 93
.3	55 43	70 60	87 79	100 100	(.22) 100 100
.4	64 52	81 75	99 99		
.5	72 62	91 89	(.41) 100 100		
.6	80 71	(.58) 100 100			
.7	87 80				
.8	93 88				
.9	98 96				
.94	100 100				

We notice in this table that the tractive force is generally a greater percentage of the maximum, for that speed, than the amount of water used, but in many cases the two values are quite close together. We therefore conclude that for approximately estimating the water consumption of a locomotive when not operating at its full capacity, that is, greatest available tractive force for the speed in question, we can safely consider that the amount of water used bears the same ratio to the maximum steam capacity of the boiler that the available tractive force bears to the maximum available tractive force at the speed being discussed; as the latter can be found by means of plate 29 or a similar construction, we thus have a ready means of determining approximately the water consumption at any intermediate power and speed.

In some recent tests made with compound 2—8—2 type engines having 200,000 pounds normally on drivers, and about 5,000 square feet of heating surface, upon a heavy grade division in Colorado, it was found that when engines were so loaded that the maximum tractive force was called into play throughout the trip (as would be the case where the engine was given full tonnage for a grade that was uniform throughout the run) the consumption of water averaged about 600 gallons, or 5,000 pounds, per mile, which, at 10 miles an hour (the average speed) gave a total of 50,000 pounds per hour—10 pounds per square foot of heating surface. When, however, the average grade was about 40 per cent of the controlling grade for which the engines were loaded, requiring (with speed resistance) about half the full tractive power of the locomotive, the consumption averaged only about 300 gallons per mile, or one-half the previous quantity, which gives us a very good check upon our assumptions in the premises. The amounts per mile are the hourly rates divided by the speed. If these figures are needed, however, we should use the percentages shown in the table. That is, for 55 per cent of the maximum tractive force at 10 miles an hour, 43 per cent of the maximum evaporation should be taken. The reason for this is the greater amount of work done by a given quantity of steam when used expansively, as at early cut-off, than when at full stroke with no expansion, about 25 per cent being the amount of this increase. At extremely early cut-off, as 10 per cent, the cylinder condensation again reduces this economy. This will be further considered in connection with the amount of water used per horsepower hour, and which will show for simple engines the greatest economy at about one-third cut-off. In selecting the proper column of the table for locomotives whose boiler capacity permits the operation at full stroke faster than 10 miles an hour, the maximum cut-off for the speed should be considered in preference to the speed heading given in the table. Thus, if a .58 cut-off can be maintained at 30 miles an hour, use the 20-mile column.

WATER PER HORSEPOWER HOUR.

In our study of Fig. 100 we found that the quantity of water (or steam) used per horsepower hour varied between wide limits, depending upon the point of cut-off and the speed of the engine; the maximum quantity of water was fixed by the evaporative power of the boiler, and the work performed varied greatly. In the chapter on steam action, the steam consumption per indicated horsepower hour was illustrated by plates 15 and 15a, and these show clearly the effect of speed and expansion ratio. However, there are other points to be considered, which affect the water rate. The valve and gear, lagging of cylinders, design of ports and passages all unite to complicate the matter, so that it cannot be expected that absolute uniformity can be secured in various types of engines. The Lake Shore & Michigan Southern Railway reported from

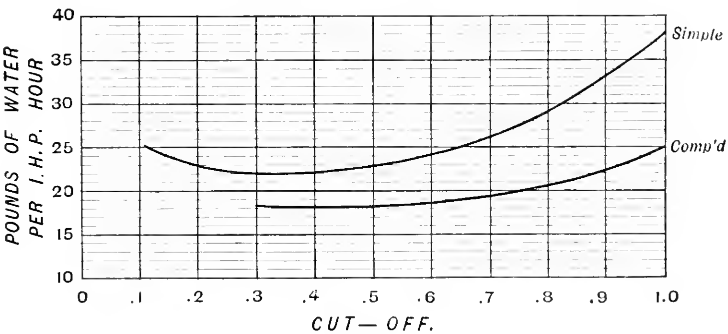


Fig. 101.

some tests with slide and piston valves, that the former used 34.9 pounds of water per horsepower hour, and the latter 31.7 pounds, or 10 per cent more for the slide than the piston valve; however, both of these values are high, as recent tests upon the Furness Railroad in England gave from 20 to 21 pounds per I. H. P. hour for average time that steam was used. With compound locomotives, 18 pounds was thought by the late D. L. Barnes to be the lowest that could reasonably be expected. Fig. 101 is presented to indicate what would be considered very good results in practice; no doubt, that better figures are occasionally obtained, but on the other hand, we have shown that many cases occur where the performance is very much inferior.

That the speed of the engine has an effect upon the water rate is shown by plate 15a, but the discrepancy between the different curves is so great that it is impossible to harmonize them. In fact, the published results of tests to demonstrate this feature are so meager, that no definite law can at this time be stated, and while something in this line might be attempted by calculation, the results of careful tests would be much more reliable and satisfactory. For purposes of estimate, therefore, we believe that Fig. 101 can be used at all speeds, bearing in mind that it is based on indicated horsepower. In the figure, the best rate given for simple engines is 22 pounds per I. H. P. hour and 18 pounds for compounds, or 18 per cent economy. This is somewhat better than the general result of comparative tests, which often indicate from 10 to 15 per cent. At full stroke the compound shows about the same as the single expansion engine at half stroke, and when we consider that the real expansion in this case is about two volumes, we find an explanation for this agreement. The water rate indicated by the figure is not the amount accounted for by indicator, but as measured from the supply to the boiler.

QUANTITY OF WATER EFFECTED BY PRESSURE.

In our discussion above, it has been considered that the quantities of water were based upon a boiler pressure in the neighborhood of 200 pounds per square inch, as this is current locomotive practice, although 225 pounds pressure is in limited use. An increase in pressure always means a gain in economy, as far as the heat cycle of operations in the cylinder is concerned, though there may be other difficulties in the use of high pressure steam, which will more than offset the thermal efficiency. The well-known formula for the efficiency of a heat engine $\frac{T_1 - T_2}{T_1}$, in which T_1 is the absolute temperature of the entering steam, and T_2 the absolute temperature of the exhaust steam, indicates clearly the advantage of high temperature (and consequently high pressure) steam. R. Clausius, in his "Mechanical Theory of Heat," says: "In order to get the greatest advantage from engines driven by heat, the most important point is to increase the temperature interval $T_1 - T_2$."

As the exhaust pressure is fixed within close limits, high pressure steam requires a greater expansive ratio than low pressure steam; in fact, it is in this that we obtain the economy. This also means a greater difference in the temperature of the initial steam and the walls of the cylinder, causing an increase in condensation, which reduces the economy greatly. There are several ways of preventing this condensation, or at least reducing it. One method, that of providing steam jackets, has been given a limited number of trials, without producing very satisfactory results. Another method is by compounding the cylinders, and this is the system most generally used at the present day to decrease condensation, by reducing the difference in temperatures of initial and exhaust steam, and so maintaining a "hotter" cylinder, as compared with the temperature of the incoming steam. The economy of compounds compared with simple engines has been discussed in the last section, but it is well to remember that this type of engine permits us to use high pressure steam with less thermal loss than the single-expansion locomotive. As stated before, however, there are other points to be considered than the mere gain in thermal efficiency.

In 1898 Prof. Goss presented to the Master Mechanics' Association a statement of the theoretical quantity of steam required per indicated horsepower hour for a perfect engine, operating against 1.3 pounds per square inch back pressure, and furnished with steam at various pressures above the atmosphere. These values are given below:

Boiler (gauge) pressures..	50	100	150	200	250	300
Pounds of steam per indicated horsepower hour..	26.0	19.5	16.5	15.0	14.0	13.3

Of course, these values are never obtained in practice. Several years ago the Caledonian Railway made some tests to determine the most economical pressure in a special passenger service. The engines used for this purpose had the following proportions:

Size of cylinders	18 by 26 inches
Diameter of drivers (four in number)	78 inches
Heating surface	1,202 square feet
Grate area	19.5 square feet

The pressures tested were 150, 175 and 200 pounds per square inch, and the results showed 10.94 per cent more steam used per horsepower hour at 175 than at 200 pounds, and 22.45 per cent more at 150 than at 200 pounds.

As the pressures are increased, however, leaks of all kinds become more numerous, and also expensive, as they occur under a higher pressure. It is notoriously more difficult to keep flues, staybolts, etc., tight as the pressure is increased. Cylinder lubrication is harder to accomplish properly, and the packing of piston rods and valve stems requires constant attention, and even then, the results are usually anything but satisfactory. Under these conditions it is easy to comprehend how some or all of these troubles can more than overbalance the saving of high pressures.

SUPERHEATING.

The third method of reducing cylinder condensation is by superheating the steam, and when it gets into the cylinder, this excess heat is given up before condensation commences—if the superheat be high enough and the cut-off not excessively early, there may even be no condensation, but the expansion will merely reduce the temperature to that of saturated steam. We have seen that cylinder condensation causes great losses under certain conditions of working—conditions that would otherwise be conducive of economy, as, for instance, increasing the difference of temperature, $T_1 - T_2$, so that if this could be avoided, there would be a double gain.

Superheated steam also effects an economy by reason of its increased volume, although it requires more heat to effect the change in volume, of a given weight of steam, but the increase in volume is in much more rapid proportion than the increase of heat, as the greater portion of the latter has been absorbed in evaporating the water, and has gone into latent heat.

All the tests of superheaters on locomotives show a much greater saving in water than in fuel; in other words, the engine economy is increased while the boiler efficiency is decreased, due, of course, to the additional heat required per pound of steam, which would be expected to show a reduced rate of evaporation in a locomotive, where the superheater often is so located that it deprives the water in the boiler of a number of heat units which would otherwise be available for the generation of steam.

Two types of superheaters have been giving considerable service, the Schmidt and the Pielock. The former is made in two ways; one with a nest of small tubes or pipes concentrically arranged in several rows in the bottom of the smokebox, and heated by means of a special flue about 8 inches in diameter, allowing fire from the firebox to pass forward and heat the pipes, through which the steam is made to pass on its way from the throttle to the cylinders; the other by means of loops of small (about 1 inch) pipes extending backward through several rows of large flues (5 inches diameter) in the upper portion of the boiler, the steam passing through these on its way to the steam chests. The first form of the Schmidt superheater is used on the Prussian State Railway; the second form on the Canadian Pacific Railway. Four locomotives on the Prussian State so equipped gave such satisfactory results that a large number of new engines were fitted up in the same way. The heating surface of the superheater amounted to 300 square feet, the heating surface of the boiler being 1,140 square feet. The compound engines against which the superheater was tested were of the two-cylinder type—in the simple engine the steam at 170 pounds pressure was delivered to the cylinders at a temperature of about 825 degrees Fahrenheit—that is with 450 degrees of superheat. The results of a nine days' trial in express train service showed 25 per cent economy in water consumption over the compound engines, and 10.5 economy in fuel. On the Canadian Pacific, the simple engine with superheater made an average saving in fuel of 31 per cent over the simple and 10.6 per cent over the compound engines (two-cylinder) with which it was tested.

The Pielock superheater consists of a cubical box, placed in the center of the boiler, directly under the dome, and forming a water-tight compartment about the flues. The steam is taken in at the top of the box and after being led in a winding path about the flues is delivered superheated to the throttle valve in the dome. This is also being tested on the Prussian State Railway. The heating surface of the boiler is reduced by the amount in the superheater—in the case under consideration amounting to 226 square feet, the total heating surface of the boilers being about 1,300 square feet. In the tests reported by Herr Strahl to the Association of German Engineers, with a boiler pressure of 170 pounds, feed water temperature of 50 degrees Fahrenheit, and a temperature (Fahrenheit) of 500 degrees in the dome for the simple engine and 446 degrees for the two-cylinder compound, with the superheater, the saving in water and coal amounted to 16 and 12.3 per cent, respectively, for the simple engine and 10 and 3.5 per cent for the two-cylinder compound when compared with the same size of engines without the heater. In these tests the usual simple flat slide valves were retained, and no trouble was experienced with the temperatures used; in the Schmidt system, however, where the superheating was much greater, special piston valves and forced lubrication were applied.

The engines being tested took turns in hauling the same train, exchanging every day, and the average results from the runs considered reliable were used for the comparisons. In comparing the volumes of steam used by the cylinders in the different trips, it was found that practically the same volume of steam was used in the locomotive with a superheater as in the locomotive without, and that the saving in steam corresponded to the increased specific volume given by the superheating; it was also found that the economy depended only on the superheating, and therefore was the same for the same degree of superheat whether compound or single expansion locomotives were compared, assuming, of course, that locomotives of the same class and type were compared with each other.

From the above tests and remarks, it follows that the same volume of steam did the same amount of work in the cylinders, whether it was saturated or superheated. Theoretically the expansion curve of superheated steam drops more rapidly from the cut-off point than does the adiabatic expansion line of saturated steam, but the greater cylinder condensation of the latter practically reduces this curve so that it is nearly identical with the former.

With the foregoing statement of the facts of the test, it is easy to define the economy in water which should be expected from any degree of superheating, providing that we know the rate of expansion or increase in volume due to the superheating. The expansion of dry or superheated steam follows very nearly the same laws as perfect gases, and the volumes of such gases, at constant pressure, have been found to vary as the absolute temperatures to which they are subjected, the unit volume being considered at the melting point of ice, 32 degrees Fahrenheit, or $32 + 461 = 493$ degrees absolute Fahrenheit. Thus, if

v_0 = the volume of 1 pound of gas at 32 degrees Fahrenheit, or t_0 degrees;

v = the volume of 1 pound of gas at another temperature t Fahrenheit.

We have from the above law the equation $\frac{v}{v_0} = \frac{461 + t}{461 + t_0}$, and if v' and t' be any other greater corresponding volume and temperature, we can also write

$$\frac{v'}{v} = \frac{461 + t'}{461 + t} \qquad (111)$$

If, as stated, the saving in steam corresponded to the increased volume $v' - v$, the economy will be represented by $\frac{v' - v}{v} = \frac{v'}{v} - 1$, when we let v and t be the volume and temperature (Fahrenheit) of 1 pound of saturated steam and v' and t' the same for 1 pound of superheated steam. Thus, in the test reported, with saturated steam at 170 pounds gauge, the temperature $t = 375$ and the superheated temperature $t' = 500$ degrees, we have $\frac{v'}{v} = \frac{461 + 500}{461 + 375} = \frac{961}{836} = 1.15$, or a saving of 15 per cent; the actual saving reported was 16 per cent, the increased amount being due, no doubt, to cylinder condensation being largely avoided with superheated steam.

With formula 111 as a guide, it is easy to construct a table showing what economy in water could be made with various amounts of superheat and at different pressures, the table giving this data for 175, 200 and 225 pounds boiler pressure and temperatures ranging from 400 to 800 degrees Fahrenheit.

WATER ECONOMY OF STEAM HEATED TO t′ DEGREES, COMPARED TO SATURATED STEAM AT A NORMAL TEMPERATURE OF t DEGREES.

Pressure.	175 Pounds.		200 Pounds.		225 Pounds.	
Sat. Temp. t.	377° Fahrenheit.		388° Fahrenheit		397° Fahrenheit.	
Sup. temp. t′.	Superheat.	Saving.	Superheat.	Saving.	Superheat.	Saving.
400	23	3 %	12	1.5%	3°	0.5%
450	73	9	62	7.5	53	6.
500	123	15	112	13.	103	12.
550	173	21	162	19.	153	18.
600	223	27	212	25.	203	24.
650	273	33.	262	31	253	29.5
700	323	39.	312	37.	303	35.
750	373	45	362	43.	353	41.
800	423	51.	412	48	403	47.

As the temperatures due to superheating are raised, difficulties are encountered which may prevent a full realization of the economy indicated—radiation losses will be greater and lubrication rendered more difficult, whereby leaks past the pistons and valves may occur through cutting of the packing rings, etc., all of which will reduce the saving in steam used.

Superheating may be attained by generating steam at one pressure and wiredrawing it down to a lower pressure before admitting it to the cylinders. It can readily be demonstrated that such a proceeding is not a rational one for a locomotive. Suppose that we generate steam at 300 pounds and operate the pistons at 200 pounds pressure. The total heat in 1 pound of steam at 300 pounds pressure is 1,210 heat units from water at 32 degrees, and in 200 pound steam, 1,200 heat units. In reducing the pressure (provided no work is performed) there will be 10 heat units per pound available for superheating, and as the specific heat of dry steam is .48, we have $\frac{10}{.48} = 21$ degrees of superheating. By the table we find that the saving in water would be only about 3 per cent, and we know that the saving in fuel would be still less, which gives little gain for the great increase in boiler pressure and its attendant difficulties.

WASTE OF WATER.

The water necessary for the movement of the pistons in the cylinders is often augmented by wastes of various kinds. Many of these wastes are due to improper care of the engines, either on the road or when in the house; others are due to the water which is used, and others still are probably chargeable to high steam pressures and forced rates of evaporation. When the latter are combined with poor water, it is impossible to escape trouble, and while roads using good water have little increase of these difficulties with high pressures, poor water combines to cause large increase of leaks in shell, flues, mud rings, crown bolts, staybolts and from cracks in sheets. In a recent trip over one of the transcontinental lines, on some divisions it was a rare occurrence to find an engine that was not leaking. It cannot always be said that such leaks are due entirely to poor maintenance, as the water and fuel may be of such a nature that expert work on the boilers every trip will not keep them tight. On the Arizona division of the Santa Fe, the engines running west out of Needles obtained quite good water—those running east, water that was materially worse. A large force of boilermakers had to be maintained to keep the east end engines in service, and when they got in such shape that they could not climb the grades without causing many failures, they were transferred to the west end, and would run satisfactorily for several months. The installation of a water-treating plant (Kennicott) at the most important supply station on the eastern district changed all this at once—the engines can now be run indiscriminately east or west, and the force of boilermakers was more than cut in half.

Steam blows are common sources of leakage; when it escapes into the atmosphere from the piston and valve stem packing, it manifests itself very clearly, and with high pressures there is more of this manifestation than is desirable. Besides a waste of steam, there is a great obstruction of the view ahead to the enginemen, making it dangerous to operate the engine. Any mechanism which induces or permits an unusual amount of vibration of the piston rod is sure to cause trouble with leaky packing; therefore the guides should be kept well closed to prevent vertical movement of the crosshead. Piston valves with inside admission also prevent blowing around the valve stem. Blows that occur inside the cylinder past the packing rings of the piston or in the steam chest by the valve do not make themselves visible, but can generally be detected by the sound of the exhaust. Most roads using compound locomotives, where the process of locating a blow is more difficult than in a simple engine, furnish their engineers with detailed instructions regarding the detection of this kind of a leak. The method generally consists in blocking the engine in different positions of the crank and noting the presence or absence of steam escaping from the cylinder cocks or stack. Careful engineers will do this periodically, without waiting to detect the blow by ear from the sound of the exhaust.

Priming is another serious source of waste. Waters containing soluble salts in quantities of more than 30 grains to the gallon generally give trouble from foaming. Such waters have to be carried very low in the glass, frequently going entirely out of sight, and when the engine is working hard, the foaming tendency induces the formation of large steam bubbles or clots apparently next to the firebox sheets, causing overheating, and subsequent cracking when the water returns. This priming is frequently so serious that the whistle cannot be sounded without closing the throttle, in order to reduce the level of the water in the boiler. Water is carried over into the cylinders, cutting the valves and seats, and the piston packing and cylinders. If the case is sufficiently aggravated, the cylinder heads may be broken, but if not, it is almost sure to destroy the piston rod and valve stem packing, causing great leaks and serious wastes of steam.

Injectors that do not take up the overflow, waste water, but as this is only slightly heated, the fuel waste is small—the other leaks above noted all represent considerable fuel consumption in raising the water to the boiling point.

Safety valves often waste steam when the manipulation of the fire is done in a careless manner. This occurs mostly when the engine is standing on a sidetrack, or running down hill. The amount of steam blown off by a 2½-inch safety valve in one minute represents the evaporation caused by burning 15 pounds of coal or, in round numbers, is equal to 100 pounds of water, all of which has been evaporated at the pressure of the boiler. If each engine on a road having 1,000 locomotives blows off at the rate of 10 minutes a day only, the total is equivalent to 1,000,000 pounds of water and 150,000 pounds, or 75 tons, of coal wasted per day! This can all be prevented by the proper manipulation of the injectors, dampers and firedoor.

It is clear from the foregoing that the waste of water may be very great, and that most of the waste will be of steam, which represents fuel as well. While these various leaks cannot be even approximately estimated as to their amount and economical value, yet all parties concerned should be interested in seeing that they are reduced to a minimum.

WATER SCOOPS.

In connection with our study of the quantity of water used by locomotives, it is interesting and profitable to consider the action of water scoops, by which the necessary water is taken while running. For a number of years this was employed only in fast passenger service, but recently, fast freights have also been equipped for the same purpose, thus saving the time and expense of stops for water. In construction, there is quite a difference in the details of this mechanism, as adopted by different roads, but the general arrangement is the same, and too well known to need specific illustration. The conditions of operation can be discussed from a mathematical standpoint, but as there are so many uncertainties in obtaining exact functions, a very close correspondence with actual tests cannot be expected. In Fig. 102 let us consider that

v = speed of engine in feet per second;

v_1 = velocity of water into lower end of scoop, relatively to the tender, in feet per second;

h = height to which water must be elevated, in feet;

a = dip of scoop, below surface of water, in feet;

b = width of scoop, in feet;

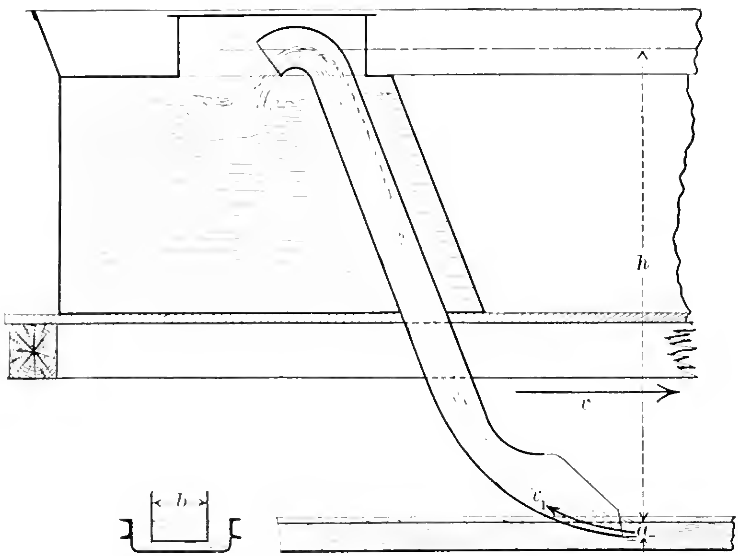


Fig. 102.

f = resistance coefficient, for entry and friction of water in pipe;

Then we have the total head, due to the velocity of the engine $\frac{v^2}{2g}$, and this evidently equals the sum of the head due to velocity of water at point of influx, $\frac{v_1^2}{2g}$; plus the friction head, $f\frac{v_1^2}{2g}$; plus the height which the water must be lifted, h;

or the velocity of influx, v_1, is dependent upon the difference of the opposing heads, so that we can write

$$\frac{v_1^2}{2g} = \frac{v^2}{2g} - f\frac{v_1^2}{2g} - h \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (112)$$

As the mouth of the scoop is quite sharp, the entry head will be nearly unity, and as the pipe is large and short, and may not run full, the friction of the water will be small, both features combining to make the value of f quite inconsiderable. Moreover, there is always a wave formed ahead of the scoop, increasing the actual height of dip a, so that for approximate purposes we can ignore the quantity $f\frac{v_1^2}{2g}$. This permits us to write equation 112, $\frac{v_1^2}{2g} = \frac{v^2}{2g} - h$, or transposing, $v_1^2 = v^2 - 2gh$, and finally

$$v_1 = \sqrt{v^2 - 2gh} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (113)$$

In applying this formula to road conditions, it will be necessary to remember that

$v = 1.466\,V$ and $v^2 = 2.15\,V^2$, where

V = speed in miles per hour.

In 1901 the New York Central made a series of tests to determine the quantity of water actually taken at various speeds, the tender scoop having a height h of 9 feet and a width b of 1 foot, the dip a, or immersion of scoop below the surface of the water averaging 3½ inches at 30 miles an hour and 4½ inches at 40 miles. The amount of water actually taken in a distance of 1,200 feet, the track tank being 1,400 feet long (allowing thereby 100 feet at each end for lowering and raising the scoop) averaged 2,300 gallons at 30 miles and 3,200 gallons at 40 miles an hour. Below 15 miles an hour water could not be taken with any regularity.

Now, applying formula 113, we have for 30 miles an hour, $v_1 = \sqrt{900 \times 2.15 - 2 \times 32.2 \times 9} = \sqrt{1{,}935 - 580} = \sqrt{1{,}355} = 37$ feet per second, and the quantity of water taken per second is $\frac{3.5}{12} \times 1 \times 37 = 10.8$ cubic feet, or $10.8 \times 7.5 =$

81 gallons. At 30 miles an hour the speed is $30 \times 1.466 = 44$ feet per second, so that the scoop is in the water for $\frac{1,200}{44} =$ 27.2 seconds, and the water taken will be $81 \times 27.2 = 2,200$ gallons, 100 less than by test, or an error of about 4½ per cent.

At 40 miles per hour we obtain

$$v_1 = \sqrt{1,600 \times 2.15 - 2 \times 32.2 \times 9} = \sqrt{2,860} = 53.5,$$

and therefore $\frac{4.5}{12} \times 53.5 \times 7.5 \times \frac{1,200}{40 \times 1.466} = 3,100$ gallons taken, 100 short of the actual results.

To find the minimum speed at which any water can be taken it is only necessary to put equation 113 equal to zero, thus $v_1 = \sqrt{v^2 - 2\,g\,h} = 0$, so $v^2 = 2\,g\,h$, and

$$v = \sqrt{2\,g\,h} \qquad (114)$$

which being solved for $h = 9$ feet gives us

$$v = \sqrt{2 \times 32.2 \times 9} = \sqrt{580} = 24 \text{ feet per second, or}$$

$\frac{24}{1.466} = 16.4$ miles per hour, which also checks with the results of the tests.

QUALITY OF WATER.

Very much could be written upon this important subject, and still leave it incomplete. While the matter should be studied from a chemical standpoint, yet the results from using a variety of waters are so largely and intimately connected with locomotive operation, it is felt that the treatise would not be complete without a few general statements regarding them.

In a general way, we may form five classifications of boiler waters, as follows:

1. Practically pure.
2. Forming soft scale
3. Forming hard scale.
4. Corroding.
5. Foaming.

In the first class may be considered all waters that have not any of the characteristics of the other classes. They may contain sewage or other matter that would make them unsuitable for culinary or drinking purposes, but as they would not trouble the boiler, they would be practically pure for this purpose. While not generally available throughout this country, yet in certain regions, principally in lake districts, where there are many fresh ponds, as in northern Wisconsin and Michigan, in the territory of the great lakes, and in mountain regions where the streams formed from the melting snows of winter do not percolate and dissolve the soil, water practically pure is found in abundance. While Lake Michigan water has some scale-forming substances, yet it is very good when compared with much of the water available for locomotives. Boilers using such waters will give little trouble, if properly made, either during operation or when in the roundhouse or shops; washing out will cause little delay at terminals, and flues and fireboxes should last from five to 15 years. There is probably no one thing that would be appreciated so much by the motive power and transportation departments of a railroad cursed with bad water as the advent of practically pure water, were it possible.

In the second class may be placed those waters which form a soft scale in the boiler, this being due to the presence of the carbonates of lime and magnesia, one or both. These materials are freely soluble in water which contains carbonic acid gas in solution. When such water is boiled the carbonic acid is driven off by the heat, and the material is deposited on the inside of the boiler. The carbonate scales are not hard, but quite bulky, as the proportion of water of crystallization is large. This is the white, chalky matter that is washed out of boilers when in the roundhouse. When examined on the surface of flues and firebox, it is thick, but quite easily removed. It is a very poor conductor of heat—in fact, one of the best boiler laggings on the market is largely formed of carbonate of magnesia. Unfortunately, it forms naturally on the inside of the boiler, where it prevents the transmission of heat to the water, instead of outside, where it would be desirable to prevent the transmission of heat from the water.

The third class embraces principally the sulphates of lime and magnesia, which form a very hard scale on the inner surfaces of the boiler; sometimes this scale is as hard as porcelain. The deposit is heavy and smooth—it is not thrown down until the temperature of the water is about 300 degrees Fahrenheit, when it unites with the other deposits, forming a hard cement-like coating on flues and firebox. It is also a poor conductor, causing a waste of fuel, and it is very difficult to remove it from the surfaces to which it attaches itself.

Besides the prevention of heat transfer to the water, the scale permits the sheets, flues, etc., to become overheated for the same reason, and this causes numerous leaks and engine failures. The scale may, if not properly removed by washing, become so thick as to permit the sheets to become fire-cracked, owing to the non-transfer of heat to the water. The only way in which any kind of service can be obtained when using such waters in an untreated or natural condition, is by frequent and constant washouts—sometimes every round trip. This in itself is not only an item of expense, but the delay to the power is a great drawback, as it requires a layover of at least six hours, and sometimes more, to effect a careful cooling down, washing out, and firing up again; at the best, this process is detrimental to the life and tightness of the boiler.

The fourth classification includes waters that contain free acid, generally sulphuric from the drainage of coal mines, etc., or carbonic acid in solution, which often attacks the metal; also the chlorides of lime and magnesia, the latter being much more active in this manner than the former. The results are pitting and eating away of the sheets and tubes, it not being uncommon to have holes appear completely through sheets. In many respects, this is the most dangerous of the several different troubles existing, as it is liable to attack any part of the boiler, and often centralizes upon those parts most difficult of inspection, so that the metal is reduced to a dangerous thickness before it is discovered. Boilers using such water require the most careful and frequent inspection. The presence of free acid in the water generally makes itself known by the dark red color of the liquid that runs from the water leg when the plugs are withdrawn. The vegetable acids, like tannin, are not very troublesome, and may even neutralize the effect of scaling waters, when mixed in the same boiler.

From an operating standpoint, the fifth group is more troublesome than any of the others, and it is also the most difficult to cure. Foaming causes broken cylinder heads, broken pistons, rings and valves, blowing packing, both on piston rods and valve stems, as well as the piston packing itself, besides badly cutting cylinders and valve seats. At times, boilers have been known to foam so badly that the whistle could not be blown without closing the throttle. In addition to this, there is the waste of water and heat which is carried over with the steam, and which is useless for performing work. Then again, the water must be carried very low in the glass, especially when working hard; in climbing a heavy grade, it is not uncommon with foaming waters, to have the water disappear entirely from the gauge glass; when the summit is turned, the water runs ahead, the throttle is closed, allowing it to settle, and the danger of burning the crown sheet is greatly increased. While the actual effect upon the boiler itself is not as serious as the other troubles enumerated, yet it constitutes a very great drawback to satisfactory operation.

Foaming is caused principally by matter in solution or suspension—and the materials in solution are generally sulphate of soda and chloride of sodium and calcium. In rainy seasons, the mud and suspended matter in turbulent waters increase the foaming, but not to as great an extent as the solubles mentioned. While stationary boilers can often use water containing as much as 60 or 80 grains of foaming matter to the gallon without causing serious trouble, with locomotives the motion causes a constant churning and increases the tendency to foam, so that if the content be over 30 grains per gallon we are quite sure to have trouble. Foaming often indirectly causes the rapid destruction of firebox sheets. One case in the writer's experience may be quoted. A large passenger engine, whose boiler contained 3,700 square feet of heating surface, had its firebox completely ruined in one year, after making about 60,000 miles. The upper and central portions of the sidesheets were covered with innumerable cracks, seemingly started from the water side. While the scale was not at all heavy, the water had given much trouble by foaming. Experiments indicated that when working hard, with a bright fire, there were times when a film of steam, ¼ to ⅜ inch in thickness, separated the water from parts of the sheet, particularly at the central portion, where the activity of the fire was greatest. From this it was easy to diagnose the cause of the cracks in the sheet—it became overheated when working hard, even though the water level was a foot or more above the damaged portion, by the temporary absence of water, and when the latter did return, it cooled the sheet suddenly, and in time this process caused the numerous cracks observed. In the case mentioned, the water space was not restricted, being 4 inches at the mud ring, and greater above.

In order to illustrate these remarks practically, we give below the analysis of a sample water in each group—of course, these cannot be representative of all waters in each group, but merely show how the troubles are caused. In each column, a star (*) precedes the quantity of the ingredient that causes the trouble indicated in the heading of the column.

ANALYSIS OF WATERS IN GRAINS PER GALLON.

Group numbers	1	2	3	4	5	2, 3 and 5
Matter in solution	Pract. Pure.	Soft Scale.	Hard Scale.	Corrosion.	Foaming.	Scale and Foaming.
Lime carbonate	1.15	*9.61	11.37	4.37	2.28	*42.75
Lime sulphate	0.78	3.03	*23.50	11.08	0.37	*42.59
Lime chloride				*4.37		
Magnesia carbonate	1.11	*7.80	1.46	0.70	0.62	*19.01
Magnesia sulphate			*17.15		0.20	*32.10
Magnesia chloride		0.23		*2.51		
Soda carbonate					*2.85	
Soda sulphate	0.56		14.34		*46.19	*62.05
Soda chloride	0.01	0.36	2.22	60.65	*1.40	*1.21
Total incrusting	4.03	20.76	53.29	24.31	8.12	137.90
Total non-incrusting	1.21	0.59	16.56	60.65	52.98	62.05
Locality	Wis.	Ill.	Cal.	Ariz.	Wyo.	Minn.

By the totals given, it will be noticed that a few unimportant materials have been omitted in the tabulation. No. 4 will foam as well as corrode, due to the large quantity of common salt in solution.

TREATMENT OF WATER.

After making a short review of the troubles caused by impure water, the next point is naturally, "What can be done to reduce these evils as much as possible and avoid the delays and expenses of operation incident thereto?" In each case the analysis of the water must be the deciding factor in framing an answer. "Washout experts" have contended that by extreme care—so difficult to enforce among the exigencies of a congested traffic—boilers could be maintained for a long time in service, but the delays due to holding the engine for such careful washouts would sum up, perhaps, more time actually out of service than would ordinarily be imagined, actually causing an increase in capital expenses by requiring more locomotives to fill the temporary vacancies. Washing out is important, and should always be thoroughly and conscientiously done, but it is not the only thing that should be done. When the scale bakes and hardens on the tubes and sheets it is extremely difficult to dislodge, even with a strong stream of water, and then there are so many parts of the boiler which cannot be well reached or inspected.

In some of the groups mentioned chemical action will change the quality of the water from one class to another. This was formerly done in the locomotive boilers or tenders, and while it is better to do it there than not do it at all, yet there is very much left to be desired by such treatment. The softening of water consists in changing the matter in solution from one chemical combination to another, and by this change there is a mass of sediment and deposit formed in the boiler; while this may not form scale, it does form a sludge, and requires constant blowing off on the road and washing out in the house. If we can prevent this material going into the boiler at all, we save that much delay in cleaning the boiler between runs.

The waters of group 1 need no treatment, as they will cause no trouble in the boiler.

Those of group 2 are best treated by dosing them with slaked lime in solution; when this is added to water containing carbonate of lime or magnesia it unites with the carbonic acid in the water, and which is the cause of solubility of the carbonates, forming carbonate of lime, which being insoluble in the water freed from its carbonic acid gas, settles down as a white precipitate. The carbonates held in solution by the carbonic acid gas are thrown out of solution as soon as the gas is removed by the slaked lime, and joined the precipitate formed by the slaked lime. The clear water may be drawn off and safely used for boiler purposes. Where the water contains only carbonates as incrustants this treatment transfers it from group 2 to group 1.

In group 3 the question is not so easily settled. The treatment is to use soda-ash or carbonate of soda in solution. This starts a chemical reaction by means of which the sulphate of lime (or magnesia) and carbonate of soda react and form carbonate of lime and sulphate of soda. The carbonate of lime now settles as in group 2 when the free carbonic acid was removed from the water, but the sulphate of soda remains in solution and passes into the boiler. If the solution be strong, or if it become strong by concentration in the boiler, it causes foaming, and the only remedy is to blow off frequently while the engine is hot, and to change the water between trips, so as to keep down the strength of the solution. This means a great loss of water, as the boiler must be filled up fresh, and also refilled to make up for the blowing off on the road. If done while running it is very hard on the paint of the tender and cars that closely follow it. Thus we see that treatment of class 3 is apt to throw the water into class 5.

These two methods of treatment are represented by two cases in western Iowa, in which the principal ingredients are given below:

Matter in Solution.	Group 2.		Group 3.	
	Before.	After.	Before.	After.
Lime carbonate	*14.68	2.16	*21.39	2.26
Lime sulphate	3.60	....	*6.22	....
Magnesia carbonate	2.51	2.05	1.18	0.88
Magnesia sulphate	1.20	....	*13.33	....
Soda carbonate	....	1.22	....	....
Soda sulphate	....	7.77	5.58	*26.32
Soda chloride	1.15	2.38	1.21	1.27

The * denotes the troublesome ingredient in each case. While the first water becomes fairly pure by the treatment, the second is transferred from a scaling water to a foaming one, and if the original amount of soda sulphate had been much greater the water would be worse after treatment, as far as the operation of the engine was concerned, than before.

Mr. Howard Stillman, engineer of tests of the Southern Pacific Company, gives the following very sound advice:

"Do not ordinarily attempt to treat a water containing less than 12 grains per gallon of total matter classed as incrustating unless of an unusually corrosive nature such as the unstable chlorides of lime and magnesium."

"It is not commercially profitable to treat a water if the total alkalies (salts of soda or potash), naturally contained and resultant, exceed 30 grains per gallon."

It is a well-known fact to locomotive engineers that treated water is sometimes more troublesome than untreated, due to the increased tendency to foam. This was experienced on the Western Division of the Santa Fe, where the scaling waters of Colorado were changed to foaming, with its consequent delays and engine failures.

Waters of the fourth class can generally be neutralized by adding lime, soda ash or some other alkaline material. As the quantities of such corrosive matter are usually small, there is not so much likelihood of inducing foaming troubles. It is absolutely necessary, however, that some treatment be given, otherwise the boilers will be ruined. The tendency of the treatment is to throw the water from class 4 to class 5.

Group 5 includes the waters that give us the greatest trouble in operation, and which, we have seen, is also likely to include many waters after going through the expense of treating them. It is important that this should be considered before such treatment is started. As to remedying this trouble there seems at present to be but one way of accomplishing it; that is, by distillation. The materials that cause foaming, usually the salts of soda and potash, termed "alkalies," remain in solution, and cannot be removed by any inexpensive chemical treatment. As with sea water (which contains about 2,200 grains to the gallon, 1,700 of which are sodium chloride) the only way in which it can be made usable for boilers is by distillation, and this is used in the navies of the world in order to replenish the waste from the boilers. This process has generally been branded as prohibitive from a cost view, but an apparatus has lately been produced by Prof. Goss which bids fair to prove this a fallacy. With a model it has been found possible to obtain from 60 to 80 pounds of water with the heat generated by burning one pound of coal. With the latter at one dollar a ton, the cost would run about 13 cents per thousand gallons, and there are many localities where fuel can be had for much lower figures, a case recently coming to the author's notice where 25 cents a ton would cover the fuel cost. This would put the distilled water down to about 4 or 5 cents per 1,000 gallons, an amount not greatly in excess of many of the treated or softened waters of the day.

The above remarks are not intended in any way as a criticism of the several very desirable methods of treating water now before the public, but they are intended as a caution to those who expect a speedy relief from all boiler troubles upon the completion of their plants, without taking steps to prevent simply a correlation of trouble. Each particular water must be analyzed and studied, and the general results of one railroad cannot be assumed to be applicable to another, without a complete understanding of the detail conditions existing in both cases.

As an illustration of this point, at the 1903 convention of the Master Mechanics' Association, it was stated that by means of water treatment on one of the important western roads, the district that was formerly the worst on the system, from a water standpoint, had become the best, and that where formerly flues lasted only from three to six months, and boilermakers were called upon to work on the engines every time they came to a terminal, they were now getting from 18 to 24 months' service from flues, and that the boilers gave no trouble whatever. When we examine the analyses of the waters of this district, however, we find that the incrusting solids are almost entirely carbonate of lime and magnesia, many of the waters containing no sulphates, or at the most 3 or 4 grains to the gallon. There is no reason why this water should not be satisfactorily treated. In a similar district, where the sulphates constituted a large number of grains to the gallon, we would certainly experience trouble from foaming, as was cited above.

In February, 1903, Mr. G. M. Davidson presented, in a paper before the Western Railway Club, some tabulated data on water treatment, which are produced here, with some alterations and enlargement. These are designed to show briefly the various kinds of water with the troubles that follow its use in the raw state, the method of caring for the boiler if the water be not treated, the logical treatment to be provided before delivering it to the tender, the possible trouble that may be expected after this treatment, and the care of the boiler to obviate this secondary trouble. It is, of course, assumed that the quantities in solution are sufficiently great to cause the annoyances stated:

EFFECTS OF IMPURE WATER IN BOILERS.

Group No.	1	2	3	4	5
Trouble	None	Soft scale	Hard scale	Corrosion	Foaming
Cause.	Practically pure	Lime carb'nt Magnesia "	Lime sulphate Magnesia "	Acids, Chlorides	Alkali, Mud
Care of boilers	Ordinary	Through washouts frequently	Through washouts frequently	Close inspection frequently	Blow out and change water frequently
Remedy	None needed	Slaked lime	Soda ash with or without slaked lime	Slaked lime or soda ash	Distillation, alum
Possible after trouble	None	Should be none	Foaming	Foaming	Some corrosion if all distilled water be used
Boiler treatment	Ordinary	Ordinary	Blow out and change water	Blow out and change water	Mixture with other waters

CARE OF BOILERS.

From what has preceded, it is apparent that systematic and intelligent care of locomotive boilers is of the greatest importance, and that in territories afflicted with poor water, it is doubly important. On some roads in the eastern part of this country, boilers are washed out only once a month—in some portions of the West they are washed out every other day and water changed the alternate days. This means a large waste of water, if it is proper to call it such, but it seems inevitable under the existing conditions.

The proper blowing out by the engineer is important, in order to prevent undue concentration of the material in solution. Some roads prescribe that this blowing off is to be done while running, and others at terminals. If it be desired to get rid of sediment or sludge, such as mud, soft scale, etc., the blowing off should be done at terminals, after the water has had a chance to settle somewhat; if, however, it is simply concentration of the solubles, then it can be done with advantage on the road. Care and intelligence on the part of the engineer in drawing a full tank from the good water stations, and either running or taking a small quantity from the poor sources of supply, will greatly aid the work of caring for the boilers.

The manner of cooling down and washing out is of much importance. Often this work is done hurriedly under pressure from the dispatcher, who is in haste to get a train off, and the boiler suffers in consequence. Then, again, the labor employed upon this work is often underpaid and unintelligent, and good results cannot be expected. The following extracts are taken from the "Instructions for Boiler Washers" in use on the Santa Fe System:

"Boilers should be thoroughly cooled before being washed, when time will permit. When they are cooled in natural way without the use of water, the steam should be blown off, but the water must be retained above the top of crown sheet and boiler allowed to stand until the temperature of the steel in the firebox is reduced to about 90 degrees, or so that it feels cool to the hand; then draw off water and wash. When the engine cannot be spared from service sufficiently long for it to be cooled in this manner before washing, proceed as follows:

"When there is sufficient steam pressure to work it, start the injector and fill the boiler with water until the steam pressure will no longer work the injector. Then connect water pressure hose to feed hose between engine and tender, and fill boiler full, allowing the remaining steam pressure to blow through syphon cock or some other outlet at top of the boiler. Open blow-off cock and allow water to escape, but not faster than it is forced in through the check, so as to keep the boiler completely filled until the temperature of the steel in the firebox is reduced to about 90 degrees; then remove all plugs and allow boiler to empty itself.

"Begin washing flues by side holes of boiler opposite front end of crown sheet. Wash top of crown sheet at front end, then between rows of crown bars (when so provided) and bolts, directing stream towards back end of crown sheet. After washing through holes near front end of crown sheet, use holes in their respective order toward the back of the crown sheet. This is to work the mud and scale from the crown sheet toward the side and back legs of the boiler and prevent depositing it on the back end of flues. Next wash crown sheet from boiler head, using the swivel connection with hose and right-angled nozzle, inserting to the front end of crown sheet, and slowly drawing back and revolving it at the same time, so as to wash top of boiler and all radial stays or both, as well as crown sheet. (This refers to radial stayboxes.)

"Then wash back end of flues through holes in connection sheet, and afterwards water space between back head and door sheet through holes in back head, with angle nozzle. Inside arch flues should also be washed thoroughly from the back head and scraped with proper form of scraper.

"Now wash through holes near check valves at front end of boiler, using straight and angle nozzles, with swivel connection, and then wash through holes in bottom of barrel near rear end, using the straight nozzle directly against the flues, reaching as far as possible in all directions. Then use the bent nozzle through front hole in bottom of barrel and also straight nozzle in same manner, to clean flues and space between flues and barrel.

"If there are washout plugs in the front flue sheet, the washing through them should be done before washing through the bottom holes of barrel, and should be done by means of a long pipe nozzle of sufficient length to reach to the back flue sheet. If the holes are among the flues, the nozzle should be a bent one, and should be revolved as it is drawn from the back end to the front end.

"After washing the barrel completely, clean the back end of arch flues, making sure that they are free from scale at that end. Then use bent nozzles in the side and corner holes of water legs, revolving same thoroughly to clean the side sheets, and finally clean off all scale and mud from the mud ring by means of straight nozzles in the corner holes. It must not be assumed that because the water runs clear from the hose that the boiler is clean, but all spaces must be examined carefully with rod and light, and if necessary, use a pick, steel scraper, or other tools to remove accumulating scale."

It is wise to screw a bushing in the several washout holes to avoid bruising the thread with the tools and pipes; also the washout pressure should not be less than 100 pounds to the square inch.

CHAPTER IX.

FUEL CONSUMPTION.

As the cost of fuel is usually the largest single expense connected with the operation of locomotives it is highly important that this subject receive a full and thorough treatment in this work, particularly as it is desired to make the study complete without necessitating laborious researches by the student among other volumes. On this account it is deemed advisable to briefly examine the composition of the various kinds of fuel used by locomotives, their economical combustion and thermal efficiency, as well as the quantity required in different units of work, time, distance, etc., and as American practice is principally discussed in this work, our examination will be chiefly confined to the fuels used in this country, which, however, comprise a number of kinds and grades.

COMPOSITION OF FUELS.

Four different kinds of fuel are used upon locomotives in this country, wood, coal, coke and oil, but there are many grades or varieties of each, with varying composition. The use of wood is now generally limited to small roads and unimportant branches, as coal has come to be the almost universal fuel. In Mexico, where the price of coal is high, much wood is used for locomotives, and in some parts the roots of the mesquite are dug up and burned. In some of the heavily wooded districts in this country a considerable quantity of wood is used, but the amount is probably diminishing each year, even for firing up, as fuel oil and air jets have largely taken the place of cord wood.

As a combustible wood is divisable into two general classes —hard and heavy woods, as oak, elm and ash, and soft and light woods, as pine, birch and poplar. All wood contains large proportions of moisture, from 20 to 40 per cent being commonly found in wood that has not been specially dried, and even when thoroughly desiccated it will, upon exposure to the atmosphere, a' sorb about 15 per cent of water.

The chemical composition of wood from various trees is remarkably similar, and in general runs about as follows, for dried and ordinary fire wood:

Constituent.	Dried Wood.	Ordinary Wood.
Carbon	50 Per Cent	37.50 Per Cent
Hydrogen	6 "	4.50 "
Oxygen	41 "	30.75 "
Nitrogen	1 "	0.75 "
Ash	2 "	1.50 "
	100 Per Cent	75.00 Per Cent
		Moisture .... 25.00 "
		100.00 Per Cent

Wood intended for locomotive use should always be dried to get rid of the moisture as far as possible, and should be kept protected from the rain, though this is seldom done.

Coal is so largely used for locomotives, and, in fact, all steam purposes, that it has almost become the universal fuel in this country, yet it is by no means a satisfactory one, particularly as there is so much absolute dirt that is palmed off for coal, causing engine failures and delays to trains. Coal can be divided into five general classes, and these, with the arbitrary limits of fixed carbon and volatile matter, in percentages of combustible (that is, free from ash or moisture), for each group, are as follows:

Class of Coal.	Per Cent of Fixed Carbon.	Per cent of Volatile Matter.
Anthracite	100 to 92	0 to 8
Semi-anthracite	92 to 87	8 to 13
Semi-bituminous	87 to 75	13 to 25
Bituminous	75 to 50	25 to 50
Lignite	below 50	over 50

The chemical composition of the several classes of coal mentioned may be represented by the analyses of fuels typical of the different groups—of course, variations will occur above and below the figures given.

APPROXIMATE COMPOSITION OF TYPICAL KINDS OF COAL IN PER CENTS.

Constituent.	Anth.	Semi-Bit.	Pit., Pa.	Bit., Ohio	Bit., Ill.	Lignite.
Carbon	86.0	84.0	75.0	67.0	56.0	55.0
Hydrogen	1.0	4.2	5.0	4.8	5.0	4.0
Oxygen	1.0	3.4	8.0	10.0	11.0	15.0
Nitrogen	.5	.8	1.0	1.2	1.0	1.0
Sulphur	.5	.6	1.6	1.5	3.0	1.0
Ash	10.0	6.0	8.0	8.0	13.0	5.0
Moisture	1.0	1.0	1.4	7.5	11.0	14.00

While the complete or ultimate analysis is interesting, as a rule the proximate analysis is more commonly used, and this divides the fuel into four general constituents, viz., fixed carbon, volatile matter, ash and moisture, and the proportion of fixed carbon and volatile matter decides to what general class the coal belongs.

The use of coke is restricted to certain portions of roads or service, such as tunnels, cities, etc., where the emission of smoke is particularly objectionable. As a general fuel, it is not desirable on account of its slow ignition and its great bulk for a given weight. The composition varies somewhat, depending upon the length of time it is allowed to remain in the coke ovens. The best cokes are those from Connellsville, Pa., and the Pocahontas district, in Virginia, and they have the following approximate analyses:

COMPOSITION OF COKE.

Constituent.	Connellsville.	Pocahontas.
Fixed carbon	89 Per Cent	93 Per Cent
Ash	10 "	6 "
Sulphur	1 "	1 "

Here it is seen, the combustible is practically carbon, and nothing else. Gas coke is sometimes used, as it is cheaper than that manufactured expressly to sell as coke, as it is a by-product.

Of recent years, oil has become a very successful rival of coal in California and Texas, where large fields being opened, have placed the price of that commodity considerably below coal. In Pennsylvania and Ohio, the low price of coal prevents fuel oil being considered seriously for locomotives, but in Texas, and especially in California, the opposite case exists —in the latter state a few years ago coal for locomotives was valued at $5.00 or $7.00 a ton, while the many oil fields have reduced the price of oil recently to 25 cents a barrel, in some cases, four barrels being usually considered as containing the heat equivalent of a ton of coal. It has a great advantage in the matter of handling, as compared with coal, but it is much more severe on the firebox.

The chemical composition of pètroleum is remarkably uniform, even that obtained from points far distant from each other. The annexed table gives the analysis of California (Kern River), Texas (Beaumont), and an average of 15 samples from different sources, which were analyzed by M. Sainte-Claire Deville:

ANALYSIS OF PETROLEUM.

Constituents, etc.	California.	Texas.	Deville.
Carbon	84.4 Per Cent	84.6 Per Cent	84.7 Per Cent
Hydrogen	11.0 "	10.9 "	13.1 "
Oxygen	3.3 "	2.9 "	2.2 "
Nitrogen	.7 "	......	......
Sulphur	.6 "	1.6 Per Cent	......
Specific gravity	0.962	0.924	0.870
Flashing point	228° Fah.	180° Fah.	......
Burning point	258° "	208° "	......

As compared with coal, it is apparent that the large quantities of hydrogen in oil will give it a much greater heating value—this will be discussed under a later heading. The Texas oil is lighter and more inflammable than the California oil, and much more care is needed in handling it; the vapors given off by this oil are very poisonous, and tanks which have contained it cannot be safely entered for cleaning or repairs until they have been thoroughly steamed or washed out.

COMBUSTION.

Combustion is simply the chemical combination of the constituents of a fuel, principally carbon and hydrogen, with oxygen, producing oxides, and accompanied by heat (which is the result of the chemical action), and which in turn performs the useful work of generating steam, when the operation is performed in the firebox of a boiler. Only the active constituents unite with the oxygen in useful work, that is, the carbon and hydrogen—nitrogen remains inert, and the sulphur does more harm in fouling the fire than it benefits by generating heat, although it has a low calorific value. The moisture and ash are also useless from a heating standpoint, and their presence means so much loss or waste.

The oxygen necessary to maintain and support combustion is derived from the air which is composed of 23.6 per cent oxygen and 76.4 nitrogen by weight, so that for each pound of oxygen delivered to the furnace we must supply $\frac{1}{.236} =$ 4.24 pounds of air.

As with the nitrogen in the fuel, that in the air is of no value as a heat agent, as it merely dilutes the products of combustion. Air being generally measured by volume, it is convenient to convert its weight into cubic feet, but here the temperature must be considered. At 32 degrees Fahrenheit it requires 12.39 cubic feet of air to make one pound in weight, and as we found in formula 111, the volume at any other temperature, and at atmospheric pressure, will be proportional to the absolute temperatures, or

$$v = 12.39 \frac{461 + t}{493} \quad \ldots\ldots (115)$$

where v and t are the volume and temperature under consideration.

These values we can tabulate for convenient use, and we obtain the volume in cubic feet (v) for one pound of air at atmospheric pressure and at temperatures (t) in Fahrenheit degrees, as follows:

t =	0	32	40	50	62	70	80	90	100
v =	11.58	12.39	12.59	12.84	13.14	13.34	13.59	13.85	14.10

The hydrogen in the fuel unites with enough oxygen to form water, which passes off as steam, this reaction being shown as follows: $H_2 + O = H_2O$ = water, where H is the symbol for hydrogen and O that for oxygen.

In the same manner the carbon unites with oxygen to form either carbonic oxide, when the combustion is imperfect, or carbonic acid if the combustion be complete; if we use the symbol C for carbon, we can write these reactions

$$C + O = C\,O = \text{carbonic oxide};$$

$$C + O_2 = C\,O_2 = \text{carbonic acid}.$$

In the above equations, the quantities by weight can be determined from the arrangement of the atoms, by using the atomic weight of the different elements. These atomic weights are: Oxygen, 16; hydrogen, 1; carbon, 12; sulphur, 32.

Thus, in burning hydrogen to water the quantities by weight involved are $1 \times 2 + 16 = 18$, or two pounds of H + 16 pounds of O produce 18 pounds of water, and every pound of hydrogen requires 8 of oxygen for its complete combustion. So, for carbon, the complete combustion to carbonic acid requires $12 + 16 \times 2 = 44$, or 12 pounds of C and 32 pounds of O make 44 pounds of carbonic acid, and each pound of carbon needs $\frac{32}{12} = 2.66$ pounds of oxygen. If it receives only half of this amount, carbonic oxide is formed, $12 + 16 = 28$, or $\frac{16}{12} = 1.33$ pounds of oxygen to 1 of carbon. Now, if we let the number of pounds of carbon and hydrogen in any fuel be represented by the symbols C and H, we can find the amount of oxygen, O, needed for complete combustion by the formula $O = 2.66\,C + 8\,H$, and as there is often a quantity of oxygen in the fuel itself, we can consider that this will unite with the hydrogen as far as it will go, and if O = the oxygen in the fuel, the amount of oxygen to be supplied from an outside source will be $2.66\,C + 8\,(H - \frac{O}{8})$. This oxygen, however, must be supplied by the atmosphere, of which it will require 4.24 pounds to provide 1 pound of oxygen, so that the necessary amount of air will be in pounds

$$\text{Air} = 4.24 \times 2.66\,C + 4.24 \times 8\,(H - \frac{O}{8}),$$

$$= 11.3\,C + 34\,(H - \frac{O}{8}), \text{ which is commonly written}$$

$= 12\,C + 36\,(H - \frac{O}{8})$, or, if we desire, the cubic feet of air at 62 degrees Fahrenheit necessary, instead of the weight, we have cu. ft. air $= 13.14 \times 12\,C + 13.14 \times 36\,(H - \frac{O}{8})$

$$= 158\,C + 473\,(H - \frac{O}{8}) \qquad \text{.........................(116)}$$

If we apply this formula to the different kinds of fuel that we have listed, taking the analyses given, we obtain the cubic feet of air at 62 degrees Fahrenheit and at atmospheric pressure that will be actually used in the combustion of one pound of the fuel. It is customary to consider that twice this quantity, or at least 50 per cent more must be supplied to insure complete combustion, though it cannot be used for chemical action in excess of the amount really required for this purpose.

Cubic feet of air at 62° F. required for the combustion of one pound of fuel.

Petroleum	167	Pennsylvania bituminous	137
Coke	144	Ohio bituminous	124
Anthracite coal	141	Illinois bituminous	108
Semi-bituminous	151	Lignite	97
		Wood, dry	84

It is thus seen that oil requires by far the greatest amount of air for its combustion, and wood the least. This forms a guide by which the open space in the grates can be proportioned for different kinds of fuel. The resistance in passing through the bed of fuel has also a great effect upon the volume of air admitted, and for this reason coal, which packs closely together, must be kept spread thinly over the grate. The average proportion of openings in grate bars for bituminous coal is perhaps from 30 to 40 per cent, but for coke and anthracite 40 to 45 per cent is desirable. For wood burners the area of openings should be greatly reduced, say about 15 or 20 per cent. The openings in the fuel bed itself are large with wood and this reduces the friction and increases the volume of the air drawn in with the same smokebox vacuum. While it is important to obtain sufficient air to ensure burning the carbon to CO_2 instead of CO, an unnecessarily large quantity of air reduces the temperature of the fire and cools the boiler. Mr. A. Bement has given considerable attention to the quality of locomotive smokebox gases as produced by various methods of firing. In one experiment, where the fireman was left to his own methods, the proportion of CO_2 to CO was as 11.07 is to 2.33 as the average for the trip. In another case, where a lighter fire was carried and supplied more frequently, with numerous shakings of the grates, the results were, for the average percentage of the trip, 12.43 CO_2 and no CO. This points out a method of watching the firing by means of the analysis of smokebox gases, as is done frequently in stationary plants, but which is laborious upon locomotives. The "Econometer" was designed to give a continuous reading of the percentage of CO_2 in the escaping gases, but we do not know of any continuous type for locomotives; besides, there would ordinarily be no one to watch the results on each engine, unless the apparatus recorded the amount of carbonic acid made throughout the trip.

From formula 116 we can determine the percentage of the air that should combine with the carbon to produce CO_2. For instance, in Pennsylvania bituminous coal, with .75 C, .05 H, and .08 O, we have the cubic feet of air needed for the carbon $= 158 \times .75 = 118$, and for the hydrogen $= 473 \times$

$$(.05 - \frac{.08}{8}) = 19, \text{ or } \frac{118}{118 + 19} = .86$$

of the oxygen goes to the carbon, and as the total oxygen is only .236 of the weight of the air, we see that but $.86 \times .236 = .20$, or 20 per cent of the total needed air combines with carbon. Now, as there is generally twice as much air supplied as is chemically required for combustion, the best results expected cannot be over 10 per cent—if only 50 per cent excess is supplied, there might result 15 per cent of CO_2, but there would be danger of reducing the CO_2 by the production of CO. Mr. Bement states that in practice the amount of CO_2 in the gases of combustion varies from 11 to 15 per cent; the most economical amount can be figured for different kinds of fuel as in the last example. The excess air will reduce, of course, the percentage of CO_2 formed. For instance, the writer above mentioned gives for a certain grade of coal (kind not stated) the following relations between the percentage of CO_2 in volume and the cubic feet of air supplied per pound of coal:

Air =	150	200	300	400	500	600	700
CO_2 =	20	14½	9½	7	6	4½	4

Mr. Robert Wilson, in "Boiler and Factory Chimneys," states that in estimating the volume of the gases of combustion we can take the volume of the mixed carbonic acid, nitrogen and unburnt oxygen as equal to the original volume of air supplied to the furnace, and increase its density simply in the ratio of the sum of the weights of the air and of the carbon taken up, to the weight of air. The volume of the products of combustion is greater than the original volume of air supplied by an amount equal to the quantity of oxygen that has combined with hydrogen, but the quantity of hydrogen in ordinary fuel is so small a proportion of the total weight that it is not worth considering. The volume, of course, must be increased in proportion to the absolute temperatures, as per equation 111. The volume of one pound of air in cubic feet at high temperatures will be:

t =	100	200	300	400	500	600	700	800
v =	14.10	16.60	19.12	21.63	24.15	26.66	29.17	31.68
t =	900	1,000	1,500	2,000	2,500	3,000 degrees F.		
v =	34.20	36.81	49.38	61.94	74.57	87.13		

all at atmospheric pressure, and the relative volumes of air supplied to furnace and gases drawn through flues will be in the ratio of the volumes given above for the corresponding temperatures; for instance, 14 cubic feet at 100 degrees will become 24 cubic feet if heated to 500 degrees Fahrenheit, if maintained at atmospheric pressure. Of course, the discharge from the stack includes the steam from the exhaust, as well as the products of combustion.

We have heretofore been considering the amount of air, etc., needed for the proper combustion of one pound of fuel—

a locomotive, however consumes many pounds a minute, so that the quantities of air used and gases produced are very great. The rate of combustion is generally referred to in pounds burned per square foot of grate area per hour, although much of it is not burned, but passes through the stack in an unconsumed or partially burnt condition.

The rate of combustion, when forced to its maximum, has been stated in the chapter on "Steam Capacity," and the greatest rates generally obtained in practice are about as follows:

Bituminous coal	200 pounds per hour
Anthracite coal (large)	100 pounds per hour
Anthracite coal (small)	60 pounds per hour

Ordinarily, the rate of combustion is much less than these figures, as the amounts stated above are only reached when the engine is working on a heavy grade or at high speed; as would be expected from this statement, the rate is generally greater in passenger than in freight service. Some tests on the Great Northern Railway showed 127 pounds for passenger and 101 pounds for freight service per square foot of grate per hour while throttle was open, with bituminous coal. On the Michigan Central, the rate averaged for various tests from 60 to 90 pounds per hour. Special fuel tests on the Furness Railroad of England recently averaged from 45 to 60 pounds per square foot per hour, with bituminous coal. As we shall see later on, the economy varies with the rate; it is probably true that 100 pounds an hour is a fairly large average consumption under ordinary conditions, when it represents the average for a long run, though double that amount will often be used for comparatively short distances.

For anthracite coal the average rate of combustion is probably about 60 per cent of the maximum stated. In tests made on the Lackawanna with anthracite culm, the average combustion for several trips in passenger service ran from 33 to 44 pounds per hour—with large sizes, it would probably average 60 pounds.

As previously stated, there seems to be no definite limit to the amount of oil that can be burned, and as grates (as such) are not used in oil burners, we have no equivalent expression for the rate of combustion. It can be consumed at the rate of 1.5 pounds per square foot of heating surface per hour, and with the usual proportions of heating surface to grate area, this would be equivalent to somewhere about 100 pounds per square foot of grate. The intense heat is very hard on the fireboxes when the combustion is forced and the steam generation increased greatly above that with coal. From formula 94 we can determine the draft necessary to produce any given rate of combustion, the maximum values requiring by the formula, a smokebox vacuum of about 7½ inches of water. Strong draft raises the lighter particles of coal and ejects them from the stack partially burned. The amount so ejected depends greatly upon the nature of the fuel, being greatest for wood and lignites, and least for large sizes of anthracite. From some tests made at Purdue University, it was found that with severe drafts, the percentage of sparks ejected was very large. These experiments were made with five grades of bituminous coal, and the percentage of fuel fired which was represented by the weight of sparks ejected from the stack under various draft conditions is given below:

Draft in Inches of Water.	Kind of Coal.	Cinders Per Cent of Coal Fired.
1.51	E	5.2
5.13	E	23.1
5.73	E	21.4
1.70	D	5.0
5.40	D	17.2
6.98	D	20.0
1.77	C	3.0
5.34	C	13.6
5.74	C	14.5
1.70	A	5.7
5.67	A	16.1
6.32	A	15.1
1.89	B	4.0
5.67	B	16.2
6.40	B	18.0

When these values are plotted, the average may be expressed by the following formula. Let, as before, d = draft in smokebox in inches of water; c = percentage of coal fired which passes out of stack as cinders, then

$$c = 3\,d \qquad (117)$$

that is, for each inch of water draft, there will be 3 per cent of fuel sent, partially unburnt, through the stack.

That the amount of coal burned depends principally upon the quantity of steam used and exhausted through the stack can be demonstrated by combining equations 94 and 95. In 94 let us substitute f for the numerical coefficient of c, so that we have $d = f c$; also in 95 substitute f_1 for the coefficient of q, obtaining $d = f_1 q$. Now, by equating we can write

$$f c = f_1 q, \text{ or } c = \frac{f_1}{f} q \qquad (118)$$

that is, the pounds of coal "c" that can be burned per square foot of grate per hour are proportional to the pounds of steam "q" passing through the exhaust pipe, per second, which is equivalent to saying that the coal consumption varies in a locomotive as the steam consumption of the same locomotive. This might be construed as meaning that the steam generated was a direct function of the coal burned, and while this is so in a general way, it is by no means strictly true, as the generation of steam per pound of fuel burned decreases as the rate of combustion increases, as will be seen further on.

THERMAL VALUE OF FUEL.

The heating value of a fuel depends upon the quantities of carbon and hydrogen which it contains—the other constituents have little effect. Sulphur generates a small amount of heat, but it is generally neglected in making computations from analyses. Nitrogen is inert, and oxygen helps to support the combustion of the hydrogen present. If already combined with hydrogen in the form of moisture, it merely forms bulk, and does not add to the heat generated—in fact, it absorbs heat, as it must be converted into steam or vapor before the fuel can burn. Ash also merely acts to reduce the proportion of useful elements, and it also nullifies a quantity of useful heat by dropping hot into the ashpit.

It has been found by experiment that elementary substances will always produce the same quantity of heat if burned completely in the presence of oxygen, and the measure of heat generally adopted in this country and Great Britain is the "British thermal unit," often simply written "B. T. U." This unit is defined as that quantity of heat which will raise one pound of pure water one degree Fahrenheit in temperature. The total heat of the principal elements found in fuel and as determined by experiment is stated herewith, per pound of the element:

Carbon ....14,500 British thermal units
Hydrogen ....62,100 British thermal units
Sulphur .... 4,000 British thermal units

The total heat of combustion of one pound of a fuel is found to be the sum of the quantities of heat which the combustible elements contained in the fuel would produce if burned separately. If a fuel contains oxygen as well as hydrogen, eight parts by weight of the former unite with one part of the latter to form water, which exists as such in the fuel, and this does not add to the total heat of combustion. If there is, however, an excess of hydrogen beyond what is required to form water with the oxygen, the remaining hydrogen does add to the total heat of combustion, and may be reckoned in estimating its value. If we again let the chemical symbols represent the quantities of these elements in pounds in one pound of fuel, we can, from the above data, write the total heat

$$\text{B. T. U.} = 14{,}500\,C + 62{,}100\left(H - \frac{O}{8}\right) \qquad (119)$$

This is known as Dulong's formula, and is almost universally used in estimating the heating value of fuels from the ultimate analysis. The sulphur is omitted in the formula as written above, though it is sometimes included. As this substance is objectionable in fuel it is considered better to omit it in an estimate of heat value.

This formula is used as follows: In order to determine the heat value of Texas oil from the analysis given previously as $C = .846$; $H = .109$; and $O = .029$, we write our equation

$$14{,}500 \times .846 + 62{,}100\left(.109 - \frac{.029}{8}\right) = 18{,}812 \text{ B. T. U.},$$

that is, from the amount of hydrogen, deduct one-eighth of the amount of oxygen and multiply the remainder by 62,100, and to this add the amount of carbon multiplied by 14,500. The result is the total heat generated by burning one pound of the fuel. Fig. 103 gives a graphical determination of Dulong's formula. In it the carbon, hydrogen and oxygen are considered as making all together 100 per cent, or a unit of weight, so that to use it correctly, the other elements must be deducted, and the three mentioned considered as forming the

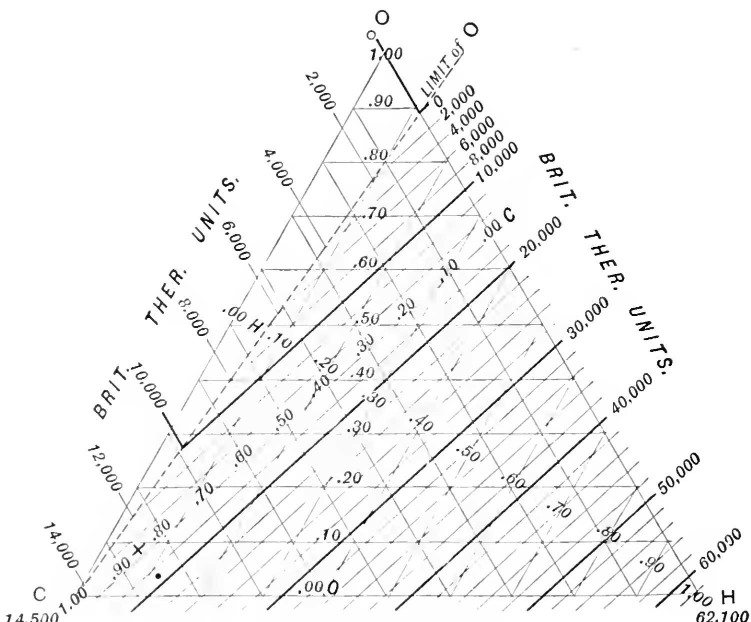


Fig. 103.

combustible. The proportion of each element, C, H or O, is designated by the distance on the diagram from the base opposite the apex marked with the symbol. For instance, when C = 1.00, or the material is nothing but carbon, the corresponding point will be at the lower left-hand apex, as this point is distant 1.00 parts by scale from the line or side connecting O and H. If H = 1.00, the location is at lower right-hand apex. If the fuel be carbon and hydrogen only and does not contain oxygen, the point must be located on the base line.

Thus, marsh gas, H_4C, which is .25 hydrogen and .75 carbon, is found on the base line, .25 of the distance from C and .75 from H. The values on the triangle lines designate the value or proportion of the element. Also a compound of .70 C, .20 H and .10 O would be found at the intersection of the horizontal line marked .10 (O), the line inclining to the right marked .20 (H) and the line inclining to the left marked .70 (C). The close diagonal lines show the theoretical heat units in one pound of the mixed elements, C, H and O. The broken line marked "limit of O" shows the dividing point of oxygen, or where the oxygen and hydrogen neutralize each other and produce no heat. To the left of this line heat will be derived only from the proportion of carbon, which accounts for the heat lines running at a different angle.

To illustrate, by observing that the point on base line where C = .75 and H = .25 is crossed by the heat line 26,000, we estimate the heat caused by the combustion of one pound of marsh gas at about 26,000 units. For the Texas oil, we must proceed as follows:

$$\begin{aligned} C &= .846 \\ H &= .109 \\ O &= .029 \\ \hline & .984 \end{aligned}$$

As this total is nearly unity, we can locate the point as shown by the dot and find the value about 19,000 heat units. (By calculation we found 18,812.)

Taking the analysis of Pennsylvania bituminous coal, we find that the C, H and O are considerably less than unity; thus:

$$\begin{aligned} C &= .75 \\ H &= .05 \\ O &= .08 \\ \hline & .88 \end{aligned}$$

The rest is incombustible, and must be omitted. In our diagram, each part must be considered in its relation to .88 or its percentage of .88, so that the carbon becomes $\frac{.75}{.88} = .85$, hydrogen .06, and oxygen .09. This gives the location shown at "x" in the figure, or 15,000 heat units, but as one pound of fuel contains only .88 of the elements considered, the fuel itself will produce only $.88 \times 15{,}000 = 13{,}200$ B. T. U. per pound.

In estimating the total heat in any fuel, the analysis should be at hand; this varies greatly for different mines and localities, and often for different samples from the same mine. It is needless to reproduce here tables of such analyses, as they are readily accessible, and besides, the individual case must be studied for close results.

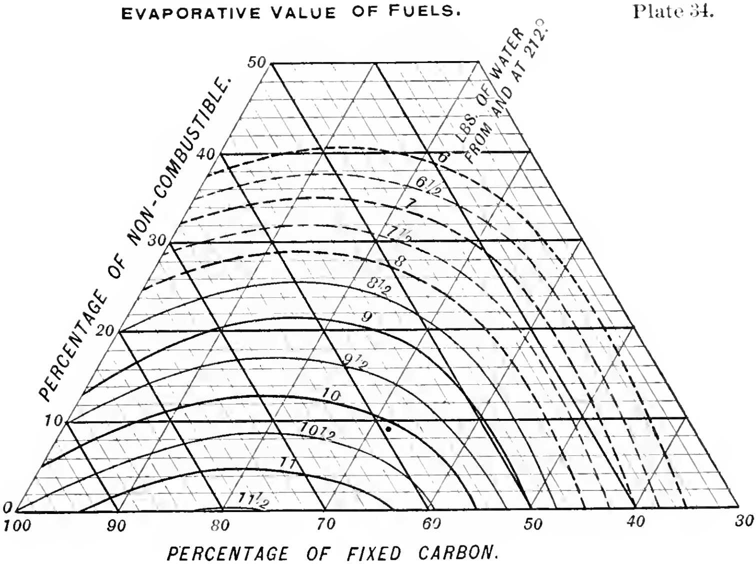
EVAPORATION.

While the total heat is useful in making comparisons between different grades or kinds of fuel, the evaporation is of much greater practical value; it depends upon a variety of conditions, any one of which may alter the results in a marked manner, so that we will first consider the evaporation resulting from slow and complete combustion of the fuel, as would be expected in a stationary boiler working under a moderate draft, and in care of a skillful fireman.

The evaporative value of coal is generally derived from the proximate analysis, the constituents being classified as fixed carbon, volatile matter, ash and moisture, so that these four quantities make up 100 per cent. Of course, absolutely reliable results cannot be expected. The heat units determined are divided by 966, the latent heat of steam at atmospheric pressure, which gives the equivalent evaporation from and at 212 degrees. As the efficiency of the boiler will probably average about 70 per cent, seven-tenths of the evaporation thus found will represent the actual in practice. Plate 34 gives this data in graphical form, the evaporative value (from and at 212 degrees) being such as would be expected from each pound of the fuel in a modern steam boiler, when the rate of combustion was about three-fourths of a pound of fuel per square foot of heating surface per hour. In a locomotive this would be an extremely low rate, but where it was so maintained, the evaporation would probably be nearly as shown in the plate.

The diagram is constructed upon the same general principles as Fig. 103. The amounts of ash and moisture, added together, constitute the non-combustible portion, which are measured by the horizontal lines. The sloping lines (from left to right) designate the percentage of fixed carbon, as shown at bottom. To use the chart, simply find the intersection of the lines corresponding to the percentages of fixed carbon and non-combustible matter in the fuel, and the evaporation will be found by interpolation from the curves. The solid line water values are constructed in accordance with data given in Kent's Mechanical Engineer's Pocket Book, and

EVAPORATIVE VALUE OF FUELS. Plate 34.



the broken lines upon the same basis, but it is not known that the latter (broken lines) will be as reliable as the former (solid lines) as they extend beyond the range of Kent's table.

The following example will indicate the use of the diagram; the proximate analysis of Connellsville bituminous coal indicated

1.26%	moisture,
30.12%	volatile matter,
59.61%	fixed carbon,
8.23%	ash,
99.22	

the rest being sulphur, or .78%. The non-combustibles are $1.26 + 8.23 = 9.49$, say 9½. Fixed carbon, 59½, approximately. The value is located as shown by the dot, and the evaporation is found to be about 10 pounds of water per pound of coal, from and at 212 degrees Fahrenheit. The following table gives the proximate analyses of representative coals of the several kinds enumerated, the values being in percentages of the whole fuel:

PROXIMATE ANALYSES OF COALS.

Kind.	Locality.	Moisture.	Vol. Mat.	Fixed Carb.	Ash.
Anthracite	Pennsylvania	3.5	4.0	84.0	8.0
Anthracite	New Mexico	2.0	9.0	76.0	13.0
Semi-anthracite	Pennsylvania	1.0	9.0	83.0	6.0
Semi-bituminous	Pennsylvania	1.0	18.0	73.0	7.0
Semi-bituminous	Virginia	1.0	20.0	76.0	3.0
Bituminous	Pennsylvania	1.5	34.0	57.0	8.0
Bituminous	Virginia	1.5	35.0	56.0	6.0
Bituminous	Ohio	4.0	35.0	54.0	7.0
Bituminous	Kentucky	4.0	34.0	55.0	7.0
Bituminous	Illinois	10.0	35.0	43.0	12.0
Bituminous	Missouri	6.5	37.5	48.0	8.0
Bituminous	Kansas	3.0	36.0	55.0	6.0
Lignite	Wyoming	8.0	39.0	42.0	11.0
Lignite	Utah	9.0	42.0	44.0	5.0
Lignite	Oregon	15.0	43.0	33.0	9.0
Lignite	New Mexico	10.0	42.0	42.0	6.0

As stated previously, the evaporation indicated by the combustion of one pound of fuel in plate 34 is only obtained when the rate is low—much lower than generally obtains in a locomotive performing road service. As the rate of combustion increases, the evaporation per pound of fuel diminishes. This is due to several things, perhaps the most important being the large amount of unburnt fuel thrown from the stack by the heavy blast of the exhaust. Another is the rapid motion of the gases, due to the increased draft, whereby they do not have as much time to impart their heat to the tubes of the boiler, and the water outside of them; also the incomplete combustion of the fuel on account of insufficient air.

The rate of combustion is measured commonly in two ways, one in pounds of fuel burned per square foot of grate area per hour, and the other per square foot of heating surface per hour. These are often used indiscriminately, but the author believes that there is a logical use for each unit. In considering questions of fuel combustion alone, and without relation to the evaporative efficiency, it is desirable to use the square foot of grate area as a unit, because this is intimately and directly connected with the burning of the fuel. When, however, we desire to investigate the question of water evaporation, the heating surface is the medium through which the heat is transmitted, and it is better to use that as the unit. We saw above that the maximum rate of combustion was taken at 200 pounds per square foot of grate area per hour, and as plate 34 gives the steam efficiency at low rates, we must

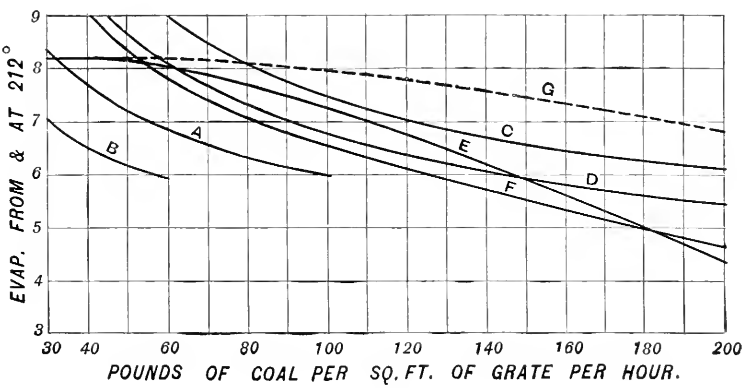


Fig. 104.

determine the amount of water evaporated at intermediate rates of combustion. While it is recommended always to figure evaporation on heating surface, some data based on grate area are presented in Fig. 104. The curves designated by letters have the following significance:

A = Large sizes of anthracite.
B = Small sizes of anthracite.
C = Semi-bituminous of Pennsylvania and Virginia.
D = Bituminous of Indiana and Illinois.
E = Bituminous of Indiana (Brazil block).
F = Baldwin Locomotive Works.
G = Goss experiments with diminished grates.

Curve F is given by the Baldwin Works in their "Recent Construction," but the kind of coal is not stated. E is from experiments made at Purdue University with their test locomotive, and G is from the same engine during a series of experiments made by decreasing the grate area, by blocking off portions, but keeping the quantity of coal burned per hour constant for the locomotive—it is therefore constant for the heating surface, and it is noticed that the evaporation varies less than in any of the other cases. The small drop at high rates of combustion per unit of grate is due to the fact that the only losses are in sparks and incomplete combustion, whereas, the same total quantity of fuel being burned, the gases have the same volume and velocity through the flues, and the heating surface being constant, has the same opportunity to absorb the heat of combustion. While evaporation based on unit of heating surface does not specifically comprise spark losses, yet, as the ratio between grate area and heating surface has a certain average value, the question of draft will be in a measure cared for on this basis. It is found that the results of various tests, when plotted on a chart using combustion rate per unit of heating surface coincide much more closely than when the rate per unit of grate area constitutes the abscissa, probably due to the fact that the rate based on heating surface varies more than that on grate area, as indicated by lines E and G.

Plate 35 shows the variation in evaporative ratios due to different rates of combustion per square foot of heating surface per hour, in water from and at 212 degrees Fahrenheit. As before, the letters designate:

A = Large sizes Pennsylvania anthracite.
B = Small sizes Pennsylvania anthracite.
C = Pennsylvania and Virginia semi-bituminous.
D = Indiana and Illinois bituminous.

As previously stated, these values cannot be depended upon absolutely in any case, but as indicating the general manner in which the evaporation depends upon the rate of combustion they are very useful, as will appear when the quantity of coal is to be determined. The evaporative power of fuel oil will be discussed when considering oil-burning locomotives.

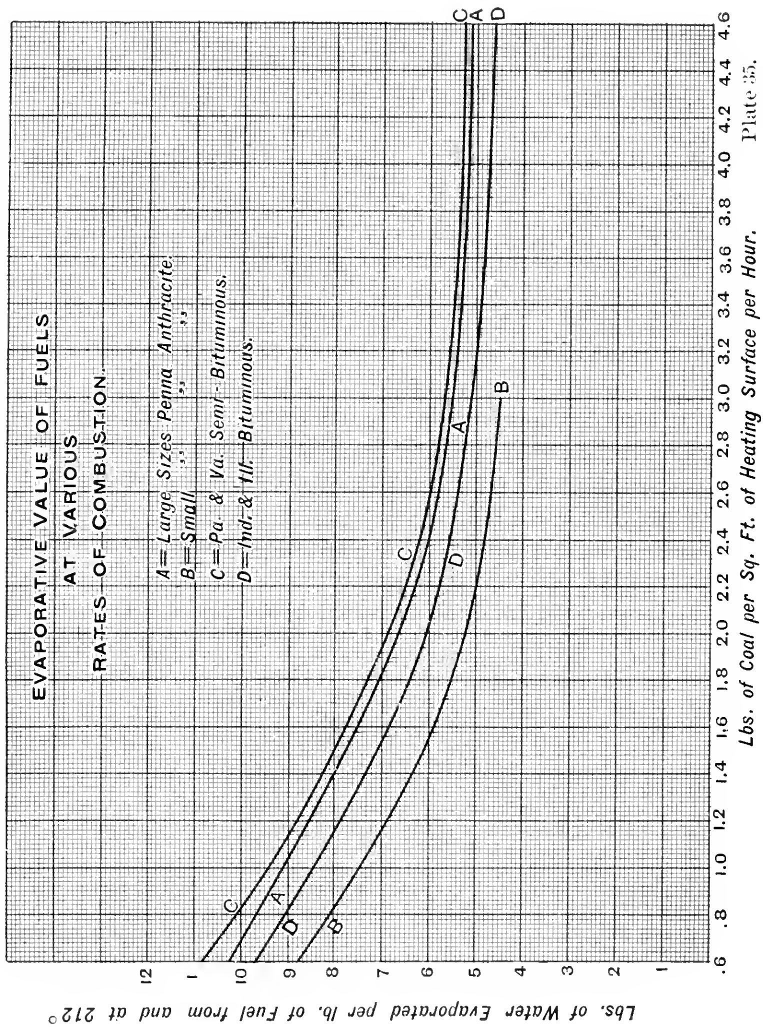

EVAPORATIVE VALUE OF FUELS
AT VARIOUS
RATES OF COMBUSTION.
A = Large Sizes Penna Anthracite.
B = Small ,, ,, ,,
C = Pa. & Va. Semi-Bituminous.
D = Ind. & Ill. Bituminous.
A
B
C
D
.6
.8
1.0
1.2
1.4
1.6
1.8
2.0
2.2
2.4
2.6
2.8
3.0
3.2
3.4
3.6
3.8
4.0
4.2
4.4
4.6
Lbs. of Coal per Sq. Ft. of Heating Surface per Hour.
1
2
3
4
5
6
7
8
9
10
11
12
Lbs. of Water Evaporated per lb. of Fuel from and at 212°

Plate 35.

QUANTITY OF COAL USED.

Having determined the evaporative power of fuels, we are now in position to pursue our study and investigate the amount needed to perform various quantities of work. We saw that there was a limit to the amount of coal that could be burned in a firebox, depending upon the size of grate and grade of fuel. This fixes the maximum quantity that can be consumed, and also corresponds to the maximum evaporation as a total, or per unit of heating surface. The lower quantities of fuel depend upon the work being done by the steam, and the economy of steam itself affects the amount of fuel. Thus we have seen that at different expansive ratios, the work which can be performed by a pound of steam is different, and at different combustion ratios, the amount of steam generated by a pound of coal is also different. This combination gives us a great variety of conditions in a given locomotive, and we must determine how the results will be correspondingiy affected.

In order to obtain a clear idea of the variation in quantity of fuel to perform a unit of work, we must combine plate 35 and Fig. 101. As an example we will take Virginia semi-bituminous coal for the fuel, and consider its combustion in a simple locomotive boiler in which the heating surface is 80 times the grate area. At a maximum rate of 200 pounds per square foot of grate per hour, the rate will be $\frac{200}{80} = 2.5$ pounds per square foot of heating surface per hour, and we can also consider the lower rates of 2, 1.5 and 1 pound per hour.

Lbs. coal per sq. ft. heating surf per hour		2½	2	1½	1
Evaporation from and at 212° (pl. 35)		6.1	6.9	8.0	9.4
Lbs. water per lb. coal at 200 lbs. pres		5.1	5.7	6.7	7.8
	Cut-off				
Lbs. coal per I. H. P. hour = $\frac{\text{Lbs. water per I. H. P. hour (fig. 101)}}{\text{Lbs. water per lb. coal (as above)}}$	1	4.9	4.4	3.7	3.2
	.2	4.45	4.0	3.4	2.9
	.3	4.3	3.85	3.3	2.8
	.4	4.35	3.9	3.35	2.85
	.5	4.5	4.0	3.45	2.95
	.6	4.7	4.2	3.6	3.1
	.7	5.1	4.55	3.9	3.35
	.8	5.7	5.1	4.3	3.7
	.9	6.5	5.8	4.9	4.25

We see from this that we may have coal rates per indicated horsepower hour varying by more than 100 per cent, depending upon the combination of expansion and combustion ratios. The table can be extended for compound engines, and also worked up for other fuels and for different boiler proportions, if desired, by following the method shown.

For estimating the amount of coal burned when exerting a definite tractive force and running at a selected speed, we can have recourse to the method given for determining the water used. The following table exhibits the ratios of tractive force and fuel consumed to the total or maximum tractive force and coal consumption at that speed for several different speeds, as deduced from the series of tests made upon the Chicago & Northwestern apparatus:

Cut-off.	Speed in Miles Per Hour.									
	10		20		30		40		50	
	% T. F.	% coal.	% T. F.	% Coal.	% T. F.	% Coal.	% T. F.	% Coal.	% T. F.	% Coal.
.1	25	15	29	22	34	31	40	43	40	68
.2	43	23	52	33	64	49	85	79		
.3	55	32	67	45	83	67	?		?	
.4	66	39	77	57	95	95				
.5	72	48	87	70						
.6	80	58	97	90						
.7	87	71								
.8	93	84								
.9	98	95								

For general use, plate 36 is presented (at back of book). This has been worked up from the same tests, but is in shape to suit the ordinary proportions of simple engines; that is, such engines as will have their tractive force represented without great error by plate 29. It is based upon the understanding that the curve of available tractive force in plate 29, from B to C, represents the limit of capacity of the boiler, and therefore the maximum coal consumption throughout. As this is defined by the maximum rate of combustion, multiplied by the grate area, the corresponding tractive forces and speeds will require 100 per cent of the maximum fuel consumption, and the 100 per cent line in plate 36 is the same as the line B C in plate 29. If this curve were a regular hyperbola, like the line D E in plate 29, the curves representing percentages less than 100 could be simple hyperbolas, such that the products of their co-ordinates at any and every point would bear the same proportion to the products of the co-ordinates of the 100 per cent line that the quantity of steam generated per square foot of heating surface per hour at the reduced rate of combustion bears to the quantity generated at the maximum rate of combustion. However, all these curves are modified hyperbolas (as explained in connection with plate 29) and their construction is similar to that of the line B C. The ordinates give the percentage of the theoretical tractive force, so that the solution of various problems becomes quite simple. Take, for instance, the Chicago & Northwestern Class R locomotive, with a theoretical tractive force of 31,300 pounds and a grate area of 29 square feet. The maximum coal consumption with Illinois bituminous should be $29 \times 200 = 5{,}800$ pounds. (The maximum in the test was 5,874 pounds per hour.) Any of the speeds and tractive forces indicated by the 100 per cent line would require about 5,800 pounds per hour; as, 25,000 pounds $= \frac{25{,}000}{31{,}300} = 80$ per cent T. T. F. at 60 revolutions per minute. If the same tractive force were desired at 30 revolutions, the coal per hour would be about $5{,}800 \times .40 = 2{,}320$ pounds. Also for 18,000 pounds $\frac{18{,}000}{31{,}300} = 58$ per cent at 106 revolutions, the full rate would be required, while for 9,000 pounds $= \frac{9{,}000}{31{,}300} = 28$ per cent at the same speed, we should expect $.32 \times 5{,}800 = 1{,}860$ pounds per hour, as the 28 and 106 lines intersect at a value of 32 per cent of the maximum fuel rate. In the actual tests, there were 1,953 pounds burned per hour under these conditions. Considering the many variables in a problem of this kind, accuracy cannot be expected, but it is probable that the figures will come as close to results as it would be reasonable to anticipate. If the boiler capacity is such that plate 29 does not fairly represent the existing conditions this plate and also number 36 must be reconstructed as explained. This is likely to be necessary with compound engines, where the supply of steam is large for the volume of the high pressure cylinders.

Some of the features of economy will be taken care of by plate 36, especially if the curves be constructed to suit the different cases. Large grate areas induce economy by reducing the rate of combustion, and this would be covered when the point B in plate 29 was located with the assistance of Fig. 91. A series of tests made on the Southern Pacific Railway with engines closely alike, but one having a ratio of heating surface to grate area of 85.77 and the other 67.75, showed in the neighborhood of 12 per cent economy for the large grate (the second engine mentioned); when the rates of combustion were considered in connection with plate 35, it was found that the increased evaporation is what would be expected from the reduced combustion.

The economy of piston valves will not be so clearly indicated, unless the data be very carefully laid out from the start. M. Edouard Sauvage, in a paper presented at the March, 1904, meeting of the Institution of Mechanical Engineers stated that "piston valves have been found advantageous on account of providing larger steam passages than flat valves, thus reducing wiredrawing, both for admission and exhaust; the valves referred to had inside admission. Compared with similar locomotives having flat valves and handling the same traffic during a prolonged period, the piston valve engines have shown an economy of 10 per cent in coal consumption."

If the curves be laid out to suit compound engines, the economy should be shown by the diagram. At the meeting above referred to, it was variously stated that compound engines saved from 10 to 20 per cent of fuel. On some of our western roads the economy in fuel consumption by compound engines was from 10 to 25 per cent. This largely depends, however, upon the profile of the road. It has been found that where the division consists of a long gradient in one direction, that the steam used by a compound locomotive in order to make it run swiftly down hill may very nearly equal the economy of the up-hill trip, thereby leaving a very small balance in its favor. The most advantageous line for such an engine is evidently one where steam can be used for the whole running distance. As a rule, compound engines require more careful maintenance than simple engines, and a better class of mechanics to do the work needed, and in some sections of this country it is difficult to obtain and retain the quantity and quality of labor necessary; in other words, the engines that require the least attention, are in many ways those that are most desirable.

The economy of superheating has been discussed in the last chapter. Reports of various tests, however, differ greatly, some showing high fuel economy and others apparently none at all. Without complete information regarding the details, it is impossible to assimilate these antagonistic statements. Theoretically, we should expect fuel economies as follows: As previously stated, it was found in the tests on the Prussian State Railway, that equal work consumed equal volumes of saturated and superheated steam, wherefore, if we let

G = the weight of saturated steam consumed per hour;

G′ = the weight of superheated steam consumed per hour,

and consider v and v′ as the volumes of one pound of saturated and superheated steam, respectively, as in formula 111, then we have equal volume for equal work expressed by the equation $G v = G' v'$ and also $\frac{G'}{G} = \frac{v}{v'}$.

Furthermore, let us assume that

Q = the number of heat units required per hour for saturated steam,

Q′ = the same for superheated steam,

l = the total heat required to produce one pound of saturated steam at the desired pressure and temperature t from water at 32° Fahrenheit.

l′ = the total heat of one pound of superheated steam at the same pressure, but at a temperature t′ from water at 32°;

c = the specific heat of dry steam = .48.

Then we can obtain l from the ordinary steam tables, and $l' = l + c\,(t' - t)$. If we consider the heat units above 32°

in one pound of the feed water = q we will have $\underset{\sim}{Q} = G(l - q)$ and $\underset{\sim}{Q}' = G'(l' - q) = G'[l - q + c(t' - t)]$,

hence $\dfrac{\underset{\sim}{Q}'}{\underset{\sim}{Q}} = \dfrac{G'(l' - q)}{G(l - q)}$; but $\dfrac{G'}{G} = \dfrac{v}{v'}$ and from equation III,

$\dfrac{v}{v'} = \dfrac{461 + t}{461 + t'}$, so, by combination, we have

$$\frac{\underset{\sim}{Q}'}{\underset{\sim}{Q}} = \frac{461 + t}{461 + t'} \times \frac{l - q + .48\,(t' - t)}{l - q} \qquad \text{...............} (120)$$

As $\dfrac{\underset{\sim}{Q}'}{\underset{\sim}{Q}}$ is the ratio between the required heat units per hour for equal work, it will also represent the coal ratio, so that the saving will be expressed as a ratio to the fuel used with saturated steam by $1 - \dfrac{\underset{\sim}{Q}'}{\underset{\sim}{Q}}$.

If we consider that the water is delivered to the boiler at 92 degrees Fahrenheit, q = 60 degrees = 92 — 32, and for steam at 175 pounds pressure, t = 377° and l = 1,197, so that for 600 degrees temperature of superheated steam, or 223 degrees of superheat (600 — 377 = 223) equation 120 becomes

$$\frac{\underset{\sim}{Q}'}{\underset{\sim}{Q}} = \frac{461 + 377}{461 + 600} \times \frac{1{,}197 - 60 + .48 \times 223}{1{,}197 - 60} = \frac{838}{1{,}061} \times \frac{1{,}244}{1{,}137}$$

= .865, and the saving, 1 — .865 = .135, or 13.5 per cent. In this manner the coal economy upon a theoretical basis has been calculated for the pressures and temperatures used to state the water economy in the last chapter.

FUEL ECONOMY OF SUPERHEATED STEAM COMPARED TO SATURATED STEAM AT SAME PRESSURE.

Pressure.	175 lbs.	200 lbs.	225 lbs.
Temp. t'	Saving.		
400°	2. Per Cent	1. Per Cent	0. Per Cent
450°	5.5 "	4. "	3. "
500°	9. "	7.5 "	6. "
550°	11 5 "	10. "	9. "
600°	13.5 "	12.5 "	12. "
650°	16. "	15. "	14.5 "
700°	18. "	17. "	16.5 "
750°	20. "	19. "	18.5 "
800°	21.5 "	21. "	20.5 "

The capacity of the fireman must be considered in connection with the quantity of coal fired, but no hard and fast rule can be formulated to limit the capabilities of this personage; it is largely a question of endurance. Coal has been fired at the rate of three tons, or 6,000 pounds an hour, but how long this can be maintained depends entirely upon the sturdiness of the man and the surrounding conditions. This subject was investigated by a committee of the Master Mechanics' Association, and replies indicated that grates of 60 square feet were being supplied with soft coal and from 80 to 95 square feet with hard or anthracite coal, but this gives only a vague idea of the amount of coal handled. As 6,000 pounds an hour means a scoop averaging 16 pounds of coal every 10 seconds, it seems as if this figure was nearly the limit for one man, but with a mechanical stoker, where the coal is merely fed into a hopper, and does not have to be spread over a fire, with perhaps a 10-foot throw and a door to be opened and closed, very much larger amounts can be fed.

EFFECT OF LOAD AND SPEED.

Plate 36 enables us to make a study of the effect of variation in load and speed upon the quantity of fuel used. It is recognized at once that an increase in the load to be hauled will mean an increase in the fuel consumption, both per mile and hour, and also that an increase in speed will cause an increase in the quantity of fuel per hour, but as revenue is usually considered and collected on the basis of ton mileage, it is particularly interesting to discover the law governing fuel consumption upon the basis of tons behind the tender hauled one mile. While faster trains generally command a higher rate for hauling, yet the ton mile is still the basis of comparison generally adopted. Locomotive accounts are also kept largely upon the ton-mile basis, especially the fuel charge, so that it is highly important to understand how the speed and loading affect this account.

If we consider a locomotive with a train, burning so much fuel per mile, and we add 100 tons to this train, it will naturally require more fuel per engine mile or train mile, but for every mile run, there will be 100 ton miles additional hauling performed—if the fuel increase were directly proportional to the weight of train behind tender, the consumption per ton mile would be constant, but as this is not the case, some conditions of loading will be conducive to fuel economy, per ton mile, and others will be the reverse.

To illustrate the solution of this problem by means of plate 36, let us consider a hypothetical locomotive having the following general characteristics:

Theoretical tractive force	25,000 pounds
Weight of engine and tender	100 tons
Grate area	25 square feet
Diameter of drivers	56 inches
Fuel	Bituminous coal

It is also assumed that the boiler and cylinders are so proportioned that the curve of available tractive force on plate 29 applies without sensible error, which also means that plate 36 can be correctly used. Now, suppose that this engine be given 2,000 tons back of tender to haul over a level division, then the total weight of train will be $2,000 + 100 = 2,100$ tons, and if the speed be fixed at 10 miles an hour, the resistance will be $2,100 \times 5.5 = 11,550$ pounds, and the percentage of the theoretical tractive force $= \frac{11,550}{25,000} = 46\%$. With 56-inch drivers there will be 60 revolutions per minute at the speed named, and from plate 36 we find that the intersection of 46 per cent and 60 revolutions corresponds to 34 per cent of the maximum fuel consumption, which will be $25 \times 200 = 5,000$ pounds per hour, and 34 per cent of this amount will be $= .34 \times 5,000 = 1,700$ pounds per hour. As the speed is 10 miles per hour, this will be $\frac{1,700}{10} = 170$ pounds per mile, and as the load hauled is 2,000 tons, we have $\frac{170}{20.00} = 8.5$ pounds of coal per 100 ton miles of train hauled. By repeating this operation we obtain the coal rate for different weights of train back of tender. This has been done and is exhibited in Fig. 105 by the line marked "level." It is seen that the coal per ton mile diminishes slowly as the load is increased, largely due to the fact that the weight of the engine and tender is a smaller proportion of the total weight hauled with increasing train loads, until the late cut-off demanded finally overcomes the reduction due to the diminishing proportion of the engine and tender,

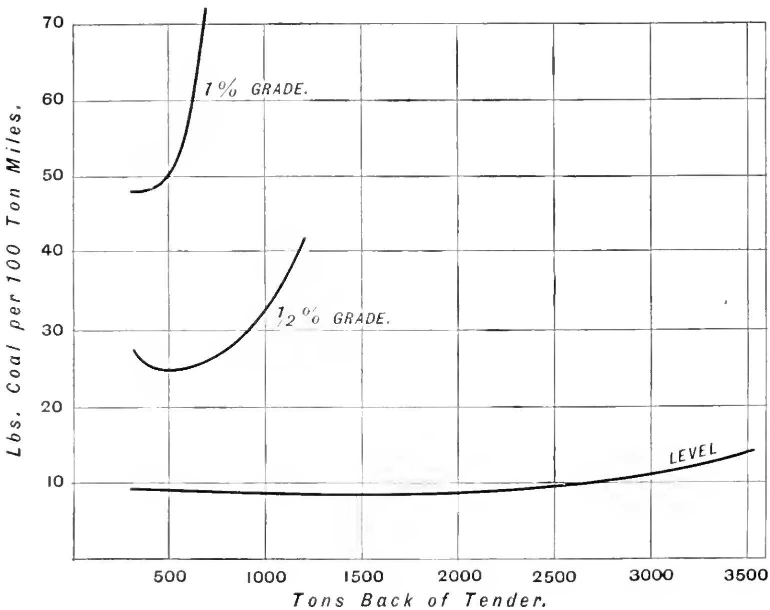


Fig. 105.

and results in requiring an increased fuel consumption. Thus the coal per ton mile will be greater for 3,500 tons than for 2,500 tons by about 50 per cent—at the same time it will be less for 1,500 than for 500 tons, per ton mile. The curves marked ½% and 1% grade show that under these circumstances the increase is much greater, although the proportions between the minimum and maximum rates are nearly the same in the three cases illustrated. Of course, in practice we perhaps never have a uniform gradient throughout the run of the engine, and it must be loaded for the controlling grade; this reduces the loading for the average grade so that the full load for the controlling grade may be the most economical one for the average grade. Thus, if a long level stretch had a few miles of ½ per cent grade, so that the engine could not take over 1,200 tons, the average being nearly level, would indicate a more economical load throughout the run than 500 tons, which would take less coal on the hill portion. To figure the coal on a section of varying gradients, each portion consisting of a different grade must be calculated separately. The speed of each part must be considered in connection with the load; in Fig. 105 all speeds were taken at 10 miles an hour.

Very surprising results often occur from changes in the rating of the engine, at times quite different from what were anticipated. The writer remembers a case on the Chicago & Northwestern where the reduction of a grade increased the fuel rate per ton mile for the division. While this was contrary to the general expectation, it was proved theoretically to be the natural result. As it is an interesting case, the analysis will be given.

The division upon which these conditions existed was 202 miles long, and may be considered as composed of the following sections:

Section.	Length	Average Grade.
A to B	57 miles	5½ feet per mile
B to C	6 "	Level
C to D	14 "	Down grade
D to E	34 "	5¼ feet per mile
E to F	12 "	Down grade
F to G	43 "	6½ feet per mile
G to H	23 "	Down grade
H to I	13 "	18½ feet per mile

The controlling grade was 60 feet per mile, and the original rating was 1,050 tons from A to G and 750 tons from G to I. The coal consumption calculated, as above explained, in detail, was as follows:

Section.	Tonnage.	Ton Miles.	Coal Per 100 Ton Miles.	Total Coal in Pounds.
A to B	1,050	59 800	12 Lbs.	7,180
B to C	1,050	6,300	9 "	567
C to D	1,050	Down grade		
D to E	1,050	35,700	12 "	4,290
E to F	1,050	Down grade		
F to G	1,050	45 100	12 "	5,400
G to H	750	Down grade		
H to I	750	9,750	17 "	1,660
				19,097

$$\text{Total} = 1{,}050 \times 166 \text{ miles} = 174{,}000 \text{ ton miles}$$
$$750 \times 36 \text{ miles} = 27{,}000 \text{ ton miles}$$
$$201{,}000 \text{ ton miles}$$

and $\frac{19{,}097}{201{,}000} = 9.5$ pounds of coal per 100 ton miles for the run over the division.

The grade reductions were effective in both directions, but our computation considers westbound trains only; also while the maximum or controlling grades were reduced, there was little change in the average grade of the various sections. After the revision, the controlling grades were 37 feet per mile, and the train load increased to 1,250 tons. The consumption then figured as below:

Section.	Tonnage.	Ton Miles	Coal Per 100 Ton Miles	Total Coal in Pounds.
A to B..........	1,250	71,500	13 pounds	9,300
B to C..........	1,250	7,500	8½ pounds	640
C to D..........	1,250	Down grade..	....	....
D to E..........	1,250	42,500	13 pounds	5,520
E to F..........	1,250	Down grade...	....	....
F to G..........	1,250	54,000	13 pounds	7,020
G to I..........	1,250	45,000	21 pounds	9,450
				31,930

Totals = 202 × 1,250 = 252,500 ton miles, and $\frac{31{,}930}{252{,}500} =$ 12.6 pounds of coal per 100 ton miles, an increase in the fuel rate of nearly one-third; the section from G to I averaged 18½ feet for the 36 miles under the revision. The expenses of engine and train crews were not increased, nor were most of the other operating charges, so the gross results showed greater economy in operation, but the increase in coal consumption per ton mile caused considerable comment, until it was demonstrated to be a perfectly logical occurrence.

At the present time, the question of speed and its relation to fuel consumption is of particular interest. The cost of running trains at high speed was discussed in connection with a paper by Mr. F. A. Delano by the Western Railway Club in January, 1900. This was principally in connection with passenger service and Mr. F. H. Clark presented some data upon the fuel consumed by such trains. He took into consideration, for a period of six months, four trains operating at speeds varying between 30 and 50 miles per hour, and gave the quantity of coal used in this time.

Train.	Number of Cars.	Average Speed.	Number of Stops.	Mileage.	Tons Coal Used Per Car.
A..............	6.76	31.33	18	34,480	170.38
B..............	6.24	35.70	8	34,480	176.11
C..............	2.95	45.40	7	34,480	369.83
D..............	3.88	48.00	7	31,100	328.35

Comparing trains A and D, we have an increase of over 90 per cent of coal burned for an increase of 53 per cent in speed.

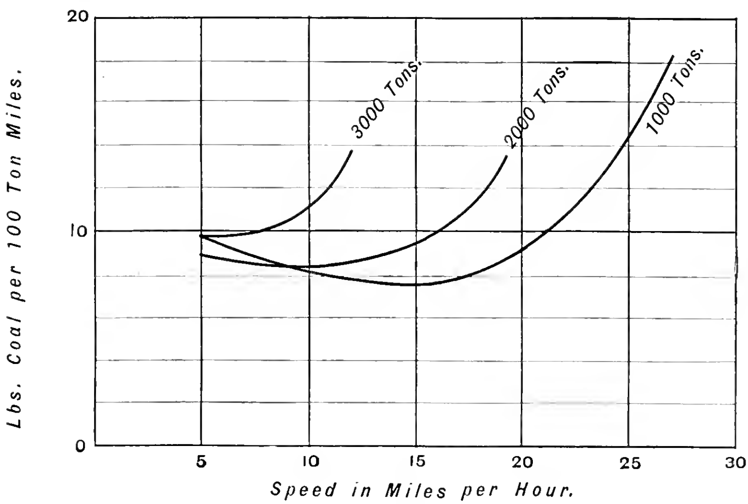


Fig. 106.

Fig. 106 has been constructed in the same way as Fig. 105, using plate 36 as a foundation. The track is supposed to be level in each case, and three trains are illustrated, of 1,000, 2,000 and 3,000 tons back of tender. It will be noticed that the 1,000-ton train requires nearly 10 pounds of coal per 100 ton miles, when the speed is 5 miles per hour, and only 7.7 pounds at 15 miles, while the consumption increases to 14 pounds at 25 miles an hour. At 27 miles the rate is 18 pounds, and this is the limit of speed for the typical locomotive with 1,000 tons back of tender. For this load, it is seen that 15 miles an hour is the most economical speed from a fuel standpoint on a level road. With 2,000 tons, the best rate is found at 10 miles an hour, and with 3,000 tons, at 5 miles.

From this figure, the great excess in fuel consumed in stock and fast freight trains over the slow freights can readily be understood; in the 1,000-ton curve the rate at 25 miles is 84 per cent greater than at 15 miles an hour. This demonstrates the absolute futility of making comparisons or dealing out reprimands based upon the ton mileage rate alone, as the speed element is of great importance in fixing the coal consumption, and a larger proportion of fast trains over a division one month than another will increase the fuel charges without any additional credit to ton mileage handled.

In order to make a careful study of any special case, diagrams should be made to suit the particular engine and conditions involved.

WASTE OF FUEL.

In spite of all the precautions taken to prevent it, the waste of coal on a railroad is very great. Purchased in enormous quantities, and distributed with a lavish hand, the whole tendency is to cause a disregard for the actual value of coal. Some of the wastes were considered in connection with the question of water supply, such as scale forming upon the heating surfaces and preventing the transmission of heat, also the loss of heat due to steam or water leaks in the engine or boiler. These will not be again referred to, as it is evident that leaks or waste of steam or hot water are directly drains upon the coal pile without any benefit being received therefrom. There are large wastes, however, in which the steam takes no part, such as the generation of smoke and carbonic oxide, and the emission of sparks in large quantities, not to mention wastes of coal that never passes the fire door.

Perhaps smoke attracts more attention to the waste in the firebox than any other visible reminder. When a fuel rich in volatile matter is charged into a hot furnace, the light hydrocarbons are distilled from the coal. If some of the carbon is unconsumed, owing to an insufficient supply of air, it passes off as smoke. The amount of free carbon, even in the densest smoke, is small, perhaps not over ½ per cent by weight, as it has great coloring power—the greatest waste is in the production of carbonic oxide, as previously explained, due to a lack of oxygen, and as smoke indicates the shortage of air, it is a valuable guide to the efficiency of the fire. Then the temperature must be high enough to cause the ignition of the coal gas distilled from the coal, and in order to keep the surface of the fire uniformly hot, the coal must be well scattered over the grate, so that the gases will burn as they are distilled and the firebox will not become chilled, even in parts; to insure quick burning, the coal should be broken up into small-sized pieces—not over 3 inches cube—before it is shoveled through the fire door. The fireman should be on the alert to take advantage of the physical conditions of the road, and should fire lightly and regularly, with the door closed gradually—that is, cracked for a few seconds until there has been sufficient air admitted to consume the fresh distillates. The gauge should be constantly observed and the supply of air regulated principally by the dampers. The engineer must cooperate with his fireman by handling the engine in an intelligent manner and by communicating constantly with him, keeping him informed of his intended movements. In order to facilitate the work, fire doors should be at a convenient height and of suitable size, the steam gauge should be in comfortable view, day and night, and blower or smoke consumer valves should be quick acting and convenient of access. Many contrivances are used upon locomotives in order to reduce the smoke in the limits of large cities; these consist chiefly of the brick arch and air injectors, by which jets of air are forced into the firebox through openings in the water legs, although the latter are often more diluters than consumers. The Bates or narrow, constantly open fire door, with or without an inside deflector, is also found advantageous on many roads. However, more depends upon the man with the scoop than any other one or several things combined. Some firemen will supply the fire evenly and carefully without any signs of fatigue, while others will labor at the "one shovel system" until they tire themselves out, and then, in disgust, throw in six or eight and sit down to rest. This is, perhaps, the most difficult problem connected with locomotive operation—to secure and retain competent and efficient firemen—there are plenty of "coal heavers," but a conscientious and intelligent fireman is too often a curiosity. The work is laborious, without doubt, which is all the more reason for providing the men with such comforts and conveniences as are possible.

Much can be done in the matter of supplying uniform qualities of coal to the different divisions—that is, in keeping one kind of coal on one division, and, if necessary, another kind on some other division; this avoids trouble from improper grates and enables the men to accustom themselves to the fuel used.

The foregoing remarks apply principally to bituminous coal, as anthracite does not make smoke—the replenishing of the fire with the latter is generally done while the throttle is closed, when the service or run is such as to permit this method being used, as in trains that make frequent stops.

The waste due to sparks or unburnt fuel passing out of the stack has been already touched upon, and it was seen that 20 per cent of the coal fired sometimes passed off in this form.

While these sparks are coal partially burnt, much heating value is still retained by them when ejected from the stack. The analysis of some sparks from Brazil block coal showed as follows:

Fixed carbon	62 to 76 per cent
Volatile matter	3 to 4 per cent
Moisture	1½ to 2 per cent
Ash	18 to 32 per cent

This indicates, as would be expected, that the light hydrocarbons were driven off, and the fuel partially consumed before it was expelled. The heating value of these sparks ranged from .75 to .91 of that of dry coal of the same weight, so that the actual heat losses were not quite as great as the spark losses. As might be expected, however, the heavier the draft, the greater the heating value of 1 pound of sparks, as well as the greater quantity of sparks ejected. This is another argument for running with as large an exhaust nozzle as possible.

These sparks are not only wasteful of coal, but dangerous to property along the right of way. All railroads are confronted with the question of fire damages, and in the dry western country, the amounts claimed are often very large. Then, too, the lignites and light coals which supply that territory give out great quantities of sparks, so that at nighttime an engine working up a hill becomes the center of a veritable fireworks display. On some roads the old form of diamond stack is used with these lignites, but they can be handled with a straight stack if a good arrangement of netting be provided. One of the great troubles is due to the netting's becoming stopped up with the cinders, and choking the draft, thus causing the engine to lose time. In such cases, in order to get a train over the road, the enginemen often either punch a hole in the netting or drop the manhole. The writer found this such a common practice in the Southwest that he introduced a 3 by 3 mesh instead of a 4 by 4 netting, his theory being that if it did not choke up, there would not exist the temptation to open the trap—this was proved by results, as the fires set out by the engines with the larger netting showed no excess over those with the finer mesh. The sparks in many cases were pointed, and remained wedged in the openings of the netting like small valves, completely closing large sections of the netting. With Illinois and Missouri coal the same road used 2 by 2 mesh, which permitted a large exhaust nozzle and corresponding fuel economy.

In 1902 some experiments were conducted to determine the "firing value" of cinders that would pass through a 2 by 2 netting, the wires being nearly 1/8 inch in diameter. The cinders were heated red hot and shot out of a cannon whose bore was 1½ inches by 6 inches, the charge consisting of 3 or 4 ounces of giant powder. The gun was pointed at an angle of 45 degrees from the horizontal, and in the direction of the wind, which was blowing about 10 miles an hour. The farthest distance to which a glowing cinder was carried was 100 feet. The test was made at night, and cinders were heard to fall 150 feet away, but they were not visible. The gun was also pointed vertically, but all the sparks fell around the cannon within a diameter of 50 feet, some of them glowing for several seconds after landing.

In order to determine the time of cooling, cinders as large as could be pushed through the 2 by 2 netting were heated to a white heat and then poured upon a wire gauze. It was found to take from 5 to 8 seconds for them to cool down to the point where they were just visibly red; a current of air blowing upon them hastened the cooling. Kansas coal was treated in the same way, with the result that 5 seconds was the longest time that any of the pieces continued to give off gas and remain lighted. The cinders heated white hot would not set fire to dry shingles, but did light a piece of paper. These experiments were carried out by the late Mr. Edward Grafstrom and Mr. W. A. Powers, mechanical engineer and chemist, respectively, of the Santa Fe system.

In order to set fire to a light object 100 feet from the track, these experiments indicated that the spark must reach it in 8 seconds or less, and it would require a velocity of 12½ feet per second, or 8½ miles an hour, to do this, so that if the spark traveled with the speed of the wind, there is hardly a probability of firing objects 100 feet away unless the wind be greater than 10 miles an hour.

Of course, other kinds of fuel may, and probably would, produce different results—especially if they are allowed to pass through holes or an imperfectly fitted netting. Here it may be well to state that nettings are often found defective in this respect—a hole large enough to admit the finger sometimes being found in the joints, or where the steam or exhaust pipes require careful fitting, and more than a mere casual inspection is needed to discover these defects.

Professor Goss in his book entitled "Locomotive Sparks" gives the deductions from some experiments made by students of Purdue University to determine the distance which sparks will travel from the track and be a fire menace. The tests were made at the top of a long 1 per cent grade, near Lafayette, Ind., and the cinders were collected in pans located at different distances from the railroad, and provided with a layer of cotton in the bottom, both to prevent the sparks blowing away and also to indicate scorching by the heat of the sparks. The pans were always placed to leeward of the train. A summary of the results is given as follows:

"The greatest number of sparks fell at from 35 to 150 feet from the center of the track.

"With few exceptions, the pans within 20 feet of the track caught few sparks.

"No scorching of the cotton in the pans was observed in any case. (The tests were made in April and May, with the temperature between 60 and 70 degrees Fahrenheit.)

"Beyond 125 feet the sparks were of such a character as to preclude any possibility of fire from them."

Professor Goss concludes that the heat-carrying power of a spark from a locomotive in good order is so small that it is doubtful whether such a spark will set fire to the roof of a building, unless it falls upon materials more finely divided than shingles; also that there is almost absolutely no danger at distances greater than 100 feet from the track; these expressions confirm the results from the Santa Fe tests before reported.

There are fires likely to occur, however, by fire falling out of the ashpan through open dampers. In dry seasons, and especially if there be wooden bridges or trestles on the line, there should be a netting trap fitted to all ashpan openings, and the enginemen should see that this is securely closed and fastened before leaving terminals. It need not interfere with the action of the regular dampers, and should be hinged so that it can be lifted out of the way when the ashpan must be cleaned out.

Some of the wastes of fuel are due to overloading tenders, so that the coal rolls off on the roadbed; also in allowing the fine dirt to accumulate in the bottom of the tender, instead of working it off gradually. The practice of wetting coal is due to an effort to keep down the dust and also prevent the dry, fine stuff passing to the stack without being consumed—in this it is effective, but no more water should be used than necessary to effect this purpose, as all such water must be evaporated in the firebox and absorbs otherwise useful heat.

OIL BURNING.

Fuel oil is used on locomotives in this country for two reasons; one to avoid smoke and the other to reduce the cost of fuel. In the first category, the Boston & Maine is perhaps the most prominent example. The Hoosac Tunnel is nearly 5 miles long, and the grades are 26 feet to the mile in each direction, the summit being near the center of the tunnel. The heavy freight traffic passing through made it impossible to maintain it free from smoke, even with a ventilating shaft and fan connected with the middle of the tunnel. Several helpers were fitted up for oil, and while in the tunnel no fresh coal is put in the firebox of the road engines (which still burn coal), the oil burner being depended upon to do most of the work, and so keep the tunnel clear. In this case, there were no considerations of fuel economy, as the oil is much more costly than the equivalent amount of coal.

With the Santa Fe and Southern Pacific railways, however, the reverse is true; the oil fields of Beaumont and Sour Lake in Texas, and of Bakersfield and Olinda in California are producing oil at such a rate that it is much cheaper as a heating agent than coal; moreover, the hot climate of parts of these states makes it much easier upon the firemen. The heating value of a ton of oil is much greater than that of a ton of coal, so that less weight must be carried to produce the same quantity of steam. Practical tests bear out the theoretical economic value of oil. When Mr. Urquhart commenced using petroleum refuse on the Grazi-Tsaritzin Railway in Southeastern Russia, the attention of railroads in this country was directed toward his experinments. Dr. C. B. Dudley inspected the locomotives in this service, and later delivered a lecture before the Franklin Institute on the subject; about this time the Pennsylvania Railroad also experimented, but the price of oil made its adoption prohibitive.

The heat-producing power of fuel oil is generally stated at from 1.4 to 1.8 that of coal, or that 1.4 or 1.8 pounds of coal are required to generate the same amount of steam as is produced by a pound of oil. But there are many kinds of coal, and this is rather indefinite. Good bituminous coal is credited with producing from 13,000 to 14,000 heat units for each pound burned, and petroleum from 19,000 to 20,000, or about 50 per cent more. Tests upon the Southern Pacific in January, 1902, between Los Angeles and Indio showed an evaporation of from 13 to 14 pounds of water per pound of oil from and at 212 degrees; this is certainly 50 per cent better than is ordinarily obtained from coal. Again, tests made on the Santa Fe between Needles and Bagdad, Cal., demonstrated that 159 pounds of oil hauled as much as 356 pounds of coal, or the oil was twice as effective as the coal. However, the coal in this test was a New Mexican lignite from Gallup, and by plate 34 we find that its water rate is about $7\frac{1}{2}$ against 10 for fair grades of bituminous coal, so that we still have $\frac{2 \times 7.5}{10} =$ 1.5 times the value of a good grade of coal. The oil ran about six barrels to the ton, so that four barrels, or 1,333 pounds, were usually estimated to be thermally equivalent to a ton of coal.

Tests were also made on the Santa Fe to determine the relative thermal values of the Kern County and Olinda oils, as well as the California and Texas oil. The comparison of California oils is here given:

Pounds of Water Evaporated Per Pound of Oil.

Boiler.	Olinda.	Kern. Co.
Stationary	11.23	11.12
Locomotive	10.65	10.67
Gravity (Baumé at 60° F.)...	17.7 to 19.0	12.1 to 13.5

In the test between California (Olinda) and Texas (Beaumont) oils, the following results were obtained with a consolidation locomotive:

POUNDS OF WATER EVAPORATED PER POUND OF OIL FROM AND AT 212 DEGREES FAHRENHEIT.

Oil.	Trip 1.	Trip. 2.	Trip 3.	Trip 4.	Average.
Texas..........	12.87	13.39	13.26	13.00	13.1
California......	13.29	13.98	13.77	13.13	13.5

The gravity of the Texas oil was 21.5 and the California oil 15.5 degrees Baumé, or 7.64 and 7.71 pounds per gallon, respectively. While the result shows slightly in favor of the California oil, as a general thing it is safe to consider them all of equal heating value.

The rate of combustion does not have to be based on grate area as with coal, as no grate is used. It has been found possible to burn oil at the rate of nearly 1½ pounds per square foot of heating surface per hour, and to obtain an evaporation of 12 or 13 pounds of water from and at 212 degrees. Just what is the limit we do not know, but this rate produces 25 per cent more steam with the same boiler than the average coal used at its maximum rate. It is a well-known fact that engines burning oil will haul their trains at a faster rate than coal burners of the same size—in fact, it has been a surprise to visitors to see the speed at which the heavy passenger trains (10 or 12 cars, half of them Pullmans) were taken up the long 95-foot grade from the Colorado River to the Arizona summit at Yampai, nearly one mile high.

BURNERS.

Nearly all the oil burners or atomizers used on locomotives in this country are similar in their general characteristics. The oil flows over a flat surface or trough, from 2 to 6 inches wide, from which it is blown, as it flows over the end, by a blast of steam about the same width or slightly wider, and from 1-32 to 3-32 inch in thickness. In some cases, the mixing of steam and oil is done in the burner, and the spray emerges from a slot the width of the burner and from ¼ to ⅜ inch thick. Mixtures of air and steam prior to atomization are special features of some makes, but the flat spray is nearly universal in this country. The original Russian burners had round nozzles. It seems as if a 3 or 4 inch burner was wide enough for the largest engines, and any additional width caused a waste of oil—indeed, the burner should not be larger than will properly supply the boiler, so as to prevent overheating of the firebox sheets. When the proper adjustment of the oil and steam valves has been made, there will be no smoke produced—merely a gray haze from the stack, but a change in the opening of the throttle or the position of the reverse lever demands an instant change in the oil supply. If less steam be used, the smokebox vacuum will be less, and the supply of air diminished, resulting in dense smoke from the stack, unless the oil be first shut off or reduced, supplying just enough to burn in the smaller quantity of air drawn in. If the supply of oil be insufficient, the pointer on the steam gauge will fall at once. The proper regulation maintains the steam pressure without emitting smoke. Frequently (about every 20 or 30 minutes) several quarts of fine sand must be introduced through a hole in the firedoor by means of a funnel, when the engine is working hard, in order to scour the soot off the flues. This causes dense smoke for a few minutes, but it soon returns to the gray haze. When the engine is standing, the oil valve needs to be barely cracked, in order to maintain pressure, and when running down hill, it is important to close the dampers, otherwise cold air will be drawn in, and ruin the firebox; for this reason, the dampers should be easily and conveniently manipulated. It is well to have a removable stop in the handle of the oil valve, which will definitely fix the proper position for standing or drifting, but which can be removed and allow the complete closing of the valve when the engine is cooled off. If the fire accidentally is extinguished while on the road, it is at once indicated by puffs of yellow smoke from the stack.

ARRANGEMENT.

Since the commencement of oil fuel in locomotives, the burner has been placed just below the mud ring, the jet being directed under a firebrick arch, built up from a shallow pan or bottom, with an opening for air, perhaps 12 inches square, just below the arch; another opening is also placed near the burner —both of these openings must be controlled by dampers, readily manipulated from the cab. The life of these arches is very variable—sometimes they last one week, sometimes two months. The jarring motion of the engine helps to destroy them even if they do not burn out. For this reason the Galveston, Houston & San Antonio Railroad has been experimenting with checker flash walls, and has succeeded in operating engines with such an arrangement for six months to a set of brick—the wall costs only about $8 to construct, whereas the arches cost $30. In both cases the sides of the firebox are lined with firebrick as high as the arch or top of wall, and the bottom is covered with brick front and back of the arch or wall, except at the air openings noted. A steam chamber is used to heat the oil in cold weather before it enters the burner, and render it more limpid.

The oil tank on the tender has a capacity of 2,000 gallons or more, for road engines, and was formerly braced so that about 10 pounds air pressure could be kept on top of the oil, causing it to flow readily to the burner. Recently, however, this pressure has been considered unnecessary. The tanks are fitted with automatic safety valves, with a wire rope or chain connection to back of engine cab, and so arranged that if the engine and its tender become separated on the road, the chain pulls out a key, allowing the valve to close immediately, under the action of a coil spring, thus stopping the flow of oil. The oil in the tank is warmed by steam from the engine; this is accomplished either by a coil of pipe in the bottom of the tank, termed an "indirect heater" or by blowing the steam directly into the oil, called a "direct heater;" in the last method, the water of condensation mixes with the oil, and arrangements must be made for draining it from the bottom of the tank, underneath the oil. As all the commercial oil contains more or less water, this draining is always important. When water passes over into the burner mixed with the oil, it causes an instantaneous extinguishment of the flame; the following oil is immediately relighted by the red hot arch bricks, and a series of small explosions is thus caused, one accompanying each "slug" of water passing through the burner.

OPERATION.

In firing up an oil burner, provided it be located where steam pressure can be obtained for use as a blower, a piece of lighted greasy waste should be thrown in the firebox; the oil valve should then be opened slightly, after which the atomizer or steam valve should be turned on enough to spray the oil, when the vapor will instantly ignite. It is dangerous to start the burner before applying the flame, as the firebox becomes filled with oil vapor which will explode with violence. If there be no steam available, then a wood fire must be made until enough steam is generated to operate the burner.

It is very important that the brickwork be maintained in perfect condition; occasionally a small piece of brick will fall and lodge in front of the burner, which will seriously interfere with the steaming, and it must be promptly removed. When the flues are sanded, the engineer should drop the lever into the corner notch, and pull the throttle wide open, for a few revolutions, in order to cause the sand to scour the flues by means of the heavy draft.

In handling oil burners on the road, the engineer and fireman must work in harmony; when the engineer intends closing the throttle, he should advise the fireman in time so that the latter can first reduce the oil supply, avoiding smoke and waste; the same applies when starting. In feeding the oil, the valve should be gradually opened wider as the working of the engine becomes harder.

It is not safe to approach an open manhole of the oil tank with a light closer than 10 feet; if it be desired to determine the height of oil in tank, a stick or rod should be inserted, and carried to the light. Should it be necessary to do any work inside of the oil tanks after they are empty, they should be first steamed out, and then washed out with cold water, before a lighted torch of any kind is taken near the opening. This is to ensure a cleansing of the tanks of all the gases which they may contain. No one should attempt to enter one of the tanks until it has been steamed and washed out as above. The vapor given off by the Texas oil is poisonous as well as explosive, and men have died in one of these empty tanks in a few moments.

ADVANTAGES AND DISADVANTAGES.

In California and Texas, the principal advantage of oil burning is the decreased cost compared with coal. This is especially true in California, where one dollar's worth of oil produces as much heat as four or five dollars' worth of coal; in

Texas, the gain is not so large. The cost of changing a coal burner into an oil burner will average about $500—somewhat more or less, depending upon the size of oil tanks, etc. The absence of smoke (if properly handled) and cinders makes it pleasanter for passengers and prevents fire in the surrounding country. This is especially important in very dry sections, like Arizona and New Mexico, where fires are so easily started. The cost of handling fuel is at least 75 per cent less than coal, and there are no ashes or clinkers to delay the engines on the road or to consume time at terminals. A large engine is as easily fired as a small one, and in a country where the summer temperature is frequently 120 degrees Fahrenheit in the shade, this has considerable importance.

On the other hand, there are numerous disadvantages. The intense heat of the oil jet is much harder on fireboxes; in heavy service, they will often burn out in two years. The repairs in this period are also much greater, they having been estimated at three times as heavy and costly as with coal burners. Rivet heads and button heads of crownbolts burn off—to such an extent is this true, that countersunk rivets and crownstays are now quite generally used. The latter reduces the protection against dropping crownsheets, but in no other way can trouble with these bolts be obviated. The single-lap seams crack and melt off owing to the double thickness of metal, and it is a common practice to cover such seams with a strip of firebrick held in place by studs. The intense heat at times, drives the water from the sheets by the rapid formation of steam, to the quick destruction of the plates. It is useless to apply patches or half sidesheets to oil burners, unless they can be covered with firebrick where the metal is of double thickness, as they will be a continual source of trouble—this is the reason that new fireboxes must be applied so often, especially with engines in hard service.

They also consume much time at terminals when washing out is necessary, or boiler makers must enter the firebox for repairs. The heat of the brickwork is so great that it requires about 12 hours to cool down, wash out or repair and fire up again, whereas coal burners can be treated in about one-half this time. Burners have been devised which it was hoped would increase the life of the firebox, but these have so far been deficient in steaming qualities. The greater amount of work which can be gotten out of an oil burner encourages over-loading and high speeds, which quickly cause deterioration of the firebox, necessitating heavy and costly repairs.

APPENDIX.

TABLES OF CIRCULAR, TUBULAR, RECTANGULAR AND I-SECTIONS.

The headings of the columns designate as follows:

A = Area of Section.
I = Moment of Inertia, Horizontal Axis.
S = Modulus of Section, Horizontal Axis.
R = Radius of Gyration, Horizontal Axis.
i = Moment of Inertia, Vertical Axis.
r = Radius of Gyration, Vertical Axis.
W = Width of Rectangular or I-Section.
T = Thickness of Flange, I-Section.

TABLES FOR CIRCLES.

Diam.	A	I	S	R
2	3.14	.78	.78	.50
2¼	3.98	1.28	1.12	.56
2½	4.91	1.92	1.53	.62
2¾	5.94	2.80	2.05	69
3	7.07	3.97	2.66	.75
3¼	8.30	5.40	3.38	.81
3½	9.62	7.36	4.21	.87
3¾	11.05	9.62	5.18	.93
4	12.57	12.57	6.28	1.00
4¼	14.19	15.92	7.53	1.06
4½	15.90	20.15	8.95	1.12
4¾	17.72	25.05	10.55	1.19
5	19.64	30.69	12.28	1.25

TABLES FOR TUBES.

O. Diam.	Thickness	A	I	S	R
2	¼	1.37	.53	.53	.62
	⅜	1.91	.66	.66	.59
	½	2.36	.73	.73	.56
2¼	¼	1.57	.82	.73	.72
	⅜	2.21	1.03	.92	.68

TABLES FOR TUBES—Continued.

O. Diam.	Thickness	A	I	S	R
2¼	½	2.75	1.16	1.03	.65
	⅝	3.20	1.23	1.09	.62
2½	¼	1.77	1.14	.91	.80
	⅜	2.51	1.46	1.17	.76
	½	3.14	1.67	1.34	.73
	⅝	3.69	1.80	1.44	.70
	¾	4.13	1.87	1.50	.67
2¾	¼	1.96	1.52	1.10	.88
	⅜	2.80	2.02	1.47	.85
	½	3.54	2.34	1.70	.81
	⅝	4.17	2.55	1.85	.78
	¾	4.71	2.68	1.95	.75
	⅞	5.16	2.75	2.00	.73
3	¼	2.16	2.07	1.38	.98
	⅜	3.09	2.69	1.79	.93
	½	3.93	3.19	2.13	.90
	⅝	4.67	3.51	2.34	.87
	¾	5.30	3.72	2.48	.84
	⅞	5.84	3.85	2.57	.81
	1	6.29	3.92	2.61	.79
3¼	¼	2.36	2.60	1.60	1.05
	⅜	3.39	3.48	2.14	1.01
	½	4.32	4.12	2.54	.98
	⅝	5.16	4.62	2.84	.95
	¾	5.90	4.94	3.04	.92
	⅞	6.53	5.15	3.17	.89
	1	7.07	5.28	3.25	.86
3½	¼	2.55	3.39	1.94	1.15
	⅜	3.68	4.56	2.61	1.11
	½	1.71	5.44	3.11	1.07
	⅝	5.64	6.08	3.48	1.04
	¾	6.48	6.58	3.76	1.01
	⅞	7.26	6.90	3.94	.98
	1	7.85	7.11	4.06	.95
3¾	¼	2.75	4.22	2.25	1.24
	⅜	3.98	5.65	3.02	1.19
	½	5.11	6.82	3.64	1.15
	⅝	6.14	7.70	4.11	1.12
	¾	7.07	8.34	4.15	1.09
	⅞	7.91	8.81	4.72	1.06
	1	8.61	9.16	1.88	1.03
4	¼	2.95	5.21	2.61	1.33
	⅜	1.27	7.17	3.58	1.29
	½	5.50	8.60	4.30	1.25
	⅝	6.63	9.77	1.88	1.21
	¾	7.66	10.60	5.30	1.17
	⅞	8.59	11.29	5.64	1.14
	1	9.43	11.79	5.89	1.12
4¼	¼	3.14	6.30	2.96	1.41
	⅜	4.57	8.56	4.03	1.37
	½	5.89	10.52	4.96	1.33
	⅝	7.12	11.95	5.63	1.29
	¾	8.25	13.12	6.18	1.26
	⅞	9.28	14.00	6.60	1.23

TABLES FOR TUBES—Continued.

O. Diam.	Thickness	A	I	S	R
4¼	1	10.21	14.64	6.90	1.20
4½	¼	3.33	7.58	3.38	1.50
	⅜	4.85	10.52	4.69	1.47
	½	6.28	12.79	5.70	1.43
	⅝	7.60	14.75	6.57	1.39
	¾	8.83	16.18	7.20	1.35
	⅞	9.96	17.35	7.72	1.32
	1	10.99	18.23	8.12	1.29
4¾	¼	3.53	9.13	3.85	1.61
	⅜	5.15	12.48	5.26	1.56
	½	6.67	15.43	6.52	1.52
	⅝	8.10	17.69	7.46	1.48
	¾	9.42	19.95	8.42	1.44
	⅞	10.64	21.08	8.88	1.40
	1	11.78	22.25	9.39	1.37
5	¼	3.74	10.54	4.22	1.67
	⅜	5.45	14.77	5.92	1.64
	½	7.07	18.12	7.26	1.60
	⅝	8.59	21.07	8.45	1.56
	¾	10.02	23.33	9.35	1.52
	⅞	11.34	25.29	10.12	1.49
	1	12.57	26.72	10.70	1.46

TABLES FOR RECTANGLES.

			Horizontal Axis.			Vertical Axis.	
Height	W	A	I	S	R	i	r
4	2	8.	10.66	5.33	1.15	2.66	0.58
	2¼	9.	12.00	6.00	1.15	3.79	0.65
	2½	10.	13.33	6.66	1.15	5.20	0.72
	2¾	11.	14.66	7.33	1.15	6.94	0.79
	3	12.	16.00	8.00	1.15	9.00	0.87
	3¼	13.	17.33	8.66	1.15	11.45	0.94
	3½	14.	18.66	9.33	1.15	14.31	1.01
	3¾	15.	20.00	10.00	1.15	17.58	1.08
	4	16.	21.33	10.66	1.15	21.33	1.15
4¼	2	8.50	12.78	6.02	1.22	2.84	0.58
	2¼	9.56	14.38	6.77	1.22	4.03	0.65
	2½	10.62	15.98	7.53	1.22	5.53	0.72
	2¾	11.68	17.58	8.28	1.22	7.37	0.79
	3	12.75	19.18	9.03	1.22	9.56	0.87
	3¼	13.81	20.78	9.78	1.22	12.17	0.94
	3½	14.88	22.38	10.54	1.22	15.20	1.01
	3¾	15.94	23.98	11.29	1.22	18.68	1.08
	4	17.00	25.58	12.05	1.22	22.66	1.15
4½	2	9.00	15.20	6.77	1.30	3.00	0.58
	2¼	10.12	17.10	7.61	1.30	4.27	0.65
	2½	11.25	19.00	8.46	1.30	5.86	0.72
	2¾	12.37	20.90	9.30	1.30	7.80	0.79
	3	13.50	22.80	10.13	1.30	10.12	0.87
	3¼	14.62	24.70	10.98	1.30	12.88	0.94
	3½	15.75	26.60	11.82	1.30	16.10	1.01

TABLES FOR RECTANGLES—Continued.

			—Horizontal Axis.—			—Vertical Axis.—	
Height	W	A	I	S	R	i	r
4½	3¾	16.87	28.50	12.67	1.30	19.78	1.08
	4	18.00	30.40	13.50	1.30	24.00	1.15
4¾	2	9.50	17.87	7.53	1.37	3.17	0.58
	2¼	10.69	20.10	8.45	1.37	4.50	0.65
	2½	11.88	22.35	9.42	1.37	6.18	0.72
	2¾	13.07	24.55	10.33	1.37	8.23	0.79
	3	14.25	26.80	11.27	1.37	10.68	0.87
	3¼	15.44	29.00	12.20	1.37	13.60	0.94
	3½	16.63	31.25	13.15	1.37	17.00	1.01
	3¾	17.81	33.55	14.12	1.37	20.88	1.08
	4	19.00	35.75	15.02	1.37	25.33	1.15
5	2	10.00	20.83	8.33	1.44	3.33	0.58
	2¼	11.25	23.40	9.35	1.44	4.74	0.65
	2½	12.50	26.05	10.42	1.44	6.50	0.72
	2¾	13.75	28.60	11.45	1.44	8.67	0.79
	3	15.00	31.25	12.50	1.44	11.24	0.87
	3¼	16.25	33.80	13.55	1.44	14.31	0.94
	3½	17.50	36.45	14.60	1.44	17.90	1.01
	3¾	18.75	39.00	15.60	1.44	21.98	1.08
	4	20.00	41.65	16.65	1.44	26.66	1.15
5¼	2	10.50	24.10	9.22	1.52	3.50	0.58
	2¼	11.81	27.10	10.33	1.52	4.98	0.65
	2½	13.13	30.15	11.48	1.52	6.83	0.72
	2¾	14.44	33.15	12.63	1.52	9.10	0.79
	3	15.75	36.15	13.77	1.52	11.81	0.87
	3¼	17.06	39.20	14.91	1.52	15.02	0.94
	3½	18.37	42.20	16.08	1.52	18.80	1.01
	3¾	19.69	45.20	17.25	1.52	23.08	1.08
	4	21.00	48.20	18.35	1.52	28.00	1.15
5½	2	11.00	27.75	10.10	1.59	3.66	0.58
	2¼	12.38	31.20	11.37	1.59	5.21	0.65
	2½	13.75	34.70	12.60	1.59	7.16	0.72
	2¾	15.12	38.10	13.90	1.59	9.54	0.79
	3	16.50	41.60	15.15	1.59	12.37	0.87
	3¼	17.88	45.10	16.43	1.59	15.73	0.94
	3½	19.25	48.50	17.65	1.59	19.70	1.01
	3¾	20.62	52.00	18.93	1.59	24.18	1.08
	4	22.00	55.50	20.20	1.59	29.33	1.15
5¾	2	11.50	31.70	11.03	1.66	3.84	0.58
	2¼	12.93	35.60	12.37	1.66	5.45	0.65
	2½	14.37	39.60	13.80	1.66	7.48	0.72
	2¾	15.80	43.50	15.15	1.66	9.97	0.79
	3	17.25	47.50	16.55	1.66	12.93	0.87
	3¼	18.68	51.50	17.90	1.66	16.45	0.94
	3½	20.12	55.50	19.35	1.66	20.60	1.01
	3¾	21.56	59.50	20.70	1.66	25.28	1.08
	4	23.00	63.40	22.06	1.66	30.66	1.15
6	2	12.00	36.00	12.00	1.73	4.00	0.58
	2¼	13.50	40.50	13.50	1.73	5.68	0.65
	2½	15.00	45.00	15.00	1.73	7.81	0.72
	2¾	16.50	49.50	16.50	1.73	10.40	0.79
	3	18.00	54.00	18.00	1.73	13.50	0.87
	3¼	19.50	58.50	19.50	1.73	17.18	0.94

TABLES FOR RECTANGLES—Continued.

Height	W	A	I (Horizontal Axis)	S (Horizontal Axis)	R (Horizontal Axis)	i (Vertical Axis)	r (Vertical Axis)
6	3½	21.00	63.00	21.00	1.73	21.50	1.01
	3¾	22.50	67.50	22.50	1.73	26.38	1.08
	4	24.00	72.00	24.00	1.73	32.00	1.15
6¼	2	12.50	40.70	13.05	1.80	4.16	0.58
	2¼	14.06	45.80	14.65	1.80	5.92	0.65
	2½	15.63	50.90	16.30	1.80	8.13	0.72
	2¾	17.19	56.00	17.92	1.80	10.83	0.79
	3	18.76	61.10	19.55	1.80	14.06	0.87
	3¼	20.32	66.20	21.20	1.80	17.89	0.94
	3½	21.88	71.30	22.85	1.80	22.40	1.01
	3¾	23.44	76.40	24.45	1.80	27.48	1.08
	4	25.00	81.50	26.10	1.80	33.33	1.15
6½	2	13.00	45.75	14.08	1.88	4.33	0.58
	2¼	14.62	51.45	15.80	1.88	6.16	0.65
	2½	16.25	57.15	17.60	1.88	8.46	0.72
	2¾	17.88	62.85	19.35	1.88	11.28	0.79
	3	19.50	68.65	21.15	1.88	14.62	0.87
	3¼	21.12	74.45	22.90	1.88	18.60	0.94
	3½	22.75	80.15	24.70	1.88	23.30	1.01
	3¾	24.37	85.85	26.40	1.88	28.58	1.08
	4	26.00	91.55	28.16	1.88	34.66	1.15
6¾	2	13.50	51.25	15.18	1.95	4.50	0.58
	2¼	15.18	57.62	17.10	1.95	6.39	0.65
	2½	16.86	64.05	19.00	1.95	8.79	0.72
	2¾	18.55	70.50	20.90	1.95	11.70	0.79
	3	20.25	76.95	22.80	1.95	15.18	0.87
	3¼	21.93	83.35	24.70	1.95	19.31	0.94
	3½	23.62	89.70	26.60	1.95	24.15	1.01
	3¾	25.31	96.05	28.50	1.95	29.68	1.08
	4	27.00	102.50	30.35	1.95	36.00	1.15
7	2	14.00	57.20	16.35	2.02	4.67	0.58
	2¼	15.75	64.35	18.40	2.02	6.63	0.65
	2½	17.50	71.50	20.45	2.02	9.12	0.72
	2¾	19.25	78.65	22.50	2.02	12.13	0.79
	3	21.00	85.80	24.48	2.02	15.74	0.87
	3¼	22.75	92.95	26.53	2.02	20.02	0.94
	3½	24.50	100.10	28.60	2.02	25.05	1.01
	3¾	26.25	107.25	30.65	2.02	30.78	1.08
	4	28.00	114.40	32.70	2.02	37.33	1.15

TABLES FOR I-SECTIONS, 4 INCHES HIGH.

(½-inch Web.)

W	T	A	I (Horizontal Axis)	S (Horizontal Axis)	R (Horizontal Axis)	i (Vertical Axis)	r (Vertical Axis)
2	½	3.50	7.29	3.65	1.44	.70	.45
	1	5.00	9.66	4.83	1.39	1.35	.52
	1½	6.50	10.54	5.27	1.27	2.01	.56
2¼	½	3.75	8.07	4.03	1.46	.98	.51
	1	5.50	10.83	5.41	1.40	1.92	.59

TABLES FOR I-SECTIONS, 4 INCHES HIGH—Continued.

(½-inch Web.)

			—Horizontal Axis.—			—Vertical Axis.—	
W	T	A	I	S	R	i	r
2¼	1½	7.25	11.85	5.92	1.28	2.85	.63
2½	½	4.00	8.83	4.41	1.49	1.33	.58
	1	6.00	12.00	6.00	1.41	2.63	.66
	1½	8.00	13.16	6.58	1.28	3.92	.70
2¾	½	4.25	9.61	4.80	1.50	1.76	.64
	1	6.50	13.16	6.58	1.42	3.49	.73
	1½	8.75	14.47	7.23	1.29	5.21	.77
3	½	4.50	10.38	5.19	1.52	2.28	.71
	1	7.00	14.33	7.16	1.43	4.52	.80
	1½	9.50	15.79	7.89	1.29	6.75	.84
3¼	½	4.75	11.15	5.57	1.53	2.89	.78
	1	7.50	15.50	7.75	1.44	5.74	.88
	1½	10.25	17.10	8.55	1.29	8.59	.91
3½	½	5.00	11.92	5.96	1.54	3.59	.85
	1	8.00	16.66	8.33	1.44	7.15	.95
	1½	11.00	18.41	9.20	1.29	10.70	.99
3¾	½	5.25	12.70	6.35	1.55	4.43	.92
	1	8.50	17.83	8.91	1.44	8.82	1.02
	1½	11.75	19.73	9.86	1.29	13.21	1.06
4	½	5.50	13.47	6.73	1.56	5.36	.99
	1	9.00	19.00	9.50	1.45	10.68	1.09
	1½	12.50	21.01	10.52	1.30	16.01	1.13

TABLES FOR I-SECTIONS, 4¼ INCHES HIGH.

(½-inch Web.)

			—Horizontal Axis.—			—Vertical Axis.—	
W	T	A	I	S	R	i	r
2	½	3.63	8.49	4.00	1.53	.70	.44
	1	5.13	11.36	5.35	1.49	1.35	.51
	1½	6.63	12.54	5.90	1.37	2.01	.55
2¼	½	3.88	9.38	4.42	1.55	.98	.50
	1	5.63	12.72	5.99	1.50	1.92	.58
	1½	7.37	14.10	6.64	1.38	2.85	.62
2½	½	4.12	10.26	4.83	1.58	1.33	.57
	1	6.12	14.08	6.63	1.52	2.63	.65
	1½	8.12	15.66	7.37	1.39	3.92	.70
2¾	½	4.37	11.15	5.25	1.60	1.76	.63
	1	6.62	15.45	7.28	1.53	3.49	.72
	1½	8.87	17.22	8.11	1.39	5.21	.77
3	½	4.63	12.03	5.66	1.61	2.28	.70
	1	7.13	16.81	7.91	1.53	4.52	.79
	1½	9.63	18.77	8.83	1.39	6.75	.84
3¼	½	4.88	12.93	6.08	1.62	2.89	.77
	1	7.63	18.18	8.55	1.54	5.74	.87
	1½	10.37	20.33	9.56	1.40	8.59	.91
3½	½	5.13	13.80	6.50	1.64	3.59	.84
	1	8.13	19.54	9.20	1.55	7.15	.94
	1½	11.13	21.89	10.30	1.40	10.70	.98
3¾	½	5.38	14.69	6.92	1.65	4.43	.91

TABLES FOR I-SECTIONS, 4¼ INCHES HIGH—Continued.

(½-inch Web.)

W	T	A	Horizontal Axis. I	S	R	Vertical Axis. i	r
3¾	1	8.63	20.90	9.84	1.56	8.82	1.01
	1½	11.88	23.45	11.05	1.40	13.21	1.05
4	½	5.62	15.58	7.33	1.66	5.36	.98
	1	9.12	22.26	10.47	1.56	10.68	1.08
	1½	12.62	25.01	11.78	1.41	16.01	1.12

TABLES FOR I-SECTIONS, 4½ INCHES HIGH.

(½-inch Web.)

W	T	A	Horizontal Axis. I	S	R	Vertical Axis. i	r
2	½	3.75	9.85	4.37	1.62	.71	.43
	1	5.25	13.25	5.89	1.59	1.36	.51
	1½	6.75	14.78	6.57	1.48	2.02	.55
2¼	½	4.00	10.86	4.83	1.65	.99	.49
	1	5.75	14.82	6.60	1.61	1.93	.58
	1½	7.50	16.61	7.39	1.49	2.86	.62
2½	½	4.25	11.87	5.28	1.67	1.34	.57
	1	6.25	16.39	7.28	1.62	2.64	.65
	1½	8.25	18.44	8.20	1.50	3.93	.69
2¾	½	4.50	12.88	5.73	1.69	1.77	.62
	1	6.75	17.97	8.00	1.63	3.50	.72
	1½	9.00	20.27	9.02	1.50	5.22	.76
3	½	4.75	13.88	6.17	1.71	2.29	.69
	1	7.25	19.55	8.69	1.64	4.53	.79
	1½	9.75	22.10	9.83	1.50	6.76	.83
3¼	½	5.00	14.90	6.63	1.72	2.90	.76
	1	7.75	21.12	9.38	1.65	5.75	.86
	1½	10.50	23.93	10.62	1.51	8.60	.90
3½	½	5.25	15.91	7.08	1.74	3.60	.83
	1	8.25	22.69	10.08	1.66	7.16	.93
	1½	11.25	25.76	11.45	1.51	10.71	.98
3¾	½	5.50	16.92	7.53	1.75	4.44	.90
	1	8.75	24.27	10.78	1.67	8.83	1.00
	1½	12.00	27.59	12.26	1.51	13.22	1.05
4	½	5.75	17.93	7.97	1.76	5.37	.97
	1	9.25	25.84	11.50	1.67	10.69	1.07
	1½	12.75	29.42	13.09	1.52	16.02	1.12

TABLES FOR I-SECTIONS, 4¾ INCHES HIGH.

(½-inch Web.)

W	T	A	Horizontal Axis. I	S	R	Vertical Axis. i	r
2	½	3.88	11.27	4.74	1.70	.71	.42
	1	5.38	15.27	6.42	1.69	1.36	.50
	1½	6.88	17.20	7.24	1.58	2.02	.54
2¼	½	4.13	12.40	5.22	1.73	.99	.49
	1	5.88	17.07	7.18	1.70	1.93	.57

TABLES FOR I-SECTIONS, 4¾ INCHES HIGH—Continued.

(½-inch Web.)

W	T	A	Horizontal Axis. I	S	R	Vertical Axis. i	r
2¼	1½	7.63	19.32	8.13	1.59	2.86	.61
2½	½	4.38	13.55	5.70	1.75	1.34	.56
	1	6.38	18.88	7.95	1.72	2.64	.64
	1½	8.38	21.46	9.04	1.60	3.93	.69
2¾	½	4.63	14.65	6.17	1.78	1.77	.61
	1	6.88	20.65	8.70	1.73	3.50	.71
	1½	9.13	23.55	9.92	1.61	5.22	.76
3	½	4.88	15.80	6.65	1.80	2.29	.68
	1	7.38	22.47	9.47	1.74	4.53	.78
	1½	9.88	25.69	10.82	1.61	6.76	.83
3¼	½	5.13	16.90	7.11	1.81	2.90	.75
	1	7.88	24.24	10.20	1.75	5.75	.85
	1½	10.63	27.78	11.70	1.62	8.60	.90
3½	½	5.38	18.05	7.60	1.83	3.60	.82
	1	8.38	26.05	10.97	1.76	7.16	.92
	1½	11.38	29.91	12.59	1.62	10.71	.97
3¾	½	5.63	19.25	8.10	1.85	4.44	.89
	1	8.88	27.92	11.76	1.77	8.83	.99
	1½	12.13	32.10	13.53	1.63	13.22	1.04
4	½	5.88	20.35	8.57	1.86	5.37	.96
	1	9.38	29.68	12.50	1.78	10.69	1.06
	1½	12.88	34.19	14.40	1.63	16.02	1.11

TABLES FOR I-SECTIONS, 5 INCHES HIGH.

(½-inch Web.)

W	T	A	Horizontal Axis. I	S	R	Vertical Axis. i	r
2	½	4.00	12.83	5.13	1.79	.71	.42
	1	5.50	17.46	6.97	1.78	1.36	.50
	1½	7.00	19.83	7.92	1.68	2.02	.54
2¼	½	4.25	14.07	5.62	1.82	.99	.48
	1	6.00	19.47	7.77	1.80	1.93	.57
	1½	7.75	22.23	8.88	1.69	2.86	.61
2½	½	4.50	15.39	6.15	1.85	1.34	.55
	1	6.50	21.55	8.62	1.82	2.64	.64
	1½	8.50	24.72	9.89	1.70	3.93	.68
2¾	½	4.75	16.60	6.63	1.87	1.77	.61
	1	7.00	23.55	9.42	1.84	3.50	.71
	1½	9.25	27.10	10.85	1.71	5.22	.75
3	½	5.00	17.92	7.16	1.89	2.29	.68
	1	7.50	25.63	10.25	1.85	4.53	.78
	1½	10.00	29.58	11.84	1.72	6.76	.82
3¼	½	5.25	19.14	7.65	1.91	2.90	.74
	1	8.00	27.62	11.07	1.86	5.75	.85
	1½	10.75	31.97	12.80	1.72	8.60	.89
3½	½	5.50	20.45	8.18	1.93	3.60	.81
	1	8.50	29.71	11.90	1.87	7.16	.92
	1½	11.50	34.45	13.80	1.73	10.71	.97
3¾	½	5.75	21.67	8.67	1.95	4.44	.88

TABLES FOR I-SECTIONS, 5 INCHES HIGH—Continued.

(½-inch Web.)

			Horizontal Axis.			Vertical Axis.	
W	T	A	I	S	R	i	r
3¾	1	9.00	31.70	12.70	1.88	8.83	.99
	1½	12.25	36.83	14.75	1.73	13.22	1.04
4	½	6.00	22.99	9.18	1.96	5.37	.95
	1	9.50	33.79	13.52	1.88	10.69	1.06
	1½	13.00	39.32	15.72	1.74	16.02	1.11

TABLES FOR I-SECTIONS, 5¼ INCHES HIGH.

(½-inch Web.)

			Horizontal Axis.			Vertical Axis.	
W	T	A	I	S	R	i	r
2	½	4.13	14.51	5.53	1.87	.71	.41
	1	5.63	19.81	7.55	1.87	1.36	.49
	1½	7.13	22.68	8.64	1.78	2.02	.53
2¼	½	4.38	15.91	6.06	1.91	.99	.48
	1	6.13	22.10	8.42	1.90	1.93	.56
	1½	7.88	25.44	9.69	1.79	2.86	.60
2½	½	4.63	17.37	6.62	1.94	1.34	.54
	1	6.63	24.43	9.30	1.92	2.64	.63
	1½	8.63	28.25	10.77	1.81	3.93	.68
2¾	½	4.88	18.77	7.14	1.96	1.77	.60
	1	7.13	26.72	10.20	1.94	3.50	.70
	1½	9.38	31.02	11.83	1.82	5.22	.75
3	½	5.13	20.17	7.67	1.98	2.29	.67
	1	7.63	29.00	11.05	1.95	4.53	.77
	1½	10.13	33.78	12.89	1.83	6.76	.82
3¼	½	5.38	21.62	8.24	2.00	2.90	.74
	1	8.13	31.35	11.95	1.96	5.75	.84
	1½	10.88	36.60	13.95	1.83	8.60	.89
3½	½	5.62	23.02	8.77	2.02	3.60	.80
	1	8.62	33.62	12.82	1.98	7.16	.91
	1½	11.62	39.36	15.00	1.84	10.71	.96
3¾	½	5.88	24.42	9.30	2.04	4.44	.87
	1	9.13	35.91	13.70	1.99	8.83	.98
	1½	12.38	42.12	16.07	1.85	13.22	1.03
4	½	6.13	25.82	9.83	2.05	5.37	.94
	1	9.63	38.20	14.56	2.00	10.69	1.05
	1½	13.13	44.88	17.07	1.85	16.02	1.10

TABLES FOR I-SECTIONS, 5½ INCHES HIGH.

(½-inch Web.)

			Horizontal Axis.			Vertical Axis.	
W	T	A	I	S	R	i	r
2	½	4.25	16.35	5.94	1.96	.72	.41
	1	5.75	22.40	8.13	1.97	1.37	.49
	1½	7.25	25.80	9.38	1.88	2.03	.53
2¼	½	4.50	17.90	6.50	1.99	1.00	.47
	1	6.25	24.96	9.05	2.00	1.94	.56

TABLES FOR I-SECTIONS, 5½ INCHES HIGH—Continued.

(½-inch Web.)

			—Horizontal Axis.—			—Vertical Axis.—	
W	T	A	I	S	R	i	r
2¼	1½	8.00	28.92	10.50	1.90	2.87	.60
2½	½	4.75	19.50	7.08	2.02	1.35	.53
	1	6.75	27.57	10.00	2.02	2.65	.63
	1½	8.75	32.09	11.67	1.91	3.94	.67
2¾	½	5.00	21.00	7.63	2.05	1.78	.59
	1	7.25	30.08	10.94	2.04	3.51	.69
	1½	9.50	35.17	12.78	1.92	5.23	.74
3	½	3.25	22.60	8.21	2.08	2.30	.66
	1	7.75	32.68	11.90	2.05	4.54	.76
	1½	10.25	38.35	13.93	1.93	6.77	.81
3¼	½	5.50	24.20	8.80	2.10	2.91	.73
	1	8.25	35.30	12.82	2.07	5.76	.83
	1½	11.00	41.52	15.10	1.94	8.61	.89
3½	½	5.75	25.70	9.34	2.11	3.61	.79
	1	8.75	37.81	13.75	2.08	7.17	.90
	1½	11.75	44.59	16.22	1.95	10.72	.96
3¾	½	6.00	27.30	9.92	2.13	4.45	.86
	1	9.25	40.42	14.70	2.09	8.84	.97
	1½	12.50	47.77	17.37	1.95	13.23	1.03
4	½	6.25	28.90	10.51	2.15	5.38	.93
	1	9.75	43.03	15.65	2.10	10.70	1.04
	1½	13.25	50.94	18.52	1.96	16.03	1.10

TABLES FOR I-SECTIONS, 5¾ INCHES HIGH.

(½-inch Web.)

			—Horizontal Axis.—			—Vertical Axis.—	
W	T	A	I	S	R	i	r
2	½	4.38	18.30	6.38	2.04	.72	.40
	1	5.88	25.10	8.75	2.07	1.37	.48
	1½	7.38	29.10	10.15	1.98	2.03	.52
2¼	½	4.63	19.97	6.96	2.08	1.00	.47
	1	6.38	27.90	9.73	2.09	1.94	.55
	1½	8.13	32.57	11.32	2.00	2.87	.59
2½	½	4.88	21.73	7.58	2.11	1.35	.52
	1	6.88	30.80	10.70	2.12	2.65	.62
	1½	8.88	36.13	12.57	2.02	3.94	.67
2¾	½	5.13	23.40	18.15	2.13	1.78	.58
	1	7.38	33.60	11.70	2.13	3.51	.68
	1½	9.63	39.60	13.78	2.03	5.23	.74
3	½	5.38	25.15	8.77	2.16	2.30	.65
	1	7.88	36.50	12.70	2.15	4.54	.75
	1½	10.38	43.17	15.00	2.04	6.77	.81
3¼	½	5.63	26.95	9.38	2.19	2.91	.72
	1	8.38	39.40	13.70	2.17	5.76	.82
	1½	11.13	46.74	16.28	2.05	8.61	.88
3½	½	5.88	28.70	10.00	2.21	3.61	.78
	1	8.88	42.30	14.72	2.18	7.17	.89
	1½	11.88	50.30	17.50	2.06	10.72	.95
3¾	½	6.13	30.50	10.60	2.23	4.45	.85

TABLES FOR I-SECTIONS, 5¾ INCHES HIGH—Continued.

(½-inch Web.)

			Horizontal Axis.			Vertical Axis.	
W	T	A	I	S	R	i	r
3¾	1	9.38	45.20	15.75	2.20	8.84	.96
	1½	12.63	53.87	18.72	2.06	13.23	1.02
4	½	6.38	32.15	11.18	2.25	5.38	.92
	1	9.88	48.00	16.70	2.20	10.70	1.03
	1½	13.38	57.33	19.95	2.07	16.03	1.09

TABLES FOR I-SECTIONS, 6 INCHES HIGH.

(½-inch Web.)

			Horizontal Axis.			Vertical Axis.	
W	T	A	I	S	R	i	r
2	½	4.50	20.37	6.79	2.12	.72	.40
	1	6.00	28.00	9.33	2.16	1.37	.48
	1½	7.50	32.63	10.87	2.08	2.03	.52
2¼	½	4.75	22.27	7.42	2.16	1.00	.46
	1	6.50	31.17	10.39	2.19	1.94	.55
	1½	8.25	36.57	12.19	2.11	2.87	.59
2½	½	5.00	24.17	8.06	2.20	1.35	.52
	1	7.00	34.34	11.45	2.21	2.65	.62
	1½	9.00	40.50	13.50	2.12	3.94	.66
2¾	½	5.25	26.10	8.70	2.23	1.78	.58
	1	7.50	37.50	12.50	2.24	3.51	.68
	1½	9.75	44.45	14.82	2.13	5.23	.73
3	½	5.50	27.95	9.32	2.25	2.30	.65
	1	8.00	40.77	13.59	2.25	4.54	.75
	1½	10.50	48.38	16.12	2.14	6.77	.80
3¼	½	5.75	29.90	9.97	2.28	2.91	.71
	1	8.50	43.84	14.61	2.27	5.76	.82
	1½	11.25	53.32	17.44	2.15	8.61	.88
3½	½	6.00	31.75	10.58	2.30	3.61	.78
	1	9.00	47.00	15.66	2.29	7.17	.89
	1½	12.00	56.26	18.75	2.16	10.72	.95
3¾	½	6.25	33.70	11.23	2.32	4.45	.84
	1	9.50	50.17	16.72	2.30	8.84	.96
	1½	12.75	60.20	20.07	2.17	13.23	1.02
4	½	6.50	35.55	11.85	2.34	5.38	.91
	1	10.00	53.34	17.78	2.31	10.70	1.03
	1½	13.50	64.14	21.38	2.18	16.03	1.09

TABLES FOR I-SECTIONS, 6¼ INCHES HIGH.

(½-inch Web.)

			Horizontal Axis.			Vertical Axis.	
W	T	A	I	S	R	i	r
2	½	4.63	22.63	7.24	2.21	.72	.39
	1	6.13	31.11	9.95	2.25	1.37	.47
	1½	7.63	36.41	11.65	2.18	2.03	.51
2¼	½	4.88	24.70	7.90	2.25	1.00	.45
	1	6.63	34.61	11.07	2.29	1.94	.54

TABLES FOR I-SECTIONS, 6¼ INCHES HIGH—Continued.

(½-inch Web.)

			Horizontal Axis.			Vertical Axis.	
W	T	A	I	S	R	i	r
2¼	1½	8.38	40.80	13.07	2.21	2.87	.58
2½	½	5.13	26.80	8.57	2.29	1.35	.51
	1	7.13	38.12	12.20	2.32	2.65	.61
	1½	9.13	45.18	14.46	2.23	3.94	.66
2¾	½	5.38	28.90	9.25	2.32	1.78	.57
	1	7.63	41.62	13.34	2.34	3.51	.67
	1½	9.88	49.57	15.90	2.24	5.23	.73
3	½	5.63	30.95	9.90	2.34	2.30	.64
	1	8.13	45.12	14.43	2.36	4.54	.74
	1½	10.63	53.95	17.27	2.25	6.77	.79
3¼	½	5.88	33.05	10.58	2.37	2.91	.70
	1	8.63	48.62	15.58	2.37	5.76	.81
	1½	11.38	58.35	18.68	2.26	8.61	.87
3½	½	6.13	35.15	11.25	2.40	3.61	.77
	1	9.13	52.12	16.70	2.39	7.17	.88
	1½	12.13	62.72	20.08	2.27	10.72	.94
3¾	½	6.38	37.20	11.90	2.42	4.45	.83
	1	9.63	55.62	17.80	2.41	8.84	.95
	1½	12.88	67.11	21.50	2.28	13.23	1.01
4	½	6.63	39.30	12.57	2.43	5.38	.90
	1	10.13	59.12	18.95	2.42	10.70	1.02
	1½	13.63	71.50	22.88	2.29	16.03	1.08

TABLES FOR I-SECTIONS, 6½ INCHES HIGH.

(½-inch Web.)

			Horizontal Axis.			Vertical Axis.	
W	T	A	I	S	R	i	r
2	½	4.75	24.95	7.68	2.29	.73	.39
	1	6.25	34.35	10.57	2.34	1.38	.47
	1½	7.75	40.40	12.42	2.28	2.04	.51
2¼	½	5.00	27.20	8.37	2.34	1.01	.45
	1	6.75	38.15	11.74	2.38	1.95	.54
	1½	8.50	45.21	13.90	2.31	2.88	.58
2½	½	5.25	29.40	9.04	2.37	1.36	.51
	1	7.25	41.95	12.90	2.41	2.66	.61
	1½	9.25	50.12	15.42	2.33	3.95	.66
2¾	½	5.50	31.65	9.75	2.40	1.79	.57
	1	7.75	45.75	14.08	2.43	3.52	.67
	1½	10.00	54.83	16.85	2.34	5.24	.73
3	½	5.75	33.95	10.45	2.43	2.31	.63
	1	8.25	49.65	15.28	2.45	4.55	.74
	1½	10.75	59.73	18.35	2.36	6.78	.79
3¼	½	6.00	36.35	11.18	2.46	2.92	.70
	1	8.75	53.55	16.48	2.47	5.77	.81
	1½	11.50	64.65	19.90	2.37	8.62	.87
3½	½	6.25	38.55	11.87	2.49	3.62	.77
	1	9.25	57.35	17.64	2.49	7.18	.88
	1½	12.25	69.46	21.35	2.38	10.73	.94
3¾	½	6.50	40.75	12.53	2.51	4.46	.83

TABLES FOR I-SECTIONS, 6½ INCHES HIGH—Continued.

(½-inch Web.)

W	T	A	Horizontal Axis. I	S	R	Vertical Axis. i	r
3¾	1	9.75	61.15	18.83	2.51	8.85	.95
	1½	13.00	74.27	22.85	2.39	13.24	1.01
4	½	6.75	43.05	13.25	2.53	5.39	.89
	1	10.25	64.95	20.00	2.51	10.71	1.02
	1½	13.75	79.08	24.32	2.40	16.04	1.08

TABLES FOR I-SECTIONS, 6¾ INCHES HIGH.

(½-inch Web.)

W	T	A	Horizontal Axis. I	S	R	Vertical Axis. i	r
2	½	4.88	27.50	8.15	2.37	.73	.38
	1	6.38	37.85	11.21	2.44	1.38	.46
	1½	7.88	44.65	13.23	2.38	2.04	.50
2¼	½	5.13	29.87	8.85	2.41	1.01	.44
	1	6.88	41.99	12.45	2.47	1.95	.53
	1½	8.63	49.92	14.78	2.41	2.88	.57
2½	½	5.38	32.35	9.57	2.45	1.36	.50
	1	7.38	46.18	13.67	2.50	2.66	.60
	1½	9.38	55.25	16.37	2.43	3.95	.65
2¾	½	5.63	34.90	10.33	2.49	1.79	.56
	1	7.88	50.40	14.95	2.53	3.52	.66
	1½	10.13	60.60	17.98	2.43	5.24	.72
3	½	5.88	37.35	11.07	2.52	2.31	.63
	1	8.38	54.60	16.20	2.55	4.55	.73
	1½	10.88	65.95	19.55	2.46	6.78	.78
3¼	½	6.13	39.85	11.81	2.55	2.92	.69
	1	8.88	58.80	17.42	2.57	5.77	.80
	1½	11.63	71.25	21.14	2.47	8.62	.86
3½	½	6.37	42.20	12.50	2.57	3.62	.76
	1	9.37	62.90	18.65	2.59	7.18	.87
	1½	12.37	76.50	22.70	2.49	10.73	.93
3¾	½	6.63	44.55	13.22	2.60	4.46	.82
	1	9.88	67.05	19.90	2.61	8.85	.94
	1½	13.13	81.75	24.24	2.48	13.24	1.00
4	½	6.88	47.00	13.93	2.61	5.39	.88
	1	10.38	71.25	21.13	2.62	10.71	1.01
	1½	13.88	87.10	25.80	2.51	16.04	1.07

TABLES FOR I-SECTIONS, 7 INCHES HIGH.

(½-inch Web.)

W	T	A	Horizontal Axis. I	S	R	Vertical Axis. i	r
2	½	5.00	30.20	8.62	2.45	.73	.38
	1	6.50	41.57	11.90	2.53	1.38	.46
	1½	8.00	49.20	14.05	2.48	2.04	.50
2¼	½	5.25	32.85	9.38	2.49	1.01	.44
	1	7.00	46.12	13.18	2.57	1.95	.53

TABLES FOR I-SECTIONS, 7 INCHES HIGH—Continued.

(½-inch Web.)

			—Horizontal Axis.—			—Vertical Axis.—	
W	T	A	I	S	R	i	r
2¼	1½	8.75	55.02	15.70	2.51	2.88	.57
2½	½	5.50	35.50	10.14	2.54	1.36	.50
	1	7.50	50.67	14.48	2.60	2.66	.60
	1½	9.50	60.84	17.40	2.53	3.95	.65
2¾	½	5.75	38.15	10.90	2.58	1.79	.56
	1	8.00	55.25	15.80	2.63	3.52	.66
	1½	10.25	66.65	19.05	2.55	5.24	.72
3	½	6.00	40.80	11.66	2.61	2.31	.62
	1	8.50	59.75	17.08	2.65	4.55	.73
	1½	11.00	72.47	20.70	2.57	6.78	.78
3¼	½	6.25	43.45	12.40	2.63	2.92	.68
	1	9.00	64.35	18.40	2.67	5.77	.80
	1½	11.75	78.29	22.38	2.58	8.62	.86
3½	½	6.50	46.10	13.18	2.66	3.62	.75
	1	9.50	68.85	19.65	2.69	7.18	.87
	1½	12.50	84.10	24.03	2.59	10.73	.93
3¾	½	6.75	48.75	13.94	2.69	4.46	.81
	1	10.00	73.45	21.00	2.71	8.85	.94
	1½	13.25	89.92	25.65	2.60	13.24	1.00
4	½	7.00	51.40	14.68	2.71	5.39	.88
	1	10.50	77.95	22.30	2.72	10.71	1.01
	1½	14.00	95.74	27.35	2.61	16.04	1.07

INDEX.

A

B

C

D

E

F

G

*The left-hand column gives the tens and the upper horizontal line the unit figure. The intersection of the horizontal line and the vertical column gives the page upon which any desired formula may be found. For example, to find formula 67: Follow line 60 to the right to column numbered 7 at the top. The number, 180, is that of the page on which the formula appears.

H

I

J

K

L

Q

R

*Plates 11, 12, 13, 16, 18, 19, 26, 27, 31, 32 and 36 will be found in supplement form at the back part of the book.

S

U

V

PLATE 11
PLATE 12

Plate 11.

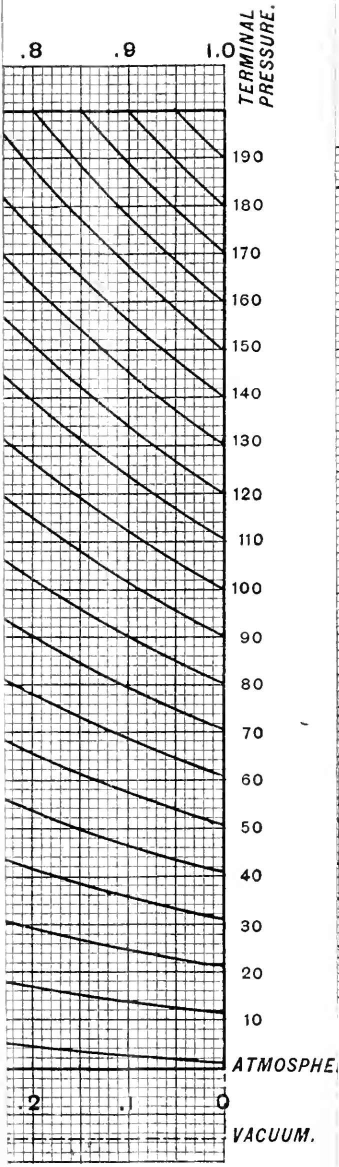
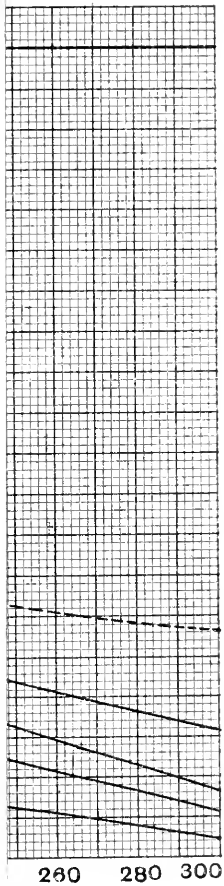
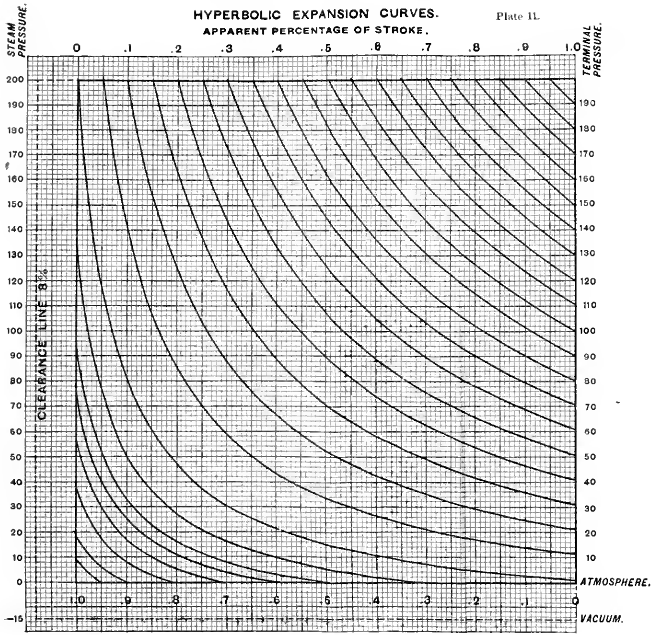


Plate 12.




HYPERBOLIC EXPANSION CURVES.
APPARENT PERCENTAGE OF STROKE.
Plate 11.
STEAM PRESSURE.
TERMINAL PRESSURE.
0 .1 .2 .3 .4 .5 .6 .7 .8 .9 1.0
200 190 180 170 160 150 140 130 120 110 100 90 80 70 60 50 40 30 20 10 0 −15
CLEARANCE LINE 8%
1.0 .9 .8 .7 .6 .5 .4 .3 .2 .1 0
ATMOSPHERE.
VACUUM.

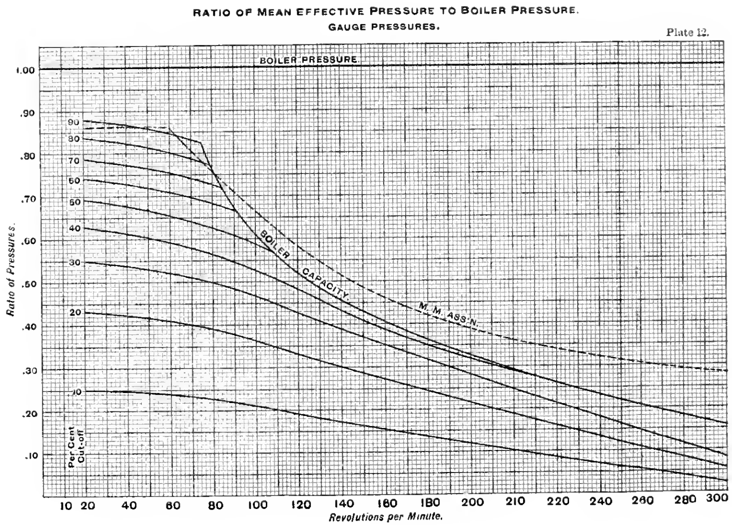
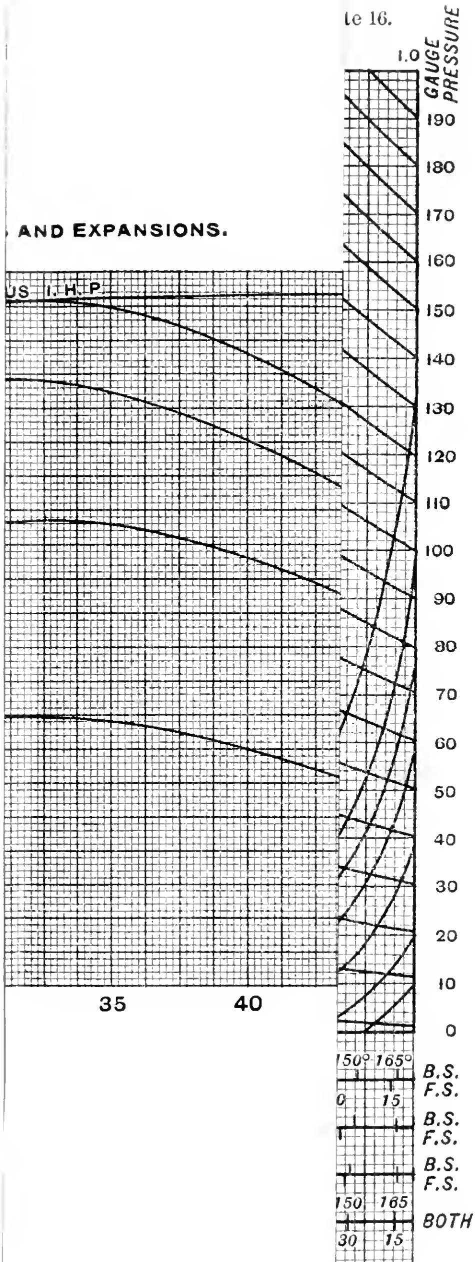

RATIO OF MEAN EFFECTIVE PRESSURE TO BOILER PRESSURE.
GAUGE PRESSURES.
Plate 12.
BOILER PRESSURE.
1.00 .90 .80 .70 .60 .50 .40 .30 .20 .10
Ratio of Pressures.
90 80 70 60 50 40 30 20 10
Per Cent Cut-off
BOILER CAPACITY.
M. M. ASS'N.
10 20 40 60 80 100 120 140 160 180 200 210 220 240 260 280 300
Revolutions per Minute.

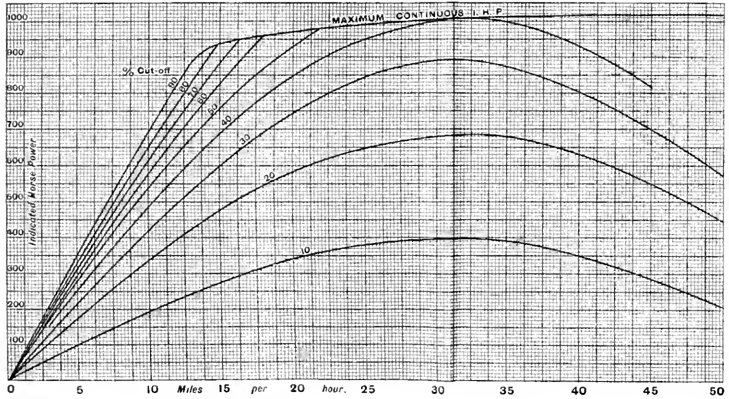
PLATE 13
PLATE 16

te 16.

AND EXPANSIONS.



INDICATED HORSE POWER AT VARIOUS SPEEDS AND EXPANSIONS. Plate 13.



INSTANTANEOUS PISTON PRESSURES.

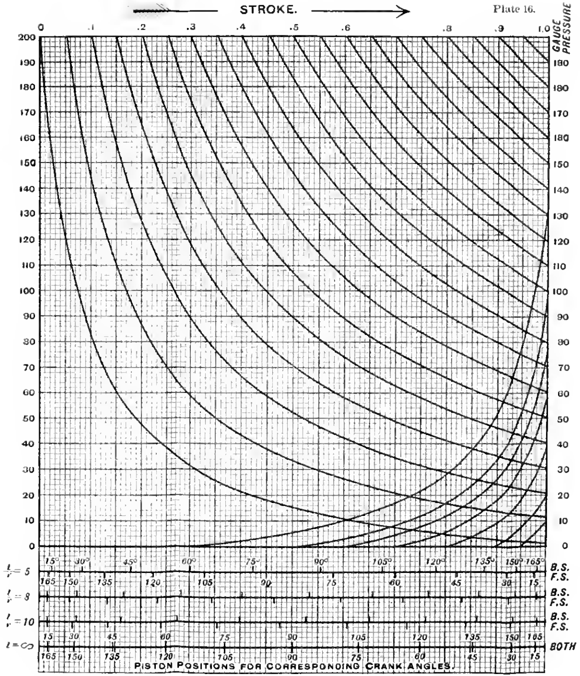


PLATE 18
PLATE 26

Plate 18.

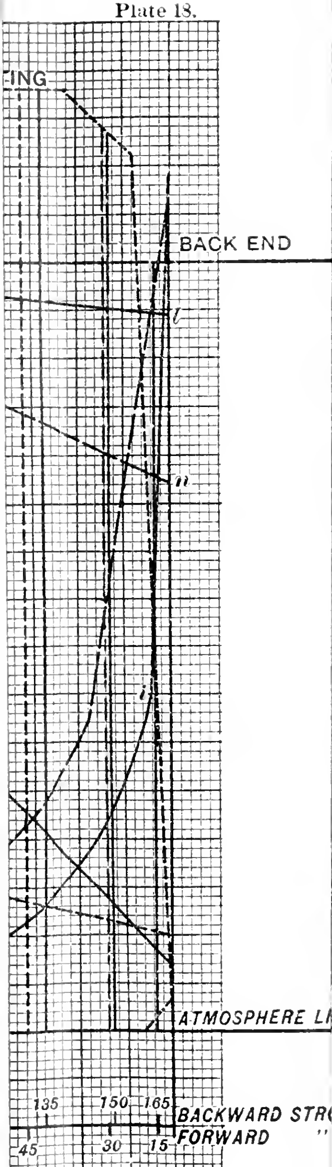
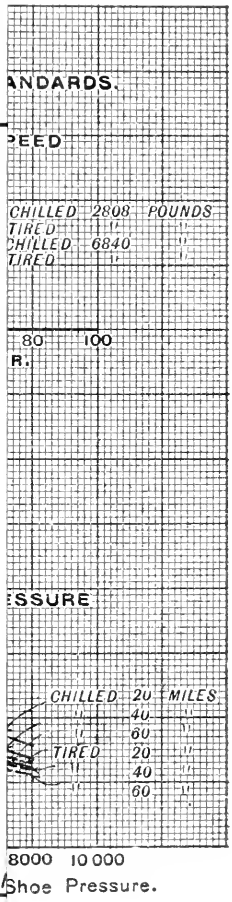
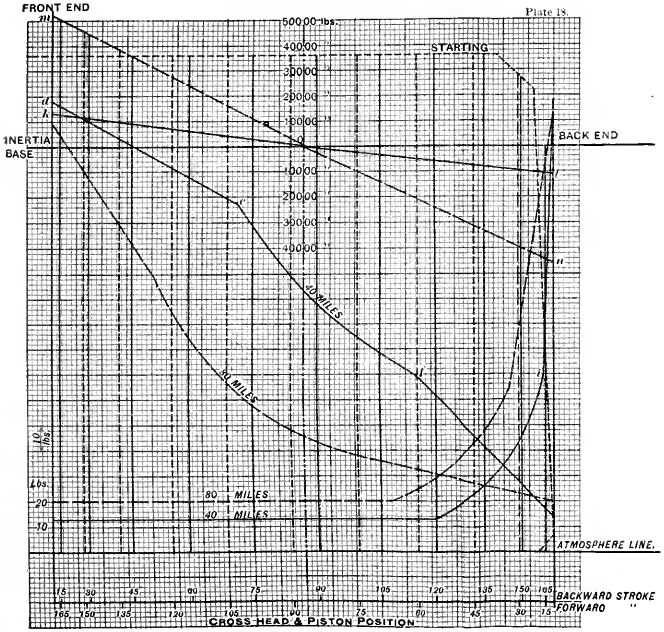


Plate 26.



RECIPROCATING FORCES.



FORCES OF RETARDATION

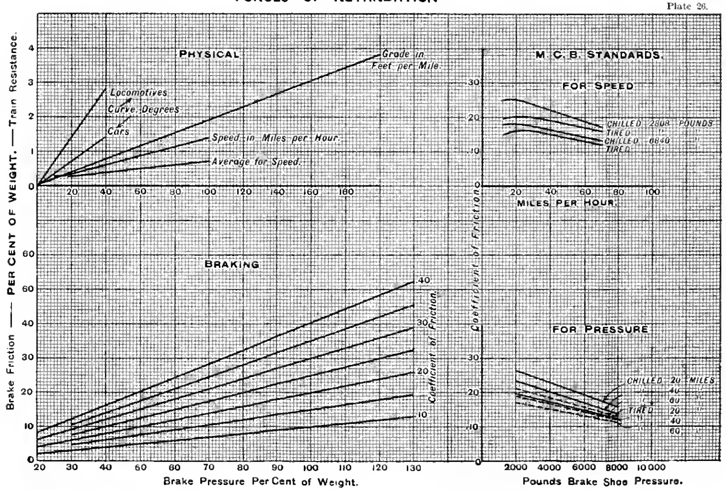
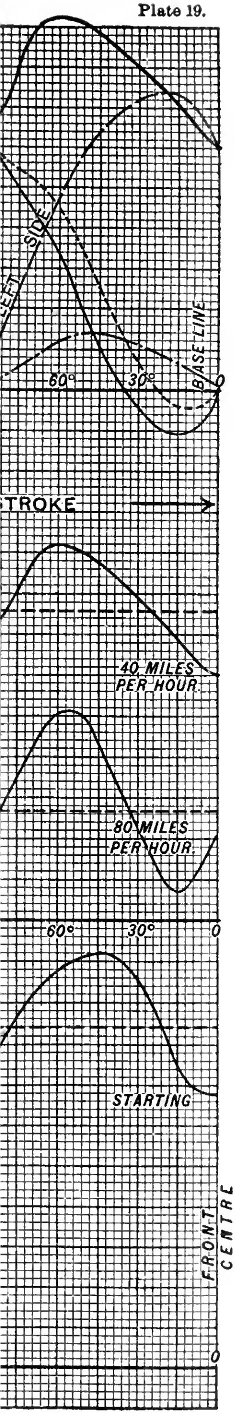


PLATE 19
PLATE 31
PLATE 32

Plate 19.



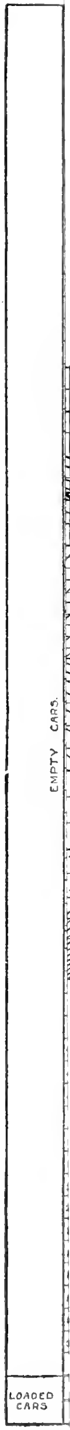
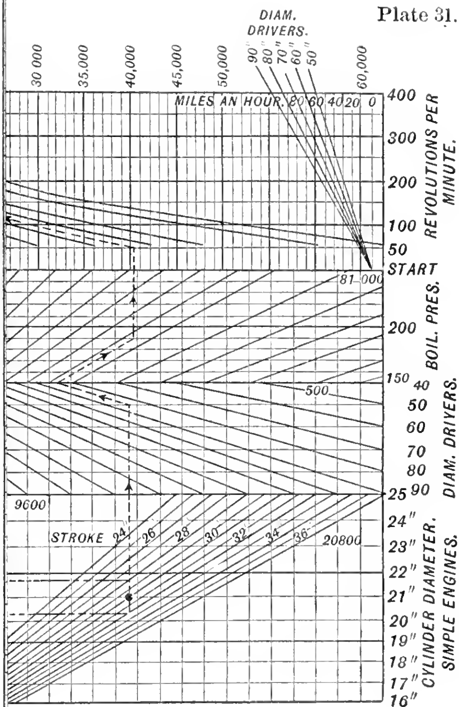


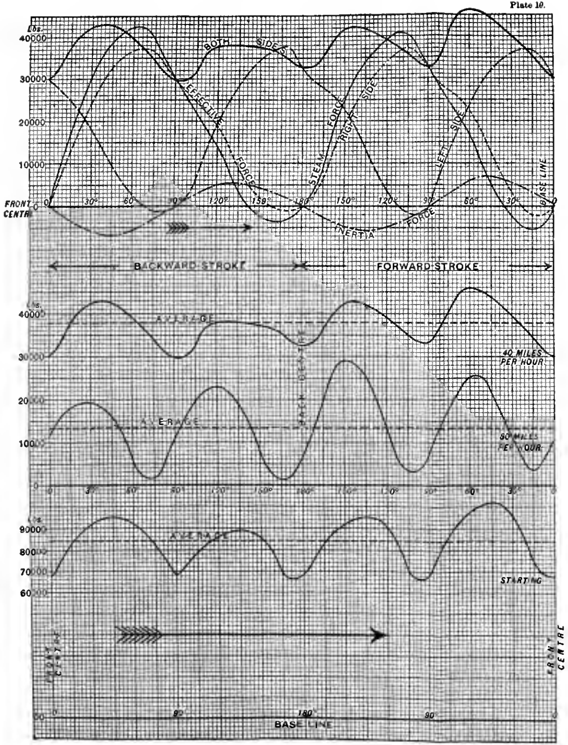
Plate 31.



TRACTIVE FORCE,
ERENCE OF DRIVERS.

ROTATIVE FORCES.

Plate 19.



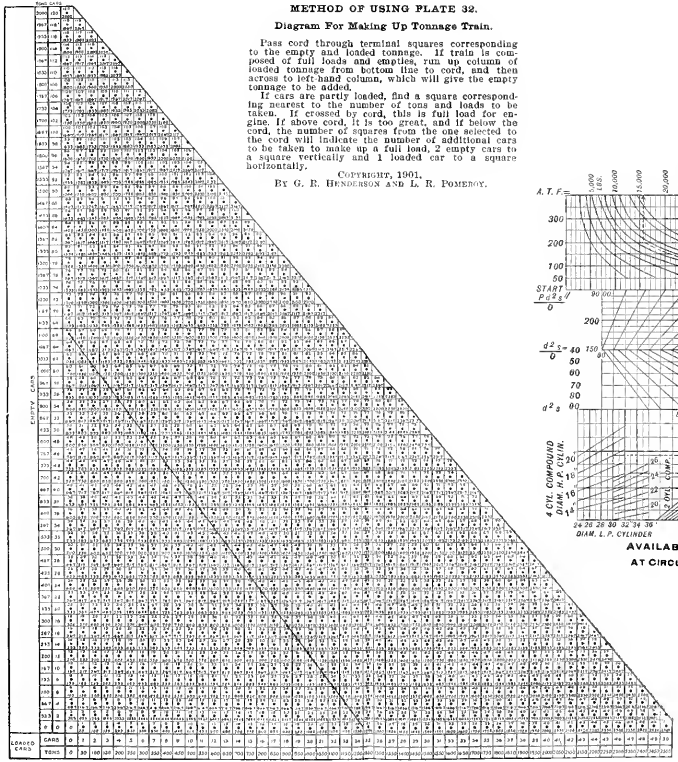


PLATE · 32

METHOD OF USING PLATE 32.

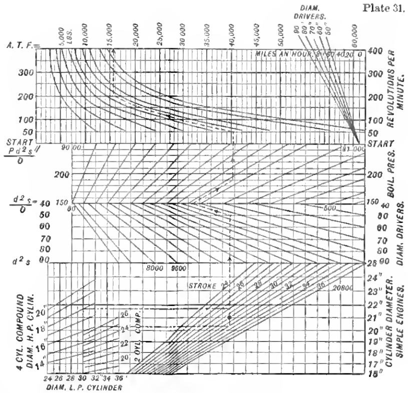
Diagram For Making Up Tonnage Train.

Pass cord through terminal squares corresponding to the empty and loaded tonnage. If train is composed of full loads and empties, run up column of loaded tonnage from bottom line to cord, and then across to left-hand column, which will give the empty tonnage to be added.

If cars are partly loaded, find a square corresponding nearest to the number of tons and loads to be taken. If crossed by cord, this is full load for engine. If above cord, it is too great, and if below the cord, the number of squares from the one selected to the cord will indicate the number of additional cars to be taken to make up a full load, 2 empty cars to a square vertically and 1 loaded car to a square horizontally.

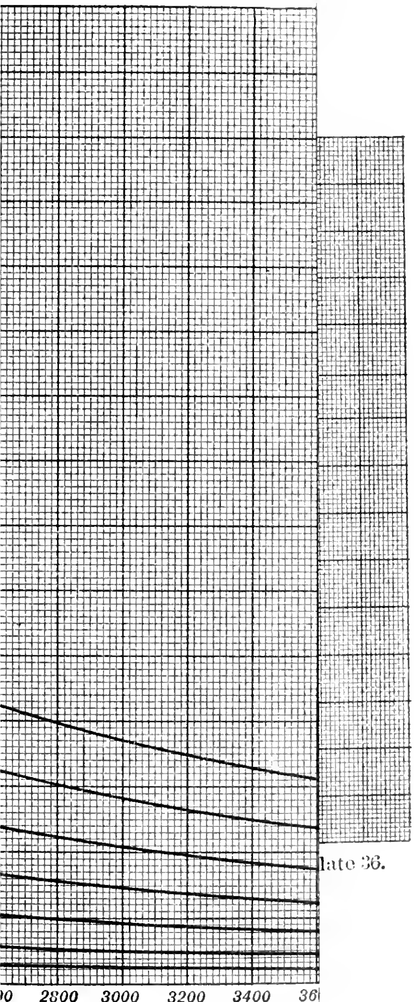
By G. R. HENDERSON AND L. R. POMEROY.

Plate 31.



AVAILABLE TRACTIVE FORCE,
AT CIRCUMFERENCE OF DRIVERS.

PLATE 27
PLATE 36



late 36.

RETARDATION OF TRAINS.