REVIEW OF I. C. R. R. LOCOMOTIVE, NO. 962

...BY...

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THESIS

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REVIEW OF LOCOMOTIVE NO. 962, I. C. R. R.

Outline: -

(I) Short historical review of locomotive in general.

(2) Brief discussion of types, and requirements of locomotives in general, with statement of requirements of #962 in particular. Dimensions of #962.

(3) Train resistance, and resistance of the atmosphere.

(4) Detailed statement of requirements of locomotives in general, which may apply to locomotive designed.

(5) Rapid Steaming Qualities; a discussion of boiler and fire box proportions; also sizes of stacks and nozzles.

(6) Collection of formulas that may apply to locomotive design. Relation between different formulas shown by curves plotted on cross section paper. Comparison of above proportion with the actual dimensions of #962 and others.

(7) PLATES. DIAGRAMS OF TANGENTIAL EFFORT INCLUDING EFFECT OF INERTIA OF RECIPROCATING PARTS. Dimensions of several modern locomotives, tabulated.

(8) Investigation of unbalanced pressures on valve face.
Historical Review.

The invention and early development of the Locomotive in England was mainly due to the absolute necessity for a better and more efficient means for handling her rapidly increasing traffic. Its introduction and operation in this country was largely due to commercial rivalry between certain seaports. The opening of the Erie Canal gave New York an immense advantage, commercially, over such cities as Baltimore, Charleston, Boston and Philadelphia, and compelled these cities, in self defense, to attempt means of communication of their own with the interior. It found a much warmer welcome in this country than in England, where strong prejudice existed against its introduction.

One of the first machines built, which in object or arrangement resembled the modern Locomotive, was the tramway engine of Richard Trevethick (Cornwall, Eng. 1803). It had horizontal cylinders connected to the drivers by gears, used comparatively high pressure steam, and exhausted into a chimney. Its handling capacity was about eleven tons, but it proved more expensive than horse power. In 1825, Col. J. Stevens of Hoboken, N.J. designed and built a rack rail engine with horizontal cylinders about 4 X 12". It was mounted on wooden wheels, was provided with vertical tubular boiler, and burned wood.

In 1828 America saw its first practical locomotive,--- Stephenson's "America", imported by the Delaware & Hudson Canal Co. The "Stourbridge Lion", however, was the first locomotive in this country to actually run. The "America" was somewhat similar to the
famous "Rocket". It had a 4' 1" X 9' 6" boiler, and cylinders 9" X 24" set on an angle of 33° with the horizontal. Two 1' 7" flues opened into the chimney base. The frame was bar iron. Later Stephenson adopted the "Sandwich" frame of wood and iron and later the plate frame. The bar iron frame, however, came to be preferred in this country over the plate frame because of the adaptability to rough tracks.

The "Stourbridge Lion" (Walking beam engine), had a vertical cylinder near rear end of tubular boiler, beam and mechanism being placed above. A vertical connecting rod drove the rear driver. The two drivers were connected by a horizontal tie-rod. It was provided with a smoke box and exhausted into a chimney. The design was bad, as the weight of the heavy reciprocating parts caused a pound on track and unsteady running.

In 1829 Peter Cooper built the "Tom Thumb" for the B.& O.R. R. and shortly after, the "Best Friend". One of the best of the early locomotives built in this country was built by him for the South Carolina R. R. It had inclined 6 X 6 cylinders, inside connected, drivers 4' 9", boiler vertical, weight 4 1/2 tons.

Stevenson's "Rocket" used copper tubes in boiler because of conductivity of that metal to heat. Cylinders were inclined, direct connected; a blast nozzle was used and the fire box was surrounded by water. A speed of 24 miles per hour was attained.

Many of the early locomotives of this country were made by the West Point Foundry Ass'n. One of their engines, the "West Point" (bar, tubular 4' 9" four complete drivers, 6 X 6 cylinders) hauled 5 cars and 117 passengers 2 1/2 miles in eight minutes.
Their next, the "DeWitt Clinton" for the Mohawk & Hudson R. R. was an inside connected "America", (4'6" drivers, 5 1/2 X 16 cylinders, 30 copper tubes.). It made 15 miles per hour with three coaches.

A $4,000.00 prize offered by the B. & O. was captured by Messrs. Davis & Gartner. This firm introduced the "Grasshopper" type; for many years used on that road.

In August 1831, the "John Bull" was received from England. It was rather bare as received, so that a pilot, cab, and other details were added which made it look a little like the American type of to-day. This is the locomotive that made the trip to the World's Fair. The valve gear was somewhat crude, it being necessary for the engineer to run the valve by hand for a few strokes, in reversing.

From this time on, locomotive practice in this country and in England began to differ, each country adapting its product to its needs. One of the first departures from English designs was the use of the truck, first suggested by Horatio Allen, and shortly after put into practice by J. B. Jervis, in a locomotive for the Mohawk & Hudson R. R. Another important step was the introduction of the link motion. This was first invented by W. T. James of Newport in 1823, but was afterwards independently invented in the shops of Stephenson, and by him given the prominence it deserved. It was soon generally adopted, though in spite of some opposition.

The first attempt at counterbalancing locomotives was made by the Roger's works, in their "Sandusky". They next reduced the gauge from 4'10" to 4'8 1/2" in the "Clinton". They then tried to improve weight distribution by placing drivers in rear of fire box,
but found that this threw too much weight on front trucks and too little on drivers, causing loss of tractive power. Next the plan was tried of placing part of the tender weight on the drivers.

About this time Baldwin put the cranks on an arm of the driver and abandoned secondary coupling rod cranks.

In 1835, H. R, Campbell designed the first strictly "American" type engine, with four drivers and four leading truck wheels. It was inside connected; the forward drivers were without flanges; about the same ratio of adhesive to total weight was used as now, 8 to 13. Garret & Eastwick in 1837, improved this type by having a separate frame for the four drivers, which was pivoted on springs to main frame. The latter slid vertically over the journal boxes. But as the opposite sides were tied together, the lateral unevenness of the track played havoc with the frame, which prompted Eastwick & Harrison to introduce equalizing levers the year after.

The separate frame was not used but a horizontal lever was introduced; the ends connected to abutments on boxes; the center connected to the frame by a spring. They were independent and so could equalize the strain caused by the lateral irregularities of the track. It was thought desirable to keep the truck wheels close together, which caused builders to try placing the cylinders above the wheels, sloping them backward and downward about 20°.

Between 1850 and 1854 the question of movable link, or movable block, was settled in favor of the former. In 1845, "Rogers" changed to the inside framing, improved the arrangement of equalizing levers, and increased the size of drivers to 5 ft. Sand boxes came in 1847.
In 1850 Rogers introduced the wagon top boiler and set the cylinder saddle between truck wheels, giving horizontal motion and simplifying the problem of counterbalancing. In 1857 Bissell extended the leading truck to the rear and pivoted it there. The object was to give better action on curves. In order to increase tractive power, Hudson in 1863 designed the first "Mogul". This was followed by a 45 ton, ten wheeler, by Alex. Mitchell. With the application of the air brake and injector we have (in essential features) the modern locomotive as seen in use today. In its early stages nearly every simple type of construction known was experimented on; with the exception of the water tube boiler, corliss gear, and other stationery type mechanism. The horizontal and vertical boiler with small and large flues, with and without smoke box, exhaust and air blast, walking beam, geared, grasshopper and direct connected engines, inside and outside cylinders, vertical and oblique, were tried on both continents, and the results thus independently arrived at, while differing in detail are quite similar, except in so far as different conditions of road bed and usage necessitated different design. The English locomotive is more rigid than the American, while lighter loads, or very firm, solid track may justify the use of a single large driver. There has been a steady increase of weight and of height of center of gravity. This latter condition has been found and then proven to give steadier riding qualities than a low center of gravity.

The present tendency is toward increase in weight. Locomotives now receive different treatment than formerly. Trains are not only heavier but faster, and a locomotive is in almost contin-
uous service, the engineers' alternating. Again, a break down is getting to be a serious matter, because of the delay which may be caused to other trains.

Therefore, the modern locomotive must be made to stand service, and strength and durability must be obtained either by the use of good quality steel and iron, or by weight of parts, or both.
American.

Mogul.

Ten Wheel.

Consolidation.
Types and Requirements.

Types of locomotives vary with the service required; hence we have the freight and passenger, the logging, the switch, the suburban and the mountain or rack rail locomotives. Considering American freight and passenger locomotives as constituting one type, we have the following classification:

The 8 wheel locomotive (or American type) has four driving and truck wheels. Used both for passenger and freight service.

The Mogul has 6 driving and two truck wheels. Used principally for freight service, especially fast freight.

The 10 wheeler has 6 driving and 4 truck wheels. For fast freight service.

The consolidation, 8 drivers and two truck wheels. For heavy freight service especially on steep grades.

It is to be noted in connection with this classification that in practice, diameter of drivers decreases as their number increases.

In designing a locomotive for use on a given road it is necessary to know:

1. The kind of service required, whether passenger, fast passenger or light or heavy, fast or slow freight.
2. Minimum and maximum weight of trains and conditions of rolling stock.
3. Required speed.
4. Grades, curves, and conditions of track.
5. Gauge of track, and weight on drivers that rails will stand.
6. Kind of fuel used.

7. Height of tunnels, and bridges.

In general:

Conditions (1) and (2) above enable us to determine the type, class, and weight of proposed locomotive. (3) has to do with size of drivers, piston speed, and steam distribution. (1), (2), (3), and (4) give us boiler requirements. (5) must be considered in connection with weight and number of drivers (hence class of engine) and with counterbalancing. In connection with boiler requirements (6) determines details of furnace and grate. (7) limits height of smoke stack, and cab, and may affect size of drivers, as affecting total height of locomotive.

This brings us to the special case under consideration,—that of I. C. R. R. Locomotive 962, (Brooks,) for fast passenger and mail service between Chicago, Champaign and Centralia. Country is prairie land, grades moderate.

Conditions of rolling stock is good.

Minimum weight of trains are 240,000 lbs.

Maximum " " " 800,000

Ordinary " " " 300,000

It was desired to get as high speed as possible, consistent with safety and sufficient tractive power to handle full trains.

Condition of track,—fair. Rock ballast.

Gauge of track, 4 ft 8 1/2".

75 to 90# rails.
18 Ties to 30' rail  Ties 8' X 6' X 8' oak.

Fuel, bituminous (Odin) lump.  Quality good.  3" down.

Height of bridges,- entirely clear.
DIMENSIONS OF I. C. R. R. LOCOMOTIVE #962.

Steam pressure 200#.
Wheel base, engine, 23' 7".
Driving wheel base, 8' 9".
Driving wheels, diameter of centers, 68". Cast steel. Wheels - 75"
Tires, Krupp crucible steel.
Engine truck wheels, 36" diameter.
Tractive power at 90% B. P. - 10% friction = 18190 #.
Coefficient of adhesion = 4.4.
Smallest ring 62" outside.
Thickness of cylindrical plates -
   waist - 9/16, 5/8, 1/2, 7/16"
   Throat sheet, 5/8"
Longitudinal seams quintuple riveted.
Vertical seams double riveted.
Fire box - kind - sloping between frames.
   " " length inside 107 5/8"
   " " width inside 36 3/8"
   " " material steel.
   " " side sheets 5/16"
   " " crown sheet 3/8"
Tube sheets, front - 5/8" back 1/2"
Water spaces, width - 3 1/2", sides - 4", front and back.
Tubes, diameter, 2". length 11' 7"
   " number 274, # 12 B. W. G.
Heating surface, tubes, 1649.4 sq. ft.
   " fire box 152.2 sq. ft.
   " total 1801.6 sq. ft.
Grates, shaking fingers bars.
Piston rods Ewald C. C. bloom iron.
" " Diam. 3 1/2".
" " and valve stem packing, U. S. Met.
Piston packing, Dunbar.

Driving axle, hammered iron.
" " journal 8 1/2" diam. X II" long.

Engine, truck axle, hammered iron.
" " journal 5 1/2" diam. X I2" long.

Connecting rod, I section hammered iron.

Valves Allen American balanced.

LUBRICATION - guide adjustable needle cups.
rods, spindle feed, cups forged on.
cylinder, #9 old style Nathan, cups on steam chest.

Injector, one #9 Monitor and one #8 Ohio.

Safety valves two 3" Ashton pop. One open and one muffled.

Water capacity of tender, 4260 U. S. gal.
TRAIN RESISTANCE.

Experimental results give coefficient of resistance in terms of load = .0015 to .0096 for 5 to 60 M. P. H., or pull = \( l = \frac{\sqrt{s}}{650} \) Giving per ton of 2240 lbs. 3.1 # to 19.2# respectively. For curvature .5 # per degree and per ton.

\[
5280 \sin \theta = ma \\
L \sin \theta = \frac{mL}{5280} = .00019 L m.
\]

L = weight of train alone, - tons.
R = resistance of train in lbs.

Adding resistances, \( R = L(0.0015(I + \frac{s}{650}) + 0.00025c + 0.00019n) \) the + or - depending upon whether ascending or descending grades.

Resistance of engine is, similarly, (where \( w = \) wt. of engine and tender.) \( R = w(0.0015(I + \frac{s}{650}) + 0.005c + 0.00091n) \)

The coefficient of \( c \) being doubled for American locomotives.

Then total resistance = \( (w + L)(0.0015(I + \frac{s}{650}) + 0.00025c + 0.00091n) \), which should = tractive power for the cut off used for given speed.

The above assumes that locomotive resistance is the same, (proportional to weight), as resistance to trains.

By still another formula \( R = \frac{I}{4} \) vel. in M. P. H. \( + 2 \); thus for 50 M. P. H., \( R = 14.5# \) and for 90 M. P. H. \( R = 24.5# \).

Knowing train resistance we may find the power necessary to overcome it, at required rate of speed; ie. H. P. transmitted = resistance \( X \) ft. per minute run \( \frac{a}{33000} \).

Find resistance from above formula for train resistance only, without locomotive. To uniformly accelerate train to this
speed, as in leaving a station use, "force = mass \times \text{accel.}". Then force \times \text{distance run} + 33000 = \text{H. P.} \text{ for accel.} \quad (b)

Add (a) and (b), also engine resistance; this should = tractive power \times \text{distance} + 33000, when running at that speed. Formula (a) of course considers velocity at some instant and (b) assumes acceleration constant at that same instant.

Meyers, in "Modern Locomotive Construction" gives for slow speed, (7 to 9 M. P. H.) 7 1/2\# per ton.

Otherwise

Resistance per 2000\#, straight level track,

\[ R = \left( \frac{\text{M. P. H.}}{171} \right)^2 + 6. \] \quad (c)

For grade \( R = \text{rise, (ft. per mile)} \times 0.3788 \), added to result obtained from formula (c).

Resistance due to curves = 1/2\# per ton per degree of curvature.

The degree of curvature when radius is less than 500 ft. = radius in feet + 5730, approximately.

To find train resistance per unit weight of locomotive plus load, we may note speed at stated intervals of time as train slows down on a level track with the brakes off, thus finding negative acceleration, and then apply the formula

\[ P = M \frac{f \times \text{wt.}}{gXf}. \]

\( P \) = force.

\( M \) = mass.

\( f \) = acceleration.

\( g \) = acceleration of gravity.

Making weight = unity we have force required to accelerate one lb. of engine and train load. If on a grade add or subtract from this force a \% of same = \% grade. A trial of this kind showed
for locomotive #962 and eight cars at a speed ranging from 58 to 52 M. P. H. an average negative acceleration of .2 M. P. H. per minute. 6.2 M. P. H. acceleration per minute = .1516 ft. acceleration per second. Then, force = \( \frac{1}{2} \times .1516 = \frac{.0047}{32.2} \)
r resistance per lb. wt. of locomotive and load.

\[ .0047 \times 2240 = 10.53\text{# per ton. Kent gives 12# per ton for similar conditions.} \]

Knowing the train resistance and therefore the required tractive power, we may find cylinder diameter by solving equation of tractive power, for \( d \). Thus;

\[
P = \frac{d p s}{D} \quad d = \frac{P D}{ps} \]

\( d = \text{cyl. diam.} \)

\( D = \text{diam of drivers.} \)

\( p = \text{steam press.} \)

\( S = \text{stroke = 2 x crank radius.} \)

\( P = \text{tractive force required to overcome train resistance.} \)

\( D = \text{diameters of drivers is here required which may be selected by experience, for given conditions and requirements, and by joint considerations of speed of train, piston velocity, and velocity of steam through ports, requiring again, assumptions as to port openings, which can only be based on current practise.} \)

This method however will give some idea of general proportions upon which to base our selection of data, calculation of weight of reciprocating parts, etc. involved in the design of locomotive cylinders and valves by methods of "High speed" Steam engine" and "Design".
Shows negative acceleration, train slowing down on level track. Used in finding train resistance, $f = mass \times accel$. See p. 15+16.
Experiments on resistance offered by atmosphere to motion of train. Made on 962. Pressure per sq. ft. \(= 0.002 V^2\).
Calibration of wind-pressure-gauge.
REQUIREMENTS. of locomotive in general.

RAPID STEAMING QUALITIES.

Quality of combustion. Rate of combustion.
Grate area. Heating surface.
Prevention of priming. (Dome and safety valve not over fire box).
Circulation of water.
Conductivity and prevention of radiation.
Draft, involving arrangement of exhaust nozzle.

PROPER STEAM DISTRIBUTION.

Arrangement of steam pipes and attachments.
Design and dimensions of valves.
" " " valve gear.
" " arrangement of exhaust apparatus.

TRACTIVE POWER.

Weight on drivers. No of drivers. Coeff. of adhesion.
Turning effect of pistons.

SPEED.

Properly designed and balanced reciprocating parts.
Proper steam distribution.
Proper proportions between drivers and cylinders.
Steadiness. Good steaming Qualities.

STEADINESS.

Good construction and good condition of road bed.
Proper counterbalancing.
Flexibility of frame and proper arrangement of equalizing levers. Position of Center of Gravity.
STRENGTH AND PROPER PROPORTION OF PARTS.

Mathematical analysis.

Results of experience.

SAFETY.

General arrangement, signals etc.

Proper design of boilers, stays and parts.

Design and arrangement of attachments and piping.

Design of parts for strength.

Brakes.


FACILITY OF OPERATION.

Convenient location of boiler and eng. attachments.

Convenient couplings.

Accessibility of parts. Roomy cab, etc.

ECONOMY.

Quality of fuel. Few repairs.

" " feed water.

Accessibility of parts for repair.

Type of boiler. Steam distribution.

" " furnace. Management (starting, stopping, etc).
Rapid Steam Qualities.

In all boilers it is desirable to make the heating surface such that the temperature of escaping gases, as they leave the last points of contact with the plates, shall be higher than the temperature of the steam at the required pressure. The higher this temperature, the greater the economy in space; the lower, to a certain limit, the greater the economy in fuel. Therefore, a locomotive boiler as commonly designed, allows a great deal of heat to be carried off with the escaping gases, especially when working with strong draught. For this reason, together with the fact that water will absorb more heat from a given surface than that surface (if iron) will absorb from heated gases, in cases of strong draught retarders, and ribbed (Serve's) tubes have been found to increase economy from 12 to 19 per cent.

The bounding surfaces of the fire box are the most efficient steam producers. A flat plate parallel, or a curved plate concave to the fire, are superior to convex plates or plates held at an angle with the direction of radiation. Tubes should be arranged in vertical rows so that steam may rise to the surface in as nearly a vertical direction as possible. To effect this, lay out the tubes on lines inclined at 60° with the horizontal, and in vertical rows, the distance from center to center = \(\text{diameter} + \frac{1}{2}''\), or

\[
\text{distance from center to center} = 15 \left(\frac{\text{no. tubes}}{15}\right) \times \left(\frac{\text{diameter}}{16}\right) + 1''
\]

Meyer suggests that diam. = \(\frac{1}{65}\) length. This will give results a trifle larger than common practice. However, in order to have sufficient draught to prevent choking of the flues by soot and ash
it has been found necessary in practice to restrict the total area of the tubes to about \( \frac{1}{8} \) the grate area. Therefore, since area varies as square of the diameter, and heating surface directly as diameter; the diameter of tubes or the number of tubes will be limited. The common practice is to make the tubes 3" diam; the number equalling \( \frac{\text{total tube area}}{\text{area single tube}} \). If the required number of tubes then be laid off as explained, the diam. of boiler may be found, a space above tubes \( \frac{1}{3} \) diam. boiler being kept clear for steam.

In locomotives burning soft coal it is necessary in order to permit sufficient depth of fire, to drop the fire box between driving axles. The width of frames thus limits its size one way, and the distance between axles, limits its size the other way.

Knowing the amount of coal ordinarily burned in a locomotive of the type under consideration, per lb. of water evaporated per hour, and knowing the required power or necessary steam consumption of the engine, we may calculate the amount of coal required per hour. Dividing this by lbs. coal commonly burned per \( \frac{\text{sq. ft.}}{\text{grate}} \), we will have necessary grate surface. All of which shows that the empirical formula, 1 sq. ft. grate to every 600 lbs. tractive effort, which is found to agree fairly well with practice, is reasonable as having a rational basis. For hard coal divide the tractive force by 500, this gives for "963", 30 sq. ft., slightly over the actual area, 27.2 sq. ft. For "999" we get 25.4 , the actual area being 30.4. The small calculated value is due to reduction of tractive power due to large 86" driver. This together with the fact that a fast pass-
enger engine with extra large drivers may be expected to make about the same R. P. M., would seem to argue against basing grate area upon tractive power. In the case of a locomotive with unusually large or small drivers it would be well to use another means for deciding grate area, dependent more upon the steam consumption used.

Perfect combustion while not at all impossible is not desired, the cost involved in saving the final heat units being greater than they are worth. What is desired is to get the most heat possible into the water, where the furnace is restricted as in locomotive boilers it is impossible to get economy and power at the same time. If fuel is cheap in the locality where the locomotive is to run, power is desired rather than great economy of fuel. If expensive, economy is to be desired even if trains are shorter, more frequent, and wage bill higher.

The most power is to be had when the grate is as large as possible, and the crown sheet is fairly close to the fire, with tubes large in area of opening, and 10' or 11' long. Blast should be as strong as can be had without causing excessive back pressure in cylinder. The exhaust should be variable, so as to avoid tearing up the fire in starting. An economical boiler must have as large a heating surface as is possible, especially in the fire box. Combustion must be slow, as forcing a boiler beyond a proper limit reduces its efficiency. Tubes should be small and long, say 15' X 1 1/3".

For either power, the fire box heating surface should be as large as possible, and the tube heating surface ordinarily as large as can be, without interfering with draught by friction. One grate
will do for either boiler as it may be blocked off for economy.

The intensity of radiant heat varies as the square of the distance from the source of heat. Therefore, deep leg boilers, while somewhat more economical than short leg boilers, are not as powerful as the latter, especially with anthracite coal. A brick arch is often used to cause heating of the gases as they come from the fuel. It also makes a longer way for the flame, and gives the gas a chance to burn before it is extinguished by the tubes. An objection to it is that it shields the crown sheet from radiant heat, thus reducing power somewhat. But this objection is slight as compared with the better combustion resulting, the mingling and heating of the gases, and the better conditions of their passage through the tubes. (as loco. boilers)

The combustion space of internally fired boilers should be large with bars not too high. This allows the air to mix with the gases and permits of a thick fire. About 3 ft. is commonly allowed from surface of fire to plates. If the distance is too great there will be a loss of radiant heat, (the intensity of which varies inversely as the square of the distance). If too little the plates may be damaged.

The rate of combustion per sq. ft. grate, depends upon draught, quality of fuel, and skill of firemen. In locomotive boilers it will vary with weight of train, speed, and grades.
A somewhat recent type of fire box is that designed by J. E. Wotten, for the combustion of anthracite and lignite. The fire box extends clear over the wheels, giving a grate area of 65 to 85 sq. ft. Very fine coal may be used, as the draught over this area is so slight as not to raise it. The grate is composed of alternate water tubes and bars, or shaking grates may be used. The crown sheet is low, being 2' 5" from grate in certain existing express locomotives with this type of boiler. A 27" combustion chamber leads to the 184, 1 3/4" tubes.

The heating surface is,

\[
\begin{align*}
\text{Fire box} & \quad 135 \text{ sq. ft.} \\
\text{Flue tubes} & \quad 932 \text{ " } \\
\text{Heat surface} & \quad 1117 \text{ sq. ft.}
\end{align*}
\]

\[
\frac{\text{heat surface}}{\text{grate area}} = 14.7.
\]

Average coal consumption per sq'. grate = 34 1/4.

Kent, quoting A. E. Mitchell in paper before N. Y. Railroad Club, says, "Square feet of boiler heating surface for bituminous coal should be less than 4 times the diameter in inches of a cylinder 1" steam larger than the cylinder to be used."

The following table is calculated for this rule.

<table>
<thead>
<tr>
<th>Column 1 refers to sheet of dimensions</th>
<th>1</th>
<th>2</th>
<th>3</th>
</tr>
</thead>
<tbody>
<tr>
<td>For Brooks practice, (963.)</td>
<td></td>
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<td></td>
</tr>
<tr>
<td>&quot; &quot;</td>
<td>41</td>
<td>1134</td>
<td>1511</td>
</tr>
<tr>
<td>&quot; &quot;</td>
<td>3</td>
<td>1018</td>
<td>1398</td>
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<td>&quot; &quot;</td>
<td>13</td>
<td>1237</td>
<td>1569</td>
</tr>
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<td>5</td>
<td>1195</td>
<td>1587</td>
</tr>
<tr>
<td>&quot; &quot;</td>
<td>6</td>
<td>1257</td>
<td>1832</td>
</tr>
<tr>
<td>&quot; &quot;</td>
<td>11</td>
<td>1257</td>
<td>1933</td>
</tr>
<tr>
<td>&quot; &quot;</td>
<td>7</td>
<td>1257</td>
<td>1833</td>
</tr>
</tbody>
</table>

See tabulated list of representative passenger locomotives,- Plates.
For Penn. R. R. practice............ : Calculated: Actual
" " " " " " " " " " " " " " " " " " : 17: 1257 : 1572
" " " " " " " " " " " " " " " " " " : 40: 1195 : 1918
" Baldwin " " " " " " " " " " " " " " " " " " : 22: 1357 : 1579
" " " " " " " " " " " " " " " " " " : 25: 1385 : 1912
" B. & O. " " " " " " " " " " " " " " " " " " : 2: 1134 : 1511

\[
\frac{12 \times 14668}{12} = 28155
\]

Average ... 1217 1679.3

Which shows that while the rule may be all right as regards minimum surface, the result obtained fall below the best examples of modern practice in fast engines. For closer agreement we may make our multiplier 5.44 instead of 4. Even then, however, the amount of surplus will have to be varied, depending on quality of fuel, ratio of fire box surface to tube surface and requirements regarding load, and capacity or economy.

For grate area, (Kent. Am. Mach.) gives the rule:

Displacement of one piston cu. ft. X 6 1/2 = (sq. ft.) grate area.

A comparison of results of this formula with areas in existing locomotives shows that the multiplier should be 7.5 to 7.6 instead of 6.5.
Total Heating Surface

Piston Displacement - cu. ft.

Numbers refer to dimension sheet.
Ttractive Force.
Diameter of Stack:

Meyer recommends that area of smallest cross-section of stack = \( \frac{1}{17} \) grate area. Comparing with our table of columns, we find:

<table>
<thead>
<tr>
<th>No.</th>
<th>Grate Area</th>
<th>Stack Area</th>
<th>( \frac{\text{Stack Area}}{\text{Grate Area}} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>42</td>
<td>27.2</td>
<td>254.47</td>
<td>0.107</td>
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<tr>
<td>2</td>
<td>28.33</td>
<td>247.45</td>
<td>0.114</td>
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<tr>
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<td>31.35</td>
<td>201.06</td>
<td>0.156</td>
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<tr>
<td>6</td>
<td>26.2</td>
<td>153.94</td>
<td>0.17</td>
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<tr>
<td>7</td>
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<td>201.06</td>
<td>0.136</td>
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<tr>
<td>8</td>
<td>17.8</td>
<td>201.06</td>
<td>0.088</td>
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<td>254.47</td>
<td>0.1</td>
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<tr>
<td>13</td>
<td>28.17</td>
<td>213.82</td>
<td>0.131</td>
</tr>
<tr>
<td>16</td>
<td>22.6</td>
<td>143.14</td>
<td>0.16</td>
</tr>
<tr>
<td>19</td>
<td>25.29</td>
<td>153.94</td>
<td>0.164</td>
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<tr>
<td>21</td>
<td>30.7</td>
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<td>22</td>
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<tr>
<td>37</td>
<td>26.41</td>
<td>188.69</td>
<td>0.14</td>
</tr>
</tbody>
</table>

The average ratio, 0.132, shows that the value 1/17 agrees very well with modern practice.
To find area of exhaust nozzle, if double, divide area of grate \( n \) square inches by 400. If single, by 200.

If necessarily short, the exhaust pipe should be double.

If long, as is ordinarily the case with long smoke boxes, the pipe will either be single or have nozzles converging slightly toward center of stack. If single, bridge should be high enough to prevent as much as possible, back pressure in one cylinder caused by exhaust from the other. Care should be taken that area of nozzle is not so small, relatively, to area of single pipes at bridge, as to cause this. The bore of nozzles should be cylindrical for about an inch at the end, to prevent spreading of exhaust steam. Passages in the pipe should be smooth, with easy continuous curves. A thin orifice is a good one, in that it is easily removed, and may be bevelled off on outside toward top, for passage of gases.
There should be at least three nozzles with every engine.

A general, rough idea of the size may be obtained from above rules, several nozzles made of sizes varying more or less from this, and the one selected which upon trial gives the best results, considering requirements of that particular locomotive as regards speed, power, or economy.
Design of the Machine Parts.

In order to compare the results obtained by the use of various existing formulas for the design of engine parts with current practice, these formulas are either given and one or two representative cases calculated and results compared with practice, or, they are plotted on cross-section paper, existing sizes being represented by dots or a curve connecting them.

Locomotives being subjected to very much rougher usage than stationary engines, it becomes a question as to how far we may be guided by formulas intended for the latter. In many cases, as when an existing formula has a rational basis, it will be an easy matter to adjust constants to correspond with common practice. Most of the plates are self-explanatory as, for example, the plate on diameters of piston rods.
Thickness of Cylinder.

In locomotives the piston speed is very high, and in spite of precautions ashes may sometimes be drawn into the cylinders. The thickness of cylinder walls must be such that fracture, due to pressure or jarring, cannot take place. Sufficient thickness to allow of reboring must be added, e.g. $\frac{1}{4}$ or $\frac{1}{8}$ inch. Calculations based upon steam pressure alone would give absurdly low results, when compared with thicknesses used in practice, and which at once allow for pressure, wear, re-boring, jigs, and strains caused either by contraction in the mould or by expansion and contraction due to heat, when in use. The following formulae, supposed to give fair results, are plotted on accompanying sheets:

\[
t = 0.0004pd + 0.5 \quad (a)
\]
\[
t = 0.0033pd \quad (b)
\]
\[
t = 0.002pd + 0.6 \quad (c)
\]
\[
t = 0.0001pd + 0.15\sqrt{D} \quad (d)
\]
\[
t = 0.03\sqrt{pd} \quad (e)
\]
\[
t = \frac{D + 2.5}{1900} \quad (f)
\]
\[
t = 0.00033pd + 0.8 \quad (g)
\]
Formulas (b) and (d) gave fair results, but are a trifle low. About \(\frac{1}{2}\) inch added would give somewhat better results. (c) gives results which are about right. All of the rest are decidedly high.

_Cylinder heads._

Those used on "961 class are well stiffened. Seating (Kent) objects to webs because a nick in their outer edge, which is in tension, may start a crack. This is in accordance with J.C. experience, several covers having broken in service. The tensile strength of the bolts used is commonly much higher than is necessary to resist steam pressure alone. The strain due to improper adjustment, or, because of unequal packing, to one bolt carrying more than its share of load, may be greater than that due to steam pressure. Sheds are preferred to bolts, because their renewal does not involve tearing out the cylinder lagging.
Cylinder flanges should be heavier than cover flanges, in order that the least possible damage may be done if a cylinder head is blown off.
Diameter of Cyfinder

Thickness of Cylinder

\[ T = \frac{(D+2.5)}{1900} P \] (Tredgold.)

Diameter of Cylinder
Diameter of Cylinder inches.

Thickness of Cylinder.

\[ T = 0.0023PD + 0.6 \] (Weisbach)
Thickness of Cylinder:

\[ T = 0.004pD + 0.5 \text{ (SeaToH)} \]

Diameter of Cylinder.
Thickness of Cylinder vs. Diameter of Cylinder

\[ T = 0.00033 pD \] (Quoted in Kent)
Thickness of Cylinder.

Diameter of Cylinder.

\[ T = 0.001 \rho D t \cdot \frac{1}{13V^8} \] (Van Buren).
Mani Bearing.

For lubrication, Meyer divides total weight on one driven by 160, which gives projected area of journal. This is quite a little lower than is used on the engine tabulated, which run about 215 * per. sq. inch.

Mani Axle.

The formula for diameter of shafts, to resist torsion,
is\[ d = \sqrt[3]{\frac{5.17}{5}} \]

\( T = \) torsional moment in inch pounds, \( S = \) shearing resistance of material, \( d = \) diameter of shaft. To calculate a shaft by this formula would require that one side be supposed to transmit its total power to the opposite side of engine. But since, in the design of locomotives, there is a relation between "torsional moment" and tractive force calculated from friction of wheels on rails, we may express extruding sizes as a function of tractive force. This torsional moment = Cyl. Area x Pressure x \( \frac{1}{2} \) stroke =

(approx) tractive power, \( o_r, = \) tractive power x constant \( c, = c \).
Placing all constants outside of the parenthesis, and combining, we have, \( a = k(t^3) = k^{3/2} \), where \( k \) is the cube root of all constants combined. Assuming a ratio of wheel to crank radius \( r = 6 \), and remembering that, in equating cylinder power to traction force a high coefficient of friction is used, we have \( a = 0.3 \sqrt{r} \). The traction force \( t \) should here be figured on the basis of a coefficient of friction \( = 0.2 \). This gives for \( a = 7.5 \).
Cranks, Puis.

Marka gives the formula, \( l = 1.038 \frac{\text{I.H.P}}{L} \).

Whitman recommends \( l = .9075 \frac{\text{I.H.P}}{L} \).

\( l \) = length in inches, \( f \) = coefficient of friction = .03 to .1 for different degrees of lubrication. \( L \) = length of stroke, feet.

Above are for marine practice.

For locomotive cranks puis, Marka gives \( l = .00000247f \text{P.N.D} \).

\( p \) = M.E.P. \( N \) = number of single strokes per minute. \( D \) = cylinder diameter. Whitman gives, \( l = \text{R.P.M.} \times \text{mean total load} \),

Rankine, \( l = \frac{p(V+20)}{44800d} \)

\( V \) = velocity of rubbing surface, feet/minute.

The above formulas are intended to give length which will prevent heating. An increase in length is more effective in this respect than an increase in diameter, as the large bearing surface required is thus obtained without an increase in relative velocities of rubbing surfaces. But considering the strength and rigidity limit the length that may be used.
For strength, \( d = \sqrt[3]{\frac{5.1 \text{ pl}}{t}} \)  
\( d = \text{diameter of pin} \)  
\( P = \text{maximum load on piston} \)  
\( t = \text{maximum allowable stress per square inch} \)  

For rigidity, \( d = 0.945 \sqrt[3]{\frac{(\text{HP})^2}{L \cdot N}} \)

If the pin is calculated for rigidity \( d \) it will be more than as strong as if calculated for strength.

Meyer, assuming \( d = 1\frac{1}{8}L \), which is about the ratio used in practice, and adding \( \frac{1}{8} \) or \( \frac{1}{16} \) for wear, allows 1600 lbs. per square inch projected area on main crank pin, and makes side rod pin diameter = .67 of this.

The above formula for rigidity gives results which are much smaller than are commonly used. The formula \( d = 0.0238 \sqrt[3]{\frac{\text{pl}}{t}} \) gives very fair results, and for \( \frac{t}{d} = 1.1 \) gives in ordinary cases (above 100 psi pressure) about the same results as Meyer's formula, \( d = \frac{P}{L \times 1600}, \text{ i.e. 1600 psi per square inch = allowable pressure on piston} \).
The formula for length, \( l = 0.0000247 f P N D^2 \) gives approximate results for high \( M.E.P. \), but is not to be depended upon unless the \( M.E.P. \) be taken of such value as to make the formula come out all right! The best way seems to be to use either Meyer's or Mark's formula, assuming \( \frac{e}{d} = 1 \) to \( 1.1 \), and adding a small fraction for wear and to bring the size to a convenient division of inches. Mark's formula is \( d = 0.0238 \sqrt{\frac{P e}{d}} \) \( P = \) max. load or piston). In Meyer's formula, \( d = \frac{P}{l \times 1600} \), if \( l = 1.1 d \), we have \( d = \sqrt{\frac{P}{1.1 \times 1600}} = \sqrt{\frac{P}{1760}} \). An inspection of the curves representing these two formulas shows that, for steam pressure between 175 and 200 "., very good results may be obtained, for 9 out of 12 of our engines listed on Amention Sheet having this pressure have crank pins corresponding to either curve when brought up to next higher even measure. It would be much easier to remember that crank pins for high pressures should be 5 1/2 " long, than to use this formula: but the good results obtained for this pressure indicates
\[ d = \sqrt[3]{\frac{5.1Pd}{t}} = \sqrt[3]{\frac{5.1}{t}} \sqrt{P\frac{d^3}{d^3}} = 0.0238 \sqrt{\frac{Pd}{d^3}} \]

\[ P = \text{total hor. press.} = \text{max. load on pist} \]

\[ t = \text{load (allowable) per. sq. in. of metal.} \]

\[ l = \frac{\text{length}}{\text{diam.}} \quad \text{(Thurston.)} \]
Diam. crank pin calculated for rigidity. (Marks)

\[ d = 0.066 \sqrt{\frac{P}{l^2D^2}} \]

- \( P \) = max. press.
- \( l \) = assumed length.
- \( D \) = cyl. diam.
Length of pin of 962° and others.

\[ L = 0.0000247 f(MEP) ND^2 \]
\[ f = 0.7 \]
\[ D = \text{diam. cyl.} = 18'' \]
\[ N = 71 \text{RPM}. \]

(Thomps.)
Diam. of crank pin.

\[ d = 0.0238 \sqrt{\frac{P}{d}} \quad \text{(Marks)} \]

\[ P = \text{total press. on piston.} \]

\[ d = \sqrt{\frac{P}{1750}} \quad \text{(Meyer)} \]

\[ P = \text{maximum steam pressure.} \]
that the formulas are applicable to other pressures,
high or low.

Crosshead Pivis. These are made short, in order to reduce
the binding effect produced by side play in passing around
curves. This in turn, in order to allow sufficient rubbing
surface to prevent friction, necessitates so great a diameter
that, if the pin be designed for rubbing surface, it will
be more than strong enough. Meyers allows 2880 * per. sq.
inch projected area for working pressure. This is well
within the figures used in practice, as a glance at the
dimension sheet will show. Ratio of length to diameter is
1 or a trifle greater.
Area of Cross head slides.

Seaton advises a pressure of less than 100 psi per sq. inch.

Rankine: \( P_0 = \frac{14800}{V+20} \)

\( P_0 = \) pressure per sq. inch.
\( V = \) vel. ft. per min.

Whitham: Area of slides = \( \frac{3.85 + d^2 P}{P_0 V w^{0.5}} \)

\( P_1 = \) press. per sq. inch in piston.
\( w = \frac{\text{Connecting rod}}{\text{crank}}. \)

Current British practice is about 40.

Phurston's formula implies 80, about.

Rankine gives 72.

Rank: "There is perfect agreement among the authorities as to the formula for area of slides, \( A = \frac{\text{Plan}}{P_0} \), but the value of \( P_0 \) varies from 35 to 500. The proper value of \( P_0 \) would seem to be 40 or 50."
Area of cross head guide.

\[ \text{Area} = \frac{.1854 \times D^2 \times p}{p_0 (\text{in}^{-1})} \] (Whitham)

- \( D \) = diam. piston.
- \( p \) = max. steam pressure.
- \( p_0 \) = allowable press. per sq. in. on guide.
- \( \tau_1 \) = corr. load / crank.

\( p \) = max. steam pressure.
Modern practice.
Sizes taken from 30 representative locomotives.
Piston Rods.
The rod crosses, showing sizes used on locomotives given in table of dimensions, seem to be very widely scattered. This, however, is due to the fact that a rather large horizontal scale is used. 3\(\frac{\text{3}}{\text{8}}\)^{\text{\textdegree}}\, \text{diam.} seems to be about the average. It is interesting to note that locomotives with the same size cylinders but much different boiler pressure, have in many cases the same size piston rod. Thus 3\(\frac{\text{1}}{\text{4}}\)" is used for 19" cylinder and 190, 180, 165 \& 13. P, and for 18" cylinder, 200, 180, 165, 160 \& boiler pressure. Manks' formula, \(d = \sqrt[6]{\frac{1}{3}D^2 p}\) would answer well for these results, if about .35" be added to result thus obtained. The slope is about right. A little stiffness is sacrificed in order to reduce weight of reciprocating parts, and, in turn, the excess weight of counterbalance.
Riveted Joints.

Punched holes must be made of such size with reference to the thickness of plate that the punch will not crush, i.e., diameter $\geq \frac{t S_s}{0.25 S_c}$ where $S_s$ is the shearing strength of the metal, $t$ is thickness, and $S_c$ is allowable crushing strain on punch. Practically any size may be chosen for drilled holes. Efficiency of joint, however, necessitates holes of larger size than above formula calls for, as we shall see later. Kent gives tabulated results of tests of drilled, punched and rolled, and punched plates, showing clearly the superiority of the former two methods over the latter. Experiments on hand and hydraulic riveting, while slightly in favor of hand riveting as regards ultimate strength, show it to be much
inferior as regards first visible skip. (Ratio about \( \frac{1}{2} \)).

For a lap joint with a single row of rivets, the formula

\[
d = \frac{4 Sc t}{\pi S_s}
\]

for diameter, and

\[
a = \frac{\gamma m Sc}{11 n S_s} \left(1 + \frac{m Sc}{S_t}\right) t
\]

for pitch,

where \( d \) = diameter of rivet, \( m \) is number of rivets in pitch

\( a \), and \( n \) is number of rivet sections receiving shear,

in same space, give \( d = 2.04 t \), \( a = 4.75 t \), and

efficiency \( \gamma = 0.57 \). For lap joint, double riveted,

\( d = 2.04 t \), \( a = 7.48 t \), \( \gamma = 0.73 \). This assumes \( S_t = 9,000 \).

\( S_s = 7,500 \), \( Sc = 12,000 \).

Honey gives:

Single lap, \( d = 2.33 t \). \( a = 6.7 t \).

Double lap, \( d = 2.33 t \). \( a = 11 t \) -- iron

\( a = 9.33 t \) -- steel.
A rule which has long been used in practice is: 

\[ a = d + \frac{\pi d^2}{4t} \quad \text{for single and} \quad a = d + \frac{\pi d^2}{4t} \quad \text{for double riveting.} \]

Ordinarily, the efficiency of riveted joints is as shown by Peabody,

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>58%</td>
<td>67.5%</td>
<td>78.6%</td>
<td>82.4%</td>
</tr>
</tbody>
</table>

Knowing the efficiency of the chosen joint we may calculate thickness of boiler plate. In order to keep the plate as thin as possible an arrangement of rivets should be chosen which will give as high an efficiency as possible and still remain tight at calked edge.
Riveted joints.
Pitch, overlap, diam. & space.

Thickness of plates, inches.
Double riv butt, with double strips.

Wilson.

\[ P = \frac{4d}{t} + d \]

\[ P = 3.3d \]

Diam. of rivet

Thickness of strip

Thickness of plate
Single riveted lap joints.

\[ p = \text{pitch}, \quad d = \text{diam. of rivet}, \]
\[ a = \text{area of rivet}, \quad t = \text{thickness of plates}. \]
The curves of diameters and pitch are self-explanatory.

The average as recommended by Meyer is higher than that given by Wilson, Fowey, or by calculation. The curve representing Wilson's values for practice is near the average of the formulas given in each case.
The valve used is the Allen American balanced valve. An overshoot port doubles the port opening within certain limits, and at the same time doubles the quickness of opening. This is well shown by Zener diagrams for different positions of link. See Plates. This valve, as used by the S.C.R.R. also very rapidly, which leads to the conclusion that it is improperly balanced. The following investigation is made with a view to determining as nearly as may be possible the exact pressures on valve face when valve is used under ordinary running conditions. In general, the method consists in finding the areas exposed by steam chest pressure, multiplying them
by that pressure, adding the area exposed to downward exhaust pressure (inside of balance ring) times that pressure, and subtracting upward pressures, as of pressure of steam in overshot port against over shot port area, and exhaust pressure times area of top of exhaust port. The pressures existing at different parts of the valve travel are taken from actual indicator diagrams, and piston positions corresponding to the assumed valve positions are found by reference to Zener diagrams.
Pressure of Value on Seat. See drawing of Value.

Area of ring, outside measurement, in ordinary position, is 148.5 sq. m.

Area of uncovered or unbalanced portion of value = 106.49.

The ring projects 5/8" beyond value. Area of each segment thus formed = 2.37. Area of both segments = 4.74. Area of ring = 148.5. Area covering value = 148.5 - 4.78 = 143.72.

Area of value face = 20 x 12.5 - .215 (corners rounded) = 249.78.

Area of uncovered portion of value = 249.78 - 143.72 = 106.06 sq. m., as stated above, line 4.

Projected area of cotton of overshot, deducting webs = 165.5.

" " " " " " top " " 174.4

Difference, = (area of overshot ports) = 8.6

Area of inside of ring, exposed to exhaust pressure = 130.2"
Beginning with valve at the end of its travel, full
squeeze, we will follow it across its seat toward the right,
finding the total unbalanced pressure. This will equal
(a) the steam chest pressure x excess of area of valve
face over balance ring, plus (b) exhaust pressure x excess
of area of inside of ring over area of exhaust port in
valve, minus (c) steam pressure x area of cover shot ports
(this = excess of area of top of cover shot - projected - over
bottom of cover shot, and pressure acts upward),
minus (d) the pressure in cylinder x area of either steam
chest port which may be covered by valve, during
compression, or minus (e) steam pressure during expansion
x area of covered port, minus (f) steam pressure in chest
x area of overhanging portion of valve which is covered
or balanced by ring. Area of balanced portion of over-shot
port is subtracted from this, as it has already been
included in (c).
Thus, when valve is in extreme left position, (distance from mid-position = \(3\frac{3}{4}\)"), full gear, the pressure drops:

Steam pressure \(\times\) unbalanced portion of valve, \(= 106.8 \times 200" = 21,360"\)

Excess area of balance nigg over exhaust port in valve, \(\times\) back pressure \(\times\) left of cylinder.

The pressure up is:

- Balanced overhanging portion of valve minus balanced over shot port \(\times\) steam pressure, \(= 23.45 \times 200" = 4690"\)

Area of over shot ports \(\times\) steam pressure \(= 21.25 \times 200" = 4250"\).

Area of valve bridge between over shot and exhaust ports \(\times\) back pressure, \(= 36\) B.P.

As the valve travels to the right, the pressure drops due to upward pressure decreasing as overhanging edge decreases. After \(\frac{1}{2}"\) movement (dist. from mid-position \(= 3\frac{3}{4}"\) the pressures are

Downward, same as before, unless back pressure has changed. See indicator card for any special case and find corresponding piston positions from Zenneur circles.
Upward, balanced overhang - balanced over shot port. x steam pressure, = 17.95 x 200 = 3590." 
Over shot ports x pressure = as before 4250. 
Value bridge, 17 5/8 x back pressure. 
When right hand edge begins to overhang, (distance from mid position = 3.03") the pressures are 
Downward, 21200 + effect of back pressure in top of balance area. 
Upward, 15.05 x 200 + 4250 = 7260". 
Value bridge, area exposed, = 6.36 x back pressure. 
When valve is 2 3/4" from mid position, the pressures are 
Down, 21200 + H2.25 x back pressure. 
Up, 11.15 x 200 + 4250 = 6480." 
Value bridge, 935 x back pressure.
When over shot on the right begins to open, valve being 2 3/8" from mid-position, the pressures are,

Down, 21200 + 42.25 x back pressure.
Up, 11.12 x 200 + 4250 = 6474.

Valve bridge, 17 x back pressure.

At 2" from mid-position, pressures are,

Down, 21200 + 22.25 x back pressure.
Up, 5.34 x 200 + 4250 = 5318.

Valve bridge, 6.36 x back pressure.

When right edge closes right port, (1.32" from mid-position)

Down, 21200 + 42.25 x back pressure.
Up, 2.5 x 200 + 4250 = 4750.

Valve bridge left side, 10 1/2 x back pressure.

With no over travel, (1.93" from mid-position)

Down, 21200 + 42.25 x back pressure.
Up, due to over lap of bridge over left port, and pressure of steam now expanding in right end of cylinder, against both over shot port areas.
= 17 x back pressure + 21.75 x steam pressure in right cylinder.
At 1/2" from mid position:
Down, 21200, exhaust being closed.
Up, area of left port x compression pressure,
= 29.75 x pressure, +
Area of right port x expansion pressure, = 29.75 x pressure
Up, 29.75 x compression pressure + 29.75 x expansion pressure.

Indicator card #1, taken June 16, 1897, time 2:59 P.M.,
left side, will be taken to show the application of
the above method of finding pressures on valve.
Steam chest pressure is assumed as equal to maximum
boiler pressure for slow running, and because the boiler
pressure sometimes exceeded that at which the safety
values were supposed to blow off. Cut off for this
card is approximately \( \frac{1}{3} \). The train had just left a station and the speed was moderate, therefore the back pressure was not as high as when running at high speeds, even with a shorter cut-off. Crank position's corresponding to the chosen valve positions were obtained from Zeuner diagrams, and corresponding cylinder pressures obtained from indicator card.

Value m. extreme position, \( \frac{1}{3} \), Pressure down = 23,735 \( \frac{\text{mm}}{\text{Hg}} \), Pressure up = 5696 \( \frac{\text{mm}}{\text{Hg}} \), Resultant, down, = 18039 \( \frac{\text{mm}}{\text{Hg}} \),

Area of bottom of valve-ports, \( = 123 \text{ sq} \). " per. sq. m. valve face = 147 \( \frac{\text{mm}}{\text{Hg}} \).

Valve position at 1.32"

Pressure down = 21950 \( \frac{\text{mm}}{\text{Hg}} \),

" up = 4939 \( \frac{\text{mm}}{\text{Hg}} \),

Resultant down = 18011 \( \frac{\text{mm}}{\text{Hg}} \),

" per. sq. m. valve face = 146 \( \frac{\text{mm}}{\text{Hg}} \).
Valve position at 93°:

Pressure down = 22,000\#  
\[ \frac{\mu_l}{\mu_p} = \frac{3535}{\text{per. sq. in.}} \]
Resultant down = 18465\#  
\[ \text{per. sq. in.} = 150\#. \]

Valve position \( \frac{1}{4} \)°:

Pressure down = 21,200\#  
\[ \frac{\mu_l}{\mu_p} = \frac{3863}{\text{per. sq. in.}} \]
Resultant down = 17457\#  
\[ \text{per. sq. in.} = 142\#. \]

Mid position: practically the same.
The excessive pressures here shown to exist might be remedied by the use of a larger balance ring, necessitating a turn a larger cover plate. On the square form of balance might be adopted to advantage, the advantages in regard to leakage, of the round over the square form being unmaterial, provided the latter is properly constructed.

Summary of pressures.

<table>
<thead>
<tr>
<th>Valve, inches from mid position</th>
<th>Pressure per sq. inch valve face</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.32&quot;</td>
<td>146&quot;</td>
</tr>
<tr>
<td>.93&quot;</td>
<td>150&quot;</td>
</tr>
<tr>
<td>.25&quot;</td>
<td>142&quot;</td>
</tr>
<tr>
<td>0.0</td>
<td>142&quot;</td>
</tr>
<tr>
<td>Extreme. 1.02&quot; Chosen</td>
<td>1.47&quot;</td>
</tr>
<tr>
<td></td>
<td>5/7 2.7&quot;</td>
</tr>
<tr>
<td></td>
<td>Average 1.45.4&quot;</td>
</tr>
</tbody>
</table>

Conclusion -

Value pressure downward averages 145" per sq. in. valve face, which pressure is excessive.