STEAM-ENGINE DESIGN.

FOR THE USE OF

MECHANICAL ENGINEERS, STUDENTS, AND DRAUGHTSMEN.

BY

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"Practice varies; but principles are eternal."

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STEAM-ENGINE DESIGN.

INTRODUCTION.

1. Types.—The efficiency of a steam-engine for any particular duty depends largely upon the type selected. In choosing any particular type, we must endeavor to secure one which has the greatest number of advantages and the least number of disadvantages. Since the designer is familiar with the various types of engines in use, only the conditions governing him in making his selection will be enumerated.

These conditions are, briefly:

1. Clearance.
2. Piston speed.
3. Friction.
4. Economy of fuel.
5. Weight and complexity of moving parts.
6. Accessibility for repairs.
7. Radiating surface.

2. Clearance is the volume included between the piston, when at the end of its stroke, and the valve-seat. The distance between the piston, when at the end of its stroke, and the cylinder-head is sometimes (inaccurately) called clearance. This latter is *piston* and not *engine* clearance.

Clearance is measured as a certain percentage of the stroke displacement of the piston. As the clearance volume is nearly constant for any fixed diameter, the percentage is greater for short- than for long-stroke engines. Hence it is well to have the stroke as long as possible, in order to eliminate the prejudicial effect of clearance. If the stroke is fixed, the clearance volume will be nearly directly proportional to the square of the diameter of the cylinder.

CHAPTER I.

DESIGN OF PISTON AREA, PISTON, THICKNESS OF CYLINDER, BOLTS, ETC., FOR A NON-COMPOUND ENGINE.

8. Diameter of Cylinder.—Having determined upon the power of the engine, its type, speed of piston, initial and terminal pressures of steam in the cylinder, we are prepared to begin the design of the parts for the duty required of them.

From § 3 we have

$$a = \frac{33000 \times \text{I. H. P.}}{p_e l n},$$

in which a = area of the piston in square inches;

I. H. P. = indicated horse-power of the engine;

p_e = mean effective or unbalanced pressure on each square inch of the piston in pounds;

l = length of stroke in feet;

n = number of strokes per minute;

Let p_1 = initial absolute pressure of steam in pounds per square inch at the cylinder;

p_B = initial absolute pressure of steam at the boiler;

p_2 = terminal absolute pressure of steam, or driving pressure at the end of the stroke;

p_3 = mean absolute back pressure against the piston per square inch;

p'_m = mean absolute driving pressure throughout the stroke in pounds per square inch.

If the pipe conveying the steam from the boiler to the steam-chest is short, direct, and lagged, the fall will not exceed $\frac{1}{13}$ of p_B, or

$$p_1 = \tfrac{12}{13} p_B.$$

Since the steam filling the clearance volume expands with that in the cylinder proper, we will divide the volume of clearance by the area of the piston, and find the length of cylinder

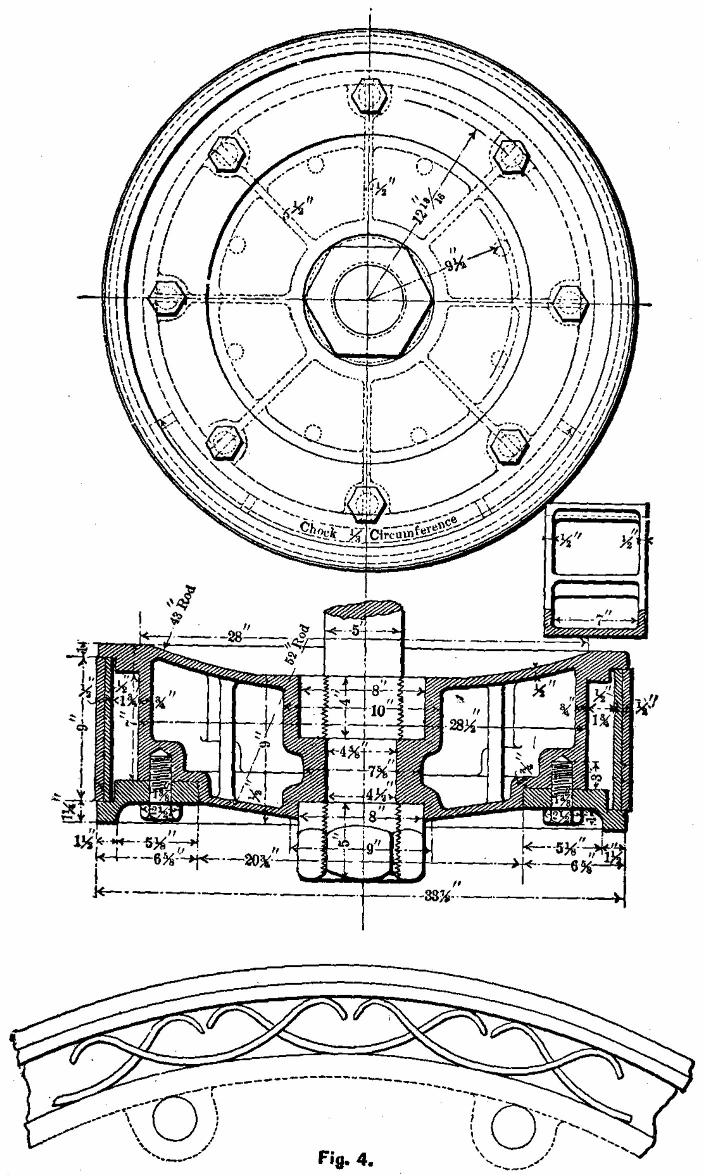

Fig. 4.

the late Mr. George H. Corliss by swelling the outer thickness about the middle of the cylinder's length.

Whenever the two cylinders are made in two parts, it is necessary to make a steam-tight joint at the free end of the bushing. This is done in various ways, but frequently the joint is made so tight that there is left no space for free expansion of the bush over the casing. In such a case the bushing has no special advantages. This is likely to occur only with long-stroke engines. To overcome this objection, Mr. E. D. Leavitt, Jr., casts the outer cylinder in two parts, and connects them by a swelled copper joint, extending circumferentially around them.

Fig. 18 is a drawing (from *Engineering*, xxii.) of the high-pressure jacketed cylinder of the U. S. S. *Nipsic*. The flat surfaces are to be proportioned by formulæ given in Chapter IV. The jacket-joint is made by brass or Babbitt metal rings which are forced into place by the screws, the screws abutting against the inside of the cylinder-head. The head has a manhole. In Fig. 60 is illustrated a form of plain cylinder.

15. EXAMPLE.—Design the cylinder and cylinder-head for $D = 37.25$, $P = 69$ lbs.

$$\text{Thickness of cylinder-head} = \frac{37.25 \times 69 + 500}{2000}$$

$$= 1.54 \text{ inches (Seaton, § 11).}$$

$$\text{Thickness of cylinder-head at flange} = \frac{37.25 \times 69 + 500}{1500}$$

$$= 2.05 \text{ inches.}$$

$$\text{Thickness of cylinder} = 0.03\sqrt{37.25 \times 69}$$

$$= 1.52 \text{ (Van Buren).}$$

Diameter of bolt-circle for the cylinder-head $= 37.25 + 2 \times 1.52 + 2 \times$ diameter of the bolt $= 42$ inches, on the assumption that the bolts are about $\frac{7}{8}$ inch in diameter.

The bolts should not be more than 6 inches apart. If they are 5.5 inches pitch there will be 24 bolts required.

The safe tensile strength of the 24 bolts must be equal to the maximum load on the cylinder-head, or

Sectional Elevation of Cylinders and Steam Chests

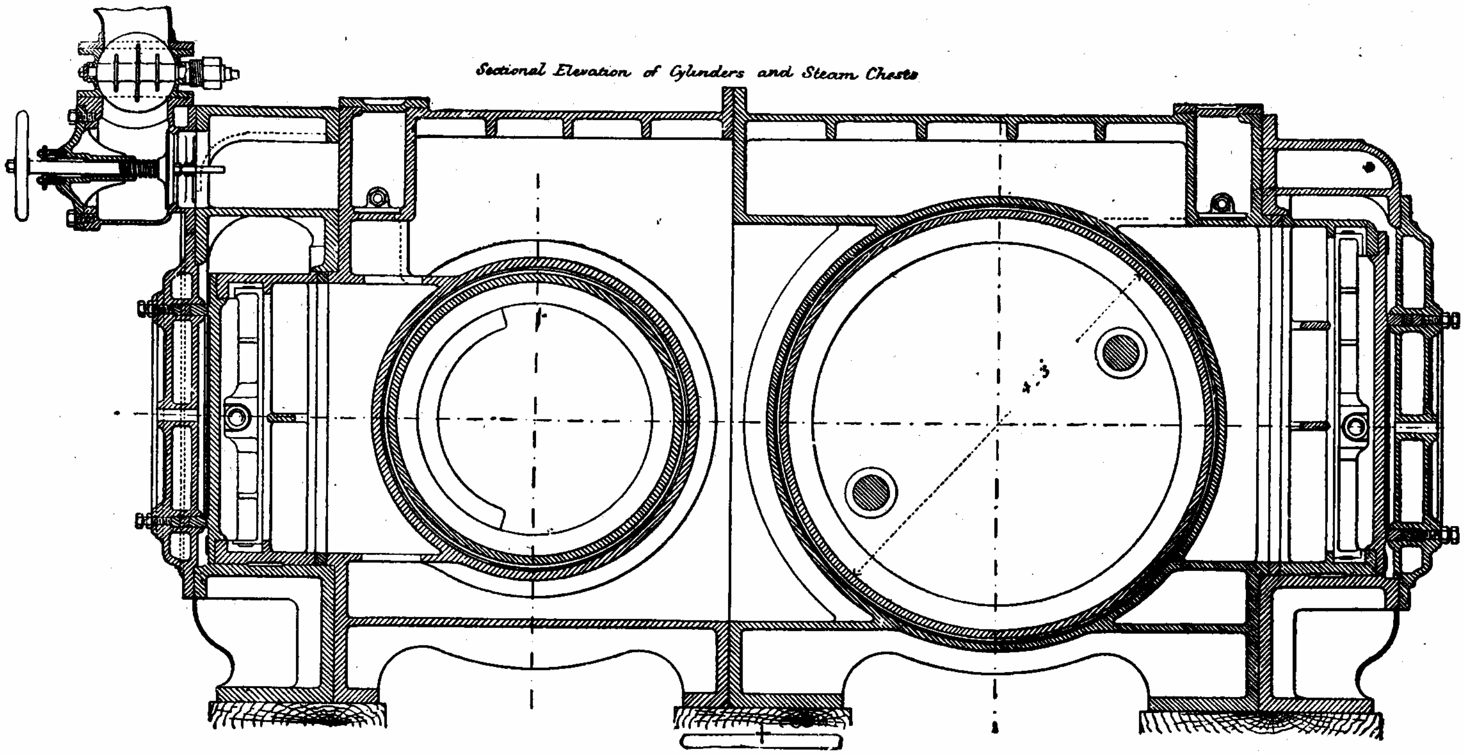

Fig. 18

$$24 \times \frac{\pi d^2}{4} \times 5000 = 69 \times 0.7854 \times (37.25)^2;$$

$$d = 0.89 \text{ inch} = \text{effective diameter of bolt.}$$

In this, 5000 is taken as the safe tensile strength of wrought-iron, and no allowance is made for the counter-bore of the cylinder.

17. Design of Valve-ports.—The ports or passages through which the steam passes in going from the valve-chest to the cylinder must be short, direct, of easy curvature, and large enough to prevent "wire-drawing" of the steam.

Length of port = 0.8 diameter of cylinder;
Area of exhaust port = $\frac{3}{2}$ area of steam-port.

Steam flowing at the velocity of 6000 ft. per min. (see § 3, and Rankine's *Steam Engine*) passes through the steam-port into the cylinder, where the piston is moving at from 100 to 1000 ft. per min. Hence, applying the "law of continuity" of fluids to a current of steam,

$$a = \frac{AV}{v} = \frac{\text{area piston} \times \text{piston's speed}}{6000};$$

where a = area of valve-port in sq. inches;
A = " " piston in square inches;
v = velocity of steam flowing through the port = 6000 ft. per min.;
V = velocity of piston in ft. per min.

The length of the port is long in comparison with its width, and the steam passages short and direct, in order that the clearance volume may be small.

CHAPTER II.

DESIGN OF SLIDE-VALVES.

18. Kinds of Valves.—A slide-valve regulates the admission and emission of steam to and from the cylinder. In order that it may be operated so as to produce the best effect, the steam space in the boiler must be large enough to give a uniform flow, and the communicating steam-pipe must not be too small or long, or have sharp bends. Valves are called oscillating, rotating, poppet, reciprocating, etc., according to their movement and form. The plane, reciprocating valve is called the slide-valve, and is in general use.

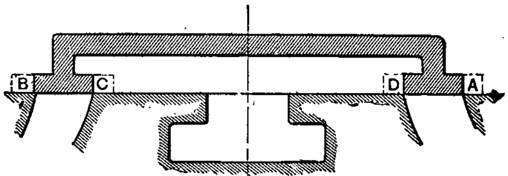

Fig. 21.

19. Square Slide-valves.—The slide-valve moves over a plane surface called the *valve-seat*. The under side of the valve is called its *face*. The square slide-valve has three ports in its seat, one at each end for the live steam to enter, and through which the steam is finally emitted as exhaust, and discharged into the central passage called the exhaust-port. The movement of the valve is controlled by an *eccentric*. This is a disk set eccentric to the shaft on which it is secured. The *throw of the eccentric* is the distance between the centre of the shaft and centre of the eccentric. The throw is usually equal to one

$AD = 4$ ft.; the scale for the eccentric orbit is $HB = 3$ **in.**; and the scale for pressures, 45 lbs. = 1 inch.

The steam enters at crank position OA, and closes at $\frac{5}{8}$ stroke, or OC; therefore the steam-port is wide open when the crank

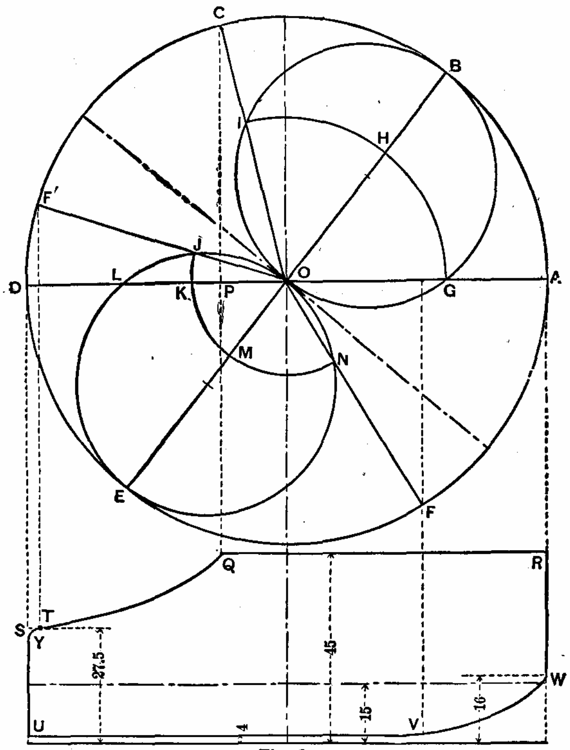

Fig. 27.

is midway between A and C, or at OB. OB is the valve-circle, OG is the steam-lap $= OH$, and HB is the 3-in. steam-port width given. On the scale that $BH = 3$ in., the lap $OH = 5$ in. The throw of the eccentric is $(OH + HB =)$ 8 in. The exhaust-valve closes when the crank is at OF, or 12 in. from the end of the stroke, compressing steam of $\left(\frac{30 - 22}{2} =\right)$ 4 lbs. pressure to $\left(4 \times \frac{12 + 4}{4} =\right)$ 16 pounds.

Steam-lap at the inner end of the valve $= 1.11$ in. $= KO$;
Exhaust-lap at the inner end of the valve $= 1.22$ in. $= HO$;
Point of cut-off for stroke from inner end $= 20$ in.

Fig. 30 is a sketch, from Seaton's *Manual of Marine Engineering*, of this valve.

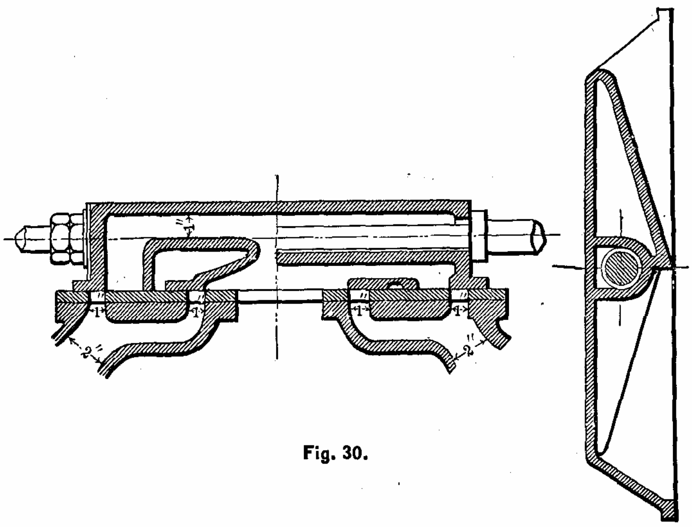

Fig. 30.

EXAMPLE 4. Fig. 31 is a sketch of a valve in use. The stroke of the piston is 36 in.; throw of the eccentric, 3 in.; length of the connecting-rod, 72 in.; clearance, $\frac{1}{12}$ of the stroke-displace-

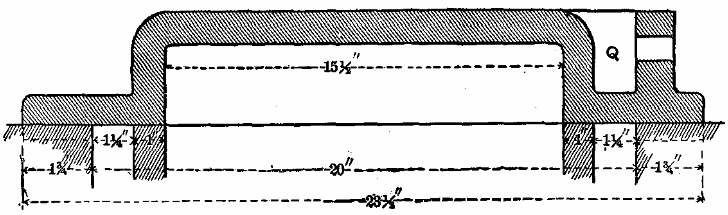

Fig. 31.

ment; the steam-port is open $\frac{1}{4}$ in. when the engine is on either centre; initial absolute pressure of steam, 45 lbs. per square inch; and the back pressure, 4 lbs. Required the point of cut-off for each stroke, position of the piston when the steam-port

APPENDIX.

134. Strength of Materials.—The following table is given on the authority of Rankine, Kirkaldy, Fairbairn, Wade, Barlow, Telford, Anderson (Com. B. A.), Shock, Lloyd, Hodgkinson, Trautwine, and others. The values refer to the ultimate resistance of the material in pounds per square inch of cross-section.

METAL.	Tension.	Compression.	Shearing.	Transverse or breaking-across.	Modulus of elasticity.
Iron, cast, average	16,500	86,000	19,000 to 28,000	17,000	18,270,000
" " with scrap	28,000				
" " pig	13 000				
" " first melting	20,800				
" " 2d "	24,700	99,680			
" " 3d "	26,800	140,000			
" " hot blast		111,300			
" " cold "		99,200			
Iron, wrought, bars, English	50,000 to 66,000	33,000 to 36,000	40,000 to 46,000	40,000	28,000,000
" " " American	45,000 to 70,000	"	"	"	"
" " large forgings	35,000				
" " hammered bars		33,000 to 36,000			
" " plates	45,000 to 65,000				
" " " beams				40,000 to 42,000	
Steel, plate, American	50,000 to 95,000				
" " Bessemer	95,000 to 112,000				
" rolled and hammered ingots	125,000				
" bar	95,000 to 120,000		82,800		36,000,000 to 42,000,000
" " tempered	214,400	50,000 to 60,000			
" chrome	180,000				
" cast		225,000			
Steel, hematite bar		159,578			
" American black diamond		102,500			
Copper, wrought	33,600	103,000			
" sheet	30,000				
" cast	20,000	117,000			
Gun metal, bronze	33,000 to 36,000				10,000,000
Bronze, 8 Cu, 1 Sn					9,900,000
Brass, cast	18,000	16,480			9,170,000
Aluminum bronze, 90 Cu, 1 Al	73,181				
Phosphor "	34,465				

135. Saturated Steam Table.—The following table is taken from Northcott's treatise on *The Steam-Engine:*

Absolute pressure in pounds per square inch.	Temperature of boiling-point in degrees Fahr.	Cubic feet of steam weighing one pound.	Latent heat of vaporization per pound of steam generated under a constant pressure.	Total heat units required to generate one pound of steam from water at 32° F. under a constant pressure.
0.0	−461.2	....	....	....
0.085	32	3390.0	1091.70	1091.70
1.00	102	332.6	1043.02	1113.05
1.25	109.6	269.3	1037.98	1115.62
1.50	115.8	226.8	1033.41	1117.26
1.75	121.3	196.1	1029.58	1118.94
2.00	126.4	173.0	1026.02	1120.49
2.50	134.8	140.1	1020.17	1123.06
3	141.6	118.0	1015.43	1125.14
3.5	147.8	102.16	1011.09	1127.02
4	151.1	90.12	1007.38	1128.63
4.5	157.8	80.67	1004.10	1130.07
5	162.3	73.50	1000.95	1131.44
5.5	166.4	66.77	998.08	1132.69
6	170.1	61.50	995.49	1133.82
6.5	173.5	57.03	993.11	1134.86
7	176.9	53.16	990.73	1135.90
7.5	180.0	49.79	988.55	1136.84
8	183.0	46.83	986.44	1137.75
9	188.4	41.87	982.66	1139.40
10	193.3	37.87	979.22	1140.89
11	197.8	34.60	976.06	1142.26
12	202.0	31.85	973.12	1143.55
13	205.9	29.51	970.36	1144.72
14	209.6	27.50	967.78	1145.87
14.7	212.0	26.36	966.08	1146.60
15	213.1	25.86	965.31	1146.93
16	216.3	24.32	963.06	1147.91
17	219.5	22.97	960.81	1148.89
18	222.5	21.76	958.68	1149.80
19	225.3	20.68	956.76	1150.62
20	228.0	19.70	954.79	1151.47
21	230.7	18.82	952.89	1152.30
22	233.3	18.01	951.05	1153.09
23	235.8	17.27	949.28	1153.85
24	238.2	16.60	947.58	1154.58
25	240.5	15.97	945.96	1155.29
26	242.7	15.39	944.40	1155.96
27	244.8	14.86	942.91	1156.60
28	246.8	14.36	941.50	1157.21
29	248.7	13.89	940.15	1157.79
30	250.5	13.46	938.87	1158.34
32	254.0	12.67	936.39	1159.41
34	257.4	11.97	933.98	1160.45
36	260.7	11.35	931.61	1161.46
38	263.9	10.79	929.36	1162.43
40	267.0	10.28	927.16	1163.38

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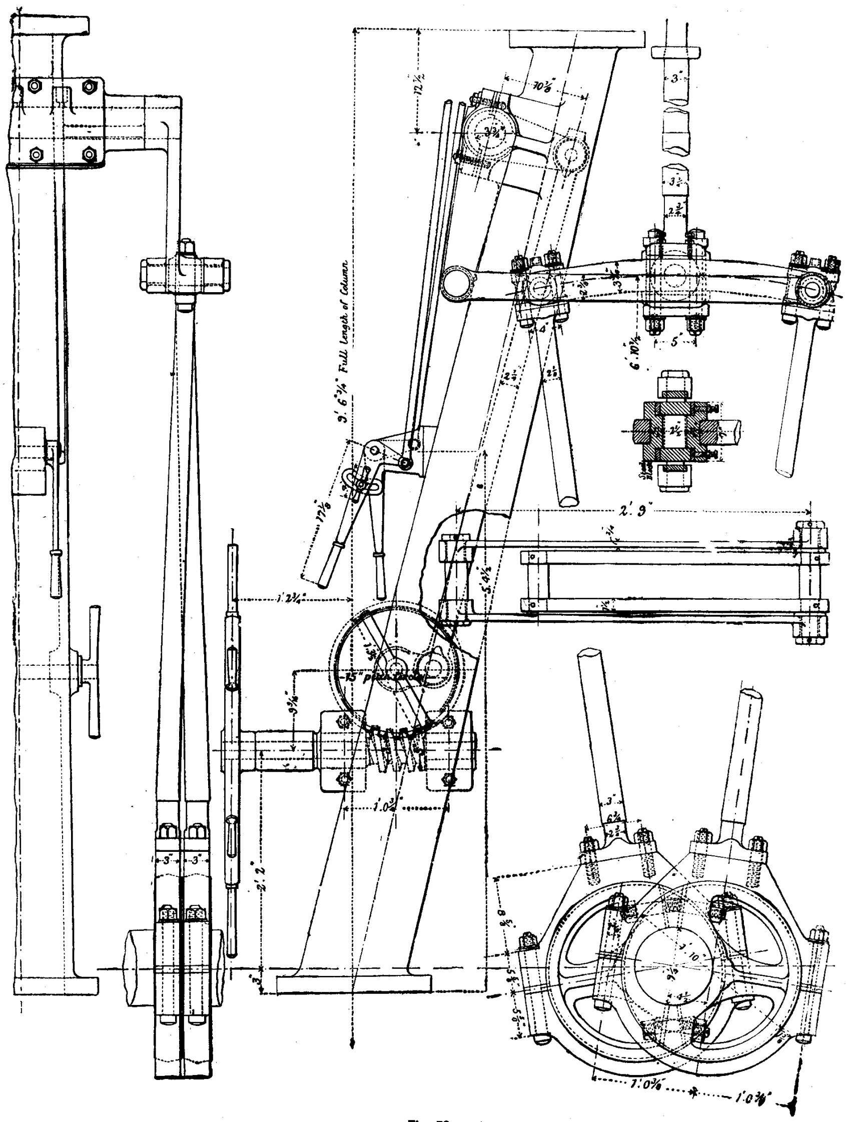

Fig. 78.

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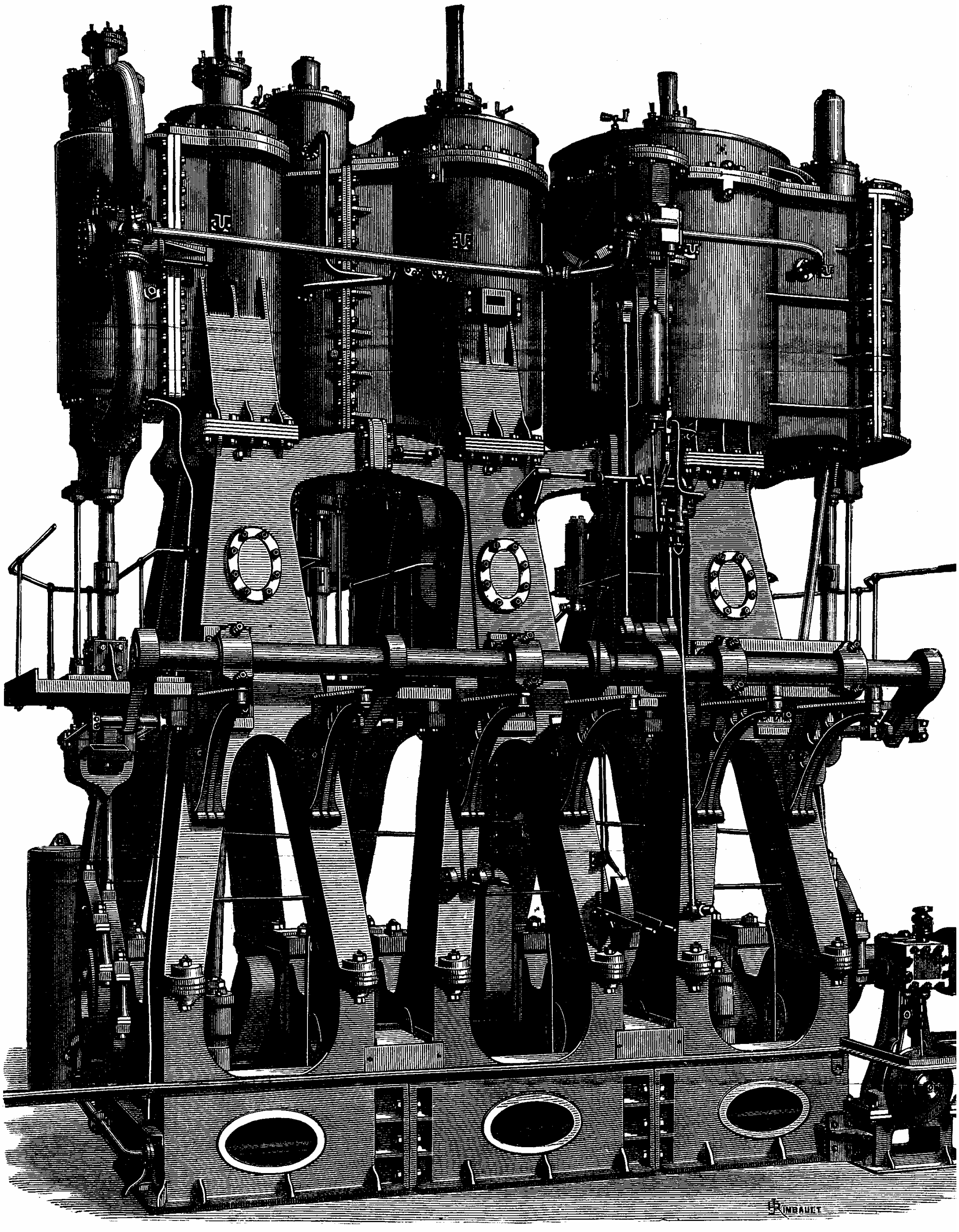

Fig. 91.

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TRIPLE-EXPANSION ENGINE OF SS. *Aberdeen*.