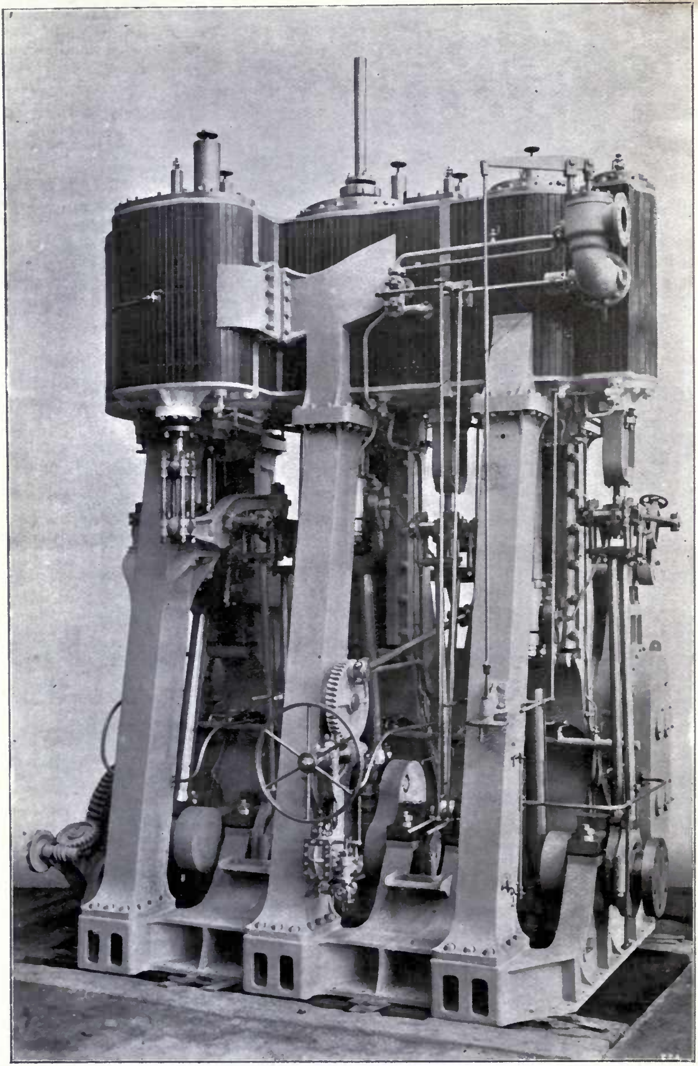
THE

PRACTICAL ENGINEER'S HAND-BOOK,

COMPRISING

A TREATISE ON MODERN ENGINES AND BOILERS—MARINE, LOCOMOTIVE, AND STATIONARY.



TRIPLE-EXPANSION MARINE-ENGINES OF 1200 I.H.P.

BY PALMER & Co., LIMITED, JARROW-ON-TYNE.

[*For Description see page* 415.

H. H. W.

COMPANION TO "THE WORKS' MANAGER'S HAND-BOOK."

THE

PRACTICAL ENGINEER'S HAND-BOOK

COMPRISING A TREATISE ON

MODERN ENGINES AND BOILERS

MARINE, LOCOMOTIVE, AND STATIONARY.

AND CONTAINING

A LARGE COLLECTION OF RULES AND PRACTICAL DATA RELATING TO RECENT PRACTICE IN DESIGNING AND CONSTRUCTING ALL KINDS OF ENGINES, BOILERS, AND OTHER ENGINEERING WORK.

THE WHOLE CONSTITUTING

A COMPREHENSIVE KEY TO THE BOARD OF TRADE AND OTHER EXAMINATIONS FOR CERTIFICATES OF COMPETENCY IN MODERN MECHANICAL ENGINEERING.

BY

WALTER S. HUTTON,

CIVIL AND MECHANICAL ENGINEER,

AUTHOR OF "THE WORKS' MANAGER'S HANDBOOK FOR ENGINEERS," "STEAM BOILER CONSTRUCTION," ETC.

With upwards of 370 Illustrations.

FIFTH EDITION, CAREFULLY REVISED, WITH ADDITIONS.

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PREFACE.

THIS volume is designed as a companion to the Author's book, "THE WORKS' MANAGER'S HAND-BOOK," and, like that work, contains a quantity of matter not originally intended for publication, but collected for the Author's own use in the practical construction of a variety of modern engineering work.

The Author has endeavoured to give in simple language, bereft of complex mathematical formulæ, and in a concise and condensed form, the most useful practical information, freely illustrated by well-executed engravings, which elucidate and shorten description; and to explain the true principles of, and give the principal rules for, the construction of engines and boilers, and the economical production of steam-power.

The whole comprises the results of recent researches and experiments in various branches of engineering, and contains a large collection of Tables and practical data invaluable to all engaged in designing, constructing or estimating for, engines, boilers, and other engineering work.

The principal subjects treated are as follows:—

Air, gas, and water; air-motors, gas-engines, and water-motors; fuel, combustion, products of combustion; heating-power and evaporative-power of fuel, evaporation of water to steam; heat and heat-energy; methods of burning liquid-fuel and refuse-fuels.

Evaporative effect of natural draught and forced draught in the furnaces of steam-boilers.

Propulsion of modern steam-ships, with description of, and rules for various propellers.

Evaporative-performance of various kinds of modern steam-boilers.

Boiler-explosions, their causes: and the usual manner in which boilers explode.

Rules and practical data for the construction of modern locomotive, stationary, and marine engines and boilers.

Rules and practical data for triple-expansion and quadruple-expansion engines.

Strength and specific gravity of a variety of materials, collated from careful experimental data.

It is hoped the volume will be found to be considerably in advance of any work of a similar description, and that, besides being a useful reference-book for engineers, it will prove extremely valuable to those who, having had the necessary practice, aspire to hold a Board of Trade Certificate for Competency in Mechanical Engineering.

PREFACE TO THE FIFTH EDITION.

THIS new Edition has been carefully revised, and improved by the addition of a little new matter to some of the sections.

W. S. HUTTON.

June, 1896.

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AIR, WIND, AND WIND-MOTORS; WATER AND WATER-MOTORS; HEAT AND FUEL; GAS AND GAS-ENGINES; COMBUSTION, ETC.

THE

PRACTICAL ENGINEER'S HAND-BOOK.

SECTION I.

AIR, WIND, AND WIND-MOTORS; WATER AND WATER-MOTORS; HEAT AND FUEL; GAS AND GAS-ENGINES; COMBUSTION, ETC.

ATMOSPHERIC AIR AND COMPRESSED AIR.

Atmospheric Air is a transparent invisible fluid surrounding the earth to a height of about 45 miles. Air is a very slow conductor of heat.

The Air is composed of nitrogen 78·5 parts; oxygen, 20·6; aqueous vapour, ·86; and carbonic acid, ·04 parts. Air contains about 4 grains of moisture per cubic foot.

Nitrogen does not support combustion or respiration, and has no taste or smell. It is lighter than common air; its specific gravity being ·9736.

Oxygen is essential to the support of animal life and combustion. It is one-tenth heavier than common air; its specific gravity being 1·1056.

Carbonic Acid is produced by the fermentation of animal and vegetable substances, the respiration of animals and the processes of combustion.

Carbonic Acid is composed of 2 atoms of oxygen and 1 of carbon. Its specific gravity is 1·529.

The Effect of Carbonic Acid is to destroy animal life and extinguish flame. When the atmosphere contains 8 per cent. of carbonic acid there is danger of suffocation, and air mixed with 10 per cent. of this gas will extinguish lights.

Carbonic Acid Gas is frequently found in mines, where it is called choke-damp, or black-damp.

The Weights and Volumes of the two principal Gases of which the air is composed, under the pressure of one atmosphere at 32° Fahr., are as follows:—

The weight of one cubic foot of nitrogen = ·07859 lb. or 1·258 ounces.
do. do. oxygen = ·08926 lb. or 1·428 do.
The volume of one pound weight of nitrogen = 12·727 cubic feet.
do. do. oxygen = 11·205 do.

The Weight of Atmospheric Air is ·08072 lb., or 1·2916 ounces, or 565 grains, per cubic foot, at 32° Fahr., under the pressure of one atmosphere. At 62° Fahr. the weight of one cubic foot is 32 grains less or 533 grains, or 1·2171 ounces or ·076098 lb.

The Weight of Air under the pressure of one atmosphere at 32° Fahr. is 773 times lighter than that of water of the same temperature, and at 62° Fahr. it is 820 times lighter than water of the same temperature. For ordinary calculations air is frequently taken as 770 times lighter than water.

The Weight of Air is about two-thirds more than that of gaseous steam, and about fourteen times greater than that of hydrogen.

The Weight of Air decreases with the height above the surface of the earth, its weight at a height of $3\frac{1}{2}$ miles being only one-half; at 7 miles, one-fourth; at $10\frac{1}{2}$ miles, one-eighth; and at a height of 14 miles, one-sixteenth of its weight on the surface of the earth.

The Volume of one pound of Air under the pressure of one atmosphere at 32° Fahr. is 12·386 cubic feet, and at 62° Fahr. it is 13·14 cubic feet.

Compression of Air.—The power, independent of friction, leakage, and resistance of valves, necessary to compress a given volume of air by isothermal compression is theoretically obtained by the formula*:—

$$W_t = \left[\left(\frac{1 + \log_{\epsilon} R}{R} \times P\right) - A\right] \times V_1 \quad . \quad . \quad .$$

where

W_t = the work performed by isothermal compression.

R = the ratio of compression = absolute end-pressure divided by the absolute pressure of the atmosphere.

P = the pressure in the receiver = end-pressure in compressing-cylinder.

V_1 = volume traversed by the compressing-piston.

A = the back pressure of the atmosphere = 14·7 lbs. per square inch.

Example: Required the amount of power expended in compressing one cubic foot of air of 45 lbs. pressure per square inch above the atmosphere = 59·7 lbs. total pressure?

Then

$$R = \frac{59{\cdot}7}{14{\cdot}7} = 4{\cdot}0612.$$

$\log_{\epsilon} R$, or the hyperbolic log. $R = 1{\cdot}4015$.

$P = 59{\cdot}7$ lbs. per square inch $= 59{\cdot}7 \times 144 = 8596{\cdot}8$ lbs. per square foot.

V_1 is, for isothermal compression, inversely as the pressure and
$= R \times$ the number of cubic feet of compressed air produced,
$= 4{\cdot}0612 \times 1$ cubic foot $= 4{\cdot}0612$ cubic feet.

$A = 14{\cdot}7$ lbs. pressure per square inch $= 14{\cdot}7 \times 144 = 2116{\cdot}8$ lbs. pressure per square foot.

* The Author is indebted for the above formulæ to the Articles on Compressed Air in "The Engineer," of Feb. 6 and 13, 1885.

$$W_t = \left[\left(\frac{1 + 1{\cdot}4015}{4{\cdot}0612} \times 8596{\cdot}8 \text{ lbs.}\right) - 2116{\cdot}8 \text{ lbs.}\right] \times 4{\cdot}0612 \text{ cub. ft.}$$
$$= [({\cdot}5913 \times 8596{\cdot}8) - 2116{\cdot}8] \times 4{\cdot}0612 \text{ cubic feet.}$$
$$= 2966{\cdot}7 \text{ lbs.} \times 4{\cdot}0612 \text{ cubic feet.}$$

Therefore

$W_t = 12048$ foot-pounds.

To find the amount of steam power necessary to compress a given amount of air, it is necessary to consider the losses caused by friction, leakage, and heating of the air. The theoretical power may be taken as that due to adiabatic compression, but friction and leakage as due to isothermal compression.

By adiabatic compression the mean forward pressure is found by Professor Cotterills' formula, where for adiabatic compression of air $n = \gamma = 1{\cdot}41$.

$$P_m = \frac{\gamma}{\gamma - 1} \times \frac{P_1}{r} - \frac{1}{\gamma - 1} \times P_2 \quad . \quad . \quad . \quad . \quad .$$

Where

P_m = the mean forward pressure per square inch, absolute.

P_1 = the pressure resulting from compression = 59·7 lbs. per square inch = 8596·8 lbs. per square foot.

P_2 = the pressure of the atmosphere = 14·7 lbs. per square inch = 2116·8 lbs. per square foot.

$\gamma = 1{\cdot}41$ (a number on which the velocity of sound depends, irrespective of any special theory of heat, found by experiment to have this value for air and other simple gases).

$$= \frac{\text{specific heat of air at constant pressure}}{\text{specific heat of air at constant volume}} = \frac{183{\cdot}4}{130{\cdot}2} = 1{\cdot}41.$$

r = ratio of compression, the volume of air to be compressed at 14·7 lbs. pressure and 60 deg. F. being, as in the former case, 4·0612 cubic feet: the resulting volume is found in the following way:—

Temperature due to compression = T_1

$$= \left(\frac{\text{ultimate pressure}}{\text{initial pressure}}\right)^{\cdot 29} \times \text{absolute initial temperature.}$$
$$= \left(\frac{59{\cdot}7}{14{\cdot}7}\right)^{\cdot 29} 521 \text{ deg.} = 4{\cdot}0612^{\cdot 29} \times 521.$$

Log. $T_1 = {\cdot}29 \times \log. 4{\cdot}0612 + \log. 521$ deg.;
and temperature = 782·3 deg. F.

Rise of temperature = 782·3 − 521 = 261·3 deg. F. and 261·3 + 60 = 321 deg. F.

$$\textit{Resulting volume} = \frac{14{\cdot}7 \text{ lbs.}}{59{\cdot}7 \text{ lbs.}} \times \frac{4{\cdot}0612 \text{ cub. ft. } (782{\cdot}3^\circ - 28^\circ)^*}{521^\circ - 28^\circ}$$
$$= {\cdot}246 \times 6{\cdot}213 = 1\ 53 \text{ cubic feet.}$$

$$\textit{Ratio of compression} = \frac{4{\cdot}0612}{1\ 53} = 2{\cdot}654.$$

* If the standard volume of these calculations were taken at 32 deg. F. the changes of volume would be proportional to the absolute temperature, but if taken at 60 deg. F., 60 deg. − 32 deg. = 28 deg. has to be deducted from all the absolute temperatures.

Therefore

$$P_m = \left(\frac{1{\cdot}41}{{\cdot}41} \times \frac{59{\cdot}7}{2{\cdot}654}\right) - \left(\frac{1}{{\cdot}41} \times 14{\cdot}7 \text{ lbs.}\right)$$
$$= (3{\cdot}439 \times 22{\cdot}49) - (2{\cdot}439 \times 14{\cdot}7 \text{ lbs.})$$
$$= 77{\cdot}346 - 35{\cdot}843 = 41{\cdot}498 \text{ lbs. per square inch.}$$

Mean effective pressure $= 41{\cdot}498 - 14{\cdot}700 = 26{\cdot}798$ lbs. per square inch $= 26{\cdot}798 \times 144 = 3859$ lbs. per square foot.

Total power exerted in compressing $4{\cdot}0612$ cubic feet of air adiabatically $= 4{\cdot}0612$ cubic feet $\times$ 3859 lbs. per square foot $= 15{,}672$ ft.-lbs.

Assuming friction, leakage, and resistance of valves to be about 24 per cent., in addition to the load, then each foot-pound in the compressor requires $1{\cdot}24$ foot-pounds steam-pressure, and the friction due to 12048 foot-pounds will $= 12048 \times {\cdot}24$ . . . . 2892 ft.-lbs.

Total power required for 1 cubic foot of compressed air . 18564 ft.-lbs.

The Temperature of the Atmosphere is greatest at the surface of the earth, and decreases with the height above the surface; the decrease being at the rate of 1° Fahr. for every 340 feet of vertical height.

The atmosphere receives scarcely any of its warmth directly from the sun's rays, but is heated almost entirely by the ground on which it rests, and it is therefore in the condition of the water in a boiler where the heat is applied from below.

The Temperature of the Air on the surface of the Earth varies with the height above the level of the sea, and with local circumstances. It decreases from the Equator to the Poles.

The greatest Heat in the Air seldom exceeds 150° Fahr., and the greatest cold is seldom more than 74° Fahr. below freezing point.

Air is increased in Volume by Elevation of Temperature: the volume varies as the absolute temperature when the pressure is constant. Absolute temperature is measured from absolute zero, which is 461° below zero of Fahrenheit's scale, at which point air has no elasticity, therefore it has been adopted as that of absolute zero. The absolute temperature is found by adding 461 to the temperature indicated by a Fahrenheit-thermometer.

The increased volume of a constant weight of air of which the initial volume = 1 taken at 32° Fahr., heated to a given temperature under atmospheric pressure, or $14{\cdot}7$ lbs. per square inch, may be found by this *Rule:*—

$$\textit{Increased volume of Air} = \frac{\text{Given temperature} + 461}{32 + 461}$$

If the temperature be taken at 62° instead of 32°, the divisor is $62+461 = 523$.

Example: Required the increased volume of a constant weight of air at 75° Fahr., of which the initial volume = 1 at 32° Fahr.

Then $\frac{75^\circ + 461}{32^\circ + 461} = 1{\cdot}087$, the increased volume of air by expansion.

Table No. 1 has been calculated by this rule.

The volume of a constant weight of air for any pressure, when the volume at a given pressure is known, the temperature remaining constant, may be found by this *Rule*:—

$$\textit{New volume of Air} = \frac{\text{Given volume} \times \text{given pressure}}{\text{New pressure}}$$

The product of the pressure and volume of air is proportional to the absolute temperature. The pressure of air varies directly as the absolute temperature when the volume is constant, and inversely as the volume when the temperature is constant.

The pressure of a constant weight of air for any other volume and temperature, when the pressure is known for a given volume and temperature, may be found by this *Rule*:—

New pressure of Air =

$$\frac{\text{Given pressure} \times \text{Given volume} \times \text{New absolute temperature.}}{\text{New volume} \times \text{Given absolute temperature.}}$$

Table 1.—INCREASED VOLUME OF ATMOSPHERIC AIR BY EXPANSION DUE TO ELEVATION OF TEMPERATURE.

At	32°	Fahr.	vol. equal	1·000	At	100°	Fahr.	vol. equal	1·137
,,	35°	,,	,,	1·006	,,	110°	,,	,,	1·158
,,	40°	,,	,,	1·018	,,	120°	,,	,,	1·178
,,	45°	,,	,,	1·026	,,	130°	,,	,,	1·198
,,	50°	,,	,,	1·036	,,	140°	,,	,,	1·219
,,	55°	,,	,,	1·046	,,	150°	,,	,,	1·239
,,	60°	,,	,,	1·056	,,	160°	,,	,,	1·259
,,	65°	,,	,,	1·066	,,	170°	,,	,,	1·279
,,	70°	,,	,,	1·077	,,	180°	,,	,,	1·300
,,	75°	,,	,,	1·087	,,	190°	,,	,,	1·320
,,	80°	,,	,,	1·097	,,	200°	,,	,,	1·341
,,	85°	,,	,,	1·107	,,	210°	,,	,,	1·361
,,	90°	,,	,,	1·117	,,	212°	,,	,,	1·365
,,	95°	,,	,,	1·127	,,	230°	,,	,,	1·402

The Mean Pressure of the Atmosphere at the level of the sea is equal to 14·7 lbs. per square inch, being the weight of a column of air one inch square, of the height of 27,800 feet at 32° Fahr. of uniform density equal to that of air at the level of the sea. This is called one atmosphere of pressure, and it is taken in round numbers at 15 lbs. pressure per square inch.

The Atmospheric Pressure is equal to 144 square inches × 14·7 lbs. = 2116·8 lbs. per square foot.

A Column of Mercury 29·922 inches high—or in round numbers 30 inches high—at 32° Fahr. will equal or balance the pressure of the atmosphere, at the mean level of the sea.

A Column of Water at 62° Fahr. 1 inch square and 33·947 feet high —or in round numbers 34 feet high—will equal or balance the pressure of the atmosphere.

A Column of Air $\frac{27800}{14{\cdot}7} = 1893$ feet high, at 32° Fahr. of uniform density equal to that of air at the level of the sea, will equal a pressure of 1 lb. per square inch.

A Column of Mercury $\frac{29{\cdot}922}{14{\cdot}7} = 2{\cdot}035$ inches high at 32° Fahr. will equal a pressure of 1 lb. per square inch.

A Column of Water $\frac{33{\cdot}947}{14{\cdot}7} = 2{\cdot}309$ feet high—or in round numbers 2·31 feet high; or $2{\cdot}31 \times 12 = 27{\cdot}72$ inches high at 62° Fahr. will equal a pressure of 1 lb. per square inch.

The Atmospheric Resistance to moving bodies increases as the square of the velocity of the body: and the atmospheric resistance in lbs. per square foot of frontage of a moving body increases as the square of the velocity in feet per second multiplied by ·0017. *Example:* What amount of atmospheric resistance is opposed to the front of a locomotive engine having a frontage-area of 30 square feet, at a speed of 40 miles an hour?

Then $\frac{1760 \text{ yards} \times 3 \text{ feet} \times 40 \text{ miles}}{60 \text{ minutes} \times 60 \text{ seconds}} = 58{\cdot}66$, and $58{\cdot}66^2 \times {\cdot}0017 \times 30$ feet area $= 175{\cdot}49$ lbs. atmospheric resistance opposed to the front of the locomotive engine.

The Atmospheric Resistance on Railways, independent of side winds, at ordinary speeds, is equal to about one-half the gross resistance, or from 3 lbs. to 4 lbs. per ton of the weight of the engine, tender and train. With average side winds a constant total resistance of 8 lbs. per ton of the weight of the engine, tender and train, is usually adopted in calculations of this kind.

The Weather-Barometer is an instrument for measuring the atmospheric pressure, the pressure being measured by a column of mercury which rises or falls as the weight of the atmosphere increases or diminishes. It consists of a straight glass tube, 33 inches long, closed at the top, containing mercury; the lower end dips into a cup of mercury placed at the bottom of the tube. There is a space at the top of the tube, free from air and moisture, and the mercury is raised in the tube by the atmospheric pressure on the mercury in the cup; the level of the mercury in the tube varies with the heaviness or lightness of the atmosphere. The graduations on the instrument indicate weather as follows:—

Height of the Mercury in Inches.		
28	indicates	Stormy weather.
28·5	,,	Much rain.
29	,,	Rain.
29·5	,,	Change of weather.
30	,,	Fair weather.
30·5	,,	Set fair.
31	,,	Very dry.

A Barometer has sometimes been used instead of a vacuum-gauge, but it differs in construction from the weather barometer, and consists of a glass tube bent in the form of an inverted syphon. One leg of the syphon-shaped tube is open to the atmosphere and contains mercury; the end of the other syphon-leg is connected by a pipe to the condenser, and the mercury rises in this leg according to the amount of vacuum and falls according to the amount of vapour. When not in operation the mercury will stand at the same level in both legs of the syphon. If there were a perfect vacuum in the condenser, as each pound of vacuum represents 2 inches of mercury, the pressure of the atmosphere in one leg would cause the mercury to rise in the other leg of the syphon connected to the condenser to a height of 30 inches, when it would balance the atmospheric pressure and indicate 15 lbs. of vacuum; if the mercury rose to 26 inches it would be 13 lbs of vacuum, and so on, the indicated pressures being those below atmospheric pressure.

The Ordinary Weather-Barometer can be used for ascertaining the heights of mountains. The mercury falls on being taken to a height above the ground at the rate, approximately, of one-tenth of an inch for every hundred feet of vertical height, because the atmospheric pressure decreases with the height above the surface of the ground.

The Aneroid-Barometer is the best instrument for ascertaining heights, because it is extremely sensitive and contains no liquid. It consists of a round metal box, exhausted of air, in the top of which a very thin and flexible sheet-metal plate is placed, which yields to the pressure of the atmosphere, and actuates a system of multiplying levers and a spring connected to an index, which moves on a scale.

The Pressure of the Atmosphere in lbs. per square inch corresponding to the Height of a Barometer may be found by multiplying the weight of a cubic inch of mercury, $= \cdot 4908$ lb., by the height of the barometer in inches. The following Table has been calculated by this rule.

Table 2.—PRESSURE OF ATMOSPHERIC AIR AT DIFFERENT HEIGHTS OF THE BAROMETER.

Height of the Barometer.	Pressure of the Air in lbs. per Square Inch.	Pressure of the Air in lbs. per Square Foot.	Height of the Barometer.	Pressure of the Air in lbs. per Square Inch.	Pressure of the Air in lbs. per Square Foot.
Inches.			Inches.		
27	13·25	1908·23	29·25	14·35	2067·24
27·25	13·37	1925·89	29·50	14·47	2084·92
27 50	13·49	1943·56	29·75	14·60	2102·58
27·75	13·61	1961·23	30	14·70	2116·80
28	13·74	1978·90	30·25	14·84	2137·92
28·25	13·86	1996·56	30·50	14·96	2155·59
28·50	13·98	2014·24	30·75	15·09	2173·26
28·75	14·11	2031·91	31	15·22	2190·94
29	14·23	2049·58	31·5	15·46	2226·27

PRESSURE AND VELOCITY OF WIND.

Wind is air in motion, due to the disturbance of the equilibrium in some part of the atmosphere, caused by a difference in temperature of adjacent countries; the air in one part having become heated expands, and being lighter, rises, and the motion of the cooler air in rushing in to supply its place produces a current, the velocity of which depends upon the difference between the temperatures.

The Velocity of the Wind increases with altitude. The velocity at different altitudes may be calculated by the following *Rule*:— $\frac{V}{v} = \sqrt[4]{\frac{H}{h}}$ in which V, v, H, h, are the velocities and heights at the lower and upper levels respectively.

The Force of Wind increases as the square of its velocity. The ratio of the different forces exerted by two winds of different velocity is found approximately by squaring the ratio of their velocities. Thus, a wind blowing at the rate of 60 miles an hour has a velocity three times greater than that of another wind moving at the rate of 20 miles an hour, and the former will exert a force approximately equal to $3 \times 3 = 9$ times greater than that of the latter.

Pressure and Velocity of Wind.—The force or pressure of wind may be found by the following *Rule*, deduced from the results of recent experiments.

Pressure of Wind in lbs. per square foot$=$(velocity in feet per second)^2 $\times \cdot 0017$.

Example: Required the pressure of the wind in lbs. per square foot when its velocity is 42 feet per second.

Then $42 \times 42 \times \cdot 0017 = 3$ lbs. pressure per square foot.

The Velocity of the Wind in feet per second may be found by the following *Rule*:—

$$\text{Velocity of wind in feet per second} = \sqrt[2]{\frac{\text{Pressure in lbs. per square foot.}}{\cdot 0017}}$$

Example: Required the velocity of the wind in feet per second when its pressure is 40 lbs. per square foot, then $\sqrt[2]{\frac{40}{\cdot 0017}} = 154$, the wind's velocity in feet per second, and its velocity in miles per hour $=$ $\frac{154 \text{ feet per second} \times 60 \text{ seconds} \times 60 \text{ minutes}}{1760 \text{ yards} \times 3 \text{ feet}} = 105$ miles per hour, the wind's velocity.

A column of water 1 inch high exerts a pressure on the base of $5 \cdot 196$ lbs. per square foot, therefore a pressure of 1 lb. per square foot would equal $\frac{1 \text{ lb.}}{5 \cdot 196 \text{ lbs.}} = \cdot 192$ inch of water pressure.

Table 3 has been calculated by these rules. It will be found useful in making calculations of the pressure of wind on structures; the power of fans and blowers, and of natural draught and forced draught in chimneys.

Table 3.—PRESSURE AND VELOCITY OF WIND.

Wind-Pressure.		Wind's Velocity.		
Pressure in lbs. per Square Foot.	Pressure in Inches of Water.	Velocity in Feet per Second.	Velocity in Miles per Hour.	Description of Wind.
lbs.	Inches.			
$\frac{1}{4}$	·048	12·13	8·27	Gentle wind.
$\frac{1}{2}$	·096	17·15	11·69	Pleasant wind.
$\frac{3}{4}$	·144	21	14·31	Fresh breeze.
1	·192	24·25	16·53	Strong breeze.
$1\frac{1}{2}$	·288	29·71	20·02	
$1\frac{3}{4}$	·311	32·08	21·87	Gale.
2	·384	34·30	23·38	
$2\frac{1}{2}$	·480	38·34	26·14	Brisk gale.
3	·577	42·00	28·63	
$3\frac{1}{2}$	·672	45·37	30·93	Strong wind.
4	·769	49	33·60	
5	·962	54	36·81	Moderately high wind.
6	1·154	59	40·22	High wind.
7	1·347	64	43·63	
8	1·539	69	47·04	
9	1·731	73	49·81	
10	1·924	77	52·50	Very high wind.
11	2·116	80	54·55	
12	2·309	84	57·27	
13	2·501	88	60·00	Storm.
14	2·693	91	62·04	
15	2·886	94	64·09	
16	3·078	97	66·13	Great storm.
17	3·271	100	68·18	
18	3·463	103	70·22	
19	3·655	106	72·27	
20	3·850	109	75·26	Very great storm.
25	4·810	122	83·18	Tempest.
30	5·772	133	90·68	Hurricane.
35	6·734	144	98·56	
40	7·696	154	105·00	Great hurricane.
45	8·658	163	111·21	
50	9·620	172	117·27	Violent hurricane.
55	10·582	180	122·72	
60	11·544	188	128·03	Very violent hurricane.
65	12·506	196	133·63	
70	13·468	203	138·41	
75	14·430	210	143·18	
80	15·392	217	147·95	Tornado.
85	16·354	224	152·72	
90	17·316	230	156·81	
95	18·278	237	161·59	
100	19·240	245	167·04	

WIND-MOTORS.

Windmills are efficient and economical motors for intermittent work, or where the nature of the work admits of its being suspended during a calm, which seldom lasts more than two days at a time. Taking the average of the running of windmills for a year, it has been found that they may be depended upon working constantly for at least one-third of the time, or 8 hours out of 24. The average velocity of the wind being 15 miles an hour.

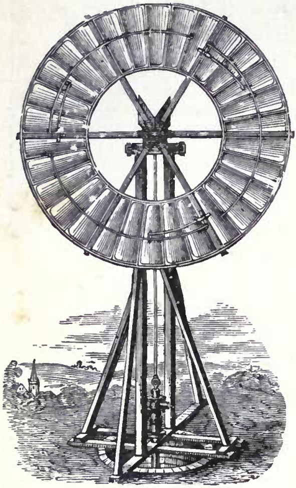


Fig. 1.—Windmill.

Windmills, shown in Fig. 1, may be applied to working pumps, for supplying water for domestic use, irrigation, drainage, compression of air, or driving dynamo machines. They are fitted with self-adjusting sails and self-winding tackle to keep the mill always in wind, and require no attention except filling the lubricators, which are arranged to hold a supply of lubricant to last one or two months.

The Length of the Whip, or radial arms of the sails of a windmill, depends upon the power, situation, and velocity required; the sails are frequently made rectangular, and the length is equal to five times the breadth.

The Whip is divided into seven equal parts, six of those parts, from the extremity, being the length of the sails.

The weather-board, or wind-board, is equal to one-fifth of the sail's breadth.

The Shaft on which the sails are fixed may have an angle with the horizon between 10° and 15°.

The Total amount of Sail-Surface should not exceed one-fourth of the whole disc-surface described by the whip or radial arms of the sail.

In order to gain the greatest amount of the wind's impulsive effect to produce circular motion by the sails of a windmill, the total surface of sails presented to the wind should be about seven-eighths of the circle's surface which is formed by their motion, and each sail should be angled to the plane of motion as follows, the whip or back being divided into six equal parts :—

Distance from the centre of motion .	1	2	3	4	5	6
Angle with the plane of motion . . .	24	21	18	14	9	3

The Horse-power of a Windmill, H. P., may be found by this *Rule* :—

$$\text{H. P.} = \frac{\begin{pmatrix}\text{Total area of sails}\\ \text{in square feet}\end{pmatrix} \times \begin{pmatrix}\text{Velocity of the wind}\\ \text{in feet per second}\end{pmatrix}^3}{1{,}100{,}000}$$

The total area, A, in square feet of the sails of a windmill may be found by this *Rule* :—

$$A = \frac{\text{Horse-power} \times 1,100,000}{(\text{wind-velocity in feet per second})^3}.$$

Table 4.—Efficiency of Windmills turning 8 Hours a Day at a Speed of 15 Miles an Hour when used for working Pumps.

Diameter of Windmill.	Number of Revolutions per Minute.	Actual Horse-power developed.	Quantity of Water raised daily to a Height of 25 Feet.	Quantity of Water raised daily to a Height of 50 Feet.	Quantity of Water raised daily to a Height of 100 Feet.
Feet			Gallons.	Gallons.	Gallons.
12	55	$\frac{1}{4}$	13500	6750	3375
15	50	$\frac{1}{3}$	20000	10000	5000
18	45	$\frac{1}{2}$	40000	20000	10000
20	40	$\frac{3}{4}$	50000	25000	12500
22	36	1	60000	30000	15000
24	32	$1\frac{1}{4}$	80000	40000	20000
30	28	$1\frac{1}{2}$	110000	55000	27500
36	24	$1\frac{3}{4}$	160000	80000	40000
40	22	2	200000	100000	50000

The Pantanemone or Universal-Windmill, shown at Fig. 2, is a recent continental invention in wind-motors. Two plane surfaces in the form of semi-circles are mounted at right angles to each other upon a horizontal shaft, and at an angle of 45° with respect to the latter.

These motors it is said work satisfactorily whatever may be the direction of the wind. One of them has been working in the vicinity of Poissy for several years, where it lifts about 40,000 litres of water to a height of 20 metres every 24 hours, in a wind of a velocity of from 7 to 8 metres per second. Another one raises about 150,000 litres of water to the Villejuif Reservoir at a height of 10 metres every 24 hours, in a wind of a velocity of 5 to 6 metres per second.

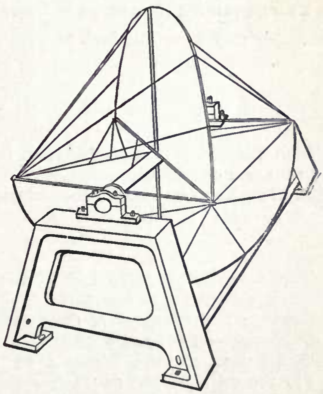


Fig. 2.—Pantanemone.

The Horse-power, H. P., of the Pantanemone may be found by this *Rule* :—

$$\text{H. P.} = \frac{\begin{pmatrix}\text{Total area of sails}\\ \text{in square feet}\end{pmatrix} \times \begin{pmatrix}\text{Velocity of the wind}\\ \text{in feet per second}\end{pmatrix}^3}{1,200,000}$$

The total Area, A, in square feet, of the sails of the Pantanemone, may be found by this *Rule*:—

$$A = \frac{\text{Horse-power} \times 1{,}200{,}000}{(\text{wind-velocity in feet per second})^3}.$$

WATER AND WATER-MOTORS.

Water is composed of two gases, hydrogen and oxygen, in the proportion of one part of hydrogen and eight parts of oxygen. Water dissolves more substances than any other agent: it freezes at a temperature of 32° Fahr.

The Weight of a cubic foot of Ice is from 57·5 to 58 lbs., its specific gravity being about ·922. Water in freezing expands about $\frac{1}{12}$th of its original volume as water, its expansive force at the moment of freezing being estimated at about 32,000 lbs. per square inch. A cubic foot of ice floating in water has about $\frac{11}{12}$ths of its volume under water, and $\frac{1}{12}$th of its volume above water, and a square foot of ice of any thickness requires a weight equal to $\frac{1}{11}$th of its weight to sink it to the surface of the water. The compressive strength of hard pure block-ice is about 20 tons per square foot. The specific heat of ice is ·504.

Snow weighs 6 lbs. per cubic foot when freshly fallen, and averages 12 lbs. per cubic foot when moderately saturated with rain. Snow has 12 times the bulk of water, and its specific gravity is ·084.

The Standard Measures of water are as follows:—

The weight of one cubic inch of pure water at	62° Fahr. =	252·595 grains.
,, ,, ,,	32° ,, =	·03612 lb.
,, ,, ,,	39·1° ,, =	·036125 lb.
,, ,, ,,	62° ,, =	·03608 lb.
,, ,, ,,	212° ,, =	·03451 lb.

The Weight of one cylindrical inch of pure Water at 62° Fahr. = ·4533 ounce or ·02883 lb.

The Weight of one cubic foot of pure Water at	32° Fahr.	= 62·418 lbs.
,, ,, ,, ,,	39·1° ,,	= 62·425 lbs.
,, ,, ,, ,,	62° ,,	= 62·355 lbs.
,, ,, ,, ,,	212° ,,	= 59·640 lbs.

The weight of a cubic foot of water is usually taken at 62·4 lbs. per cubic foot for ordinary calculations.

A cubic foot of water at the temperature of maximum density weighs 998·8 ounces—it is usually taken at 1000 ounces.

One Gallon of pure Water at 62° Fahr. weighs 10 lbs., and its volume is 277·123 cubic inches, or ·160372 cubic foot.

The Volume of 1 lb. of pure Water at	32° Fahr.	= 27·684 cubic inches.
,, ,, ,,	39·1° ,,	= 27·680 ,,
,, ,, ,,	62° ,,	= 27·712 ,,
,, ,, ,,	212° ,,	= 28·978 ,,

The volume of one ton of pure water = 35·9 cubic feet.

The volume of pure water at 62° in cubic inches × ·0036 = capacity in gallons.

The pressure of a column of water 1 foot high $= \frac{1·000}{2·309} =$ ·433 lb., or 6·928 ounces, per square inch pressure.

The pressure of a column of water 33·947 feet, or say 34 feet high = the pressure of the atmosphere, or 14·7 lbs.

The pressure of a column of water $\frac{33·947 \text{ feet high}}{14·7 \text{ lbs.}} =$ 2·309 feet high, or say 2·31 feet high = 1 lb. per square inch pressure.

Expansion of Water by Heat.—Water expands as the temperature rises above the temperature of maximum density, 39·1° Fahr., it also expands in nearly an equal ratio, as the temperature falls below this point, down to the freezing point.

Table 5.—TEMPERATURE AND VOLUME OF WATER.

	Temperature.			
At	40°	Fahrenheit,	its volume	1·00000
„	55°	„	„	1·00054
„	65°	„	„	1·00137
„	75°	„	„	1·00255
„	85°	„	„	1·00404
„	95°	„	„	1·00583
„	105°	„	„	1·00800
„	115°	„	„	1·01121
„	125°	„	„	1·01275
„	135°	„	„	1·01551
„	145°	„	„	1·01847
„	155°	„	„	1·02162
„	165°	„	„	1·02500
„	175°	„	„	1·02845
„	185°	„	„	1·03209
„	195°	„	„	1·03590
„	205°	„	„	1·03984
„	212°	„	„	1·04600

The Evaporation from the Surface of Water in Lakes, Canals and Rivers in this country probably averages a total of 8 inches in spring, 12 inches in summer, 7 inches in autumn, and 4 inches in winter: equal to 31 inches per annum. The annual evaporation in some districts has been found to average 75 per cent. of the annual rainfall, and in others it has been found to exceed the annual rainfall.

Weight of Sea-water.—The weight of one cubic foot of sea-water at 62° = 64 lbs.: 35 cubic feet of sea-water = 1 ton.

The weight of a cubic yard of sea-water = 15 cwt. 1 qr. 20 lbs., or nearly $15\frac{1}{2}$ cwt.

The weight of fresh water compared with that of sea-water is as 1 to 1·026, or as 39 to 40.

The Mean Specific Gravity of Sea-water is 1·026 : in the Black Sea it is 1·016 : in the Indian Ocean, 1·0264 : in the North Atlantic Ocean, 1·0267 : in the South Atlantic Ocean, 1·0268 : in the Red Sea, 1·0286 : and in the Mediterranean Sea 1·029.

The Composition of Average Sea-water is as follows:—

Water	96·6	parts.
Chloride of Sodium	2·6	,,
Chloride of Magnesia	·4	,,
Sulphate of Soda	·37	,,
Carbonate of Lime	·02	,,
Sulphate of Lime	·01	,,
	100·00	

Table 6.—COMPONENTS OF THE SALTS OF SEA-WATER.

Ingredients.	PER 100 PARTS OF TOTAL SALTS.	PER 100 PARTS OF HALOGEN CALCULATED AS CHLORINE.	
	Dittmar.	Dittmar.	Forchammer.
Chlorine	55·292	99·848	Not determined.
Bromine	·1884	·3402	,, ,,
Sulphuric Acid ($S\ O_3$)	6·410	11·576	11·88
Carbonic Acid ($C\ O_2$)	·152	·2742	Not determined.
Lime (Ca O)	1·676	3·026	2·93
Magnesia (Mg O)	6·209	11·212	11·03
Potash ($K_2\ O$)	1·332	2·505	1·93
Soda ($Na_2\ O$)	41·234	74·464	Not determined.
Basic equivalents to the Halogens	12·493		
Total Salts	100	180·584	181·1

The mean quantity of solid substances—chiefly salt—held in solution by sea-water is 3·4 per cent., three-fourths of which is common salt.

The Quantity of Salt in Sea-water varies in different seas. The waters of the White Sea, the Baltic Sea, and the Polar Seas contain very little salt, but the Red Sea contains a large quantity of salt. The Red Sea contains 4·32 per cent. of salt: the Baltic Sea, 5 per cent.: and the sea at Cronstadt contains 2 per cent. of salt.

The Ice of Sea-water contains no salt, because water in freezing parts with all its impurities.

Ordinary Sea-water contains $\frac{1}{88}$ part of its weight of salt, called 1 degree of saltness.

The weight of Salt in a Gallon of Sea-water may be ascertained as follows: If thirty-three pounds weight of sea-water be evaporated, it will leave one lb. of salt or $\frac{1}{33}$rd of its weight. One gallon of sea-water weighs $10\frac{1}{4}$ lbs., $\frac{1}{33}$rd part of which is salt, therefore the quantity of salt contained in one gallon is $\frac{10{\cdot}25 \text{ lbs.} \times 16 \text{ ounces}}{33} = 4{\cdot}97$ ounces, or in round numbers 5 ounces of salt per gallon of sea-water.

Sea-water Boils at $213^\circ{\cdot}2$ Fahr. under the pressure of one atmosphere, or when the mercury in the weather-barometer stands at 30 inches. The boiling point of sea-water varies with the quantity of salt held in solution, and rises in proportion to its concentration as brine; it also varies with the rise and fall of the weather-barometer.

Saturated Brine boils under the pressure of one atmosphere at $226^\circ{\cdot}4$ Fahr.

The point of Saturation of ordinary Sea-water is $\frac{12}{32}$ of its weight, when the water is so full of salt that it will hold no more, and it is therefore rapidly precipitated.

The Boiling Point of Sea-water may be calculated from its density as follows: It is found that $\frac{1}{33}$rd part of salt increases the boiling point to the extent of $1^\circ{\cdot}2$ Fahr.; the boiling point of fresh water being 212° Fahr., that of ordinary sea-water will equal $212^\circ + 1^\circ{\cdot}2 = 213^\circ{\cdot}2$ Fahr. at atmospheric pressure, or when the weather-barometer stands at 30 inches. The following Table has been calculated in this way, each degree of salt representing 5 ounces of salt per gallon of sea-water.

Table 7.—Boiling Points of Sea-water of various Densities under one atmosphere of Pressure.

Part of Salt.	Degrees of Salt.	Ounces of Salt per Gallon.	Boiling Point in Degrees Fahr.
$\frac{1}{33}$	or 1 degree of Salt or	5	$213^\circ{\cdot}2$
$\frac{1\frac{1}{2}}{33}$	„ $1\frac{1}{2}$ „ „	$7\frac{1}{2}$	$213^\circ{\cdot}9$
$\frac{2}{33}$	„ 2 „ „	10	$214^\circ{\cdot}4$
$\frac{2\frac{1}{2}}{33}$	„ $2\frac{1}{2}$ „ „	$12\frac{1}{2}$	$215^\circ{\cdot}1$
$\frac{3}{33}$	„ 3 „ „	15	$215^\circ{\cdot}6$
$\frac{3\frac{1}{2}}{33}$	„ $3\frac{1}{2}$ „ „	$17\frac{1}{2}$	$216^\circ{\cdot}3$
$\frac{4}{33}$	„ 4 „ „	20	$216^\circ{\cdot}8$
$\frac{4\frac{1}{2}}{33}$	„ $4\frac{1}{2}$ „ „	$22\frac{1}{2}$	$217^\circ{\cdot}5$
$\frac{5}{33}$	„ 5 „ „	25	218°
$\frac{6}{33}$	„ 6 „ „	30	$219^\circ{\cdot}2$
$\frac{7}{33}$	„ 7 „ „	35	$220^\circ{\cdot}4$
$\frac{8}{33}$	„ 8 „ „	40	$221^\circ{\cdot}6$
$\frac{9}{33}$	„ 9 „ „	45	$222^\circ{\cdot}8$
$\frac{10}{33}$	„ 10 „ „	50	224°
$\frac{11}{33}$	„ 11 „ „	55	$225^\circ{\cdot}2$
$\frac{12}{33}$	„ 12 Saturated brine	60	$226^\circ{\cdot}4$

The above Table gives the boiling points when the mercury in the weather barometer stands at 30 inches or atmospheric pressure, and in using the Table an allowance must be made when the barometer stands above or below that point. As 1 degree of salt increases the boiling point to the extent of 1·2, it will be sufficiently near in most cases to deduct two-thirds of 1·2 or ·8 from the boiling point for every $\frac{1}{2}$ inch the barometer registers below 30 inches, or to add the same for every $\frac{1}{2}$ inch the mercury stands above 30 inches.

Example: The boiling point given in the Table is 215·1 for $2\frac{1}{2}$ degrees of salt when the barometer stands at 30 inches, required the boiling point when the barometer stands at $30\frac{1}{2}$ inches, or $\frac{1}{2}$ inch higher. Then $215{\cdot}1 + {\cdot}8 = 215{\cdot}9$, the boiling point of water containing $2\frac{1}{2}$ degrees of salt when the mercury in the barometer stands at $30\frac{1}{2}$ inches.

DENSITY OF SEA-WATER. SALINOMETERS. BLOWING-OFF.

The Working Density or Saltness of Water, when sea-water is used in marine boilers, is from $1\frac{1}{2}$ to 2 degrees of saltness, or from $\frac{1\frac{1}{2}}{33}$ to $\frac{2}{33}$rds its weight of salt: the maximum density seldom exceeds $\frac{3}{33}$rds.

A Boiler is said to be Salted when there is an accumulation of salt on the tubes and heating surface. Four degrees of saltness, or $\frac{4}{33}$rds, deposits salt rapidly. Salting is prevented by scumming and by frequently and regularly blowing off a portion of the boiler-water.

Scum-Cocks.—The salt and dirt floating on the surface of the water in a marine-boiler can be blown into the sea by means of the scum-cocks. These cocks are fixed on the shell of the boiler, and are connected to a cock fixed on the ship's side by a pipe. From each scum-cock a pipe is carried inside the boiler, having a dished end placed a little below the working level of the boiler, which collects the refuse from the surface of the water. The scum-cocks are used whenever the surface of the water is considered to be dirty, or when the limit of density is reached. They must be shut before the water level has fallen too low. Neglect of the cocks leads to their sticking fast, and renders the boiler liable to become salted and the tubes to become burnt.

A Hydrometer or Salinometer is an instrument for measuring the density or degree of saturation of the water when sea-water is used in a marine-boiler. It consists of a bulb of glass or metal having a graduated stem at the top, and a stem at the bottom filled with mercury to make the instrument swim upright. It acts by sinking into the water more or less according to the degree of saltness of the water, the salter the water the less will it sink. The salinometer is graduated into 33rds, representing 5 ounces of salt per gallon of water. To graduate the stem, the zero point is marked at which the instrument floats in fresh water, it is also marked at the level at which it floats in sea-water of the average degree of saltness. The space between these two fixed points is divided into ten parts or degrees of the salinometer, and the graduations are extended to 35°. Every 10° represents $\frac{1}{33}$rd of saltness; 10° represents the density of sea-water: $15^\circ = 1\frac{1}{2}$ the density: 20° = twice the density: $25^\circ = 2\frac{1}{2}$ times the density, and 30° = three times the density of sea-water.

A Hydrometer or Salinometer cannot be used at any temperature indiscriminately, because it is graduated for a fixed temperature, and it should only be used in water of the temperature for which it was marked; therefore it is necessary to test the water with a thermometer before using the instrument, when accuracy is required. Hydrometers or Salinometers are usually marked to suit a temperature of 200° Fahr., this being about the temperature of the boiler-water immediately after being drawn off for testing; therefore it is only necessary to use a thermometer with the instrument when great accuracy is required. To test the boiler-water in the absence of a salinometer, draw off and boil a small quantity of the water, and test it with a thermometer to ascertain its boiling point in the open air, from which the degree of saltness corresponding to its boiling point may be ascertained from Table 7.

Blowing-off is practised in marine boilers which are fed with sea-water, to prevent the degree of saltness of the water exceeding a particular density, the supersalted water being got rid of by blowing off a portion of the boiler-water into the sea, which is replaced with sea-water of ordinary density.

The Quantity of Feed-Water required when blowing-off is practised, is equal to the sum of the quantity of water evaporated to steam and the quantity blown off.

The Quantity to be Blown Off to maintain a constant density, may be found by dividing the number of cubic feet of feed-water by the number of degrees of saltness.

Example: A marine boiler is to be kept at two degrees of saltness. How much water must be blown off? Then if 1 = the quantity of feed-water, $1 \div 2 = \cdot 5$, or one-half the quantity of feed-water must be blown off to prevent the degree of saltness rising above $\frac{2}{32}$, or 10 ounces per gallon.

The Quantity to be Blown Off may be calculated from the water evaporated, by this *Rule:* Subtract 1 from the number of degrees of saltness, and by the remainder divide the quantity of water evaporated.

Example: If 1200 gallons of water be evaporated to steam, what quantity of brine must be blown off, that the water in the boiler may be maintained at $\frac{3}{33}$, or three degrees of saltness. Then $\frac{1200}{3-1} = 600$ gallons must be blown off.

The Quantity of Water Evaporated may be calculated from the quantity blown off, by this *Rule:* Subtract 1 from the number of degrees of saltness, and multiply the remainder by the quantity of water blown off. *Example:* If 600 gallons be blown off, what number of gallons have been evaporated to steam, the water in the boiler being maintained at $\frac{3}{33}$, or three degrees of saltness?

Then $3-1=2 \times 600 = 1200$ gallons, the quantity of water evaporated.

The Quantity of Water Blown Off may be calculated from the weight of salt per gallon, by this *Rule:* Divide the number of ounces of salt per gallon of feed-water by the number of ounces per gallon of boiler-water, and the quotient will give the proportion of feed-water to be blown off. *Example* 1: The feed-water contains 4·2 ounces of salt per gallon, and the water in the boiler contains 12 ounces of salt per gallon. What proportion of the quantity of the feed-water should be continuously blown off?

Then $\frac{4\cdot2 \text{ ounces of salt per gallon of feed-water}}{12 \text{ ounces of salt per gallon of boiler-water}} = \cdot35$, or a little more than one-third of the feed-water should be blown off.

Example 2: If the feed-water from a surface condenser has only a trace of salt, say ·04 ounce per gallon, and it is required to work the boiler at ·8 ounce of salt per gallon, what percentage of the feed-water must be blown off after the water in the boiler has reached this degree of saltness?

Then $\frac{100 \times \cdot04 \text{ ounce of salt per gallon of feed-water}}{\cdot8 \text{ ounce of salt per gallon of boiler-water}} = 5$ per cent.

The Saltness of the Water in the boiler may be calculated from the weight of salt per gallon of feed-water, by this *Rule:* Divide the number of ounces of salt per gallon of feed-water by the fraction of the feed continuously blown off, the quotient will be the number of ounces of salt per gallon of the water in the boiler.

Example: The feed-water contains 4·2 ounces of salt per gallon, and ·35 of the feed is continuously blown off. What quantity of salt does the water in the boiler contain?

Then $\frac{4\cdot2 \text{ ounces of salt per gallon of feed-water}}{\cdot35 \text{ proportion of feed-water blown off}} = 12$ ounces of salt per gallon contained in the water of the boiler.

The Saltness of the Feed-water may be calculated from the quantity of salt in the boiler-water, and the quantity of feed-water blown off, by this *Rule:* Multiply the quantity of salt in ounces per gallon of water in the boiler, by the proportion of feed-water blown off. The product will be the quantity of salt in ounces per gallon of feed water. *Example:* If the water in the boiler contains 9 ounces of salt per gallon, and one-eighth of the feed-water is continuously blown off, what quantity of salt is contained in the feed-water?

Then 9 ounces per gallon of boiler-water × ·125 blow-off = 1·125 ounces of salt per gallon, is contained in the feed-water.

It may also be ascertained as follows:—

$$1 : 9 :: \cdot125 = 1\cdot125 \text{ ounces per gallon of feed-water.}$$

The pressure of Steam required to expel the Brine at a depth below the surface of the sea, may be ascertained as follows:—A pressure of 1 lb. per square inch is equal to a column of sea-water =

$\frac{2\cdot31 \text{ height of a column of fresh water}}{1\cdot026 \text{ specific gravity of sea-water}} = 2\cdot251$ feet high at 62° Fahr., hence the *Rule:*—Divide the depth in feet of the level of the water in the boiler, below the surface of the sea, by 2·251, the quotient will be the force or pressure in pounds per square inch required to expel the brine. It may also be ascertained as follows: the pressure of the atmosphere = 14·7 lbs. being balanced by a column of sea-water =

$$\frac{34 \text{ feet height of a column of fresh water}}{1\cdot026 \text{ specific gravity of sea-water}} = 33\cdot138 \text{ feet high.}$$

As 33·128 feet : depth in feet below sea-surface :: 14·7 lbs. = the pressure of the water in lbs. per square inch; and pressure of water + 14·7 = the total pressure at the given depth.

Example 1: What pressure of steam is required to expel the brine, the level of the water in a marine boiler being 12 feet below the surface of the sea?

Then 12 ÷ 2·251 = 5·33 lbs. pressure, or as 33·138 : 12 : : 14·7 = 5·32 lbs. pressure.

Example 2: What pressure of steam is required to expel the brine from a marine boiler having its water-level 11 feet below the surface of the sea, the pressure of steam being 25 lbs. per square inch?

Then $\frac{11}{2{\cdot}251}$ = 4·88 lbs. pressure of water, and 4·88 + 14·7 = 19·58 lbs. total pressure at 11 feet, and the force available to expel the brine is 25−19·58=5·42 lbs. per square inch.

The Loss of Fuel and Heat by Blowing-off may be found as follows:—

Let N = the number of times the density of the boiler-water is greater than that of the feed-water.

„ T = the temperature of the water in the boiler in degrees Fahr.

„ t = the temperature of the feed-water in degrees Fahr.

Then, the loss per cent. of the fuel used=

$$\frac{T-t}{(N-1)\,(1115-{\cdot}3\times\overline{T-t})+(T-t)}.$$

The loss per cent. of the total heat in the boiler=

$$\frac{100\times\overline{T-t}}{(N-1)\,(1115-{\cdot}3\times\overline{T-t})+(T-t)}.$$

Example: The density of the water in a marine boiler is 1·8 times greater than that of the feed-water, the temperature of the boiler-water is 245° Fahr., and that of the feed-water 45° Fahr. Required the loss per cent. of the fuel used by blowing off, and also the loss per cent. of the total heat in the boiler?

Then $\frac{245-45}{(1{\cdot}8-1)\quad(1115^\circ+{\cdot}3\times\overline{245-45})+(245-45)}$ = 1·8—1=·8, and 245 × ·3 = 73·5 + 1115 = 1188·5 — 45 = 1143·5 × ·8 = 914·8 + (245—45) = 1114·8 and 245—45=200, then $\frac{200}{1114{\cdot}8}$ = ·178, the loss per cent. of the fuel used.

The loss per cent. of the total heat in the boiler will be

$\frac{100\times\overline{245-45}}{(1{\cdot}8-1)\,(1115^\circ+{\cdot}3\times\overline{245-45})+\overline{245-45}}$ = 1·78 per cent. of heat lost by blowing off.

The Loss of Fuel and Heat by Blowing-off may also be found as follows:—

Let B = the number of cubic feet of water blown off every 3 hours.

„ E = the number of cubic feet of water evaporated every 3 hours.

„ T° = the temperature of the water in the boiler in degrees Fahr.

„ t° = the temperature of the feed-water in degrees Fahr.

The number of cubic feet of water entering the boiler every 3 hours will equal B + E.

The total units of heat in steam of 212° Fahr. $= 1115° + \overline{\cdot 3 \times 212} =$ $1115° + 63\cdot6 = 1178\cdot6$ units total heat. To evaporate E cubic feet of water to steam will require $(1178\cdot6-t°)$ E units of heat.

To boil the B feet of water blown off will require $(T°-t°)$ B units of heat.

The total loss will be $(T°-t°)$ B.

The total quantity of heat used will be $(1178\cdot6-t°)$ E + $(T°-t°)$ B, as out of $(1178\cdot6-t°)$ E+$(T°-t°)$ B there is lost $(T°-t°)$ B.

The loss of heat by blowing-off$=\dfrac{(T°-t°)\,B}{(1178\cdot6-t°)\,E+(T°-t°)\,B}$

The loss per cent. of the total heat in the boiler$=\dfrac{100\,(T°-t°)\,B}{(1178\cdot6-t°)\,E+(T°-t°)\,B}$

Example: A marine boiler is blown out every hour: 140 gallons being expelled each time, and 420 gallons are evaporated in the same time. The temperature of the water in the boiler is 242°·8 Fahr., and that of the feed-water is 42°·8 Fahr. Required the loss per cent. of the total heat in the boiler?

Then 140 gallons blown off per hour equal $140 \times 3 = 420$ gallons, or $420 \div 6\cdot25 = 67\cdot2$ cubic feet of water blown off every 3 hours. 420 gallons of water evaporated per hour $= 420 \times 3 = 1260$ gallons, or $1260 \div 6\cdot25$ $= 201\cdot6$ cubic feet of water evaporated every 3 hours.

Then $\dfrac{100 \times (242°\cdot8-42°\cdot8) \times 67\cdot2}{(1178°\cdot6-42°\cdot8) \times 201\cdot6+(242°\cdot8-42°\cdot8) \times 67\cdot2} = 1178\cdot6-42\cdot8=$ $1135\cdot8 \times 201\cdot6 = 228{,}977\cdot28 + (242\cdot8-42\cdot8) \times 67\cdot2 = 242{,}417.28$ units, the loss of heat by blowing-off: and $100 \times (242\cdot8-42\cdot8) \times 67\cdot2 = 100 \times 200 \times$ $67\cdot2 = 1{,}344{,}000$, then $\dfrac{1{,}344{,}000}{242{,}417\cdot28} = 5\cdot54$ per cent. of the total heat in the boiler lost by blowing off.

When the temperatures are expressed in degrees Centigrade, the same rule can be used as the last, by changing the constant 1178·6 Fahr. in the last rule, to 637·2 C., that being the number of units of heat in 1 lb. of steam at 100° C., or 212° Fahr.

The Density of the Water in the Boiler may be calculated from the quantity blown off by the following *Rule:* where L=loss per cent. of the fuel used.

The density of the boiler-water above that of the feed=

$$\frac{T°-t°}{L\,(1115+\cdot3\,T-t)}+1.$$

Example: If the loss by blowing off is equal to $\frac{1}{8}$th—or ·125—of the fuel used, the temperature of the steam being 248° Fahr., and that of the feed-water 108° Fahr., at what density should the boiler-water be maintained above that of the feed-water?

$$\frac{248-108}{\cdot125\,(1115+\cdot3 \times T-t)}+1.$$

Then $248 \times \cdot 3 = 74 \cdot 4$, and $1115 + 74 \cdot 4 = 1189 \cdot 4 - 108 = 1081 \cdot 4 \times \cdot 125 = 135 \cdot 175$; and $248^\circ - 108^\circ = 140$, then $\frac{140}{135 \cdot 175} = 1 \cdot 4 + 1 = 2 \cdot 4$, the amount the density of the boiler-water should exceed that of the feed-water.

MEASUREMENT OF FLOWING WATER.

The Height of the Fall should be measured from the level of the water in the head-race to the level of the water in the tail-race.

Measurement of Flowing Water by a Notched-Board.—The quantity of water in a stream, available for driving a turbine, may be measured by means of a notched board as shown in Fig. 3, with which proceed as follows :—

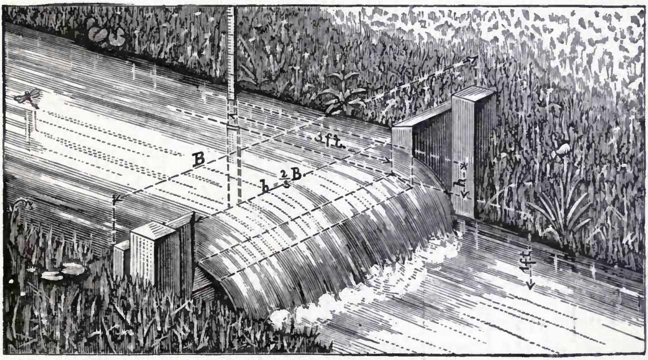


Fig. 3.—Notched-board for measuring flowing water.

Place a board across the stream at a point where the water flows very slowly; cut a notch in the board sufficient in depth to pass all the water to be measured, and not more than two-thirds of the width of the stream in length. The edges of the notch on the bottom and sides must be bevelled to almost a sharp edge towards the downstream side as shown, and the surface of the water below the notch should not be less than one foot. About 3 feet behind the notch drive a stake into the bottom of the water-course, the top of the stake being level with the notch. When the water has reached its greatest depth, measure the depth, marked h in figure 3, from the surface to the top of the stake, by a thin-edged rule. The following Table will show the quantity of water in cubic feet per minute for weirs from one inch to twenty-five inches in depth. The number in the Table corresponding to the depth h when multiplied by the length of the notch in inches, will give the quantity of water in cubic feet per minute. For instance, if the depth of the notch be 5 inches and its length 40 inches, the multiplier given in Table 8 is 4·5; and the quantity of water delivered in cubic feet per minute is = 4·5 × 40 = 180 cubic feet.

Table 8.—MULTIPLIERS FOR FINDING THE QUANTITY OF WATER FLOWING OVER A NOTCHED-BOARD OR WEIR.

Depth of Weir (h).	Multiplier.	Depth of Weir (h).	Multiplier.	Depth of Weir (h).	Multiplier.	Depth of Weir (h).	Multiplier.
1	·40	7¼	7·84	13¼	19·42	19¼	33·94
1¼	·55	7½	8·25	13½	19·97	19½	34·60
1½	·74	7¾	8·66	13¾	20·52	19¾	35·27
1¾	·93	8	9·10	14	21·09	20	35·94
2	1·14	8¼	9·52	14¼	21·65	20¼	36·60
2¼	1·36	8½	9·96	14½	22·22	20½	37·28
2½	1·59	8¾	10·40	14¾	22·79	20¾	37·96
2¾	1·83	9	10·86	15	23·38	21	38·65
3	2·09	9¼	11·31	15¼	23·97	21¼	39·34
3¼	2·36	9½	11·77	15½	24·56	21½	40·04
3½	2·63	9¾	12·23	15¾	25·16	21¾	40·73
3¾	2·92	10	12·71	16	25·76	22	41·43
4	3·22	10¼	13·19	16¼	26·36	22¼	42·13
4¼	3·52	10½	13·67	16½	26·97	22½	42·84
4½	3·83	10¾	14·16	16¾	27·58	22¾	43·56
4¾	4·16	11	14·67	17	28·20	23	44·28
5	4·50	11¼	15·18	17¼	28·82	23¼	45·00
5¼	4·84	11½	15·67	17½	29·45	23½	45·71
5½	5·18	11¾	16·20	17¾	30·08	23¾	46·43
5¾	5·54	12	16·73	18	30·70	24	47·18
6	5·90	12¼	17·26	18¼	31·34	24¼	47·91
6¼	6·29	12½	17·78	18½	31·98	24½	48·65
6½	6·65	12¾	18·32	18¾	32·63	24¾	49·39
6¾	7·05	13	18·87	19	33·29	25	50·13
7	7·44						

Measurement of Flowing-water by the Velocity of the Water and Sections of the Stream.—Choose a length of the stream, say about 50 or 100 feet, along which the section is as uniform as possible, and find the area of the section by multiplying the width by the average depth. It is advisable to take several sections in the chosen length, from which to find the average section.

A stake should be fixed at each end of the measured length, and a float, consisting of a bottle, cork, or piece of wood, must be thrown into the middle of the stream a little above the first stake, and the time noted which it takes to pass from the one stake to the other. This should be several times repeated, and the average time taken, so as to get a more accurate result.

From these data the quantity of water passing can be found, as illustrated in the following example: The sectional area of a stream is 20 square feet, and a float passes over a measured length of 90 feet in 36 seconds. To find the quantity of water passing, multiply the area of the stream by the measured length, and also by 60, and divide the product by the time in seconds taken by the float in passing over the chosen length. Thus, $20 \times 90 \times 60 \div 36 = 3000$ cubic feet per minute. From this a deduction (amounting to about 20 per cent. for earthen banks) must be made to allow for loss of velocity at the sides and bottom through friction. In the above example, 20 per cent. of 3000 is equal to 600, and the true quantity passing will be $3000 - 600 = 2{,}400$ cubic feet per minute.

Measurement of Flowing-water by Discharge through an Orifice.—This method may often be used in situations where a sluice already exists.

The sluice must be raised, so that all the water coming down just passes through, and the length and depth of the opening carefully measured, together with the depth from the surface of the water in the head-race to the centre of the orifice, and from these data the quantity of water may be calculated by the following rule for the discharge through an orifice under a given head. *Rule:* Multiply the area of the aperture in square feet by the square root of the head in feet and by 5·1, the product will be the quantity discharged in cubic feet per second.

Example: Required the quantity of water in cubic feet per second discharged through an orifice or sluice, 15 inches wide and 1 foot 6 inches high, the head, or depth from the surface of the water in the head-race to the centre of the orifice, being 16 feet. Then $\sqrt[2]{16} = 4$ and 1·25 foot wide $\times$ 1·5 foot high $\times 4 \times 5{\cdot}1 = 38{\cdot}25$ cubic feet of water discharged per second. The velocity of the water is $\sqrt[2]{16} = 4 \times 5{\cdot}1 = 20{\cdot}4$ feet per second.

The Driving Power of Flowing-water being gravity, the power of a stream of water depends upon the height of the fall and the quantity of water flowing per minute. The theoretical horse-power of a stream of water may be found by this *Rule:*—

$$\text{Theoretical horse-power of stream} = \frac{\text{cubic feet of water falling per minute} \times 62{\cdot}5 \text{ lbs.} \times \text{fall in feet.}}{33000}$$

A deduction of 25 per cent. must be made from the result obtained by this rule to allow for the power absorbed by friction and for leakage, the remainder will be the actual power which should be developed by a good water-motor.

Example: Required the horse-power of a stream of water, passing 1600 cubic feet of water per minute over a fall of 33 feet, the height being measured from the level of the water in the head-race to the level of the water in the tail-race.

Then,

$$\frac{1600 \text{ cubic feet} \times 62{\cdot}5 \text{ lbs.} \times 33 \text{ feet height of fall}}{33000 \text{ lbs.}} = 100 \text{ horse power;}$$

the theoretical power of the stream and $100 \times {\cdot}75 = 75$ horse-power the actual power which should be obtained from the stream by a good water-motor.

WATER-MOTORS.

A Turbine, when correctly designed and constructed, is the best and most efficient motor for the utilization of water power. The chief types of turbines and their modes of action may be briefly described as follows:—

Classes of Turbines.—There are two classes of turbines, called respectively *Pressure* and *Impulse* turbines, the difference between them being that whereas in the former the water acts in part by impulse and in part by pressure, in the latter the water acts entirely by its impulse.

Whitelaw's Re-action Wheel is the simplest form of turbine. It has two arms formed in the shape of an Archimedean spiral, like Figs. 4 and 5. The water is supplied from the underside of the wheel, as shown in Fig. 5, through the centre of the arms, and flows horizontally outwards

Figs. 4 and 5.—Whitelaw's re-action wheel.

through the arms towards the periphery, and leaves the motor in a direction tangential and opposite to the direction of rotation. The water acts by re-action, and the most efficient speed of the wheel, at the discharging orifices at the extremity of the arms, is equal to the velocity due to the height of the fall. When the wheel is running at this speed there is a loss of efficiency of at least 16 per cent. arising from the backward velocity of the water as it leaves the wheel. The horse-power of this turbine may be found by the following *Rule:* Multiply the effective quantity of the water flowing through the wheel in cubic feet per minute by the height of the fall in feet and divide the product by 700.

Example: Required the horse-power of a Whitelaw-turbine, produced by 1000 cubic feet of water with a fall of 21 feet.

$$\text{Then } \frac{1000 \text{ cubic feet} \times 21 \text{ feet fall}}{700} = 30 \text{ horse power.}$$

The following are the rules for proportioning this turbine, with two properly formed jets:—

Width of each discharging orifice W.

$$W = \sqrt{\frac{135 \times \text{number of horse-power.}}{1000\ H \sqrt{H}}}$$

Where H = the height of the fall or head of water.
Width of each arm = 4 W.
Diameter of machine = 50 W.
Diameter of central opening = 10 W.

$$\textit{Number of revolutions per minute} = \frac{149{\cdot}44 \sqrt{H}}{\text{Diameter of the machine}}.$$

The efficiency of this turbine in practice seldom exceeds 55 per cent. It is seldom used now on account of its low efficiency, imperfect regulation and unstable speed, and it has been superseded by more efficient motors.

Causes of Loss of Efficiency of Turbines.—No turbine, however good and perfect in its action, can utilize all the power in a stream of falling water, as there are various losses of efficiency common, in a greater or less degree, to every turbine. These losses of efficiency arise from:—(1.) Shocks and collisions due to changes of section and curvature, and collision of the water on the tips of the guide and wheel-vanes. In a good turbine the changes of section and curvature have little or no effect, as all parts of the motor would be designed so as to secure gradual changes of section and curvature, but in faultily constructed ones their influence may be considerable. The loss arising from collision of the water on the tips of the guide-vanes should be inappreciable, but that due to collision on the tips of the wheel-vanes cannot in practice be quite eliminated, but should not exceed from $1\frac{1}{2}$ to 3 per cent.; (2.) Friction of the moving water on the surfaces of the motor. This is generally the most important source of loss in a turbine, indeed were it not for skin friction, a turbine might easily be made to give an efficiency of 90 or 95 per cent.; (3.) The energy carried away by the water as it leaves the wheel; (4.) Friction of the footstep or bearing of the turbine.

Guide-vanes of Turbines.—Fourneyron conceived the idea of giving the water an initial forward velocity before it entered the wheel or moving part of the turbine, and he effected this by means of fixed guide-vanes, which now form one of the fundamental parts of every turbine.

Water-paths and Velocities of a Turbine.—In Fig. 6, A represents the fixed part of a parallel flow turbine containing the guide-vanes, and B the moving part or wheel of the turbine; C D and E F are the centre lines of a guide-cell and wheel-bucket respectively. In this figure the water enters the guide-ports from the head-race or turbine-case in a direction parallel to the axis or shaft of the turbine, flows along the curved guide-blades, and is by them directed into the wheel with its proper velocity and direction. If c be the velocity and direction of the water as it leaves the guide-ports, on entering the moving wheel, this velocity is resolved into two components, viz., u_1, the velocity of the wheel at its receiving circumference, and c_1 the velocity of the water relatively to the moving wheel;

c and c_1 are generally known as the absolute and relative velocities of the water as it enters the wheel, and in order that the water may enter the wheel with as little collision as possible, the first portion of the wheel-vanes must be tangential to the direction of the velocity c_1. The discharge portion of the wheel-vane is straight so as to guide the water clear away from the turbine, and to secure a high efficiency the absolute direction of the water as it leaves the wheel must be at right angles to the direction of motion. The velocity of the wheel on its discharging circumference being known,—in a parallel-flow turbine the velocity of the wheel is necessarily the same on both the receiving and discharging circumferences and

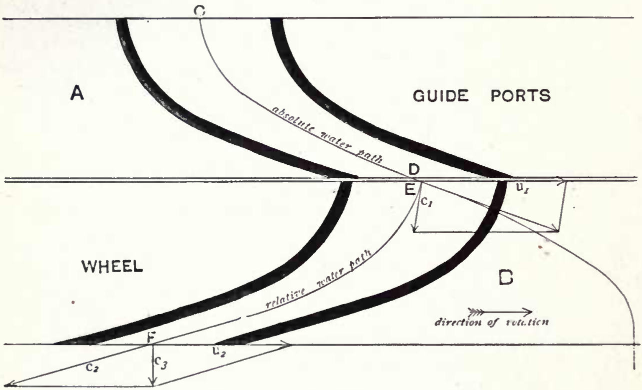


Fig. 6.—Diagram showing ports and buckets of a Jonval turbine, with water paths and velocities.

the direction of rotation—and also the absolute direction of the water as it leaves the wheel, it will be found that the smaller the discharging angles of the vanes the smaller becomes the loss of velocity and consequently of efficiency, as the absolute velocity (c_3) of the water as it leaves the wheel represents a loss of fall. The choice of the discharge angle depends upon various circumstances. In some cases this angle may be comparatively large, so that the turbine will pass a large quantity of water with a small diameter of wheel. A large angle, however, means an increased loss of efficiency, and should therefore only be adopted in cases where the water supply is abundant and a high efficiency is not necessary. Where a high efficiency is required, the angle must be made as small as possible. In Figure 6, u_1 is the velocity of the wheel at its discharging circumference, and c_2 and c_3 the relative and absolute velocities respectively of the water as it leaves the wheel: but the velocities and angles differ to a greater or less extent in almost every turbine.

In Pressure-Turbines the proportions between the velocities and angles are such that the water leaves the guides with a velocity due to about one-half the head or fall, the remaining portion of the fall acting by pressure. In order to keep up the correct distribution of pressure and velocity in the turbine, and thus maintain the efficiency, there must be a definite rate of flow through the wheel, the water must be admitted continuously over the entire circumference, and the wheel-buckets are necessarily filled with water under pressure.

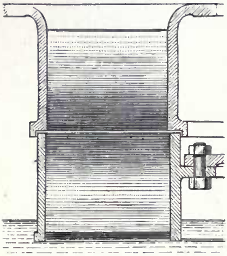
Pressure-turbines may very conveniently be divided into *inward, outward, and parallel-flow turbines*, accordingly as the water enters and leaves the motor.

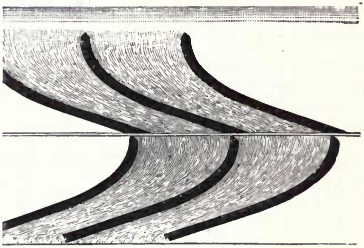
Regulation of Turbines.—As it frequently happens that the water available for driving a turbine varies considerably, often sinking down in summer to one-half or one-fourth of the usual supply, it becomes necessary to provide some means of regulating a turbine so that it will give a good efficiency not only with a full, but also with a very reduced supply. Many turbines will yield a high efficiency when fully supplied with water, but become almost useless when the supply sinks down to one-half or less, and hence the subject of regulation becomes of very great practical importance. In cases where the supply remains constant, or nearly so, but the power required varies, the turbine may be constructed to pass the water requisite for the maximum power required, and a stop, sluice, ring, or throttle-valve used to regulate the speed. These methods of regulation are not economical, but are often sufficient if the water supply is abundant, and are generally much cheaper than an efficient regulator applied to the turbine itself. A proper regulation should always be applied if the water supply varies.

Regulation of Inward-Flow Turbines.—In *inward-flow* wheels the water enters the guide ports on their outer circumference, flows towards the axis of the turbine, and leaves the wheel along its inner circumference. There are various methods of regulating *inward-flow* turbines, the commonest being by making the guide blades movable, so that by slightly turning them the discharge angle is altered and more or less water discharged as the case may be. This mode of regulation is often convenient, but is open to many objections, the chief one being, that although the discharge area of the guide-ports is altered, that of the wheel is in no way lessened, and this necessarily causes a loss of efficiency. Another objection is, that the speed of maximum efficiency of the wheel varies with the inclination of the guide blades, and thus for good working, the wheel should run at a different speed when the opening of the guide blades is altered.

Regulation of Outward-Flow Turbines.—In *outward-flow* wheels, the water enters the wheel on its inner, and leaves it on its outer circumference, and the regulation consists of a cylindrical sluice working between the guides and wheel. This manner of regulation is very imperfect and far from economical, as when the turbine is using the reduced supply, only the area of the guides is altered, that of the wheel remaining the same. Great contraction takes place as the water leaves the guides, and the water on passing into the wheel has to enter a space much larger that it can fill with the velocity it has attained. A great loss of efficiency thus takes place, and in order to diminish this, in the better forms of the Fourneyron turbine, the wheel and guides are divided into two or more tiers, and by shutting off one or more of these tiers to suit the decreased supply a fair efficiency is obtained, provided that the open tiers are fully supplied with water. The chief objections to this regulation are, that the skin friction is much increased, and the turbine is rendered very liable to be choked by leaves. The efficiency of the best Fourneyron turbine is a little lower than that of an inward or parallel-flow turbine.

The Jonval or Parallel-Flow Turbine is in many respects simpler than either of the preceding types. In this turbine, the water enters the guides parallel to the axis of the shaft, and after passing through the motor leaves the wheel in the same direction. Figs. 7 and 8 are sections of the ports and buckets of a Jonval turbine. For low falls this turbine is much





Figs. 7 and 8.—Sections of the ports and buckets of the Jonval turbine.

cheaper than any other, both in the cost of the motor itself and in its erection. As the water passes parallel to the axis of the motor, a smaller turbine case, where one is necessary, is required for a given diameter of wheel than would be requisite for an inward-flow turbine.

The Jonval, in common with all other pressure-turbines, does not admit of a perfect adjustment for a reduced water supply, but it frequently admits of a better one than either of the other types. On low falls it may be provided with vertical slides fitted into the guide ports, one slide to each port. These slides are raised and lowered by suitable mechanism, either singly, or three together. When the turbine is applied to medium falls, a slide may be constructed so as to close two guide ports simultaneously.

The efficiency of a Jonval turbine is as high or perhaps higher than any other pressure-turbine, a well-constructed wheel will yield an efficiency of from 75 to 78 per cent.

Most pressure-turbines work equally well whether totally immersed or free from the tail-water, consequently they are especially suitable for situations where considerable fluctuation takes place in the levels of head or tail-water. They are also capable of being arranged so that part of the fall, provided the portion so used does not exceed 25 to 30 feet, acts by suction. In such a case the turbine can be arranged with its shaft either vertical or horizontal; the suction pipe must be perfectly air tight, and with its lower end always below the surface of the tail-water. A Jonval turbine with horizontal shaft and adjustment is shown in Fig. 9. Excellent as pressure turbines are for low or medium falls with constant or nearly constant water supplies, they should not be applied to high falls, as under such circumstances their speed of rotation becomes very great, and the wear, tear, and repairs are much increased.

Impulse-Turbines differ from pressure-turbines, inasmuch as in the former the water leaves the guides with the velocity due to the entire head or fall, and thus acts entirely by impulse. Since the water acts by impulse alone, the wheel-buckets do not require to be filled with water, and the turbine may consequently be so constructed that the water glides along the concave surfaces of the wheel-vanes without touching the convex sides, as shown in Fig. 11. As the Girard turbine is now the only impulse-turbine

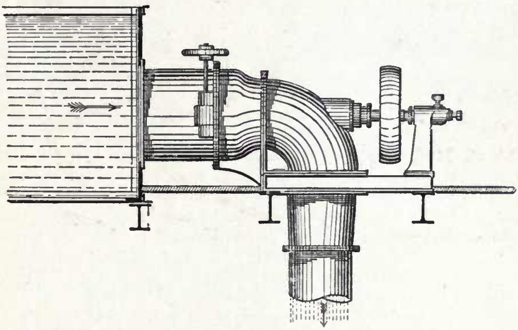
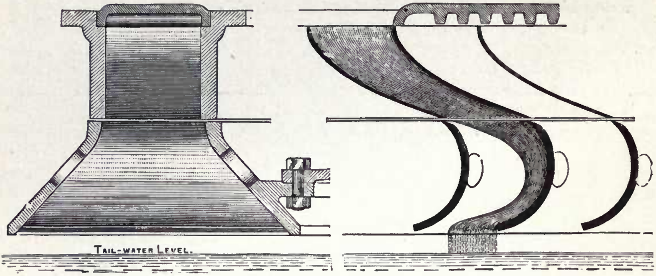


Fig. 9.—Jonval turbine, with horizontal shaft.

of importance, it is only necessary to refer to the various forms of this motor.

The Girard-Turbine, of which Figs. 10 and 11 are sections of the ports and buckets, should be placed so that the bottom of the wheel is just clear of the tail-water when the turbine is working; the object of this being to secure a perfectly free discharge for the water as it leaves the wheel. An inch or two of clearance between the bottom of wheel and top of tail-water is ample. In the Girard-turbine, it is not necessary that the water should enter the wheel over its entire circumference, consequently the injection may take place over only a small portion, and the turbine will still retain its high efficiency. This property is of great advantage, as it enables a perfect regulation to be applied to the motor, for if the turbine is constructed for the maximum quantity of water available it can be adjusted for decreased supplies by merely closing some of the guide ports. This turbine may be applied to any fall, from about 6 feet upwards, and for high falls it is the only one capable of giving a high efficiency.

One of the best known turbine-makers, Mr. W. Günther, of the Central Works, Oldham, recommends this turbine for low and medium falls with variable supplies of water, for medium falls with small varying quantities of water, and for high falls. For low and medium falls with constant or nearly constant water supplies, and for situations where the rivers are subject to frequent floods, he adopts the Jonval turbine.



Figs. 10 and 11.—Sections of ports and buckets of the Girard-turbine.

Axial and Radial Girard-Turbines.—There are two chief types of the Girard-turbine, known respectively as the axial and radial. In the former the water enters and leaves the motor in a direction parallel to the axis, as in the Jonval turbine, while in the latter it enters the wheel on its inner and leaves on its outer circumference. The axial type may be used for low, medium, and high falls, and is invariably constructed with the shaft vertical. For low falls this turbine is made with full injection, that is, injection over the entire circumference of the wheel, and the regulation consists of vertical slides fitted to each guide port, the slides being raised or lowered by suitable gearing worked from the turbine-house or other convenient place.

For medium falls, full or half injection, according to circumstances, is used, and the regulation is effected by a slide arranged so as to close two guide ports simultaneously.

For high falls and small quantities of water this turbine is made with partial injection, that is, the water is only admitted on a fraction of the circumference, and the diameter of the wheel is increased so as to secure a moderate number of revolutions. A high-fall Girard turbine with partial injection is necessarily more expensive than a turbine of small diameter with full injection, but as the former runs at a much slower speed it is not subject to the wear, tear, and friction of a quick running motor, it gives a higher efficiency, and is much more durable.

The Radial Girard-Turbine is generally only used for high falls, and is then constructed with a horizontal shaft. Partial injection is always applied, and the method of regulating the turbine generally consists of a slide closing the guide ports one after the other. Turbines of the horizontal type have frequently been constructed for falls exceeding 500 feet, and worked with perfect success, thus testifying to the adaptability of this motor to high falls. It is scarcely necessary to add that under such high heads a pressure turbine would run at such a high speed as to become almost unmanageable.

A Girard-turbine arranged with a vertical shaft, for a medium fall and variable water supply, is shown in Fig. 12. In this arrangement the guide-channels are on the whole circumference, and the adjustment for varying the supply is so arranged that the turbine can work with either full or partial

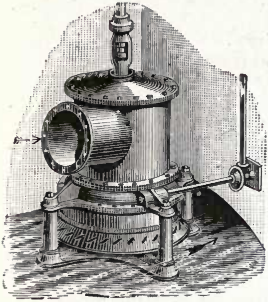


Fig. 12.—Girard-turbine with vertical shaft for medium falls.

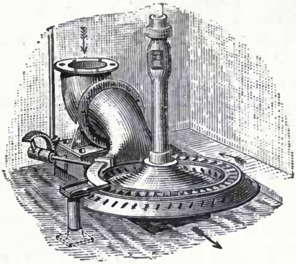


Fig. 13.—Girard-turbine with vertical shaft for high falls.

injection. For low falls the outer case is dispensed with, and the guide-channel cylinder is fixed direct to the bottom of the head-race.

A Girard-turbine arranged with a vertical shaft for high falls and varying water supplies, partial injection being employed in this case, is shown in Fig. 13.

Another Girard-turbine for high falls and variable supplies, is shown in Fig. 14. In this arrangement the shaft is placed horizontally, and the water enters the wheel on the inner circumference, and leaves it on the outer circumference. The wheel is protected by a wrought-iron case, which is not shown in the woodcut. This arrangement is especially adapted for large powers under very high falls, and partial injection is always used.

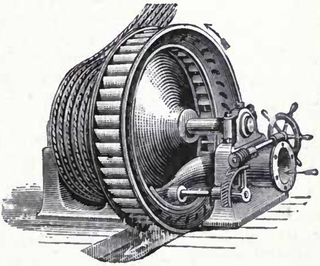


Fig. 14.—Girard-turbine with horizontal shaft for high falls. Cover removed.

The proportions of a Turbine should be adapted to the fall, quantity of water, location, and circumstances under which it has to work. The following general rules for the Jonval-turbine may be modified as circumstances require.

Rules for Proportioning Jonval's Turbine.—To find the Diameter at the Centre of the Buckets, or Centre of Motion D.—*Rule:* Multiply the square root of the height of fall in feet by 1000, and divide the product by the number of the revolutions of the turbine per minute.

Number of buckets in the turbine wheel = six times the square root of the diameter of the centre of motion D. Number of guides = 5 times the square root of D. Depth of buckets = one-eighth of D. Radius of the curved portion of the bucket = depth of bucket × 1·25. Depth of guides = one-sixth of D. Radius of the curved portion of guides = depth of guide × 1·75.

To find the total horse-power of a fall of water delivered on a Turbine.—Rule: Multiply the quantity of water passed through the turbine in cubic feet per second by the height of the fall in feet, and multiply the product by ·1134. These wheels, as previously stated, yield as high as 78 per cent. of the total power expended, when well constructed.

Example: Required the horse-power of a fall of water 28 feet high, discharging 15 cubic feet of water per second, taking the efficiency at 65 per cent., a very low estimate.

$$15 \times 28 \times \cdot 1134 = 47 \cdot 628 \text{ total horse-power,}$$

of which 47·628 × ·65 modulus = 30·958, or say 31 horse-power would be available as actual horse-power in a Jonval-turbine, and this fall would require a turbine, making say 190 revolutions per minute =

$$(1000\sqrt{28 \text{ feet fall}}) \div (190 \text{ revolutions}) = 27 \cdot 85 \text{ inches diameter.}$$

Water-Jet Motors.—The application of a jet of water to a number of cups or buckets attached to the rim of a wheel is a simple and excellent method of producing motive power. It is, however, essential to economy and efficiency that the formation of the buckets shall not permit the lodgment and carrying-over of water which has lost its impelling power, termed dead-water, and that the water shall be received without shock and discharged without velocity. It is also necessary that the buckets be small in area and number, because the loss by friction is proportional to the area of the wetted surface. The wheel should be driven by the impulse of the jet, or by a uniform and continuous pushing-action of the water. When it is driven by the impact of the jet, or by a succession of blows of the water, such as result from variations in the angle of the impingement of the jet, there is a considerable loss of efficiency. The velocity of the bucket should be equal to one-half the velocity of the impelling jet.

Buckets of Water-Jet Motors.—The action of a jet of water in striking a flat plate at right angles is shown in Fig. 14A. The water divides and forms a wedge of dead-water, and the direction of the discharge precludes a complete stoppage of the water. The action of a jet of water in striking a curve-shaped bucket is shown in Fig. 14B. The same wedge-shaped formation of dead-water exists as in the preceding case, but there is a reversal of the stream which allows it to be almost completely checked and exhausted of its energy.

Fig. 14A.—Water-jet striking a flat plate.

Fig. 14B.—Water-jet striking a curve-shaped bucket.

The Pelton Water-Wheel.—In the double-curve or wedge-centre bucket, designed by Pelton, shown in Fig. 14C, the loss due to water-wedge is avoided, because a piece of metal projects into the middle of the stream and occupies the space of the water-wedge. The jet is applied tangentially, and becomes divided into two parts, one turning to the right and the other to the left; the direction of both being almost completely reversed before the water leaves the bucket. The wheel revolves vertically, and is highly efficient, as will be seen from the following table. To facilitate the escape of the spent-water, and to utilize all of the head of water, the stream is usually applied to the lower side of the wheel.

Fig. 14c.—Water-jet striking a Pelton-bucket.

Table 9.—Results of Tests of a Pelton Water-Wheel.

Size jet. Inch.	Running pressure.	Revolutions.	Actual water used.	Actual Horse-power developed.	Theoretical Horse-power possible.	Efficiency percentage.
$\frac{5}{8}$	100	775	14·78	5·349	6·422	83·30
$\frac{5}{8}$	103	780	15·00	5·563	6·715	82·90
$\frac{3}{4}$	125	880	23·05	10·730	12·520	85·69
$\frac{3}{4}$	102	775	20·82	7·845	9·226	85·02
$\frac{3}{4}$	100	775	20·61	7·756	8·957	86·59
$\frac{3}{4}$	125	900	23·05	10·670	12·520	85·16
$\frac{3}{4}$	100	780	20·61	7·717	8·957	86·15

The power of the Pelton-wheel is independent of its diameter, and depends upon the head and volume of the water.

The Pelton-wheel with a single nozzle may be employed for heads of water as low as 30 feet, if the power required is small. When large power

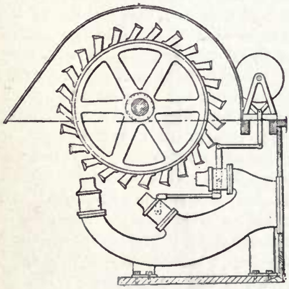


Fig. 14d.—Pelton Water-wheel with multiple nozzles. By the Pelton Water-wheel Co., San Francisco.

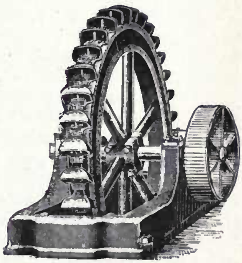


Fig. 15.—Pelton Water-wheel. By the Pelton Water-wheel Co., San Francisco.

is required from a comparatively low head of water, a wheel with multiple nozzles is employed, as shown in Fig. 14d.

The Pelton-wheel, shown in Fig. 15, is especially adapted for heads above 50 feet; and there is no limit to the height of head of water under which it will efficiently work. For instance, a Pelton water-wheel of 36 inches diameter, formed of a solid steel disc, with phosphor-bronze buckets riveted to the rim, as shown in Fig. 15A, is working under a vertical head of 2100 feet, equivalent to a pressure of $2100 \times \cdot433 = 910$ pounds per square inch. The wheel makes 1150 revolutions per minute; the peripheral velocity is 10,804 feet per minute, or about 120 miles an hour.

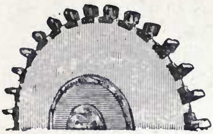


Fig. 15A.—Pelton Water-wheel formed of a solid disc.

At an electric light works, there are eight Pelton-wheels of 24 inches diameter, each weighing 90 pounds, and capable of developing 175 horse-power, or nearly two horse-power for each pound weight of the wheel. The speed is 1000 revolutions per minute, and the head of water is 820 feet, equal to a pressure of $820 \times \cdot433 = 355$ pounds per square inch.

Other Pelton-wheels are working under pressure of from 700 to 1000 pounds per square inch. The latter is equal to a head of $1000 \times 2\cdot31 = 2310$ feet, under which a wheel of 18 inches diameter, weighing 30 pounds, will develop 21 horse-power, with a nozzle-tip of $\frac{1}{4}$ inch diameter.

Considerably more power is stored in the rivers and streams of the world than is sufficient for all the industrial purposes of mankind. It is astonishing that only a small percentage of the available power has hitherto been utilized, considering that efficient motors are obtainable, adapted to all heads and purposes, for the economical utilization of water-power.

HEAT AND FUEL.

A Thermometer is an instrument for measuring temperatures. It consists of a glass-tube having a bulb at the foot, containing either mercury or alcohol: mercury being used for ordinary temperatures, and alcohol for very low temperatures, because it remains fluid and does not solidify at the greatest known cold. A scale is placed at the side of the tube, graduated into degrees, which indicates the expansion of the fluid in the tube, from which the temperature is read off. The temperature of melting ice being constant at all temperatures, it is used for marking the zero point of centigrade and Réaumur thermometers: in Fahrenheit thermometers the zero point is placed 32° below this, at about the temperature of a mixture of salt and snow. As distilled water under the same pressure, in a vessel of the same kind, always boils at the same temperature, it is used for marking the boiling point of thermometers. After the mercury is introduced, it is boiled to expel air and moisture, and the tube is hermetically sealed. The action of the thermometer is due to the change of bulk or volume to which bodies are subject with a change of temperature; they expand with heat, and contract with cold, thus indicating a high or low temperature.

The Fahrenheit Thermometer is used in this country and in America. The number 0° on its scale represents the greatest degree of artificial cold that could be produced at the time the thermometer was invented. The number 32° represents the freezing point, or temperature of melting ice, and 212° the temperature of boiling water, in both cases under atmospheric pressure. From the freezing-point to the boiling-point there are 180 degrees.

The Centigrade Thermometer is used in France and other parts of the continent; number 0° on its scale represents the temperature of melting ice, and 100° the temperature of boiling water. From the freezing-point to the boiling-point there are 100 degrees.

The Réaumur Thermometer is used in Russia and Turkey, &c. Number 0° on its scale represents the temperature of melting ice, and 80° the temperature of boiling-water. From the freezing-point to the boiling-point there are 80 degrees.

To convert Degrees Fahrenheit into Degrees Centigrade.—*Rule:* Subtract 32, multiply the remainder by 5, and divide the product by 9.

To convert degrees Centigrade into Degrees Fahrenheit.—*Rule:* Multiply by 9, divide the product by 5, and add 32 to the quotient.

To convert Degrees of Centigrade into Degrees Réaumur.—*Rule:* Multiply by 4 and divide the product by 5.

To convert Degrees of Réaumur into Centigrade.—*Rule:* Multiply by 5 and divide the product by 4.

To convert Degrees of Fahrenheit into Degrees of Réaumur.—*Rule:* Subtract 32, multiply the remainder by 4, and divide the product by 9.

To convert Degrees of Réaumur into Degrees of Fahrenheit.—*Rule:* Multiply by 9, divide the product by 4, and add 32 to the quotient.

A Thermometer is used by marine engineers with the salinometer to test the temperature of the water drawn from the boiler for the purpose of ascertaining its density, and to test the temperature of the feed-water, and of the air in the engine-room.

High Temperatures beyond the range of a Thermometer may be ascertained approximately, by heating a bar of wrought-iron to the temperature required to be ascertained, and then quenching it in cold water, when the rise of temperature of the water will enable the unknown temperature required, to be calculated by the following rule, which assumes the specific heat of wrought-iron to be one-ninth that of water:—

Let T = the temperature of the water produced by quenching the iron.
t = the original temperature of the cooling-water.
W = the weight of the cooling-water in lbs.
w = the weight of the bar of wrought-iron in lbs.
X = the unknown temperature required.

$$X = \left[\frac{(T - t) \times W \times 9}{w}\right] + T.$$

Example: A bar of wrought-iron weighing 20 lbs. was inserted in the chimney of a steam boiler, and when heated, was quenched in 30 lbs. of water at 55° Fahr., thereby raising the temperature of the water to 93° Fahr. Required the temperature of the chimney.

Then $\frac{(93^\circ - 55^\circ) \times 30 \text{ lbs. of water} \times 9}{20 \text{ lbs. weight of wrought-iron}} = 513^\circ$ Fahr., and 513 + 93 = 606° Fahr., the temperature of the chimney.

The Standard Temperatures of Water are as follows:—

The freezing-point under one atmosphere is	32°	Fahr. or	0	Cent.
The point of maximum density . . .	39°·1	„ „	4°	„
The British standard temperature . .	62°·0	„ „	16°·66	„
The boiling-point under one atmosphere .	212°	„ „	100°	„

The temperature used in calculating the specific gravity of bodies is usually 62° Fahr.

Notable Temperatures.—Melting ice, 32° Fahr.; boiling water, 212° Fahr., under the pressure of one atmosphere or in the open air; steam at 60 lbs. pressure per square inch by the steam-gauge, 307°·5 Fahr.; superheated steam 380° to 400° Fahr. Smoke in the funnel of a marine-boiler, 552° to 600° Fahr.; water in the hot-well, 100° to 120° Fahr.; boiler-furnaces, 2500° to 3000° Fahr. Dull cherry-red heat, 1470° Fahr.; full cherry-red, 1700° Fahr.; orange-colour, 2000° Fahr.; white-heat, 2370° Fahr.; bright white-heat, 2500° Fahr.

The Mean Temperature of a place is the mean of its annual temperature averaged for a number of years. The mean daily temperature is obtained by dividing the sum of 24 hourly observations by 24—the temperature being taken of the air and not of the ground. The mean temperature of 24 successive hours is approximately equal to the temperature at 9 o'clock A.M., and the mean temperature of the day—from 9 o'clock, A.M., to 5 o'clock, P.M.—is approximately equal to the temperature at 12 o'clock noon.

Temperature of the River Thames.—Sir C. B. Airy found from observations of the temperature of the Thames extending over many years, that on the average of thirty-three years the temperature of the Thames—51·7°—is higher than the air at the Royal Observatory—50°·2—by 1½ degrees. During the seven months, May to November, this difference averages 2°; and during the winter, December to April, only 0·7°. On the average of the thirty-three years, July gives the highest monthly mean river temperature—65·7°—and January the lowest—39·4°. The high temperature of 73·1° was recorded as the average for June, 1846, and of 75·4° on July 20th, 1859.

Temperature of Seas and Lakes.—The temperature of the surface of the sea varies with the seasons and with the direction of the ocean currents. The surface temperature of tropical seas is generally the same as that of the air, but that of polar seas is higher than that of the air. The average winter temperature of the sea round the coast of England is higher than that of the land. The mean annual temperature of the surface of the sea round England is 49° Fahr.; the mean surface temperature of the Indian Ocean is 89° Fahr.; and of the Red Sea 94° Fahr. The temperature of maritime land is influenced by the winds which come from the sea, as the air resting upon the surface of the sea acquires its temperature, and is distributed over the land.

The Temperature of Deep-Sea-Water is considerably less than that of the surface water; the temperature decreases as the depth increases. The temperature of the bottom of the sea in both temperate and tropical climates averages 36° Fahr. The temperature of the bottom of lakes averages 39° Fahr.

The surface water is the warmest part of the sea, its temperature extends to different depths in different seas, forming a stratum generally from 100 to 300 fathoms deep, below which the water becomes colder towards the bottom, as will be seen from Table 10, which contains the results of observations in various parts of the world.

Table 10.—TEMPERATURE OF SEAS AND LAKES AT VARIOUS DEPTHS.

	Surface Temperature. Fahr.		Fthms. Fahr.
North Pacific Ocean	. 70°	Temperature at a depth of	2500=34°
South Pacific Ocean	. 67°	,, ,,	1950=31°
North Atlantic Ocean	. 40°	,, ,,	3000=34°
South Atlantic Ocean	. 50°	,, ,,	300=33°
Equatorial Ocean	. 78°	,, ,,	300=39°
,,	78°	,, ,,	600=35°
,,	78°	,, ,,	900=32°
Lake of Geneva	. 78°	,, ,,	950=42°
Lake Sabatino, Rome	. 77°	,, ,,	490=44°
Loch Lomond, Scotland	50°	,, ,,	500=42°

Springs of Water assume the temperature of the ground through which they pass. Shallow springs have the same temperature as the air, but deep springs assume the temperature of the stratum of constant temperature, or the mean annual temperature of the place.

Warm Springs of water rise from a depth below the stratum of constant temperature. Their temperature is due to the internal heat of the earth, and is an approximate indication of the depth from which the water rises. Warm or thermal springs are largely impregnated with mineral matter, such as magnesia, soda, iron, lime, manganese, potash, bromine, lithia, iodine, and other substances. The maximum temperatures of a few noted thermal springs are: Great Geyser, Iceland, 261° Fahr.; Chandes-Aignes, 180° Fahr.; Wiesbaden, 160° Fahr.; Baden-Baden, 155° Fahr.; Lucca, 130° Fahr.; Bath, 120° Fahr., and Buxton, 82° Fahr.

The Temperature of the Earth at the surface nearly equals that of the air, but below the surface the temperature varies greatly at different localities and in different geological formations. Limestone is the coolest formation. The two coolest mines or tunnels are in limestone, viz., Chanarcillo Mines and Mont Cenis Tunnel.

Underground Temperatures.—The normal temperature of the earth at a depth of about 30 feet in this country, and at a depth of 55 feet in warm climates, is constant, and equal to the mean annual temperature of the air at that place. Below that depth the temperature gradually increases with the depth, and although it varies in amount in different kinds of rock, its average rate of increase is 1° Fahr. for every 55 feet in depth, except in exceptionally hot mines, where it sometimes increases 1° Fahr. for every 30 feet in depth. In a deep bore-hole near Schladebach, Germany, the temperature at a depth of 4567 feet was 120° Fahr. In a deep artesian-well at Pesth, the temperature at a depth of 3120 feet was 158° Fahr. In a deep bore-hole at the Waterworks, Richmond, Surrey, the temperature at a depth of 1447 feet was found to be $76\frac{3}{4}$° Fahr. In a shaft in the Aberdare Valley the temperature was found to be:—At a depth of 546 feet = 56° Fahr.; 780 feet = $59\frac{1}{2}$° Fahr.; 1020 feet = 63° Fahr., and at 1272 feet deep = $66\frac{1}{2}$ Fahr. At the Denton Colliery the temperature at a depth of 1317 feet was 66° Fahr.

Hot Mines.—The mines on the Cumstock Vein, Nevada, are extremely hot; at depths of from 1500 to 2000 feet the thermometer placed in a freshly-drilled hole will register 130° Fahr., and small bodies of water run for years at 170° Fahr., and large bodies of water at 155° Fahr. The temperature of the air is kept down to 110° Fahr. by forcing in fresh air cooled over ice. In one of the mines the temperature increased as follows:—

100 to 1000 feet deep increase 1° Fahr. in				20 feet.
100 to 1800	"	"	"	30·5 feet.
100 to 2300	"	"	"	30·3 feet.

Iced water is drunk by the miners in these hot mines, with apparently no bad results.

Deep Mines.—The high temperature of deep mines forms an obstacle to their working. At the Comstock Mines, Nevada, some years ago, the miners could only work a few minutes at a time on account of the great heat. At the New Almaden Silver Mine, California, the temperature at a depth of 600 feet was 115° Fahr. A coal mine in Durham, 1814 feet deep, has a temperature at the bottom of 78° Fahr.; at another, near Manchester, 2150 feet deep, the temperature is 75° Fahr.; at a copper mine in Cornwall, 2100 feet deep, the temperature is 88° Fahr.; and water is obtained from a well at Grenoble, France, 1797 feet deep, at a temperature of 81·7° Fahr.

The Internal Heat of the earth may be seen from the following examples, taking in each case the surface temperature at 42° Fahr., and the rate of increase of temperature at 1° Fahr. for every 60 feet in depth:—

Water will boil at a depth of

$$\frac{212^\circ \text{ boiling point} - 42^\circ \times 60 \text{ feet}}{1760 \text{ yards} \times 3 \text{ feet}} = 1{\cdot}93 \text{ miles; and}$$

Brass will melt at a depth of

$$\frac{1650^\circ \text{ melting point} - 42^\circ \times 60 \text{ feet}}{1760 \text{ yards} \times 3 \text{ feet}} = 18{\cdot}27 \text{ miles.}$$

Quantities of Heat are expressed in units of weight of water heated one degree.

The British Unit of Heat, or Thermal Unit, is the quantity of heat necessary to raise the temperature of one pound of water at 32° Fahr. one degree Fahr.—that is, from 32° to 33°. Dr. Joule found that by the expenditure of one unit of heat, 772 lbs. weight could be raised one foot high. The mechanical measure of heat is, therefore, taken at 772 foot-pounds for one unit of heat. Heat and mechanical energy are mutually convertible, and heat requires for its production, or produces by its disappearance, mechanical energy, in the proportion of 772 foot-pounds for each unit of heat.

The Specific Heat of a body means its capacity for heat, or its power of storing heat; or the quantity of heat required to raise the temperature of the body one degree Fahr., compared with that required to raise the temperature of an equal weight of water one degree. Water is taken as the standard for comparison of specific heat, and its specific heat exceeds that of nearly all other bodies. The specific heat of all solid and liquid substances is nearly constant for temperatures up to 212° Fahr.; but above that point the specific heat increases as the temperature rises. The specific heats of solid and liquid substances at ordinary temperatures, are given in the following table:—

Table 11.—Specific Heat of Solid and Liquid Bodies, and of Gases, from the Experiments of Regnault, Pouillet, Petit, and Dulong, Dalton, Despretz, and Laplace.

Water at 32° Fahr. =	1·0000	Chloride of calcium	·1642
Iridium	·1886	Zinc, 32—572° F.	·1015
Manganese	·1442	Lead, ·0314: Gold	·0325
Cast-iron	·1299	Silver	·0570
Steel, soft, ·1166: hard	·1185	Antimony, 32—572° F.	·0547
Wrought iron, 32—212° F.	·1099	Bismuth	·0308
„ 32—392° F.	·1152	Cadmium	·0567
„ 32—662° F.	·1256	Platinum at 212° F.	·0335
Copper, 32—212° F.	·0952	„ 572° F.	·0343
„ 32—572° F.	·1014	„ 2192° F.	·0381
Brass, ·0940: Tin	·0569	Mercury, solid	·0320
Zinc, ·0955: Cobalt	·1069	„ liquid	·0334
Molybdenum	·0721	„ 32—572° F.	·0350
Palladium	·0593	Nickel	·1087
Uranium	·0619	Sulphate of potash	·1901
Tungsten	·0364	„ lead	·0873
Quicklime	·2170	Protochloride of mercury	·0689
Magnesian limestone	·2175	Perchloride of tin	·1016
Chalk	·2149	Diamond	·1469
White marble	·2159	Sapphire	·2174
Stonework	·1972	Bromine, ·0840: Iodine	·0542
Brickwork	·1918	Tellurium	·0516
Glass, 32—212° F.	·1770	Oak, ·5710: Fir	·6510
„ 32—572° F.	·1900	Pear tree	·5020
Coke, ·2030: Coal	·2412	Olive oil	·3010
Anthracite	·2017	Turpentine	·4700
Graphite, natural	·2019	Acetic acid, concentrated	·6580
„ from blast fur-		Vinegar, ·9200: Alcohol	·6590
naces	·4970	Essence of orange	·4890
Magnesia	·2216	„ lemon	·4880
Ice, ·5040: Soda	·2311	„ juniper	·4770
Animal black	·2609	Benzine, 59—68° F.	·3932
Charcoal	·2415	Ether, oxalic	·4555
Phosphorus, 32—212° F.	·2504	„ sulphuric, density	
Sulphur	·2026	·76°	·6600
„ recently cast	·1845	Chloride of calcium, solu-	
Nitrate of silver	·1436	tion	·6448
„ potass	·2388	Wood spirit, 59—68° F.	·6010
„ soda	·2783	Sulphuric acid	·6613
„ barytes	·1523	Water from 32—212° F.	1·0052
Chloride of lead	·0665	Air	·2380
„ tin	·1476	Nitrogen, ·2445: Oxygen	·2190
„ zinc	·1362	Hydrogen	3·4050
„ magnesium	·1946	Gaseous steam	·4757
„ manganese	·1426	Ammoniacal gas	·5086
„ sodium	·2200	Olefiant gas	·3706

The Specific Heat of Bodies varies considerably, as may be seen from the previous table. Woods average one-half of the specific heat of water; coal averages one-fourth, and coke, stone, brick, glass and sulphur each average one-fifth the specific heat of water. The metals have the least specific heat. The specific heat of bismuth is ·03084, therefore, the quantity of heat that would raise a given weight of bismuth through one degree Fahr. would only raise the temperature of the same weight of water through ·03084 of a degree. The specific heat of mercury is ·033, and the quantity of heat that would raise the temperature of mercury one degree, would only raise the temperature of the same quantity of water ·033 of a degree; hence the same quantity of heat that would raise 1 lb. of water 1 degree, would raise the temperature of 30 lbs. of mercury 1 degree. The specific heat of iron is only about one-ninth that of water, therefore nearly 9 lbs. of iron would be raised to a given temperature, by the same quantity of heat which would be required to raise 1 lb. of water to the same temperature.

The specific heat of the same body is less in the solid than in the liquid state, for instance:—

The specific heat of water is 1·000 liquid and ·504 solid.
Ditto mercury ·0333 „ ·0319 „
The specific heat of water in a gaseous state, or steam, is ·662.

Capacity for Heat means the quantity of heat required to raise the same weight of different bodies through the same number of degrees of temperature. If the same weight of several different substances be heated to the same degree of temperature and tested in an ice-calorimeter, their capacity for heat will be determined by the quantity of ice melted by each substance. Water has a greater capacity for heat than any other substance.

The Calorimeter is an instrument for determining the total amount of heat in a body, or its specific heat. The ice-calorimeter consists of three concentric vessels of tin, in the central one is placed the heated substance to be tested, and the other two are filled with pounded ice. The ice surrounding the central vessel is melted by the heated substance, and the ice in the outer vessel excludes the heating influence of the external air. Each compartment is fitted with a cock to draw off the water produced from the liquefaction of the ice. The water from the melted ice in the compartment surrounding the central vessel will be proportional to the heat stored in the substance in the calorimeter; and if the weight of melted ice be divided by the number of degrees through which the substance has fallen, it will give the quantity of ice which the substance would melt by falling through one degree. A substance in cooling from a given temperature to zero, gives out as much heat as it absorbs in being heated from zero to the given temperature.

The Calorimeter used in Boiler-Tests, in its simplest form, consists of a barrel provided with a stirring-arm revolving on a vertical shaft for mixing hot and cold water. The barrel is placed upon a weighing machine, and is supplied with a certain weight of cold water, into which steam is discharged from a pipe connected to the boiler. The rise of temperature of the mixture of water and steam is indicated by a thermometer, and when the temperature fixed as a basis of calculation is reached, the weight of the water is taken. The difference between the weight of the barrel of water before and after the addition of the steam, gives the weight of condensed water received as steam from the boiler. If—

> W = the original weight of water in the calorimeter,
> w = the weight of condensed water, or water added by heating with steam,
> t = the temperature of the original water in the calorimeter,
> t_1 = the temperature of the water after the admission of steam to the calorimeter,
> T = the temperature of the steam admitted to the calorimeter,
> l = the latent heat in the steam of boiler-pressure :—

then the heat imparted to the water by the steam will $= W(t_1 - t)$. The sensible heat imparted to the water by the steam $= w(T - t_1)$. If the portion $w(T - t_1)$ representing the sensible heat of the water added by the condensation of the steam from the boiler, be subtracted from $W(t_1 - t)$, representing the total heat given to the water in the calorimeter, the remainder will be the heat-units used for evaporation in the steam at boiler-pressure, which divided by the latent heat of steam, l, will give the weight of water in lbs. which the heat of steam of that pressure will evaporate. Representing the weight of water in lbs. by x, then

$$x = \frac{W(t_1 - t) - w(T - t_1)}{l}$$

If E = the heating efficiency of the steam supplied, compared with saturated steam between the same limits of temperature, H = the total heat of the steam at the observed pressure, and Q = the quality or dryness of the steam :—Then

$$E = \frac{W(t_1 - t)}{w(H - t_1)}$$

$$Q = 1 - \frac{(H - t_1)(1 - E)}{l}$$

or

$$Q = \frac{1}{l}\left\{\frac{W}{w}(t_1 - t) - (T - t_1)\right\}$$

Then when Q is less than 1, the percentage of moisture in the steam is $= 100(1 - Q)$. When Q is greater than 1, the number of degrees that the steam is superheated $= 2{\cdot}0833\, l\,(Q - 1)$.

Latent Heat.—When a solid body is heated and ultimately passes into the liquid state under the influence of heat, the temperature of the body rises until it reaches the melting point, when the temperature remains constant whatever the intensity of the heat may be, and the heat thus absorbed by the body in changing its condition becomes latent, and is not sensible to the thermometer, its only effect being to maintain the body in its liquid state. This is called the latent heat of fusion or liquefaction, and represents the number of units of heat absorbed by 1 lb. of the solid in passing to the liquid state; when, on the contrary, the liquid passes into the solid state, the latent heat is disengaged or restored. Every body, under the same pressure, solidifies at a fixed temperature, which is the same as that of fusion or liquefaction, and the temperature remains constant during solidification.

The Latent Heat of a Non-Metallic Substance may be found by M. Person's *Rule:* Subtract the specific heat of the substance in its solid state from the specific heat in its liquid state, and multiply the remainder by the number of the degrees Fahrenheit of the melting point, plus 256. The product is the latent heat of fusion or liquefaction in heat units.

Table 12.—LATENT HEAT OF FUSION OR LIQUEFACTION OF SOLID BODIES.

Description of Substance.	Latent Heat in Units of Heat.	Authority.
Ice	142·6	Person.
Chloride of calcium	73·0	,,
Phosphate of soda	120·0	,,
Phosphorus	9·0	,,
Spermaceti	148·0	,,
Wax	175·0	,,
Sulphur	17·0	,,
Nitrate of soda	113·0	,,
Nitrate of potass	85·0	,,
Tin	25·6	,,
Cadmium	25·6	,,
Bismuth	22·7	,,
Lead	9·86	,,
Zinc	50·60	,,
Silver	37·90	,,
Cast-iron	233·00	Clement.
Platinum	46·00	,,
Mercury	36·00	,,

The latent heat of water is 142·6, or in round numbers 143 units of heat, being the number of units of heat absorbed by ice during the process of melting; the amount of heat thus absorbed would have raised the same weight of water 143 degrees. Hence to melt one pound of ice requires as much heat as would raise 143 lbs. of water 1 degree Fahr.

The Temperature resulting from a Mixture of Water of different Temperatures may be found by this *Rule:* Divide the sum of the products of the weight in lbs. of each quantity of water by its temperature, by the weight of the mixture in lbs. *Example:* If 30 lbs. of water at 32°

Fahr. be added to 40 lbs. of water at 212°, what will be the temperature of the mixture? Then

30 lbs. × 32° = 960	Thermal units in the 1st quantity of water.
40 lbs. × 212° = 8480	Thermal units in the 2nd quantity of water.

$$9440 \div 40 \text{ lbs.} + 30 \text{ lbs.} = \frac{9440}{70} = 134^{\circ}{\cdot}85 \text{ Fahr.,}$$

the temperature of the mixture.

The Temperature resulting from a mixture of Ice and Water may be found as follows: To melt one pound of ice will absorb 143 thermal units, that being the latent heat of water. Then let

I = the weight of ice in lbs.
W = the weight of water in lbs.
T = the temperature of the water.

$$\text{The resulting temperature} = \frac{I\,(143-32)-(T \times W)}{I+W.}$$

Example: If 10 lbs. of ice be mixed with 10 lbs. of water at 212° Fahr., what is the resulting temperature? Then $143-32=111 \times 10=1110$ and $212 \times 10=2120-1110=1010 \div 20$ lbs. $=50^{\circ}{\cdot}5$ Fahr., the temperature of the water.

The Weight of Ice to be added to Water to Cool or lower its Temperature may be found as follows:—

Let T°=the temperature of the water to be cooled.
t°=the temperature the water requires to be cooled down to.
W=the weight of water in lbs. to be cooled.

$$\text{Weight of ice} = \frac{(T-t) \times W}{(143-32)+t.}$$

Example: How many pounds of ice must be mixed with 10 lbs. of water at 212° Fahr., to obtain water at a temperature of 50°·5 Fahr.? Then

$$212^{\circ}-50^{\circ}{\cdot}5=161{\cdot}5 \text{ and} = \frac{161{\cdot}5 \times 10}{(143-32)+50{\cdot}5} = \frac{1615}{161{\cdot}5} = 10 \text{ lbs. of ice.}$$

The Temperature resulting from mixing Mercury with water may be found as follows, when the temperature of the mercury is *less* than that of the water:—

The specific heat of water is 1.
The specific heat of mercury is ·033.
T=the temperature of the water.
t=the temperature of the mercury.

$$\text{The temperature of the mixture will be} = \frac{T-t}{1+{\cdot}033} + t.$$

Example: If 1 lb. of mercury at 52° Fahr. be placed with 1 lb. of water

212° Fahr., what is the resulting temperature? Then $\frac{212^\circ - 52^\circ}{1 + \cdot033} = \frac{160}{1\cdot033} =$ $154\cdot8 + 52^\circ = 206\cdot8^\circ$ Fahr., the temperature of the mixture.

When the temperature of the mercury is greater than that of the water, the temperature of the mixture may be found as follows:—Every 30·3 degrees given up by the mercury will only heat the water 1 degree, and the difference of the temperatures of the mercury and the water divided by $30\cdot3 + 1$, and added to the temperature of the water, will give the temperature of the mixture.

Example 1: If 1 lb. of mercury at 212° Fahr. be placed with 1 lb. of water at 52° Fahr., what is the resulting temperature?

Then $212 - 52 = 160$, and $\frac{160}{31\cdot3} = 5^\circ 11 + 52^\circ = 57^\circ\cdot11$ Fahr., the temperature of the mixture.

Example 2: If 1 lb. of mercury at 212° Fahr. be placed with 10 lbs. of water at 42° Fahr., what is the resulting temperature?

Then $212 - 42 = 170$, and $\frac{170}{31\cdot3 \times 10 \text{ lbs.}} = \cdot54 + 42 = 42^\circ\cdot54$ Fahr., the temperature of the mixture.

Example 3: How much mercury at a temperature of 212° Fahr. will be required to melt 20 lbs. of ice?

Then 143 latent heat of water × 20 lbs. = 2860, and

$$\frac{2860}{212^\circ \times \cdot033 \text{ specific heat of mercury}} = 408\cdot8 \text{ lbs. of mercury.}$$

Laws of Expansion of Metals by Heat.—Metals expand equally in all directions, only when of uniform homogeneous texture, free from laminations and impurities. The rate of expansion is not constant for each metal, but varies with its mixture; for instance, four different mixtures of gun-metal, heated to the same temperature, were found to expand respectively per degree of heat = ·000010461, ·000010576, ·000010645, ·000010783. The expansion of a metal is greatest when it is pure and homogeneous, less when it is fibrous or porous, and least when it is either in a burnt or rotten state. When a metal has coarse fibres, the expansion will be greater along than across its fibres.

Expansion is increased by rolling, or compressing the metal. as it closes the pores and makes the texture of the metal more uniform. The amount of force exerted by heat and cold in the expansion and contraction of a metal, is equal to that which would be required to stretch or compress it to the same extent by mechanical means.

The Expansion of Metals for every degree Fahr., of increase of temperature, is frequently taken at six parts in one million parts for cast iron, that is 6 inches in a length of one million inches: at 7 for wrought iron: 8 for steel not tempered: 9 for brass: 10 for tempered steel: 12 for tin and lead.

Example: If a bar of wrought iron, 10 feet long, be heated from 62° Fahr. to 212° Fahr. required the length due to expansion by heat.

Then $212° - 62° = 150$ degrees increase of temperature, and $150° \times 7 = 1050$ inches increase of length due to expansion in a length of one million inches, and $\frac{1050 \times 10 \text{ feet} \times 12 \text{ inches}}{1000000} = \cdot126$ inch the additional length of the bar due to expansion by heat.

Example 2: A multitubular boiler is 10 feet long, the plates are wrought-iron, the temperature of the bottom of the shell is 238° Fahr., and that of the remainder of the shell 308° Fahr. If the difference between the expansion of the top and bottom of the shell be apportioned at $\frac{1}{4}$th for compression on the top of the shell and $\frac{3}{4}$ for tensile strain on the bottom of the shell:—How much will the elongation be, in parts of the length of the boiler?

Then taking the temperature of the air outside the boiler at 58° Fahr.

The expansion of the top of the boiler will be

$$= \frac{(308° - 58°) \times 7 \times 120 \text{ inches long}}{1000000} = \cdot2100 \text{ inch.}$$

The expansion of the bottom of the boiler will be

$$= \frac{(238° - 58°) \times 7 \times 120 \text{ inches long}}{1000000} = \cdot1512 \text{ inch.}$$

Leaving a difference of $\cdot2100 - \cdot1512$ . . $= \cdot0588$ inch.

Then $\cdot0588 \div 4 = \cdot0147$, the allowance for the compression of the top-plates, and $\cdot0147 \times 3 = \cdot0441$, the allowance for the tensile strain on the bottom-plates.

Then $\cdot0441 + \cdot1512$, the expansion of the bottom plates $= \cdot1953$, the total expansion, and $\cdot1953 \div 120$ inches length of boiler $= \cdot00162$.

The expansion of a metal-pipe is sometimes employed as a means of automatically working the valve of a drain-pipe for draining condensation-water from steam-pipes. The valve-seat is placed at the bottom of the drain-pipe, and the valve is held by iron rods connected to a crossbar fixed on the drain-pipe above the valve. When the drain pipe contains steam, it expands and closes the valve; when it contains water, the pipe cools and recedes from the valve, and the water escapes. Taking steam at 212° Fahr., and condensation-water at $112° = 100°$ difference of temperature, a wrought-iron pipe 10 feet long would expand $= 120$ inches $\times 100° \times \cdot00000658 = \cdot0789$ inch.

Average Expansion of Substances by Heat in length and volume. The results of experiments by various authorities are given in Table 13, the use of which may be illustrated by the following examples:

Example 1: How much will a bar of average quality wrought-iron, 20 feet long, expand in length when heated 150 degrees?

Then 20 feet $\times$ 12 $=$ 240 inches $\times$ $\cdot00000658$ $\times$ 150 degrees $= \cdot23688$ inch, or nearly $\frac{1}{4}$ inch, making the bar 20 feet $0\frac{1}{4}$ inches long.

Example 2: How much will 64 cubic feet of oil expand when heated 100 degrees, and what would the volume of the oil be?

Then 64 cubic feet $\times$ $\cdot00044445$ $\times$ 100 degrees $= 2\cdot8444$ cubic feet the expansion, and the volume of the oil would be $64 + 2\cdot8444 = 66\cdot8444$ cubic feet.

Table 13.—Linear, Superficial, and Cubical Expansion of Substances, &c., by Heat, per Degree Fahrenheit, from 32° Fahrenheit.

Superficial expansion, or expansion in length and breadth, is twice the linear expansion; and cubical expansion or expansion in length, breadth, and depth, is three times the linear expansion.

Description of Substance.	Temperature. Fahr.°	Linear Expansion.	Superficial Expansion.	Cubical Expansion.
Permanent gases . .	32—212	·00069417	·00138834	·00208251
Water	392—572	·00018905	·00037810	·00056715
Water	212—392	·00017067	·00034134	·00051201
Water	32—212	·00008807	·00017614	·00026421
Alcohol and nitric acid .	32—212	·00015153	·00030306	·00045459
Oil	32—212	·00014815	·00029630	·00044445
Turpentine and ether .	32—212	·00012967	·00025934	·00038901
Sulphuric acid . . .	32—212	·00011112	·00022224	·00033336
Salt solution . .	32—212	·00009252	·00018504	·00027756
Zinc, hammered . .	32—212	·00001728	·00003456	·00051840
Zinc, cast	32—212	·00001636	·00003272	·00049080
Lead	32—212	·00001586	·00003172	·00004758
Tin, hammered . .	32—212	·00001508	·00003016	·00004524
Hard solder . . .	32—212	·00001436	·00002872	·00004308
White-metal . . .	32—212	·00001325	·00002650	·00003975
Tin, cast	32—212	·00001210	·00002420	·00003630
Compressed gun-metal .	32—212	·00001130	·00002260	·00003390
Silver, hammered . .	32—212	·00001118	·00002226	·00003354
Silver, pure . . .	32—212	·00001063	·00002120	·00003189
Copper	32—572	·00001095	·00002190	·00003285
Copper	32—212	·00000958	·00001816	·00002874
Brass-sheets and plates .	32—212	·00001078	·00002156	·00003234
Brass, cast . . .	32—212	·00001047	·00002094	·00003141
Phosphor-bronze . . .	32—212	·00001067	·00002134	·00003201
Gun-metal	32—212	·00001062	·00002124	·00003186
Gold, hammered . .	32—212	·00000837	·00001674	·00002511
Gold, pure	32—212	·00000821	·00001642	·00002463
Wrought-iron . . .	32—572	·00000892	·00001784	·00002676
Wrought-iron, average .	32—212	·00000658	·00001316	·00001974
Wrought-iron, best . .	32—212	·00000685	·00001370	·00002055
Cast-iron	32—212	·00000630	·00001260	·00001890
Malleable cast-iron . .	32—212	·00000636	·00001272	·00001908
Cast-steel	32—212	·00000615	·00001230	·00001845
Cast-steel castings .	32—212	·00000621	·00001242	·00001863
Hardened steel . .	32—212	·00000695	·00001390	·00002085
Compressed-steel . .	32—212	·00000650	·00001300	·00001950
Mild-steel boiler-plates .	32—212	·00000672	·00001344	·00002016
Mild-steel boiler-plates Siemens-Martin, best	32—450	·00000700	·00001400	·00002100
Roman cement . .	32—212	·00000800	·00001600	·00002400
Bismuth	32—212	·00000780	·00001560	·00002340

Table 13 *continued.*—EXPANSION OF SUBSTANCES, &c.

Description of Substance.	Temperature. Fahr.°	Linear Expansion.	Superficial Expansion.	Cubical Expansion.
Wire, brass . . .	32—212	·00001074	·00002148	·00003222
Wire, iron . . .	32—212	·00000745	·00001490	·00002235
Mercury . . .	32—212	·00003334	·00006668	·00010002
Glass	392—572	·00000665	·00001330	·00001995
Glass	212—392	·00000550	·00001100	·00001650
Glass	32—212	·00000482	·00000964	·00001446
Antimony . . .	32—212	·00000610	·00001220	·00001830
Palladium . . .	32—212	·00000560	·00001120	·00001680
Platinum . . .	32—572	·00000525	·00001050	·00001575
Platinum . . .	32—212	·00000500	·00001000	·00001500
Marble . . .	32—212	·00000620	·00001240	·00001860
Granite	32—212	·00000440	·00000880	·00001320
Kentish rag-stone .	32—212	·00000452	·00000904	·00001356
Bath stone . . .	32—212	·00000405	·00000810	·00001215
Stock-bricks . .	32—212	·00000306	·00000612	·00000918
Fire-bricks . . .	32—212	·00000235	·00000470	·00000705
Slate	32—212	·00000577	·00001154	·00001731
Gutta percha . . .	25— 60	·00008450	·00016900	·00025350
Ice	−17—+30	·00002857	·00005714	·00008571

Table 14.—CONTRACTION OF WROUGHT-IRON ON SUDDEN COOLING, FROM THE EXPERIMENTS OF WRIGHTSON, HOWE, AND OTHERS.

Size of Bar or Wire	Initial length of Bar or Wire.	Number of times quenched.	Percentage of Contraction.
	Inches.	Times.	Per cent.
Bar, $1\frac{1}{8}$ inches square . .	30·05	5	·76
Bar, $1\frac{1}{8}$ inches square . . .	30·05	15	2·26
Hoop, $1\frac{1}{8}$ inches wide . .	57·70	5	·61
Hoop, $1\frac{1}{8}$ inches wide . . .	57·70	20	2·25
Wire, ·022 inch thick . . .	71·00	20	·31
Wire, ·012 inch thick . . .	72·75	20	·09
Wire, ·009 inch thick . . .	92·75	5	·06
Bar, 1 inch square	36·00	1	·001
Bar, 1 inch square . . .	36·00	6	·005
Bar, 1 inch square . . .	36·00	12	·008
Bar, 1 inch square . . .	36·00	20	·012
Tire, 6 feet diameter . . .	226·25	5	·009
Tire, 6 feet diameter . . .	226·25	10	·015
Tire, 6 feet diameter . . .	226·25	15	·022

Contraction of Wrought-iron by Sudden Cooling.—If a bar of wrought-iron be suddenly cooled from a bright-red heat, the contraction which occurs is considerably greater than the expansion previously caused by heating the bar, so that its final length is considerably less than its initial length. If the heating and cooling of the bar be repeated, say 30 times, a further permanent contraction occurs at each successive cooling. Thick bars contract considerably more than thin ones, as will be seen from the results of experiments given in the previous Table, showing that bars $1\frac{1}{8}$ inch square contracted about seven times as much as wire of less than $\frac{1}{32}$ inch in thickness.

COAL AND OTHER FUELS.

Coal is the product of the decomposition of vegetable matter, and is composed of carbon, hydrogen, oxygen, nitrogen, and sulphur. Its impurities consist of silica, alumina, magnesia, lime, oxide of iron and earthy matter. The more carbon the coal contains the more heat it will yield. Coal deteriorates rapidly by exposure to the atmosphere, owing to the gradual escape of the constituent gases.

Weight of Coal.—Coal in its natural bed weighs about 80 lbs. per cubic foot, and in its broken or loose state it averages 48 lbs. per cubic foot, small or screened coals weigh 43 lbs. per cubic foot. Hard coals contain from 5 to 9 per cent. of volatile matter, and soft coals from 15 to 35 per cent. of volatile matter. Those coals contain the most volatile matter which swell up and present a volcanic appearance in burning. All coals contain a certain percentage of water; and they yield in burning from $1\frac{1}{2}$ to 10 per cent. of ash.

Anthracite Coal is of homogeneous structure and jet-black colour. It yields intense heat, and contains from 90 to 95 per cent. of carbon, and from 5 to 9 per cent. of volatile matters. It ignites with difficulty, makes hardly any smoke, and emits no sulphurous fumes; it burns with a short feeble flame, falls to pieces much in burning, and does not cake; specific gravity, 1·36 to 1·60.

Bituminous Coals are lighter than anthracite, and consist of several kinds, viz., clear-burning, flaming, and fuliginous coals.

Clear-burning Coals are fragile, ignite with difficulty, and burn slowly with a short, clear, bluish flame, and contain from 15 to 25 per cent. of volatile matter. Some bituminous coals, on being heated, cake, and others swell up and fuse.

Flaming Coals are black and glossy, ignite with difficulty, and burn away rapidly with a long white flame; they contain from 20 to 35 per cent. of volatile matter.

Fuliginous Coals ignite easily, and burn away rapidly with a long yellow smoky flame, and contain upwards of 35 per cent. of volatile matter.

Semi-bituminous Coals are of dull-black colour, with from 10 to 20 per cent. of volatile matter, do not cake, and burn with greater flame than anthracite.

Bituminous or Gaseous Coals are hard and strong, of dull lustre and brownish-black colour.

Newcastle Coal is a flaming coal, it burns faster and makes more smoke than Welsh coal. In order to obtain the same evaporative economy, shorter fire-grates are required in burning Newcastle or long flaming coal, than for a coal containing a less quantity of volatile ingredients, such as Welsh coal, because the generation of heat is spread over a greater length of surface by the long flame of the flaming coal, than the short flame of Welsh coal. The furnace-bars should be spaced wider apart for Welsh coal than for Newcastle coal, as it requires more air.

Composition of Coal.—The average composition of several kinds of coal is given in the following Table :—

Table 15.—Composition, Weight, &c , of Coal.

Constituents, &c.	Anthracite Coal.	Aberdare Coal.	Welsh Coal.	Newcastle Coal.	Lancashire Coal.	Derbyshire Coal.	Yorkshire Coal.	Scotch Coal.
Carbon, per cent.	92·00	88·28	86·26	83·60	80·70	80·00	79·90	79·50
Hydrogen	3·80	4·24	4·66	5·28	5·50	4·85	4·83	5·58
Oxygen	1·00	1·65	2·60	4·65	8.48	9·90	10·10	8·33
Nitrogen	1·00	1·66	1·45	1·22	1·12	1·35	1·40	1·14
Sulphur	·70	·91	1·77	1·25	1·50	1·10	1·00	1·45
Ash	1·50	3·26	3·26	4·00	2·70	2·30	2·77	4·00
Specific gravity	1·37	1·32	1·31	1·25	1·27	1·30	1·29	1·26
Weight of a cubic foot in lbs. in solid state	85·60	82·50	81·90	78·10	79·40	81·20	80·60	78·70
Weight of a cubic yard in tons, in solid state	1·031	·994	·987	·941	·957	·978	·972	·948
Average bulk of one ton, heaped, in cubic feet	41	42	43	46	45	44	44·5	45·5
How it burns	With difficulty.	Slowly.	Slowly.	Quickly	Quickly	Quickly	Quickly and cakes.	Quickly
Draught required	Quick.	Quick.	Quick.	Ordinary.	Ordinary.	Ordinary.	Brisk.	Ordinary.
Quantity of smoke	None.	Scarcely any.	Very little.	Large.	Large.	Large.	Large.	Very large.

The Specific Gravity of Coal may be ascertained as follows : suspend from the under side of the pan of a weighing-balance a piece of coal by means of a horse hair, and weigh it both in and out of water, divide its weight in the air by the loss of weight when in the water, and the quotient will be the specific gravity.

Example : A piece of coal weighs . . . . 561 grains
Its loss of weight when weighed in water is . . . 445 grains
Then $561 \div 445 = 1{\cdot}26$ specific gravity of the coal compared with water.

Table 16.—Weight of Coal.

Specific Gravity.	Weight of Coal in the Natural Bed, per Inch thick per Acre.	Weight of a Cubic Foot of Loose Coal.	
		Large Coal.	Small Coal.
	Tons.	lbs.	lbs.
1·10	111·25	43	38
1·15	116·53	45	39
1·20	121·76	47	41
1·25	126·84	49	43
1·30	131·92	51	44
1·35	136·59	53	46
1·40	141·95	55	48
1·45	146·75	57	49
1·50	152·00	59	51

The Weight of Coal in its natural bed, per inch in depth per acre—allowing for loss in working—available in a given area of seam is approximately equal to the weight of the same surface and depth of rainfall, or about 100 tons per acre; and the weight in tons of an acre of coal 1 inch thick is approximately equal to 100 times its specific gravity.

The Heating Power of Coals, or their average evaporative power in pounds of water from 212° converted into steam by 1 lb. of coal, is as follows:—

Anthracite coal	9·10
Bituminous clear-burning coal	9·16
Bituminous flaming coal	8·10
Bituminous fuliginous coal	8·10
Semi-bituminous coal	9·00
Gaseous coal	6·20

Warlich's Patent Fuel is a mixture of small coal 1 ton, tar 10 gallons, or 242 lbs., moulded into blocks and baked at a temperature of 800° Fahr. for ten hours; the blocks in baking lose 5 per cent. of their weight.

Wylam's Fuel is a mixture of small coal or slack 92 parts, finely ground dry pitch 8 parts. The mixture is forced with an Archimedean screw through a retort maintained at a dull red heat, and afterwards moulded under pressure into blocks.

Mezaline's Fuel is a mixture of finely ground small coal and pitch; during the process of grinding it is softened by superheated steam. Fuel prepared in this way loses 4 per cent. of its weight when exposed to a high temperature.

Barker's Fuel is a mixture of one ton of small coal, thirty gallons of farina-mucilage—consisting of farina 1 part, water 4 parts, and a small quantity of carbolic acid—and a small quantity of powdered pitch. The mixture is baked for 9 hours at a temperature of 300° Fahr.

Holland's Fuel is a mixture of small coal, lime and cement; hence it makes when burnt a considerable quantity of ash.

Penrose and Richard's Patent Coke is a mixture of 60 per cent. of anthracite, 35 per cent. of bituminous coal, and 5 per cent. of pitch, ground and mixed together; the yield of coke is 80 per cent. of the weight of the charge. This coke is very hard, of steel-grey colour, and is about 23 per cent. heavier than Welsh coke.

Weight and Bulk of Coal.—The weight of a cubic foot of heaped coal varies from 44 to 58 lbs. The average weight of a variety of different coals was found to be 50 lbs. per cubic foot.

Weight of Coke.—The weight of a cubic foot of heaped coke is 30 lbs.

The Bulk of one ton of Heaped Coal varies from 37 to 50 cubic feet. The average bulk is 45 cubic feet per ton.

Coal of medium density averages $1\frac{1}{2}$ cubic yards per ton heaped.

Peat of medium density averages 8 cubic yards per ton heaped.

The bulk of one ton of heaped coke averages 80 cubic feet.

The Admiralty allowance for the Bulk of Coal equals 40 cubic feet per ton of 2240 lbs., and 48 cubic feet per ton of 2700 lbs.

Patent Fuels are lighter than coal. The average weight is 74 lbs. per cubic foot solid, and 65 lbs. per cubic foot heaped. The bulk of one ton of patent fuel heaped averages 35 cubic feet.

Petroleum-Refuse Fuel requires a space of 41 cubic feet to contain one ton.

The stowage-capacity of Patent Fuels, Liquid Fuels, and Stores is given in the following table:—

Table 17.—Weight and Specific Gravity of Patent Fuels, Liquid Fuels, and Stores.

Description.	Weight per Cubic Foot.	Weight per Cubic Yard.	Specific Gravity.
	Lbs.	Tons.	
Wylam's patent fuel	68·8	.829	1·10
Lyon's	70·6	·851	1·13
Bell's	71·2	·858	1·14
Livingstone's	71·2	·858	1·14
Warlich's	71·9	·867	1·15
Holland & Green's . . .	81·2	·978	1·30
Vinegar distilled weighs . .	68	...	1·009
Milk	64·3	...	1·030
Fresh water	62·425	...	1·000
Wine	62	...	·993
Tallow	59	...	·950
Linseed oil	58·7	...	·940
Ice	58	...	·930
Rape seed oil and whale oil . .	57·4	...	·920
Alcohol, proof strength . .	57·4	...	·920
Olive oil	57·1	...	·915
Gunpowder, average	56·7	...	·910
Petroleum	54·9	...	·880
Turpentine	54·2	...	·870
Naphtha	53·1	...	·850
Cotton waste	11	...	·170

Dry Pine Wood requires 110 cubic feet to stow one ton, and weighs 21 lbs. per cubic foot.

Bulk of Gunpowder, cubical contents of 100 lbs. weight of gunpowder = 1·774 cubic feet, or 3064 cubic inches.

The Quantity of Coal a Bunker will contain may be found by this *Rule:* Multiply the length, width, and depth, in feet together, and the product will be the contents of the bunker in cubic feet, then divide by 45, the number of cubic feet in a ton of coal of average bulk, and the quotient will be the quantity in tons which the bunker will contain. The section of a coal-bunker is shown in Fig. 16.

Fig. 16.—Coal-bunker.

When the cubical contents in feet and two dimensions only are given to find the third—

The Length of a Bunker may be found by dividing the cubical contents in feet by the product of the breadth and height in feet.

The Width of a Bunker may be found by dividing the cubical contents in feet by the product of the length and height in feet.

The Height of a Bunker may be found by dividing the cubical contents in feet by the product of the length and width in feet.

The Cubical Contents or Space required in Cubic Feet to contain Coal may be found by multiplying the number of tons by 45.

Example 1: Required the quantity of coals contained in a coal bunker 16 feet long, 8 feet wide, and 7 feet 6 inches high.

Then $16 \times 8 \times 7 \cdot 5 = 960 \div 45 = 21$ tons, 6 cwt., 2 qr., 12 lb. of coal.

Example 2: Required the quantity of coals contained in a coal-bunker 20 feet long, 10 feet deep, 7 feet wide at the top, and 5 feet wide at the bottom. First find the mean width by adding the two widths together and dividing by two. Then $7 + 5 = 12 \div 2 = 6$ the mean width and $20 \times 10 \times 6 = 1200 \div 45 = 26$ tons, 13 cwt., 0 qr., 23 lbs. of coal.

Example 3: A coal-bunker is required 8 feet wide and 6 feet deep, what length should it be to hold 20 tons of coal? To find the length divide the cubical contents by the product of the width and height. Then $8 \times 6 = 48$, and 20 tons $\times$ 45 cubic feet $= 900 \div 48 = 18$ feet 9 inches, the length required.

Practical Analysis of Coal.*—The sample should represent an average of the whole quantity—no less than one pound can be used. This must first be ground in an iron mortar and sifted through a fine sieve. What remains must again be ground and sifted until all passes through.

Estimation of Water.—Weigh off three grammes of the powdered sample, and heat in an air-bath at 120°C. for twenty minutes; then weigh (after cooling). Afterwards heat up again, weighing every ten minutes until the weight is constant. Then the loss in weight = water.

Bituminous coals increase in weight by oxidation during the heating, so too great care cannot be exercised in this part of the analysis.

Unless the percentage of water is specially desired it need not be regarded. Under all circumstances it is best to calculate all results in the dry material, for which purpose heat up the sample for forty minutes and place in a desiccator to cool.

Volatile and Combustible Material.—Place the dry sample in a weighing flask. Deliver from it 0·500g. coal into a porcelain crucible, and heat for ten minutes over the strongest Bunsen burner, the crucible being kept covered all the time. Cool and weigh.

Loss = volatile and combustible material + $\frac{1}{2}$ the sulphur of the FeS_2.

Fixed Carbon.—Take from the weighing flask about 0·500gr. and place it in a tarred, open platinum dish. Heat gently at first over Bunsen burner, then more strongly, and finally at highest heat, until all the carbon is burned off.

Loss = volatile matter + $\frac{1}{2}$ S + F. carbon. The difference is the fixed carbon. The residue = ash.

Examine ash closely as to colour and texture.

Sulphur.—Oxidize 2g. of coal with 20c. c. fuming nitric acid and $\frac{1}{2}$g. chlorate of potash in a porcelain dish. Cover with inverted funnel for two hours at a very low heat, bring on to a filter, wash with boiling water, and precipitate with $BaCl_2$. Wash the precipitate in acetate of ammonia by

* See an article by Mr. A. K. Glover, in the "Scientific American."

boiling up with it and decanting several times. From the $BaSo_4$ calculate the sulphur.

Estimation of Total Carbon and Available Hydrogen.—Employ a hard, infusible glass combustion tube 40cm. long and about 15mm. in diameter, drawn to a point. Fill one-third the length with *dry fused chromate of lead finely powdered.* Then by means of a small delivery tube insert 0·200g. of coal into the combustion tube. Mix well the coal and chromate by means of a wire stirrer, and finally add more chromate, stirring still, until the tube is filled to the extent of 35cm. At the anterior end place a thick coil of copper gauze, to decompose the nitrous oxide that may be formed. Then attach a chloride of calcium tube, carrying two bulbs, to the tube by rubber cork. To the chloride of calcium tube attach a U tube containing nitrate of lead, to catch the sulphurous acid formed, and lastly the potash bulbs filled with strong caustic potash. Proceed carefully as in any other delicate organic analysis, keeping the copper gauze at a bright red heat. The posterior part of the chromate should be heated the hottest. From the increase in weight of the potash bulbs calculate the total carbon (fixed and volatile) and from the $CaCl_2$ tube the hydrogen.

Never use more than 0·250g. of coal. In coals carrying as high as 20 per cent. of volatile material 0·100g. is sufficient. Too much care in using *dry* chromate of lead cannot be exercised, otherwise too much hydrogen will be set down for the coal.

Heating Power of Carbon and Hydrogen.—Taking as a *heat-unit* the heat necessary to raise the temperature of one pound of water through one degree centigrade, one pound of carbon (C) in burning to carbonic acid (CO_2) disengages 8080 thermal-units: and one pound of hydrogen (H) in burning to water (H_2O) disengages 34,460 thermal-units. Hence the following approximate *Rule* for finding the heating power. For a more exact *Rule* see p. 57.

Valuation in Heating Power.—Multiply the percentage of total carbon by 8080, and the available hydrogen by 34,460. Divide each result by 100, and add together.

Example: Required the heating power of coal containing 80 per cent. carbon and 5 per cent. hydrogen.

$$\text{Then } \frac{(80 \times 8080) + (5 \times 34{,}460)}{100}$$

$$\begin{array}{r} 6464{\cdot}00 \\ 1723{\cdot}00 \\ \hline 8187{\cdot}00 \end{array} = \text{calorific or heating power.}$$

If the English unit of heat, or the heat necessary to raise the temperature of one pound of water through one degree Fahr., be taken as the heat unit instead of that given above, then the multiplier for the above *Rule* will be 14,500 for the carbon, and 62,535 for the hydrogen

As a further means of comparison, it is often advisable to record the amount of coke yielded by a given sample. The coke is the difference between 100 per cent. and the volatile matter. Beyond the above analysis nothing is wanted to enable a right judgment to be formed as to the value of a sample of coal.

COMBUSTION.

Combustion is the chemical combination of substances—principally carbon and hydrogen—with oxygen, the supporter of combustion, attended with the evolution of heat and light.

The chief constituents of coal are carbon and hydrogen, neither of which can be consumed while they remain united, the carbon or carbonaceous portion is combustible only in its solid state, and the hydrogen, or bituminous portion is combustible only in the gaseous state.

Carbon is perfectly consumed when combined with 2·666 parts of oxygen to form carbonic acid gas, and partly consumed when combined with one-half that quantity of oxygen to form carbonic oxide gas or smoke.

Hydrogen is the lightest body known: it is $14\frac{1}{2}$ times lighter than air. It is the main element in the gas evolved from burning coal, and its combustion produces flame. Hydrogen in burning combines with eight parts of oxygen to form nine parts of water. Bituminous coal contains from 5 to 6 per cent. of hydrogen, and as each pound of hydrogen in combustion combines with 8 lbs. of atmospheric oxygen, it produces 9 lbs. of water in the form of steam; and each hundred-weight of coal produces about 50 lbs. of water.

The gaseous products of the complete combustion of 1 lb. of fuel, are as follows:—

1 lb. of carbon unites with 2·66 lbs. of oxygen to form 3·66 lbs. of carbonic acid.

1 lb. of hydrogen unites with 8 lbs. of oxygen to form 9 lbs. of steam.

1 lb. of sulphur unites with 1 lb. of oxygen to form 2 lbs. of sulphurous acid.

The sulphur may be omitted in ordinary calculations.

When Heat is applied to Bituminous Coal the first result is its absorption by the coal and the disengagement of gas, from which flame is exclusively derivable. The constituents of this gas are hydrogen and carbon, and the unions form two gases, viz., carburetted hydrogen, and bi-carburetted hydrogen commonly called olefiant gas. These inflammable gases require certain quantities of atmospheric air to effect their combustion, and they are only combustible in proportion to the degree of mixture and union which is effected between them and the oxygen of the air. The chemical assimilation of the volatilised coal-elements is retarded by contact with cold surfaces, and assisted by hot surfaces. Air can be heated instantaneously, and when the heating-surfaces of a combustion-chamber are highly heated, the air is instantaneously heated to the same degree of tenuity as the hot gases, they therefore instantly unite and produce intense heat. When a sufficient supply of air is not provided to the furnace, the result is waste of fuel from the escape of unconsumed fuel, in the gaseous and smoky state, to the chimney. If the supply of air be correctly regulated, there will be perfect combustion producing carbonic acid; but if the supply of air be deficient, the combustion will be imperfect and carbonic oxide will be produced.

The Heat developed by Fuels in burning, may be ascertained by calculation when their chemical composition is known. The chemical composition of various combustibles is given in the following Table.

Table 18.—CHEMICAL COMPOSITION OF COMBUSTIBLES.

Description.	Carbon.	Hydrogen.	Oxygen.	Ash.
Coke, average of 25 analyses	·935	...	...	·065
Charcoal, from wood, average	·925	...	...	·075
Oil of turpentine	·882	·118	...	...
Pine-wood oil	·867	·106	·027	...
Paraffin, or petroleum-oil	·855	·145	...	...
Petroleum, average	·840	·110	·050	...
Creosote, or tar-refuse	·825	·100	·075	...
Spermaceti	·817	·123	·060	...
Beeswax	·815	·139	·046	...
Coal, average	·804	·052	·079	·041
Schist oil	·800	·117	·083	...
Resin	·791	·102	·107	...
Asphalte	·790	·095	·087	·028
Tallow	·788	·118	·094	...
Sperm oil	·787	·110	·103	...
Olive oil	·770	·134	·096	...
Lignite, bituminous	·749	·075	·130	·046
Lignite, perfect	·690	·050	·202	·058
Lignite, imperfect	·602	·053	·290	·055
Peat, perfectly dry	·600	·060	·300	·040
Peat, dry, average	·570	·060	·320	·050
Peat, dry, medium quality	·460	·050	·437	·053
Alcohol	·520	·135	·345	...
Wood-pimps, or bundles of twigs, dry	·510	·060	·400	·030
Wood, perfectly dry, average	·500	·060	·418	·022
Wood-bark, dry	·477	·063	·432	·028
Wood, pine, dry	·445	·055	·485	·015
Wood, oak, dry	·430	·053	·496	·021
Sawdust, dry	·410	·043	·531	·016
Straw, barley, dry	·380	·054	·510	·056
Straw, wheat, dry	·360	·050	·540	·050
Straw, oats, dry	·348	·050	·557	·045

Heating-Power of Carbon.—One lb. of carbon burning to carbonic acid develops 14500 units of heat according to Peclet, and 12906 units of heat according to Dulong.

One lb. of hydrogen develops 62535 units of heat; the heat developed by hydrogen is $62535 \div 14500 = 4{\cdot}31$ times as great as that of carbon.

One lb. of hydrogen requires 8 lbs. of oxygen.

Atmospheric air contains 20 per cent. of oxygen.

The Heat developed by a Combustible, or its Calorific Power, may be calculated from its chemical constituents by this formula:—

$$\text{Units of heat} = \left(14500 \times \text{percentage of carbon}\right) + \left\{62535 \times \left(\%\ \text{hydrogen} - \frac{\%\ \text{oxygen}}{8}\right)\right\}.$$

Coke contains ·935 carbon, but no hydrogen or oxygen as will be seen from Table 18. Therefore coke in burning will develop 14500 × ·935 = 13557 units of heat per lb. of coke; this is its maximum calorific power.

The Effect of Oxygen in a Combustible containing hydrogen, is to diminish its heating power. The oxygen is combined with the hydrogen in the ratio of 8 to 1 forming water instead of leaving free the heat which the hydrogen gives off in burning. When the quantity of oxygen in a combustible is less than that ratio, it combines with one-eighth of its weight, and the excess of hydrogen, above that neutralized by the oxygen, yields its proportion of heat, which may be calculated in the following manner:—

Heating Power of Peat.—As oxygen requires one-eighth part of its weight of hydrogen, the oxygen in peat will require one-eighth of its hydrogen or ·320÷8 = ·04 hydrogen, the hydrogen in the peat is ·06; therefore, the excess of hydrogen available for yielding heat will be ·06 — ·04 = ·02, and the heat this fuel will yield will be as follows:—

$$\begin{aligned} \text{Peat average, carbon} &= {\cdot}570 \times 14500 = 8265 \text{ units} \\ \text{Hydrogen in excess} &= {\cdot}02 \times 62535 = 1250 \text{ units} \\ &\qquad\qquad \overline{9515} \text{ units of heat} \end{aligned}$$

per lb. of peat.

Heating Power of Coal.—Coal composed of ·804 carbon: ·052 hydrogen: ·079 oxygen, will yield

$$\begin{aligned} &\text{Carbon} && {\cdot}804 \times 14500 = 11658 \text{ units} \\ &\text{Hydrogen} \left(\overset{\text{Hydrogen.}}{{\cdot}052} - \overset{\text{Oxygen.}}{\frac{{\cdot}079}{8}}\right) && = {\cdot}0422 \times 62535 = 2639 \text{ units} \\ & && \qquad\qquad \overline{14297} \text{ units of heat} \end{aligned}$$

per lb. of coal: its maximum calorific power.

Heating Power of Sawdust.—Sawdust in burning requires large fire-grate surface, the grate-bars should be placed low, and thin grate-bars, having narrow spaces between them, should be used. A thick layer of sawdust should be kept on the bars which will develop the following quantity of heat:—

Sawdust when in a dry state, on an average is composed of ·410 carbon; and 1 lb. of hydrogen for each 8 lbs. of oxygen; and as the hydrogen is thus neutralised by the oxygen, it develops no heat. Hence sawdust will yield: carbon ·410 × 14500 = 5945 units of heat per lb. of sawdust, its maximum calorific power.

Heating Power of Liquid Fuel.—Petroleum-oil is largely used for this purpose.

Paraffin or petroleum-oil, composed of ·885 carbon and ·145 hydrogen, will yield—

$$\begin{aligned} \text{Carbon} \quad {\cdot}885 \times 14500 &= 12832 \text{ units} \\ \text{Hydrogen} \ {\cdot}145 \times 62535 &= 9067 \text{ units} \\ &\ \ \overline{21899} \text{ units of heat per lb. of oil,} \end{aligned}$$

its maximum calorific power.

The heating power of petroleum, it will be seen from the above, is more than 50 per cent. greater than that of coal.

Creosote-oil, or residue from the distillation of tar, is also used as liquid fuel. Its composition is given in Table 18; its calorific power is less than that of petroleum-oil. Creosote-oil is liable to crystallize at temperatures below 120° Fahr., and it is necessary to place a small steam-pipe inside the pipe through which the oil flows when using it as liquid fuel.

Gaseous Fuel.—The composition of average coal-gas is as follows:—

Hydrogen	51·00
Marsh-gas	35·30
Carbon oxide	7·58
Olefines	3·55
Nitrogen	2·27
Oxygen	·30
	100·00

The Maximum Heating-Power of Average Coal-Gas has been found by experiment to be 700 heat-units per cubic foot of gas. The heating power of ordinary 16-candle-power coal-gas is 630 units of heat per cubic foot. As 30 cubic feet of coal-gas at 62° Fahr. = 1 lb., the maximum heating power of 1 lb. of coal-gas is $700 \times 30 = 21000$ heat-units, or more than 50 per cent. greater than that of coal of average quality.

Table 19.—Quantity of Gas in Cubic Feet obtained from Coal.

Description of Coal.	Specific gravity of Gas.	Cubic feet of Gas obtained per ton of Coal.	Weight in lbs. of Gas per ton of Coal.
Boghead	·752	15000	866
Capeldrae	·577	14400	638
Lesmahago	·618	13200	627
Arniston	·626	12600	606
Newcastle	·475	11648	423
Welsh	·737	11424	645
Pelaw	·444	11424	389
Pelton	·437	11424	387
Bickerstaff, Liverpool	·475	11424	415
Wigan	·528	11400	461
Garesfield	·398	10500	321
Powell	·459	10165	357
Forest of Dean	·360	10133	279
South Staffordshire	·320	9600	235
Derby's Deep Main	·424	9400	308
Derby's Soft Coal	·528	7500	303
Leeds	·530	6500	264

The volume of gas in cubic feet may be reduced to weight in lbs. by multiplying the volume by the specific gravity of the gas and by the constant number ·076.

One Ton of Crude Petroleum will yield 21000 cubic feet of good gas.

Pintsch-Gas, or Gas from Oil, is successfully used for lighting railway carriages and for purposes where coal-gas is not available.* It is made by decomposing petroleum or shale-oil, and is compressed to about 10 atmospheres. During compression, liquid is deposited in a chamber immediately attached to the pumps, and to a much larger extent in the reservoir in which the gas is stored, and from which it is delivered into the iron recipients (drums) attached to the railway carriages. This liquid is commonly called "hydrocarbon," and for convenience it is designated "pump hydrocarbon" and "reservoir hydrocarbon." Notwithstanding the nature of the material used in making oil-gas, the "hydrocarbon" is practically free from paraffins, containing but traces of hydrocarbons insoluble in sulphuric acid; it essentially consists of benzene and toluene, mixed with hydrocarbons of the C_nH_{2n} and C_nH_{2n-2} series. The "reservoir hydrocarbon" especially is saturated with gas, and on passing this into bromine, a solid bromide, of the composition $C_4H_6Br_4$, is obtained, which melts at 116° C, and is but slightly volatile. It seldom contains less than about 50 per cent. of benzene and toluene. The "pump hydrocarbon" differs from that deposited in the reservoir, only in being richer in the less volatile constituents.

In the Pintsch System of Manufacturing Oil-Gas, two cast-iron D-shaped retorts are set one above the other; the largest size used being 6 feet 4 inches long, 10 inches wide, and $9\frac{1}{4}$ inches deep. The oil is run into the upper retort at one end, falling upon an iron-tray, which is loosely fitted into the retort; and to complete the decomposition, the vapours are passed through the second lower retort. The temperature at which the retorts are worked is very high—a bright cherry-red. The oil is run in at the rate of about $12\frac{1}{2}$ gallons per hour, and about 80 cubic feet of gas is yielded per gallon of oil.

In the Keith System of Manufacturing Oil-Gas, the retort is generally 6 feet long, 5 inches broad, and 10 inches deep, and is constricted in the middle; it is made shallower in the middle and proportionally broader, so that the sectional area is the same as at the end. The oil is caused to flow down an inclined trough, so that it strikes the retort near the constriction where the temperature is highest. It is stated that from 100 to 150 cubic feet of gas may be obtained from 1 gallon of oil, according to the quality of the latter; and with 12 retorts 2000 cubic feet of gas can be produced per hour, of 50-candle-power. The results depend upon the quality of the oil, and the manner in which the decomposition is effected.

GAS ENGINES AND HOT-AIR ENGINES.

Gas Engines possess many advantages over small steam engines, and compare favourably with them as regards the cost of fuel. The "Otto" gas engine, shown in Fig. 17, is single-acting, the cylinder being open at the front end. The first forward stroke of the engine draws an explosive

* See a Paper by Professor Armstrong, in the "Journal of the Society of Chemical Industry."

mixture of air and gas into the cylinder, which, on the return of the piston, or first inward stroke, is compressed to about one-third its volume. At the beginning of the second out-stroke, the compressed charge is ignited, and the expanding gases propel the piston to the end of the stroke, the products of combustion being expelled at the second in-stroke. The tested consumption of coal gas by the "Otto" gas engine, when new, may be averaged

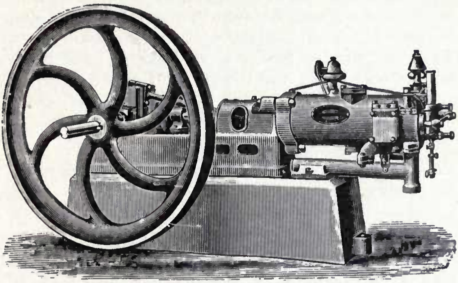


Fig. 17.—"Otto" horizontal gas-engine.

at from 18 to 22 cubic feet per indicated horse-power per hour for the large sizes, to 25 cubic feet per indicated horse-power per hour for the smaller sizes. At intermittent hoisting work, using coal-gas costing 3*s*. per 1000 cubic feet, the cost of working a $3\frac{1}{2}$ horse-power engine, taking an average of three months' consumption of gas, has been found to be as low as tenpence per day. The cylinder of the "Otto" gas engine is shown in section at Fig. 18. The slide is shown in the position when it is about

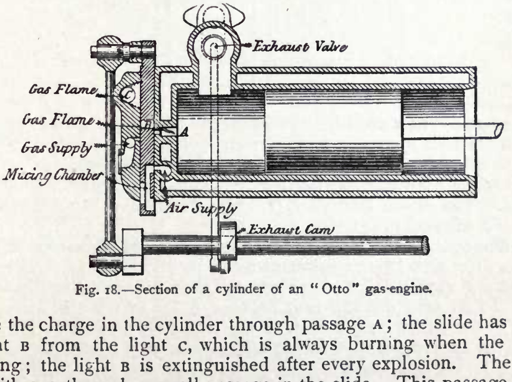


Fig. 18.—Section of a cylinder of an "Otto" gas-engine.

to ignite the charge in the cylinder through passage A; the slide has carried the light B from the light C, which is always burning when the engine is working; the light B is extinguished after every explosion. The light B is fed with gas through a small passage in the slide. This passage will be closed just before light B reaches chamber A, and it ignites the charge.

In careful tests of a single "Otto" gas engine of 30 indicated horse-power, the total consumption of fuel was found to be only 1·2 lb. per indicated horse-power, and 1·5 lb. per brake horse-power per hour.

Dowson's Water-gas is used for large gas engines. It is very much cheaper than coal-gas, its cost being only about three pence per thousand cubic feet; in an "Otto" gas engine using this gas, the consumption is as low as 1·3 lb. of anthracite coal per indicated horse-power per hour. An analysis of this gas by Professor Foster is given in the following Table, which also contains for comparison an analysis of ordinary coal-gas of 16 candle-power, from which it will be seen that coal-gas has nearly four times the calorific power of the other, the comparative explosive force of the two gases in atmospheric air being as 3·8 : 1, that is, the coal-gas has 3·8 times the energy of the other.

Table 20.—Composition and Calorific Power of Dowson's Water-Gas, and of ordinary 16 Candle-power Coal-Gas.

Composition of the Gas.	Ordinary Coal-Gas, 16 Candle-Power.			Dowson's Generator-Gas, or Water-Gas.		
	Volume per Cent. at 0 deg. Cent. and 760 mm.	Weight in Grammes of 100 Litres.	Calorific Power of 100 Litres.	Volume per Cent. at 0 deg. Cent. and 760 mm.	Weight in Grammes of 100 Litres.	Calorific Power of 100 Litres.
Hydrogen	51·81	4·63	159,559	18·73	1·67	57,689
Marsh gas	35·25	25·20	329,187	0·31	0·22	2,899
Olefiant gas	3·53	4·41	52,664	0·31	0·38	4,633
Carbonic oxide	8·95	11·20	27,854	25·07	31·36	77,992
,, acid	...	...	...	6·57	12·91	
Oxygen	0·08	0·11	...	0·03	0·04	
Nitrogen	0·38	0·47	...	48·98	61·27	
	100·00	46·02	569,264	100·00	107·85	143,213

Water-gas is now used for a great variety of purposes, such as singeing yarns and fabrics, type-founding, varnish making, heating water, cooking, baking bread, &c., and in all cases it is found that by allowing about *four volumes of the generator gas* for one of coal gas, the same heating effect is obtained. The above theoretical calculations of the calorific power are therefore confirmed in practice.

The Pressure produced by the Explosion of Gaseous Mixtures has been determined by experiments made by Mr. D. Clerk,* the results of which afford useful data for gas engines, and are given in the following Table. In no case did the heat accounted for by the explosion-pressure amount to more than 77 per cent. of the total heat present as inflammable gas; in the majority of cases it was a little over 50 per cent.

* See a Paper read before the Institution of Civil Engineers, by Mr. D. Clerk, F.C.S., "On the Explosion of Homogeneous Gaseous Mixtures."

Table 21.—PRESSURES PRODUCED BY THE EXPLOSION OF MIXTURES OF INFLAMMABLE GASES WITH ATMOSPHERIC AIR.

OLDHAM COAL-GAS AND AIR MIXTURES.
Average Temperature of Gases before Ignition taken at 17 degrees Centigrade.
Pressure, atmospheric (14·7 lbs.).

Proportion of Gas by Volume.	Maximum pressure in lbs. per Square Inch above the Atmosphere.	Maximum Temperature, Centigrade.	Time of Explosions.
	lbs.	Degrees.	Seconds.
$\frac{1}{15}$	40	806	·45
$\frac{1}{14}$	51·5	1033	·31
$\frac{1}{13}$	60	1202	·24
$\frac{1}{12}$	61	1220	·17
$\frac{1}{10}$	78	1557	·08
$\frac{1}{8}$	87	1733	·06
$\frac{1}{7}$	90	1792	·04
$\frac{1}{6}$	91	1812	·05
$\frac{1}{5}$	80	1505	·16

GLASGOW COAL-GAS AND AIR MIXTURES.
Temperature of Gas before Ignition 18 degrees Centigrade.
Pressure, atmospheric (14·7 lbs.).

Proportion of Gas by Volume.	Mean pressure in lbs. per Square Inch above the Atmosphere.	Maximum Temperature, Centigrade.	Time of Explosions.
	lbs.	Degrees.	Seconds.
$\frac{1}{14}$	52	1047	·28
$\frac{1}{12}$	63	1265	·18
$\frac{1}{10}$	69	1384	·13
$\frac{1}{8}$	89	1780	·07
$\frac{1}{6}$	96	1918	·05

Hydrogen and Air Mixtures. Temperature of Gases before Ignition, 16 degrees Centigrade.
Pressure, atmospheric (14·7 lbs.).

Proportion of Gas by Volume.	Maximum.	Maximum Temperature, Centigrade.	Time of Explosions.
$\frac{1}{7}$	41	826 to 909	·15
$\frac{1}{5}$	68	1358 ,, 1539	·026
$\frac{2}{7}$	80	1615 ,, 1929	·01

The Highest Temperature in a Gas Engine has been estimated at 3444° Fahr. absolute, the temperature of the exhaust gases being estimated at 1230° Fahr.; the highest theoretical efficiency being $= \frac{3444 - 1230}{3444} =$ 64·2 per cent. Such a high efficiency is not realised in practice, the loss of heat through the sides of the cylinder being very great, owing to its having to be kept cool in order to enable the piston to work properly. The practical efficiency of gas engines is from 12 to 22 per cent.

Hot-Air Engines are used to a limited extent for the development of small power. In these engines heat is applied to air enclosed in a cylinder, the air expands and drives a piston. The air in doing work loses heat, and the efficiency of the engine depends upon the difference of temperature through which it works; the greater the fall from the higher to the lower temperature, the greater the power developed. The efficiency cannot be greater than—

$$\frac{\text{Higher absolute temperature} - \text{lower absolute temperature}}{\text{Higher absolute temperature.}}$$

The best hot-air engines consume about 8 lbs. of coke per horse-power per hour; a $\frac{1}{2}$-horse power engine consumes about 40 lbs. of coke in working ten hours.

The "Rider" Hot-Air Engine, shown in Fig. 19, is a simple and efficient small motor. It has two single-acting cylinders placed vertically; there are two plungers, marked C and D, which are coupled by means of connecting rods J to cranks keyed at right angles to each other on a crank-shaft common to both; one plunger D, called the power plunger, works in a cylinder kept permanently hot by means of a fire, while the other C, called the compression plunger, works in a cylinder surrounded by a water-jacket E. The two cylinders are connected by a wide passage H, fitted with plates, and constituting the regenerator. There is no change of air in this engine, but the relative motions of the two plungers cause a constant alteration in volume and movement of the air between the two cylinders. Any leakage is made good through the inlet valve shown at the bottom of the cool cylinder C. The working of the engine includes four operations:*—1. A small compressed bulk of air has heat restored to it by the regenerator, the pressure rising during the process; the back pressure in C counterbalances the forward pressure in D, so that this is a period of displacement, taking place during the first quarter of a revolution as the plunger D moves up from the bottom centre. 2. The small bulk of air receives a fresh supply of heat at the highest temperature, expands, doing useful work by pressure on D, and subsequently on C to a less extent. This takes place during the second quarter of a revolution, the piston D arriving at the top centre. 3. A large expanded bulk of air is transferred from B to A, and its heat abstracted and stored in the generator. The pressure falls, and the forward pressure on C during the last half of the upward stroke counter-balances the pressure in D. This is a period of displacement, which takes place during the third quarter of the revolution. 4. The large bulk of air is compressed into the small space C. The back pressure against C is not counterbalanced by the for-

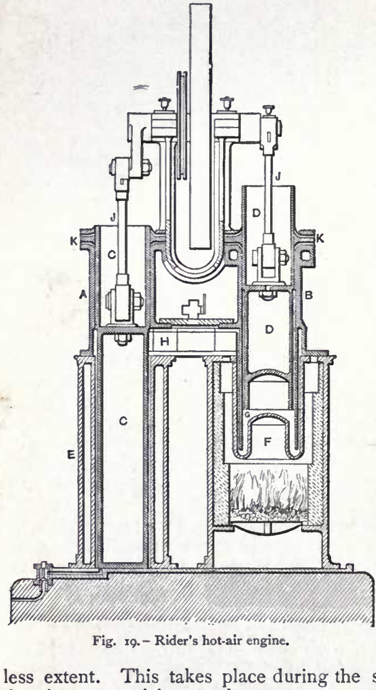


Fig. 19.—Rider's hot-air engine.

* See article on this engine in the "Mechanical World."

ward pressure in D; hence during this period work is required from the fly-wheel to compress the air, but at the same time heat is withdrawn while the fluid is at the lowest temperature, and the work required to compress the fluid while cold, and losing heat, in the fourth period is considerably less than the work given out by the fluid while expanding when hot, and receiving heat, in the second period.

The Effective Power of this Engine represents the difference between that given out in the second revolution and that required to compress the air in the fourth. The engines made of this type are of $\frac{1}{4}$, $\frac{1}{2}$, and 1 horse-power, the number of revolutions being from 100 to 150 per minute, and the indicated maximum pressure from 20 lb. to 22 lb. The regenerator consists of cast-iron plates about $\frac{1}{8}$ in. thick, with a space between of $\frac{1}{32}$ inch; in the $\frac{1}{2}$ horse-power engine, it measures about 11 inches by 4 inches by $4\frac{1}{2}$ inches. The heaters run good, if used with care, from two to five years; the cost of a new heater is about 23*s*., and the labour of one man for a day to put it in place. The air appears to get heated to the temperature corresponding to dull-red heat of cast-iron, or about 1000° Fahr. At one test of a $\frac{1}{2}$ horse-power engine, when working a pump, about 675 gallons of water were raised per hour to a height of 90 feet with 4 lbs. of coal. This performance is equal to, say 10.128 ft. lb. per minute, or ·307 horse-power; and the consumption of coal per effective horse-power was 13 lb. At another trial, 4 lbs. of coal were sufficient for $\frac{1}{2}$ horse-power duty in the water, or 8 lbs. per horse-power. The mean effective pressure in the working cylinder was 16·8, and in the cooling cylinder (making allowance for the difference in strokes) 6·47, giving 10·33 lbs. per square inch nett. The 1 horse-power engine has plungers 10 in. in diameter, and 13 in. stroke, and occupies a floor space of 4 ft. 4 in. by 2 ft. 8 in., and measures to the top of the fly-wheel 7 ft. 6 inch. The weight complete is about 28 cwt. 2 qr., and it gives out about one effective horse-power. The power cylinder has an iron jacket for about half its length, so formed that the entering air passes well round the cylinder and between the jacket, and then impinges directly on the hot heater bottom F, thus getting thoroughly heated in its passage, and at the same time the cylinder remains relatively cool. The packing round the plungers consists of leather collars held in position by a ring.

HEAT EVOLVED BY COMBUSTIBLES.

The Total Quantities of Heat evolved by the Combustion of Different Combustibles, the results of experiments by MM. Favre and Silbermann and others, are given in the following Table.

Table 22.—HEATING POWER OF VARIOUS COMBUSTIBLES.

	Heat in degrees Fahr. to which 1 lb. of water will be raised by 1 lb. of combustible.
Acetate, C_6, H_6, O_4	9616
,, of Alcohol, Valeric, C_{20}, H_{20}, O_4	14349
Acetone, C_6, $H_6 + O_2$	13149
Acid, acetic, $O_4 + C_4$, H_4, 6310; butyric acid, $O_4 + C_8$, H_8	10122
,, ethalic, $O_4 + C_{32}$, H_{32}	16956

Table 22 *continued.*—HEATING POWER OF VARIOUS COMBUSTIBLES.

	Heat in degrees Fahr. to which 1 lb. of water will be raised by 1 lb. of combustible.
Acid, formic, O_4+C_2, H_2	3600
,, phrenic, C_{12}, H_6, O_2	14117
,, stearic, O_4+C_{38}, H_{38}	17676
,, valeric, O_4+C_{10}, H_{10}	11591
Alcohol, $H\ O_2+C_4, H_4$	12932
,, ethalic, $H\ O_2+C_{32}, H_{32}$	19133
,, valeric, $H\ O_2+C_{10}, H_{10}$	16126
Aldehyde, ethalic, C_{32}, H_{32}, O_2	18616
,, stearic, C_{38}, H_{38}, O_2	18893
Amylene, C_{20}, H_{20}, 20347; C_{22}, H_{22}	20279
Aragonite, combined, gives	+68·9
,, separated, absorbs	−554·6
,, separated after combination, absorbs	−485·6
Butyrate of methylene, C_{50}, H_{10}, O_4	12238
Carbon burnt with peroxide of azote at 10°	20085
,, from C to $C\ O_2$	14545
,, ,, gas retorts	14486
,, ,, sugar from C to $C\ O_2$	14472
Cetine, C_{32}, H_{32}	19942
Decomposition of oxide of silver absorbs	−39·8
,, of peroxide of azote	19963
,, of water oxygenated 1 gr. oxygen	2346
Diamond, 13987; Diamond, heated	14182
Essence of citron, C_{20}, H_{16}	19727
,, ,, turpentine, C_{20}, H_{16}	19534
Ether, acetic, C_8, H_8, O_4	11327
,, butyric, C_{12}, H_{12}, O_4	12764
,, sulphuric, $H\ O_2+C_8, H_8$	16249
,, valeramilic, C_{20}, H_{20}, O_4	15379
,, valeric, $H\ O_2+C_{20}, H_{20}$	18339
Formiate of alcohol, C_6, H_6, O_4	9503
,, ,, methylene, C_4, H_4, O_4	7556
Gas, marsh, C_2, H_4, 23514; Gas, olefiant, C_4, H_4	21344
Graphite from high mines, 14014; No. 2	13927
Graphite, natural, No. 1, 14061; No. 2	14007
Hydrogen at 59°	62032
Iceland spar for $C\ O_2$ and C to O absorbs	−554·6
Metamylene, C_{40}, H_{40}	19672
Oxide from carbon, $C\ O_2$	4325
Paramylene, C_{10}, H_{10}	20684
Spirit of wood, $H\ O_2+C_2, H_2$	9543
Sulphur at instant of crystallization	4066
,, native melted, 3998: Sulphur of carbon	6121
Térébène, C_{20}, H_{16}	19194
Valeriate of alcohol, C_{14}, H_{14}	14103
,, ,, methylene, C_{14}, H_{14}, O_4	13277

LIQUID-FUEL.

Petroleum-Refuse and Creosote-Oil are used as liquid-fuel. They are usually burnt upon a fire-grate covered with fire-bricks, and the liquid-fuel is blown into the furnace from below, by spray-injectors, steam being used for injecting the spray. The spray-injectors have long narrow orifices, the usual width being from ·02 inch to ·08 inch. The air for combustion is generally forced into the furnace, partly above and partly under the fire-grate, by fans or blowers, and the fire-box has frequently two combustion-chambers, constructed with brick-work, which offers a slight resistance to the free exit of the ignited gases, and retains them as long as possible in the combustion chambers, thus ensuring complete admixture with the air.

Weight of Petroleum-Refuse.*—As the quality of petroleum varies considerably, it is necessary to employ a hydrometer and thermometer for testing the specific gravity and temperature of petroleum-refuse; the specific gravity varying with the temperature. The following Table contains the specific gravity and weight per cubic foot of petroleum-refuse; the heaviest petroleum-refuse has a specific gravity of ·921, or a weight of 57·412 lbs. per cubic foot when at freezing-point, thus requiring a space of 39 cubic feet to contain a ton. The lightest at a temperature of 95° Fahr. has a specific gravity of ·889, or a weight of 55·24 lbs. per cubic foot, requiring a space of $40\frac{1}{2}$ cubic feet to contain a ton.

*Table 23.—Specific Gravity and Weight per Cubic Foot of Petroleum-Refuse at Various Temperatures. Water=1·0000 Specific Gravity, at $17\frac{1}{2}$° Centigrade or $63\frac{1}{2}$° Fahrenheit.

Temperature.			Specific Gravity.	Weight in lbs. per Cubic Foot.	Temperature.			Specific Gravity.	Weight in lbs. per Cubic Foot.
Centigrade.	Réaumur.	Fahrenheit.			Centigrade.	Réaumur.	Fahrenheit.		
°	°	°		lbs.	°	°	°		lbs.
0	0·0	32	·9110	56·61	18	14·4	64·4	·8997	55·84
1	0·8	33·8	·9103	56·55	19	15·2	66·2	·8991	55·84
2	1·6	35·6	·9097	56·50	20	16·0	68·0	·8984	55·81
3	2·4	37·4	·9091	56·50	21	16·8	69·8	·8978	55·74
4	3·2	39·2	·9085	56·42	22	17·6	71·6	·8972	55·74
5	4·0	41·0	·9078	56·36	23	18·4	73·4	·8965	55·68
6	4·8	42·8	·9072	56·36	24	19·2	75·2	·8959	55·62
7	5·6	44·6	·9066	56·30	25	20·0	77·0	·8953	55·62
8	6·4	46·4	·9060	56·30	26	20·8	78·8	·8947	55·55
9	7·2	48·2	·9053	56·20	27	21·6	80·6	·8940	55·55
10	8·0	50·0	·9047	56·14	28	22·4	82·4	·8934	55·48
11	8·8	51·8	·9041	56·14	29	23·2	84·2	·8928	55·43
12	9·6	53·6	·9034	56·11	30	24·0	86·0	·8922	55·43
13	10·4	55·4	·9028	56·05	31	24·8	87·8	·8915	55·37
14	11·2	57·2	·9022	56·05	32	25·6	89·6	·8909	55·30
15	12·0	59·0	·9016	55·99	33	26·4	91·4	·8903	55·30
16	12·8	60·8	·9009	55·92	34	27·2	93·2	·8896	55·24
17	13·6	62·6	·9003	55·92	35	28·0	95·0	·8890	55·24

* See a Paper read before the Institution of Mechanical Engineers, by Mr. Thos. Urquhart.

Liquid-Fuel is very much used in Russia. On one line of railway about one hundred and fifty locomotives and a number of stationary boilers are burning this description of fuel, with the most satisfactory results, in special furnaces or combustion-chambers constructed on Mr. Urquhart's system of burning liquid-fuel. The following is a brief description of the apparatus used for feeding the furnaces and of the construction of several varieties of combustion-chambers by which the efficient and economical combustion of petroleum-refuse is satisfactorily obtained. The dimensions given in the woodcuts are French measures.

Fig. 20.

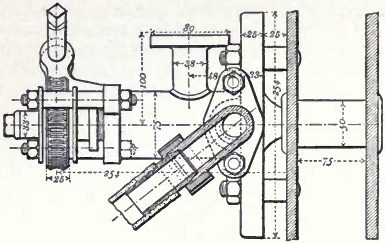
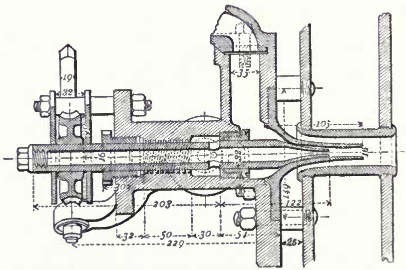


Fig. 21



Figs. 20 and 21.—Urquhart's liquid-fuel injector.

Urquhart's Liquid-Fuel Injector,* or pulveriser, is shown in Figs. 20 to 23. The orifice through which the petroleum flows, is adjustable, and can be swept out by the steam, so that if the outlet becomes choked, the opening can be enlarged, and this can be thoroughly cleansed without interfering with the working, otherwise than by increasing the flow of oil for a few moments. A tube is fixed through the double walls of the fire-box to admit the nose of the injector, around which a space is left for air to be carried in by the jet. The oil runs down a pipe, which ends in the external nozzle of the injector, while the steam passes through the inner nozzle, which it enters through a ring of holes, the steam and oil cavities being separated by a stuffing-box packed with asbestos. This packing is renewed once a month. The steam supply is regulated by a valve on the pipe and independent of the injector, while the oil supply is increased or diminished by screwing the steam nozzle backwards and forwards in the external nozzle, and varying the section of the annular passage. This is effected by a worm and wormwheel, the latter of which is connected to the steam nozzle by a feather-key, while the former is on a shaft which terminates in a convenient position close to the fireman's

* The Author is indebted to "Engineering" for the above description and drawings of Mr. Urquhart's Patent Apparatus for Burning Liquid Fuel.

hand. Skill and care in regulation are two most important points in the use of this fuel on railways, in order to avoid smoke, and, at the same time, not to admit an excessive quantity of cold air. Every change in the opening of the regulator or the position of the link requires a corresponding alteration of the oil admission, while when the engine is at rest or running down an incline with the steam off, the flame must be extinguished, and the firebox shut up by closing the dampers. Even under these conditions the steam will continue to rise from the gradual emission of the heat stored in the brickwork of the firebox. When the stoppage is ended, or the incline passed, the flame is lighted again by first turning on the steam, and then gradually admitting the oil, which catches fire from the hot bricks without any explosion. The dampers can then be opened and the full supply of oil turned on, according to the load and the character of the road. Spocket

Fig. 22

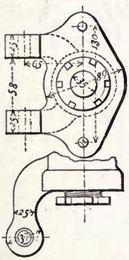
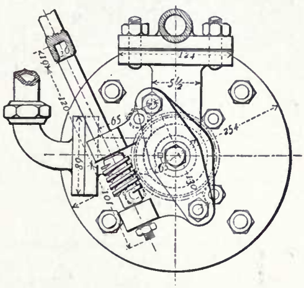


Fig. 23.



Figs. 22 and 23.—Urquhart's liquid-fuel injector.

wheels and pitched chains convey the motion of the regulating wheel from the cab to the injector.

Mr. Urquhart's apparatus is entirely outside the firebox, and is screened from the radiant heat of the flame, while the earlier injectors were more or less inside the furnace and exposed to the fierce temperature which carbonised the oil on the outlet of the jet, and contributed even more than the dirt to the choking of the nozzle.

Special Combustion-Chamber.—Important as is the construction of the injector, however, the arrangement of the firebox or combustion-chamber is more so, particularly in a locomotive with its infinite capacity for leakage at numberless tubes, stays, and rivets. The experience of forced draught on torpedo boats has shown that there is a decided limit to the temperature which a tube-plate can bear, and that if very highly heated flame be driven against it, there will be continual trouble. In the case of petroleum the flame is so hot that the sight-hole through which it is observed has to be covered with coloured glass, to protect the eyes of the fireman, and in the early experiments, it was found that the nuts on the inside of the fire-box crown dropped off, so great was the heat brought to bear upon them. Further, while the direct impact of flame is detrimental to the plates, it is also unfavourable to perfect combustion of the gases. In the most recent types of the Siemens furnace a great advantage has been

Fig. 24.

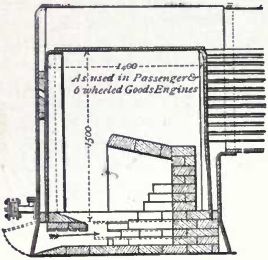


Fig. 25.

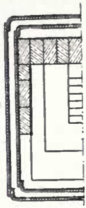
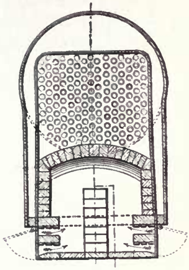


Fig. 26.



Figs. 24—26.—Combustion-chamber for liquid-fuel.

found in keeping the flame free both of the brick-work and of the objects to be heated, and although this of course cannot be followed in a boiler furnace, yet it is possible to let the principal part of the combustion take

Fig. 27.

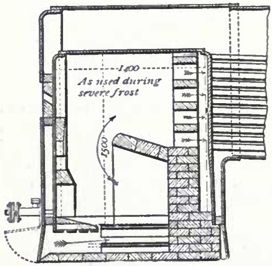


Fig. 28.

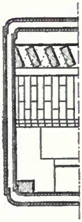
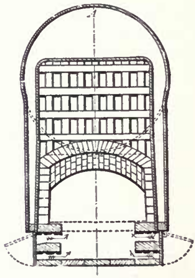


Fig. 29.



Figs. 27—29.—Combustion-chamber used in winter for liquid-fuel.

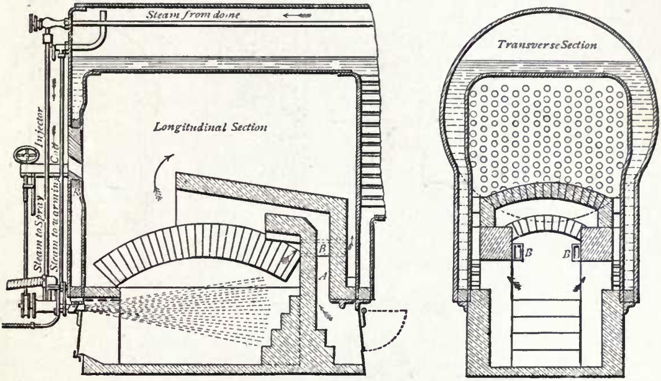
place in an area inclosed by glowing walls, which exercise no check upon the complete oxidation of the gases, but allow them to attain the full temperature before they meet the comparatively cold metal plates.

Several Forms of Combustion-Chambers are used by Mr. Urquhart, according to the class of engine to which they are to be applied, and to the teachings of experience. Five varieties are illustrated in Figs. 24 to 41.

Figs. 24 to 26 show the fire-box as used in passenger and six-wheeled goods engines. Opposite the lower part of the fire-box there is built a stout wall, which is continued along each side (Fig. 26), terminating at the top with a skewback. An arch is thrown from side to side, and forms a top to this combustion-chamber which is entirely open at the front. The

Fig. 30.

Fig. 31.



Figs. 30 and 31.—Combustion-chamber for liquid-fuel.

jet plays on to the back wall, while the flames turn up under the arch above, filling the box and entering the tubes in tongues of fire. There are three ashpit doors, one at the trailing end and one at each side, and these can be opened to any extent by a chain and notched lever. Sufficient air to commence the combustion enters with the spray, which is protected by a brick-work screen from the main draught, until it reaches the mouth of the combustion-chamber. Figs. 27, 28, 29 represent a modified arrangement for use during the severe frosts which prevail in Russia. The difference between the two is not great; the latter, however, has the back wall carried up to the roof of the box, and perforated with numerous openings which are set at an angle, as shown in Fig. 28, and diffuse the heat over the tube-plate with great uniformity. The side of the box is partially protected, but the midfeather, shown in Fig. 26, is wanting. Figs. 30, 31, 32, show another and more recent construction of combustion-chamber, in which no side-doors are cut in the ash-pan, the original front and back doors being utilised. The air which enters through the front door is passed up through a thin brick-channel A, and becomes heated before it comes in contact with the gases. Two cast-iron boxes, BB, are built into

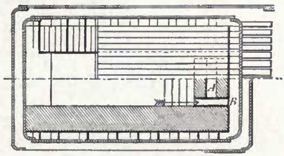


Fig. 32.—Combustion-chamber for liquid-fuel.

the brick-work, in order to let a small quantity of flame gain access to that part of the tube-plate below the tubes. The entire heating surface of the sides of the box is likewise utilised by keeping the walls of the chamber a short

Fig. 33.

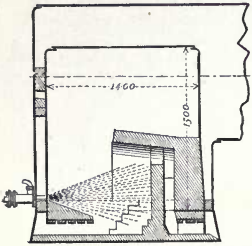


Fig. 34.

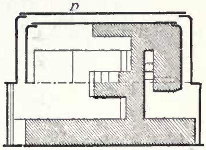
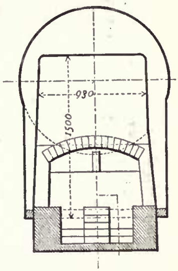


Fig. 35.



Figs. 33—35.—Combustion-chamber for liquid-fuel.

distance from them, and allowing the flame to play between the two. The spray injector is placed in the ash-pit, as shown in Fig. 31, which is made very deep, and the hollow stay shown in Fig. 21, is not used. Figs. 33, 34, and 35, illustrate another modification in which there is preliminary

Fig. 36

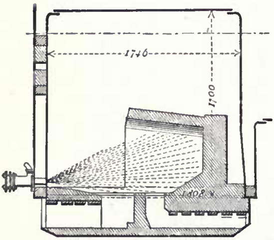


Fig. 37.

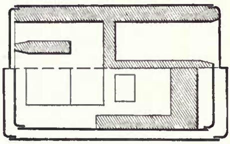
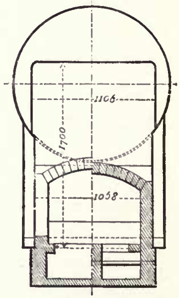


Fig. 38.



Figs. 36—38.—Combustion-chamber for liquid-fuel.

heating of the air, while Figs. 36, 37 and 38, are engravings of the combustion-chamber of an eight-wheeled locomotive engine weighing 48 tons.

The Brick-work of the Combustion-Chambers acts as a reservoir of heat, tending to keep the furnace at a uniform temperature, and to maintain the combustion under the most favourable conditions. It likewise serves to light the spray when it has been turned out during a stoppage. When the boiler is cold the jet cannot be used, of course, as there is no steam to propel the oil forwards. Under such conditions it is customary to raise the steam by a temporary attachment to a stationary boiler, and then to create a draught by turning on a blower in the chimney. A handful of burning waste is then placed in the combustion-chamber, and the steam and oil are turned on in succession. Mr. Urquhart has also applied brick combustion-chambers to all the stationary boilers, twenty-five in number, under his control; the method adopted with Cornish boilers is shown in Figs. 39, 40, and 41.

The Petroleum is carried in a tank in the tender of the locomotive,

Fig. 39. Fig. 40.

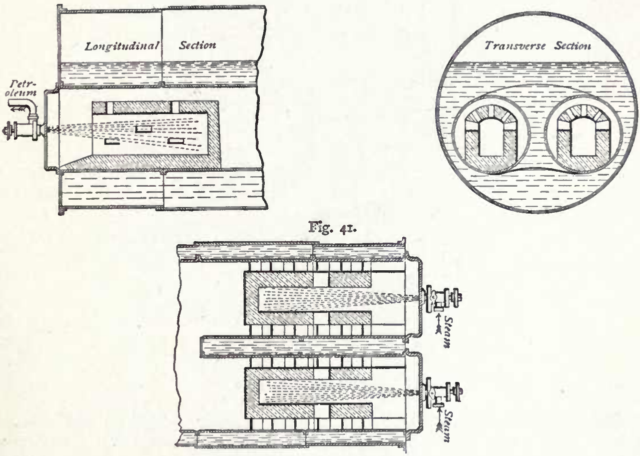


Fig. 41.

Figs. 39—41.—Combustion-chamber for burning liquid-fuel in Cornish-boilers.

formed by placing a bulkhead at the head of the coal-space, and a cover over the top of it. There is a pocket made in the bottom to receive the water, which the oil picks up in its transit in barges, and the separation is aided by the heat which is received from the feed-water tanks on the tender, and from a special steam-coil. When the temperature falls 12 degrees below freezing-point the use of artificial heat is imperative. For a six-wheel locomotive the capacity of the tank is $3\frac{1}{2}$ tons, a quantity sufficient for 250 miles, with a train of 480 tons gross, exclusive of engine and tender.

Creosote-oil is the residue from the distillation of tar in the production of anthracene, quinone, and alizarin. The heating-power of creosote-oil has been found in practice to be double that of ordinary good steam-coal, that is, one ton of oil is equal to two tons of coal. It costs about twenty-five shillings per ton, makes very little smoke in burning, and when used in a well-arranged furnace requires little attention. In Sadler's system of burning creosote-oil as liquid fuel in a circular furnace, the furnace-tube is lined with fire-brick for a portion of its length so as to form a brickwork-tube, the end of which is closed by a baffle-wall perforated with holes 3 inches square. The oil is forced into the furnace by a jet of steam, by means of a nozzle similar in construction to that of a Giffard's injector. The draught produced by the jet of steam is checked by the baffle-wall at the end of the furnace, and by an ordinary damper.

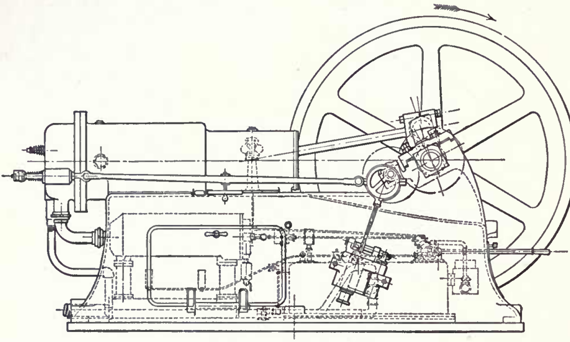


Fig. 42.—Priestman's Oil-engine.

PETROLEUM-ENGINES.

Oil-Engines occupy small space, require little attention, are readily started, and work economically with ordinary mineral oil. There are several forms of these engines, some of which may be briefly described as follows:—

The Priestman Oil-Engine.—This engine, shown in Fig. 42, is mounted on a hollow bed-plate forming a tank of sufficient capacity to contain a supply of oil to run the engine a day. Air is pumped into the tank. The oil is delivered in a state of fine spray to a vapourising chamber, where it is converted into vapour by heat supplied by a lamp on starting the engine, and afterwards by exhaust vapour which envelopes the chamber.

The engine has a four-cycle movement, that is, the piston on its first or forward stroke draws in a charge of vapour through an inlet-valve. This charge is compressed on the second stroke, and at the moment of compression an electric spark, from a small battery, explodes the charge and drives the piston out, thus making the third stroke. The fourth stroke drives the spent vapour out of the cylinder through an exhaust-valve.

The engine works satisfactorily with common mineral oil. The quantity of oil consumed is about one pint per brake-horse-power per hour. When

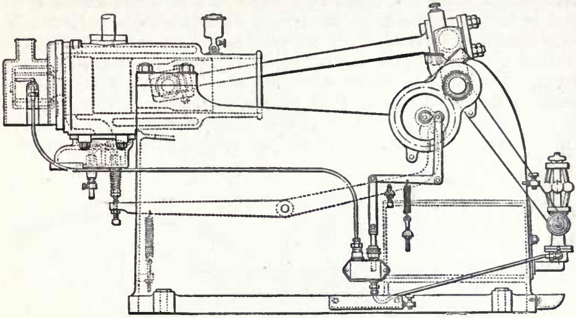


Fig. 43.—Hornsby's Oil-engine.

using oil costing fivepence per gallon, the cost of a horse-power is from one-halfpenny to three farthings per hour.

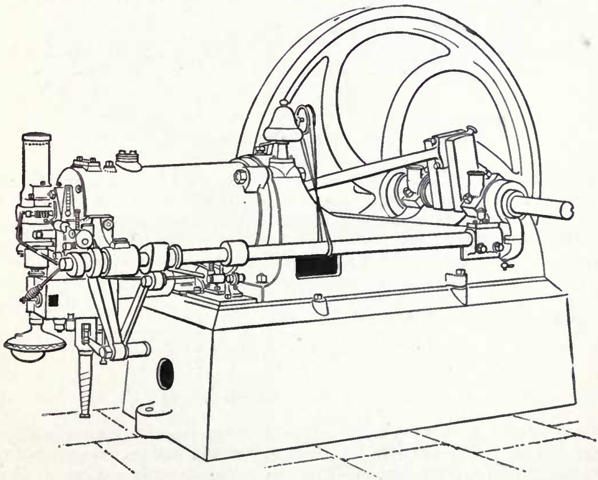


Fig. 44.—Crossley's Oil-engine.

Hornsby's Oil-Engine.—This engine, shown in Fig. 43, has a tank in the bed-plate containing oil. At each stroke a small quantity of oil is pumped into a vapourising chamber, connected by a short pipe to the back of the cylinder. This chamber is heated by a lamp on starting the engine, and afterwards by the heat of the explosion. There is one explosion in four strokes, or two revolutions. A mixture of air and oil-vapour is drawn into the cylinder on the first stroke of the piston, and compressed on its backward stroke. The compressed charge is ignited by the hot metal, and explosion takes place in the vapouriser.

Crossley's Oil-Engine.—This engine, shown in Fig. 44, resembles a gas-engine in general appearance. The oil is pumped into a vapouriser, which is heated by a lamp. The firing is effected by a horizontal hot tube.

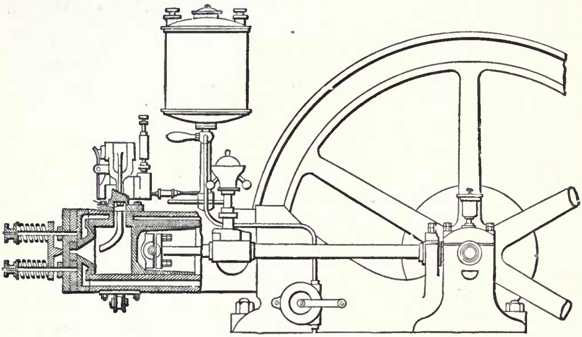


Fig. 45.—Spiel's Oil-engine.

Spiel's Oil-Engine.—This engine, shown in Fig. 45, has a four-cycle movement. The piston on its out-stroke draws in a charge of air and petroleum; it then returns compressing this mixture, which is exploded as the crank passes the back centre; in the next stroke the combustion and expansion of the charge takes place, while the fourth and last stroke of the cycle expels the products of combustion. Thus there is one acting stroke in every four, the energy stored up in the fly-wheel carrying it through the other three.

VEGETABLE-REFUSE FUEL.

In countries where coal is dear and vegetable-refuse abundant, it is important to utilise the latter as fuel for steam boilers. Vegetable substances, such as sugar-cane refuse, cotton-stalks, reeds, dry-grass, fibrous plants, peat, and brushwood make excellent fuel, and although greatly inferior to coal, can compare favourably with wood, as regards calorific power, when burnt as fuel in steam boilers properly constructed for burning this description of fuel.

Boilers using Vegetable-Refuse as Fuel require to have the furnace or fire-box twice as large as that required for coal, and the total heating surface at least one-half larger than that required for coal-burning. The fire-bars should be thin and spaced about six inches apart; and they should be placed diagonally across the furnace. The fuel should be fed continuously in a thin layer to the fire, and a large supply of atmospheric air must be admitted to the furnace to produce perfect and continuous combustion.

Heating-power of Vegetable-Refuse Fuel.—The vegetable substances above-mentioned, when in a perfectly dry state, on an average are composed of ·34 carbon, ·5 hydrogen, and ·40 oxygen. As the hydrogen

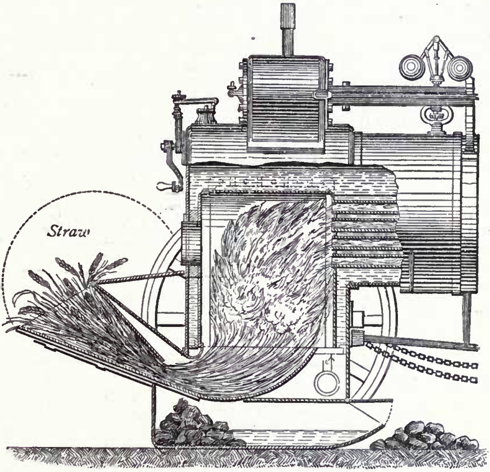


Fig. 46.—Ruston's straw-burning apparatus.

and oxygen are in the combining proportions for forming water they develop no heat. Hence these vegetable substances will yield :—Carbon $\cdot 34 \times 14500 = 4930$ units of heat per lb. of fuel : its maximum calorific power. Taking the maximum units of heat in 1 lb. of coal at 14297, the weight of straw required to develop the same heat as 1 lb. of coal will be $14297 \div 4930 = 2\cdot 94$ lbs., or nearly 3 lbs. In practice rather more than this is required, and 4 lbs. of straw are equal to 1 lb. of coal.

Ruston's Straw-burning Apparatus applied to a portable-engine is shown in Fig. 46. The straw is fed into a hopper attached to the fire-box, and is ignited on entering the furnace: as the fire is fed from the bottom of the furnace it burns like a torch, the fire not being damped down when fed, as would be the case if the straw were put upon the top of the fire. The combustion is forced by a steam-jet, and complete combustion of the fuel is obtained. The apparatus is simple and effective, and it can be used for burning other kinds of vegetable-refuse fuel as well as straw.

Coal-dust, Coke-dust, Breeze, and similar Refuse-Fuels which are of no value as fuel in ordinary furnaces, may be efficiently and economically burnt in Perret's Furnace, with water-cased grate, as shown

Fig. 47.

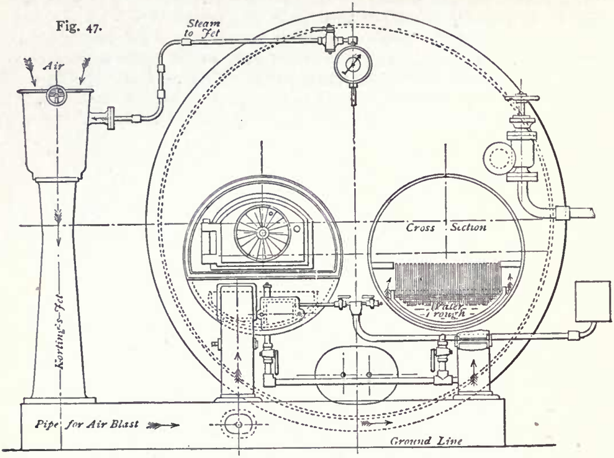
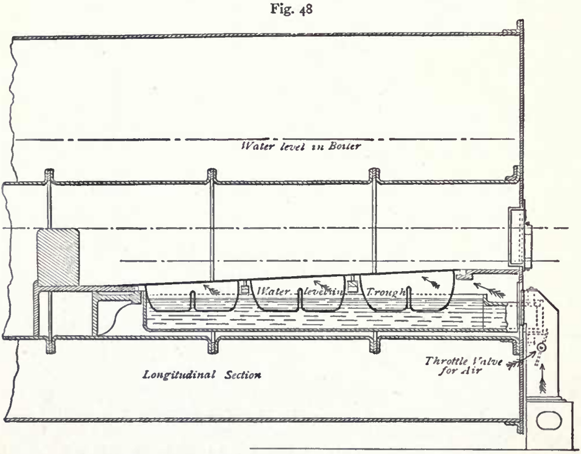


Fig. 48



Figs. 47 and 48.—Perrett's furnace for burning dust-fuel.

in Figs. 47 and 48. The ash-pit of this furnace is closed, and is worked under pressure supplied either by a fan or steam-jet. The fire-bars are spaced only about $\frac{1}{10}$th of an inch apart; they are made very deep, and the bottoms of the bars are immersed in water as shown. An air blast of a pressure of from one-half inch to one inch of water is admitted above the surface of the water in the troughs, and underneath the fire-bars, and passes through the fuel. From 20 lbs. to 30 lbs. of fuel can be burnt in this furnace per square foot of fire-grate per hour.

The evaporative power of coal-dust and breeze are given in the following Table :—

Table 24.—Comparative Trials of Coal-dust and Breeze in Perret's Furnace.

Fuel	Fuel Consumed.	Water Evaporated	Pounds of Water Evaporated per Pound of Fuel.	Cost per 1000 Gallons Evaporated.
	lbs.	lbs.	lbs.	s. d.
1. Breeze . . .	2800	14,300	5·1	4 4½
2. Equal weights of breeze and coal dust . . .	3008	17,400	5·8	5 9
3. Coal dust . .	2016	15,300	7·6	5 10½
4. Washed coal dust .	2351	15,820	7·1	6 3½

Test of Perret's Furnace. — A Lancashire boiler, fitted with Perret's furnace burning coal-dust, was carefully tested for eleven hours. The boiler was 21 feet long × 7 feet diameter, with a total heating surface of 582 square feet: fire-grates 4 feet long, and 1 foot 10½ inches wide, each equal 7½ square feet of surface, or 15 square feet in all. The coal burnt during the trial was 2231 lbs.; equal 14·8 lbs. per square foot of grate per hour, and ·383 lbs. burnt per square foot of heating surface per hour. Thickness of fuel on the bars, 5 inches : water in fuel, 4·17 per cent.: air used per lb. of fuel, 14·37 lbs.: average pressure of blast = ½ inch of water: pressure of steam, 46·2 lbs. per square inch. Ashes, 58 lbs.: clinkers, 176 lbs.: or a total waste of 234 lbs. = 10·5 per cent. Useful coal consumed, 1902 lbs. Temperature of smoke at exit from the boiler, 376·5° Fahr.

The coal-dust was analyzed and found to contain as follows :—

Table 25.—Composition of Radford's Navigation Coal-Dust, used in the Test of Perret's Furnace.

	Per Cent.				
Carbon. .	84·79 or	1892 lb.	of the day's consumption of	2231 lb.
Hydrogen .	4·55 ,,	102 ,,	,,	,,
Oxygen .	2·17 ,,	48 ,,	,,	,,
Nitrogen . .	0·96 ,,	21 ,,	,,	,,
Sulphur .	0·57 ,,	13 ,,	,,	,,
Ash . . .	2·79 ,,	62 ,,	,,	,,
Water . .	4·17 ,,	95 ,,	,,	,,

Results of the Test.*—The evaporative power was 12·8 lb. at 212° Fahr. Air-blast velocity was 2248 feet per minute through a tube of ·297 feet sectional area. Taking the weight of 1 cubic foot of air at 58° Fahr. to be ·0767 lbs., the weight of air passed during the eleven hours was $(2248 \times \cdot 297) \times \cdot 0767\ (60 \times 11) = 33798$ lbs. The carbon in the day's charge required $1892 \times \frac{8}{3} = 5045$ lbs. of oxygen to form 6937 lbs. of carbon dioxide, and the hydrogen in the day's charge required $102 \times 8 = 816$ lbs. of oxygen to form 918 lbs. of water, so that the total oxygen required to burn the coal was 5861 lbs. But the charge of fuel itself contained 48 lbs.; hence the oxygen required for the air was 5813 lbs. But the air actually supplied contained $33{,}798 \times \frac{23}{100} = 7773$ lbs. of oxygen. Deducting that absolutely required for combustion, viz., 5813 lbs., the excess of oxygen supplied was 1960 lbs., and this is equivalent to $1960 \times \frac{100}{23} = 8522$ lbs. of air, whilst the nitrogen present in the air, whose oxygen was used, was $5813 \times \frac{77}{23} = 19461$ lbs.

The proportion in which the heat of the fuel was used and wasted in the above test is shown by the following balance sheet:—

Table 26.—Balance Sheet of Heat Units (Fahr.).

Per Cent.	Heat Absorbed.	Heat Units.
63·5	By feed water evaporated, 1582 gallons at 46·2 lbs. gauge pressure. Total heat from 32 deg. in steam at 46·2 lbs.$=1167\cdot5$. Temperature of feed$=93$ deg.; hence $\{1167-(93-32)\} \times 15{,}820$	17,504,830
15·8	*Furnace Gases.*—Excess air$=8522$ lbs.$\times \cdot 238$, specific heat$\times(376\cdot5-58=318\cdot5)$ . . .	645,993
	Nitrogen with consumed oxygen$=19{,}461$ lbs.$\times \cdot 244$ (sp. ht.)$\times 318\cdot5$	1,512,392
	Carbon dioxide formed$=6937$ lbs.$\times \cdot 2164$ (sp. ht.)$\times 318\cdot5$	478,122
	Water from H in coal$=918$ lbs.$\times[(212-58)\times 1+966+(376\cdot5-212)\times \cdot 4805]$	1,100,682
	Water in coal$=95\times[(212-60)\times 1+966+(376\cdot5-212)\times \cdot 4805]$	113,715
	Evaporation from troughs under grate $416\cdot06\times[(212-60)\times 1+966+(376\cdot5-212)\times \cdot 4805]$.	498,031
5·3	Loss by radiation (say$=$night) $133{,}647\times 11$.	1,470,117
15·4	Unaccounted for: Heat in cinders, carbon in smoke, cinders unconsumed . . .	4,261,987
		27,585,869

* Kindly given to the Author by Messrs. B. Donkin & Co., London.

Heat evolved: 2231 lbs. of coal, evolving $12 \cdot 8 \times 966$ heat units per pound, will give during combustion $2231 \times 12 \cdot 8 \times 966 = 27{,}585{,}869$ units.

COMBUSTION AND CONSUMPTION OF FUEL IN STEAM BOILER FURNACES.

Firing.—The fire should be as thick as the quality of the coals will allow; a thin fire is wasteful, because parts of the grate are liable to become uncovered and permit air to pass through the furnace unconsumed. The shape of the fire should be concave, that is much thicker at the sides than the middle, which ensures proper admixture of the gases and economical

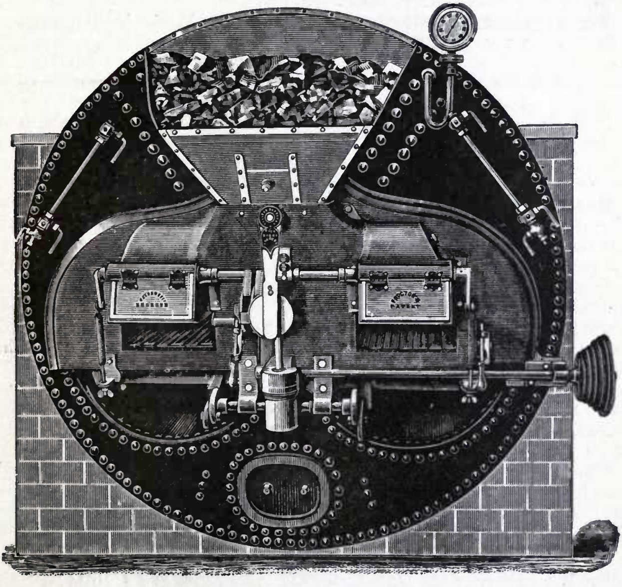


Fig. 49.—Proctor's Mechanical-Stoker.

combustion of the fuel with the least smoke. Side firing is best in most cases, that is, the coal is thrown on each side of the fire alternately, leaving one side of the fire always bright, by which means the temperature of the furnace is reduced as little as possible. It is better to fire frequently, putting a little coal on at a time, than to put large charges of coal on the fire at long intervals. Hand-firing is preferable for round coal, but for burning slack in the most economical way, mechanical-firing gives the best results in many cases.

A Mechanical Stoker provides a steady evaporation, although it is frequently not so rapid as in hand-firing: it produces a steady supply of steam, and is an excellent preventer of smoke. It dispenses with the frequent opening of the furnace door, thereby preventing large volumes of cold air impinging upon the hot plates. The Mechanical-Stoker, shown in Fig. 49. is a very good imitation of hand-firing, it being so arranged that it delivers the coal upon the fire with varied forces, by means of a radial shovel actuated by a shaft making a partial rotary motion. The machine is very simple, has few working parts, and provides an evaporation nearly as rapid as hand-firing; in one case 9·41 lbs. of water were evaporated per lb. of coal from feed water of 110° Fahr. with a very common description of slack.

Air required for Combustion.—Atmospheric air is composed of 1 lb. of oxygen and $3\frac{1}{2}$lbs. of nitrogen by weight, or 1 cubic foot of oxygen and 4 cubic feet of nitrogen by volume. Nitrogen being a neutral gas is present as a dilutent simply, and passes through the fire without chemical alteration. For every cubic foot of oxygen required in combustion, 5 cubic feet of air must be supplied. For the complete combustion of 1 lb. of hydrogen, 500 cubic feet of air are required, and for the complete combustion of 1 lb. of carbon 160 cubic feet of air are required. For the combustion of 1 lb. of sulphur 60 cubic feet are required.

The quantity of Air chemically consumed is 150 cubic feet per lb. of coal consumed, but in order to ensure complete combustion of coal, and prevent the formation of carbonic-oxide instead of carbonic acid, or the formation and discharge of smoke, it is necessary to admit a much larger quantity of air to the furnace than is theoretically required, so that each particle of gaseous combustible matter may be supplied with its due equivalent of oxygen. The supply of air to a furnace should equal double the quantity chemically consumed, or 300 cubic feet of air at 62° Fahr. per lb. of coal.

The weight of Air per lb. of Coal required in practice to secure sufficient dilution of the gases to ensure their combustion, and absence of smoke, is 24 lbs. with natural or chimney draught, and 18 lbs. with forced or artificial draught.

The Products of Perfect Combustion are steam and carbonic acid, each being invisible and incombustible, steam is formed from the hydrogen gas given out by the coals combining with its equivalent of oxygen from the air, in the proportion of 2 volumes of hydrogen to 1 of oxygen, or by weight as 1 to 8. Carbonic acid is formed from the carbon of the coal combining with its equivalent of oxygen from the air, in the proportion of 2 volumes of oxygen to 1 of carbon, or by weight as 16 to 6.

The Products of Imperfect Combustion are, carbonic-oxide, invisible but combustible, and smoke, partly invisible and incombustible. Carbonic-oxide is formed from the carbonic-acid first produced, receiving another volume of carbon in passing through the fire, which last volume of carbon is unconsumed, and forms the combustible carbonic-oxide, whilst carbonic acid, having its carbon consumed, is incombustible. Carbonic-oxide burns with a pale blue flame, and its presence in a furnace denotes imperfect combustion, from a deficient supply of air, as it indicates that only 8 parts of oxygen instead of 16 parts have united with 6 parts of carbon.

The Vitiation of the Air by the products of Combustion is chiefly caused by the carbonic acid and steam produced by combustion. Petroleum and solar oil produce least of both substances, and tallow, coal, and coal-gas most. Coal-gas also contains sulphur, which forms sulphurous and sulphuric acid to the injury of plants and polished metallic surfaces. The quantities of carbonic acid and steam produced by various kinds of combustibles, or the effects of artificial light on air in closed rooms, may be ascertained from the following Table.

Table 27.—Composition of various Combustibles.—The total Heat evolved by their Combustion; the Weight of Oxygen; the Quantity of Air Consumed, and the Products of their Combustion.

Description of Combustible.	Composition, per Cent.			Weight of Oxygen consumed per lb. of Combustible.	Quantity of Air chemically consumed per lb. of Combustible.	Products of Combustion from 1 lb. of Combustible.		Units of Heat evolved by the Combustion of 1 lb. of the Combustible.
	Carbon.	Hydrogen.	Oxygen.			Carbonic Acid.	Water.	
				lb.	lb.	lb.	lb.	Units
Coal, average	80·4	5·2	8·0	2·46	12·00	2·95	·47	14300
Coal-gas . .	70·5	3·2	...	3·25	14·16	2·60	2·05	21000
Tallow . .	78·8	11·8	9·4	2·95	12·85	3·19	1·35	18120
Stearine . .	76·3	12·4	11·3	3·45	15·00	3·17	1·32	17950
Wax . . .	81·5	13·6	4·9	3·24	14·14	3·08	1·17	19950
Sperm . .	80·5	13·5	6·0	3·18	13·84	3·05	1·18	19680
Paraffin . .	85·5	14·5	...	3·48	15·14	2·90	1·08	21460
Colza-oil . .	77·3	13·3	9·4	3·08	13·40	2·86	1·25	18840
Solar-oil . .	77·5	13·5	9·0	2·97	12·79	2·84	1·16	18990
Petroleum .	85·3	14·7	...	3·20	13·92	2·72	1·13	21560

Smoke is formed during combustion, from the hydrogen and carbon which have not received their respective equivalents of oxygen from the air, and thus pass off unconsumed.

Prevention of Smoke.—Coals can be burnt without smoke by providing a proper supply of air to the furnace, to secure sufficient dilution of the gases to ensure their combustion, and to prevent the formation of smoke; but when smoke has been produced, it can neither be consumed nor converted to heat. The colour of smoke depends upon the carbon passing off in its dark pulverised state, but the quantity of heat carried away is not dependent upon the carbon alone, but also upon the invisible but combustible gases—hydrogen and carbonic-oxide, so that whilst the colour may indicate the amount of carbon in the smoke, it does not indicate the amount of heat lost. The prevention of smoke considerably increases the economic value of the fuel and the evaporative power of the boiler.

The Supply of Air to the Boiler Furnace may be regulated by means of a furnace-door, having a sliding grid on the outside, as shown in Fig. 50. A perforated box-baffle-plate is fixed on the inside of the door for admitting air above the fire. The sum of the perforations should not be less than 3 inches per square foot of fire-grate surface. After firing, the ventilating-grid in the door is opened for about a minute, and the proper supply of air is admitted to the furnace; by this means, and with careful hand-firing, the emission of smoke may be effectually prevented.

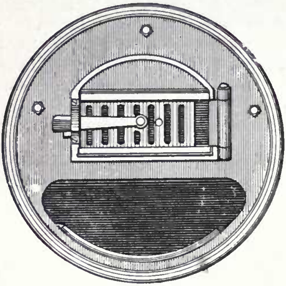


Fig. 50.—Furnace-front.

Quantity and Temperature of Air in Steam Boiler Furnaces.—The quantity of air required is, as already stated, 300 cubic feet at 62° Fahr. per lb. of coal consumed. The air enters the fire at 62° Fahr., and leaves it at about 1250° Fahr., and expands to $3\frac{1}{4}$ times its initial volume; the maximum temperature of the furnace is about 2400° Fahr., so that the air departs at about one-half of the maximum temperature of the furnace. After leaving the furnace the air decreases in temperature and volume on its way to the chimney, where its temperature is from 552° to 600° Fahr., and its volume about 600 cubic feet.

The Quantity of Hot Gases evolved by Combustion at any special temperature per hour, may be found from the following formula:—

Let Q = lbs. of coal consumed per hour.
C = volume of cold air reduced to the temperature 32° Fahr. required in cubic feet per lb. of coal.
V = volume of heated gases per hour.
T = temperature in the chimney.
t = the temperature of the external air.

Then $V = Q\,C\,[1 + \cdot 00203\,(T - t)]$.

Weight of Gases and Vapours.—The weights and volumes of gases and vapours at 62° Fahr., under an atmospheric pressure of 30 inches of mercury, are as follows:—

Atmospheric air	weighs	·076098	lb. per cubic ft. & the volume of 1 lb. is	13·14	cubic ft.
Hydrogen gas	,,	·005264	,, ,,	189·73	,,
Nitrogen gas	,,	·073795	,, ,,	13·55	,,
Oxygen gas	,,	·084133	,, ,,	11·90	,,
Vapour of water	,,	·047398	,, ,,	21·00	,,
Carbonic acid gas	,,	·116365	,, ,,	8·60	,,
Carbonic oxide gas	,,	·073632	,, ,,	13·65	,,
Vapour of alcohol	,,	·120876	,, ,,	8·30	,,

Vapour of oil of turpentine	weighs	·362103	lb. per cubic ft. & the volume of 1 lb. is	2·80 cubic ft.
Vapour of sulphuric ether	„	·197016	„ „	5·00 „
Vapour of mercury	„	·529911	„ „	1·90 „
Vapour of benzine	„	·206152	„ „	5·10 „
Chloroform	„	·404613	„ „	2·60 „
Olefiant gas	„	·073395	„ „	13·84 „
Ammoniacal gas	„	·044726	„ „	23·10 „
Light carburetted hydrogen	„	·041943	„ „	24·66 „
Coal-gas	„	·033300	„ „	30·00 „

The Weight of a cubic foot of Gaseous Steam is ·622—or about $\frac{5}{8}$—of that of a cubic foot of air, of the same pressure and temperature.

The Volume of the Products of Combustion can be calculated as follows. In the case of a combustible composed only of carbon, 1 lb. of carbon unites with 2·66 lbs. of oxygen to form 3·66 lbs. of carbonic acid, the volume of 1 lb. of carbonic acid it appears from the above Table is 8·6, then $3·66 \times 8·6 = 31·476$ cubic feet, the volume of gas produced. The volume of the oxygen consumed will equal $2·66 \times 11·9$, its volume from the above Table, $= 31·654$, from which it appears that the volume of carbonic acid gas produced by the combustion of 1 lb. of carbon is the same as that of the oxygen consumed. Hence when a fuel is composed only of carbon, the volume of gas in the chimney will be the same as that of the air entering the furnace, except that it will be expanded to the volume equal to its increased temperature.

When a combustible contains hydrogen, as previously stated, 1 lb. of hydrogen unites with 8 lbs. of oxygen and forms 9 lbs. of steam, such as coal containing ·052 of hydrogen which unites with $·052 \times 8 = ·416$ lb. of oxygen to form $·052 + ·416 = ·468$ lb. of water, which—taking the volume of the vapour of water as given in the above Table at 21 cubic feet per lb.—will give $·468 \times 21 = 9·828$ cubic feet of vapour at 62° Fahr.; and as 300 cubic feet are required for combustion per lb. of coal, the volume of gases produced at 62° Fahr. will be $300 + 9·828 = 309·828$ cubic feet per lb. of fuel. Assuming the temperature of the chimney of a steam-boiler to be 552° Fahr., at which temperature the volume of air is double that at 62° Fahr., the volume of vapour, air, and gases in the chimney will be $309·828 \times 2 = 619·65$ cubic feet. In round numbers there will be 620 cubic feet of the gaseous products of combustion at 552° Fahr. per lb. of coal, in the chimney.

The Volume of the Gaseous Products of Combustion in the furnace of a steam-boiler per horse-power, may be calculated from the above data:—Assuming 10 lbs. of coal to be consumed per nominal horse-power per hour, then $620 \times 10 = 6200$ cubic feet of gases will be discharged by the chimney per horse-power per hour.

Chimney Draught is due to the motion of the air caused by the difference in weight of the air inside and that outside the chimney. The draught can be checked by closing the damper or opening the furnace-door. The resistances to draught, are the passage of the air through the interstices of the coals on the fire-grate, and the friction of the air and gases against the sides of the furnace and flues.

The Draught is most efficient when the temperature of the heated gases passing up the chimney is 552° Fahr. The gases only then weigh one-half that of the air outside the chimney at 62° Fahr. This refers to natural draught.

The Temperature of the Smoke in the Funnel of well-arranged marine steam-boilers with natural draught is 600° Fahr.

When Flame issues from the Top of the Funnel, it is due to the burning of gases which have escaped from the furnace without being consumed, owing to a deficiency in the supply of air to the furnace. It is detrimental because it denotes loss of heat and waste of fuel, besides causing injury to the funnel.

The Loss of Fuel due to a High Temperature in the Funnel of a marine steam-boiler may be calculated by this *Rule:* Multiply the normal consumption of coal in tons per day, by the difference between what the temperature in the funnel should be, and what it actually is, and divide the product by 2200, the quotient will give the increase of consumption per day, due to the high temperature in the funnel.

Example: The consumption of coal at the beginning of a voyage of a steamship was 30 tons per day, the temperature of the funnel then being 556° Fahr., but after a few days the temperature of the funnel increased to 706° Fahr., for the same supply of steam under the same conditions, what was the increased consumption of coal due to the increase of temperature in the funnel?

Then $\frac{30 \text{ tons per day} \times (706^\circ - 556^\circ)}{2200} = 2{\cdot}045$ tons per day, increase of consumption, thus raising the consumption of fuel from 30 tons per day to $30 + 2{\cdot}045 = 32{\cdot}045$ tons per day.

The Temperature of the Funnel of a marine steam-boiler may be calculated from the increased consumption of fuel due to its excessively high temperature by this *Rule:* Multiply the difference between what the temperature in the funnel should be, and what it actually is, by 2200, and divide the product by the normal consumption of coal in tons per day, the quotient will give the increase of temperature in the funnel.

Example: The temperature of the funnel at the beginning of a voyage of a steamship was 556° Fahr., the consumption of coal then being 30 tons per day, but after a few days it increased to 32·045 tons per day, for the same supply of steam, under the same conditions, what was the increase of temperature in the funnel?

Then $\frac{2200 \times (32{\cdot}045 \text{ tons} - 30 \text{ tons})}{30 \text{ tons normal consumption per day}} = 150^\circ$ Fahr. increase of temperature in the funnel, thus raising its temperature from 556° Fahr. to $556^\circ + 150^\circ = 706^\circ$ Fahr.

The Normal Temperature of the Gases escaping from the Funnel of a marine steam-boiler will be increased at the rate of 22° Fahr. for 1 per cent. increase in the cost of evaporation. *Example:* Taking the data from the previous example, the increase is 32·045 tons—30 tons = 2·045 tons. This increase on 30 tons will equal $\frac{30}{2{\cdot}045} = 14{\cdot}67$ and $\frac{100}{14{\cdot}67} = 6{\cdot}87$ per cent., therefore the increase of temperature above the normal temperature, will be $6.82 \times 22 = 150°$ Fahr., and this added to the original temperature will give $150° + 556° = 706°$ Fahr., or the same result as obtained by the previous *Rule*.

The Consumption of Coal in Marine Boilers varies as the distance steamed, multiplied by the square of the speed of the ship.

Example: A steamship made a voyage of 900 miles at a speed of 10 knots per hour, the total consumption of fuel being 100 tons. Required the consumption of fuel, C., for a voyage of 1500 miles at a reduced speed of 8 knots per hour.

Then $C = 900 \times 10^2 : 1500 \times 8^2 :: 100$ tons;

or $\dfrac{1500 \text{ miles} \times 8^2 \text{ knots} \times 100 \text{ tons}}{900 \text{ miles} \times 10^2 \text{ knots}} = 106$ tons. consumption of fuel at the reduced speed.

The Consumption of Coal in Marine Boilers per unit of time, varies as the cube of the speed of the ship.

Example 1: A steamship attained a speed of 10 knots an hour with a consumption of 7 tons of coal per day. Required the consumption, C, at a reduced speed of 8 knots per hour.

Then $C = 10^3$ knots : 8^3 knots : : 7 tons; or $\dfrac{8^3 \text{ knots} \times 7 \text{ tons}}{10^3 \text{ knots}} =$ 3·584 tons.

Example 2: On one voyage a steamship attained a speed of 8 knots an hour with a consumption of 10 tons of coal per day. After undergoing repairs, this ship attained, on another voyage, a speed of 9 knots an hour. Required the consumption, C., of coal at the increased speed.

Then $C = 8^3$ knots : 9^3 knots : : 10 tons; or $\dfrac{9^3 \text{ knots} \times 10 \text{ tons}}{8^3 \text{ knots}} =$ 14·23 tons.

The Speed of a Steamship due to a given consumption of Coal may be found by the converse of the last *Rule:*

Example 1: Taking the data from example 2 above. A steamship attained a speed of 8 knots per hour with a consumption of 10 tons of coal per day. Required the speed attained with a consumption of 14·23 tons of coal per day. Then $\dfrac{14.23 \text{ tons} \times 8^3 \text{ knots}}{10 \text{ tons}} = 729$ and $\sqrt[3]{729} = 9$ knots per hour, the speed due to the increased consumption of coal.

Example 2: Taking the data from example 1 above. A steamship attained a speed of 10 knots per hour with a consumption of 7 tons of coal per day. Required the speed attained with a consumption of 3·584 tons of coal per day. Then $\dfrac{3.584 \text{ tons} \times 10^3 \text{ knots}}{7 \text{ tons}} = 512$ and $\sqrt[3]{512} =$ 8 knots per hour, the speed due to the decreased consumption of coal.

The Consumption of Coal in a Voyage of a steamship may be found by this *Rule:* Multiply the speed in knots per hour by 24, the product will be the number of knots the ship makes per day. Then as the number of knots sailed per day is to the number of knots in the voyage, so is the consumption of coal per day to the consumption of coal on the voyage.

Example: A steamship attained a speed of 10 knots per hour with a consumption of 20 tons of coal per day. Required the consumption, C, of coal in a voyage of 1000 knots.

Then the speed per day will be 10 knots × 24 hours = 240 knots, and C = 240 knots : 1000 knots : : 20 tons, or $\frac{1000 \text{ knots} \times 20 \text{ tons}}{240 \text{ knots}} = 83{\cdot}3$ tons.

The Quantity of Coal left at the end of a Voyage of a ship steaming at a uniform speed, may be found as follows. As the number of knots made is to the number of knots in the voyage, so is the coal which has been consumed in the knots made. The result will be quantity of coal consumed during the voyage.

Example : A steamship is provided with a stock of 350 tons of coal for a voyage of 2400 knots, it was found after steaming 800 knots that 100 tons of coal had been consumed. How much coal will there be left at the end of the voyage, the speed being the same for the whole voyage?

Then 800 knots : 2400 knots : : 100 tons coal : quantity of coal consumed during voyage, or $\frac{2400 \text{ knots} \times 100 \text{ tons}}{800 \text{ knots steamed}} = 300$ tons of coal consumed during the voyage, which deducted from the stock at starting on the voyage, or 350 − 300 = 50 tons, the quantity of coal that will be left at the end of the voyage.

The Consumption of Coal per indicated horse-power of the engine may be ascertained by this *Rule :* Multiply the number of tons of coal consumed per day by 2240, and divide the product by 24, the quotient will be the number of lbs. of coal consumed per hour, which divided by the indicated horse-power of the engine will give the consumption of coal in lbs. per indicated horse-power per hour.

Example : The consumption of coal during the voyage of a steamship was 140 tons per day, the average indicated horse-power of the engines was 5400. Required the consumption of coal per indicated horse-power per hour.

Then 140 tons × 2240 ÷ 24 = 13066 lbs. of coal consumed per hour, and $\frac{13066 \text{ lbs.}}{5400 \text{ horse-power}} = 2{\cdot}23$ lbs., the coal consumption per indicated horse-power per hour.

The number of Baskets of Coal burnt in a watch on a steamship may be ascertained by this *Rule :* Multiply the consumption of coal in tons per day by 2240, and divide the product by the weight of coal contained in the basket, the quotient will be the number of basketfuls of coal consumed per day, which divided by the number of watches per day will give the number of basketfuls burnt in a watch.

Example : The total consumption of coal on a steamship is 140 tons per day, how many basketfuls of coal, each weighing 60 lbs., will be burnt in a watch of 4 hours?

Then 140 tons × 2240 lbs. = 313600 lbs. of coal consumed per day, and $\frac{313600 \text{ lbs.}}{60 \text{ lbs. weight of basketful}} = 5226{\cdot}6$ basketfuls per day. As the watch is 4 hours there will be 24 ÷ 4 = 6 watches in the day, then 5226·6 basketfuls per day ÷ 6 = 871·1 basketfuls per watch.

The number of Basketfuls of Coal burnt per day during the voyage of a steamship being given, the consumption of coal in tons per day may be ascertained by this *Rule :* Multiply the weight of coal contained in the basket by the number of basketfuls, and divide the product by 2240, the quotient will be the coal-consumption in tons per day.

Example: The consumption of coal on a steamship during a voyage was 5226·6 basketfuls per day, each weighing 60 lbs. Required the coal-consumption in tons per day.

Then $\frac{5226 \cdot 6 \text{ basketfuls} \times 60 \text{ lbs. each}}{2240 \text{ lbs. per ton}} = 140$ tons of coal consumed per day.

The Consumption of Coal in tons during the voyage of a steamship may be ascertained by dividing the number of cubic feet of coal used by 45, the number of cubic feet in a ton.

Example: A steamship after 7 days' steaming was found to have used coal that occupied a space 18 feet in length of a coal-bunker which is 25 feet long, 14 feet wide, and 15 feet high. Required the daily consumption of coal in tons.

Then $\frac{14 \text{ feet width} \times 15 \text{ feet height} \times 18 \text{ feet length of coal burnt}}{45 \text{ cubic feet of coal per ton} \times 7 \text{ days}} =$ 12 tons daily coal-consumption.

The Consumption of Coal by Locomotives per indicated horse-power per hour increases with the speed of the engine. If the speed of the engine be doubled and the weight of the train remain the same, the quantity of fuel burned in a given time would also be doubled; but as only one-half the time would be occupied in travelling a mile, the consumption of fuel per mile would be the same in both cases. Therefore the rate of consumption of fuel per train-mile varies with the weight of the train and is independent of the speed. The consumption of coal per train-mile varies considerably on different railways, as will be seen from the following Table, collated from recent practice:—

Table 28.—Average Consumption of Coal per Train-Mile by Passenger Locomotives on various English Railways.

	Brighton and South Coast.			Great Western Express.	Great Northern Express.	Midland Express.	North-Western.	
	Light Express.	Tank Engine.	Heavy Express.				Express.	Compound.
	lbs.	lbs.	lbs.	lbs.	lbs.	lbs.	lbs.	lbs.
Average consumption of coal per mile . .	22	24	30	25	27	28	34	38

The consumption of coal per train-mile by goods engines averages from 45 to 52 lbs.

In an experiment made to ascertain the power required to haul a train, weighing with the engine $335\frac{3}{4}$ tons, from Brighton to London, Mr. Stroudley found that 1 lb. of coal would convey 1 ton weight of the train $13\frac{1}{2}$ miles, at an average speed of 43·38 miles per hour, over the Brighton Railway, the rate of consumption being 2·03 lbs. of coal per indicated horse-power per hour: the evaporation was 12·95 lbs. of water per lb. of coal.

Consumption of Petroleum-Refuse Fuel.--A series of careful trials were made by Mr. Thomas Urquhart to ascertain the mean consumption of petroleum-refuse fuel in locomotive-engine boilers, during continuous trips in winter and summer.*

The mean result for the whole year for six-wheeled coupled goods-engines is a coal consumption of 69·80 lbs. as compared with 43·19 lbs. of petroleum, at a cost of 10·212 pence for coal, and 5·459 pence for petroleum; being an advantage of 38 per cent. in weight and 46 per cent. in cost of petroleum-refuse. The mean result per train-mile, with four-wheeled coupled passenger-engines is 39·38 lb. of coal, against 29·62 lb. of petroleum, at a cost of 5·672 pence for coal, and 3·808 pence for petroleum; being an advantage of 25 per cent. in weight, and 33 per cent. in cost of petroleum-refuse. Petroleum is also successfully used in the above-named boilers as an anti-incrustator; about 4 lbs. of petroleum are used for every 100 miles run, and the boilers are washed out every 600 miles.

Table 29.—Composition, Heating Power, and Theoretical Evaporative Power of Pennsylvanian and Russian Crude Petroleum Oil.

Crude Petroleum Oil.	Pennsylvanian.	Russian.		
		Light.	Heavy.	Naphtha-Refuse.
	Per cent.	Per cent.	Per cent.	Per cent.
Carbon	84·9	86·3	86·6	87·1
Hydrogen	13·7	13·6	12·3	11·7
Oxygen	1·4	·1	1·1	1·2
	100	100	100	100
Specific gravity at 32° Fahr., water=1000	·886	·884	·886	·928
	Units.	Units.	Units.	Units.
Heating-power in British thermal units	19210	22628	19410	19260
	lbs.	lbs.	lbs.	lbs.
Theoretical evaporation at 120 lbs. or 8 atmospheres pressure, in lbs. of water per lb. of fuel	16·2	17·4	16·4	16·2

* See a paper read before the Institution of Mechanical Engineers by Mr. Thomas Urquhart.

Table 30.—Comparative Trials of a Locomotive Engine Burning Petroleum, Anthracite, Bituminous Coal and Wood.

Description of Fuel used.	Consumption, including Lighting-up in Winter.				Consumption, including Lighting-up in Summer.		
	Total.	Per Train-Mile.	Cost of Fuel per Train-Mile.	Temperature and Weather.	Total.	Per Train-Mile.	Cost of Fuel per Train-Mile.
	lbs.	lbs.	Pence.		lbs.	lbs.	Pence.
Anthracite . .	31779	81·90	11·96	−6° to −8½° Fahr. Strong side-wind.	...	...	...
Bituminous coal .	37558	96·53	14·09		14084	72·60	10·60
Petroleum-refuse .	9462	48·77	5·49		6176	31·84	3·58
Anthracite . .	12640	65·15	9·52	21° to 12° Fahr. Light side-wind.	12784	65·90	9·62
	cubic ft.	cubic ft.					
Wood, in billets .	1072	5·52	8·50		...	...	...
	lb.	lb.					
Petroleum-refuse .	7223	37·23	4·19		6104	31·46	3·54

Prices of Fuel used.—Petroleum refuse, 21*s.* per ton; anthracite and bituminous coal, 27*s.* 3*d.* per ton; wood in billets, 42*s.* per cubic sajene = 343 cubic feet, equivalent to 1·47 penny per cubic foot.

Dimensions of Locomotive Engines.—Cylinders, $18\frac{1}{8}$ inches diameter, and 24 inches stroke; wheels, 4 feet 3 inches diameter. Total heating-surface, 1248 square feet. Total adhesion-weight, 36 tons. Boiler-pressure, 120—135 lbs.

The rate of Combustion in Steam Boilers is the weight of fuel in lbs. per hour burnt on each square foot of fire-grate; it depends upon the quantity of draught and the combustibility of the fuel, and it varies with different classes of boilers.

The Average rate of Combustion in Steam Boilers per square foot of fire-grate per hour is as follows:—

Egg-ended boilers, externally fired . . .	6 to 10 lbs.
Water-tube boilers	7 ,, 14 ,,
Galloway-boiler	8 ,, 15 ,,
Portable-engine boilers	9 ,, 16 ,,
Vertical boilers	10 ,, 30 ,,
Cornish boilers	12 ,, 14 ,,
Lancashire boilers	14 ,, 18 ,,
The nozzle boiler	15 ,, 26 ,,
Marine boilers with natural draught . . .	16 ,, 24 ,,
Marine boilers with steam jet	28 ,, 30 ,,
Marine boilers with air pressure	34 ,, 40 ,,
Locomotives burning coal	40 ,, 65 ,,
Torpedo-boat boilers	60 ,, 70 ,,
Locomotives burning coke	65 ,, 110 ,,

The rate of Combustion of various Coals in lbs. of coal burnt per square foot of fire-grate per hour, with natural draught, is as follows :—

Welsh steam coal	20 to 21	lbs. per sq. ft. of fire-grate per hour.
Newcastle steam coal	24 „ 30	„ „ „
Yorkshire steam coal	25 „ 30	„ „ „
Lancashire steam coal	26 „ 28	„ „ „

The Force developed by Combustion, or its dynamical value, may be ascertained by multiplying the number of units of heat developed by the complete combustion of 1 lb. of fuel by the mechanical equivalent of each unit of heat, or 772. Hence the force developed by 1 lb. of carbon in burning to carbonic acid is equal to $14500 \times 772 = 11,194,000$ foot pounds.

The Force developed by the complete Combustion of one lb. of good Coal, taking its calorific power at 14000 thermal units per lb. of coal, is equal to $14000 \times 772 = 10,808,000$ foot pounds, or equal to 10,808,000 pounds raised one foot high.

The Horse-power theoretically developed by the heat yielded by the complete combustion of 1 lb. of coal per hour will equal—

$$\frac{10808000 \text{ foot pounds}}{33000 \text{ foot pounds per minute} \times 60 \text{ minutes}} = 5{\cdot}4$$ horse-power, representing a consumption of coal per indicated horse-power of ·185 lb , a result which has not yet been attained in practice, as the best engines only perform about one-tenth of that duty.

The Actual Power developed by the Combustion of one lb. of Coal in Practice is on an average equal to one-eleventh part of the power theoretically developed by the coal in first-class compound condensing engines, or $·185 \text{ lb.} \times 11 = 2{\cdot}035$ lbs., or say 2 lbs. coal-consumption per indicated horse-power per hour; but in some cases it has been less than this, or about $1\frac{1}{2}$ lb. In high-pressure non-condensing engines the actual power developed is on an average only about one-twentieth part of that theoretically due to the coal, or $·185 \times 20 = 3{\cdot}7$ lbs. coal-consumption per indicated horse-power per hour.

The Consumption of Coal per Nominal Horse per Hour averages 10 lbs in ordinary stationary engines.

Small Vertical Engines with vertical boilers, on an average consume 6 lbs. of coal per indicated horse-power per hour.

Stationary Engines, non-condensing, on an average consume from 3 to 4 lbs. of coal per indicated horse-power per hour.

Stationary Engines, condensing, on an average consume from $1\frac{3}{4}$ to $2\frac{1}{2}$ lbs. of coal per indicated horse-power per hour.

Portable-Engines on an average consume 4 lbs. of coal per indicated horse-power per hour.

Stationary and Portable Compound Non-condensing Engines on an average consume from $2\frac{1}{2}$ to 3 lbs. of coal per indicated horse-power per hour.

Locomotive Engines on an average consume from 2 to $2\frac{1}{2}$ lbs. of coal per indicated horse-power per hour.

Double Expansion Compound Marine Engines on an average consume from $2\frac{1}{4}$ to $2\frac{3}{4}$ lbs. of coal per indicated horse-power per hour.

Triple Expansion and Quadruple Expansion Marine Engines on an average consume from $1\frac{1}{4}$ to $1\frac{1}{2}$ lbs. of coal per indicated horse-power per hour.

The Difference between the Theoretical and Actual Force developed by steam-boilers is partly due to imperfect combustion and partly to the small portion of the total heat developed by combustion which, in practice, can be applied to the heating surfaces of boilers and absorbed by the water, the remainder being wasted, owing in many cases to the heating surfaces of the boiler not being arranged in the best manner for effectually depriving the products of combustion of their heat.

In order to obtain high economical results from the combustion of fuel, it is necessary in designing steam-generators to understand the true principles of boiler-construction, regarding the combustion of fuel and the liberation, absorption, radiation, conduction and convexion of heat. Combustion has been previously treated, and the following subjects may now be briefly considered.

Action of Flame in Boiler-Flue Tubes.—Peclet says, in speaking of the heating of liquids by gas, as for example, in the case of a steam-boiler, and of that portion which does not receive the direct rays of heat radiated from the fuel on the fire-grate, that is to say in the flues, the quantity of heat which traverses the plate is invariably determined by the difference in temperature on its opposite sides; the absorbing and emissive powers of the two surfaces of the plate, and above all, by the movements of the sheets of gas which are in contact with the metal. It will be found in all cases that the rapid renewal of layers of liquid or gas which touch the surface of the metal plate, has a great influence on the transmission of heat; but this circumstance is much more important in the case of gases than in the case of liquids. Suppose, for example, that the products of combustion escape at a high temperature into a circular flue surrounded with water which we wish to heat, the sheets of gas which are in contact with the plate are cooled down with great rapidity, but all the little elementary currents having a direction parallel with the axis of the flue, the sheets of gas change places very slowly, because the only cause for change lies in the acquisition of density which results from cooling; but this change of temperature only takes place in the upper portion of the flue, and it tends to produce displacement with a slow speed. Hence if the section of a flue be considerable, and the speed at which the gases move through it be high, the greater part of the central veins will not come in contact with the surface of the plate, and they will therefore preserve their original temperature.

The Flame in passing from the Furnace through the Flue-Tube of a Cornish Boiler acts in a similar manner to the above. The atoms of heated gas come in contact with the top of the flue-tube, impart heat to it, and as they move along the flue they cool and sink away from it and are replaced by others; and it is probable that the central portion of the volume of heated gases sweeps through the flue and reaches the end of the boiler without coming in contact with the surface of the tube and without any sensible diminution in temperature. The gases in turning round the boiler-end come in contact with bends and angles and are thereby broken up and yield heat which would otherwise escape to the chimney unused.

This shows that the flame operates at a great disadvantage when travelling through a horizontal flue-tube in a direction parallel to its axis, and means should be adopted to diffuse the heat by breaking up and subdividing the stream of heated gases in order to cause fresh portions of the gases to come in contact with the surface of the tube, which is best effected by placing Galloway-tubes in the flue. If these tubes are properly arranged and placed at proper intervals the flame not only impinges against the face or surface of the tubes facing the fire, but where there is a good body of flame it laps well round the tubes, becomes split up, and is brought into contact with the main flue-tube.

The Flame in passing through Small Tubes, such as boiler-tubes, assumes a winding manner which considerably adds to their efficiency as heating surface, because the whirling motion brings fresh portions of the products of combustion against the surface of the tubes. The flame will pass right through boiler-tubes of ordinary diameter and length, with a proper supply of air to the furnace and a moderate draught.

The Products of Combustion Escape to the Chimney at a High Temperature, that being necessary to obtain a good natural draught, the effect of heat in the chimney being to increase the draught and to accelerate combustion. The heated gases pass up the chimney on an average at a temperature of from 400° to 425° Fahr. in boilers of portable engines; at from 552° to 600° Fahr. in Cornish and Lancashire boilers and marine boilers, with natural draught, and at from 400° to 450° Fahr., with well-arranged forced draught. The temperature in the smoke-box of the boiler of a locomotive engine generally varies from 450° to 600° Fahr. when the engine is working lightly, and from 800° to 900° Fahr. when the engine is hard worked, according to the load. In the case of boilers with forced draught the products of combustion may be utilised in heating the air supplied to the furnace; if the air abstract from the escaping gases, in passing through a heating chamber placed in the chimney, say 200° of heat, and 18 lbs. of air be required per lb. of coal, then $200° \times 18$ lbs. $\times \cdot 238$ specific heat of air $= 856 \cdot 8$ units of heat per lb. of coal consumed, will be recovered by the air from the waste gases which would otherwise escape up the chimney and be wasted.

The Proportion of the Total Heat Developed on the Fire-grate which is absorbed by the surfaces surrounding the fire, varies considerably in different boilers. Peclet found that one-half the heat of a fire of peat was given out by radiation. The radiant power of coal and other fuels may be taken at the same as that of peat, so that of the total heating power of coal and other fuels one-half the heat is given out by radiation and the remainder is absorbed by the air in passing through the fire, which in the case of a steam boiler is partly recovered by passing the products of combustion through flues surrounded by absorbing or heating surfaces.

The Temperature of the Products of Combustion at Different Stages between the Fire and the Chimney, may be ascertained by a series of calculations which may be illustrated by taking the case of the plain cylindrical egg-ended steam-boiler, shown in Fig. 51, and calculating the temperature of the gaseous products of combustion at various points of the length of the boiler. Then, let the external diameter of the boiler be 4 feet; length of boiler, 37 feet; length of fire-grate, 5 feet; and width of fire-grate, 4 feet. Taking the maximum calorific power of coal of average quality at 14300 units per lb., and allowing 10 per cent. for imperfect combustion, it leaves $14300 - 1430 = 12870$ units, the available quantity of heat. The quantity of air required with natural draught will be 24 lbs. for each pound of coal, and the temperature of the air on admission to the fire will be, say, 60° Fahr. In externally-fired boilers only one-fourth of the heat developed by the combustion of fuel on the fire-grate is given out by radiation to the boiler over the fire, the remainder, or three-fourths, being carried off by the air which passes through the furnace. Therefore, $12870 \div 4 = 3217$ units of heat will be absorbed by the surface of the boiler, and the remainder, or $12870 - 3217 = 9653$ units will escape past the bridge of the furnace. The temperature of the air and the gaseous products of combustion when passing from the fire over the furnace-bridge, will be

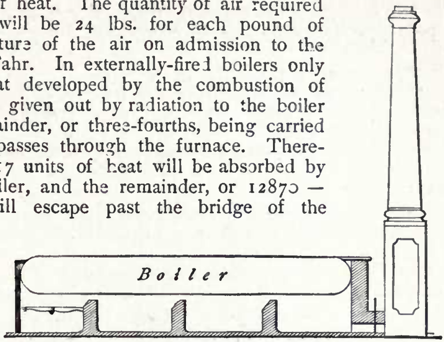


Fig. 51.—Method of setting plain cylindrical or egg-shaped boilers.

$$= \frac{9653 \text{ units of heat}}{24 \text{ lbs. of air} \times \cdot 238 \text{ the specific heat of air}} + 60^\circ \text{ initial temperature} = 1750^\circ \text{ Fahr.}$$

If the absolute pressure of the steam in the boiler be 70 lbs. per square inch the temperature of the boiler will be 303° Fahr., and the quantity of heat abstracted from the gases on their way to the chimney at different points in the length of the boiler, will be in proportion to the difference between the temperature of the escaping gases at each point, and the temperature of the boiler. As the gases give out the greatest heat after passing the furnace-bridge for a distance equal to about the length of the fire-grate, it may be assumed that one-fourth of this difference of temperature will be given up from point to point for that distance, and one-eighth of this difference from point to point for the remainder of the length of the boiler, and the temperature at the different points may be calculated as follows, supposing no heat is lost in heating the setting:—

The temperature of the gases at the furnace-bridge = 1750° Fahr.; temperature of boiler = 303° Fahr.,

Then, $1750 - \frac{1750 - 303}{4} = 1388^\circ$ Fahr., the temperature of the first point beyond the bridge.

$1388 - \frac{1388 - 303}{4} = 1117^\circ$ Fahr., the temperature of the second point beyond the bridge.

$1117 - \frac{1117 - 303}{8} = 1005^\circ$ Fahr., the temperature of the third point beyond the bridge.

$1005 - \frac{1005 - 303}{8} = 917^\circ$ Fahr., the temperature of the fourth point beyond the bridge.

And so on for the remaining points in the length of the boiler, the results being as follows :—

Temperature of the Gaseous Products of Combustion at, and beyond the Bridge of the Furnace of the Boiler shown in Fig. 51 :—

	Degrees Fahr.
Temperature at the bridge of the furnace	1750
Temperature at a distance of 2 ft. 6 in. beyond the bridge . .	1388
" " " 5 " 0 " " " . .	1117
" " " 8 " 0 " " " . .	1005
" " " 11 " 0 " " " . .	917
" " " 14 " 0 " " " . .	840
" " " 17 " 0 " " " . .	773
" " " 20 " 0 " " " . .	714
" " " 23 " 0 " " " . .	663
" " " 26 " 0 " " " . .	618
" " " 29 " 0 " " " . .	578
" " " 32 " 0 at end of boiler or entrance to the flue of the chimney	544

The walls shown in Fig. 51 behind the furnace-bridge, form baffling-bridges to insure the contact of the gaseous products of combustion with the heating or absorbing surface of the boiler, and to retard the escape of the gases and promote their admixture with the air. This is the best method of setting plain cylindrical or egg-ended boilers of proper length. It will be seen from the above temperatures at various points in the length of the boiler, that it is necessary to make this class of boiler of sufficient length to prevent the gases escaping to the chimney before they have been thoroughly deprived of their heat. For instance, if the boiler had only extended to a length of 11 feet beyond the bridge, the gases would escape to the chimney at 917° Fahr., causing a loss of $917-544 = 373°$ of heat. The quantity of heat lost when the gases enter the chimney at a temperature of 544° Fahr. is $= 544-60 \times$ (24 lbs. of air $\times \cdot 238$ specific heat of air) $= 2765$ units, and the quantity of heat lost if the gases entered the chimney at 917° Fahr. would be $= 917 - 60 \times (24 \times \cdot 238) = 4896$ units, showing a loss of heat $= 4896-2765 = 2131$ units greater with the short than with the long boiler above described, which allows the gases to escape at 32 feet beyond the bridge at a sufficiently high temperature to ensure a proper draught.

Absorption is the power of taking in heat. Coated surfaces absorb more readily than uncoated surfaces. Good radiators of heat are good absorbers, and bad radiators are bad absorbers of heat. For instance, lampblack is a good absorber of heat and also a good radiator, hence the same numbers which express the relative radiant powers of any series of substances will also express their relative absorbent powers.

Heat is Communicated from one body to another in three ways :—

1st. By direct contact, called conduction.
2nd. By right lines, called radiation.
3rd. By carrying, called convection.

Conduction is the power that substances possess of conducting heat from other bodies in immediate contact with them, the power varying according to the nature of the substance. The conducting power of metals when pure is nearly the same for heat and electricity.

Woods conduct better in the direction of the fibre than across the fibre. The conducting power of the bark of a tree is lower than that of the wood.

Non-Conductors of Heat, or, more properly, slow or bad conductors or retainers of heat are the following, viz., stones, glass, terra-cotta, brickwork, straw, white paper, wool, hair, felt, &c., each being successively lower in their conducting powers according to careful experiments, the results of which are given in the following Table :—

Table 31.—Heat Conducting Power of Various Substances by Wiedmann and Frantz, and Others.

Silver	100	Steel	11·6
Copper	73·6	Lead	8·5
Gold	53·2	Platinum	8·4
Tin	14·5	Rose's alloy	2·8
Iron	11·9	Bismuth	1·8
Stone	100	Wood-ashes	3·7
Glass	50	Mahogany sawdust	3·6
Terra-cotta	30	Hemp	3·3
Brickwork	25	Cork-chips	3·1
Coal-ashes, fine	10·8	White writing paper	2·5
Chalk, powdered	6·3	Cotton wool or lint	2·4
Chaff	5·6	Sheep's wool	2·3
Bran	4·5	Hair felt	2·1
Straw, chopped	4·0	Eiderdown	2·0

Radiation is the heating effects produced by direct rays from a hot body through space, as light is from that of a luminous body, the heating effect being inversely the square of the distance from the hot body. If at any given distance a certain heating effect is produced, at twice that distance the power or effect will be one-fourth, and at three times the distance one-ninth; but the radiating powers of substances vary in effect with the nature and colour of their surfaces. Thus, polished iron does not, at the same temperature, give out so much heat by radiation as when its surface is in a corroded state. Hence the cylinder covers of engines should be kept constantly bright to prevent loss of heat by radiation. A white surface will not diffuse heat by radiation equal in quantity to that of a darker colour. Bodies coated with a thin plate of bright metal suffer very little loss from radiation of heat. Leslie found that a tin vessel filled with hot water and covered with lampblack possessed a radiating power of 100. The following Table contains the comparative radiating power of other materials.

Table 32.—Radiating Power of Materials.

Surface covered with lampblack	100
,, ,, white lead	100
,, ,, writing paper	98
,, ,, resin	96
,, ,, ordinary white glass	91
,, ,, China ink	88
,, ,, red lead	80
,, ,, plumbago	75
,, ,, isinglass	75
,, ,, tarnished lead	44
,, ,, scratched tin	22
,, ,, mercury	20
,, ,, polished lead	19
,, ,, polished iron	15
,, ,, tin-plate	12
,, ,, gold, silver, copper, each	12

Other authorities consider that the radiating power of the metals given above are about 70 per cent. too large.

The Loss of Heat by Radiation from Steam-pipes is considerable, even when the pipes are clothed with a non-conducting material, as will be seen from the following Table, which contains the results of careful experiments to determine the loss by radiation from steam-pipes protected with various non-conducting materials. The best results were obtained with hair-felt surrounded with a covering of burlap, and it was found that coverings owe their efficiency chiefly to their interstices being filled with air, but not being open enough to permit a circulation.

Table 33.—Radiation from Steam-pipes Covered with Various Non-conducting Materials, the Results being given in Pound-Fahrenheit Heat-Units Radiated per Hour, from a Surface of One Square Foot, the results of Experiments by Professor Ordway.

Material.	Diameter, Inches.	Weight per Foot, Ounces Avoir.	Pound-Fahr., Heat Units per Sq. Ft. per Hour.
Hair-felt, burlap	5·37	21·4	51·0
,, ,,	4·5	13·2	56·6
Asbestos-paper, hair felt, duck	4·5	17·3	59·7
,, ,, paper-duck	4·5	19·9	62·5
,, ,, ,,	5·0	18·4	63·8
,, ,, ,,	4·0	17·2	70·6
,, ,, ,,	4·5	16·1	77·9
Fossil-meal with 12 per cent. cork dust	5·12	24·7	64·8
,, ,, strawboard	5·12	20·8	68·4
Paste of fossil-meal and hair	4·75	60·7	83·9
,, ,, ,,	4·25	31·9	91·0
,, ,, ,,	4·12	26·9	93·1
,, ,, asbestos	4·5	34·4	117·5

Table 33 *continued.*—RADIATION FROM STEAM-PIPES, &c.

Material.	Diameter, Inches.	Weight per Foot, Ounces Avoir.	Pound-Fahr., Heat Units per Sq. Ft. per Hour.
Best slag-wool, strawboard	4·75	13·2	66·6
Poor ,, ,,	4·25	24·1	90·5
Asbestos-paper, slag-wool, asbestos-paper	4·00	26·2	117·0
Air space, tin-plate, hair felt, duck	4·75	20·8	69·7
,, strawboard, cork, strawboard	5·12	12·0	79·0
,, ,, hair-felt, drilling	5·12	...	79·0
,, ,, ,, ,,	5·12	...	79·5
,, asbestos-paper, paper-cylinder	4·75	29·3	81·1
,, strawboard, hair-felt	4·75	12·0	82·1
,, ,, sphagnam, cotton-cloth	4·75	10·6	85·8
,, ,, pine - turnings, strawboard	5·12	14·0	89·6
,, ,, paper-pulp, strawboard	5·12	13·7	91·7
,, ,, rice-chaff, strawboard	5·12	17·1	94·6
,, ,, cotton-roving stair-pad	4·75	14·6	96·0
,, ,, wire-netting, paste of fossil-meal, asbestos	5·00	41·0	113·4
,, multiplex, with strawboard and cords	4·37	28·1	115·3
,, strawboard, cotton-roving	4·25	8·3	124·0
,, simple	2·94	...	195·4
,, ,,	4·75	...	200·0
Silicated cork-chips, drilling	5·25	14·8	71·4
,, ,, ,,	5·25	14·7	72·7
Air space, strawboard, cork-chips, strawboard	5·12	12·0	79·0
Cork in strips	3·62	6·7	105·3
Silicated pine-charcoal	5·25	34·5	78·3
,, hard wood	5·00	41·9	97·8
,, rice-chaff, cotton-cloth	4·75	22·7	80·2
,, ,, ,, ,,	5·25	27·5	85·4
Rice-chaff, strawboard	5·12	16·7	87·0
Air space, strawboard, rice-chaff, strawboard	5·12	17·1	94·5
Rice-chaff, strawboard	3·75	8·4	109·5
Paper made of wool-pulp and wool-waste	4·37	30·5	78·1
Cotton-batting	4·00	6·0	79·7
Air space, asbestos paper, paper cylinder	4·75	29·3	81·1
Straw rope, four thicknesses cotton cloth	4·50	20·2	96·9
Silicated cotton-seed hulls, drilling	5·12	51·0	107·6
Blotting-paper	4·00	28·9	127·8
Asbestos-paper	4·00	...	131·2
,, ,,	2·87	...	223·7
Anthracite coal-ashes (fine), strawboard	5·00	37·6	96·8
Bituminous ,, ,,	5·00	41·2	98·4
Anthracite ,, (coarse) ,,	5·12	57·3	131·2
Paste of fossil-meal and hair	4·75	60·7	83·9
,, ,,	4·25	31·9	91·0
,, ,,	4·12	26·9	93·1
Air space, fossil-meal and hair, asbestos	5·00	41·0	113·4
,, ,,	4·50	35·4	117·5
Carbon, plaster-of-paris, flour and hair.	4·50	33·0	106·8
Clay and vegetable-fibre	5·25	94·1	146·0
Anthracite-ashes, plaster-of-paris, flour and hair	4·75	79·2	155·4
Clay and vegetable-fibre	4·25	65·2	205·3
Naked pipe	...	...	1555·1

Convection is the power possessed by fluids of conveying heat acquired at one place to another place. Convection is caused by currents both in air and water. Smoke ascends the chimney, and ventilation is caused by the same principle. Gaseous bodies, from the great mobility of their particles, are the most rapid conveyers, although they are the slowest conductors of heat. Any body hotter than the air heats it and sets it in motion in an upward current, which may be seen rising from highly heated bodies, and the particles which rise are immediately replaced by the influx of other particles from every side. The slightest difference in temperature is sufficient to produce these effects, hence the rapidity with which air reduces solid bodies to its own temperature. A body colder than the air, such as a lump of ice, produces an opposite action; it cools the air in contact with it, which, becoming denser, descends in a continual stream, supplied by an influx of air from all sides to the ice, until the whole is melted.

Heat is distributed by Convection through the Water in a Boiler, the heat imparted to the outside of the plate from the furnace, is conducted through the plate, and the water in contact with it absorbs heat, expands, and rises from it, and colder water immediately descends, and occupies its place. Water, being a bad conductor of heat, can only be warmed very slowly by conduction. Owing to the low conducting power of water, the application of heat to its upper surface is of no effect in heating the mass of water beneath, and it may be boiled at the surface, while a lump of ice, fixed at the bottom, remains unmelted, as shown at Fig. 52.

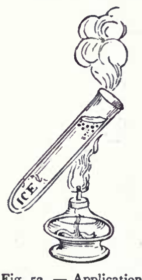


Fig. 52. — Application of heat to the surface of water.

Effect of Heat on Water.—There is no change of temperature in liquids under ordinary conditions without causing a displacement of particles. If heat be applied under a vessel of water, the particles near the bottom of the vessel being heated first and expanding, become specifically lighter and ascend; colder particles occupy their place, become heated and ascend in their turn, and thus a current is established, the heated particles rising up through the centre and colder particles descending at the sides. Hence ample area should be provided in the water-spaces of boilers for both ascending and descending currents. The heat is not conducted from particle to particle without displacement, as in the case of solids; but each particle, as fast as it receives a fresh accession of heat, conveys it to a distance, displacing other and colder particles in its progress.

When Heat is applied to the Surface of Water it is not diffused by convection, but creeps very slowly downwards by conduction. Hence water placed underneath a hot tube on being heated and becoming lighter clings to the surface above it and diffuses no heat downwards: for instance, in Cornish boilers, without Galloway tubes for promoting the circulation, the water under the flue-tube is frequently comparatively cold some time after the steam has been raised. It will be seen from this that the under portions of the internal flue-tubes of boilers are almost of no value for raising steam.

Circulation is caused by the difference of density of water in a boiler, due partly to difference of temperature, but principally to the action of steam-bubbles, which lighten the water, hence there is a difference of density of those portions which produce the most and the least steam.

SECTION II.

EVAPORATION; BOILER-SHELLS, BOILER-FURNACE-TUBES; BOARD OF TRADE, LLOYD'S, AND NUMEROUS OTHER RULES AND DATA FOR STEAM-BOILERS; BOILER-CONSTRUCTION; BOILER-EXPLOSIONS, ETC.

SECTION II.

EVAPORATION; BOILER-SHELLS, BOILER-FURNACE-TUBES; BOARD OF TRADE, LLOYD'S, AND NUMEROUS OTHER RULES AND DATA FOR STEAM-BOILERS; BOILER-CONSTRUCTION; BOILER-EXPLOSIONS, ETC.

EVAPORATION OF WATER TO STEAM IN STEAM-BOILERS.

The Evaporative Efficiency, E, of a perfect steam generator, having perfect combustion with the proper quantity of air, would be expressed by the following formula, in which T_1 and T_2 denote the temperatures of the furnace and the chimney in degrees Fahr. $E = \frac{T_1 - T_2}{T_1 + 461}$.

For instance, if the temperature of the boiler-furnace be 2400° Fahr., and that of the chimney $=550°$ Fahr., then the efficiency is:—

$$E = \frac{\text{Heat utilised}}{\text{Heat supplied}} = \frac{2400° - 550°}{2400° + 461°} = \cdot 65,$$

that is, the heat utilised is 65 per cent. of the heat supplied, the maximum efficiency theoretically possible with this range of temperature.

The Actual Evaporative Efficiency of a Steam-Boiler is measured by the amount of the total heat of combustion absorbed into the boiler and applied to the evaporation of water to steam.

$$\text{The actual efficiency} = \frac{\text{Number of thermal units transmitted to the water,}}{\text{Calorific power of the fuel.}}$$

The Evaporative Power of a Steam-Boiler is expressed by the quantity of water evaporated to steam per hour.

The Total Heat of Saturated Steam or the total heat of evaporation is the sum of the latent and sensible heats of evaporation, or the quantity of heat necessary to raise 1lb. of water from 32° Fahr., the freezing-point, to a particular temperature and to evaporate it at that temperature.

The Sensible Heat is that required to raise the temperature of water from the freezing-point to the temperature of ebullition: and the latent heat is that required to evaporate the water at the given temperature, or the heat which disappears in effecting the conversion of water into vapour. Liquids on the application of heat absorb a quantity of heat in passing from water to steam, which remains latent in the steam. This heat can be restored to sensibility and communicated to other bodies, when the vapour is condensed into liquid.

The Latent Heat, L, of Saturated Steam, at any given temperature, T, in degrees Fahr., may be found by this *Rule:* $L=966-\cdot 7(T-212^\circ)$.

Example: Required the latent heat of steam of 43 lbs. absolute pressure, the temperature of which is 272° Fahr. Then $272-212\times\cdot 7=42^\circ$, and $966^\circ-42^\circ=924$ units of latent heat per lb. of steam.

Total Heat of Steam.—The sensible heat required to raise the temperature of water from the freezing to the boiling point is 212° Fahr.—32° Fahr.=180 thermal units, the latent heat of evaporation of 1 lb. of steam at atmospheric pressure is 966, hence the total heat in 1 lb. of steam at atmospheric pressure is $966+180=1146$ thermal units, being the amount of heat expended in the generation of 1 lb. of saturated steam at 212° Fahr. from water at 32° Fahr.

The Total Heat, H, incorporated in 1 lb of saturated steam at the temperature, T, is equal to the latent heat of evaporation of steam at nearly 32° Fahr., increased by the product ·305 T. It is expressed by the formula:—$H=1082+\cdot 305\ T$.

Example: Required the total heat of steam at a temperature of 310° Fahr. Then $310\times\cdot 305=94\cdot 55$ and $1082+94\cdot 55=1176\cdot 55$, or in round numbers there are 1177 total number of units of heat in 1 lb of steam at 310° Fahr., or 77 lbs. absolute pressure.

The Volume of Steam, or number of cubic feet of steam, evaporated from 1 cubic foot of water, compared with that of water at 39° Fahr., may be calculated by the following formula of Fairbairn and Tate:—

Let V=the volume of saturated steam at the pressure, P, measured by the height of a column of mercury in inches.

$$\text{Then } V = 25\cdot 62+\frac{49513}{P+\cdot 72}.$$

Example: Required the relative volume of steam of 31 lbs. absolute pressure, or 63·15 inches pressure in inches of mercury.

Then $63\cdot 15+\cdot 72=63\cdot 87$ and $49513\div 63\cdot 87=775\cdot 23$ and $775\cdot 23+25\cdot 62=800\cdot 85$ relative volume.

The Pressure of Steam measured in Inches of Mercury may be found by the converse of the last formula as follows:—

$$P=\frac{49513}{V-25\cdot 62}\quad\cdot 72.$$

Example: Required the pressure in inches of mercury of steam of 80·085 relative volume as given in the last example.

Then $800\cdot 85-25\cdot 62=775\cdot 23$, and $49513\div 775\cdot 33=63\cdot 87$, and $63\cdot 87-\cdot 72=63\cdot 15$ inches of mercury, the pressure of the steam.

The Efficiency of a Steam-boiler is usually expressed by the weight of water evaporated to steam per lb. of fuel, from and at 212° Fahr.

The Equivalent Weight of Water Evaporated to Steam from and at 212° Fahr., the weight of water being supplied at a given temperature and evaporated at a given pressure, may be calculated as follows:—

Let H = the total heat of the steam at the given absolute pressure, which may be found in Table 79: t = the temperature of the feed-water; F =

the factor of evaporation; W = the actual evaporation per lb. of fuel; W_1 = the equivalent weight of water evaporated as from and at 212° Fahr.

C = a constant divisor = 966 for feed-water supplied at 212° Fahr., this being the standard usually adopted in Boiler-Tests. When the equivalent weight of water is to be evaporated as from a less temperature than 212° Fahr., then—

C = a constant divisor = 1078 for feed-water supplied at 100° Fahr.
C = do. 1116 do. do. 62° do.

The water in each case to be taken as evaporated at 212° Fahr.

Then $F = \frac{H+32-t}{C}$. and $W_1 = W \times \frac{H+32-t}{C}$.

Example: Steam is generated at 65 lbs. per square inch absolute pressure, the temperature of the feed-water is 100° Fahr., and 8lbs. of water are evaporated per lb. of coal. Required the equivalent weight of water evaporated as from and at 212° Fahr. H=1173 units of heat per lb. in steam of 65 lbs. absolute pressure.

Then $1173+32=1205-100=1105\div966=1{\cdot}143$, the factor of evaporation, and 8 lbs. $\times$ 1·143 = 9·144 lbs., the equivalent weight of water evaporated per lb. of fuel.

The Weight of a Cubic Foot of Steam at various temperatures is obtained by dividing 62·5 the weight of a cubic foot of water by the relative volume of steam.

Example: Required the weight in lb. per cubic foot of steam of 30 lbs. absolute pressure, the volume of which compared with one cubic foot of water at 39° Fahr. is 827 cubic feet.

Then $62{\cdot}5\div827={\cdot}0755$ lb. weight of one cubic foot.

Evaporative Power of Fuel.—The evaporation of 1 lb. of water at a temperature when evaporisation commences, or 212° Fahr. to steam of the same temperature, requires the addition of 966 units of heat. Therefore if the maximum calorific power of the fuel, or the number of thermal units it will develop per lb. in burning, be divided by 966, it will give the number of lbs. of water that each lb. of fuel is theoretically capable of evaporating to steam from and at 212° Fahr.

If the temperature of the water be less than 212° Fahr., say 100°, a further quantity of heat will be required amounting to $212-100=112$ thermal units, and this quantity must be added to the divisor, increasing it to $966+112=1078$.

Example: Required the quantity of water in lbs. which coal, having a total heat of combustion of 14297 thermal units, is capable of evaporating for each lb. of coal burnt. Then $14297\div966=14{\cdot}8$ lbs of water.

The heat developed by hydrogen is 4·31 times as great as that of carbon, hence the evaporative power of any fuel may be found by the following formula, where C,H,O represent the constituents, carbon, hydrogen, and oxygen, U = the thermal units developed by the fuel, and E represents its evaporative power.

$$\text{Then } E = \frac{14500}{966} \times \left\{ C + 4{\cdot}31\left(H - \frac{O}{8}\right)\right\} = 15\left\{ C + 4.31\left(H - \frac{O}{8}\right)\right.$$

The quantity $\frac{O}{8}$ represents the deduction to be made from the constituent hydrogen, or that portion which forms steam with the constituent oxygen. The number 14500 is the thermal units developed by one lb. of carbon in burning to carbonic acid.

Example: Required the evaporative power of coal composed of ·86 carbon; ·05 hydrogen; ·03 oxygen.

$$\text{Then } 15 \left\{ \cdot 86 \text{ carbon} + 4\cdot 31 \left(\cdot 05 \text{ hydrogen} - \frac{\cdot 03}{8} \text{ oxygen} \right) \right.$$

$= \cdot 03 \div 8 = \cdot 0072 - \cdot 05 = \cdot 0428 \times 4\cdot 31 = \cdot 184 + \cdot 86 = 1\cdot 044 \times 15 = 15\cdot 66$ lbs. of water per lb. of fuel, the maximum theoretical evaporative power of that coal, which is much greater than that realised in practice.

The Theoretical Evaporative Power of Coal and other fuels in lbs. of water evaporated per lb. of fuel from and at 212° Fahr. is given in the following Table.

Table 34.—Theoretical Evaporative Power of Fuel in lbs. of Water Evaporated per lb. of Fuel.

Description of Fuel.	Lbs. of Water evaporated to Steam from and at 212° Fahr.	Description of Fuel.	Lbs. of Water evaporated to Steam from and at 212° Fahr.
	lbs.		lbs.
Petroleum	22	Newcastle coal, good, average	14·80
Creosote-oil or tar-refuse	18	Lancashire coal	14·60
Asphalte	17·18	Derbyshire coal	14·50
Welsh coal, Ebbw Vale	16·82	Yorkshire coal	14·38
Welsh coal, Powell's Duffryn	16·34	Scotch coal, best	14·15
Welsh coal, best Aberdare	15·93	English coal, average of a number of samples of good steam-coal	14·00
Welsh coal, Llangennech	15·25	Coke	13·00
Welsh coal, Anthracite (Jones & Aubrey)	14·59	Mixture of 1 part each of broken coke and good slack	10·00
Welsh coal, average of a number of samples of good steam-coal	15·50	Peat, dry	10·00
Patent fuels	15·48	Breeze and slack, equal parts	8·00
Bituminous lignite	14·90	Wood, dry	7·50
Perfect lignite	12·00	Tan-refuse—Oak bark	6·50
Imperfect lignite	10·12	Straw	4·00
Best Newcastle steam-coal	15·30		

The Actual Evaporative Power of various Steam-Coals of good quality, and other fuels, the average results of a number of boiler-tests with natural draught, in lbs. of water evaporated from and at 212° Fahr., per lb. of fuel, is given in the following Table:—

Table 35.—Actual Evaporative Power of various Steam-Coals and other Fuels, with Natural Draught.

Description of Fuel.	Lbs. of Water evaporated per lb. of Fuel.
	lbs.
Welsh steam-coal	12·50
Crude petroleum-oil	12·25
Best Newcastle steam-coal	12·00
Best Lancashire steam-coal	11·90
Creosote-oil, or residue from the distillation of tar	11·10
Newcastle steam-coal, good average	11·00
Lancashire steam-coal, good average	10·85
Yorkshire steam-coal	10
Coal, good, average of a number of samples from various districts	9
Coal, medium quality, average of a number of samples from various districts	7½
Best slack, average of a number of samples from various districts	6½
Common slack, average of a number of samples from various districts	6
Breeze and common slack, mixed half and half by weight	5½
Tan-refuse—Oak bark	5
Coal-dust and sawdust, mixed half and half by weight	4½

The Actual Evaporative Power of Fuel in Practice is much less than the theoretical power. The total heat which may be developed by the combustion of 1 lb. of good coal will theoretically evaporate 14 lbs. of water from 100° Fahr. to 212° Fahr. With a clean boiler, good coal and skilful firing, from 10 to 12 lbs. of water at 212° Fahr., have been evaporated per lb. of coal with natural draught, and a well-arranged boiler should evaporate 10 lbs. of water per lb. of coal. Few boilers evaporate more than from 8 to 9 lbs. of water per lb. of coal burnt, and a great many only evaporate 7 lbs. of water per lb. of coal, or one-half the theoretical quantity. The difference between the theoretical and actual evaporation is chiefly due to radiation, imperfect combustion, and to the quantity of heat necessarily expended in producing chimney-draught when natural draught is used for combustion. Therefore it is not possible in practice to use all the available heat.

In a careful test of a Cornish boiler with natural draught, 812 lbs. of good small coal evaporated 6691 lbs. of water at 55° Fahr. to steam at 212° Fahr. $= \dfrac{(1178^\circ - 55^\circ) \times 6691 \text{ lbs. of water}}{812 \text{ lbs. of coal burnt}} = 9254$ units of heat per lb. of coal, and equal to $6691 \div 812 = 8·24$ lbs. of water evaporated per lb. of coal.

The Loss of Heat in a Boiler-Furnace is caused chiefly by fuel falling unburnt through the bars, and by gases escaping at a high temperature to the chimney. The available heat left for the generation of steam, being only about 60 per cent. of the heat theoretically developed by the fuel, in internally-fired boilers; the loss being 10 per cent. greater in externally-fired boilers. The evaporative efficiency varies considerably in different classes of boilers.

Effect of Blast-Pressure in a Chimney.—Experiments have been made with steam-boilers to ascertain the effect of a steam-blast in the chimney, which resulted in showing that it increased the consumption of coal on the fire grate from 40 to 50 per cent. over natural draught, with an increase of efficient power not exceeding 15 per cent., and that a steam jet was very wasteful. Mr. Longridge, in some experiments, also found the blast-pipe in a chimney to be wasteful, and that only about one-eleventh of the power escaping is utilised in drawing air through the tubes. A strong air-blast has also been tried in a chimney which scarcely gave one-half as good a result as was obtained from forced draught with an air pressure of two inches of water in the stokehold.

The Evaporative Performance of Steam-Boilers in Practice is given in the following Table, which contains the average results of a number of tests of well-arranged steam-boilers with natural draught.

Table 36.—Evaporative Performance of Steam-Boilers in lbs. of Water Evaporated per lb. of Coal from and at 212° Fahr.

Description of Steam-Boiler.	Lbs. of Water evaporated per lb. of Coal.	Description of Steam-Boiler.	Lbs. of Water evaporated per lb. of Coal.
	lbs.		lbs.
Galloway boiler . .	11·72	The Field boiler . .	10·00
Water-tube boiler . .	11·00	Marine boiler . .	8·75
Lancashire boiler, with Galloway tubes . .	10·75	Cornish boiler . . .	8·00
Portable-engine boiler .	10·50	Torpedo-boat boiler, locomotive type .	7·50
Locomotive, compound, with coke . . .	10·00	Vertical tubular boilers .	7·00
Locomotive, ordinary, with coke . . .	10·00	Vertical cross-tube boilers . . .	6·25
The nozzle boiler . .	10·00	Egg-ended boiler, externally fired . .	6·20

An evaporative efficiency of about 90 per cent. has been obtained as the result of tests of boilers of the locomotive type.

The Efficiency of the Boilers in the above Table may be ascertained by dividing the actual evaporative power by the theoretical evaporative power. Taking the theoretical evaporative power of 1 lb. of average good coal at 14 lbs. of water per lb. of coal, the evaporative performance of the Galloway boiler shows an actual efficiency of $\frac{11·72}{14} = ·83$. The efficiency of the Lancashire boiler, $\frac{10·75}{14} = ·76$; the portable-engine boiler, $\frac{10.5}{14}=.75$; the marine boiler, $\frac{8.75}{14}=.62$; the Cornish boiler, $\frac{8}{14}=.57$, and the vertical boiler with cross tubes, $\frac{6.25}{14}=.44$, or less than one-half the theoretical evaporative power. The theoretical evaporative power of coke is 13 lbs. of water per lb. of coke, and the efficiency of ordinary locomotives is $\frac{10}{13}=.76$.

Evaporative Efficiency of Locomotive-Engine Boilers.—The evaporative performance of a well proportioned locomotive boiler is given in Table 37, which contains the results obtained by Mr. Stroudley in a careful trial with one of his engines, the feed-water being heated to 188° Fahr., from which it will be seen that $10\frac{1}{2}$ lbs. of water were evaporated per lb. of Welsh coal.

Table 37.—EVAPORATIVE PERFORMANCE OF A LOCOMOTIVE-ENGINE BOILER.

Trip First, London Bridge to Brighton, 6·5 p.m.				
Average Coaches.	Fuel, lbs. per Mile.	Lbs. of Water evaporated per Foot of Grate Surface.	Lbs. of Water evaporated per Foot of Heating Surface.	Lbs. of Water per lb. of Coal.
15·6	23·2	12·30	·193	10·08
Trip Second, London Bridge to Brighton, 5 p.m.				
10·05	19·96	11·09	·173	10·50
Fire-grate Area. Square Feet.	Heating Surface of Tubes. Square Feet.	Heating Surface of Fire-box. Square Feet.	Tubes. Diameter, $1\frac{3}{4}$ in.	Total Heating Surface.
19	1100	110	206	1210
Description of fuel used on each trip—*Welsh coal.*				

The Evaporative Performance of a Locomotive-Engine Boiler under different arrangements in the supply of feed-water, are given in Table 38, which contains the results of an experiment by Mr. Drummond to determine the relative economic efficiency of a locomotive boiler when fed with feed-water by a feed-pump with water-heater, and when fed by an injector: from which it will be seen that there is very little difference in the weight of water evaporated per lb. of coal, whether the boiler is fed by a pump or by an injector.

Table 38.—Results of Trials on the North British Railway, showing the Evaporative Efficiency of a Locomotive-Engine Boiler when Fed with a Pump with Feed-Water Heater and when Fed with an Injector.

Engine.		Train.		Average per trip, Glasgow to Carlisle.			
18 in. by 26 in. Cylinders. 5 ft. Wheel. 150 lb. Boiler press. Adhesive weight, 39½ tons.	Reference Number.	Description.	Average Speed whilst in Motion.	No. of Stoppages.	Time Shunting.	Time Standing.	Average weight train, *i.e.*, van, wagons, and load.
			Miles per hour.		h. m.	h. m.	Tons.
Pumps, exhaust cocks under foot-plate	281	Midnight and mid-day through goods . . .	18·4	16·4	2 28	2 50	267
Do. do.	287	Light express goods . .	21·07	13·25	1 17	2 42	About 260
Pumps, exhaust cocks under cylinders	305	Mid-day through goods .	19·8	17·5	2 45	2 20	244·6
Injector	292	Do.	20·5	16·2	2 30	2 30	230·6
Injector	292	Midnight and mid-day through goods . . .	19·6	16·7	2 22	2 29	223·5

Consumption of Coal.					Water.									Mileage.	
Reference Number.	Lbs. of Coal per Train Mile.	Lbs. of Coal per Mile per Ton of Train.	Lbs. of Coal per Mile per Ton of Engine and Train.	Lbs. of Water per lbs. Coal.	Average number of Gallons consumed. Glasgow to Carlisle.	Ditto, including Condensed Steam.	Lbs. Water per Mile, including Condensed Steam.	Lbs. Water per Mile per Ton of Train.	Lbs. Water per Mile per Ton of Engine and Train.	Average Heat of Water supplied to Boiler.	Maximum ditto.	Minimum ditto.	Average temperature Water was supplied at.	Train Mileage on which results are based.	Mileage assisted up Inclines.
	lb.	lb.	lb.	lb.	Gal.	Gal.	lb.	lb.	lb.	deg. Fah.	deg Fah.	deg. Fah.	deg. Fah.	Miles.	Miles.
281	49·5	·185	·149	8·95	5910	6377	442·8	1·67	1·33	127	200	60	44	864	..
287	57·6	·22	·177	7·73	6080	6410	445·1	1·72	1·37	102½	172	56½	44	576	..
305	56·39	·230	·182	8·24	6313	6692	464·7	1·90	1·496	107¾	188	60	45	864	85½
292	53·36	·231	·180	8·21	6347	..	438·1	1·90	1·477	..	..	..	45	869	98⅝
292	..	..	..	..	6385	..	443·4	1·984	1·532	..	..	..	45	288	..

The Consumption of Coal was measured in numbers 305 and 292. In numbers 281 and 287, the consumption of Coal was taken from Monthly Sheets.

The Rate of Evaporation depends upon the quality of the coal, and also on the force of the draught, as the temperature of the furnace depends upon the rate of the supply of air. More steam can be raised in a given time per square foot of fire-grate from burning semi-bituminous, or Newcastle steam-coals, than from slightly bituminous, or Welsh steam coals, as will be seen from the following Table :—

The rate of Evaporation of various Coals, in cubic feet of water from 212° Fahr., evaporated per square foot of fire-grate per hour, with natural draught, is as follows :—

	Cubic feet of Water from 212° Fahr.	
Best Lancashire Steam-coal will evaporate . .	5·25	Per square foot of fire-grate per hour.
Lancashire Steam-coal, good, average, will evaporate	5·00	„ „
Best Newcastle Steam-coal will evaporate . . .	5·10	„ „
Newcastle Steam-coal, good, average, will evaporate	4·60	„ „
Best Yorkshire Steam-coal will evaporate . .	4·50	„ „
Welsh Steam-coal will evaporate	4·20	„ „

The Average Maximum Evaporative Effect of Heating Surface is 21 lbs. of water per hour, or one cubic foot of water evaporated by three square feet of heating surface.

The Evaporative Power of a Square Foot of Heating Surface varies considerably in different classes of boilers, as well as in the same class of boiler owing to its arrangement, condition and position.

The Evaporative Performance of Steam-Boilers per Square Foot of total Heating Surface per Hour is given in the following Table, which contains the average results of a number of tests of well-arranged steam-boilers.

Table 39.—Evaporative Performance of Steam-Boilers, all with Natural Draught except where otherwise stated, in lbs. of Water Evaporated per square foot of total Heating Surface per Hour.

Description of Steam Boiler.	Lbs. of Water evaporated per Square Foot of Heating Surface per Hour.	Description of Steam Boiler.	Lbs. of Water evaporated per Square Foot of Heating Surface per Hour.
	lbs.		lbs.
Egg-ended boiler, externally fired . .	2·10	Marine boilers . .	5
Vertical boiler, with cross-tubes . . .	2·20	Marine boilers, with forced draught . .	13 to 15
Vertical tubular boiler .	2·25	Locomotive boilers burning coal . . .	8 to 9
Cornish boiler . .	2·30	Locomotive boilers burning coal, average .	8
Water-tube boiler . .	2·40	Locomotive boilers burning coke . . .	9 to 13
Lancashire boiler . .	2·50	Torpedo-boat boilers, locomotive type. .	18
Portable engine boiler .	2·75		
Galloway boiler . .	3·00		
The field-boiler . .	3·25		
The nozzle-boiler . .	4·00		

The Influence of Air-pressure on Evaporation, Combustion, and Temperature, is shown in Table 40, which contains the results of experiments with Thorneycroft's torpedo-boat boilers of the locomotive type, having air pressure in the stokehold. The boiler had a barrel 53 inches diameter, with 205 tubes of $1\frac{1}{3}$ inch diameter, and 6 feet long. Heating surface of fire-box 56 square feet. Total heating surface 618 square feet. Fire-grate surface 18·9 square feet. The pressure in the chimney was about equal to that of the atmosphere.

Table 40.—Evaporative Performance of Torpedo-Boat Boilers.

Air pressure in stokehold in inches of water	2	3	4	6
Air pressure in ash-pit in inches of water	1·47	2·29	3·26	5·25
Air pressure in furnace in inches of water	1·35	1·87	3·0	4·33
Temperature of feed-water, Fahr.	53·5°	57·°	54·°	56·°
Temperature in the funnel, Fahr.	1,037°	1,192°	1,260°	1,444°
Steam pressure above the atmosphere, in lbs. per square inch	117	117	115	115
Coal consumed per hour, in lbs.	925	1177	1472	1815
Coal consumed per square foot of fire-grate, in lbs.	49	62	78	96
Square feet of heating surface per pound of coal	·668	·525	·420	·340
Water evaporated per hour, total lbs.	6530	7770	9320	10,840
Water evaporated per pound of coal, lbs.	7·06	6·6	6·33	5·97
Evaporation per pound of coal reduced to equivalent at 212° Fahr. from 100°	7·61	7·08	6·81	6·41
Evaporation per hour per square foot of heating surface	10·8	12·9	15·5	18·
Duration of experiment in hours and minutes	h. m. 2 0	h. m. 2 7	h. m. 1 39	h. m. 1 27
Coal used, Nixon's Navigation.—Ashes, 9 per cent.				

It will be seen from the above Table that an increase of air pressure from 2 to 6 inches increased the consumption of coal per square foot of fire-grate from 49 lbs. to 96 lbs., and the evaporation per hour per square foot of heating surface from 10·8 lbs. to 18 lbs.

Closed Stokeholds worked under air-pressure are better ventilated than open stokeholds, with natural draught: the power of the draught is independent of the force and direction of the wind. The speed of the fans for supplying the air to the furnaces can be regulated as may be required for the power to be developed.

Table 41.*—Results of Evaporative Tests of two Boilers of Modified Locomotive Type for Torpedo-boat Chasers.

No. of Trial.	Duration of Trial.	Surfaces.		Mean Steam Pressure.	Water.		Water Evaporated. From and at 212 deg. per Hour.			
		Heating.	Grate.		Total.	Per Hour.	Per Hour.	Per lb. of Coal.	Per sq. ft. Grate.	Per sq. ft. Heating Surface.
	h. m.				lbs.	lbs.				
1	1 32	2232	52·5	125·5	44,969	29,336	35,937	6·969	685	16·1
2	1 23	2232	52·5	125	44,004	31,809	38,998	6·889	742·8	17·47
3	1 5	2232	52·5	120	36,264	33,474	40,905	6·566	779·1	18·32
4	1 41	2232	52·5	116·2	54,900	32,613	39,821	8·566	758·5	17·84

No. of Trial.	Air Pressure.			Revolutions of Fans.		Temperature.				Weather.
	In Stokehold.	In Ash-pit.	In Uptake.	Port.	Starboard.	Stokehold.	Uptake.	Atmosphere.	Feed.	Description.
1	3·54	3·13	0·65	1283	1150	75	1150	44	37	Light westerly wind; dry, but cold.
2	3·605	3·446	0·25	877	909	66	1163	37	36	Fresh easterly wind; snow falling.
3	3·07	3·36	0·633	925	1171	77·5	1210	40	39	Light easterly wind; dry.
4	2·0	3·706	0·812	769·2	1212	87·6	1200	50	40	Light westerly breeze; fine and dry.

No. of Trial.	Coals.				Class of Coals.
	Total.	Per Hour.	Per sq. ft. Grate.	Sq. ft. of Heating Surface per lb. Coal.	—
	lbs.				
1	7907	5165	98·3	0·432	Cowpen Coal Co.'s ordinary best steam coals, not hand-picked; rather damp and small.
2	7840	5667	107·9	—	Cowpen Coal Co.'s ordinary best, as above.
3	6720	6203	118·1	—	Cowpen Coal Co.'s ordinary best, as above.
4	7826	4648·8	88·5	—	Nixon Navigation; new, dry, and good.

No. of Trial.	Description of Fire-Bars.
1	Ordinary fire-bars. No opening through fire doors. Both fans discharging into stokehold.
2	Thin fire-bars. $\frac{1}{4}$ inch air spaces. No opening through fire doors. Fires thinner. Both fans discharging into stokehold.
3	Th'n fire-bars. $\frac{1}{8}$ inch air spaces. One fan on to ash-pit; other to stokehold. Air was getting into fire doors. One boiler steaming.
4	Thin fire-bars. $\frac{1}{8}$ inch air spaces. One fan on to ash-pit; other to stokehold. No air got through fire doors. One boiler steaming.

* The above Table was given by Mr. F. C. Marshall at a Meeting of the Institution of Naval Architects.

The quantity of Water apparently Evaporated to Steam in Boiler-Tests is frequently considerably greater than that actually evaporated, in consequence of loss of water from foaming and priming, resulting in a supply of wet steam. When steam contains more than 1 per cent. of moisture it is not termed dry steam.

Foaming, or a frothy condition of the surface of the water in a boiler, is due to the agitation caused by the steam in rising through scum on the surface of the water, from dirt, foreign matter, or impurities held in solution by the water. Foaming produces less moisture in steam than priming. An allowance of five per cent. is sufficient in most cases for foaming.

Priming, or the carrying of water with the steam, may be caused by defects in the design of a boiler; violent ebullition; a sudden withdrawal of steam; insufficient steam space; and to the reduction of pressure immediately underneath the point in a boiler whence steam is drawn, which creates a difference of pressure in different parts of the boiler, and produces currents of spray. An allowance of 10 per cent. may be made for priming.

The weight of Water in lbs. Evaporated per Square Foot of Fire-Grate Surface per Hour may be found by multiplying the rate of combustion in lbs. of coal burnt per square foot of fire-grate area per hour by the number of pounds of water evaporated to steam per pound of fuel. For *Example:* A marine boiler burning 30 lbs. of coal per square foot of fire-grate surface per hour and evaporating $8\frac{3}{4}$ lbs. of water per lb. of coal will evaporate $30 \times 8{\cdot}75 = 262\frac{1}{2}$ lbs. of water to steam per square foot of fire-grate per hour.

The Area of Fire-Grate Surface in square feet, may be found by dividing the weight of water in lbs. required to be evaporated per hour by the weight of water in lbs. evaporated per square foot of fire-grate per hour.

The Area of Fire-Grate in Square Feet per lb. of Coal Burnt per square foot of fire-grate per hour, for boilers of different classes, all with natural draught, except the torpedo-boat boilers, and locomotive boilers, averages as follows:—

Portable-engine boilers .	·090	square foot per lb. of coal burnt.
Cornish boilers . . .	·083	,, ,, ,,
Lancashire boilers . .	·066	,, ,, ,,
Marine boilers . . .	·044	,, ,, ,,
Torpedo-boat boilers . .	·014	,, ,, ,,
Locomotive-engine boilers .	·012	,, ,, ,,

The Total Heating Surface in proportion to the fire-grate surface of steam-boilers, or the surface ratio, is usually as follows:—

Egg-ended boilers; heating surface from . .	10 to 16 to 1	of grate surface.
Vertical boilers; heating surface from . . .	15 to 25 ,,	,,
Cornish and Lancashire boilers; heating surface from	16 to 25 ,,	,,
Multitubular boilers; heating surface from . .	20 to 60 ,,	,,
Marine boilers; heating surface from . . .	27 to 35 ,,	,,
Portable-engine boilers; heating surface from .	40 to 90 ,,	,,
Locomotive-engine boilers; heating surface from .	60 to 90 ,,	,,

The Power Developed by a given Area of Fire-Grate Surface and Heating Surface of Marine Boilers with Natural Draught, is shown in the following Table, collated from recent practice.

Table 42.—Proportions of, and Indicated Horse-power developed by, Marine Engines and Boilers.

Particulars of Triple-Expansion Engines in Twenty-eight Steamers.										Particulars of Boilers in Twenty-eight Steamers.								
No. of Steamer.	Cylinders.				Condenser Cooling surface.	Propeller.				No. of Steamer.	Number.	Diameter.		Length.		Heating surface. Total.	Fire-grate area.	Steam pressure lbs. per sq. in.
	Diameters.			Stroke.		Diameter.		Pitch.										
No.	Inches.			Inches.	Sq. ft.	ft.	in.	ft.	in.	No.	No.	ft.	in.	ft.	in.	sq. ft.	sq. ft.	lbs.
1	40	66	100	72	11,586	22	0	28	6	1	Six	13	6	18	0	17,640	626	155
2	40	66	100	72	11,586	22	0	28	6	2	Six	13	6	18	0	17,640	626	155
3	39	61	97	66	11,000	20	10	26	0	3	Five	13	6	18	0	15,107	540	155
4	39	61	97	66	11,000	20	10	26	0	4	Five	13	6	18	0	15,107	540	155
5	23	38	61	42	2,008	16	0	17	6	5	Two	14	6	10	4½	3,972	133	160
6	25½	42	70	51	3,209	16	6	20	0	6	Two	13	0	17	6	6,162	193	180
7	21	34	55½	36	1,447	14	0	17	6	7	Two	13	6	10	0	3,350	99	160
8	22	35	59	39	1,430	15	6	15	6	8	Two	13	4	9	9	3,324	102	160
9	29	45	74	54	3,900	19	6	20	0	9	Three	12	5	16	9	6,875	240	160
10	31	48	82	54	4,150	19	6	19	0	10	Three	12	6	18	6	8,000	260	160
11	25	41	67	48	2,800					11	Two	12	6	16	4	4,645	142	160
12	21½	36	59	42	2,000	15	0	16	6	12	Three	12	0	10	3	3,852	122	160
13	32	51	82	54	12,562	16	6	23	0	13	Four	16	0	19	0	20,192	710	160
14	27	44	71	48	2,800	17	9	17	6	14	Two	13	6	16	6	6,164	220	150
15	29	45	74	60	4,020	19	0	24	0	15	Two	14	8	16	8	6,950	196	150
16	29	45	74	54	3,850	18	0	21	0	16	Two	14	3	17	0	6,960	216	160
17	23	37	64	48	2,400	16	6	18	0	17	Two	11	9	17	0	4,715	144	180
18	28	44	74	51	3,700	17	9	22	9	18	Two	14	3	18	0	8,000	264	150
19	23	36½	58	36	2,218	15	0	15	6	19	One	14	10	15	5	3,271	126	160
20	17,17	38	60	42	2,900	15	6	15	6	20	Two	12	0	15	2	4,400	168	150
21	25	39	62	36	2,700	14	0	16	3	21	Two	12	2	14	0	4,000	150	160
22	31	46	72	51	3,713	16	3	22	6	22	Three	13	0	11	4	5,076	110	150
23	22½	35½	58½	39	1,750	14	7	16	6	23	One	15	0	11	9	2,338	50	160
24	25	42	68½	48	2,763	16	10	17	9	24	Two	14	3	11	6	4,346	84	160
25	22¼	35¾	58½	48	3,530	15	6	18	0	25	Two	13	0	11	4	3,486	63	160
26	31	50	83	60	6,860	19	0	23	0	26	Two	16	3¼	12	0	6,438	154	150
27	32	53	87½	60	7,500	19	0	23	9	27	Four	14	6	11	6	8,571	210	160
28	28½	46	75	42	3,450	16	0	21	0	28	Three	14	3¼	9	11	6,618	188	160

Table 42* *continued.*—RESULTS OF TRIALS OF TWENTY-EIGHT STEAMERS, &c.

No. of steamer	Revols. per minute.	Piston speed. Feet per minute.	Indicated horse-power	Heating surface. Per I.H.P.	Heating surface. Per lb. of coal per hr.	Indicated horse-power per sq. ft. of grate.	Coal burnt per sq. foot of grate per hour.	Coal burnt per I.H.P. per hour.	Remarks.
No.	Revols.	feet.	I.H.P.	sq. feet.	sq. feet.	I.H.P.	lbs.	lbs.	
1	52·2	627	4,295	4·11	2·46	6·86	11·45	1·67	H
2	51·3	616	4,402	4·04	2·55	7·03	11·14	1·584	H
3	57·3	630	3,587	4·21	2·22	6·65	12·60	1·896	H
4	57·4	631	3,822	3·95	2·14	7·08	13·02	1·841	H
5	61	427	1,120	3·54	2·02	8·43	14·75	1·75	
6	61·3	521	1,700	3·62	2·40	8·82	13·25	1·505	
7	64	384	900	3·72	2·31	9·09	14·67	1·612	H
8	70	455	1,065	3·12	2·38	10·42	13·70	1·312	
9	56	504	2,250	3·055	2·04	9·38	14·00	1·494	H
10	61·5	553	2,600	3·075	2·04	10·00	15·10	1·505	H
11	58	464	1,300	3·57	2·26	9·16	14·46	1·580	
12	67	469	1,100	3·50	2·29	9·02	13·79	1·529	
13	58·5	526	3,670	5·50	3·64	5·17	7·80	1·510	H
14	63	504	1,680	3·67	2·12	7·65	13·18	1·723	H
15	53·8	538	2,360	2·94	1·78	12·03	19·85	1·650	
16	64	576	2,550	2·73	1·82	11·80	17·70	1·500	
17	62	496	1,500	3·14	2·00	10·40	16·31	1·568	
18	62	527	1,727	4·63	2·85	10·53	17·06	1·620	
19	76	456	1,269	2·58	1·84	10·07	14·10	1·400	P
20	75	525	1,530	2·875	1·96	9·11	13·32	1·464	P
21	73	438	1,250	3·20	2·40	8·35	11·13	1·330	P
22	72	612	2,513	2·02	1·34	22·85	33·95	1·488	D H
23	76	494	1,350	1·73	1·28	27·00	36·42	1·350	D H P
24	65	520	1.800	2·41	1·94	21·42	26·62	1·242	D H P
25	69·5	552	1,360	2·56	1·91	21·59	28·90	1·338	D H P
26	59	590	2,600	2·435	1·78	16·88	23·05	1·365	D H P
27	66	660	3,400	2·52	2·04	16·18	19·97	1·234	D H P
28	73	511	2,058	3·215	2·05	17·10	10·92	1·565	
Average of all twenty-eight				3·274	2·14	11·22	17·08	1·522	
,, of natural draught				3·560	2·25	8·91	13·92	1·573	
,, of forced draught				2·412	1·72	20·98	28·15	1·336	

D = forced draught. H = feed-heater. P = Pass-over slide valve.

Name of Vessel.	I. H. P.	Surfaces. Grate.	Surfaces. Heating.	Surfaces. Condensing.	Ratios. G. to H.	Ratios. Tubes to Grate.	I. H. P. per sq. foot. Grate.	I. H. P. per sq. foot. Heating.	I. H. P. per sq. foot. Condensing
Parisian .	6020	544	13176	9624	1:27·9	...	11·06	·396	·625
Gallia .	5300	538	13000	8300	1:24·9	1:4·7	9·85	·468	·638
Devastation	6637	742	17806	6710	1:24·	1:5·3	8·94	·378	·989
Mexican .	3400	330	10000	5500	1:30·3	1:4·85	10·3	...	...
Inchonal .	1559	107	3800	1900	1:35·	1:4·6	12·4	·349	·698
Aberdeen .	2631	252	7128	5270	1:33·	1:4·29	12·53	·370	·500
Servia . .	10350	1014	27483	15000	1:27·1	1:5·5	9·86	·383	·690
Arizona .	6357	780	19500	12540	1:25·	...	8·08	·324	·503
Monarch .	2283	247	7803	5050	1:31·	...	9·22	·293	·456

* The author is indebted for the particulars of the first 28 steamers in this Table to a Paper read before the Institution of Mechanical Engineers by Mr. Alfred Blechynden.

Relative Value of the Heating Surface of the Tubes and Fire-Box of the boiler of a locomotive engine. It was found, as the result of a number of experiments with locomotives, that, under ordinary work, the fire-box of a locomotive boiler evaporated one-fifth and the tubes four-fifths of the water.

The Proportion of the Heating Surface of the Tubes to that of the Fire-Box of a locomotive engine boiler is generally from 10 to 12 to 1. The heating surface of the fire-box of an ordinary-sized locomotive boiler is frequently about 110 square feet and the heating surface of the tubes about 1100 square feet.

The Total Heating Surface of a Locomotive Engine Boiler, to obtain an ample supply of steam, should=(diameter of one cylinder)^2 ×4, This *Rule* gives the following results:—

Diameter of locomotive cylinder 16 in.=16×16×4=1024 square feet.
" " " " 17 in.=17×17×4=1156 square feet.
" " " " 18 in.=18×18×4=1296 square feet.
" " " " 19 in.=19×19×4=1444 square feet.

Table 43.—Standard Proportions of Heating Surface of Fire-Box and Tubes of Boilers of Locomotive Engines on Various English Railways.

Railway.	Diameter of Cylinder.	Fire-Grate Area.	Heating Surface.		
			Fire-Box.	Tubes.	Total.
	Inches.	sq. ft.	sq. ft.	sq. ft.	sq. ft.
Great Northern Railway	18	17·6	122	1043	1165
Brighton Railway	17	17·8	102	1080	1182
Brighton Railway	18¼	20·65	112	1373	1485
Brighton Railway, tank engine	17	16	90	858	948
Midland Railway	18	17·5	110	1096	1206
Great Western Railway, B. G.	18	21	153	1800	1953
Great Western Railway, N. G.	17	16·25	97	1216	1313
Great Western Railway, N. G.	18	17	133	1145	1278
London and North-Western Railway	16	15	85	1013	1098
London and North-Western Railway	17	15	89	1013	1102
London, Chatham, and Dover Railway	17½	16·3	107	962	1069
South-Western Railway	18½	17·5	104	1112	1216
Great Eastern Railway	18	17·3	117	1083	1200
Manchester S. & L. Railway	17½	17	87	1057	1144

The power developed by the Heating surface of Locomotive-Engine Boilers may be estimated approximately by this Rule.—Total Heating-surface in square feet × ·5=the average indicated horse-power in ordinary service; and total heating-surface in square feet $\times \cdot 7 =$ the maximum indicated horse-power. For instance, a locomotive-engine with a total heating surface of 1429 square feet, averages in ordinary service $= 1429 \times \cdot 5 = 714$ indicated horse-power, and its maximum power is $= 1429 \times \cdot 7 = 1000$ indicated horse-power.

The Rate of Transmission of Heat through Plates depends upon the condition and thickness of the plate. The rate of conduction in thermal units through boiler-plates and tubes per square foot of surface per hour may be calculated by Rankine's formula, as follows:—

Let T = the temperature of the gaseous products of combustion in the furnace
t = the temperature of the water
A = a constant varying between 160 and 200
U = thermal units transmitted per square foot of heating surface per hour

Then $U = \dfrac{(T-t)^2}{A}$.

Example: The temperature of the gases in a boiler furnace is 2580° Fahr., and the temperature of the water in the boiler is 280 Fahr., required the number of thermal units transmitted per square foot of the heating surface per hour.

Then $\dfrac{(2580-280)^2}{200} = \dfrac{2300^2}{200} = 26450$ thermal units per hour. After leaving the furnace the gases decrease in temperature on their way to the chimney, where their temperature would be about 600°, so that the temperature of the gases at the point where they quit the boiler would be, say, 650, and the rate of transmission at that point would only be $\dfrac{(650-280)^2}{200} = \dfrac{370^2}{200} = 684\cdot5$ thermal units per hour.

Grate Surface and Heating Surface.—The relation of grate-area heating surface, and consumption of fuel and water in steam-boilers has been fully investigated by Mr. D. K. Clark, with the following results:—

1st.—For a given extent of heating surface, the economical hourly consumption of fuel or water decreases directly as the grate-area is increased, and consequently in order to maintain the same efficiency or economical effect, the total hourly consumption should be reduced at the same rate as the grate-area is increased.

2nd.—For a given area of fire-grate the total hourly consumption should vary as the square of the heating surface. That is, if we double the area of heating surfaces, we can burn four times the quantity of fuel with the same grate-area, and maintain the same evaporative efficiency or economy.

3rd.—For a given hourly consumption, the area of the fire-grate should vary as the square of the heating-surface, in maintaining the same efficiency. That is, if the heating-surface be doubled, the fire-grate area may be increased four times, and the same economical consumption maintained.

The first conclusion applies to all boilers, but the two last conclusions apply more especially to boilers of the locomotive type.

The Efficiency of the Heating-Surface of a Boiler may be determined by Mr. D. K. Clark's formula:—

$$W = ar^2 + Bc$$

Where

$W =$ lbs. of water evaporated per square foot of fire-grate per hour
$c =$ lbs of fuel consumed per square foot of fire-grate per hour
$r =$ ratio of heating surface to fire-grate-area $= \dfrac{\text{Heating-surface}}{\text{Grate-surface}}$
$a =$ a constant, specific for each kind of boiler $= \cdot016$ for marine boilers, and $\cdot0222$ for stationary boilers.
$B =$ a constant, specific for each kind of boiler $= 10\cdot25$ for marine boilers, and 9.56 for stationary boilers.

Thus the lbs. of water evaporated from and at 212° Fahr. per lb. of fuel $= \dfrac{W}{c} = w$

$$\therefore\ w = \frac{ar^2}{c} + B$$

$$\text{and the efficiency} = \frac{w \times 100}{c}$$

Example: Required the efficiency of the heating-surface of a marine boiler which consumes 16 lbs. of coal per square foot of fire-grate surface per hour; the heating surface is 35 times that of the fire-grate surface, or a ratio of 35 to 1.

Then the lbs. of water evaporated from and at 212° Fahr. per lb. of fuel, $w = \dfrac{35 \times 35 \times \cdot016}{16 \text{ lbs. of coal}} = 1\cdot225 + 10\cdot25 = 11\cdot475$ lbs., and the efficiency of the heating surface is $= \dfrac{11\cdot475 \times 100}{16 \text{ lbs. of coal}} = 71\cdot72$ per cent.

The Efficiency of the Heating-Surfaces of Boilers is frequently calculated by the following formula by Rankine, which is based on the two fundamental principles that:—

1st.—The smaller the quantity of air used per pound of fuel the higher the temperature of the fire; and that—

2nd.—The greater the difference between the temperature of the fire and that of the water the greater the efficiency of the heating surface.

$$\frac{E'}{E} = \frac{BS}{S + AF} \text{ and efficiency} = \frac{BS}{S + AF}$$

Where $E' =$ the available evaporative power, and $E =$ the evaporating power of 1 lb. of a given kind of fuel in an ordinary boiler, in which $S =$ the total area of heating surface, including feed-water heater if any; $F =$ the number of pounds of fuel burnt per square foot of fire-grate per hour. A and B are constants found by experience; A is probably approximately proportionate to the square of the quantity of air supplied per lb. of fuel; B is a fractional multiplier to allow for miscellaneous losses of heat.

For boilers with chimney draught, $B = \dfrac{11}{12}$ $A = \cdot5$.

For boilers with forced draught, $B = \dfrac{19}{20}$ $A = \cdot3$.

Example: The consumption of coal in a boiler with natural draught is 12 lbs. per square foot of fire-grate surface. The heating-surface is 30 times that of the fire-grate surface, or a ratio of 30 to 1. Required the efficiency of the heating-surface of the boiler.

Then $\dfrac{30 \times \frac{11}{12}}{30 + (12 \times \cdot 5)} = \cdot 764$, or 76·4 per cent.

In some cases B is taken at 20 per cent. for boilers with natural draught, then—

For boilers with chimney draught, $B = \dfrac{4}{5}$.

Example: In a Lancashire boiler, the coal consumption is 14 lbs. per square foot of fire-grate surface per hour, the heating surface is 25 times that of the fire-grate surface, or 25 to 1. Required the efficiency of its heating surface.

Then $\dfrac{\frac{4}{5} \times 25}{25 + 14 \times \cdot 5} = \dfrac{20}{32} = \cdot 625$, the efficiency of the heating surface of the boiler.

In Lancashire and Cornish boilers a great portion of the heating surface is frequently rendered useless, in consequence of the flues not being kept clean.

Forced Draught effects great economy in the production of steam. The sharper the draught the less is the quantity of air required for dilution, the higher the temperature of the products of combustion, and the less are their volume and velocity. The economy is due to the high temperature of the furnace which ensures such rapid and efficient diffusion and combination of the gases, that they cannot escape from the furnace unburnt, and the formation of carbonic oxide and discharge of smoke is prevented. It also permits the products of combustion to be effectually deprived of their heat and discharged into the chimney at the lowest temperature. Such a condition cannot be effected with natural draught, as a high temperature in the chimney is necessary to create a draught. By forcing the draught, the maximum rate of combustion is obtained with the minimum quantity of fuel and air, from the minimum grate and heating surface.

The following are a few of the advantages of combustion with forced draught. It enables an inferior and cheaper class of coal to be used, with the same or better results than are obtained from best coal with natural draught. It increases the temperature of the furnace, the rate of combustion, and the efficiency of the heating surface. It enables the area of the fire-grate surface and tube-surface to be reduced, and a much smaller boiler can be used, to develop the same power, than is required with a natural draught. The air required for combustion can be efficiently warmed by the waste products of combustion, which is conducive to economy in the generation of steam, as about 13 per cent. of coal is wasted by supplying cold air to the furnace.

The indicated horse-power obtainable with forced draught, per square foot of fire-grate and per ton of boiler, is double that which can be obtained from the same boiler with natural draught.

Table 44.*—NATURAL DRAUGHT EXPERIMENTS ON A HORIZONTAL RETURN-TUBE BOILER AT PHILADELPHIA.

Reference Number.	Heating Surface in Square Feet.	Grate in Square Feet.	Heating Surface. Grate Area.	Total Combustible = Fuel, less Refuse, in Pounds.	Refuse per Cent. of Fuel.	Pounds of Combustible per Foot of Grate.	Square Feet of Heating Surface. Pounds of Combustible.	Pounds of Water per Pound of Combustible from and at 212 deg.	Funnel Temperature.	Pounds of Air per Pound of Combustible.	Efficiency, taking Calorific Value of Combustible at 16.	Per Cent. of Error. Rankine.	Per Cent. of Error D. K. Clarke.
59	638	36	17·7	417	21	11·57	1·53	12·77	Lead melts . .	16	·798	−13·5	−16·3
47	950	36	26·4	536	21·5	14·88	1·77	12·75	,, ,, . .	16	·797	−10·4	−13·7
63	326	36	9·1	68	21·3	2·43	3·7	13·6	383	18	·85	− 5·1	−20·6
61	482	36	13·4	310	21·4	8·62	1·55	12·28	Lead melts . .	19	·767	− 9·6	−13·8
48	950	36	26·4	549	20·3	15·24	1·73	12·17	,, ,, . .	20	·760	− 6·6	− 9·8
16	950	36	26·4	256	19·6	7·12	3·7	12·71	510	20½	·795	+ 1·5	− 7·1
64	950	36	26·4	507	19·5	14·09	1·87	11·72	Lead and zinc melt	21	·733	− 1·3	− 5·8
19	950	36	26·4	601	18·6	16·69	1·58	9·96	,, melts . .	21½	·748	− 7·0	+ 9·5
60	560	36	15·5	403	18·7	11·28	1·40	11·81	,, ,, . .	22½	·738	− 8·8	−10·3
7	950	36	26·4	282	23·7	7·84	3·36	12·45	512	22½	·778	+ 2·4	− 6·2
66	950	36	26·4	519	18	14·42	1·83	11·75	Lead melts . .	23	·734	− 2·0	− 6·21
37	950	36	26·4	628	22·2	17·44	1·51	11·26	Zinc ,, . .	23½	·704	− 2·2	− 3·4
58	716	36	19·9	482	18·9	13·4	1·48	11·67	Lead ,, . .	23½	·729	− 6·0	− 81
24	950	21·6	44	400	22·5	18·51	2·375	11·62	,, ,, . .	23½	·726	+ 4·1	+ 2·6
57	804	36	22·3	505	20·2	14·03	1·59	11·58	,, ,, . .	24	·724	− 3·8	− 6·6
65	950	36	26·4	546	18·9	15·18	1·74	11·55	,, ,, . .	24	·722	− 1·5	− 4·93
10	950	24	39·5	424	22	17·68	2·24	11·52	,, ,, . .	24½	·720	+ 4	+ 1·2
23	950	28·8	33	438	20·9	15·2	2·16	11·52	,, ,, . .	24½	·72	+ 3·3	− 1·1
9	950	30	31·7	260	18·9	8·65	3·65	12·2	510	25	·762	+ 5·9	− 0·7
18	950	36	26·4	135	13·5	3·75	7·03	13·08	371	25	·818	+ 4·5	− 2·1
32	950	36	26·4	609	16·2	18·58	1·42	10·78	Zinc melts . .	26	·673	+ 0·6	+ 0·6
12	950	18	52·8	344	22·8	19·1	2·76	11·52	570	27	·720	+ 7·6	+ 9·2
8	950	30	31·7	535	21·7	17·82	1·77	10·79	Lead melts . .	29	·675	+62	− 3·2
26	950	36	26·4	663	17	18·43	1·43	10·44	,, and zinc melt	30	·653	+ 3·8	+ 3·9
22	950	36	26·4	565	19·4	15·68	1·68	10·6	,, melts . .	30½	·662	+ 6·6	+ 3·4
73	950	36	26·4	370	13·6	10·26	2·56	11·81	477	31	·738	+ 3·8	− 4·06
21	950	36	26·4	635	17·6	17·64	1·5	10·45	Lead melts . .	31½	·653	+ 5	+ 4·1
13	950	18	52·8	152	20·3	8·43	6·25	12·29	399	32½	·768	+10·4	+26·4
15	950	13·5	70·4	109	23·3	8·08	8·7	12·53	360	33½	·783	+10·6	+60·1
72	950	36	26·4	244	13·6	6·79	3·9	11·96	427	33½	·747	+ 8·5	− 0·59
28	950	36	26·4	668	15·7	18·56	1·42	9·47	Zinc melts . .	34	·592	+ 1·4	+15·7
17	950	36	26·4	645	16·1	17·92	1·47	9·99	Lead ,, . .	35	·624	+ 9·8	+ 8·8
14	950	13·5	70·4	257	20·4	19·02	3·69	11·26	478	36	·704	+14·5	+28
11	950	24	39·5	209	16·4	8·71	4·78	11·65	431	36	·728	+13·3	+12·6
20	950	36	26·4	437	16·7	12·15	2·17	11·05	478	38	·790	+ 7·8	+ 1·1
27	950	36	26·4	733	18·1	20·35	1·295	7·51	Zinc melts . .	46	·47	+40·4	+44·6
29	950	36	26·4	772	18·2	21·43	1·23	7·41	,, ,, . .	48	·463	+40·6	+47·9
71	950	36	26·4	123	15	3·42	7·71	11·25	337	54	·703	+22·3	+20·09

Ref. No.	Remarks.	Ref. No.	Remarks.	Ref. No.	Remarks.
59	Four upper rows of tubes plugged.	58	Three upper rows of tubes plugged.	12	Grate reduced.
63	Eight upper rows of tubes plugged.	24	Grate reduced.	9	,, ,,
61	Six upper rows of tubes plugged.	57	Two upper rows of tubes plugged.	8	,, ,,
64	Ferrules in tubes.	65	Ferrules in tubes	73	Ferrules in tubes.
60	Five upper rows of tubes plugged.	10	Grate reduced.	13	Grate reduced.
66	Ferrules in tubes.	23	,, ,,	15	,, ,,
		32	Ferrules in four upper rows of tubes.	72	Ferrules in tubes.
				14	Grate reduced.
				11	,, ,,
				71	Ferrules in tubes.

* The Author is indebted for Tables 44—46 to a Paper read before the North-East Coast Institution of Engineers and Shipbuilders by Messrs. J. Patterson and M. Sanderson.

Table 45.—NATURAL DRAUGHT TRIALS ON BOILERS AT KIMBERLEY WATERWORKS.

Reference No.	Heating Surface in square feet.	Grate in square feet	Heating Surface. Grate Area.	Total Coal in lb.	Lb. of Coal per foot of Grate.	Square ft. of Heating Surface. Lb. of Coal.	Lb. of Water per lb. of Coal from and at 212°.	Funnel Temperature.	Lb. of Air per lb. of Coal.	Efficiency, taking Calorific Value of Coal at 14·805.	Per Cent. of Error. Rankine.	Per Cent. of Error. D. K. Clarke.
C1	644	15	43	120	8	5·36	11·23	529	24	·758	+10·5	24
D5	644	22½	28·6	150	6·70	4·3	11·17	509	24	·754	+ 8·7	9·2
D1	644	2 ½	28·6	176	7·84	3·64	10·87	547	25	·734	+ 9·8	9·7
D3	644	22½	28·6	100	4·44	6·44	11·18	444	28½	·755	+ 2·6	18
B4	644	10½	61	80	7·62	8·05	10·78	454	31½	·728	+18·4	67
C2	644	15	43	100	6·7	6·44	11·49	368	31½	·776	+ 9·5	28
D4	644	22½	28·6	80	3·56	8·05	11·25	392	32	·760	+13·4	23·8
C3	644	15	43	80	5·3	8·05	11·52	349	33	·778	+10·8	37
B3	644	10½	61	120	11·43	5·36	10·47	460	34	·707	+18·5	47
C4	644	15	43	60	4	10·73	11·07	313	44½	·747	+17·1	59
B5	644	10½	61	60	5·71	10·73	10·02	310	61	·676	+29·4	106
B6	644	10½	61	40	3·81	16·1	8·15	261	111	·550	+61·4	217
C5	644	15	43	45	3	14·31	8·32	256	131½	·562	+57·4	57·4

These experiments show that an excessive supply of air to a furnace rapidly diminishes the efficiency of the heating-surface of a boiler. The supply of air can be reduced to a minimum, and the efficiency of the heating surface is increased, by forcing the draught. In combustion with natural draught, about one-fourth of the heat developed in a furnace is expended in creating a draught.

Table 46.—FORCED DRAUGHT EXPERIMENTS ON A HORIZONTAL RETURN-TUBE BOILER AT PHILADELPHIA.

Reference Number.	Heating Surface in Square Feet.	Grate in Square Feet.	Heating Surface. Grate Area.	Total Combustible = Fuel, less Refuse in lb.	Refuse per cent. of Fuel.	Lb. of Combustible per foot of Grate.	Square ft. of Heating Surface. Lb. of Combustible.	Lb. of Water per lb. of Combustible from and at 212°.	Funnel Temperature.	Lb. of Air per lb. of Combustible.	Efficiency, taking Calorific Value of Combustible at 16.	Per Cent. of Error. Rankine.	Blast.
50	950	27	35·2	513	22·4	19	1·85	12·92	Zinc melts.	13	·808	−10·8	Fan.
45	950	36	26·4	797	18·9	22·15	1·19	11·36		22½	·710	− 9·1	Fan.
49	950	36	26·4	780	24·2	21·66	1·22	11·18		23½	·699	− 7·1	Fan.
51	950	36	26·4	659	18·2	18·3	1·44	11·23		23½	·702	− 3·1	Jet.
44	950	36	26·4	778	20·8	21·62	1·22	11·06		24½	·691	− 6·1	Fan.
41	950	36	26·4	805	16·3	22·37	1·18	10·0		31	·625	+ 2·9	Jet.
42	950	36	26·4	780	18·5	21·67	1·22	9·88		31½	·618	+ 5·0	Fan.
43	950	36	26·4	783	17·7	21·76	1·22	9·25		35½	·578	+12·3	Fan.

The Performance of Steam Ships with Natural Draught and with Forced Draught in the Boilers are given in Tables 47 and 48. It will be seen from column 7 of Table 47, that in the ship with natural draught only $10\frac{1}{4}$ indicated horse-power was developed, per square foot of fire-grate; but that with forced draught from 16 to 17 horse-power were developed per square foot of fire-grate, the boilers being practically the same in the two cases.

Table 47.—Performance of Steam Ships, each fitted with the same Size of Boilers, with Ordinary Open Stokeholds and Natural Draught, and with Closed Stokeholds and Forced Draught.

Ship.	Load on Safety Valves.	Indicated Horse-Power.	Weight of Boilers.	Area of Firegrate.	Indicated Horse-Power per Square Foot of Firegrate.	Indicated Horse-Power per Ton of Boiler.
	lb.		tons.			
Open Stokeholds.						
Natural Draught.						
Inflexible	60	8484	756	829	10·21	11·22
Colossus . . .	64	7492	594	645	11·62	12·61
Phaeton	90	5588	462	546	10·23	12·1
Closed Stokeholds.						
Forced Draught.						
Howe	90	11725	632	756	15·54	18·5
Rodney (9 boilers) .	90	9544	474	567	16·83	20·1
Mersey . . .	110	6628	306	399	16·61	21·7
Scout	120	3370	174	207	16·28	19·3
Trafalgar (estimated) .	135	12000	514	609	20·00	23·3

Note.—The weight of boiler given includes weight of water, funnel, uptakes, fittings, spare gear, &c.

Table 48.—Results of Trials of similar Steamships with Open Stokeholds and Natural Draught, and with Closed Stokeholds and Forced Draught.

Particulars.	Open Stokeholds (Natural Draught). "Inflexible."	"Colossus."	"Phaeton."
Duration of trial in hours . . .	6	5	5
Number of boilers used . . .	12	10	8
Mean steam pressure in boilers . lb.	61·06	61·52	85·35
Mean pressure in cylinders in lbs. per sq. in. — High pressure	29·55	40·66	43·56
Mean pressure in cylinders in lbs. per sq. in. — Low "	9·833	12·09	11·43
Mean revolutions per minute . .	73·26	89·96	100·26
Mean speed of piston, in ft., per minute	586	585	802
Indicated horse-power	8483	7492	5588
Area of firegrate used in square feet .	829	645	546
I.H.P. per square foot of firegrate . .	10·21	11·62	10·23
Heating surface per I.H P. in square feet — Tubes .	2·20	1·97	2·23
Heating surface per I.H P. in square feet — Total .	2·63	2·33	2·61
Coal used per I.H.P. per hour, in lbs.	2·06	2·55	2·39
" hour, in tons . . .	7·80	8·53	5·96
Remarks	Blast used last half hour only.	Blast used throughout the trial.	Natural draught only.

Table 48 *continued.**—Results of Trials of Similar Steamships, &c.

Particulars.	Forced Draught.					
	"Mersey."	"Scout."	"Rodney."		"Howe."	"Caroline."
Duration of trial in hours	3	4	4	3	4	6
Number of boilers used	6	4	12	9	12	2
Mean steam pressure in boilers . . . lb.	107·8	113 09	93·06	92·74	89·21	84·52
Mean air pressure in boiler rooms, in inches of water	2·02	1·52	1·4	1·89	2·05	1·5
Mean pressure in cylinders in lbs per square inch — High pressure	56·53	61·42	59·92	49·73	59·51	43·9
Mean pressure in cylinders in lbs per square inch — Low pressure	22·82	24·31	12·8	12·1	13·43	12·79
Mean revolutions per min.	122 34	152·33	103·42	100·13	106·63	77·8
Mean speed of piston, in feet, per minute . .	795	762	776	751	800	389
Indicated horse-power . .	6628	3370	11158	9544	11725	983
Area of firegrate used in square feet . . .	399	207	756	567	·756	545
I H.P. per sq. ft. of firegrate	16·61	16·28	14 75	16·83	15·51	18·02
Heating surface per I.H.P. in sq. ft. — Tubes	1·56	1·63	1·54	1·35	1·46	1·24
Heating surface per I.H.P. in sq. ft. — Total	1·77	1·83	1·82	1·6	1·73	1·43
Coal used per I.H.P. per hour, in lbs. . . .	2·48	2·6	2·2	...	2·16	2·54
Coal used per hour, in tons	7·33	3·92	11	...	11·30	11·1

Note.—The indicated horse-power recorded, is that developed by the main engines only, and does not include the indicated horse-power expended in working the feed and circulating pumps, blowing fans, and other auxiliary machinery.

The maximum air pressure = 2·02 inches of water in the above trials, is equal to a pressure of about $\frac{1}{13}$ lb. per square inch.

The cubic capacity of one boiler of the Howe is 1344 cubic feet.

The indicated horse-power given by the boiler was 644 with natural draught, and 977 with forced draught, the indicated horse-power per cubic foot of boiler was ·479 with natural draught, and ·727 with forced draught.

The performance of steamships of various nations with natural draught and with forced draught is shown in a compact form in the following Table, which gives the method of forcing the draught, the consumption of coal per indicated horse-power, and the percentage of gain.

* The Author is indebted for Tables 47 and 48 to the "Reports of the Proceedings of the Institution of Naval Architects."

Table 49.*—Performance of Steamships with Natural and with Forced Draught.

Name of Vessel.	I. H. P.		Grate Surface.		I. H. P. per sq. ft. Grate.		Coal per I. H. P.		Per cent. of gain of I. H. P. per sq. ft. of Grate.	Method of forcing the Draught.
	Natural draught.	Forced draught.	Natural draught.	Forced draught.	Natural draught.	Forced draught.	Natural draught.	Forced draught.		
French Navy—										
Foudroyant	6016	8088	732	732	8·22	11·05	2·20	2·51	34·4	Jet in chimney.
Duperré	6075	8010	730	730	8·33	10·99	2·20	2·42	31·9	” ”
Bayard	3255	4064	416	416	7·82	9·67	2·31	2·73	24·8	” ”
Naiade	2663	3215	340	340	7·83	9·46	2·42	2·98	20·7	” ”
Iphigenie	2230	2762	284	284	7·85	·73	2·09	2·09	23·9	” ”
Sfax (estimated)	5000	7500	538	538	9·29	14·00	...	...	50·	Closed fire-room.
English Ironclads—										
Collingwood	8360	9558	...	...	...	...	...	...	...	” ”
Rodney	8262	11156	...	...	...	...	...	...	35·03	” ”
Renown (estimated)	5500	8500	570	...	9·65	15·00	...	...	54·5	” ”
Benbow (estimated)	7500	9800	...	...	...	...	...	...	30·7	” ”
Conqueror	4658	4023	574	293	7·96	13·41	...	...	68·5	” ”
English Cruisers—										
Leander	4233	4658	545	545	7·77	8·55	2·01	1·98	10·04	Jets wide open.
Heroine	1127	702	110	55	10·25	12·8	...	...	24·9	” ”
Hyacinth	1195	1445	110	110	10·87	13·13	...	...	20·9	” ”
Satellite	1116	1397	110	82·5	10·15	16·9	...	...	66·5	Closed fire-room.
English Despatch Boats—										
Alacrity	2157	3173	...	...	...	...	2·1	2·77	47·0	” ”
Surprise	2104	3017	...	...	...	...	2·6	2·78	43·4	” ”
Archer class (estimated)	2400	3500	224	224	10·7	15·6	...	...	45·2	” ”
Scout class (estimated)	2100	3200	204	204	10·2	15·6	...	...	52·4	” ”
Curlew class (estimated)	850	1200	...	...	...	...	...	...	41·2	” ”
Mean of three vessels	1875	2800	...	...	...	...	...	...	33·0	” ”
German Navy—Blitz	1794	2808	298·5	298·5	6·01	9·43	...	...	56·4	” ”
Brazilian Navy—Riachuelo	6926	7336	Boilers. 10	Boilers. 8	...	...	...	...	27·0	” ”
Aquidabam	5270	6201	...	...	...	...	...	...	...	” ”
U. S. Navy—Dolphin	1648	2253	270	270	6·1	8·35	...	...	36·7	Open fire-room and blowers.

* The Author is indebted for the above Table to "The Engineer."

Forced Draught with Warm Air.—In Howden's system of forced combustion in steam-boilers, the air supplied to the furnaces is warmed by passing it through a series of tubes, placed in the path of the escaping gases

Fig. 53.

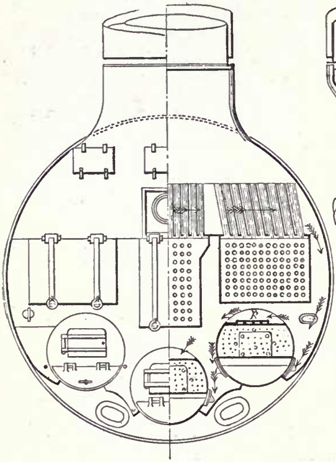


Fig. 54.

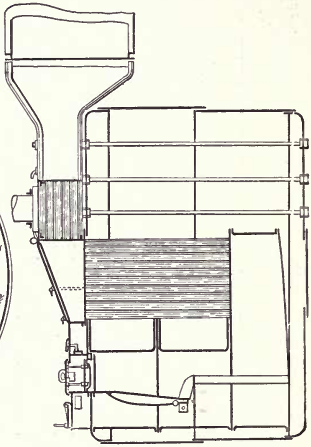
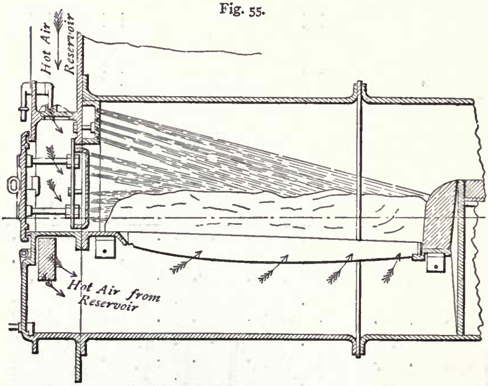


Fig. 55.



Figs. 53—55.—Howden's system of forced draught.

or products of combustion. The arrangement of the furnace is shown in Figs. 53—55. The ashpit is closed, as shown in Fig. 55, and the hot air is supplied to the fire both above and below the bars. Quick gasifying of the fuel and complete combustion is obtained by the following means:—The air in the ashpit, with a given area of air-space through the fire-bars, and a given average depth of fuel, is maintained at a pressure designed to pass a quantity of air through the fuel sufficient to gasify it, and bring it to the surface largely in the form of carbonic oxide. The air in the casing between the two furnace doors is maintained at a considerably higher pressure than in the ashpit, and is thus received by the distributing boxes inside the furnace-plate and inner furnace door. The air then, at a considerable temperature, and at a high velocity, issues in minute streams from small holes in the interior side of the air-boxes, their aggregate area being proportioned to the normal work of the furnace, and their position arranged to cause the air to strike the fuel with force equally over the surface within the limits of the fire-bars. By means of these differential pressures, the weight of air required for the complete combustion of a given weight of fuel can be made much less than is necessary in an ordinary furnace, while, with the complete stage of combustion being chiefly on or above the surface of the fuel, a clear white flame and intense heat is generated where most effective for radiation, and most innocuous in its effect on the furnace-bars.

Two voyages were made by a steamship under similar conditions, one being made with boiler-furnaces worked with natural draught, and the other with boiler-furnaces worked with Howden's system of forced draught. The results of this trial are given in the following Table:—

Table 50.—Evaporative Performance of a Marine Boiler fitted with Howden's arrangement of Forced Draught compared with a Larger Marine Boiler with Natural Draught.

Boiler with Natural Draught.							
Voyage.	Draught.		Average Revolutions.	Average Indicated Horse-Power.	Coal.	Consumption per Twenty-Four Hours.	Weather.
	Aft.	Forward.					
	ft. in.	ft. in.				tons.	
1. Homewards	20 3	19 3	56	564	Welsh.	13½	Fair.
2. Outwards .	20 4	18 10	59	Not taken.	Ryhope.	15	,,
Boiler with Forced Combustion, or Forced Draught.							
Voyage.	Draught.		Average Revolutions.	Average Indicated Horse-Power.	Coal.	Consumption per Twenty-Four Hours.	Weather.
	Aft.	Forward.					
	ft. in.	ft. in.				tons.	
1. Outwards .	20 4	18 10	57	Not taken.	Scotch.	11	Fair.
2. Homewards	20 3	19 6	60	623	Welsh.	9½	Fair, and head wind

Table 50 *continued.*—EVAPORATIVE PERFORMANCE OF A MARINE BOILER, &c.

Dimensions of Boilers.

Dimensions.	Boiler with Natural Draught.	Boiler with Forced Draught.
Length without uptake	17 ft.	11 ft.
Diameter	12 ft. 6 in.	14 ft.
Steam domes	Two	None.
Number and diameter of furnaces	Four, 3 ft. 5 in.	Three, 3 ft. 4 in.
Number, length, and diameter of tubes	372, 6 ft. $4\frac{1}{2}$ in. by $3\frac{1}{2}$ in.	210, 8 ft. by 3 in.
Tube surface	2173 sq. ft.	1319 sq. ft.
Length of firebars, over all	5 ft. 6 in.	4 ft. $1\frac{1}{2}$ in.
Aggregate firegrate	75 sq. ft.	36 sq. ft.

The air-heating tubes for the forced draught were 40 in number 2 ft. 3 inches long, and $3\frac{1}{4}$ inches external diameter.

Forced Combustion in Steam-Boilers on Howden's system will, it is estimated, give an evaporative power per square foot of fire-grate of about 25-indicated horse-power with compound engines, at 80 lbs. per square inch pressure of steam, and 30-indicated horse-power with triple expansion engines at about 140 lbs. per square inch pressure of steam, at rates of consumption not exceeding 1·35 lbs. and 1·1 lbs. respectively per indicated horse-power per hour.

Forced Combustion by Induced Draught.—In Martin's system of induced draught, fans are placed in the base of the funnel, by means of which the air is exhausted through the furnace and flue-tubes of the boiler: the draught is thus rendered independent of the height of the funnel. Trials were made of this system of forcing combustion, with a steel marine-boiler 5 feet 6 inches diameter and 6 feet long, having a single furnace-tube 2 feet 3 inches diameter, with 44 tubes 3 inches diameter and 4 feet 6 inches long.

Heating surface.—Tubes, 97 square feet; furnace, 17 square feet; combustion chamber, 38 square feet.

The draught was induced by a pair of fans 24 inches diameter, driven by friction-gear; the inlets of the fans were 12 inches diameter, a larger inlet being required than is necessary for cold air, on account of the expansion of the gases. The centre portion of the shaft of the fan, which passes through the uptake, was protected from heat by a sleeve and hollow couplings, so that the bearings of the shaft are not affected by heat. The fans made 1400 revolutions per minute. The results of the trials are given in the following Table:—

Table 51.—RESULTS OF TRIALS OF A BOILER WITH MARTIN'S SYSTEM OF INDUCED DRAUGHT, AND WITH NATURAL DRAUGHT.

Particulars.	1	2
Duration of trial	4 hours.	4 hours.
Description of coal used	(Nixon's Navigation)	(Nixon's Navigation)
Total coal consumed . . lbs.	500	350
Heating surface in square feet . . 152	...	...
Grate surface in square feet . . . 6·75	...	...
Lb. of coal per foot of grate	18·5	12·96
Temperature of feed water	72°	72°
Total water evaporated at actual temperature of feed . . lbs.	5678	2788
Water evaporated per lb. of coal at actual temperature of feed . . lb.	11·35	7·96
Water evaporated per lb. of coal at 212°	13	9·12
Water evaporated per square foot of grate	210	103·2
Steam pressure per square inch	70 lbs.	60 lbs.
Revolutions of fan per minute	1400	...

The temperature was found to be on one occasion = 1950° Fahr. in the furnace; 736° Fahr. in the tubes, 24 inches from the smoke box end; 612° Fahr. in the smoke-box; and 442° Fahr. in the funnel.

The induced draught produced a clear brisk fire and complete combustion of the fuel; and the tubes remained free from cinders.

BOILER-PLATES AND RIVETED-JOINTS; BOILER-SHELLS AND FURNACE-TUBES.

Arrangement of Boiler-Plates.—Externally-fired boilers have the plates arranged so that the laps of the ring-seams do not face the fire. Internally-fired boilers have parallel belts of plates arranged with alternately outer and inner belts. In locomotive boilers there are three belts of plates of diminishing diameter arranged telescopically: the first belt is the largest, and it is lapped inside the fire-box shell; the second belt is lapped inside the first, and the third inside the second.

Strength of Boiler-Plates.—The tensile strength of ordinary good wrought-iron boiler-plates is 21 tons per square inch along the grain, and 18 tons per square inch across the grain. With an elongation of about 7 per cent., they should admit of being bent hot, without fracture, along the grain to an angle of 130°, and across the grain to an angle of 100°; and they should bend cold without fracture to the following angles:—Plates $\frac{7}{16}$-inch thick to 55° lengthways of the grain, and 25° across the grain; $\frac{3}{8}$ inch to 70° lengthways, and 35° across; $\frac{5}{16}$-inch to 80° lengthways, and 45° across; $\frac{1}{4}$-inch 90° lengthways, and 55° across the grain. The tenacity of best Yorkshire or Lowmoor iron is 24 tons along the grain and 22 tons across the grain; with an elongation of about 12 per cent., these plates should admit of being bent double, either along or across the grain when red hot.

Soundness of Plates.—In order to ascertain whether plates are internally sound and free from blisters and laminations, they are tested in three ways. First, the plate is placed on edge and tapped all over with a light hammer, when a sharp ringing sound indicates a sound plate, and a dull heavy sound indicates the presence of defects or laminations. Second, the plate is supported horizontally at its four corners, and the upper surface is covered with fine sand: on tapping the plate lightly with a hammer, if the plate be sound the sand will be thrown off by the vibration, but if laminated the sand will remain stationary on the defective portions. Third, the plate is heated to redness and placed aside to cool, when the laminated or blistered portions will turn black while the sound portions retain their redness.

Steel Boiler Plates should be of the mildest quality of steel, containing such a low percentage of carbon as to be incapable of acquiring any degree of temper when heated and suddenly cooled. Steel-plates should be worked at a cherry-red heat: at a higher temperature the plates are liable to burn and become brittle. It is essential in working steel-plates hot, to obtain a uniform cherry-red heat, as the ductility is considerably lessened if they are worked at a blue heat; hence if steel-plates are improperly worked they are much less reliable than wrought-iron plates.

The Tensile Strength of Steel Boiler-Plates ranges from 26 tons to 32 tons per square inch, with an elongation of from 22 to 30 per cent. As thick plates require more carbon than thin ones, to enable them to stand the same mechanical tests, the thicker the plate the milder the steel, and the less the tenacity should be. For plates over $\frac{3}{4}$-inch thick, the tenacity should not exceed 28 tons per square inch, otherwise the amount of carbon in the steel may admit of the plates acquiring some degree of temper when heated and cooled, and cause them to become brittle.

The Rivet-Holes of Steel-Plates should be drilled: and when the plates contain a high proportion of carbon, they should be annealed before being bent, in a plate-bending machine, by heating them to an uniform red heat in a furnace, after which the fire should be allowed to die out, and the plates should remain in the furnace until quite cold.

Hydraulic-Riveted Joints are much stiffer and tighter under pressure than hand-riveted joints, and no caulking is necessary if the faces of the plates are clean and form a metal-to-metal joint. Visible slip does not commence in hydraulic-riveted joints until double the pressure has been applied at which slip commences in a hand-riveted joint.

Riveted Joints.—Mr. Fairbairn found the strength of riveted joints compared with that of the entire plate to be as follows:—

Strength of entire plate .	= 100
Strength of double-riveted joint	= 70
Strength of single-riveted joint	= 56

Taking the strength of the entire wrought-iron boiler-plate at 21 tons per square inch along the fibre, the breaking strength of double-riveted iron boilers, with seams properly breaking joints, $= 21 \times \cdot 70 = 14 \cdot 7$, or about $14\frac{3}{4}$ tons per square inch, and of single-riveted boilers $= 21 \times \cdot 56 = 11 \cdot 76$, or $11\frac{3}{4}$ tons per square inch. The strength of the plates across the fibre is about 15 per cent. less than the above. To obtain the most perfect riveted joint, it would be necessary to make the strength of the net-section of the plates, after the rivet-holes are made, equal to the shearing strength of the rivets.

Proportions of Riveted Joints.—The following Table of steam-tight riveted joints gives good proportions for iron plates, iron rivets, and lap joints, for pressure up to 100 lbs. per square inch, and 200 lbs. cold-water test, all the rivet-holes, above $\frac{5}{8}$ inch, being $\frac{1}{16}$ inch larger than the rivets.

Table 52.—PROPORTIONS OF RIVETED LAP-JOINTS FOR WROUGHT-IRON PLATES AND IRON RIVETS.

Thickness of Plate.	Diameter of Rivet.	Pitch of Rivets.		Breadth of Lap.	
		Single-Riveted Joints.	Double-Riveted Joints.	Single-Riveted Joints.	Double-Riveted Joints.
Inches.	Inches.	Inches.	Inches.	Inches.	Inches
$\frac{3}{16}$	$\frac{3}{8}$	1	$1\frac{3}{8}$	$1\frac{1}{4}$	2
$\frac{1}{4}$	$\frac{1}{2}$	$1\frac{1}{4}$	$1\frac{3}{4}$	$1\frac{3}{4}$	$2\frac{1}{2}$
$\frac{5}{16}$	$\frac{5}{8}$	$1\frac{3}{8}$	$2\frac{1}{4}$	2	3
$\frac{3}{8}$	$\frac{11}{16}$	$1\frac{3}{4}$	$2\frac{1}{2}$	$2\frac{1}{4}$	$3\frac{1}{2}$
$\frac{7}{16}$	$\frac{3}{4}$	2	$2\frac{3}{4}$	$2\frac{5}{8}$	$4\frac{1}{4}$
$\frac{1}{2}$	$\frac{13}{16}$	$2\frac{1}{8}$	$2\frac{7}{8}$	3	$4\frac{1}{2}$
$\frac{9}{16}$	$\frac{7}{8}$	$2\frac{3}{8}$	$3\frac{1}{8}$	$3\frac{1}{8}$	$4\frac{3}{4}$
$\frac{5}{8}$	$\frac{15}{16}$	$2\frac{1}{2}$	$3\frac{1}{4}$	$3\frac{3}{8}$	5
$\frac{11}{16}$	1	$2\frac{3}{4}$	$3\frac{3}{8}$	$3\frac{1}{2}$	$5\frac{1}{4}$
$\frac{3}{4}$	$1\frac{1}{16}$	$2\frac{7}{8}$	$3\frac{1}{2}$	$3\frac{3}{4}$	$5\frac{1}{2}$
$\frac{13}{16}$	$1\frac{1}{8}$	3	$3\frac{5}{8}$	$3\frac{7}{8}$	$5\frac{3}{4}$
$\frac{7}{8}$	$1\frac{3}{16}$	$3\frac{1}{4}$	$3\frac{3}{4}$	$4\frac{1}{4}$	6
$\frac{15}{16}$	$1\frac{1}{4}$	$3\frac{3}{8}$	4	$4\frac{1}{2}$	$6\frac{1}{4}$
1	$1\frac{5}{16}$	$3\frac{1}{2}$	$4\frac{1}{8}$	$4\frac{5}{8}$	$6\frac{1}{2}$
$1\frac{1}{16}$	$1\frac{3}{8}$	$3\frac{3}{4}$	$4\frac{1}{4}$	$4\frac{7}{8}$	$6\frac{3}{4}$
$1\frac{1}{8}$	$1\frac{7}{16}$	4	$4\frac{3}{8}$	$5\frac{1}{4}$	7

The Percentage of Strength of the Riveted Joint in terms of that of the solid plate may be found by the following *Rule:* Subtract the diameter of the rivet-hole from the pitch of the rivets, and divide the remainder by the pitch of the rivets.

The Percentage of Strength of the Rivets to that of the solid plate may be found by the following *Rule:* Multiply the square of the diameter of the rivet-hole by ·7854 and by the number of rows of rivets, and divide by the product of the pitch of the rivets by the thickness of the plate.

The Percentage of Strength of the Longitudinal Seams of a boiler, when the diameter of rivet is not known, may be estimated approximately by the following formula, the rivets being in single-shear.

Let t = the thickness of plate in inches.
p = the pitch of the rivets in inches.
n = the number of rows of rivets.
S = percentage of strength of the longitudinal seams.

$$\text{Then } S = 88 - \frac{220 \times t}{n p + 3 t}$$

Example: The thickness of plate of a boiler is $\frac{3}{8}$ inch, the pitch of the rivets is 3 inches, there are two rows of rivets in single-shear: what is the percentage of strength of the longitudinal seams, the diameter of the rivets not being known?

Then $88 - \frac{220 \times \cdot 375 \text{ inch thickness of plate}}{2 \text{ rows} \times 3 \text{ inches pitch} + (3 \times \cdot 375)} = 76$, the percentage of strength of the seam as compared with the solid plate.

Rivet-Holes.—The effect of punching the rivet-holes for riveted-joints is to weaken the metal round the holes, and to diminish the tensile strength of the plates to the extent of from 5 to 10 per cent. in soft wrought-iron plates, and from 20 to 25 per cent. in hard wrought-iron plates. The tensile strength of steel plates is diminished to the extent of from 20 to 28 per cent. by punching the rivet-holes for the joints of the plates. Thick plates are more injured by punching than thin ones.

Lloyd's Proportions for Single, Double, and Treble-Riveted Joints are given in Tables Nos. 53—56, for iron-plates, iron-rivets, lap-joints, and drilled-holes; only 90 per cent. of the rivet-section is considered to be effective in drilled-holes. For Lloyd's Rules for Riveted Joints see page 162.

Table 53.—Lloyd's Proportions for Single-Riveted Joints.

Thickness of Iron Plate.	Diameter of Iron Rivet.	Pitch.	Lap.	Percentage.	
				Rivet.	Plate.
Inch.	Inches.	Inches.	Inches.		
$\frac{5}{16}$	$\frac{11}{16}$	$1\frac{3}{4}$	$2\frac{1}{16}$	67·9	60·7
$\frac{3}{8}$	$\frac{13}{16}$	$2\frac{1}{16}$	$2\frac{7}{16}$	67·0	60·0
$\frac{7}{16}$	$\frac{15}{16}$	$2\frac{1}{4}$ & $\frac{3}{32}$	$2\frac{13}{16}$	67·4	60·1
$\frac{1}{2}$	1	$2\frac{7}{16}$	3	64·9	58·9
$\frac{9}{16}$	$1\frac{1}{8}$	$2\frac{5}{8}$ & $\frac{\cdot 3}{32}$	$3\frac{3}{8}$	65·1	58·6
$\frac{5}{8}$	$1\frac{3}{16}$	$2\frac{3}{4}$ & $\frac{1}{32}$	$3\frac{9}{16}$	63·7	57·3

Table 54.—Lloyd's Proportions for Double-Riveted Joints.

Thickness of Iron Plate.	Diameter of Iron Rivet.	Pitch.	Lap.	Percentage.	
				Rivet.	Plate.
Inch.	Inches.	Inches.	Inches.		
$\frac{3}{8}$	$\frac{11}{16}$	$2\frac{3}{8}$ & $\frac{3}{32}$	$3\frac{7}{16}$	80·2	72·1
$\frac{7}{16}$	$\frac{3}{4}$	$2\frac{1}{2}$ & $\frac{3}{32}$	$3\frac{3}{4}$	77·7	71·1
$\frac{1}{2}$	$\frac{13}{16}$	$2\frac{5}{8}$ & $\frac{1}{16}$	$4\frac{1}{16}$	77·2	69·4
$\frac{9}{16}$	$\frac{15}{16}$	$3\frac{1}{8}$	$4\frac{11}{16}$	78·5	70·0
$\frac{5}{8}$	1	$3\frac{1}{4}$	5	77·3	69·2
$\frac{11}{16}$	$1\frac{1}{16}$	$3\frac{3}{8}$	$5\frac{5}{16}$	76·4	68·5
$\frac{3}{4}$	$1\frac{1}{8}$	$3\frac{1}{2}$ & $\frac{1}{32}$	$5\frac{5}{8}$	75·0	68·1

Table 55.—LLOYD'S PROPORTIONS FOR TREBLE-RIVETED JOINTS.

Thickness of Iron Plate.	Diameter of Iron Rivet.	Pitch.	Lap.	Percentage.	
				Rivet.	Plate.
Inch.	Inches.	Inches.	Inches.		
$\frac{1}{2}$	$\frac{3}{4}$	$3\frac{1}{8}$ & $\frac{1}{32}$	$4\frac{1}{2}$	83·9	76·2
$\frac{9}{16}$	$\frac{13}{16}$	$3\frac{1}{4}$ & $\frac{1}{32}$	$4\frac{7}{8}$	84·3	75·2
$\frac{5}{8}$	$\frac{7}{8}$	$3\frac{7}{16}$	$5\frac{1}{4}$	83·9	75·4
$\frac{11}{16}$	1	$4\frac{1}{16}$	6	84·4	75·4
$\frac{3}{4}$	$1\frac{1}{16}$	$4\frac{1}{4}$	$6\frac{3}{8}$	83·4	75·0
$\frac{13}{16}$	$1\frac{1}{8}$	$4\frac{3}{8}$ & $\frac{1}{32}$	$6\frac{3}{4}$	83·3	74·5
$\frac{7}{8}$	$1\frac{3}{16}$	$4\frac{1}{2}$ & $\frac{3}{32}$	$7\frac{1}{8}$	82·5	74·1
$\frac{15}{16}$	$1\frac{1}{4}$	$4\frac{3}{4}$ & $\frac{1}{32}$	$7\frac{1}{2}$	82·1	73·8
1	$1\frac{5}{16}$	$4\frac{15}{16}$	$7\frac{7}{8}$	82·2	73·4

Table 56.—LLOYD'S PROPORTIONS FOR DOUBLE BUTT-STRAPS, DOUBLE-RIVETED DRILLED HOLES.

Thickness of Iron Plate.	Diameter of Iron Rivet.	Pitch.	Breadth of Strap.	Percentage.	
				Rivet.	Plate.
Inch.	Inches.	Inches.	Inches.		
$\frac{1}{2}$	$\frac{5}{8}$	$2\frac{1}{2}$ & $\frac{1}{16}$	$6\frac{1}{4}$	83·8	75·6
$\frac{9}{16}$	$\frac{11}{16}$	$2\frac{3}{4}$ & $\frac{1}{32}$	$6\frac{7}{8}$	82·9	75·2
$\frac{5}{8}$	$\frac{3}{4}$	3	$7\frac{1}{2}$	82·4	75·0
$\frac{11}{16}$	$\frac{13}{16}$	$3\frac{1}{8}$ & $\frac{3}{32}$	$8\frac{1}{8}$	82·1	74·7
$\frac{3}{4}$	$\frac{15}{16}$	$3\frac{1}{4}$ & $\frac{1}{16}$	$9\frac{3}{8}$	84·1	75·4
$\frac{13}{16}$	1	$4\frac{1}{32}$	10	84·0	75·2
$\frac{7}{8}$	$1\frac{1}{16}$	$4\frac{1}{4}$	$10\frac{5}{8}$	83·3	75·0
$\frac{15}{16}$	$1\frac{1}{8}$	$4\frac{3}{8}$ & $\frac{3}{32}$	$11\frac{1}{4}$	83·1	74·8
1	$1\frac{3}{16}$	$4\frac{3}{4}$	$11\frac{7}{8}$	82·6	74·6

The Board of Trade Proportions for Single, Double, and Treble-Riveted Joints for Steel Plates, steel rivets, lap-joints, and drilled-holes are given in the following Tables, Nos. 57—60.

Table 57.—BOARD OF TRADE PROPORTIONS FOR SINGLE-RIVETED JOINTS.

Thickness of Steel Plate.	Diameter of Steel Rivet.	Pitch.	Lap.	Percentage.	
				Rivet.	Plate.
Inch.	Inches.	Inches.	Inches.		
$\frac{5}{16}$	$\frac{11}{16}$	$1\frac{5}{8}$ & $\frac{1}{32}$	$2\frac{1}{16}$	71·8	58·4
$\frac{3}{8}$	$\frac{13}{16}$	$1\frac{7}{8}$ & $\frac{1}{16}$	$2\frac{7}{16}$	71·4	58·1
$\frac{7}{16}$	$\frac{15}{16}$	$2\frac{1}{4}$	$2\frac{13}{16}$	70·1	58·3
$\frac{1}{2}$	$1\frac{1}{16}$	$2\frac{1}{2}$ & $\frac{1}{32}$	$3\frac{3}{16}$	70·3	58·0
$\frac{9}{16}$	$1\frac{1}{8}$	$2\frac{1}{2}$ & $\frac{3}{32}$	$3\frac{3}{8}$	68·1	56·6
$\frac{5}{8}$	$1\frac{3}{16}$	$2\frac{5}{8}$ & $\frac{1}{32}$	$3\frac{9}{16}$	66·7	55·2

Table 58.—Board of Trade Proportions for Double-Riveted Joints.

Thickness of Steel Plate.	Diameter of Steel Rivet.	Pitch.	Lap.	Percentage.	
				Rivet.	Plate.
Inch.	Inches.	Inches.	Inches.		
$\frac{3}{8}$	$\frac{3}{4}$	$2\frac{5}{8}$ & $\frac{1}{16}$	$3\frac{3}{4}$	87·7	72·1
$\frac{7}{16}$	$\frac{13}{16}$	$2\frac{3}{4}$	$4\frac{1}{16}$	86·2	70·4
$\frac{1}{2}$	$\frac{7}{8}$	$2\frac{7}{8}$	$4\frac{3}{8}$	84·1	69·5
$\frac{9}{16}$	1	$3\frac{5}{16}$	5	84·3	69·8
$\frac{5}{8}$	$1\frac{1}{16}$	$3\frac{3}{8}$ & $\frac{1}{32}$	$5\frac{5}{16}$	83·2	68·7
$\frac{11}{16}$	$1\frac{1}{8}$	$3\frac{1}{2}$	$5\frac{5}{8}$	82·6	67·8
$\frac{3}{4}$	$1\frac{1}{4}$	$3\frac{7}{8}$ & $\frac{1}{16}$	$6\frac{1}{4}$	83·1	68·2

Table 59.—Board of Trade Proportions for Treble-Riveted Joints.

Thickness of Steel Plate.	Diameter of Steel Rivet.	Pitch.	Lap.	Percentage.	
				Rivet.	Plate.
Inch,	Inches.	Inches.	Inches.		
$\frac{1}{2}$	$\frac{13}{16}$	$3\frac{3}{8}$	$4\frac{7}{8}$	92·2	75·9
$\frac{9}{16}$	$\frac{7}{8}$	$3\frac{1}{2}$	$5\frac{1}{4}$	91·6	75·0
$\frac{5}{8}$	$\frac{15}{16}$	$3\frac{5}{8}$	$5\frac{5}{8}$	91·0	74·1
$\frac{11}{16}$	$1\frac{1}{16}$	$4\frac{1}{4}$	$6\frac{3}{8}$	91·0	75·0
$\frac{3}{4}$	$1\frac{1}{8}$	$4\frac{3}{8}$ & $\frac{1}{32}$	$6\frac{5}{4}$	90·2	74·4
$\frac{13}{16}$	$1\frac{3}{16}$	$4\frac{1}{2}$ & $\frac{1}{16}$	$7\frac{1}{8}$	89·6	74·3
$\frac{7}{8}$	$1\frac{1}{4}$	$4\frac{3}{4}$	$7\frac{1}{2}$	88·5	73·6
$\frac{15}{16}$	$1\frac{1}{4}$	$4\frac{1}{2}$	$7\frac{1}{2}$	87·2	72·2
1	$1\frac{5}{16}$	$4\frac{5}{8}$ & $\frac{1}{16}$	$7\frac{7}{8}$	86·5	71·9

Table 60.—Board of Trade Proportions for Double Butt-Straps, Double-Riveted.

Thickness of Steel Plate.	Diameter of Steel Rivet.	Pitch.	Lap.	Percentage.	
				Rivet.	Plate.
Inch.	Inches.	Inches.	Inches.		
$\frac{1}{2}$	$\frac{11}{16}$	$2\frac{3}{4}$ & $\frac{1}{16}$	$6\frac{7}{8}$	92·4	75·5
$\frac{9}{16}$	$\frac{3}{4}$	3	$7\frac{1}{2}$	91·5	75·1
$\frac{5}{8}$	$\frac{13}{16}$	$3\frac{1}{4}$	$8\frac{1}{8}$	90·2	75·0
$\frac{11}{16}$	$\frac{7}{8}$	$3\frac{3}{8}$	$8\frac{3}{4}$	90·6	74·0
$\frac{3}{4}$	$\frac{15}{16}$	$3\frac{1}{2}$ & $\frac{3}{32}$	$9\frac{3}{8}$	89·6	73·6
$\frac{13}{16}$	1	$3\frac{3}{4}$	10	90·12	73·3
$\frac{7}{8}$	$1\frac{1}{8}$	$4\frac{3}{8}$	$11\frac{1}{4}$	90·8	74·2
$\frac{15}{16}$	$1\frac{3}{16}$	$4\frac{1}{2}$ & $\frac{1}{16}$	$11\frac{7}{8}$	90·6	73·9
1	$1\frac{1}{4}$	$4\frac{3}{4}$ & $\frac{1}{16}$	$12\frac{1}{2}$	90·0	74·0

*** For the Board of Trade rules for riveted-joints, see page 150.

Riveted-Joints in Soft Steel-Plates.—Professor Kennedy in a paper read before the Institution of Mechanical Engineers,* gave the results of 14 series of experiments made by the Research Committee of that Institution, on the strength of riveted joints, of which the following is an abstract. The conclusions given below all refer to joints made in soft steel-plate with steel-rivets, the plates being in their natural state, that is, unannealed.

Further, it should be said that all dimensions, thicknesses of plate, &c., were measured by the most accurate means available; and that in every case the rivet or shearing area has been assumed to be that of the *holes*, not the nominal (or real) area of the rivets themselves. Also in every case the strength of the metal in the joint has been compared with that of strips cut from the same plates, and not merely with nominally similar material. It is thought that if these points had always been attended to, many of the discrepancies in published riveted-joint experiments would never have appeared.

1. The metal between the rivet-holes has a considerably greater tensile resistance per square inch than the unperforated metal. This excess tenacity amounted to more than 20 per cent. (both $\frac{3}{8}$-inch and $\frac{3}{4}$-inch plates) when the pitch of the rivet was about 1·9 diameters. In other cases $\frac{3}{8}$-inch plate gave an excess of 15 per cent. at fracture with a pitch of 2 diameters, of 10 per cent. with a pitch of 3·6 diameters, and of 6·6 per cent. with a pitch of 3·9 diameters; and $\frac{3}{4}$ inch plate gave 7·8 per cent. excess with a pitch of 2·8 diameters.

2. The shearing resistance of the rivet-steel is a matter upon which. as has been pointed out, further experiment is required. It may be taken as established that the resistance per square inch in double shear, is as great as that in single shear, so that allowance need not be made for the two shearing planes not being equally stressed. In *single*-riveted joints, however, the bending of the plates will put considerable tensile stress in the rivets; and this may diminish their apparent shearing resistance. In single-riveted joints it may be taken that about 22 tons per square inch is the shearing resistance of rivet-steel,† when the pressure on the rivets does not exceed about 40 tons per square inch. In double-riveted joints, with rivets of about $\frac{3}{4}$-inch diameter, most of the experiments gave about 24 tons per square inch as the shearing resistance, but the joints in Series XIII. went at 22 tons. In Series XIII. the larger rivets also went at a low load; but in the other double-riveted joints with large rivets these latter remained unbroken at a stress of 22 tons per square inch.

3. The size of the rivet-heads and ends plays a most important part in the strength of the joints, at any rate, in the case of single-riveted joints. An increase of about one-third in the weight of the rivets (all this increase, of course, going to the heads and ends) was found to add about $8\frac{1}{2}$ per cent. to the resistance of the joint, the rivets remaining unbroken at 22 tons per square inch, instead of shearing at a little over 20 tons. The additional

* Abstract of results of experiments on Riveted Joints, with their applications to practical work. By Professor Alexander B. W. Kennedy, honorary life-member of the Institution of Mechanical Engineers.

† In one pair of single-riveted joints only (Nos. 383 and 384 in Series VI.) a shearing resistance of over 24 tons per square inch was reached; in none of the others did it exceed 22·5 tons.

strength is no doubt due to the prevention of so great *tensile* stress in the rivets through distortion of the plates.

4. The strength of a joint made *across* a plate is equal to that of one made in the usual direction. (Both this conclusion and the last preceding are stated as the result of a very limited number of experiments; but there seems no reason to doubt their general truth.)

5. The intensity of bearing pressure on the rivets exercises, with joints proportioned in the ordinary way, a very important influence on their strength. So long as it does not much exceed 40 tons per square inch (measured on the projected area of the rivets) it does not seem to affect their strength; but pressures of 50 to 55 tons per square inch seem to cause the rivets to shear in most cases at stresses varying from 16 to 18 tons per square inch.

This conclusion is based on the experiments of Series X., in which the *margin* was made equal to the diameter of the drilled-hole. For ordinary joints, which are to be made equally strong in plate and in rivets, the bearing pressure should, therefore, probably not exceed 42 or 43 tons per square inch. For double-riveted butt-joints, perhaps, as will be noted later, a larger pressure may be allowed, as the shearing stress may probably not exceed 16 to 18 tons per square inch when the plate tears. But in this case it would probably be wise to increase the margin.

6. A margin (or net distance from outside of holes to edge of plate) equal to the diameter of the drilled hole has been found sufficient in all cases hitherto tried.

7. To attain the maximum strength of a joint the breadth of lap must be such as to prevent it from breaking zigzag. Such a method of fracture must inevitably be accompanied by unequal stresses in the plate straight between the rivet-holes, and by consequent diminution of strength. It has been found that the net metal measured zigzag should be from 30 to 35 per cent. in excess of that measured straight across, in order to ensure a straight fracture. This corresponds to a diagonal pitch of $\frac{2}{3}p + \frac{d}{3}$, if p be the straight pitch, and d the diameter of the rivet-hole. To find the proper breadth of lap for a double-riveted joint, it is probably best to proceed by first setting this pitch off, and then finding from it the longitudinal pitch, or distance between the centres of the lines of rivets.

Table 61.—Slip of Riveted Joints.

Rivet Diameter.	Type of Joint.	Riveting.	Slipping Load per Rivet.
$\frac{3}{4}$-inch	Single-riveted	Hand	2·5 tons
,,	Double-riveted	,,	3·0 to 3·5 tons
,,	,, ,,	Machine	7 tons
1-inch	Single-riveted	Hand	3·2 tons
,,	Double-riveted	,,	4·3 tons
,,	,, ,,	Machine	8 to 10 tons

8. Visible slip or "give" occurs always in a riveted joint at a point very much below its breaking load, and by no means proportional to that load.

A careful collation of all the results obtained in measuring the slip indicates pretty clearly that it depends upon the number and size of the rivets in the joint, rather than anything else; and that it is tolerably constant for a given size of rivet in a given type of joint. The loads *per rivet* at which a joint will commence to slip visibly are *approximately* as given in Table 61.

To find the probable load at which a joint of any breadth will commence to slip, it is only necessary to multiply the number of rivets in the given breadth by the proper figure taken from the last column of the Table above. It will be understood that the above figures are not given as exact; but they represent very well the results of the experiments in all series from VIII. to XIII.; except Series X., in which the average (for 1-inch rivets) was much lower than that given above. In this series, however, the proportions of the joints were intentionally somewhat abnormal; and it is perhaps not to be expected that in this respect the results should agree with those of the other experiments.

This result as to the slipping of a joint, although perhaps unexpected, is not contrary to what ought to have been expected. For experiments show that, long before stresses are reached which could visibly stretch the plates of a joint, there will be quite measurable shear of the rivet. The visible slip therefore will consist almost wholly of this shear, the magnitude of which will depend primarily on the number and size of the rivets in the

Table 62.—Shear of Rivet-Steel Pins, 1 Inch Diameter.

Shearing Stress in Lbs. per Square Inch.	Test Nos.:—					
	343	344	345	346	347	348
	Amount of Shear in Inches.					
0	0·0	0·0	0·0	0·0	0·0	0·0
6365	·010	·013	·016	·022	·055	·021
12730	·022	·028	·030	·034	·066	·032
19100	·034	·040	·042	·048	.078	·043
25460	·055	·060	·060	·071	·091	·062
28320	·066	...	...	...	...	...
31830	·080	·086	·082	·091	·113	·183
35010	·093	...	...	...	...	...
38190	·113	·113	·108	·114	·140	·108
41380	·141	...	...	...	...	...
44550	·168	·152	·142	·170	·171	·155
47740	·200	...	...	...	...	...
50910	·242	·200	·196	·248	·238	·222
54110	...	...	...	...	...	...
Breaking Load per sq. in. Lb.	54110	54930	55240	52 830	56670	53530
Breaking Load per sq. in. Tons.	24·15	24·52	24·66	23·59	25·29	23·90

joint. Anything that will hold the plates up better together, such as hydraulic pressure on the rivets, might be expected to diminish this shear or delay its commencement, exactly as seems to have happened. The Table 62 gives the result of experiments on this matter which were made along with those given in the Committee's first report, but which have not previously been published in the Proceedings of the Institution. The experiments are on 1-inch turned pins of rivet steel, tested in the single-shear apparatus already described. Of course the shear would commence later, and be at first smaller in extent, when the pin was replaced by an actual rivet, and when the plates were thus forcibly held together, instead of being quite free, except so far as held from motion by the resistance to shear.

9. The value of machine-riveting as compared with hand-riveting, in cases when sound hand-riveting is possible, lies mainly, if not entirely, in the fact that it doubles the load at which the slip of a joint commences. This conclusion is subject to modification by future experiments with the use of higher pressure in closing the rivet, which may probably still further raise the slipping load, so that the advantage of machine-riveting may quite possibly be even greater than it is here assumed to be; but there is no indication that it is likely to affect the ultimate strength of the joint. The question of *friction* in joint, which has not been specially experimented on by the Committee, no doubt comes in the same way. The friction induced by the rivet will affect the point at which slip commences; but can hardly have much, if any, relation to the breaking load.

It is thought that the load at which visible slip commences is probably proportional to the load at which leakage would begin in a boiler. Looked at this way, it will be seen that the great value of hydraulic riveting appears to lie rather in the increased security and stiffness it gives at ordinary working loads than in any actual raising of the breaking load. From a practical point of view the former is probably the more, and not the less, important function.

Table 63.—RATIO OF THE DIAMETER OF THE RIVET HOLE TO THE THICKNESS OF PLATE AND OF PITCH TO DIAMETER OF HOLE IN $\frac{3}{8}$-INCH PLATE.

Original Tenacity of Plate. Tons per Square Inch.	Shearing Resistance of Rivets. Tons per Square Inch.	Ratio $\frac{d}{t}$	Ratio $\frac{p}{d}$	Ratio $\frac{\text{Plate Area}}{\text{Rivet Area}}$
30	22	2·48	2·30	0·667
28	22	2·48	2·40	0·785
30	24	2·28	2·27	0·713
28	24	2·28	2·36	0·690

10. The experiments point to very simple rules for the proportioning of joints of maximum strength, which will be mentioned before any other joints are discussed. Assuming that a bearing pressure of 43 tons per square inch may be allowed on the rivet, and that the excess tenacity

of the plate is 10 per cent. of its original strength,* Table 63 gives the values of the ratios of diameter d of the hole to thickness t of plate $\left(\frac{d}{t}\right)$, and of pitch p to diameter of hole $\left(\frac{p}{d}\right)$, in joints of maximum strength in $\frac{3}{8}$-in. plate.

Summed up and rounded off, this means that the diameter of the *hole* (not the diameter of the rivet cold) should be $2\frac{1}{3}$ times the thickness of the plate, and the pitch of the rivets $2\frac{3}{8}$ times the diameter of the holes.† In mean also it makes the plate-area 71 per cent. of the rivet-area.

If a smaller rivet be used than that here specified, the joint will not be of uniform and therefore not of maximum strength; but with any other size of rivet the best result will be got by use of the pitch obtained from the formula formerly cited,—

$$p = a\frac{d^2}{t} + d$$

where, as before, d is the diameter of the hole.

The value of the constant a in this equation is as follows:—

For 30-ton plate and	22-ton rivets,	$a = 0\cdot524$
,, 28	,,	22	,,	,,	$0\cdot558$
,, 30	,,	24	,,	,,	$0\cdot570$
,, 28	,,	24	,,	,,	$0\cdot606$

or in the mean, the pitch $p = 0\cdot56\frac{d^2}{t} + d$.

It should be noticed that with too small rivets this gives pitches often considerably smaller in proportion than $2\frac{3}{8}$ times the diameter.

For double-riveted lap-joints, a similar calculation to that given above, but with a somewhat smaller allowance for excess tenacity on account of the large distance between the rivet-holes, shows that for joints of maximum strength the ratio of diameter to thickness should remain precisely as in single-riveted joints; while the ratio of pitch to diameter of hole should be 3·64 for 30-ton plates and 22 or 24-ton rivets, and 3·82 for 28-ton plates with the same rivets.

Here, still more than in the former case, it is likely that the prescribed size of rivet may often be inconveniently large. In this case the diameter of rivet should be taken as large as possible; and the strongest joint for a given thickness of plates and diameter of hole can then be obtained by using the pitch given by the equation

$$p = a\frac{d^2}{t} + d,$$

where the values of the constant a for different streng.ns of plate and rivet may be taken as follows:—

* The excess strength is taken lower than the average result of the experiment, because it is probable enough that the steel used had more than the average softness.

† The small difference here from the constants formerly given is due to the assumption, now quite justified, of a somewhat greater bearing pressure than was then allowed.

Table 64.—PROPORTION OF DOUBLE-RIVETED LAP-JOINTS, IN WHICH

$$p = a\frac{d^2}{t} + d.$$

Thickness of Plate.	Original Tenacity of Plate. Tons per Square Inch.	Shearing Resistance of Rivets. Tons per Square Inch.	Value of Constant. a.
$\frac{3}{8}$-inch.	30	24	1·15
,,	28	24	1·22
,,	30	22	1·05
,,	28	22	1·12
$\frac{3}{4}$-inch.	30	24	1·17
,,	28	24	1·25
,,	30	22	1·07
,,	28	22	1·14

Practically we may say that, having assumed the rivet diameter as large as possible, we can fix the pitch as follows, for any thickness of plate from $\frac{3}{8}$ to $\frac{3}{4}$ inch:—

For 30-ton plate and 24-ton rivets }
,, 28 ,, 22 ,, } $p = 1·16\frac{d^2}{t} + d$

,, 30 ,, 22 ,, $p = 1·06\frac{d^2}{t} + d$

,, 28 ,, 24 ,, $p = 1·24\frac{d^2}{t} + d$

In double-riveted butt-joints it is impossible to develop the full shearing resistance of the joint without getting excessive bearing pressure, because the shearing area is doubled without increasing the area on which the pressure acts. In a previous report it was shown that, considering only the plate resistance and the bearing pressure, and taking this latter as 45 tons per square inch, the best pitch would be about four times the diameter of the hole. It is probable that a pressure of from 45 to 50 tons per square inch on the rivets will cause shearing to take place at from 16 to 18 tons per square inch. Working out the equations as before, but allowing excess strength of only 5 per cent. on account of the large pitch, we find that the proportions of double-riveted butt-joints of maximum strength under given conditions are those of the following Table:—

Table 65.—PROPORTIONS OF DOUBLE-RIVETED BUTT-JOINTS.

Original Tenacity of Plate. Tons per Square Inch.	Shearing Resistance of Rivet. Tons per Square Inch.	Bearing Pressure. Tons per Square Inch.	Ratio $\frac{d}{t}$	Ratio $\frac{p}{d}$
30	16	45	1·80	3·85
28	16	45	1·80	4·06
30	18	48	1·70	4·03
28	18	48	1·70	4·27
30	16	50	2·00	4·20
28	16	50	2·00	4·42

Practically, therefore, it may be said that we get a double-riveted butt-joint of maximum strength by making the diameter of hole about 1·8 times the thickness of the plate, and making the pitch 4·1 times the diameter of the hole. These are very nearly the proportions which were used for the $\frac{3}{8}$-inch joints in Series XI. to XIII.; for the $\frac{3}{4}$-inch joints the diameter of the rivet was (as with the lap joint) less than indicated by theory. In thick plates, where it is thought impossible or inconvenient to make the rivet-holes so large as 1·8 times the thickness, the best pitch for any assumed diameter of rivet cannot be found by the method formerly used; for here we have not a given maximum shearing stress to work to, but rather the shearing stress which in a given joint causes a given maximum pressure on the rivets. The best ratio of pitch to diameter of hole in double-riveted butt-joints of maximum strength for *any* assumed diameter of hole d is therefore the same as that given in the last Table, or in mean, 4·1.

11. All the experiments hitherto made have necessarily connected themselves with the question of *strength*, and the proportions just given belong to *joints of maximum strength*. But in a boiler the one part of the joint, the plate, is much more affected by time than the other part, the rivets. It is therefore not unreasonable to estimate the percentage by which the plates might be weakened by corrosion, &c., before the boiler would become unfit for use at its proper steam-pressure, and to add correspondingly to the plate-area. Probably the best thing to do in this case is to proportion the joint not for the actual thickness of plate, but for a nominal thickness less than the actual by the assumed percentage. In this case the joint will be approximately one of uniform strength by the time it has reached its final workable condition; up to which time the joint as a whole will not really have been weakened, the corrosion only gradually bringing the strength of the plates down to that of the rivets. Thus, suppose a single-riveted lap-joint in $\frac{5}{8}$-inch plate is in question, and it is considered that corrosion will make this equal to only a $\frac{1}{2}$-inch plate before the boiler-pressure has to be lowered. The rivet should then be proportioned as if the plate had a thickness of 0·5 inch, which would give, for 30-ton plate and 22-ton rivets, a diameter of hole of 1·24 inch. Assume this as too large to be convenient, and take the diameter of hole as 1 inch. Then the pitch will be

$$p = \frac{0·524}{0·5} + 1 = 2·05 \text{ inches.}$$

The ratio of plate to rivet area to start with will be 0·835, means of course that the plate is in excess; but the ratio will diminish until it reaches 0·667, when the strength of the plate has become equivalent to that of one only $\frac{1}{2}$-inch thick, as was required. The efficiency of the joint would be 45 per cent., whereas the best efficiency of a joint in $\frac{5}{8}$-inch plate with 1-inch holes ($p = 1·84$ inches) would be 50 per cent., and the best possible efficiency of a single-riveted lap-joint in $\frac{5}{8}$-inch plate under the given condition of strength would be about 62 per cent.

It is hardly necessary to point out how strongly these figures indicate the necessity of using as large rivets as possible, and of taking every possible means to reduce the allowance necessary for corrosion. For a boiler such as has been discussed is absolutely no stronger than one of $\frac{1}{2}$-inch plate throughout, if only the thickness of the latter could be kept unreduced at the joints.

The above conclusions, as before stated, all refer to joints made in soft-steel plates—unannealed—with steel rivets.

STRENGTH OF CYLINDRICAL BOILER-SHELLS.

The Strength of a Cylindrical Shell *without seams* may be found by this *Rule* :—

$$\text{Pressure in lbs. per sq. inch} = \frac{\text{Tensile strength of the plates in lbs.} \times \text{thickness of the plate in inches} \times 2}{\text{Internal diameter of the shell in inches.}}$$

Example : If one-sixth of the tensile strength of the plates, or 8000 lbs. strain be allowed on a cylindrical boiler-shell, without seams, of 50 inches diameter, composed of iron-plate $\frac{5}{16}$ inch thick, what pressure of steam may be used inside the shell?

Then $\frac{8000 \text{ lbs.} \times 2 \times \cdot 3125 \text{ inch thick}}{50 \text{ inches internal diameter of shell}} = 100$ lbs. per square inch. But as boilers are composed of belts of plates with riveted seams, it is necessary to take the strength of the riveted-joints into consideration.

The Maximum Working Pressure of a Cylindrical Boiler-Shell may be found by the following *Rule :* Multiply the ultimate tensile strength of the plate in lbs. per square inch by the thickness of the plate in inches, and by the percentage of strength of the riveted joint to that of the solid plate, and divide the product by the product of the internal radius of the boiler-shell in inches by the factor of safety.

Example : Required the maximum working strength of a wrought-iron cylindrical boiler-shell, 5 feet 6 inches internal diameter, thickness of plate $\frac{3}{8}$ inch, the strength of the single-riveted lap joint being 56 per cent. of the solid plate : ultimate tensile strength of the plate = 21 tons, factor of safety = 6.

Then $\frac{21 \text{ tons} \times 2240 \text{ lbs.} \times \cdot 375 \text{ plate} \times \cdot 56}{33 \text{ inches radius} \times 6 \text{ factor of safety}} = 49 \cdot 89$ lbs. per square inch ; and this shell would burst with a pressure of 49·89 × 6, the factor of safety, = 299·34 lbs. per square inch.

The Internal Diameter of Boiler-Shell for a given Thickness of Plate and Pressure may be found by this *Rule :* Multiply the ultimate tensile strength of the plate in lbs. per square inch by the thickness of the plate in inches, and by the percentage of strength of the riveted joint to that of the solid plate, and divide the product by the product of the working pressure in lbs. per square inch by the factor of safety.

Example : Required the internal diameter of boiler-shell suitable for the particulars of the last example.

Then $\frac{21 \text{ tons} \times 2240 \text{ lbs.} \times \cdot 375 \text{ plate} \times \cdot 56}{49 \cdot 89 \text{ lbs.} \times 6, \text{ factor of safety}} = 33$ inches radius, and 33 inches radius × 2 = 5 feet 6 inches internal diameter.

The Thickness of Plate for a Boiler-Shell for a given Diameter and Working Pressure may be found by this *Rule:* Multiply the working pressure in lbs. per square inch by the factor of safety, and by the internal radius in inches of the boiler-shell, and divide the product by the product of the ultimate tensile strength of the plate by the percentage of strength of the riveted joint to that of the solid plate.

Example: Required the thickness of plate suitable for the boiler-shell given in the last example.

$$\text{Then } \frac{49{\cdot}89 \text{ lbs. pressure} \times 6, \text{ factor of safety} \times 33 \text{ inches radius of shell}}{21 \text{ tons} \times 2240 \text{ lbs.} \times {\cdot}56, \text{ percentage of strength of joint}}$$
$= {\cdot}375$ inch.

The Total Pressure of Steam on the Internal Surface of a Boiler may be found by multiplying the internal surface of the boiler by the pressure of the steam per unit of surface.

Example: A Lancashire boiler, 6 feet diameter, 24 feet long, with two flues each 2 feet 3 inches diameter, is worked at a pressure by the steam gauge of 60 lbs. per square inch: required the total pressure of the steam on the internal surface of the boiler.

Then,

The surface of the boiler-shell is 6 feet × 3·1416 × 24 feet . . . . =	352·39 square feet.
The surface of the two flues is 2·25 feet × 3·1416 × 24 × 2 . . . . =	339·29 „
The surface of the two flat ends of the boiler is (6 × 6 feet) − (2·25 × 2·25 feet × 2) × ·7854 × 2 =	40·64 „
Total internal surface in square feet . . .	732·32

$$\text{and, } \frac{732{\cdot}32 \text{ square feet} \times 144 \times 60 \text{ lbs. pressure}}{2240 \text{ lbs.}} = 2824{\cdot}66 \text{ tons, the}$$

total pressure of the steam on the internal surface of the boiler.

Bursting-Pressure of Cylindrical Boilers.—The strength of cylindrical shells to resist internal bursting pressure in a direction parallel to their axis may be calculated by this *Rule*—

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

Where P = The bursting pressure in lbs. per square inch.
„ T = The thickness of the shell in sixteenths of an inch.
„ D = Diameter of shell in quarter feet.
„ C = A constant, being as follows:—
1097 for single-riveted wrought-iron plates.
1372 for double-riveted wrought-iron plates.
1723 for single-riveted steel-plates.
2156 for double-riveted steel-plates.

The plates to be sound, and of good material.

Collapsing Pressure of Wrought-Iron Cylindrical Flue Tubes.—The rule usually employed for the strength of cylindrical tubes subject to external pressure, is that deduced by Fairbairn from the results of a series of experiments. It is as follows:—

$$P = \frac{806300 \times t^{2 \cdot 19}}{L \times D}$$

where P = The collapsing pressure in lbs. per square inch.
,, t = The thickness of tube in inches.
,, L = The length of tube in feet.
,, D = The diameter of tube in inches.

The following values of $t^{2 \cdot 19}$, usually required, may be useful:—

$\left(\frac{3}{16}\right)^{2 \cdot 19} = \cdot 02558.$	$\left(\frac{13}{32}\right)^{2 \cdot 19} = \cdot 13908.$
$\left(\frac{7}{32}\right)^{2 \cdot 19} = \cdot 03585.$	$\left(\frac{7}{16}\right)^{2 \cdot 19} = \cdot 16358.$
$\left(\frac{1}{4}\right)^{2 \cdot 19} = \cdot 04803.$	$\left(\frac{15}{32}\right)^{2 \cdot 19} = \cdot 19027.$
$\left(\frac{9}{32}\right)^{2 \cdot 19} = \cdot 06216.$	$\left(\frac{1}{2}\right)^{2 \cdot 19} = \cdot 21915.$
$\left(\frac{5}{16}\right)^{2 \cdot 19} = \cdot 07829$	$\left(\frac{17}{32}\right)^{2 \cdot 19} = \cdot 25027.$
$\left(\frac{11}{32}\right)^{2 \cdot 19} = \cdot 09646.$	$\left(\frac{9}{16}\right)^{2 \cdot 19} = \cdot 28364.$
$\left(\frac{3}{8}\right)^{2 \cdot 19} = \cdot 11671.$	$\left(\frac{5}{8}\right)^{2 \cdot 19} = \cdot 35725.$

Instead of the 2·19 power, the square of the thickness is usually taken as sufficiently correct in practice, in which case the formula becomes—

$$P = \frac{806300 \times t^2}{L \times D}.$$

Another rule sometimes used to find the collapsing pressure of wrought-iron furnace-tubes in lbs. per square inch, is—

$$P = \frac{465314 \times t^2}{L \times D};$$

the notation being the same as in the previous example.

Working Pressure of Wrought-Iron Cylindrical Flue-Tubes.—The formula generally used for ascertaining the working pressure to be allowed for the furnaces of wrought-iron boilers is as follows:—

$$\text{Working pressure in lbs. per square inch} = \frac{89600 \times t^2}{L \times D},$$

where t, L, and D, stand for thickness of plate in inches, length of flue in feet, and diameter of tube in inches.

The above rule would appear to be intended to allow one-ninth of the collapsing-pressure obtained by Fairbairn's formula, given above.

Collapsing Pressure of Lap-Jointed Tubes.—Mr. Richards, of the Marine Department of the Board of Trade, considers Fairbairn's formula to be unsuitable for obtaining the collapsing pressure of wrought-iron furnace-tubes constructed with lap-joints, and gives the following as the proper formula for such tubes, based on Fairbairn's deductions :—

Collapsing pressure of lap-jointed furnace-tubes in lbs. per square inch =

$$\frac{322462 \times t^2}{L \times D},$$

where the notation is the same as in the previous rule.

The Collapsing Pressure of Wrought-Iron Tubes is frequently calculated by the following formula, which provides for the tubes being slightly out of the circular form, as long tubes generally are—

$$P = \frac{375023 \times t^2}{L \times D}$$

where D = the diameter in inches, L = the length in feet, t = the thickness in inches, P = the collapsing pressure in lbs. per square inch.

BOARD OF TRADE RULES FOR STEAM BOILERS.

Iron Steam-Boilers and Superheaters.

The Surveyor to fix Pressures on Safety Valves.—The Surveyor is required by the Act to fix the limits of weight to be placed on the safety valves of passenger steam-ships. In performing this very responsible and onerous duty he must be very careful, as, in the event of accident, it will be necessary for him to satisfy the Board of Trade that he used due caution. On the one hand he must be careful as regards safety, and on the other hand he must not unduly reduce the pressure on a boiler. The Surveyor himself having fixed the limits of the weight, is then required to declare that in his judgment the boiler and machinery are sufficient for the service intended, and in good condition, and that they will be sufficient for twelve months, or such other period as he may, in his judgment, determine. For his guidance the following directions are given, and he should not depart from them in any case, without first reporting particulars to the Board of Trade, and asking for instructions.

Working Pressure to be fixed by Calculation.—The Surveyor should fix the working pressure for boilers by a series of calculations of the strength of the various parts, and according to the workmanship and material. The Board of Trade, upon the request of certain ship-builders and ship-owners, have arranged to receive for examination by their surveyors, plans and particulars of boilers before the commencement of manufacture, by these means hoping to prevent questions arising after the boilers are finished and on board. This practice has been found to work well in saving time to the Surveyors, and in preventing expense, inconvenience, and delay to owners. The senior Engineer-Surveyors should therefore receive and report on any plans of boilers intended for passenger vessels, that may be submitted in due course with the form surveys. They are not to report on any tracing or plan that is not accompanied by that form. When the Surveyor has received plans and tracings of new boilers, or of alteration of boilers, and has approved of them, he will of course be careful in making his examination from time to time, to see that they are followed in construction. When he has not had the plans submitted, but is called in to survey a boiler, he will of course measure the parts, note the details of construction, and, if necessary, bore the plates to ascertain their thickness, &c., before he gives his declaration; and in the event of any novelty in construction, or of any departure from the practice of staying and strengthening noted in these instructions, he should report full particulars to the Board of Trade before fixing the working pressure. The Surveyor cannot declare a boiler to be safe unless he is fully informed as to its construction, material, and workmanship; he should, therefore, be very careful how he ventures to give a declaration for a boiler that he is not called in to survey until after it is completed, and fixed in the ship.

Stays.—In the case of new boilers, the Surveyors may allow a stress not exceeding 7000 lbs. per square inch of net section on solid iron screwed-stays supporting flat surfaces, but the stress should not exceed 5000 lbs. when the stays have been welded or worked in the fire.

The Areas of Diagonal Stays are found in the following way:—Find the area of a direct stay needed to support the surface, multiply this area by the length of the diagonal stay, and divide the product by the length of a line drawn at right angles to the surface supported to the end of the diagonal stay: the quotient will be the area of the diagonal stay required.

Gusset Stays.—When gusset-stays are used, their area should be in excess of that found in the above way.

Girders for Flat Surfaces.—When the tops of combustion-boxes, or other parts of a boiler, are supported by solid rectangular girders, the following formula should be used for finding the working pressure to be allowed on the girders, assuming that they are not subjected to a greater temperature than the ordinary heat of steam, and that, in the case of combustion-chambers, the ends are fitted to the edges of the tube-plate, and the back plate of the combustion-box:—

$$\frac{C \times d^2 \times T}{(W-P)\ D \times L} = \text{working pressure.}$$

W = width of combustion-box in inches.
P = pitch of supporting bolts in inches.
D = distance between the girders from centre to centre in inches.
L = length of girder in feet. d = depth of girder in inches.
T = thickness of girder in inches. N = number of supporting bolts.

$$C = \frac{N \times 1000}{N + 1} \text{ when the number of bolts is odd.}$$

$$C = \frac{(N + 1)\ 1000}{N + 2} \text{ when the number of bolts is even.}$$

The working pressure for the supporting bolts and for the plate between them should be determined by the rule for ordinary stays.

Flat Surfaces of Boilers.—The pressure on plates forming flat surfaces is found by the following formula:—

$$\frac{C \times (T+1)^2}{S-6} = \text{working pressure.}$$

T = thickness of the plate in sixteenths of an inch.
S = surface supported in square inches.
C = constant according to the following circumstances.
C = 100 when the plates are not exposed to the impact of heat or flame, and the stays are fitted with nuts and washers, the latter being at least three times the diameter of the stay and two-thirds the thickness of the plates they cover. If the diameter of riveted washers be at least two-thirds the pitch of the stays, and the thickness not less than the plates they cover, the constant may be increased to 150. When doubling-plates are fitted of the same thickness as the plates they cover, and not less in width than two-thirds of the pitch of the stays, the constant may be increased to 160. When doubling-plates cover the whole of the flat surface the case should be submitted for the consideration of the Board.
C = 90 when the plates are not exposed to the impact of heat or flame, and the stays are fitted with nuts only.
C = 60 when the plates are exposed to the impact of heat or flame, and steam in contact with the plates, and the stays fitted with nuts and washers, the latter being at least three times the diameter of the stay and two-thirds the thickness of the plate they cover.
C = 54 when the plates are exposed to the impact of heat or flame, and steam in contact with the plate, and the stays fitted with nuts only.
C = 80 when the plates are exposed to the impact of heat or flame, with water in contact with the plates, and the stays screwed into the plate and fitted with nuts.
C = 60 when the plates are exposed to the impact of heat or flame, with water in contact with the plate, and the stays screwed into the plate, having the ends riveted over to form a substantial head.
C = 36 when the plates are expcsed to the impact of heat or flame, and steam in contact with the plates, with the stays screwed into the plate, and having the ends riveted over to form a substantial head.

In cases where plates are stiffened by T or L irons, and a greater pressure is required for the plate than is allowed by the use of the above constants, the case should be submitted for the consideration of the Board of Trade.

When the riveted ends of screwed stays are much worn, or when the nuts are burned, the constants should be reduced, but the Surveyor must act according to the circumstances that present themselves at the time of survey, and it is expected that in cases where the riveted ends of screwed stays in the combustion-boxes and furnaces are found in this state it will be often necessary to reduce the constant 60 to about 36.

Compressive Stress on Tube-Plates.—The Surveyors should not in any case allow a greater compressive stress on the tube-plates than 8000 lbs., which is that used in the following formula :—

$$\frac{(D - d)\,T \times 16{,}000}{W \times D} = \text{working pressure.}$$

D = least horizontal distance between centres of tubes in inches.
d = inside diameter of ordinary tubes in inches.
T = thickness of tube-plate in inches.
W = extreme width of combustion-box in inches from front of tube-plate to back of fire-box, or distance between combustion-box tube-plates when boiler is double-ended and the box common to the furnaces at both ends.

Cylindrical Boilers.—The Board of Trade consider that boilers well constructed, well designed, and made of good material should have an advantage in the matter of working pressure over boilers inferior in any of the above respects, as unless this is done the superior boiler is placed at a disadvantage, and good workmanship and material will be discouraged. They have therefore caused the following rules to be prepared :—

When cylindrical boilers are made of the best material with all the rivet-holes drilled in place and all the seams fitted with double butt straps each of at least five-eighths the thickness of the plates they cover, and all the seams at least double riveted with rivets having an allowance of not more than 75 per cent. over the single shear, and provided that the boilers have been open to inspection during the whole period of construction, then 5 may be used as the factor of safety. The tensile strength of the iron is to be taken as equal to 47,000 lbs. per square inch with the grain, and 40,000 lbs. across the grain. But when the above conditions are not complied with, the additions in the following scale must be added to the factor 5, according to the circumstances of each case:—

A†	·15	To be added when all the holes are fair and good in the longitudinal seams, but drilled out of place after bending.
B†	·3	To be added when all the holes are fair and good in the longitudinal seams, but drilled out of place before bending.
C	·3	To be added when all the holes are fair and good in the longitudinal seams, but punched after bending instead of drilled.
D	·5	To be added when all the holes are fair and good in the longitudinal seams, but punched before bending.
E*	·75	To be added when all the holes are not fair and good in the longitudinal seams.
F	·1	To be added if the holes are all fair and good in the circumferential seams, but drilled out of place after bending
G†	·15	To be added if the holes are fair and good in the circumferential seams, but drilled before bending.
H	·15	To be added if the holes are fair and good in the circumferential seams, but punched after bending.
I†	·2	To be added if the holes are fair and good in the circumferential seams, but punched before bending.

J*	·2	To be added if the holes are not fair and good in the circumferential seams.
K	·2	To be added if double butt straps are not fitted to the longitudinal seams, and the said seams are lap and double riveted.
L	·1	To be added if double butt straps are not fitted to the longitudinal seams, and the said seams are lap and treble riveted.
M	·3	To be added if only single butt straps are fitted to the longitudinal seams, and the said seams are double riveted.
N	·15	To be added if only single butt straps are fitted to the longitudinal seams, and the said seams are treble riveted.
O	1·	To be added when any description of joint in the longitudinal seams is single riveted.
P	·1	To be added if the circumferential seams are fitted with single butt straps and are double riveted.
Q	·2	To be added if the circumferential seams are fitted with single butt straps and are single riveted.
R	·1	To be added if the circumferential seams are fitted with double butt straps and are single riveted.
S††	·1	To be added if the circumferential seams are lap joints and are double riveted.
T	·2	To be added if the circumferential seams are lap joints and are single riveted.
U	·25	To be added when the circumferential seams are lap and the strakes of plates are not entirely under or over.
V‡	·3	To be added when the boiler is of such a length as to fire from both ends, or is of unusual length, such as flue boilers; and the circumferential seams are fitted as described opposite P, R, and S, but, of course, when the circumferential seams are as described opposite Q and T, V ·3 will become V ·4.
W*	·4	To be added if the seams are not properly crossed.
X*	·4	To be added when the iron is in any way doubtful, and the surveyor is not satisfied that it is of the best quality.
Y†††	1·65	To be added if the boiler is not open to inspection during the whole period of its construction.

Where marked * the allowance may be increased still further if the workmanship or material is very doubtful or very unsatisfactory.

† When the holes are to be rimered or bored out in place the case should be submitted to the Board as to the reduction or omission of A, B, G, and I as heretofore,

‡ When boilers are comparatively short the cases should be submitted to the Board for consideration as to the omission of V as heretofore.

†† When the circumferential seams are lapped and treble riveted the case should be submitted to the Board.

††† When surveying boilers that have not been open to inspection during construction, the case should be submitted to the Board as to the factors to be used.

The Strength of the Joints is found by the following method :—

$$\frac{(\text{Pitch} - \text{Diameter of rivet}) \times 100}{\text{Pitch.}} = \left\{ \begin{array}{l} \text{Percentage of strength of plate at joint} \\ \text{as compared with the solid plate.} \end{array} \right.$$

$$\frac{(\text{Area of rivet} \times \text{No. of rows of rivets}) \times 100}{\text{Pitch} \times \text{thickness of plate.}} = \left\{ \begin{array}{l} \text{Percentage of strength of} \\ \text{rivets as compared with} \\ \text{the solid plate.*} \end{array} \right.$$

Then take iron as equal to 47,000 lb. per square inch, and use the smaller of the two percentages as the strength of the joint, and adopt the factor of safety as found from the preceding scale:

$$\frac{(47{,}000 \times \text{percentage of strength of joint}) \times \text{twice the thickness of the plate in inches.}}{\text{Inside diameter of the boiler in inches} \times \text{factor of safety.}}$$

$=$ Pressure to be allowed per square inch on the safety-valves.

In the case of Zigzag Riveting the strength through the plate diagonally between the rivets is equal to that horizontally between the rivets, when diagonal pitch $= \frac{6}{10}$ horizontal pitch $+ \frac{4}{10}$ diameter of rivet.

Plates that are Drilled in Place must be taken apart and the burr taken off, and the holes slightly counter-sunk from the outsides.

Butt-Straps.—Butt-straps must be cut from plates and not from bars, and must be of as good a quality as the shell-plates, and for the longitudinal seams must be cut across the fibre. The rivet-holes may be punched or drilled when the plates are punched or drilled OUT of place, but when drilled in place must be taken apart and the burr taken off and slightly counter-sunk from the outside. When single butt-straps are used and the rivet-holes in them punched they must be one-eighth thicker than the plates they cover.

The Diameter of the Rivets must not be less than the thickness of the plates of which the shell is made, but it will be found when the plates are thin, or when lap joints or single butt-straps are adopted, that the diameter of the rivets should be in excess of the thickness of the plates.

Dished Ends must be stayed as flat surfaces; but when they are theoretically equal to the pressure needed, when considered as portions of spheres, the stays, when solid, may have a stress of 14,000 lbs. per square inch of net section, but the stress should not exceed 10,000 lbs. when the stays have been welded or worked in the fire. Truly hemispherical ends subjected to internal pressure may be allowed double the pressure that is suitable for a cylinder of the same diameter and thickness.

All Man-holes and Openings must be stiffened with compensating-rings of at least the same effective sectional area as the plates cut out, and in no case should the plate-rings be less in thickness than the plates to which they are attached. The openings in the shells of cylindrical boilers should have their shorter axes placed longitudinally. It is very desirable that the compensating-rings round openings in flat surfaces be made of L or T iron. Cast-iron doors are not to be passed.

The neutral part of boiler-shells under steam-domes must be efficiently stiffened and stayed, as serious accidents have arisen from the want of

* If the rivets are exposed to double shear multiply the percentage as found by 1·75.

such precautions. The boilers must be tested by hydraulic pressure to twice the working pressure in the presence and to the satisfaction of the Board's Surveyors.

Circular Furnaces.—Circular furnaces with the longitudinal joints welded or made with a butt strap double-riveted, or double butt straps single-riveted :—

$$\frac{90{,}000 \times \text{the square of the thickness of the plate in inches}}{(\text{Length in feet} + 1) \times \text{diameter in inches}} = \text{working pressure}$$

per square inch, provided it does not exceed that found by the following formula :—

$$\frac{9000 \times \text{thickness in inches}}{\text{Diameter in inches}} = \text{working pressure per square inch.}$$

The second formula limits the crushing stress on the material to 4500 lbs. per square inch. The length is to be measured between the rings it the furnace is made with rings.

If the longitudinal joints instead of being butted are lap-jointed in the ordinary way, and double-riveted, then 75,000 should be used instead of 90,000, but where the lap is bevelled and so made as to give the flues the form of a *true* circle, then 80,000 may be used. When the material or the workmanship is not of the best quality, the constants given above must be reduced, that is to say—the 90,000 will become 80,000; the 80,oco will become 70,000; the 70,000 will become 60,000. When the material and the workmanship are not of the best quality, such constants will require to be further reduced, according to circumstances and the judgment of the Surveyor, as in the case of old boilers. One of the conditions of best workmanship is, that the joints are either double-riveted with single butt-straps, or single-riveted with double butt-straps, and the holes drilled after the bending is done and when in place, and the plates afterwards taken apart, the burr on the holes taken off, and the holes slightly counter-sunk on the outside.

The following examples will serve to show the application of the constants for the different cases that may arise :—

Furnaces with butt joints and drilled rivet holes.	90,000 where the longitudinal seams are welded. 90,000 where the longitudinal seams are double riveted and fitted with single butt straps. 80,000 where the longitudinal seams are single riveted and fitted with single butt straps. 90,000 where the longitudinal seams are single riveted and fitted with double butt straps.
Furnaces with butt joints and punched rivet holes.	85,000 where the longitudinal seams are double riveted and fitted with single butt straps. 75,000 where the longitudinal seams are single riveted and fitted with single butt straps. 85,000 where the longitudinal seams are single riveted and fitted with double butt straps. 80,000 where the longitudinal seams are double riveted and bevelled.
Furnace with lapped joints and drilled rivet holes.	75,000 where the longitudinal seams are double riveted and not bevelled. 70,000 where the longitudinal seams are single riveted and bevelled. 65,000 where the longitudinal seams are single riveted and not bevelled.

Furnaces with lapped joints and punched rivet holes.	75,000 where the longitudinal seams are double riveted and bevelled. 70,000 where the longitudinal seams are double riveted and not bevelled. 65,000 where the longitudinal seams are single riveted and bevelled. 60,000 where the longitudinal seams are single riveted and not bevelled.

In the case of upright fireboxes of donkey or similar boilers, 10 per cent. should be deducted from the constants given above, applicable to the respective classes of work.

Corrugated Furnaces.—The working pressure for corrugated furnaces, practically circular and machine made, provided the plain parts at the ends do not exceed 6 inches in length, and the plates are not less than $\frac{5}{16}$ in. thick, should be greater than found by the following formula:—

$$\frac{9000 \times \text{thickness in inches}}{\text{Mean diameter in inches}} = \text{working pressure per square inch.}$$

When the furnaces are riveted in two or more lengths the cases should be submitted for consideration, as it may be necessary to make a deduction.

Cylindrical Superheaters.—The strength of the joints of cylindrical superheaters and the factor of safety are found in a similar manner as for cylindrical boilers and steam receivers, but instead of using 47,000 lbs. as the tensile strength of iron, 30,000 lbs. is adopted, unless where the heat or flame impinges at, or nearly at right angles to the plate, then 22,400 lbs. is substituted. When a superheater is constructed with a tube subject to external pressure, the working pressure should be ascertained by the rules given for circular furnaces, but the constants should be reduced as 30 to 47. In all cases the internal steam-pipes should be so fitted that the steam in flowing to them will pass over all the plates exposed to the impact of heat or flame.

Superheaters should, as regards survey, be deemed to be the *most important parts* of the boilers, and must be inspected *inside* and *outside:* those that cannot be entered on account of their size or arrangement, must have a sufficient number of doors through which a thorough inspection of the whole of the interior can be made.

Special attention should be paid to the survey of superheaters, as with high pressure the plates may become dangerously weak, and not give any sound to indicate their state when tested with a hammer: the plates should therefore be occasionally drilled. Before commencing the survey, it is prudent to question the engineer officers as to the tendency of the boilers to flame: if flame is a frequent occurrence, extra care must be taken in the survey and in fixing the pressure to be allowed, as the tensile strength of the plate, when heated, is often reduced to about 4 tons per square inch. Drain-pipes must in all cases be fitted to superheaters in which a collection of water in the bottom is possible.

Safety-Valve for Superheaters.—Superheaters that can be shut off from the main boilers must be fitted with a parliamentary safety-valve of sufficient size, but the least size passed without special written authority should be 3 inches diameter.

Steel for Superheaters, &c.—If steel is proposed to be used in superheaters, the particulars should be submitted to the Board of Trade for consideration, but in all cases it should be discouraged for this purpose. This applies to the unshielded uptakes of all boilers, including ordinary vertical donkey-boilers.

The Flat Ends of all Boilers, as far as the steam-space extends, and the ends of superheaters, should be fitted with a shield, or baffle-plates, where exposed to the hot gases in the uptake: as all plates subjected to the direct impact of heat or flame are liable to get injured unless covered with water.

Steel Boilers.—The following should guide the Board's Surveyors when the general quality of the steel has been found suitable for marine boilers.

Tests.—The steelmakers or boilermakers should test one or more strips cut from *each* plate for tensile strength and elongation, and stamp both results on a part of the plate where they can be easily seen when the boiler is constructed. The Surveyor is not obliged to witness the foregoing tests, although it is very desirable that he should when his other duties will allow him to do so, but he should, however, select at least one in four of these plates, either at the steel works or the boilermakers' works, and witness the testing of at least one strip cut from each selected plate.

Rivet Tests.—A few rivets of each size should be selected, and should be turned and tested for tensile stress. The tensile stress should be from 28 tons to 32 tons per square inch, with a contraction of area of about 60 per cent.

Tensile and Bending Tests.—If for the plates from which the Surveyor selects the above proportions a greater stress is wished than is allowed for iron, tests for tensile stress and elongation should be made, and those for which no reduction of thickness is asked may be tested for resistance to bending, if preferred. In the latter case the tensile stress and elongation stamped on each plate should be reported by the Surveyor to the Board of Trade, along with the results of the bending tests.

Test Strips.—The breadth of the test strips for tensile stress should be about 2 inches, and the elongation, taken in a length of 10 inches, should be about 25 per cent., and not less than 20 per cent. The test strips must be carefully prepared and measured, and they should be cut from the plate by a planing or shaping machine.

Bending Test.—The bending tests for the plates *not* exposed to flame should be made with strips in their normal condition, but strips cut from furnaces, combustion-boxes, etc., should be heated to a cherry red, then plunged into water of about 80° and kept there until of the same temperature as the water, and then bent. The bending strips should not be less than 2 inches broad and 10 inches long, and they should be bent until they break, or until the sides are parallel at a distance from each other of not less than (3) three times the thickness of plate.

Tensile Stress, &c.—The tensile stress of the plates *not* exposed to flame should be about 28 tons, and should not exceed 32 tons, per square inch of section, and 28 tons should be the stress used in the calculations for cylindrical shells if the plates comply with all the conditions as stated herein, but when the minimum tensile stress of shell plates exceeds 28 tons and allowance is wished for the excess, the case should be especially submitted for the consideration of the Board. The tensile stress of furnace, flanging, and combustion box-plates should range from 26 tons to 30 tons per square inch.

Annealing.—All plates that are punched, flanged, or locally heated must be carefully annealed after being so treated.

Perforating and Annealing.—The rivet-holes in the furnaces and longitudinal seams of cylindrical seams of cylindrical shells should be *drilled*, but if it is wished to punch them and afterwards bore or anneal the plates in a proper furnace the particulars of the punching and boring or annealing should be submitted to the Board of Trade for consideration before being done, but all punched-holes should be made after bending.

In all cases where assent has been given for plates to be punched after bending and then annealed, the makers should stamp the plates with the words "punched after bending and then annealed," and in all cases where assent has been given for punching and afterwards boring plates the words "punched and then bored" should be stamped on the plates.

Stress on Stays.—Bars for stays should be tested. Solid steel screwed stays which have *not* been welded or otherwise worked after heating may be allowed a working stress of 9000 lbs. per square inch of net section, provided the tensile stress is from 27 to 32 tons per square inch, and the elongation in 10 inches about 25 per cent. and not less than 20 per cent. Steel stays which have been welded or worked in the fire have been found to be unreliable, therefore they should not be passed.

Constants for Flat Surfaces.—If the flanging plates and those exposed to flame comply with the foregoing conditions, the constants in the Board's rules for iron boilers may be increased as follows:—

The constants for flat surfaces when they are supported by stays screwed into the plate and riveted, 10 per cent. The constants for flat surfaces when they are supported by stays screwed into the plate and nutted, or when the stays are nutted in the steam space, 25 per cent. This is also applicable to the constants for flat surfaces stiffened by riveted washers or doubling strips, and supported by nutted stays. The constants for combustion box-girders, 10 per cent. The constants for plain furnaces, 10 per cent.

Furnaces, Corrugated, or Ribbed and Grooved.—The working-pressure of machine-made furnaces of the Fox corrugated and Morison suspension types, or of the Purves ribbed and grooved type, if they are practically true circles, and the plates not less than $\frac{5}{16}$ inch thick, may be found by the following formula:—

$$\frac{C \times T}{D} = \text{working pressure.}$$

$C = 14{,}000$ for Fox's corrugated and Brown's ribbed and grooved furnaces.
$C = 13{,}500$ for Morison's suspension furnace.
$T =$ thickness in inches.
$D =$ outside diameter in inches, measured at the bottom of the corrugations when the furnace is of the corrugated or suspension type, or over the plain parts when it is of the ribbed and grooved type.

(If the furnace is riveted in two or more lengths, the case should be submitted for consideration.)

Furnaces made up of Flanged Rings.—When horizontal furnaces of ordinary diameter are constructed of a series of rings welded longitudinally, and the ends of each ring flanged and the rings riveted together, and so forming the furnace, the working-pressure is found by the following formula, provided the length in inches between the centres of the flanges of the rings is not greater than (120 T − 12) and the furnaces are properly formed and constructed :—

$$\frac{9900 \times T}{3 \times D}\left(5 - \frac{l + 12}{60 \times T}\right) = \text{working pressure.}$$

T = thickness of plate in inches.
l = length between centre of flanges in inches.
D = outside diameter of furnace in inches.

Compressive Stress on Tube-Plates.—A greater compressive stress should not be allowed on tube plates than 10,000 lbs., which is that used in the following formula :—

$$\frac{(D - d)\ T \times 20{,}000}{W \times D} = \text{working pressure.}$$

D = least horizontal distance between centres of tubes in inches.
d = inside diameter of tubes in inches.
T = thickness of tube plate in inches.
W = extreme width of combustion-box in inches from front of tube plate to back of fire-box, or distance between combustion-box tube plates when the boiler is double-ended and the box common to the furnaces at both ends.

Plate and Rivet Section.—The rivet section, if of iron, in the longitudinal seams of cylindrical shells where lapped and at least double-riveted should not be less than $\frac{13}{8}$ times the net plate section, but if steel rivets are used their section should be at least $\frac{28}{23}$ of the net section of the plate if the tensile stress of the rivets is not less than 28 tons and not more than 32 tons per square inch. Therefore, in calculating the working pressure, the percentage strength of the rivets may be found in the usual way by the Board's rules; but in the case of iron rivets the percentages found should be divided by $\frac{13}{8}$. and in the case of steel rivets by $\frac{28}{23}$, the results being the percentages required. If the percentage strength of the rivets by calculation is less than the calculated percentage strength of the plate, calculate the working pressure by both percentages. When using the percentage strength of the plate use the nominal factor of safety suitable for the method of construction as by the Board's rules for iron boilers, but when using the percentage strength of the rivets use 5 as the factor of safety. The less of the two pressures so found is the working pressure to be allowed for the cylindrical portion of the shell.

Local heating of the Plates should be avoided, as many plates have failed from having been so treated.

Welding, &c.—Steel plates which have been welded should not be passed if subject to a tensile stress, and those welded and subject to a compressive stress should be efficiently annealed.

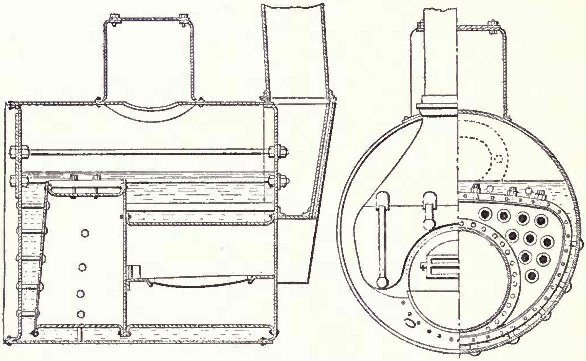
ILLUSTRATIONS OF THE BOARD OF TRADE RULES FOR BOILERS.

The Board of Trade Rules for Boilers may be illustrated by applying them to the various parts of a marine-boiler of similar construction to that shown in Figs. 56 and 57.

Girders for Flat Surfaces, or Fire-box Roof-stays.—Examples of the rule on page 146.

Example 1: What working pressure may be allowed for a steam-boiler of wrought-iron plates having a fire-box or combustion-box 34·2 inches

Fig. 56. Fig. 57.



Figs. 56 and 57.—Return-tube marine boiler.

wide; pitch of supporting bolts, $8\frac{1}{2}$ inches; distance between the girder-stays from centre to centre, 5 inches; length of girder, 3 feet; depth of girder, 7 inches; thickness of girder, 1 inch; number of supporting bolts, 3?

Then $\dfrac{750 \text{ constant} \times 7 \times 7 \text{ inches} \times 1 \text{ inch}}{(34{\cdot}2 - 8{\cdot}5 \text{ inches}) \times 5 \text{ inches} \times 3 \text{ feet}} = 95{\cdot}3$ lbs. per square inch,

the working pressure that may be allowed on the girder-stays.

Example 2: What pressure may be used for a steam-boiler made of iron-plates, having a combustion-box with girder-stays or roof-stays like Figs. 58 and 59; width of combustion-box, $45\frac{1}{4}$ inches; pitch of supporting bolts, 7 inches; distance between the girders from centre to centre, $7\frac{3}{4}$ inches; length of girder, 3 feet 6 inches, between the end supports; depth of girder, $7\frac{1}{2}$ inches; thickness of girder, $1\frac{1}{2}$ inch; number of supporting bolts, 5?

Then $\dfrac{850 \text{ constant} \times 7\frac{1}{2} \times 7\frac{1}{2} \times 1\frac{1}{2} \text{ inch}}{(45\frac{1}{4} \text{ inch} - 7 \text{ inch}) \times 7\frac{3}{4} \times 3\frac{1}{2} \text{ feet}} = 69{\cdot}12$ lbs. per square inch, the

working pressure that may be allowed on the girder-stays.

Flat Surfaces of Steam-boilers, or Sides of Fire-box.—Example of the *Rule* on page 147.

What working pressure may be used for a steam-boiler made of iron-plates, having a combustion-chamber like Fig. 60, with flat sides $\frac{1}{2}$ inch, or $\frac{8}{16}$ths of an inch thick, stayed with side-stays of $7\frac{1}{2}$ inches pitch?

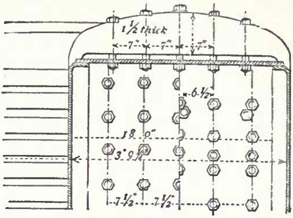


Fig. 58.—Fire-box roof-stays.

The pitch of the screwed side-stays being $7\frac{1}{2}$ inches, each stay will support a surface of $7\frac{1}{2}$ inches $\times$ $7\frac{1}{2}$ inches $= 56{\cdot}25$ square inches. As the plates are exposed to the impact of heat and flame on one side, and have water on the other, and the stays being screwed into the plate with their ends riveted over to form a substantial head, the constant will be 60.

Then $\dfrac{60 \times (8+1)^2}{56{\cdot}25 - 6} = 96{\cdot}7$ lbs. per square inch, the working pressure that may be allowed on the flat sides of the combustion box.

The Pitch of Screwed Stays for Flat Surfaces, *or side-stays of*

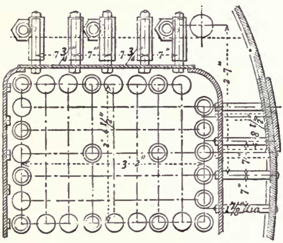


Fig. 59.—Combustion-chamber.

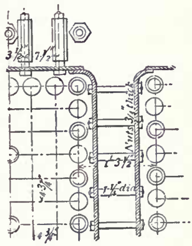


Fig. 60.—Combustion-chamber stays.

fire-box, may be found by the converse of the previous rule, using the constant suitable for each case as given at page 147.

$$\text{Pitch of stays in inches} = \sqrt[2]{\frac{\text{Constant} \times (T+1)^2}{\text{Working pressure in lbs.}} + 6}$$

where T=the thickness of the plate in sixteenths of an inch.

Example: Required the pitch of screwed-stays for the combustion-box described in the previous example?

Then $\dfrac{60 \times (8+1)^2}{96{\cdot}7 \text{ lbs. working pressure}} + 6 = 56{\cdot}25$ square inches, and $\sqrt[2]{56{\cdot}25}$ $= 7\frac{1}{2}$ inches, the pitch of stays required for that working pressure.

The Diameter of Screwed-stays for Flat Surfaces, the ends of the stays to be riveted over to form a substantial head, may be found by this *Rule:*

$$\frac{\text{Square inches of surface supported} \times \text{working pressure}}{\text{Stress per square inch of section of the stay}}.$$

The stress allowed per square inch of net section on the stays being that given at page 146.

Example: Required the diameter of a solid iron screwed stay to support the flat surface of the combustion-box described in the two previous examples?

The surface supported by each stay is $= 7{\cdot}5$ inches $\times$ $7{\cdot}5$ inches $= 56{\cdot}25$ square inches, and the maximum stress allowed on the stay is 7000 lbs. per square inch of net section.

Then $\dfrac{56{\cdot}25 \text{ square inches} \times 96{\cdot}7 \text{ lbs. working pressure}}{7000 \text{ lbs. stress per square inch}} = {\cdot}777$ and

$\sqrt[2]{\dfrac{{\cdot}777}{{\cdot}7854}} = 1$ inch diameter of the stay at the bottom of the thread.

The Working-Pressure for Screwed Stays, for flat surfaces, having the ends riveted over to form a substantial head, may be found by this *Rule:*

$$\frac{\text{Area of stay in square inches} \times \text{stress in lbs. per square inch allowed on the stay,}}{\text{Surface in square inches supported by each stay.}}$$

Example: A marine-boiler having the flat surface of the combustion-box supported by solid iron screwed stays of 8 inches pitch, has been working at 60 lbs. pressure per square inch. On examining the boiler the plates were found to be sound and suitable for that pressure, but the area of each stay was reduced by corrosion to ·4115 square inch. Required the proper working pressure of the boiler.

Then $\dfrac{{\cdot}4115 \text{ square inch} \times 7000 \text{ lbs. maximum pressure allowed}}{8 \text{ inch pitch of stays} \times 8 \text{ inch pitch of stays}} = 45$ lbs.

per square inch, the required working pressure, showing that the boiler has been worked at $60 - 45 = 15$ lbs. too high a pressure.

The Strain per Square Inch of Section of a Stay for a flat surface, may be found by this *Rule:*

$$\frac{\text{Surface supported by each stay in square inches} \times \text{pressure of the steam}}{\text{Area of each stay in square inches}}.$$

Example: The pitch of the stays of a combustion-box is 8 inches; the pressure of the steam is 45 lbs. per square inch; the area of each stay is ·4115 square inch. Required the strain on the stay per square inch of section.

Then the surface supported by each stay = the square of the pitch and
$\dfrac{8 \text{ inches} \times 8 \text{ inches pitch} \times 45 \text{ lbs. pressure}}{{\cdot}4115 \text{ square inches area of each stay}} = 7000$ lbs., the strain per square inch of section of each stay.

Compressive-Stress on Iron Tube-plates.—*Example* of the Rule on page 148. In a marine-boiler the least horizontal distance between the centres of the tubes is 5 inches; the inside diameter of the tubes is $3\frac{1}{2}$ inches; the thickness of the tube-plate is ·9 inch; the extreme width of the combustion-box from the front of the tube-plate to the back of the fire-box is 34·2 inches. What compressive stress may be allowed on the tube-plates as working pressure?

$$\text{Then } \frac{(5 \text{ inches} - 3{\cdot}5 \text{ inches}) \times {\cdot}9 \times 16{,}000}{34{\cdot}2 \text{ inches} \times 5 \text{ inches}} = 126{\cdot}31 \text{ lbs. per square inch,}$$

the working pressure which may be allowed on these tube-plates.

The Diameter of the swelled portion of a Round Stay, containing a cotter-hole, may be found by the following *Rule*:—

$$\text{Diameter of swell} = \sqrt[2]{\frac{\text{Area of stay} + \text{area of cotter-hole}}{{\cdot}7854}}$$

Example: The diameter of a stay is $1\frac{3}{4}$ inches, what size should the stay be swelled for a cotter hole $1\frac{3}{4}$ inches long by $\frac{3}{8}$ inch thick, in order to be of equal strength to the solid portion of the stay.

$$\text{Then } \sqrt{\frac{1{\cdot}75 \times 1{\cdot}75 \times {\cdot}7854 + 1{\cdot}75 \times {\cdot}375}{{\cdot}7854}} = 1{\cdot}98, \text{ or say 2 inches, the}$$

diameter of the stay after being punched.

Cylindrical Boilers with Iron-plates.—*Examples* of the Rules on page 150.

Required the strength of the joints, and the working pressure of a cylindrical boiler of 60 inches internal diameter, with iron-plates $\frac{3}{8}$ inch thick. Diameter of rivets $\frac{3}{4}$ inch, the seams having double-riveted lap-joints; pitch of rivets 3 inches.

$$\text{Then } \frac{3 \text{ inch pitch} - {\cdot}75 \text{ inch diameter of rivet} \times 100}{3 \text{ inches pitch of rivets}} = 75. \text{ The percentage}$$

of strength of plate at the joint as compared with the solid plate.

$$\text{And } \frac{{\cdot}75 \times {\cdot}75 \text{inch} \times {\cdot}7854 \times 2 \text{ rows of rivets} \times 100}{3 \text{ inches pitch} \times {\cdot}375 \text{ thickness of plate}} = 78{\cdot}5 \text{ the percentage of}$$

the strength of the rivets as compared with the solid plate.

Then taking iron as equal to 47,000 lbs. tensile strength per square inch, and using the smaller of the above two percentages as the strength of the joint and adopting a factor of safety of 5·1, it gives

$$\frac{47000 \text{ lbs.} \times {\cdot}75 \text{ strength of joint} \times ({\cdot}375 \text{ plate} \times 2)}{60 \text{ inches internal diameter} \times 5{\cdot}1 \text{ factor of safety}} = 86 \text{ lbs. pressure per}$$

square inch, which may be allowed on the safety-valves.

Circular Furnaces of Iron-plates with the longitudinal joints welded, or made with a butt-strap. *Examples* of the Rules on page 151.

Required the working pressure of an iron steam-boiler having a circular furnace-tube 30 inches diameter, and 6 feet long, plates $\frac{3}{8}$ inch thick.

Then $\frac{90000 \times \cdot375 \times \cdot375 \text{ inch}}{(6 \text{ feet} + 1) \times 30 \text{ inches diameter}}$ = 60 lbs. working pressure per square inch. The working pressure in no case to exceed that given by the following formula :—

$$\frac{9000 \times \cdot375 \text{ thickness of plate}}{30 \text{ inches diameter}} = 112 \text{ lbs. per square inch.}$$

Corrugated-Furnace of Iron-plates.—*Example* of Rule on page 152. Required the working pressure of a corrugated furnace-tube of 40 inches mean diameter, and machine made, of iron plates $\frac{1}{2}$ inch thick.

Then, $\frac{9000 \times \cdot5 \text{ inch thickness}}{40 \text{ inches mean diameter}}$ = 112 lbs. per square inch working pressure.

Corrugated-Furnaces of Steel-plates.—*Example* of the Rule on page 154. Required the working pressure of a Fox's corrugated furnace-tube, 38 inches diameter and machine made, of steel plates $\frac{1}{2}$ inch thick.

Then, $\frac{14000 \times \cdot5 \text{ inch thickness}}{38 \text{ inches outside diameter}}$ = 184 lbs. per square inch working pressure.

Compressive-Stress on Steel Tube-plates.—*Example* of the Rule on page 155. In a steel marine-boiler, the least horizontal distance between the centres of the tubes is 5 inches, the inside diameter of the tubes is $3\frac{1}{2}$ inches, the thickness of the tube-plate is ·85 inch, the extreme width of the combustion-box from the front of the tube-plate to the back of the fire-box is 38 inches. What compressive stress may be allowed on the tube-plates, as working pressure?

Then, $\frac{5 \text{ inches} - 3\cdot5 \text{ inches} \times \cdot85 \times 20000}{38 \text{ inches wide} \times 5 \text{ inches}}$ = 134 lbs. per square inch, the working pressure which may be allowed on these tube-plates.

LLOYD'S RULES FOR STEAM BOILERS.

Cylindrical Shells of Steel Boilers.—The strength of cylindrical shells of steel boilers is to be calculated from the following formula:—

$$\frac{C \times (T - 2) \times B}{D} = \text{working-pressure in lbs. per square inch,}$$

where D = mean diameter of shell in inches,
T = thickness of plate in sixteenths of an inch,
C = 20 when the longitudinal seams are fitted with double butt straps of equal width,
C = 19·25 when they are fitted with double butt straps of unequal width, only covering on one side the reduced section of plate at the outer lines of rivets,
C = 18·5 when the longitudinal seams are lap joints,
B = the least percentage of strength of longitudinal joint found as follows:—

For plate at joint $B = \dfrac{p - d}{p} \times 100.$

For rivets at joint $B = \dfrac{n \times a}{p \times t} \times 85$ wnere steel rivets are used.

$B = \dfrac{n \times a}{p \times t} \times 70$ where iron rivets are used.

where p = pitch of rivets in inches,
t = thickness of plate in inches,
d = diameter of rivet holes in inches,
n = number of rivets used per pitch in the longitudinal joint,
a = sectional area of rivet in square inches.

In case of rivets in double shear $1·75a$ is to be used instead of a.

Proper deductions are to be made for openings in shell.

All manholes in circular shells to be stiffened with compensating rings.

The shell plates under domes in boilers so fitted, to be stayed from the top of the dome or otherwise stiffened.

Note.—The inside butt strap to be at least $\frac{3}{4}$ the thickness of the plate.

Note.—For the shell plates of superheaters or steam-chests enclosed in the uptakes or exposed to the direct action of the flame, the co-efficients should be $\frac{2}{3}$ of those given in the above tables.

Stays.—The strength of stays supporting flat surfaces is to be calculated from the smallest part of the stay or fastening, and the strain upon them is not to exceed the following limits, namely:—

Iron Stays.—For stays not exceeding $1\frac{1}{2}$ inches smallest diameter, and for all stays which are welded, 6,000 lbs. per square inch; for unwelded stays above $1\frac{1}{2}$ inches smallest diameter, 7,500 lbs. per square inch.

Steel Stays.—For stays not exceeding $1\frac{1}{2}$ inches smallest diameter, 8,000 lbs. per square inch; for stays above $1\frac{1}{2}$ inches smallest diameter, 9,000 lbs. per square inch. No steel stays are to be welded.

Stay Tubes.—The stress is not to exceed 7,500 lbs. per square inch.

Flat Plates.—The strength of flat plates supported by stays to be taken from the following formula:—

$$\frac{C \times T^2}{P^2} = \text{working pressure in lbs. per square inch.}$$

where T = thickness of plate in sixteenths of an inch,
- P = greatest pitch in inches,
- C = 90 for iron or steel plates $\frac{7}{16}$ thick and under, fitted with screw stays with riveted heads,
- C = 100 for iron or steel plates above $\frac{7}{16}$ thick, fitted with screw stays with riveted heads,
- C = 110 for iron or steel plates $\frac{7}{16}$ thick and under, fitted with stays and nuts,
- C = 120 for iron plates above $\frac{7}{16}$ thick, and for steel plates above $\frac{7}{16}$ and under $\frac{9}{16}$ thick, fitted with screw stays and nuts,
- C = 135 for steel plates $\frac{9}{16}$ thick and above, fitted with screw stays and nuts,
- C = 140 for iron plates fitted with stays with double nuts,
- C = 150 for iron plates fitted with stays with double nuts and washers outside the plates, of at least $\frac{1}{3}$ of the pitch in diameter and $\frac{1}{2}$ the thickness of the plates,
- C = 160 for iron plates fitted with stays with double nuts and washers riveted to the outside of the plates, of at least $\frac{2}{5}$ of the pitch in diameter and $\frac{1}{2}$ the thickness of the plates,
- C = 175 for iron plates fitted with stays with double nuts and washers riveted to the outside of the plates, when the washers are at least $\frac{2}{3}$ of the pitch in diameter and of the same thickness as the plates.

For iron plates fitted with stays with double nuts and doubling strips riveted to the outside of the plates, of the same thickness as the plates, and of a width equal to $\frac{2}{3}$ the distance between the rows of stays, C may be taken as 175, if P is taken to be the distance between the rows, and 190 when P is taken to be the pitch between the stays in the rows.

For steel plates, other than those for combustion-chambers, the values of C may be increased as follows:—

C = 140	increased to	175,
150	„	185,
160	„	200,
175	„	220,
190	„	240.

If flat plates are strengthened with doubling plates securely riveted to them, having a thickness of not less than $\frac{2}{3}$ of that of the plates, the strength to be taken from

$$\frac{C \times (T + \frac{t}{2})^2}{P^2} = \text{working-pressure in lbs. per square inch;}$$

where t = thickness of doubling plates in sixteenths, and C, T and P are as above.

Note.—In the case of front plates of boilers in the steam-space, these numbers should be reduced 20 per cent., unless the plates are guarded from the direct action of the heat.

For steel tube-plates in the nest of tubes the strength to be taken from

$$\frac{140 \times T^2}{P^2} = \text{working pressure in lbs. per square inch;}$$

where T = the thickness of the plates in sixteenths of an inch,
P = the *mean* pitch of stay-tubes from centre to centre.

For the wide water-spaces between the nests of tubes the strength to be taken from

$$\frac{C \times T^2}{P^2} = \text{working-pressure in lbs. per square inch;}$$

where P = the horizontal distance from centre to centre of the bounding rows of tubes, and
C = 120 where the stay-tubes are pitched with two plain tubes between them and are not fitted with nuts outside the plates,
C = 130 if they are fitted with nuts outside the plates,
C = 140 if each alternate tube is a stay-tube not fitted with nuts,
C = 150 if they are fitted with nuts outside the plates,
C = 160 if every tube in these rows is a stay-tube and not fitted with nuts,
C = 170 if every tube in these rows is a stay-tube and each alternate stay-tube is fitted with nuts outside the plates.

The thickness of tube plates of combustion-chambers in cases where the pressure on the top of the chambers is borne by these plates is not to be less than that given by the following rule:—

$$T = \frac{P \times W \times D}{1600 \times (D - d)}$$

where P = working pressure in lbs. per square inch,
W = width of combustion-chamber over plates in inches,
D = horizontal pitch of tubes in inches,
d = inside diameter of plain tubes in inches,
T = thickness of tube-plates in sixteenths of an inch.

Girders.—The strength of girders supporting the tops of combustion chambers and other flat surfaces to be taken from the following formula:—

$$\frac{C \times d^2 \times T}{(L - P) \times D \times L} = \text{working-pressure in lbs. per square inch;}$$

where L = width between tube-plates, or tube-plate and back plate of chamber,
P = pitch of stays in girders,
D = distance from centre to centre of girders,
d = depth of girder at centre,
T = thickness of girder at centre. All these dimensions to be taken in inches.

Constants for Wrought Iron.

$$C = \begin{cases} 6{,}000, \text{ if there is one stay to each girder.} \\ 9{,}000, \text{ if there are two or three stays to each girder.} \\ 10{,}000, \text{ if there are four or five stays to each girder.} \\ 10{,}500, \text{ if there are six or seven stays to each girder.} \\ 10{,}800, \text{ if there are eight stays or above to each girder.} \end{cases}$$

Constants for Wrought Steel.

$$C = \begin{cases} 6{,}600, \text{ if there is one stay to each girder.} \\ 9{,}900, \text{ if there are two or three stays to each girder.} \\ 11{,}000, \text{ if there are four or five stays to each girder.} \\ 11{,}550, \text{ if there are six or seven stays to each girder.} \\ 11{,}880, \text{ if there are eight stays or above to each girder.} \end{cases}$$

Circular Furnaces.—The strength of plain furnaces to resist collapsing to be calculated from the following formula:—

$$\frac{89{,}600 \times T^2}{L \times D} = \text{working pressure in lbs. per square inch;}$$

where T = thickness of plates in inches,
D = outside diameter of furnace in inches,
L = length of furnace in feet. If strengthening rings are fitted, the length between the rings is to be taken.

If the plates do not exceed $\frac{9}{16}$ in. in thickness, the pressure, however, is not to exceed

$$\frac{8{,}000 \times T}{D} = \text{lbs. per square inch.}$$

If the plates are of steel and exceed $\frac{9}{16}$ in. in thickness, the pressure is not to exceed

$$\frac{8{,}800 \times T}{D} = \text{lbs. per square inch.}$$

If the furnaces are fitted with a single Adamson-ring at about the middle of their length, the pressure may be calculated from

$$\frac{10{,}400 \times T}{D} = \text{working-pressure in lbs. per square inch.}$$

If the furnaces are fitted with two Adamson-rings, then the pressure may be calculated from

$$\frac{11{,}400 \times T}{D} = \text{working-pressure in lbs. per square inch.}$$

If the furnaces are fitted with a series of Adamson-rings at intervals not exceeding 23 inches, the pressure may be calculated from

$$\frac{1{,}000 \times (T - 2)}{D} = \text{working-pressure in lbs. per square inch;}$$

where T = thickness in sixteenths of an inch,
D = outside diameter of furnaces.

The strength of corrugated furnaces made of steel, having a less tensile strength than 26 tons per square inch, the corrugations being 6 inches apart and $1\frac{1}{2}$ inches deep, to be calculated from

$$\frac{1{,}000 \times (T - 2)}{D} = \text{working-pressure in lbs. per square inch.}$$

The strength of furnaces made of steel, having a tensile strength between 26 and 30 tons per square inch, and corrugated on Fox's or Morison's plans, to be calculated from

$$\frac{1{,}259 \times (T - 2)}{D} = \text{working-pressure in lbs. per square inch.}$$

The strength of ribbed furnaces (with ribs 9 inches apart) to be calculated from the following formula:—

$$\frac{1{,}160 \times (T - 2)}{D} = \text{working-pressure in lbs. per square inch.}$$

The strength of spirally corrugated furnaces to be calculated from the following formula:—

$$\frac{912 \times (T - 2)}{D} = \text{working-pressure in lbs. per square inch;}$$

where T = thickness of plate in sixteenths of an inch,
and D = outside diameter of corrugated furnaces, or outside diameter of the plain parts of ribbed furnaces, in inches.

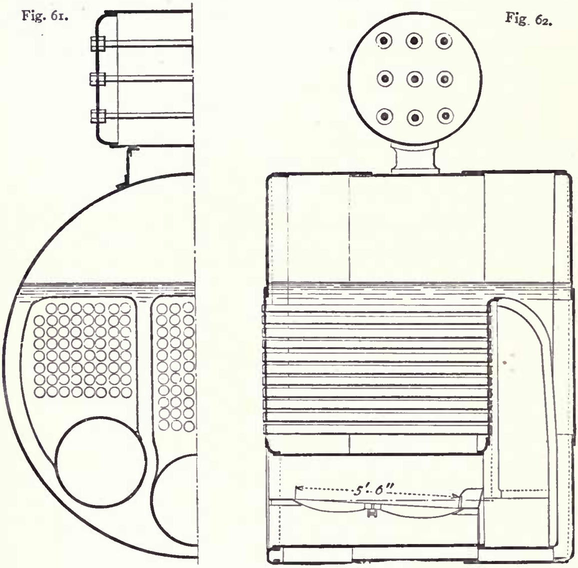
The strength of Holmes' Patent Furnaces, in which the corrugations are not more than 16 inches apart from centre to centre, and not less than 2 inches high, to be calculated from the following formula:—

$$\text{Working-pressure in lbs. per square inch} = \frac{945 \times (T - 2)}{D}$$

where T = thickness of plain portions of furnace in sixteenths of an inch,
D = outside diameter of plain parts of the furnace in inches.

ILLUSTRATIONS OF LLOYD'S RULES FOR BOILERS.

The Method of Determining the Working-Pressure on each part of a Steam-Boiler, according to Lloyd's Rules, may be illustrated by applying them to the marine-boiler shown in Figs. 61 and 62, and calculating the working pressures of the various parts. Figs. 61 and 62



Figs. 61 and 62.—Return tube marine boiler.

represent a steel-boiler 12 feet $3\frac{1}{2}$ inches diameter to work at a pressure of 90 lbs. per square inch.

Diameter of rivets $\frac{15}{16}$ inch = ·9375 inch.
Area of one rivet = ·6903 square inch.
Mean diameter of boiler-shell = 147·5 inch.
Thickness of steel-shell plates = ·75 inch.
End-plates are fitted with stays with double nuts, therefore the constant is 140.
Pitch of stays for end-plates in inches, 14·5 inches, and $14{\cdot}5^2 = 210{\cdot}25$.
Area of stay, 2·95 square inches.
Pitch of stays in combustion-chamber sides = 9 inches, and $9^2 = 81$.
Pitch of stays in combustion-chamber backs = $7\frac{5}{8}$ inches, and $7\frac{5}{8}^2 = 58{\cdot}14$.
Thickness of furnace-plates = ·53125 inch.
Length of furnace = 7·25 feet.
Outside diameter of furnace = 37 inches.

Thickness of Plating.

Circumferential shell	steel	$\frac{3}{4}$	inch thick.
Front and back upper plates	,,	$\frac{3}{4}$	,,
Back mid	,,	$\frac{11}{16}$	,,
Back lower	,,	$\frac{11}{16}$	,,
Front tube-plates	,,	$\frac{3}{4}$	,,
Front lower	,,	$\frac{11}{16}$	,,
Furnace crowns	,,	$\frac{17}{32}$	,,
Furnace bottoms	,,	$\frac{9}{16}$	,,
Inner tube-plates	,,	$\frac{11}{16}$	,,
Combustion-chamber, sides	,,	$\frac{1}{2}$	,,
Combustion-chamber, backs	,,	$\frac{7}{16}$	,,
Steam-chest shell	,,	$\frac{7}{16}$	,,
Steam-chest ends	,,	$\frac{5}{8}$	,,
Steam-chest connection	iron	$\frac{3}{4}$	,,

The Working-Pressure of each part of the Boiler may be calculated with the above data as follows :—

Working Pressure by Lloyd's Rules.

Plate-section	$\frac{3{\cdot}625 - {\cdot}9375}{3{\cdot}625} \times 100 =$	$74{\cdot}13\%$
Rivet-section	$\frac{2 \times {\cdot}6903 \times 85}{3{\cdot}625 \times {\cdot}75} \times 1{\cdot}75 =$	$75{\cdot}4\%$
Circumferential shell	$\frac{20 \times (12-2) \times 74{\cdot}13}{147{\cdot}5} =$	$100{\cdot}5$ lbs.
End-plates in steam-space	$\frac{140 \times (12)^2}{210{\cdot}25} =$	$95{\cdot}8$ lbs.
Stays in steam-space	$\frac{2{\cdot}95 \times 7500}{210{\cdot}25} =$	105 lbs.
Outer tube-plates	$\frac{140 \times (12)^2}{203} =$	$99{\cdot}3$ lbs.
Stay-tubes	$\frac{(8{\cdot}8 - 6{\cdot}5) \times 7500}{(14{\cdot}5 \times 12) - 48} =$	136 lbs.
Furnaces	$\frac{89600 \times {\cdot}28226}{7{\cdot}25 \times 37} =$	$94{\cdot}2$ lbs.
Combustion-chamber, sides	$\frac{120 \times 64}{81} =$	$94{\cdot}8$ lbs.
Combustion-chamber, backs	$\frac{110 \times 4{\cdot}9}{58{\cdot}14} =$	$92{\cdot}7$ lbs.

WORKING PRESSURE BY LLOYD'S RULES—*continued.*

Combustion-chamber, side-stays .	$\frac{1{\cdot}22 \times 6000}{78{\cdot}75}$	$=$ 92·9 lbs.
Combustion-chamber, back-stays.	$\frac{99 \times 6000}{58{\cdot}14}$	$=$ 102 lbs.
Steam-chest, plate-section . .	$\frac{2{\cdot}75 - {\cdot}8125}{2{\cdot}75}$	$=$ 70%
Steam-chest, rivet-section . .	$\frac{2 \times {\cdot}5184}{2{\cdot}75 \times {\cdot}4375} \times 85$	$=$ 73%
Steam-chest, shell	$\frac{215 \times {\cdot}4375 \times 70}{47}$	$=$ 140 lbs.

FURNACE-TUBES, PLATES AND TUBES OF BOILERS.

Corrugated Furnace-tubes, shown in Fig. 63, are much stronger to resist collapsing pressure than plain furnace-tubes, but inferior to them as regards staying the boiler-ends, and it is necessary to place longitudinal stays close to corrugated furnace-tubes. Lloyd's Rule for these tubes, given at page 167, may be illustrated by the following example. Required the working pressure of a corrugated furnace-tube with corrugations $1\frac{1}{2}$ inches deep, the thickness of the plates being $\frac{1}{2}$ inch and the greatest diameter of the furnace-tube being 36 inches.

Then $\frac{1000 \times (8-2)}{36 \text{ inches diameter}} =$ 167 lbs. per square inch, the working pressure of this furnace-tube.

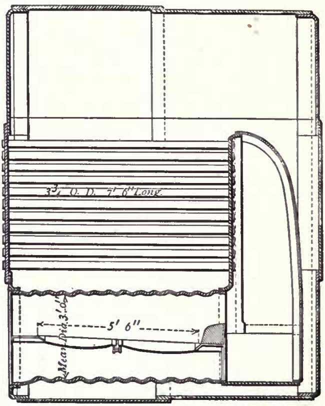


Fig. 63.—Return-tube marine boiler, with corrugated furnace-tubes.

Strength of Corrugated Furnace-tubes.—A test was made of a corrugated furnace-tube, of mild-steel, 6 feet 9 inches long, 3 feet $1\frac{7}{8}$ inches outside diameter, had 13 corrugations $1\frac{1}{2}$ inches deep, and pitched 6 inches apart from centre to centre of each corrugation. The tenacity of the steel was 27·2 tons per square inch, with an elongation of 35 per cent. in 10 inches. The results were as follows :—

Table 69.—PARTICULARS OF TEST OF A CORRUGATED FURNACE-TUBE.

Pressure in Pounds per Square inch.	Alteration of Form Transversely.						Permanent Set.			Elongation in a Length of 5 ft. 9 in.
	Horizontally.			Vertically.						
300	...	...	...	...	...	...	...	...	...	·035
400	...	...	...	...	...	...	...	...	...	·05
500	...	...	...	...	...	...	...	...	...	·07
600	·03	·015	·03	·03	·025	·03	...	...	...	·07
700	·03	·015	·03	·03	·025	·03	...	...	...	·07
800	·04	·03	·05	nil	·025	·03	...	...	...	·09
850	·10	·05	·08	·04	·025	·03	·035	·04	·035	·11
900	Total collapse ensued.									

Ribbed Furnace-Tubes shown in Fig. 64 combine the longitudinal strength of plain furnace-tubes, and the diametrical resistance to collapse of corrugated furnace-tubes. The ribs or strengthening rings are rolled on the plate, and there is no necessity, as in the case of corrugated furnace-tubes, to weld the joint for the subsequent process of corrugating. An experiment consisting of testing, with hydraulic pressure, of a furnace-tube

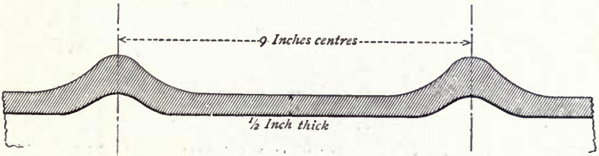


Fig. 64.—Ribbed furnace-tube.

to destruction was made with a ribbed-tube of the dimensions given in the following Table :—

Table 70.—DIMENSIONS OF A RIBBED FURNACE-TUBE TESTED TO DESTRUCTION AT THE WORKS OF MESSRS. JOHN BROWN AND CO., LIMITED, SHEFFIELD.

Thickness of Furnace Tube.	Length of Furnace Tube over all.	Length between Centres of Inner Rows of Rivets in End Angles.	Internal Diameter.					
			Front End.		Middle.		Back End.	
			Vertical.	Horizontal.	Vertical.	Horizontal.	Vertical.	Horizontal.
Inch.	ft. in.	ft. in.	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.
$\frac{1}{2}$	7 0	6 $6\frac{7}{8}$	$37\frac{25}{32}$	$37\frac{28}{32}$	$37\frac{24}{32}$	$37\frac{28}{32}$	$37\frac{25}{32}$	$37\frac{26}{32}$

Result of the Test.—The ribbed-tube collapsed opposite to the butt-strap, at a pressure of 780 lbs. per square inch. There was no appreciable deformation diametrically at 700 lbs. per square inch, the highest pressure at which the tube was gauged. It is claimed for this form of furnace-tube that besides being practically of the same strength as a corrugated furnace-tube to resist collapse, it possesses very much greater longitudinal strength, and that therefore it will not require longitudinal stays close

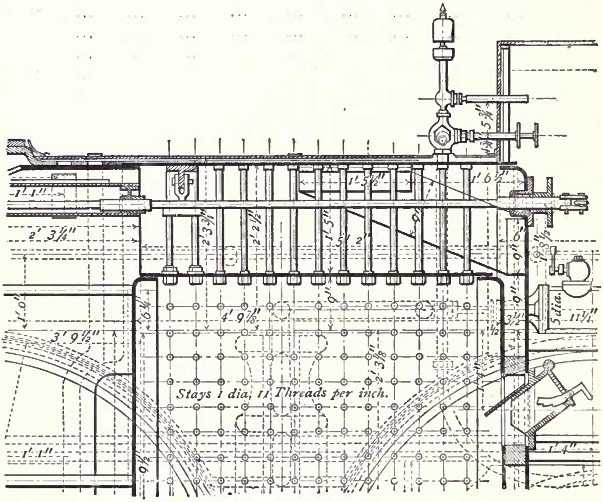


Fig. 65.—Fire-box of a locomotive-boiler.

alongside it. It also will not be liable to an accumulation of scale upon its crown to a greater extent than plain furnaces—a point of very considerable importance in the cases of vessels making very long ocean voyages. The longitudinal joint of these furnaces are best made with a butt-strap, which is, of course, placed below the fire-bar level, a truly circular form being thus obtained with a thoroughly trustworthy joint, which is not likely to give trouble by leakage as soon as a slight corrosion has occurred upon some part of the joint, as has often happened with welded joints. Further, it is considered that a welded joint is more likely to suffer from corrosion than a riveted one, on account of the material not being so homogeneous at the weld as at the other portions.

The Strength of Ribbed Furnace-Tubes is the same as corrugated furnace-tubes, and this form of furnace has been approved by Lloyd's for a working pressure as found by their formula for determining the working pressure to be allowed upon corrugated furnace-tubes.

Stay-Bolts for Fire-Box Roofs of Locomotive Engines.—The strongest and most efficient arrangement of stays for staying the roof of a locomotive fire-box is that shown in Fig. 65. The stays do not impede the circulation of the water, which can freely circulate over the hottest parts. For very high temperatures, the stay-bolt should be screwed into both plates, and have their ends riveted over to form good heads.

Roof-Stays of the Fire-Box of Locomotive Engines are frequently

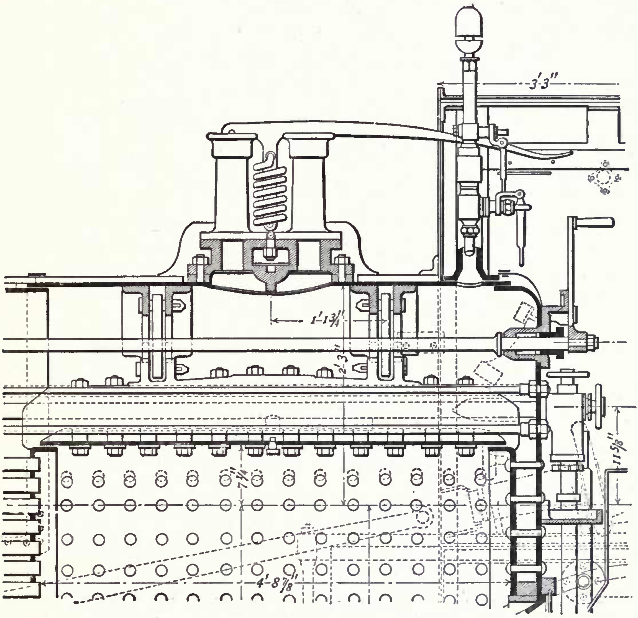


Fig. 66.—Fire-box of a locomotive-boiler.

made in the form of the girder-stay, shown in Fig. 66, in which there are 8 wrought-iron girder-stays, 2 inches thick, mean depth $6\frac{1}{2}$ inches, length of girder 53 inches between the points of support. Each girder is attached to the fire-box by best Yorkshire iron-bolts 1 inch diameter, pitched at $4\frac{3}{8}$ inches. Four of the bars are connected by means of sling-stays to angle-irons riveted to the crown of the fire-box shell. The stays are pitched $4\frac{7}{8}$ inches transversely, leaving a space of $2\frac{7}{8}$ inches between the stays.

The ultimate breaking-weight of each girder-stay may be found by

Mr. D. K. Clark's formula for the transverse strength of rectangular bars of wrought-iron loaded at the middle, which is:—

$$\text{Breaking weight in tons} = \frac{26 \times \text{breadth in inches} \times \text{depth in inches}^2}{\text{Length of span in inches.}}$$

$$\text{Then } \frac{26 \times 2 \text{ inches broad} \times 6\tfrac{1}{2} \times 6\tfrac{1}{2} \text{ inches depth}}{53 \text{ inches, length of span of girder}} = 41{\cdot}45 \text{ tons,}$$

the ultimate breaking-weight required to break each of these girder-stays when loaded at the middle; but as the weight is uniformly distributed, the ultimate breaking-weight is $41{\cdot}45 \times 2 = 82{\cdot}90$ tons.

The area of fire-box roof supported by each roof-stay is $4{\cdot}875$ inches pitch of girders $\times$ 53 inches span $= 258{\cdot}375$ square inches, and when tested by hydraulic pressure to the usual pressure of 200 lbs. per square inch, each girder-stay would be loaded to $\frac{258{\cdot}375 \times 200 \text{ lbs.}}{2240 \text{ lbs.}} = 23$ tons.

Cast-Steel Girder-Stays for Fire-Box Roofs, as shown in Figs. 67 and 68, are more efficient than wrought-iron girder-stays, as they can be made of the maximum strength and rigidity with the minimum bulk and weight.

The Thickness of Wrought-Iron Girder-Stays or Fire-Box Roof-Stays for supporting the tops of combustion-chambers and other flat-surfaces may be found by the following formula:—

$$\frac{\text{Working pressure in lbs. per square inch} \times (\text{W}-\text{P}) \text{ D} \times \text{L}}{\text{Constant} \times d^2} =$$

Thickness of the girder-stay in inches by Board of Trade Rule.

$$\frac{\text{Working pressure in lbs. per square inch} \times (\text{L}-\text{P}) \times \text{D} \times \text{L}}{\text{Constant} \times d^2} =$$

Thickness of the girder-stay in inches at the centre by Lloyd's Rule.

The constants to be taken from pages 146 and 164.

A wrought-iron roof-stay for the combustion-chamber of a marine-boiler is shown in Figs. 69, 70, 71.

The Strength of a Combustion-Chamber with a Curved Roof, as shown in Fig. 72, may be calculated, when the radius of the curvature is known, by the rules for determining the collapsing strength of furnace-tubes, the radius of the largest curvature of the roof to be taken.

The Thickness of Plate required for a Cylindrical Boiler-Shell for a given working pressure, may be found by the following *Rules*, the notation being the same as given previously for the Board of Trade and Lloyd's Rules, by which the percentage of strength of the joint is to be calculated:—

$$\left.\begin{array}{l}\text{Thickness of plate in}\\ \text{inches by Board of}\\ \text{Trade Rule.}\end{array}\right\} = \frac{\text{Working pressure in lbs.} \times \text{the inside diameter of the shell in inches} \times \text{factor of safety}}{47000 \times \text{percentage of strength of joint} \times 2}$$

$$\left.\begin{array}{l}\text{Thickness of plate in inches}\\ \text{by Lloyd's Rule.}\end{array}\right\} = \frac{\text{Working pressure in lbs. per square inch} \times \text{D}}{\text{C} \times \text{B}}$$

$$\text{or} = \frac{\text{Working pressure in lbs. per sq. inch} \times \text{mean diameter of shell in inches}}{\text{Co-efficient from table} \times \text{percentage of strength of joint.}}$$

The Thickness of Plate required for Flat-surfaces of Boilers may be found by the following formulæ :—

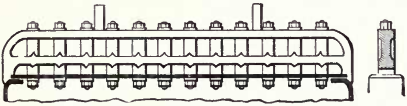
$$\frac{\text{Working pressure in lbs. per square inch} \times S - 6}{\text{Constant}} = (T+1)^2 \text{ and } \sqrt[2]{(T+1)}$$

=thickness of plate by Board of Trade Rule.

$$\frac{\text{Working pressure in lbs. per square inch} \times P^2}{\text{Constant}} = T^2 \text{ and } \sqrt[2]{T^2} = \text{thick-}$$

ness of plate by Lloyd's Rule.

The constants to be taken from pages 147 and 163.



Figs. 67 and 68.—Cast-steel fire-box roof-stays.

Fig. 69. Fig. 70.

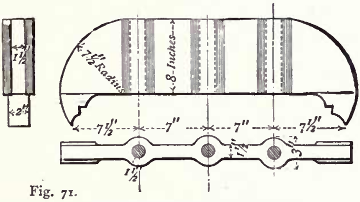


Fig. 71.

Figs. 69—71.—Wrought-iron roof-stay.

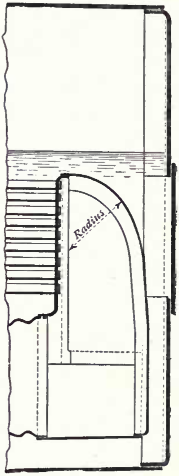


Fig. 72.—Combustion-chamber with curved roof.

The Thickness of Tube-plate may be found by the following formula :—

$$\frac{\text{Working pressure in lbs. per square inch} \times W \times D}{15000 \times (D-d)} = \text{the thickness of}$$

tube-plate in inches by the Board of Trade Rule.

The Thickness of Circular Plain Furnace-tubes, with the longitudinal joints welded or made with a butt-strap, may be found by the following formulæ :—

$$\frac{\text{Working pressure in lbs. per sq. in.} \times (\text{length in ft.} + 1) \times \text{diameter in inches}}{90000}$$

$= T^2$ and $\sqrt[2]{T^2}$ = the thickness of plate by Board of Trade Rule.

$$\frac{\text{Working pressure in lbs. per square inch} \times L \times D}{89600} = T^2 \text{ and } \sqrt[2]{T^2} = \text{the}$$

thickness of the plate in inches by Lloyd's Rule.

The Thickness of Corrugated Furnace-tubes, practically circular and machine-made, provided that the plain parts at the ends do not exceed 6 inches in length, and the plates are not less than $\frac{5}{16}$ inch thick, may be found by the following formulæ:—

$$\frac{\text{Working pressure in lbs. per square inch} \times \text{mean diameter in inches}}{9000} =$$
the thickness of plate in inches by Board of Trade Rule.

$$\frac{\text{Working pressure in lbs. per sq. in.} \times \text{greatest diameter of furnace in inches}}{1000}$$
$=(T-2)$ and $(T-2)+2=$the thickness of the plate in sixteenths of an inch for furnaces with corrugations $1\frac{1}{2}$ inches deep, by Lloyd's Rule.

The Thickness of Corrugated Steel Furnace-Tubes, when new and machine-made, and practically true circles, when the plain parts at the ends do not exceed 6 inches in length, and the plates are not less than $\frac{5}{16}$ inch thick.

$$\frac{\text{Working pressure in lbs. per square inch} \times \text{mean diameter in inches}}{12500} = \text{the}$$
thickness in inches by the Board of Trade Rule.

The Thickness of a Steel Tube-plate may be found by the following formula:—

$$\frac{\text{Working pressure in lbs. per square inch} \times W \times D}{20000 \times (D-d)} = \text{the thickness of the}$$
steel tube-plate in inches by the Board of Trade Rule.

In all the above rules for finding the thickness of plates, it is necessary to use the same constants, notation, and restrictions as previously given for the Board of Trade and Lloyd's Rules, of which these formulæ are the converse.

Boiler-Tubes.—The external diameter of the tubes of marine-boilers generally equals $2\frac{1}{2}$ inches for boilers 8 feet diameter; $2\frac{3}{4}$ inches for 9 feet diameter; 3 inches for 10 feet diameter; $3\frac{1}{4}$ for 11 feet diameter; and $3\frac{1}{2}$ inches for boilers 12 feet diameter and upwards. The space between the tubes should be equal to one-half the external diameter of the tube, in order to allow the rising steam to escape freely and be readily replaced by water. When the tubes are placed too closely together, the steam cannot be properly detached, and replaced by water, therefore the water becomes excessively charged with steam, which cannot absorb and conduct heat so readily as water; hence, the heating-surfaces are liable to become overheated and strained or deformed in shape, and the evaporative power of the boiler is diminished. The bottom rows of tubes are of very little value as heating-surfaces, because the bulk of the gases rise to the top of the fire-box and escape through the top rows of the tubes, and very little of the escaping gases pass through the bottom rows of tubes. The first foot in length at the fire-box end of a boiler-tube is much more effective heating-surface than the remainder of the length of tube; it has been found that one-half of the steam generated by tube-surface is produced from a length of tube, measured from the fire-box, equal to about one-sixth the total length of the tube.

The flame enters the tubes at about from 2100° to 2400° Fahr. and leaves about 425° Fahr. in the boilers of portable engines, and at about 600° Fahr. in Cornish, Lancashire, and marine boilers with natural draught, and at about from 700° to 800° Fahr. on an average in locomotive boilers. The length of boiler-tubes averages in marine boilers, twenty-four times their external diameter. Only the top half of boiler-tubes can be regarded as effective heating-surface.

The Heating-Surface of Boiler-Tubes or the tube-surface of boilers is the product of the external circumference of the tubes by the length between the tube-plates. The heating-surface of the tube-plates is the area of the plates minus the area of the Tube-holes, hence the following *Rules*:—

The total heating-surface of boiler-tubes in square feet =

$$\frac{3{\cdot}1416 \times \text{external diameter of tube in ins.} \times \text{length in ins.} \times \text{number of tubes}}{144}$$

The total heating-surface of each tube-plate in square feet =

$$\frac{(\text{Length in ins.} \times \text{width in ins.}) - ({\cdot}7854 \times \text{diameter}^2 \times \text{number of tubes})}{144}$$

Example.—Required the heating-surface of the tubes and tube-plates of a marine boiler with 231 tubes $3\frac{1}{2}$ inches external diameter, and 7 feet 3 inches long between the tube-plates, the tube-plates being each 12 feet by 5 feet.

$$\text{Then } \frac{3{\cdot}1416 \times 3\frac{1}{2} \text{ inches diameter} \times 87 \text{ inches long} \times 231 \text{ tubes}}{144} =$$

1342·7 square feet heating-surface of the tubes.

$$\text{And } \frac{(144 \text{ inches} \times 60 \text{ inches}) - ({\cdot}7854 \times 3{\cdot}5 \times 3{\cdot}5 \times 231)}{144} = 44{\cdot}56$$

square feet; the heating-surface of one tube-plate, and $44{\cdot}56 \times 2 = 89{\cdot}12$ square feet, the heating-surface of both tube-plates. The total surface of the tubes and tube-plates being $= 1342{\cdot}7 + 89{\cdot}12 = 1431{\cdot}72$ square feet, including both the tube-plates, but as the smoke-box tube-plate is not effective heating-surface, it should not be included in the calculation, and only one of the tube-plates should be added to the tube-surface in the above calculation to arrive at the total heating-surface.

The Heating-Surface of Boiler-Tubes may also be calculated by the following *Rule*, which gives the total heating-surface of the tubes in square feet.

Rule: Multiply the decimal number in the following Table, opposite the diameter of the tube in inches, by the length of the tube in feet, the product is the area of each tube in square feet; hence, multiply by the number of tubes, and the product is the total heating-surface of the tubes in square feet; but only about two-thirds can be taken as effective heating-surface for the production of steam.

Diameter of tubes in inches.	Decimal Numbers	Diameter of tubes in inches.	Decimal Numbers.
$1\frac{1}{2}$	·3926	$3\frac{1}{4}$	·8508
$1\frac{3}{4}$	·4580	$3\frac{1}{2}$	·9163
2	·5234	$3\frac{3}{4}$	·9817
$2\frac{1}{4}$	·5888	4	1·0471
$2\frac{1}{2}$	·6543	$4\frac{1}{2}$	1·1778
$2\frac{3}{4}$	·7197	5	1·3088
3	·7854	6	1·5708

Example.—Required the total tube-surface and effective heating tube-surface of a marine boiler containing 250 tubes $3\frac{1}{2}$ inches diameter, and 7 feet in length.

Then ·9163 × 7 feet × 250 tubes = 1603·5 square feet, total heating-surface, and $\frac{1603·5 \times 2}{3}$ = 1069 square feet effective heating-surface.

STEAM-SUPPLY AND EFFICIENCY OF STEAM BOILERS.

Steam-Space of Marine Boilers.—The steam-space should equal in capacity at least forty times that of the high-pressure cylinders of the engines, the steam for which should be taken from the smoke-box end of the boiler. The height from the top of the top row of the boiler-tubes to the top of the boiler-shell, should equal $4\frac{1}{2}$ inches for every foot in diameter of the boiler.

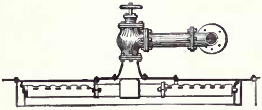


Fig. 73.—Steam collecting-pipe.

Steam should be Drawn from a Boiler through an internal perforated pipe, as shown in Fig. 73, into which the steam can flow gently and steadily, without increasing the ebullition, or causing a rapid current of steam which would carry water with it and produce priming. The total area of the perforations should be equal to double the sectional area of the pipe.

The Area and Number of Perforations required in a Steam-pipe may be found by this *Rule:* Multiply the area of the steam-pipe by 2, which will give the total area of the perforations, then divide by the area of one of the perforations, the quotient will be the number of perforations required.

Example.—A steam-pipe, 9 inches diameter, inside a steam-boiler, is required with perforations or slots, each $4\frac{1}{2}$ inches long and $\frac{1}{4}$ inch wide. How many slots are required to make an area equal to double the sectional area of the pipe?

Then 9 × 9 × ·7854 = 63·6174 square inches area of the pipe.
63·6174 × 2 = 127·2348 square inches, total area of the slots.
4·25 inches × ·25 inch = 1·0625 square inch, area of one slot.
127·2348 ÷ 1·0625 = 119·7, or say 120, the number of slots required.

The quantity of Steam used by an Engine may be found by this *Rule:* Multiply the area of the cylinder in square inches by the speed of the piston in inches per minute, and by the numerator of the fraction of the cut-off, and divide the product by 1728 × the denominator of the fraction of the cut-off of the steam.

Example 1.—How many cubic feet of steam will be used per minute, by an engine with a cylinder 10 inches diameter and 20 inches stroke, making 80 revolutions per minute, and cutting off steam at $\frac{1}{3}$rd of the length of stroke.

$$\text{Then } \frac{10 \times 10 \text{ inches} \times \cdot 7854 \times 20 \text{ stroke} \times 2 \times 80 \text{ revolutions} \times 1}{1728 \times 3} = 48 \cdot 48$$

cubic feet of steam per minute, or $48 \cdot 48 \times 60 = 2908 \cdot 8$ cubic feet per hour.

Example 2. How many cubic feet of steam will be used per hour by *a pair of engines* making 60 revolutions per minute: diameter of cylinders 50 inches, length of stroke 33 inches, cutting off steam at $\frac{3}{4}$ the length of the stroke?

$$\text{Then } \frac{50 \times 50 \text{ ins.} \times \cdot 7854 \times 33 \text{ stroke} \times 2 \times 60 \times 2 \text{ engines} \times 3 \times 60 \text{ minutes}}{1728 \times 4 \text{ denominator of cut-off}}$$

$= 404971 \cdot 8$ cubic feet of steam per hour.

The Quantity of Water evaporated in a Steam-Boiler to Steam per Hour, required to supply steam for a steam-engine, may be found by this *Rule:* Multiply the area of the cylinder in square inches, by the speed of the piston in inches per minute, by the numerator of the fraction of cut-off, and by 60 minutes: and divide the product by the volume of steam, or number of cubic feet of steam from one cubic foot of water, × by 1728 and by the denominator of the fraction of the cut-off of the steam.

Example: Required the quantity of water evaporated to steam per hour, to supply steam for a steam-engine with a cylinder 26 inches. Length of stroke, 30 inches, making 80 revolutions per minute. Pressure of steam by the steam-gauge, 90 lbs. per square inch, cut off at $\frac{5}{12}$ of the length of the stroke.

Then, it will be seen from Table 78, that the volume of steam of 90 lbs. pressure, or 105 lbs. absolute pressure, is 257, and

$$\frac{26 \times 26 \text{ ins.} \times \cdot 7854 \times 30 \text{ stroke} \times 2 \times 80 \text{ revolutions} \times 60 \text{ minutes} \times 5}{257 \text{ volume} \times 1728 \times 12 \text{ denominator of cut-off}} =$$

$143 \cdot 27$ cubic feet of water per hour required to be evaporated to steam for driving that engine.

Efficiency of Steam-Boilers.—The average performance of various types of well-arranged steam-boilers, will be found in the following Table, from which it will be seen that the boiler of a locomotive engine when used as a locomotive, gives far higher results than the same boiler fixed; this is attributed to the disintegrated or open condition of the fire, maintained by the shaking of the engine, and also to the liberation of the steam bubbles from all heating surfaces as fast as made, by the same shaking.

* Table 71.—Efficiency of Steam-Boilers.

Description of Steam-Boiler.	Lbs. of Water Evaporated. Per Square Foot of Heating Surface per Hour.	Lbs. of Water Evaporated. Per lb. of Fuel from and at 212° F.	Thermal Units. In Fuel.	Thermal Units. Transmitted per Hour per Square Foot of Heating Surface.	Thermal Units. Per lb. of Fuel.	Efficiency.	G. Figure of Merit = units per Square Foot per Hour × efficiency.
Field . . .	4·57	8·83	...	4414	8529	...	...
Field . . .	2·28	10·83	...	2202	10461	...	...
Field . . .	2·57	10·93	...	2482	10558	...	...
Portable } Cardiff .	1·52	10·23	14718	1468	9882	67	98356
Portable	2·26	10·49	14718	2183	10133	68	148444
Portable	1·76	11·81	14718	1700	11408	77	130900
Portable	3·56	9·93	...	3438	9592	...	118248
Lancashire . .	1·57	12·83	15715	1516	12393	78	108248
Lancashire . .	2·83	9·89	13833	2733	9553	68	185844
Lancashire . .	1·88	12·25	15715	1816	11833	75	136200
Jacketted . . .	4·70	7·7	14805	4595	7500	50	229750
Lancashire . .	2·57	10·9	15715	2482	10529	67	166294
Compound . .	1·43	11·51	14296	1381	11125	78	107718
Loco. (Webb) . .	9·83	10·28	14004	9495	9930	70	664650
Loco. (Marie) . .	4·62	10·65	14600	4462	10287	70	312340
Loco. } Coke .	12·57	8·22	13550	12142	7940	58	704236
Loco.	13·73	8·94	13550	13263	8636	63	835569
Loco.	6·76	10·01	13550	6530	9669	71	463630
Loco.	7·39	11·2	13550	7138	10819	77	549626
Torpedo . . .	12·54	8·37	14727	12113	8085	54	654102
Torpedo . .	14·86	7·78	14727	14354	7523	51	732054
Torpedo . . .	17·90	7·49	14727	17291	7235	49	847259
Torpedo . .	20·74	7·04	14727	20034	6800	46	921564

HORSE-POWER OF STEAM-BOILERS.

Actual Horse-power of Steam-Boilers.—The quantity of water used per indicated horse-power per hour, by the best condensing engines is about 17 lbs.; the best non-condensing engines use about one-third more than that, or 23 lbs. Allowing a margin of 50 per cent. for engines of inferior economy, the quantity becomes 23 lbs. $\times$ 1·5 = $34\frac{1}{2}$ lbs. On this basis, one actual horse-power is equal to the evaporation in a steam-boiler of $34\frac{1}{2}$ lbs. of water per hour from and at 212° Fahr. This is equal to 30 lbs. of water per hour from feed-water having a temperature of 100° Fahr to steam having a pressure of 70 lbs. per square inch above the atmosphere. The actual horse-power of boilers using steam of other pressures may be

* The above Table was given by Mr. Druit Halpin at a meeting of the Institution of Mechanical Engineers.

Table 72. — SHOWING THE EQUIVALENT EVAPORATION FROM FEED-WATER AT 100° FAHR. TO STEAM OF 70 LBS. PRESSURE, FOR VARIOUS OTHER PRESSURES AND TEMPERATURES OF FEED-WATER.

Temperature of the Feed Water in Degrees Fahr.	Pressure in Pounds per Square Inch above the Atmosphere.															
	0	10	20	30	40	45	50	60	70	75	80	90	100	120	140	160
32° . . .	1·033	1·040	1·046	1·050	1·053	1·055	1.056	1·059	1·061	1·063	1·064	1·066	1·068	1·071	1·074	1·076
40 . . .	1·025	1·033	1·039	1·043	1·046	1·048	1·050	1·052	1·054	1·055	1·056	1·058	1·060	1·064	1·067	1·069
50 . . .	1·016	1·024	1·030	1·034	1·037	1·039	1·040	1·043	1·045	1·046	1·047	1·049	1·051	1·055	1·057	1·060
60 . . .	1·008	1·015	1·020	1·025	1·028	1·030	1·031	1·034	1·036	1·037	1·038	1·040	1·042	1·046	1·048	1·051
70 . . .	·999	1·006	1·012	1·016	1·019	1·020	1·022	1·025	1·027	1·028	1·029	1·031	1·033	1·036	1·039	1·042
80 . . .	·989	·997	1·003	1·007	1·009	1·010	1·013	1·016	1·018	1·019	1·020	1·022	1·024	1·027	1·031	1·033
90 . . .	·980	·988	·993	·998	1·001	1·003	1·004	1·007	1·009	1·010	1·011	1·013	1·015	1·018	1·021	1·024
100 . . .	·971	·979	·984	·989	·992	·994	·995	·998	1·	1·001	1·002	1·004	1·006	1·009	1·012	1·015
110 . . .	·962	·970	·975	·979	·983	·985	·986	·989	·991	·992	·993	·995	·997	1·	1·003	1·006
120 . . .	·954	·961	·966	·970	·974	·976	·977	·980	·982	·983	·984	·986	·988	·991	·994	·997
130 . . .	·944	·952	·957	·961	·965	·966	·968	·971	·973	·974	·975	·977	·979	·982	·985	·988
140 . . .	·935	·943	·948	·952	·956	·957	·959	·962	·964	·965	·966	·968	·970	·973	·976	·979
150 . . .	·926	·934	·939	·943	·947	·948	·950	·952	·955	·956	·957	·959	·961	·964	·967	·970
160 . . .	·917	·925	·930	·934	·938	·939	·941	·943	·946	·947	·948	·950	·952	·955	·958	·961
170 . . .	·908	·916	·921	·925	·929	·930	·932	·934	·937	·938	·939	·941	·943	·946	·949	·952
180 . . .	·900	·907	·912	·916	·919	·921	·923	·925	·928	·929	·930	·932	·934	·937	·940	·943
190 . . .	·890	·898	·903	·907	·910	·912	·913	·916	·919	·920	·921	·923	·924	·928	·931	·934
200 . . .	·881	·888	·894	·898	·901	·903	·904	·907	·909	·911	·912	·914	·915	·919	·922	·924
210 . . .	·872	·880	·885	·889	·893	·894	·895	·898	·900	·901	·902	·904	·905	·909	·913	·915
212 . . .	·870	·877	·883	·887	·890	·892	·893	·896	·898	·899	·901	·903	·904	·908	·911	·914

calculated by means of Table 72, which gives the equivalent evaporation from feed-water of 100° Fahr. to steam of 70 lbs. pressure, for various pressures of steam.

Example of the use of the above Table: A boiler evaporates 24000 lbs. of water in one hour from feed-water at 70° Fahr. to steam of 80 lbs. pressure, per square inch, What is the equivalent evaporation from feed-water at 100° Fahr. to steam of 70 lbs. pressure per square inch? Then on a line with temperature 70 in the first column, is, under the column headed 80 lbs. pressure, the figures 1·029, which multiplied by 24000=2469·6 the equivalent evaporation from water at 100° to steam of 70 lbs. pressure per square inch, and 2469·6 ÷ 30 lbs. evaporation = 82·3 horse-power.

The Nominal Horse-power of Boilers is frequently measured by the evaporation of one cubic foot of water, or 62·4 lbs. to steam per hour.

SAFETY-VALVES FOR STEAM-BOILERS.

A Safety-valve should discharge steam so rapidly, that when the blowing-off pressure, for which the valve is set, is reached, no considerable increase in the pressure of steam can take place, however rapidly the steam is generated. When the steam is shut off from the engine, with a good fire in the furnace, or should the fires be forced, the safety-valve should prevent the pressure of the steam rising above a fixed point, and it should carry off all the steam generated, without the initial blowing-off pressure being exceeded by more than 10 per cent.

The top of a safety-valve is, where convenient, preferably made open to the atmosphere, as it can then easily be seen when the steam has risen too high in the boiler, or when the valves are leaky and are wasting steam. When the valves are bonneted, or provided with a casing over the valve, to which a waste-steam pipe is attached, a drain-pipe should be inserted at the lowest point of the pipes, to carry off the water formed from the condensed steam, otherwise water will accumulate, which, besides causing additional load on the valve, is liable to freeze in frosty weather and render the valve inoperative. Two small safety-valves are more efficient than one large one.

Flow of Steam through Safety-valves.—The velocity with which steam will flow through an orifice from a boiler, such as a safety-valve, is the same as that of a body falling by gravity from the height of a column of steam of uniform density. The formula for gravity is:—

$$V = \sqrt{2\,g\,h}, \text{ or } V = 8\sqrt{h}$$

Where V = the velocity in feet per second.
g = the velocity acquired by a body in falling freely from a state of rest, at the end of one second, being 32·2 feet per second.
h = the height in feet through which the body falls.

Example: Required the velocity with which steam of 75 lbs. per square inch, absolute pressure, will issue through an orifice, or a safety-valve.

Then, as a column of water, 2·309 feet high, will equal a pressure of 1 lb.

per square inch, the pressure will equal $60 \times 2 \cdot 309 = 138 \cdot 54$ feet of water. It will be seen from Table 78, that the volume of steam of 75 lbs., absolute pressure, is 349 times greater than water, and the height of the column of steam will be $138 \cdot 54$ feet $\times 349 = 48351$ feet; the velocity due to that height will, by the above formula, be $\sqrt[2]{48351} \times 8 = 1760$ feet per second.

The Flow of Steam through an Orifice or a Safety-valve, if there were no diminution in the area of the jet of issuing steam, would be, in cubic feet per second, through an orifice 1 square inch in area, equal to the velocity in feet per second, found by the previous rule, divided by 144. Thus, for the pressure of steam given in the last example, the velocity was found to be 1760 feet per second; therefore, the quantity of steam discharged through an orifice 1 square inch in area, would $= \frac{1760}{144} = 12 \cdot 22$ cubic feet per second of steam of 75 lbs. absolute pressure. But the area of the issuing jet of steam in passing through the narrow opening of the valve is reduced to $\cdot 64$ of the actual area, and the actual quantity discharged will be $\frac{1760 \text{ feet} \times \cdot 64}{144} = 7 \cdot 82$ cubic feet per second, or $7 \cdot 82 \times 60 \times 60 = 28152$ cubic feet of steam per hour.

The Flow of Steam through an Orifice, or Safety-valve per square inch of orifice, is frequently taken as being equal to $\frac{1}{70}$ of the absolute pressure of the steam, for pressures above 12 lbs. per square inch above the atmosphere. Hence the following Rules:—

Flow, or weight of steam in lbs. per second, per square inch of orifice =

$$\frac{\text{absolute pressure of steam in lbs.}}{70}$$

Flow, or weight of steam in lbs. per minute per square inch of orifice =

$$\frac{\text{absolute pressure of steam in lbs} \times 60,}{70} \text{ or} = \frac{\text{absolute pressure} \times 6}{7}$$

These rules give the flow of steam through the best form of orifice, that is, with a round edge, but as a square-edged orifice like a safety-valve reduces the flow to the extent of 15 per cent., the rules become:—

Flow, or weight of steam in lbs. per second, per square inch of orifice =

$$\frac{\text{absolute pressure of steam in lbs.} \times 85}{70 \times 100}$$

Flow, or weight of steam in lbs. per minute, per square inch of orifice =

$$\frac{\text{absolute pressure of steam in lbs.} \times 85 \times 60}{70 \times 100}$$

Example: What weight of steam of 75 lbs. absolute pressure per squar inch, will escape through the orifice of a safely-valve of one square inch area in one minute?

$$\text{Then } \frac{75 \text{ lbs.} \times 85 \times 60}{70 \times 100} = 54 \cdot 6 \text{ lbs.}$$

Flow of Steam through a Pipe.—In an experiment on a locomotive-boiler, fired hard, Mr. Webb found that all the steam generated was discharged through an open pipe $\frac{3}{4}$ inch diameter, without raising the pressure more than 10 lbs., and a pipe $1\frac{1}{4}$ inch diameter, was sufficient to discharge it as fast as generated without raising the pressure, which was however very high at the beginning.

Safety-valves are loaded directly by springs or weights, or indirectly by levers and weights. When the fulcrum is secured by a pin, the hole in the lever should be bushed with brass to prevent corrosion. In some cases the lever-end is arranged to turn on a case-hardened knife-edge as shown in Fig. 74, by which arrangement the friction is reduced to a minimum.

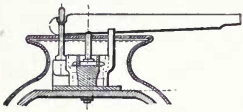


Fig. 74.—Locomotive safety-valve.

Safety-valves loaded with Lever and Weight.—In order to find the weight to be placed on the end of a safety-valve lever, to balance a given pressure of steam on the valve, it is necessary to ascertain the load on the valve due to the weight of the lever. The leverage with which the weight of the lever acts is measured by the distance of its centre of gravity from the fulcrum; the centre of gravity may be found by balancing the lever on a knife-edge, and the weight of the lever and valve may be obtained by weighing them.

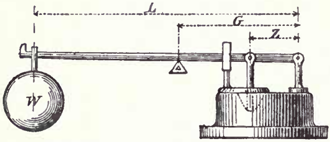


Fig. 75.—Safe.y-valve with lever and weight.

In Fig. 75, W is the weight at the end of the lever; L is the distance between the weight and the fulcrum; G is the distance of the centre of gravity of the lever from the fulcrum; Z is the distance between the centre of the valve and the fulcrum.

The Weight to be placed at the end of the Lever of a Safety-valve, the pressure on the valve, and the length of lever required for a safety-valve, may be found by the following formulæ:—

Let W = the weight in lbs. at the end of the lever.
L = the distance in inches between the weight and the fulcrum.
w = the weight of the lever in lbs.
G = the distance in inches of the centre of gravity of the lever from the fulcrum.
P = the pressure of the steam in lbs. per square inch above the atmosphere.
V = the weight of the valve in lbs.
A = the area of the valve in square inches.
Z = the distance in inches between the fulcrum and the centre of the valve.

The weight in lbs. to be placed at the end of the lever =

$$W = \left\{ (P \times A) - \left(V + \frac{w \times G}{Z} \right) \right\} \frac{Z}{L}$$

The pressure in lbs. per square inch on the valve=

$$P=\left\{\frac{(w\times G)+(L\times W)}{Z}+V\right\}\div A$$

The length of lever in inches=

$$L=\left\{(P\times A)-\left(V+\frac{w\times G}{Z}\right)\right\}\frac{Z}{W}$$

The above formulæ for safety-valves may be expressed in words as follows:—

The Weight to be placed on the end of the Lever may be found as follows:—1st: Multiply the area of the valve by the pressure in lbs. per square inch above the atmosphere, and call the product A.

2nd.—Multiply the weight of the lever by the distance of the centre of gravity of the lever from the fulcrum, and divide by the distance between the centre of the valve and the fulcrum, add the weight of the valve to the quotient, and call the result B.

3rd.—Subtract B from A, multiply the remainder by the distance between the centre of the valve and the fulcrum, and divide by the distance between the weight and the fulcrum, the quotient will be the weight to place on the end of the lever.

To Find the Pressure on the Valve in Lbs. per Square Inch above the Atmosphere.—1st: Multiply the weight of the lever by the distance of the centre of the gravity of the lever from the fulcrum, and call the product C.

2nd.—Multiply the distance between the weight and the fulcrum by the weight at the end of the lever, and call the product D, add C to D, divide by the distance between the centre of the valve and the fulcrum, and add the weight of the valve to the quotient, then divide by the area of the valve; the quotient will be the steam pressure in lbs. per square inch, at which the valve will rise.

To Find the Length of Lever, or distance between the weight and the fulcrum.—1st: Multiply the pressure of the steam in lbs. per square inch above the atmosphere by the area of the valve, and call the product E.

2nd.—Multiply the weight of the lever by the distance of the centre of gravity of the lever from the fulcrum, divide by the distance between the centre of the valve and the fulcrum, and add the weight of the valve to the quotient, and call the result F.

3rd.—Subtract F from E, multiply the remainder by the distance between the centre of the valve and the fulcrum, and divide by the weight at the end of the lever, the quotient will be the distance between the weight and the fulcrum.

Safety-valves with Flat Faces.—In calculating the weight required for a safety-valve with a flat face, the outside diameter of the face must be taken; thus a safety-valve $3\frac{1}{4}$ inches diameter, with a flat face $\frac{1}{8}$ inch wide, should be taken as $3\frac{1}{2}$ inches diameter, as it will be found that the upward pressure on the valve is that due to the area of the outside diameter over the face, and not to the inside diameter of the valve as is the case with a valve having an angular face.

The following example will show the application of the above rules for safety-valves.

Example: Required the weight to be placed on the end of the lever, the pressure on the valve and the length of lever, of a safety-valve of the following proportions:—

Weight of lever = 8 lbs.
Diameter of valve = $3\frac{1}{2}$ inches.
Length of lever = $27\frac{1}{2}$ inches.

Distances of the centre of gravity of the lever from the fulcrum = 10 inches. Weight of valve = 4 lbs.

Pressure of steam above the atmosphere = 60 lbs. per square inch.

The weight in lbs. on the lever:—

(1st). $3{\cdot}5 \times 3{\cdot}5$ inches $\times {\cdot}7854 \times 60$ lbs. $= 577{\cdot}26$ lbs. = product A.

(2nd). $\dfrac{8 \text{ lbs. weight of lever} \times 10 \text{ distance of centre of gravity from fulcrum}}{3{\cdot}5 \text{ inches distance between centre of valve and the fulcrum}}$ $= 22{\cdot}85 + 4$ lbs. $= 26{\cdot}85$ lbs. = product B.

(3rd). $\dfrac{577{\cdot}26 - 26{\cdot}85 \times 3{\cdot}5}{27{\cdot}5 \text{ inches distance between weight and fulcrum}} = 70{\cdot}052$ lbs., the weight to place on the lever for that pressure of steam.

The pressure on the valve in lbs. per square inch:—

(1st). 8 lbs. weight of lever $\times$ 10 inches distance of the centre of gravity from the fulcrum = 80 lbs. = product C.

(2nd). $27{\cdot}5$ inches length of lever $\times$ $70{\cdot}052$ weight on the lever = $1926{\cdot}43$ lbs. = product D.

Then $\dfrac{1926{\cdot}43 + 80}{3{\cdot}5 \text{ inches}} = 573{\cdot}26$ lbs., and $\dfrac{573{\cdot}26 + 4 \text{ lbs. weight of valve}}{9{\cdot}621 \text{ sq. ins. area of the valve}}$ = 60 lbs. per square inch, the pressure of steam at which that valve will rise.

The length of lever in inches:—

(1st). 60 lbs. pressure of steam $\times$ $9{\cdot}621$ area of valve in square inches $= 577{\cdot}26$ lbs. = product E.

(2nd). $\dfrac{8 \text{ lbs. weight of lever} \times 10 \text{ inches distance of the centre of gravity from the fulcrum}}{3{\cdot}5 \text{ inches distance between the centre of the valve and the fulcrum}} = 22{\cdot}85 + 4$ lbs. $= 26{\cdot}85$ lbs. = product F.

(3rd). $\dfrac{577{\cdot}26 - 26{\cdot}85 \times 3{\cdot}5}{70{\cdot}052 \text{ lbs. weight on the lever}} = 27{\cdot}5$ inches, the distance between the weight and the fulcrum, or the length of lever.

The Diameter of a solid Cast-iron Ball may be found by this *Rule:*

$$\text{Diameter in Inches} = \sqrt[3]{\frac{\text{weight of the ball in lbs.}}{\cdot 5236 \times \cdot 26 \text{ cubic inch per lb.}}}$$

Then the diameter of a cast-iron ball for the safety-valve lever described in the previous examples is $= \sqrt[3]{\frac{70\cdot 052 \text{ lbs. weight of ball}}{\cdot 5236 \times \cdot 26}} = 8\frac{1}{64}$ inches diameter.

Counter-balanced Safety-Valve Levers.—If the lever be prolonged beyond the fulcrum, and provided with a weight sufficient to balance the weight of the lever, valve, and connections, as shown in Fig. 75A, then the rules become simplified, and are as follows:—

To find the weight to be placed on the lever.—RULE: Multiply the pressure in lbs. per square inch above the atmosphere by the area of the valve, and by the distance between the centre of the valve and the fulcrum, and divide by the distance between the weight and the fulcrum.

To find the length of lever or distance between the weight and the fulcrum.—RULE: Multiply the pressure in lbs. per square inch above the atmosphere by the area of the valve, and by the distance between the centre of the valve and the fulcrum, and divide by the weight at the end of the lever.

To find the pressure on the valve in lbs. per square inch above the atmosphere.—RULE: Multiply the weight at the end of the lever by the distance between the weight and the fulcrum, and divide by the product of the area of the valve by the distance between the centre of the valve and the fulcrum.

Safety-valve Lever with Spring Balance.—When the lever is pressed down by a spring-balance, as shown in Fig. 76, the lever is generally pro-

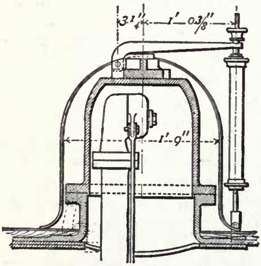


Fig. 76.—Locomotive safety-valve.

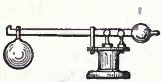


Fig. 75A.—Safety-valve with balanced lever.

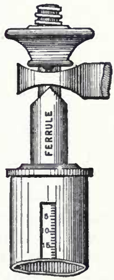


Fig. 77.—Spring-balance with a ferrule on the screw.

portioned, so that 1 lb. at its extremity balances 1 lb. per square inch on the valve. A ferrule should be placed on the screw of the spring-balance, between the nut and the casing, as shown in Fig. 77, to prevent the valve being tampered with, or overloaded by screwing down the spring beyond the blowing-off pressure for which the valve is set.

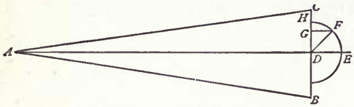


Fig. 78.—Diagram of a helical spring.

Safety-valves loaded with Direct-acting Springs.—Mr. Thomas Adams gave the following graphical method of determining the dimensions of helical springs. In fig. 78 draw A D equal to the length of the spring, and describe an isosceles triangle, having the angle B A C equal to 15 degrees; take two-thirds of B D as radius D E, describe the semi-circle and set off the angle E D F equal to 60 degrees; draw F G, which gives the pitch of the coils, and H G is the side of the square of the steel of which the springs should be made.

BOARD OF TRADE RULES FOR SAFETY-VALVES.

Safety-valves. — The provisions relating to safety-valves are in substance as follows:—Every steam-ship of which a survey is required shall be provided with a safety-valve upon each boiler, so constructed as to be out of the control of the engineer when the steam is up, and if such valve is in addition to the ordinary valve, it shall be so constructed as to have an area not less, and a pressure not greater, than the area of and pressure on that valve.

Cases have come under the notice of the Board of Trade in which steam-ships have been surveyed, and passed by the Surveyors with pipes between the boilers and the safety-valve chests. Such arrangement is not in accordance with the Act, which distinctly provides that the safety-valves shall be upon the boilers.

In all new boilers, and whenever alterations can be easily made, the valve-chest should be placed directly on the boiler; and the neck or part between the chest and the flange which is bolted on the boiler, should be as short as possible and be cast in one with the chest.

Area of Safety-valves.—The area per square foot of fire-grate surface of Government safety-valves, or (when there is more than one Government safety-valve on the boiler) the combined area of the Government safety-valves locked up, should be not less than that given in the following Table opposite the boiler pressure intended, provided the valves are not less than 3 inches in diameter. When, however, the valves are of the common description and are made in accordance with the Table, it will be necessary to fit them with springs having great elasticity, or to provide other means to keep the accumulation within moderate limits; and as boilers with forced draught require valves considerably larger than those found by the Table, the design of the valves proposed for such boilers should be submitted to the Board for consideration.

Table 73.—SAFETY-VALVE AREAS FOR VARIOUS PRESSURES OF STEAM.

Boiler Pressure.	Area of Safety-valve per Square Foot of Fire-grate.	Boiler Pressure.	Area of Safety-valve per Square Foot of Fire-grate.	Boiler Pressure.	Area of Safety-valve per Square Foot of Fire-grate.	Boiler Pressure.	Area of Safety-valve per Square Foot of Fire-grate.
lbs.	sq. in.	lbs.	sq. in.	lbs.	sq. in.	lbs.	sq. in.
15	1·250	62	·487	109	·302	155	·220
16	1·209	63	·480	110	·300	156	·219
17	1·171	64	·474	111	·297	157	·218
18	1·136	65	·468	112	·295	158	·216
19	1·102	66	·462	113	·292	159	·215
20	1·071	67	·457	114	·290	160	·214
21	1·041	68	·451	115	·288	161	·213
22	1·013	69	·446	116	·286	162	·211
23	·986	70	·441	117	·284	163	·210
24	·961	71	·436	118	·281	164	·209
25	·937	72	·431	119	·279	165	·208
26	·914	73	·426	120	·277	166	·207
27	·892	74	·421	121	·275	167	·206
28	·872	75	·416	122	·273	168	·204
29	·852	76	·412	123	·271	169	·203
30	·833	77	·407	124	·269	170	·202
31	·815	78	·403	125	·267	171	·201
32	·797	79	·398	126	·265	172	·200
33	·781	80	·394	127	·264	173	·199
34	·765	81	·390	128	·262	174	·198
35	·750	82	·386	129	·260	175	·197
36	·735	83	·382	130	·258	176	·196
37	·721	84	·378	131	·256	177	·195
38	·707	85	·375	132	·255	178	·194
39	·694	86	·371	133	·253	179	·193
40	·681	87	·367	134	·251	180	·192
41	·669	88	·364	135	·250	181	·191
42	·657	89	·360	136	·248	182	·190
43	·646	90	·357	137	·246	183	·189
44	·635	91	·353	138	·245	184	·188
45	·625	92	·350	139	·243	185	·187
46	·614	93	·347	140	·241	186	·186
47	·604	94	·344	141	·240	187	·185
48	·595	95	·340	142	·238	188	·184
49	·585	96	·337	143	·237	189	·183
50	·576	97	·334	144	·235	190	·182
51	·568	98	·331	145	·234	191	·181
52	·559	99	·328	146	·232	192	·181
53	·551	100	·326	147	·231	193	·180
54	·543	101	·323	148	·230	194	·179
55	·535	102	·320	149	·228	195	·178
56	·528	103	·317	150	·227	196	·177
57	·520	104	·315	151	·225	197	·176
58	·513	105	·312	152	·224	198	·176
59	·506	106	·309	153	·223	199	·175
60	·500	107	·307	154	·221	200	·174
61	·493	108	·304				

The Safety-valves should be fitted with Lifting-gear, so arranged that the two or more valves on any one boiler can at all times be eased together, without interfering with the valves on any other boiler. The lifting gear should in all cases be arranged so that it can be worked by hand either from the engine-room or stoke-hold.

Care should be taken that the safety-valves have a lift equal to at least one-fourth their diameter; that the openings for the passage of steam to and from the valves, including the waste steam-pipe, should each have an area not less than the area of the valve, and that each valve-box has a drain-pipe fitted at its lower part. In the case of lever-valves, if the lever is not bushed with brass, the pins must be of brass; iron and iron-working together must not be passed. Too much care cannot be devoted to seeing that there is proper lift, and free means of escape of waste steam, as it is obvious that unless the lift and means for escape of waste steam are ample, the effect is the same as reducing the area of the valve or putting on an extra load. The valve-seats should be secured by studs and nuts.

Spring Safety-valves.—When spring-loaded valves are used in place of dead-weighted valves, two separate valves are to be fitted to each boiler, except in the case of small boilers, in which the grate-surface does not exceed 14 square feet, in which case a single safety-valve may be passed, provided it is not less than 3 inches diameter. The springs and valves are to be cased in, so that they cannot be tampered with; provision must be made to prevent the valves flying off, in case of the springs breaking; and screw lifting-gear must be provided to ease all the valves.

The Size of Steel for Safety-valve Springs is to be that found by the following formula:—

$$\sqrt[3]{\frac{s \times D}{C}} = d$$

Where s = the load on the spring in lbs.

D = the diameter of the spring in inches, from centre to centre of wire.

d = the diameter, or side of square, of the wire in inches.

C = 8000 for round steel.

C = 11000 for square steel.

The springs are to be protected from steam issuing from the valves when valves are loaded by direct springs; the compressing screws should abut against metal-stops or washers, when the loads sanctioned by the Surveyors are on the valves.

SPRING-LOADED AND OTHER SAFETY-VALVES.

The Board of Trade Rules for Safety-valve Springs may be illustrated by the two following examples:—

Size of steel for the spiral spring:—

Example: A safety-valve $4\frac{3}{4}$ inches diameter is loaded with a spiral spring of round steel 5 inches diameter from centre to centre of wire coils. The boiler pressure of steam shown by the steam-gauge is 90 lbs. per square inch. Required, the diameter of the spring-steel.

The load on the spring = area of the valve in square inches × pressure of steam in lbs., and

$$\sqrt[3]{\frac{{\cdot}7854\times4{\cdot}75\times4{\cdot}75\text{ inches}\times90\text{ lbs.}\times5\text{ inches centres of the spring}}{8000\text{ constant for round steel}}}$$

= 1 inch, the diameter of the steel required for that spring.

Pressure of a spring on a safety-valve:—

$$\left.\begin{array}{r}\text{Pressure in lbs. per square inch}\\ \text{due to springs of round steel}\end{array}\right\}=\frac{8000\times(\text{diameter of steel})^3}{\text{centres of spring in inches}}$$

$$\left.\begin{array}{r}\text{Pressure in lbs. per square inch}\\ \text{due to springs of square steel}\end{array}\right\}=\frac{11000\times(\text{side of square of steel})^3}{\text{centres of spring in inches}}$$

Example: A safety-valve $4\frac{1}{2}$ inches diameter has a spring of $\frac{3}{4}$-inch round steel, $3\frac{3}{4}$ inches diameter from centre to centre of the wire coils. Required, the pressure on the valve, or load due to the spring, and also the pressure if a spring of $\frac{3}{4}$-inch square steel were employed in place of one of round steel.

Then $\dfrac{8000\times{\cdot}75\times{\cdot}75\times{\cdot}75\text{ inch diameter of steel}}{3{\cdot}75\text{ inches centres of the coil}}=900$ lbs., the load on the valve, and $\dfrac{900\text{ lbs. load on the valve}}{4{\cdot}5\times4{\cdot}5\text{ inches diameter of valve}\times{\cdot}7854}=56{\cdot}5$ lbs. per square inch, the pressure or load on the valve due to a spring of $\frac{3}{4}$-inch round steel.

Again $\dfrac{11000\times{\cdot}75\times{\cdot}75\times{\cdot}75\text{ side of square of steel}}{3{\cdot}75\text{ inches centres of the coil}}=1237{\cdot}5$ lbs., the load on the valve, and $\dfrac{1237{\cdot}5\text{ lbs. load on the valve}}{4{\cdot}5\times4{\cdot}5\text{ inches diameter of valve}\times{\cdot}7854}=77{\cdot}8$ lbs., per square inch, the pressure or load on the valve due to a spring of $\frac{3}{4}$-inch square steel.

The Size of Steel in sixteenths of an inch required for the Spring of a Safety-valve, shown in Fig. 79, may be found by the following formulæ:—

Let W = the load on the spring in lbs.
D = the diameter of the spring from centre to centre of the wire coil.
d = the diameter of round steel, or side of square of square steel.

Fig. 79.—Safety-valve spring.

Size of steel for the spiral spring:—

Then $d=\sqrt[3]{\dfrac{W\times D}{2}}$ for round steel

$d=\sqrt[3]{\dfrac{W\times D}{3}}$ for square steel.

Pressure of a spring on a safety-valve:—

$$\left.\begin{array}{l}\text{Pressure in lbs. per sq.}\\ \text{inch due to a spring}\\ \text{of round steel}\quad.\quad.\end{array}\right\}=\frac{(\text{diameter of steel in sixteenths of an inch})^3\times2}{\text{centres of the coil of spring in inches}}$$

$$\left.\begin{array}{l}\text{Pressure in lbs. per sq.}\\ \quad\text{inch due to a spring}\\ \quad\text{of square steel . .}\end{array}\right\} = \frac{(\text{diameter of steel in sixteenths of an inch})^3 \times 3}{\text{centres of the coil of spring in inches}}$$

Size of steel for spiral spring:—

Example: A safety-valve 4 inches diameter has a spring of round steel $3\frac{1}{2}$ inches diameter from centre to centre of the wire coil; the pressure of steam by the steam-gauge is 70 lbs. per square inch. Required the diameter of steel for a spring of round steel, and the side of the square for a spring of square steel.

Then the load on the spring $= 4 \times 4$ inches diameter $\times \cdot 7854 \times 70$ lbs. $=$ 879·2 lbs. per square inch,

and $\sqrt[3]{\dfrac{879\cdot2 \text{ lbs.} \times 3\cdot5 \text{ inches centres of the spring}}{2}} = 11\cdot54$ sixteenths of an inch $= \frac{11}{16}$ and $\frac{1}{32}$ inch, the diameter of round steel required for that spring,

and $\sqrt[3]{\dfrac{879\cdot2 \text{ lbs.} \times 3\cdot5 \text{ inches centres of the spring}}{3}} = 10$ sixteenths of an inch $= \frac{5}{8}$ inch, the side of the square of square steel required.

Pressure of a spring on a safety-valve:—

Example: A safety-valve $4\frac{1}{4}$ inches diameter has a spring of $\frac{13}{16}$ inch diameter steel; the diameter of the centres of the coil is $3\frac{1}{2}$ inches. Required, the pressure on the valve, or load due to the spring, and also the pressure if a spring of square steel be employed in place of one of round steel.

Then $\dfrac{13 \times 13 \times 13 \text{ sixteenths of an inch} \times 2}{3\cdot5 \text{ inches centres of the coil of spring}} = 1255\cdot4$ lbs. load on the valve, and $\dfrac{1255\cdot4 \text{ lbs. load on the valve}}{4\cdot25 \times 4\cdot25 \text{ inches} \times \cdot7854} = 88\cdot5$ lbs. per square inch, the pressure on the valve or load due to the spring of round steel.

Again $\dfrac{13 \times 13 \times 13 \text{ sixteenths of an inch} \times 3}{3\cdot5 \text{ inches centres of the coil of spring}} = 1883\cdot11$ lbs. load on the valve, and $\dfrac{1883\cdot11 \text{ lbs. load on the valve}}{4\cdot25 \times 4\cdot25 \text{ inches} \times \cdot7854} = 132$ lbs. per square inch, the pressure on the valve, or load due to the spring of square steel.

Lloyd's Regulations for Safety-valves.—Two safety-valves to be fitted to each boiler and loaded to the working pressure in the presence of the Surveyor. If common valves are used, their combined areas to be at least half a square inch to each square foot of grate-surface. If improved valves are used, they are to be tested under steam in the presence of the Surveyor; the accumulation in no case to exceed 10 per cent. of the working pressure. Each valve to be arranged so that no extra load can be added when steam is up, and to be fitted with easing-gear which must lift the valve itself. All safety-valve spindles to extend through the covers and to be fitted with sockets and cross handles, allowing them to be lifted and turned round in their seats, and their efficiency tested at any time.

Spring-loaded Safety-valves are used for marine and locomotive boilers, because they are not affected by motion or vibration. A spring-loaded safety-valve for a marine-boiler is shown in Fig. 80. The spring

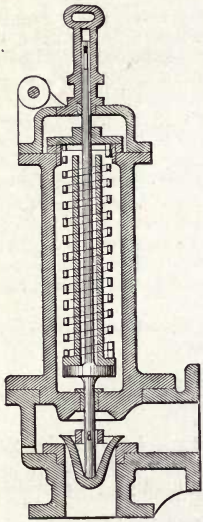


Fig. 80.—Single safety-valve.

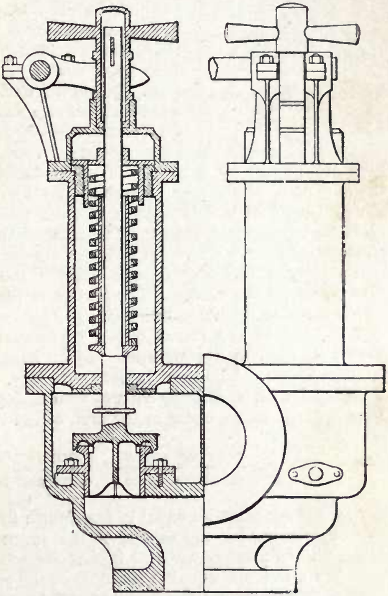


Fig. 81.—Double safety-valve.

and valve is cased in, so that no extra load can be added to the valve when steam is up and it cannot be tampered with. The compression of the spring is frequently made equal to the diameter of the valve, and a maximum lift is provided for equal to one-fourth the diameter of the valve. Easing-gear is fitted to the cap of the spindle, by which the efficiency of the valve may be tested; the valve being attached to the spindle to enable it to be raised or turned round with the spindle, on which the valve is hung loosely to enable it to vibrate freely. A double safety-valve for a marine boiler is shown in Fig. 81, which clearly shows the arrangement of gear for easing the valve.

A Spring-loaded Safety-valve for the Boiler of a Locomotive-engine is shown in Fig. 82. It is fitted with a lever and spring on Ramsbottom's principle. The advantage of this form of safety-valve is, that the point of attachment of the spring to the lever being below the points of the lever pressing on the valves, when one valve opens it tends to relieve the pressure on the other, whereas if the point of attachment were higher, the opening of one valve would put a greater weight on the other valve. When steam is suddenly shut off with a good fire burning in the fire-box, the rise of pressure in the boiler of a locomotive-engine, fitted with a pair of these safety-valves 3 inches diameter, is not more than 5 per cent.

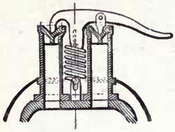


Fig. 82.—Locomotive safety-valve.

A Spring-loaded Safety-valve with Ramsbottom's Lever and Spring of the pattern used on locomotives on the London and North-Western Railway, is shown in Figs. 83—94. The valves are 3 inches diameter; they are loaded by a spring of six coils of $2\frac{7}{8}$ inches internal diameter, made of steel $\frac{11}{16}$ inch by $\frac{22}{32}$ inch. The spring is held between the large circular head of a bolt, and a circular plate at the end of a stirrup-link hung on the lever as shown. The plan of the circular head of this bolt is shown in Fig. 87. Fig. 88 is a plan of the disc against which the spring abuts at its lower end. Fig. 89 is a plan of the stirrup-link, and Fig. 90 is a plan of one of the valves. The spring is placed in a recess in the casting, which has side openings fitted with the perforated plates shown in Fig. 91, and the top of the recess has a perforated cover attached to the lever, shown in plan in Fig. 92, their object being to prevent the spring being tampered with. The lever is shown in Figs. 93 and 94.

The Lift of a Safety-valve, with a flat face, necessary to give a free escape of the steam equal to the area of the valve, is

$$\text{equal to } \frac{\text{area of valve}}{\text{circumference}} \text{ or } \frac{\text{diameter}^2 \times \cdot 7854}{\text{diameter} \times 3\cdot 1416} = \frac{\text{diameter}}{4};$$

that is, the lift must be equal to one-fourth the diameter of the valve for the area of opening for the escape of the steam to be equal to the area of the valve. Thus a safety-valve $4\frac{1}{2}$ inches diameter must lift $4\cdot 5 \div 4 = 1\frac{1}{8}$ inches to obtain an escape equal to the area of the valve.

The Pressure or Load required to Lift a Safety-valve, having a flat face, a given height when loaded for a given pressure of steam, may be found as shown by the following example:—

Example: A safety-valve, 4 inches diameter, with a flat face, is loaded with a spiral-spring having a compression of 4 inches; the working pressure of steam for which the valve is loaded is 80 lbs. per square inch. Required the lift of the valve to give an area of escape equal to one-sixth the area of the valve, and what will be the extra load on the valve due to that lift?

Then 4 inches diameter of valve $\div$ 4 = 1 inch lift to give an escape equal to the area of the valve, and one-sixth of 1 inch = ·166 inch, the lift required to give an escape equal to one-sixth the area of the valve,

$$\text{or } \frac{4 \text{ inches diameter} \times 1}{4 \times 6} = \cdot 116 \text{ inch.}$$

Fig. 83.

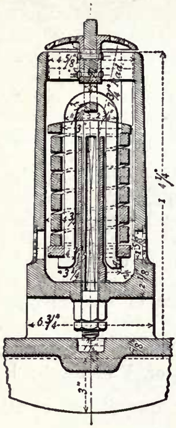


Fig. 84.

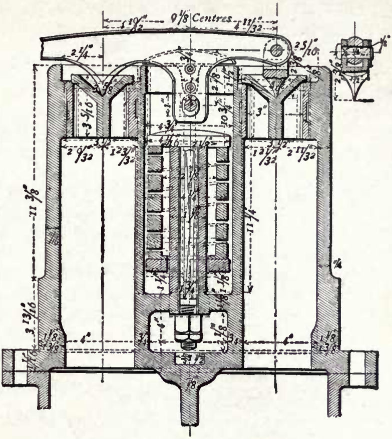


Fig 85.

Fig. 89.

Fig. 87.

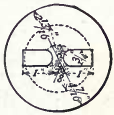


Fig. 86.

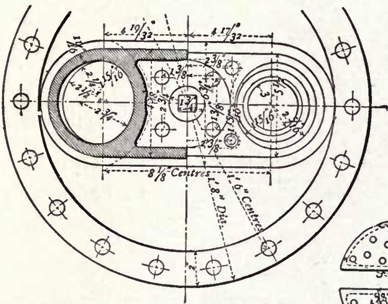


Fig. 90.

Fig. 88.

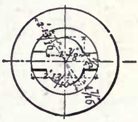


Fig. 92.

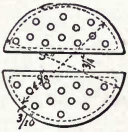


Fig. 91.

Fig. 94.

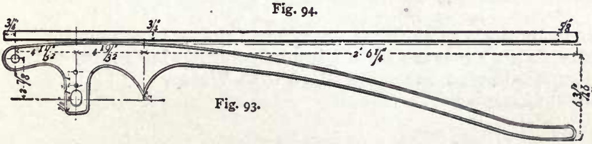


Fig. 93.

Figs. 83—94.—Details of safety-valve with Ramsbottom's lever and spring.

The compression of the spring will be increased to 4 + ·166 inch = 4·169 inches, and the pressure will be increased in the same proportion.

Then 4 : ·166 : : 80 : 3·32 lbs., the extra load on the valve, due to the compression of the spring to the extent of the lift. And 80 lbs. working load + 3·32 lbs. extra load = 83·32 lbs. the lifting load of the valve.

The Weight of Steam that will escape through an Orifice 1 inch square in area in 70 seconds is equal to the number of pounds in the gross pressure of steam per square inch, for steam above 12 lbs. pressure above the atmosphere, whence the width of opening of a safety-valve may be ascertained.

Example: Required the width of opening or height to which a safety-valve, 4 inches diameter, must rise to allow 7300 lbs. of steam to escape per hour, the boiler-pressure of the steam shown by the steam-gauge being 65 lbs. per square inch. Then the absolute pressure of the steam is 65 lbs. + 15 lbs. = 80 lbs. per square inch, and there are 60 × 60 = 3600 seconds in an hour ∴ 70 seconds : 3600 seconds : : 80 lbs. : 4114·28 lbs. weight of steam escaping in one hour through an orifice of one square inch area ∴ 4114·28 lbs. : 7300 lbs. : : 1 square inch : 1·767 square inch, the area required for the escape of 7300 lbs. of steam in one hour and $\frac{1{\cdot}767 \text{ square inch}}{4{\cdot}5 \text{ inch diameter of valve} \times 3{\cdot}1416} = {\cdot}125$ inch or $\frac{1}{8}$ inch, the lift of safety-valve required to allow the escape of that quantity of steam per hour.

The Increase of Pressure on a Spring-loaded Safety-valve due to the Increased Compression of the Spring in Blowing off Steam when the fires are bright and the engines stopped, is approximately equal to the quotient of the diameter of the valve divided by the original compression of the spring, when the area of the valve equals $\frac{1}{2}$ inch per square foot of fire-grate according to the Board of Trade Rule.

Example: The spring of a safety-valve has a compression of $4\frac{3}{4}$ inches, there are 39·28 square feet of fire-grate to the valve, the loaded pressure is 60 lbs. per square inch, the increase, irrespective of the spring, would be, say, 10 per cent.: to what pressure would the steam rise, including the effect of the spring?

Then the diameter of the valve according to the Board of Trade Rule would be $\frac{39{\cdot}28 \text{ square feet area of firegrate}}{2} = 19{\cdot}64$ square inches area,

and $\sqrt[2]{\frac{19{\cdot}64}{{\cdot}7854}} = 5$ inches diameter.

And $\frac{5 \text{ inches diameter of safety-valve}}{4{\cdot}75 \text{ compression of the spring}} = 1{\cdot}052$ lb. extra pressure due to increased compression of the spring, and 60 lbs. original pressure, + 6 lbs., or 10 per cent. increase, irrespective of the spring = 67·052 lbs. per square inch, the pressure to which the steam would rise on blowing-off.

The Extra Pressure on a Direct Spring-loaded Safety-valve due to the Waste-pipe becoming filled with Water, may be found by the following formula:

Let P = the pressure in lbs. per square inch due to the weight of the water, then as a pressure of 1 lb. per square inch is exerted by a column of water 2·309 feet high or 27·7 inches high at a temperature of 62° Fahr.

$$P = \frac{\text{Height of the column of water in feet}}{2{\cdot}309}$$

$$P = \frac{\text{Height of the column of water in inches}}{27{\cdot}7}$$

Example: The waste-steam-pipe of a spring-loaded safety-valve was found at one time to be filled with water to a height of 8 feet, and at another time to a height of 41 inches. Required the extra pressure caused in each case by the weight of the water on the safety-valve.

Then $\frac{8 \text{ feet}}{2{\cdot}309} = 3{\cdot}46$ lbs. per square inch, the pressure due to the weight of the water on the valve in one case; and $\frac{41 \text{ inches}}{27{\cdot}7} = 1{\cdot}48$ lbs. per square inch, the pressure due to the weight of the water on the valve in the other case.

Safety-valve Openings.—The consumption of coal in marine-boilers with natural draught, is not often more than 20 lbs. of coal per square foot of fire-grate surface per hour, and the water evaporated seldom exceeds 9 lbs. per lb. of coal, which corresponds to 180 lbs. per hour, or an evaporation of 3 lbs. per minute per square foot of fire-grate: under these conditions the area of opening requisite to discharge all the steam a boiler can generate, corresponds to four times the square feet of fire-grate divided by the absolute pressure, or pressure shown by the steam-gauge plus 15 lbs. The consumption of coal in stationary boilers may be taken at 15 lbs. per square foot of fire-grate surface, and the area of safety-valve orifice may be found by the following formula for each class of boiler.

$$\left.\begin{array}{l}\text{Area of orifice for a safety-valve for a}\\ \quad\text{marine-boiler} \quad . \quad . \quad . \quad . \quad .\end{array}\right\} = \frac{4 \times \text{square feet of fire-grate}}{\text{absolute pressure of steam}}.$$

$$\left.\begin{array}{l}\text{Area of orifice for a safety-valve for a}\\ \quad\text{stationary boiler} \quad . \quad . \quad . \quad .\end{array}\right\} = \frac{3 \times \text{square feet of fire-grate}}{\text{absolute pressure of steam}}.$$

The Board of Trade allowance is one-half square inch area of safety-valve for each square foot of fire-grate, hence, the lift of the valve is proportional to the diameter and inversely as the pressure. For a discharge of 3 lbs. per minute per square foot of fire-grate, the requisite lift is obtained by the following formulæ * for pressures of steam above 25 lbs. per square inch.

Let L = th lift of va! e in inches necessary for a discharge of 3 lbs. per minute per square foot of fire-grate. Then—

$$L = \frac{2 \times \text{diameter of the valve in inches}}{\text{absolute pressure of the steam}}, \text{ for flat-faced valves.}$$

$$L = \frac{2{\cdot}8 \times \text{diameter of the valve in inches}}{\text{absolute pressure of steam}}, \text{for valves with angle of seat of } 45^\circ.$$

* See a "Report on Safety Valves, presented to the Engineers and Shipbuilders of Scotland," to which the Author is indebted for some of the above and following rules and data.

Take for example a safety-valve 5 inches diameter = 19·6 square inches area, which corresponds to $2 \times 19·6 = 39·2$ square feet of fire-grate surface, which would evaporate 39·2 square feet $\times$ 3 lbs. = 117·6 lbs. of water per minute.

Then, since the area, A, in square inches requisite to discharge any weight, w, in lbs. of steam per minute at the pressure, p, is—

$$A = \frac{4\,w}{3\,p}$$

It would give, by taking the pressure, p, at 60 lbs., and the weight, w = 117·6 lbs., the area $A = \frac{4 \times 117·6}{3 \times 60} = 2·61$ square inches, which corresponds to the opening of a flat-faced valve 5 inches diameter, when the lift equals $\frac{2 \times 5}{60} = ·1667$ inches. The circumference of a valve 5 inches diameter being 15·7 inches, and $15·7 \times ·1667 = 2·61$ square inches of opening as found above.

In the Report on safety-valves above referred to it is recommended that two safety-valves be fitted to each marine-boiler, one of which should be *an easing-valve* and the dimensions of each of these valves, if of the ordinary construction, should be calculated by the following *Rule:*

Let A = area of valve in square inches.
G = grate surface in square feet.
HS = heating surface in square feet.
P = absolute pressure of steam in lbs. per square inch; or pressure shown by the steam-gauge plus 15 lbs.

$$\text{Then } A = \frac{18 \times G}{P} \text{ or } A = \frac{·6 \times HS}{P}.$$

Only one of the valves may be of the ordinary kind and proportioned by this formula, and it is to be *the easing-valve.* The other may be so constructed as to lift one-fourth of its diameter without increase of pressure, and one such valve, if calculated by the following formula, would be of itself sufficient to relieve the boiler.

Area of safety-valve in square inches.

$$A = \left.\frac{4 \times \text{grate surface in square feet}}{\text{absolute pressure of steam in lbs. per square inch}}\right\} + \text{area of guides of valve.}$$

$$\text{or } A = \left.\frac{·133 \times \text{heating surface in square feet}}{\text{absolute pressure of steam in lbs per square inch.}}\right\} + \text{area of guides of valve.}$$

The safety-valve calculated by the above formula, should be loaded 1 lb. per square inch less than the load on *the easing-valve.* If the heating-surface exceed 30 feet per foot of fire-grate surface, the size of safety-valve should be determined by the heating-surface. The valves should be flat-faced, and the breadth of face should not exceed one-twelfth of an inch.

The Silent Blow-off Ejector, shown in Fig. 95, is fixed on the side of a ship, for the purpose of discharging the waste-steam from a safety-valve into the sea, without noise or increase of pressure.

Dead-weighted Safety-Valves are directly loaded with a number of cheese-shaped weights, placed upon a strong spindle bearing on the valve, which is hung loosely on the spindle, as shown in Fig. 96: the weights are encased to prevent their being tampered with. The top of the spindle is covered by a cap, which fits loosely in a hole in the top of the cover of the chest, in order to allow the valve to be turned round on its seat by means of the cross-handle on the cap. The cap is attached to the valve-spindle by a cotter, the cotter-hole is made deeper than the cotter, to allow the spindle to rise in the cap when the valve is raised from its seat, and a forked-lever is fitted to the bottom of the cap, by which the valve may be tested by raising it when required. The area of the waste-steam pipe is equal to that of the valve. This description of safety-valve is now only used for stationary boilers. It is not suitable for marine boilers, because the effect of the weight on the valve is reduced by the heeling of the ship in rough weather, causing the valve to blow-off and waste steam; and the oscillation of the weights due to the rolling of the ship causes the valve to get out of order and leak.

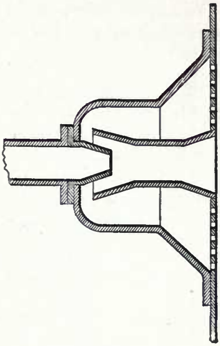


Fig. 95.—Silent Blow-off.

To find the Pressure on a Dead-Weighted Safety-Valve.—*Rule:* Divide the total weight or load in lbs. on the valve, including the weight of the valve and spindle, by the area of the safety-valve in square inches.

Example: A safety-valve $4\frac{1}{2}$ inches diameter is loaded with dead-weights as follows:—8 of 105 lbs. each, 3 of 80 lbs. each, and one of 55 lbs., the valve weighs 15 lbs., and the spindle, 20 lbs. Required the pressure in lbs. per square inch on the valve.

Then $(105 \times 8) + (80 \times 3) + 55 + 15 + 20 = 1170$ lbs., total weight, or load on the valve; and $\frac{1170 \text{ lbs. total weight}}{4{\cdot}5 \times 4{\cdot}5 \times {\cdot}7854} = 73{\cdot}5$ lbs. per square inch, the pressure on the valve.

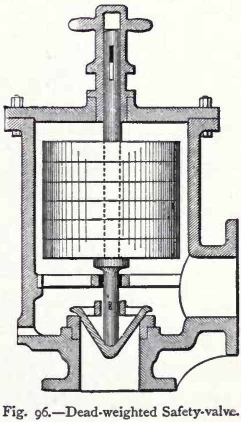


Fig. 96.—Dead-weighted Safety-valve.

To find the Weight required for a Dead-Weighted Safety-Valve.—*Rule:* Multiply the area of the safety-valve in square inches by the pressure of the steam in lbs. per square inch.

Example: Required the weight, including that of the valve and spindle, for a dead-weighted safety-valve, of 5 inches diameter, the pressure of the steam being 60 lbs. per square inch.

Then 5×5 inches $\times \cdot 7854 \times 60$ lbs. $= 1178 \cdot 1$ lbs., the total weight required, including that of the valve and spindle.

When the plates of a steam-boiler become thinned by corrosion or weakened by age, the blowing-off pressure of the safety-valve should be lowered, by reducing the load or weight on the valve to that required for the reduced working-pressure of the boiler.

The Reduction to be made in the Weight or Load, when the blowing-off pressure of a dead-weighted safety-valve is reduced, may be found by the following formula:—

Let W = the original weight or load on the safety-valve.
P = the original blowing-off pressure of the steam.
p = the reduced blowing-off pressure of steam required.
w = the reduced weight on the safety-valve for the reduced pressure of steam.

$$\text{Then } w = \frac{P - p}{P} \times W$$

Example: A dead-weighted safety-valve is loaded with a weight of 700 lbs. for a blowing-off pressure of 60 lbs. per square inch. Required the reduction to be made in the weight or load, if the blowing-off pressure be reduced to 45 lbs. per square inch.

Then $\dfrac{60 \text{ lbs.} - 45 \text{ lbs.}}{60 \text{ lbs.}} \times 700$ lbs. weight $= 175$ lbs. to be taken off the weight on the valve.

The Weight of the Dead-Weights of a Safety-Valve may be calculated approximately from their cubic contents, allowing one cubic inch of cast-iron to weigh $\cdot 26$ lb.

Example: A safety-valve, $3\frac{3}{4}$ inches diameter, is loaded with eight dead-weights of cast-iron, each $14\frac{5}{8}$ inches diameter, and $2\frac{1}{2}$ inches thick, the weight of the valve-spindle and valve is 33 lbs. Required the weight or load per square inch on the valve, and also how much additional weight must be placed on the valve to produce an additional load of 7 lbs. per square inch?

Then $14 \cdot 625 \times 14 \cdot 625 \times \cdot 7854 \times 2\frac{1}{2}$ inches thick $\times$ 8 weights $\times \cdot 26$ lbs. $= 873 \cdot 6$ lbs., the weight of the 8 cast-iron weights; and $873 \cdot 6$ lbs. $+ 33$ lbs. $= 906 \cdot 6$ lbs., the total weight of the weights, valve-spindle and valve; and $\dfrac{906 \cdot 6 \text{ lbs.}}{3 \cdot 75 \times 3 \cdot 75 \times \cdot 7854} = 82 \cdot 11$ lbs. per square inch weight on the valve; and $3.75 \times 3 \cdot 75$ diameter of valve $\times \cdot 7854 \times 7$ lbs. $= 77 \cdot 28$ lbs., the additional weight required to produce an additional load of 7 lbs. per square inch on the valve, making the total load on the valve $= 82 \cdot 11 + 7 = 89 \cdot 11$ lbs. per square inch.

FEED-PUMPS, FEED-VALVES, AND FEED-INJECTORS, FEED-WATER, AND FEED-WATER HEATERS.

Feed-water-Consumption in Steam Boilers.—The quantity of water used per hour per indicated horse-power of the engines, averages as follows:—Non-condensing engines use 40 lbs. of water per indicated horse-power per hour; condensing engines, 30 lbs.; compound condensing engines, 20 lbs.; and triple-expansion engines, and quadruple expansion-engines, use from 14 to 18 lbs. of water.

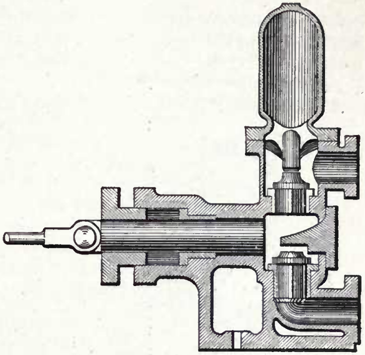


Fig. 97.—Horizontal feed-pump.

Feed-Pumps for Feeding Boilers are shown in Figs. 97 and 98. In pumping hot water the feed-pump should be placed at the same level as the supply-tank, so that the water may flow into the barrel of the pump by its own gravity. The pump-valves should be large and their lift small. A feed-pump should be capable of supplying considerably more water than is actually required for a boiler. The size of a feed-pump, capable of supplying three times the quantity of water required to be evaporated to steam for a given size of engine-cylinder, may be found by the following formula:—

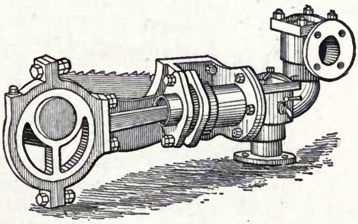


Fig. 98.—Horizontal feed-pump, with hollow ram.

Let A = the area of the pump-ram in square inches.
C = the area of the engine-piston in square inches.
L = the length of stroke in feet, before the steam is cut off.
S = the length of stroke of the pump in feet.
V = the volume of the steam from Table 78, corresponding to the absolute pressure of the steam, or pressure including that of the atmosphere.

Then $A = \dfrac{C \times L \times 6}{V \times S}$.

Example: Required the diameter of the ram of a feed pump with a 2-feet stroke, to feed the boiler of an engine having a cylinder 30 inches diameter, the steam being cut off when the piston has travelled 1·2 feet of its stroke, the pressure of steam by the steam gauge is 70 lbs. per square inch. Then the volume of steam of $70+15=85$ lbs. absolute pressure is, from Table 78, $=311$,

and $\dfrac{30 \times 30 \text{ inches} \times \cdot7854 \times 1\cdot2 \text{ feet stroke of engine} \times 6}{313 \text{ volume of steam} \times 2 \text{ feet stroke of pump}} = 8\cdot18$ square inches, the area of the ram of the feed pump, and $\sqrt[2]{\dfrac{8\cdot18}{\cdot7854}} = 3\frac{1}{4}$ inches, the diameter of the pump-ram. The capacity of a feed-pump is frequently made equal to the supply of five gallons of water per hour per indicated horse-power of the engine. The most efficient speed of a feed-pump is about 50 feet per minute.

A Feed Back-pressure Valve, or non-return Valve, through which the feed-water enters the boiler is shown in Fig. 99. The lift of the valve can be regulated by the spindle, with which the valve may be locked when required. The lift of a feed-valve is usually $\frac{1}{8}$ inch; it should not exceed $\frac{1}{4}$ inch, otherwise the valve-seating soon becomes impaired by the hammering action of the valve. The average rate of the flow of feed-water through the valves is from 200 feet to 400 feet per minute. For locomotive boilers a clack-box with a ball-valve, shown in Fig. 100, is generally used as a non-return feed-valve. Ball-valves are most suitable for the high speed of a locomotive, as they act more freely and are less liable to stick fast than mitre-valves. Feed-valves should be taken out, examined and adjusted when a boiler is cleaned.

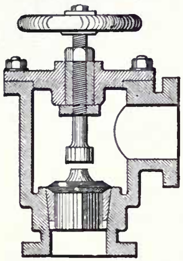


Fig. 99.—Feed back-pressure valve.

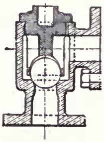


Fig. 100.—Clack-box, with ball-valve.

The Action of an Injector is due to the momentum imparted to the feed-water by a jet of steam moving at a high velocity. The steam used in working the injector is condensed and sent into the boiler with the feed-water, which by this means becomes considerably raised in temperature.

The Rise of Temperature of the Feed-water in passing through an Injector may be explained as follows:—

If 1 lb. of steam in motion were mixed with 2 lb of water at rest, the result produced would be 3 lb put in motion at one-third the original velocity of the steam. The velocity of water or steam issuing into the atmosphere from the same boiler, is equal to that acquired by a falling body in falling through the height of a column of the same water or steam giving the same effective pressure. And since the velocity acquired by a falling body is proportional to the square root of the height through which it fell, it follows that the velocity of the water and the steam would be proportional to the square roots of their relative volumes. And as the volume of steam with one atmosphere effective pressure, or 30 lbs. absolute pressure, is from Table 78, $=827$ times that of water, it would issue with the square root of $827=28\cdot76$, or say twenty-nine times the velocity of the water from the same boiler. Hence, the steam issuing would just balance twenty-nine times its own weight of water trying to issue from the boiler. The number of units of heat in 1 lb. of steam of 30 lbs. absolute pressure is from Table

79, = 1159 Fahr., and assuming the original temperature of the feed-water at 100 degs., the rise of temperature of the feed-water would be =

$$\frac{1159 \text{ units steam} - 100^\circ \text{ water}}{29+1} = 35{\cdot}3^\circ \text{ Fahr.},$$

with steam of one atmosphere effective pressure, or 30 lbs. per square inch absolute pressure. With steam of 3 atmospheres effective pressure, or 75 + 15 = 90 lbs. per square inch absolute pressure, its volume is 295 that of water; then $\sqrt[2]{295}$ = say 17; hence the issuing steam would balance 17 times its own weight of water trying to issue from the boiler. The number of units of heat in steam of this pressure = 1180 per lb. of steam, and if the original temperature of the feed-water be = 100° Fahr., the rise of temperature of the feed-water would be $= \frac{1180-100}{17+1} = 60^\circ$ Fahr.

The Diameter of the Nozzle of an Injector may be found by Rankine's rules, which are as follows:—

$$\text{Sectional area of nozzle in square inches} = \frac{\text{Cubic feet per hour of gross feed-water}}{800 \sqrt{\text{pressure in atmospheres}}};$$

$$\text{Sectional area of nozzle in circular inches} = \frac{\text{Cubic feet per hour of gross feed-water}}{630 \sqrt{\text{pressure in atmospheres}}}.$$

Example: If a boiler use 1200 lbs. of coal per hour, and each lb of coal evaporates 8 lbs. of water, what is the proper diameter of the narrowest part of the nozzle of an injector to deliver double the required feed, the pressure of steam by the steam gauge being 75 lbs. per square inch

Then $(75+15) \div 15 = 6$ atmospheres, $\sqrt{6} = 2.449$: and $\frac{1200 \times 8 \times 2}{62{\cdot}5 \text{ lbs.}} =$ 307·2 cubic feet per hour of gross feed-water

Then $\frac{307{\cdot}2 \text{ cubic feet}}{630 \times 2{\cdot}449} = {\cdot}199$ circular inch sectional area; and $\sqrt[2]{{\cdot}199}$ = ·446 inch, the diameter of the nozzle of the injector.

Quantity of Feed-water carried by an Injector per lb. of Steam.—In practice, with the best injectors, each pound of steam carries with it and forces into the boiler, about 18 lbs. of feed-water, its temperature being raised about 100° Fahr. in passing through the injector. The injector will lift its water supply from 10 to 20 feet.

An Automatic Re-starting Injector is shown in Fig. 101. It is very simple, efficient, and certain in its action, and will work continuously without attention, and if from any cause it should be stopped it will start itself again. It has three cones—a steam cone, a water or combining cone, and a delivery cone; its action may be briefly described as follows. When steam is turned on the injector, it rushes down the steam cone and through the large end of the water cone and out at the point where this cone is divided, creating a vacuum in the water chamber; the water then rushes up into the chamber and surrounds and condenses the steam, which, in the form of partially condensed steam and water, leaps across the opening between the larger and smaller ends of the water cone; this jet creates a vacuum in the overflow-chamber, when the pressure of the atmosphere at once forces the smaller end of the water cone—which is free to slide—up to the larger end, thus closing the space between the two halves of the water cone and making a continuous water and combining cone as in the ordinary injector. The combined jet of steam and water then passes out at the ordinary overflow, until the velocity is sufficiently great for it to enter the delivery cone and pass forward into the boiler. The automatic action of this injector consists of the opening and closing of the space between two halves of the water or combining cone, this space being always open except when the steam and water are both present, so that should there be any interference with the water supply the steam simply rushes out at the overflow until such time as the water comes again, when the injector immediately starts to work without any attention.

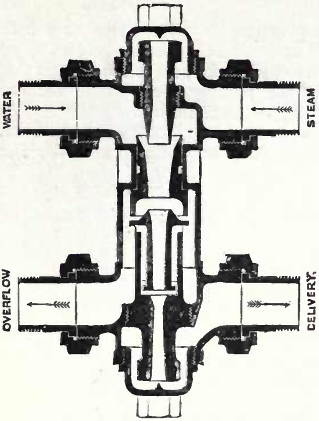


Fig. 101.—Gresham's automatic re-starting injector.

The Automatic Exhaust-Steam Injector, shown in Fig. 102, is an extremely simple, effective, and economical boiler-feeder, it being worked by the exhaust-steam from a non-condensing engine, instead of using live-steam from the boiler for this purpose. Although the exhaust-steam enters the injector only at a little above atmospheric pressure, it will feed a boiler against a steam pressure of 75 lbs. per square inch. This injector has three cones, the steam cone, the combining, mixing, or condensing cone and the delivery cone. In Fig. 102, A is the steam supply, the butterfly valve shown, being for the purpose of shutting off the steam when required. B is the steam cone, and S the steam nozzle; C is the water supply, and T the water nozzle; D is the combining, E the jet nozzle, and F the throat; K is the fixed spindle; R the regulator, and G, the overflow. The combining cone is split longitudinally for more than half its length from the smaller end; one-half hinges from a pin at its upper end, in such a manner that it hangs open when the injector is not at work: this position of the flap is shown in dotted lines, and it will be seen that a large area is presented for the passage of water when the latter is turned on. As soon as a vacuum is established in the combining cone the loose half of the nozzle is drawn inwards, and the cone behaves precisely the same as if it were not split. If the supply of water or steam fails for an instant, the flap immediately falls away, and a large opening is presented once more for the water to pass through, the exhaust-steam is induced again to enter, condensation takes place with the formation of a partial vacuum, the nozzle is drawn inwards again, and the instrument resumes work.

The Action of the Exhaust-steam Injector is as follows:—When water is turned on to the injector, it flows into the combining cone and out at the overflow; a little steam will in this manner be induced to flow into the injector, and will there be condensed, forming a partial vacuum; more steam will follow to fill up the vacuum formed, which in turn will be condensed, forming a better vacuum and a greater rush of exhaust-steam, then the steam itself becomes the working factor and rushes into the combining cone with such force as to overcome the back pressure, and, along with a portion of the water, enters the boiler, the action then results from the conversion of the latent heat of the steam into the energy of motion.

The Method of Fixing the Exhaust-steam Injector is shown in Fig. 103. A branch-pipe is taken from the exhaust-pipe of the engine and connected to the injector, which is placed vertically. The end of the overflow-pipe is placed below the surface of the water in a tank to prevent air entering the exhaust-chamber. Water is supplied to the injector from a tank placed higher than the injector, into which it flows by gravity.

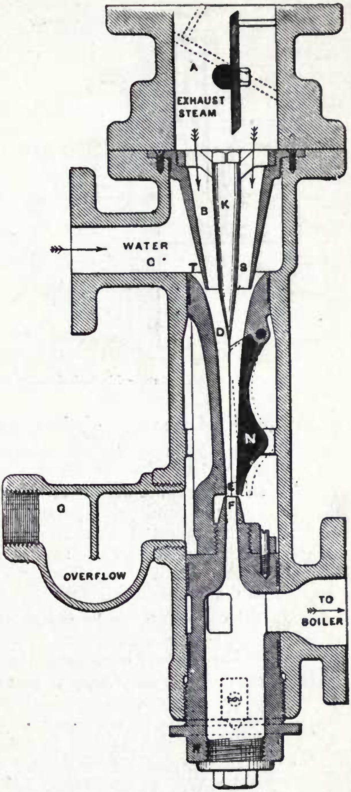


Fig. 102.—Davis, Hamer & Metcalf's Automatic exhaust-steam injector.

The Proportion of the Steam and Water Supply for the Exhaust-Steam Injector, the temperature of the mixture, &c., and the quantity of the steam condensed, may be ascertained as follows: *—

The steam in condensing gives out the same quantity of heat as the water absorbed during the process of evaporation, or which is the same thing, the units of heat expended in changing the water into steam. That

* See a paper read before "The Midland Counties Institution of Engineers," by Mr. H. Fisher.

is, the heat which 1 lb. of steam gives out in condensing would heat 966·6 lbs. of water 1° Fahr., or 9·666 lbs. 100° Fahr., and so on in proportion. Then the temperature is raised still higher, by the mixture of the

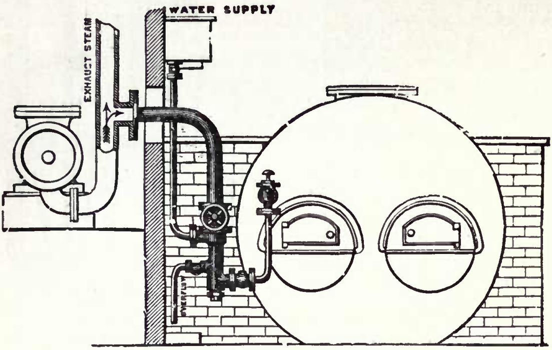


Fig. 103.—Method of fixing the exhaust-steam injector.

heated water and the water of the condensed steam. Hence the following Rules:—

Let L = the latent heat of steam.

T = the sensible heat of the steam, or the boiling point of water and the condensing point of steam under a given pressure.

w_o = the weight of steam in lbs.

w = the weight of water by the condensed steam in lbs.

W = the weight of supply-water in lbs.

t = the temperature of the supply-water.

t_o = the temperature of the heated water by the condensation of the steam.

T_o = the ultimate temperature of the mixture of water and steam.

Then, the proportion of supply-water to steam is—

$$W=\frac{L+(T-T_o)}{T_o-t} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad A$$

The temperature of the heated water by the condensation of the steam is—

$$t_o=\frac{L\times W_o}{W}+t \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad B$$

The ultimate temperature of the mixture is—

$$T_o=\frac{(t_o\times W)+(T\times w)}{W+w} \text{ or} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad C$$

$$T_o=\frac{(L\times w_o)+(T-t)\,w}{W+w}+t \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad D$$

The quantity of steam condensed in the mixture is—

$$w_0 = \frac{W \times (T_0 - t)}{L + (T - T_0)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad E$$

Assume for example that the temperature of the supply-water is 60° Fahr., and the proportion 9·666 lbs. to 1 lb. of steam; then the temperature of the heated water by the condensing of the steam is, by formula B,

$$= \frac{966{\cdot}6 \times 1}{9{\cdot}666} = 100 + 60 = 160° \text{ Fahr.}$$

The ultimate temperature of the mixture is, by formula C,

$$= \frac{(160 \times 9{\cdot}666) + (212 \times 1)}{9{\cdot}666 + 1} = \frac{1546{\cdot}56 + 212}{10{\cdot}666} = \frac{1758{\cdot}56}{10{\cdot}666} = 164°{\cdot}875 \text{ Fahr.,}$$

and by formula D,

$$\frac{(966{\cdot}6 \times 1) + (212 - 60) \times 1)}{9{\cdot}666 + 1} + 60 = \frac{966{\cdot}6 + 152}{10{\cdot}666} + 60 = \frac{1118{\cdot}6}{10{\cdot}666} = 104{\cdot}875 + 60$$

$$= 164°{\cdot}875 \text{ Fahr.}$$

The proportion of supply-water is, by formula A,

$$= \frac{966{\cdot}6 + (212 - 164{\cdot}875)}{164{\cdot}875 - 60} = \frac{966{\cdot}6 + 47{\cdot}125}{104{\cdot}875} = \frac{1013{\cdot}725}{104{\cdot}875} = 9{\cdot}666 \text{ lbs. to 1 lb.}$$

of steam.

The Quantity of Exhaust-Steam returned to a Boiler in the form of Feed-Water, by the Exhaust-Steam Injector, may be seen from the following calculation:—

If two injectors together feed 2740 gallons, or 2740 × 10 lbs. = 27400 lbs. of water per hour; and if the proportion of water to steam, calculated by Formula E, be =

$$\frac{27400 \times (153 - 52)}{966{\cdot}6 + (212 - 153)} = \frac{27400 \times 101}{966{\cdot}6 + 59} = \frac{2767400}{1025{\cdot}6} = 2698$$

Then 2698 lbs., or 1·2 ton of exhaust-steam will be returned to the boilers per hour, raising at the same time the temperature of the feed-water 101° Fahr., the economy of which may be thus expressed:—

Units of heat required to raise the supply-water from 52° Fahr. to steam at 60 lbs. pressure = 1207 − 52 = 1155, and units of heat required to raise the feed-water from 153° Fahr. to steam at the same pressure = 1207 − 153 = 1054, showing a saving of 101 units of heat, which, if applied, would increase the quantity of steam 9·5 per cent.

The Feed-Water Pipes from the Injectors to the boilers may be taken through a tube, through which the remaining exhaust-steam flows, after having passed the branch exhaust-steam pipe leading to the injectors. By this means the feed-water may be delivered to the boilers at a temperature of 204° Fahr.

Theoretically, the beneficial results of feeding the boilers with water at a temperature of 204° Fahr., compared with a temperature of 52° Fahr., are as follows:—

Units of heat required to raise the supply-water from a temperature of

52° Fahr. to steam at 60 lbs. pressure = 1207—52 = 1155; and, to raise the feed-water from a temperature of 204° Fahr. would require 1207—204 = 1003 units of heat, or 152 units of heat less than the water at the lower temperature. By 152 additional units of heat, the quantity of steam would be increased 15 per cent., which is equal to a saving of one boiler in seven.

This result has actually been attained in practice, as much steam being supplied in one case from six boilers, as could be supplied from seven boilers fed with cold water and an increased consumption of coal.

Exhaust-Steam, as a Water-Heater, is very little inferior to live-steam at 60 lbs. pressure. For instance, taking the sensible heat for that pressure at 307°·5 Fahr., and the latent heat at 900°·3 Fahr., then if the temperature of the supply-water be 52° Fahr., by formula D the temperature of the feed-water would be—

$$\frac{900{\cdot}3 + (307{\cdot}5 - 52)}{10{\cdot}1 + 1} + 52 = \frac{900{\cdot}3 + 275{\cdot}5}{11{\cdot}1} + 52 = \frac{1175{\cdot}8}{11{\cdot}1} + 52 = 156^\circ \text{ Fahr.},$$

with live steam, or only 3° Fahr. higher than would be obtained by the water being heated with exhaust-steam.

An experiment was made to determine the efficiency of an exhaust-steam injector compared with a Giffard's injector using live-steam, when the following results were obtained:—

Table 74.—Results of Tests of an Exhaust-Steam Injector compared with an ordinary Giffard's Injector.

	Average with Giffard's Injector Feeding Boiler.	Average with Exhaust Injector Feeding Boiler.
Average H.-P. . . .	208·285	241·25
Coal, per H.-P. per hour .	7·08 lbs.	5·13 lbs.
Water ,, ,, . .	7·335 galls.	5·11 galls.
Water used . . .	8986· 75 galls.	8328·75 galls.
Coal	9688 lbs.	8288 lbs.

In the case of coal per horse-power per hour, this shows that the Giffard Injector used 37·9 per cent. more coal than the exhaust-steam injector, and in the case of water per horse-power per hour, that the Giffard Injector used 43·5 per cent. more water than the exhaust-steam injector, the temperature of water delivered being from 170° to 175° Fahr.

This is, however, a special case, the average economy obtained being from 20 to 25 per cent.

The Quantity of Exhaust-Steam used by an Exhaust-Steam Injector per hour, may be found by this formula:—

Let H = the heat absorbed by the water, or the available heat less the temperature of the delivered water.

F = feed-water from tank in lbs. per hour.

G = increase of temperature, or difference between the temperatures of feed-water and delivered water.

X = exhaust-steam in lbs.

Then $X = \dfrac{F \times G}{H}$.

Example: Required the quantity of exhaust-steam in lbs. used by an exhaust-steam injector, supplied with feed-water at 68° Fahr., at the rate of 2280 gallons per hour, delivered by the injector to the boiler at 190° Fahr.

Then the total heat of steam being = 966 latent heat + 212° sensible heat = 1178, H will equal 1178 − 190 = 988° Fahr.

F will = 2280 gallons × 10 lbs. = 22800 lbs.

G will = 190° − 68° = 122° Fahr.

X will = $\dfrac{22800 \text{ lbs.} \times 122°}{988°}$ = 2815 lbs. of exhaust steam used by the injector per hour.

This represents the quantity of exhaust-steam returned to the boiler per hour, and not only is all its heat saved but also $\dfrac{2815}{2240}$ = 1 ton 5 cwt. and 15 lbs. of water. It will be seen from the above calculation, that only 22800 lbs. of water enter the injector from the tank per hour, but 22800 + 2815 = 25615 lbs. of water are fed to the boiler each hour the injector is working.

The Quantity of Water Delivered by an Exhaust-Steam Injector in gallons per hour, is approximately equal to 85 times the square of the diameter of the exhaust-pipe.

Example: The exhaust-pipe of an engine has a branch-pipe 4 inches diameter which it is proposed to connect to an exhaust-steam injector. What quantity of water in gallons per hour will be delivered by the injector?

Then 4 × 4 inches × 85 = 1360 gallons per hour.

Position of the Feed Discharge.—The feed-water should enter a boiler through a perforated pipe, placed above the level of the furnace-crown and a little below the working level of the water. The feed-water, even when heated, is always colder and heavier than the water in the boiler, hence, when introduced at a moderately high level, it descends by gravitation and promotes circulation by displacement. When the feed-water is discharged below the furnace-crown, in case anything gets under the feed back-pressure valve, the water may be forced back through the feed-pipe and valve until the level of the water in the boiler is reduced to that of the feed discharge, thus leaving the furnace-crown bare and exposed to the action of the fire, leading to overheating and collapse. When the feed discharge is below the furnace-tube it produces severe straining at the lower seams of the boiler-shell consequent on that part being kept comparatively cool.

The Feed-Water should be delivered to the boiler at as high a temperature as possible, in order to save fuel, and also to obtain uniformity of temperature throughout the boiler, and prevent wear and tear from the unequal expansion and contraction of the plates.

Cold Feed-Water produces severe Strains on the Plates, as it causes different temperatures in various parts of the boiler, resulting in unequal expansion and contraction; the strain caused in this way may be calculated as follows:—

The extension of a plate per ton of tensile strain upon it is—

A wrought-iron boiler-plate of average quality extends ·000076 of its length per ton of tensile strain.

A mild steel boiler-plate of best quality extends ·000084 of its length per ton of tensile strain.

The expansion of wrought-iron boiler-plates of average quality by heat, is from Table 13 = ·00000658, and of mild steel boiler-plates of best quality is = ·000007 per degree Fahr.

Therefore in cooling wrought-iron plates, every $\frac{\cdot000076}{\cdot00000658} = 11\cdot5$ degrees of difference of temperature produces as much contractile strain as one ton of tensile strain per square inch. And in cooling mild steel boiler plates, every $\frac{\cdot000084}{\cdot000007} = 12$ degrees of difference of temperature will produce as much contractile strain as one ton of tensile strain. If the temperature of the feed-water be 48° Fahr. and the absolute pressure of the steam in the boiler be 75 lbs. per square inch, equal, from Table 78, to a temperature of 308° Fahr., then in each case the strains in tons per square inch will be:—

$\frac{308^\circ - 48^\circ}{11\cdot^\circ 5 \text{ difference}} = 22\cdot6$ tons per square inch strain on the wrought iron plates, and $\frac{308^\circ - 48^\circ}{12^\circ \text{difference}} = 21\cdot6$ tons per square inch strain on the mild steel plates.

The above calculation shows how severely the plates may be strained by pumping cold water into a hot boiler, and the necessity of heating the feed-water to obtain uniformity of temperature.

Feed-water Heaters effect a considerable saving in fuel. It is desirable to heat the feed-water, by utilising either the heat carried off by the waste products of combustion, or the heat carried off by the exhaust-steam, instead of wasting it. In condensing engines the feed-water is usually taken from the hot-well at about 100° Fahr.; for higher temperatures a feed-water heater must be used. In non-condensing engines the feed-water can be effectively heated with the exhaust-steam either by an injection-heater, or by a surface-heater. In locomotives the feed-water may be heated by the exhaust-steam, and a saving effected of $2\frac{1}{2}$ lbs. per train mile in the consumption of coal.

The Economy Effected by Heating the Feed-water of Boilers may be calculated as follows:—If the temperature of the feed-water be 52° Fahr., the heat required to convert 1 lb. of water to steam will be = 1178° — 52° = 1126 units. To raise the temperature of the water to the boiling point, requires 1178° — 212° Fahr. = 966 units of heat; this will effect a saving of 966 ÷ 1126 = ·858, say ·86 or 14 per cent. over cold water. When the feed-water is supplied from the hot-well of a condensing engine, the temperature of the water is usually 100° Fahr., the heat required will be = 1178° — 100° Fahr. = 1078, this will only effect a saving of 1078 ÷ 1126 = ·956 or 4·4 per cent. over cold water. An economiser or water-heater placed in the flue of a boiler and heated by the waste products of combustion, would, if the temperature of the water in passing through the heater were raised to 300° Fahr., effect a saving of 1178° − 300° = 878° ÷ 1126 = ·79 or 21 per cent. over cold water.

Injection-Heaters are those in which the exhaust steam is discharged on the surface of water contained in an open or closed tank; by regulating the influx of cold water the temperature may be maintained at within a few degrees of 212° Fahr., the boiling point. The objection to this form of heater, is that, the exhaust-steam carries with it from the cylinder a quantity of grease, which combines with the carbonate of lime in bad feed-water, and is liable to cause overheating of the plates of the boiler and leaky joints. When a heater of this description is used, the cylinder of the engine should be lubricated only with good mineral oil. In cases where the feed-water is good, and not much inconvenience is caused by grease carried with the exhaust-steam, and where it is an object to save water, this method of heating is efficient. The whole of the heat contained in the steam is imparted to the water, and the water derived from the condensed steam is saved, and added to the supply of feed-water which becomes materially increased in quantity.

The Quantity of Water derived from Condensed Steam may be calculated as follows: To heat 50 gallons of water from 52° Fahr. to 212° Fahr. will require 212° − 52° = 160° × 50 gallons × 10 lbs. = 80000 units of heat, to obtain which, 80000 ÷ 966° = 82·8 lbs. of steam or 82·8 ÷ 10 lbs. = 8·28 gallons will be added to the water, and supplied from the exhaust-steam.

The Noise caused by the Injection of Steam into Water may be silenced by placing a wire-gauze nozzle full of beads or small marbles at the end of the steam-pipe.

A Water Jacket on the Exhaust-pipe is another method of heating the feed-water, the water being forced through the jacket by the feed-pump. The objection to this heater is that it does not allow sufficient time for the water to become heated, or for the impurities to deposit, and the outside of the exhaust-pipe soon becomes coated with incrustation, which diminishes the efficiency of the apparatus.

In a Coil or Worm-heater, the water is forced from the pump to the boiler through a coil of pipes, or worm, enclosed in a casing through which the exhaust-steam passes. The objections to this form of heater are, that it affords no facilities for depositing the impurities of the feed-water; it does not allow sufficient time for the water to become properly heated; the friction of the water in passing through the coil increases the wear and tear of the pumps; and the pipe is liable to become furred up and the passage of the water prevented.

Feed-water Heaters having parallel tubes fixed at both ends, are sometimes used. The objection to this form of heater is that the unequal expansion and contraction of the tubes due to the varying temperatures of the water, causes leakage.

Construction of Feed-water Heaters.—A feed-water heater should be large enough, to permit the water to be heated to occupy at least 20 minutes in passing through the heater, in order to afford time for the impurities or sediment to become deposited, because water cannot be purified instantaneously. It should be deep enough to prevent the sediment being agitated and carried into the boiler; the more impure the water is, the greater should be the depth. The feed-water should not come in contact with the exhaust-steam, but the steam should pass through tubes around which the feed-water should be made to pass. There should be no place in the tubes in which water from condensation can lodge, otherwise it will cause back-pressure and loss of power; this is best effected by bending the tubes in the form of ∩. Both ends of the tubes should be fixed in the same plate, in order to allow the tubes to expand and contract freely; and the water should either be drawn or forced through the heater. The tubes should be made of brass, and their area should be considerably in excess of that of the exhaust-pipe of the engine to which the heater is applied, otherwise it will cause back-pressure in the cylinder. The difference between the rate of expansion of brass and scale, prevents incrustation adhering to the tubes, and their heating-surface is maintained clean and effective. An efficient feed-water heater of this description is shown in Fig. 104; it will raise the temperature of feed-water to about 208° Fahr.

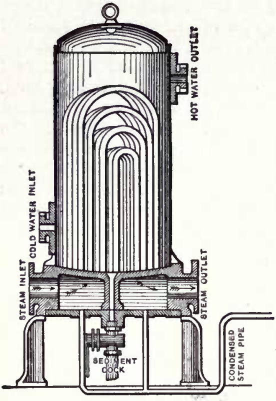


Fig. 104.—Feed-water heater.

The Area of the Tubes of a Feed-water Heater should not be less than that found by the following Rule: Multiply the square of the diameter of the exhaust-pipe of the engine, for which the heater is intended, by 1·25; the product will be the aggregate area of the aperture of tubes.

Proportions of Feed-water Heaters and Purifiers.—The following proportions of feed-water heaters have been found to answer well in practice. The proportions are much larger than actually required to heat the water, in order to allow time for purifying the feed-water in its passage through the heater.

TABLE 75.—PROPORTIONS OF FEED-WATER HEATERS.

Diameter of exhaust pipe of engine in inches	2	2½	3	4	5	6	8	10	12
Diameter of shell of heater in inches	14	16	21	24	28	30	39	45	52
Height of shell in feet . . .	4	4½	5	6	6½	7	8	10	12
Area of the aperture of the tubes in square inches . . .	5	8	12	20	33	46	80	130	180

A Feed-water Heater when used for a Marine Boiler is usually placed in the base of the funnel. The feed-water is pumped through tubes heated by the waste products of combustion, the temperature of the water being raised to the extent of from 25° to 40° with a natural draught. When a feed-water heater of this description is used with a natural draught, it is liable to obstruct and lessen the draught, and lower the evaporative power of the boiler; hence, feed-water heating can only be efficiently accomplished in marine boilers when forced draught is used for combustion.

The Principal Scale-forming Substances in Feed-water are, carbonate of lime, sulphate of lime or gypsum, and magnesia. They are soluble in cold water and in water of a moderately high temperature, but they are insoluble at a temperature of 303° Fahr., corresponding to an absolute steam pressure of 70 lbs. per square inch, as will be seen from the following Table.

Table 76.—SOLUBILITIES OF SCALE-FORMING SUBSTANCES.

Description of Substance.	Soluble in parts of Pure Water at 32° Fahr.	Soluble in parts of Pure Water at 212° Fahr.	Insoluble in Water at Degrees Fahr.
Carbonate of lime . .	62514	62485	303
Sulphate of lime . .	510	465	303
Carbonate of magnesia .	5532	9623	303

The scale-forming substances are precipitated by heating and evaporating the water. In a boiler, part of the precipitation is deposited as mud and part settles on the hot plates and forms scale, which is most objectionable when it forms on the furnace and tubes, because it resists the passage of heat to the water, and seriously detracts from the heat-transmitting power of the plates. The heat-conducting power of scale, being only about one-thirtieth of that of metal, it keeps the water from contact with the plates and causes over-heating. When water contains much scale-forming substances, the entrances to valve-boxes, valves and cocks become furred and choked, and all mountings are liable to being rendered inoperative, to leakage and breakage.

A thin coating of scale, not exceeding $\frac{1}{32}$ inch in thickness, is not injurious, and in the case of boilers fed with corrosive waters may serve to protect the plates from wasting, but scale beyond that thickness should be removed; when the scale is very thick it requires to be chipped off.

Prevention of Scale.—Chemical compounds poured into a boiler, are of no use in either removing scale or preventing its formation, and many of them are injurious to the plates. The mineral matter forming scale is first precipitated in a boiler in the form of powder, or sludge, and it should be removed before it has time to deposit on the plates and harden to form scale; this may be effected by partial blowing-off. The formation of scale may be prevented by blowing-off the water from stationary boilers, for a few minutes before stirring up the fires in the morning and before banking them up at night, with the pressure of the steam at about 5 lbs. per square inch. The best means of preventing scale is to purify the water before it enters the boiler.

Eucalyptus or Blue-Gum Leaves possess the properties of purifying bad water, and preventing and removing scale. Boilers may be cleaned by using a fluid extract from these leaves, it precipitates the salts and impurities held in solution in the water, which are deposited in the form of mud, and may be got rid of by blowing-off. It also slowly softens hard scale, and enables it to be easily removed from the plates; and it leaves a vegetable-deposit on the metal which prevents corrosion and pitting.

Hard Water may be Softened and incrustation prevented by the addition of pure caustic soda to the water, previously to its entering the boiler. Bad water may be purified and softened sufficiently for use in steam-boilers, by keeping it at a high temperature for some time before it is forced into the boiler. By this means the mud and impurities mechanically suspended in the water, and a great portion of the impurities chemically held in solution, may be got rid of. When water contains a large quantity of lime, it should, after being heated in one vessel, run through another containing broken slag, on which the lime will be deposited.

Boilers should be frequently Cleaned by washing out when cold, with water of heavy pressure, to detach mud and scale.

Feed-Water may be Purified by one of the following processes recommended by Dr. Angus Smith:—

Chalk Waters are best treated by Clark's process; that is, by caustic lime.

Mixed Chalk and Gypsum Waters can be precipitated completely by caustic soda.

Gypsum Waters may be precipitated by carbonate of soda, with the addition of a minute quantity of caustic soda.

These precipitations should be made in a separate tank, the pure water alone entering the boiler.

Water containing Carbonate of Lime only.—Treat 1000 gallons as follows:—For every grain of carbonate of lime per gallon = 1000 grains per 1000 gallons, use 1060 grains of carbonate of soda, made caustic with 560 grains of burnt lime.

Water containing Sulphate of Lime.—Treat 1000 gallons as follows:—For every grain of sulphate of lime per gallon = 1000 grains per 1000 gallons, use 779·4 grains of carbonate of soda.

Water containing both Carbonate and Sulphate of Lime, treat as follows:—For every grain of carbonate of lime per gallon, treat according to previous instructions for carbonate of lime. When the sulphate of lime does not exceed 8 to 6 of carbonate, neglect it entirely; beyond that quantity, treat according to previous instructions for sulphate of lime.

Acid Waters.—Add carbonate of soda or an alkali to the water. A degree of acidity is the same as the amount of carbonate of soda required to neutralise it; therefore, for every degree of acidity add one grain of carbonate of soda per gallon. For ·10° add ·10 of carbonate of soda per gallon, made caustic or otherwise. Carbonate of soda is preferable, the precipitation to be made in a separate vessel; the acid may be removed in this way, and the gypsum decomposed according to the previous instructions.

*** For further information on feed-water, see the author's work "Steam-Boiler Construction," published by Crosby Lockwood & Son, London.

CONSTRUCTION OF STEAM-BOILERS.

The Cornish Boiler, shown in Figs. 105 and 106, has the shell-plates arranged in parallel belts, alternately outside and inside, the fibre of the metal running circumferentially. The longitudinal seams break-joint, and are double-riveted: the circumferential seams are single-riveted. The

Fig. 105.

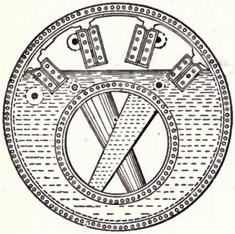
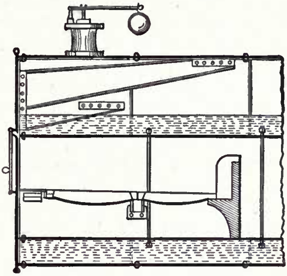


Fig. 106.



Figs. 105 and 106.—Sections of a Cornish boiler, with Galloway-tubes.

ends are each of one plate. The front-end plate is attached to the shell with outside angle-iron, and the back-end plate with inside angle-iron, the ends being stayed with gusset-stays, as shown. The bottom rivets of the

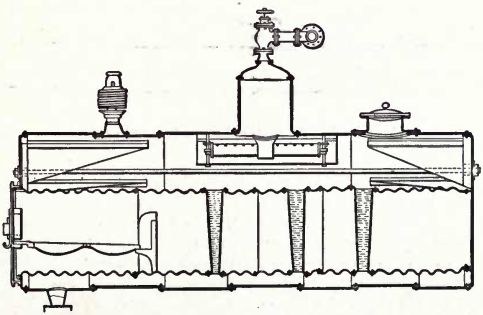


Fig. 107.—Cornish boiler, with corrugated furnace-tube, fitted with Galloway-tubes.

gusset-stays are spaced 10 inches from the rivets of the angle-iron of the furnace-tube, in order to give elasticity to the end-plates, to accommodate the expansion of the furnace-tube. The furnace-tube is fitted with flanged seams, to further accommodate expansion and strengthen the furnace-tube, and with Galloway-tubes to increase the circulation. The strains on a furnace-tube when the end-plates are held too rigidly, are very great, because the tube cannot expand longitudinally, and the expansion is accommodated by the arching or hogging of the tube, which at the same time becomes oval, the top and bottom of the tube being flattened by the strain. This bending action causes weakness in certain lines which are liable to corrode.

In the case of a Cornish boiler 20 feet long, with end-plates too rigidly stayed, which constantly leaked at the rivets of the gusset-stays on the front end-plates and at the seams of the furnace-tube when the steam was up, it was found, by fitting rods through stuffing-boxes on the crown of the shell, that the furnace-tube arched or hogged to the extent of $\frac{1}{2}$ inch at the middle of the tube.

The Cornish boiler, shown in Fig. 107, has a corrugated furnace-tube, fitted with Galloway-tubes as shown. The end-plates are in one piece, each flanged inwards where they are attached to the shell. The front end-plate is flanged outwards, and the back end-plate is flanged inwards, to receive the ends of the furnace-tubes.

Lancashire Boilers, shown in Figs. 108—110, have the end-plates in one piece. The holes for the furnace-tubes are cut out in a lathe. The space between the furnace-tubes is generally about 6 inches at the narrowest

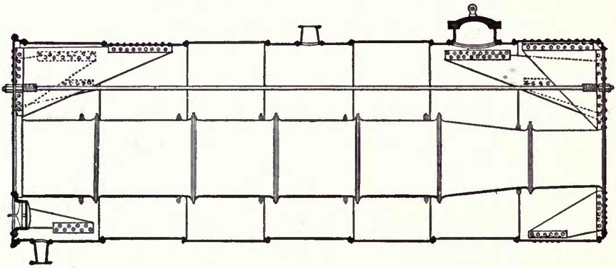


Fig. 108.—Longitudinal section of a Lancashire boiler.

part, and that between the furnace-tube and the boiler-shell about 5 inches. The end-plates and also the angle-iron, or angle-steel, are usually $\frac{1}{16}$ thicker than the plates of the shell. The arrangement of gusset-stays for staying the end-plates to the shell are clearly shown in Figs. 109 and 110. The furnace-tubes are plain; they are united at the joints with Adamson's flanged-seams, which are strong and elastic, and the rivets of the joint are not exposed to the fire. The mouth-piece of the manhole, shown in Fig. 111, is of wrought-iron, double riveted to the shell, the edge of the hole in the plate being provided with a strengthening-ring. The internal diameter of the mouth-piece is 17 inches.

The Furnace-tubes of Lancashire and Cornish Boilers are strengthened with Adamson's flanged-seam, shown in Fig. 112; with expansion-hoops, shown in Fig. 113; and with angle-iron hoops or strengthening-rings, with a water space underneath, as shown in Fig. 114.

Cornish and Lancashire boilers are good steam producers, and burn the

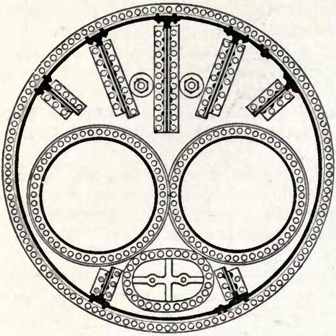


Fig. 109.—Section showing the front end-plate and gusset-stays of a Lancashire boiler.

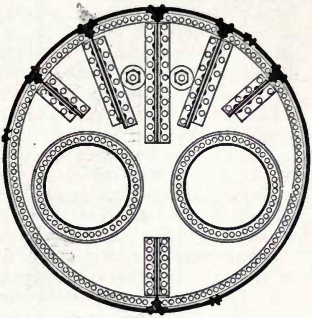


Fig. 110.—Section showing the back end-plate and gusset-stays of a Lancashire boiler.

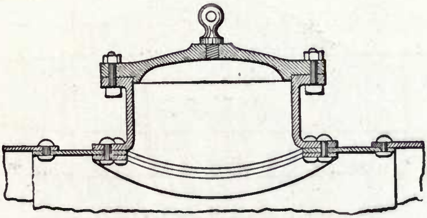


Fig. 111.—Manhole, with strengthening-ring.

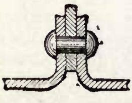


Fig. 112.—Adamson's flanged-seam.

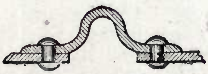


Fig. 113.—Expansion-hoop.

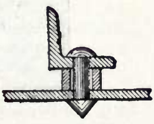


Fig. 114.—Angle-iron hoop.

commonest description of coal. They are usually fired with common slack, the consumption of which in well-arranged boilers does not exceed 3 lbs. per indicated horse-power, and is frequently considerably less.

The Method of Setting Cornish and Lancashire Boilers is shown in Fig. 115. The flues are 9 inches wide at the top, and ample room is provided for the proper cleaning of the flues, and for the external examina-

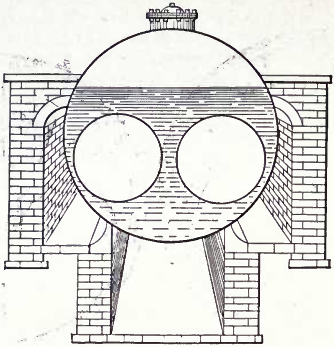


Fig. 115.—Method of setting Cornish and Lancashire boilers.

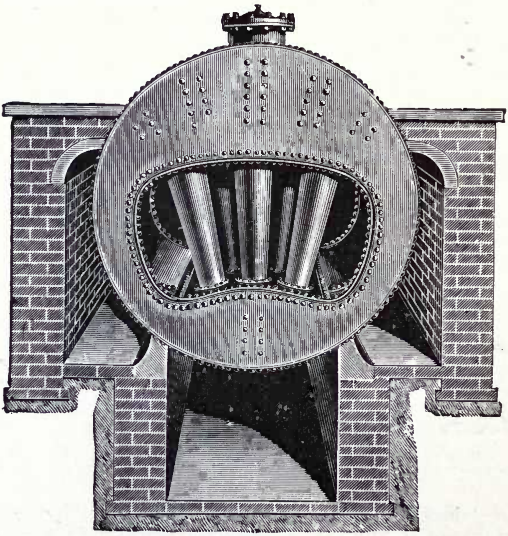


Fig. 116.—Method of setting Galloway boilers.

tion of the plates. The flues are lined with fire-brick, set in fire-clay, no mortar being used where it would come in contact with the plates. The boiler is set on fire-clay blocks, having narrow bearing surfaces in contact with the plates, to prevent water from leakage, lodging, and corroding the plates. The front end-plate is placed well above the floor, to prevent corrosion from damp or leakage: and the front wall is set back from the end-plate, so as to clear the angle-iron, in order that defects, or corrosion may not be concealed. The flame, after leaving the furnace-tube, passes under the bottom of the boiler and returns to the chimney by the side flues.

The Galloway Boiler is a very economical steam boiler. It contains about 15 per cent. more heating surface than the Lancashire type, and its economy of fuel probably accords with this ratio. In a careful test of one of these boilers, 11·72 lbs. of water were evaporated at 212° Fahr. per lb. of fuel, the percentage of water in the steam being only ·57; the cubic feet of water-space per horse-power was 14·10, and the cubic feet of steam-space per horse-power was 4·04. Fig. 116 is a back view of a Galloway boiler, showing the arrangement of the tubes and the manner the boiler is set in brickwork. The flame, after leaving the furnace-tubes, passes first round the sides and then returns underneath the boiler. This method of setting has been found to give the best results with this kind of boiler.

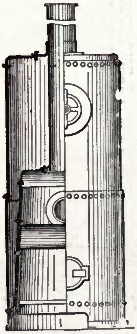


Fig. 117.—Vertical cross-tube boiler.

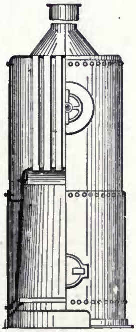


Fig. 118.—Vertical tubular-boiler.

Vertical Cross-tube Boilers, shown in Fig. 117, are very much used for driving small engines, but they are not economical in the consumption of fuel. It is usual to allow about 10 or 11 square feet of heating-surface in these boilers per horse-power.

Vertical Tubular-boilers, shown in Fig. 118, are more economical in the consumption of fuel, and better steam producers than vertical cross-tube boilers. It is usual to allow 15 or 16 square feet of heating-surface per horse-power in these boilers, to compensate for the loss of effective heating surface of that portion of the tubes above the water-line. The uptakes of vertical boilers should be provided with a fire-clay lining to prevent the plates being burnt above the water-line.

A Locomotive-Boiler is shown in Fig. 119. The barrel of the boiler

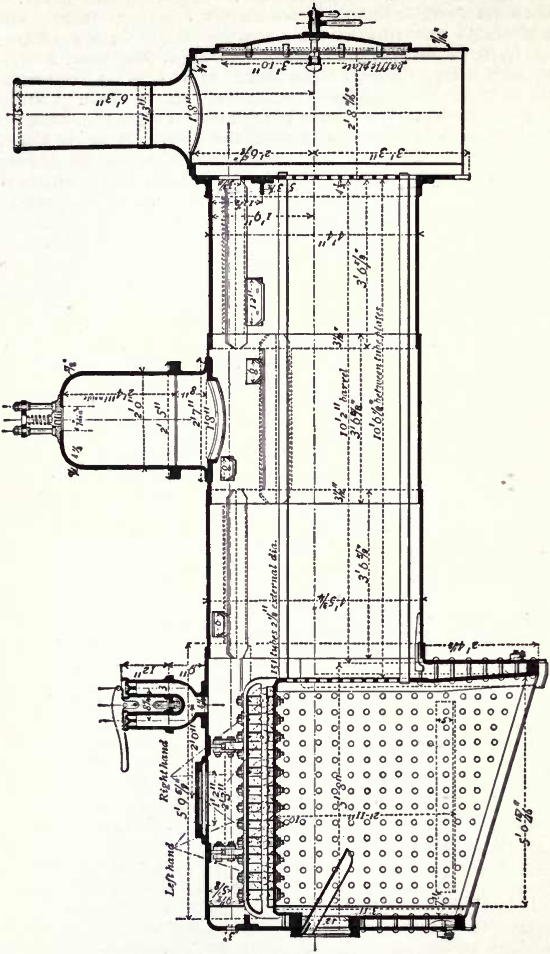


Fig. 119.—Longitudinal section of the boiler of a locomotive-engine.

is made of plates $\frac{7}{16}$ inch thick, in three rings, each of one plate, the internal diameter being 4 feet $3\frac{1}{8}$ inches at smoke-box end, and the length of the barrel 10 feet 2 inches. The longitudinal seams are butt-jointed and double-riveted, with inside and outside butt-strips $7\frac{1}{2}$ inches wide and $\frac{3}{8}$ inches thick; the joints of the rings are above the water-level. The vertical seams have double-riveted lap-joints. The front tube-plate is $\frac{7}{8}$ inch thick. The top and sides of the fire-box shell are in one plate $\frac{1}{2}$ inch thick, the back plates $\frac{1}{2}$ inch thick, and the front plates $\frac{9}{16}$ inch thick; the front plate is flanged and riveted to the barrel of the boiler. The fire-box shell-plates are double riveted at the transverse and longitudinal seams. The front tube-plate and back plate of the fire-box are stayed to the barrel by gusset-stays, and the upper part of the back plate of the fire-

Fig. 120. Fig. 121.

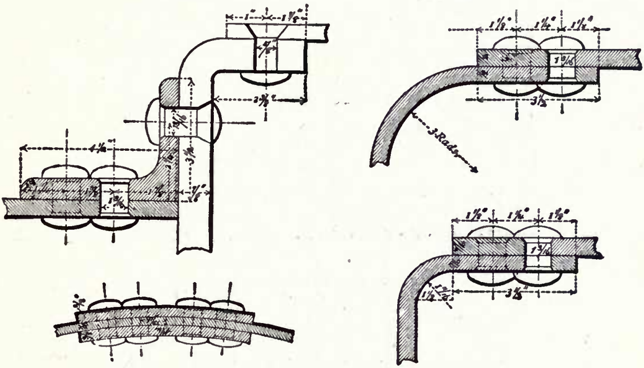


Fig. 122. Fig. 123.

Figs. 120—123.—Sections showing the riveted-joints of the locomotive boiler shown in Fig. 119.

box shell is stayed to the front tube-plate by six longitudinal stays $1\frac{1}{8}$ inch diameter. The fire-box tube-plate is connected to the boiler-barrel with seven palm-stays. The fire-box is made of copper-plates; the tube-plate is $\frac{7}{8}$ inch thick tapering below the tubes to $\frac{1}{2}$ inch thick at the bottom. The top and sides of the fire-box are in one plate $\frac{1}{2}$ inch thick, the lap of plate is $2\frac{1}{2}$ inches; diameter of rivets $\frac{13}{16}$; pitch $1\frac{3}{4}$ inch. The sides, back, and front plates of the fire-box are stayed to the fire-box casing by copper stays $\frac{7}{8}$ inch diameter, screwed 12 threads per inch, and pitched 4 inches centres. The stays are screwed into both plates, and snap-riveted on the outside, and hand-riveted on the inside, the thread being turned off the portion of the stay between the plates. There are 151 brass tubes, 10 feet $8\frac{1}{4}$ inches long, $2\frac{1}{8}$ inches external diameter, No. 12 BWG thick for 1 foot next the fire-box, then tapering to 14 BWG at the other end, the difference of thickness being on the inside of the tube; they are parallel on the outside. Steel ferules $1\frac{3}{4}$ inch long, tapering 1 in 24, are driven into the fire-box end of the tubes. Sections of the riveted-joints are shown in Figs. 120—123.

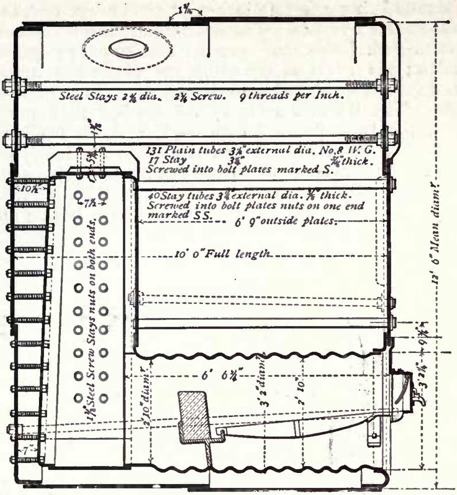


Fig. 124.—Longitudinal section of a single-ended marine boiler, by Mr. J. Dickenson, Sunderland.

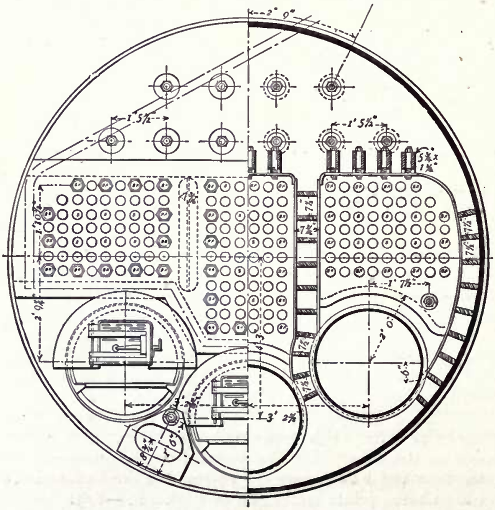


Fig. 125.—Half-end view and cross section of the single-ended marine boiler, shown in Fig. 124.

A Single-ended Marine Boiler is shown in Figs. 124 and 125. The boiler-shell is 12 feet 6 inches mean diameter, and 10 feet long, made of steel-plates, $1\frac{3}{16}$ inch thick. It has 131 plain tubes, $3\frac{1}{4}$ external diameter. No. 8 wire-gauge thick; 17 stay-tubes, $3\frac{1}{4}$ external diameter, and $\frac{5}{16}$ inch thick, screwed into both plates; and 40 stay-tubes, $3\frac{1}{4}$ external diameter, and $\frac{3}{8}$ inch thick, screwed into both plates with nuts on one end. The boiler end-plates are stayed with 10 steel-stays $2\frac{3}{4}$ inches diameter, screw-ends 9 threads per inch, pitched $17\frac{1}{2}$ inches centres, 4 stays in top row and 6 stays in bottom row. The outside rows of the steel-stays at the back of the combustion-chamber are $1\frac{1}{2}$ inch diameter, the remainder being $1\frac{3}{8}$ inch diameter. The stays are screwed 9 threads per inch, the effective diameter being 1·22 inch, and they have nuts at both ends, the pitch of the stays is $6\frac{9}{16} \times 7\frac{1}{2}$ inches. The furnace-tubes are corrugated, the plates are $\frac{1}{2}$ inch thick. The longitudinal seams of the shell are lap-jointed, as shown in Fig. 126, the area of the plate between the rivet-holes is = 79·16 per cent. of that of the solid plate, and the area of the rivets = 87·8 per cent. The total grate-surface of the boiler is 36 square feet, and the total heating-surface 1350 square feet: working pressure 150 lbs. per square inch.

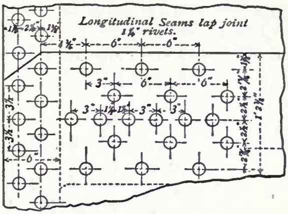


Fig. 126.—Longitudinal seams of the boiler shown in Figs. 124 and 125.

A Double-ended Marine Boiler is shown in Figs. 127 and 128. The

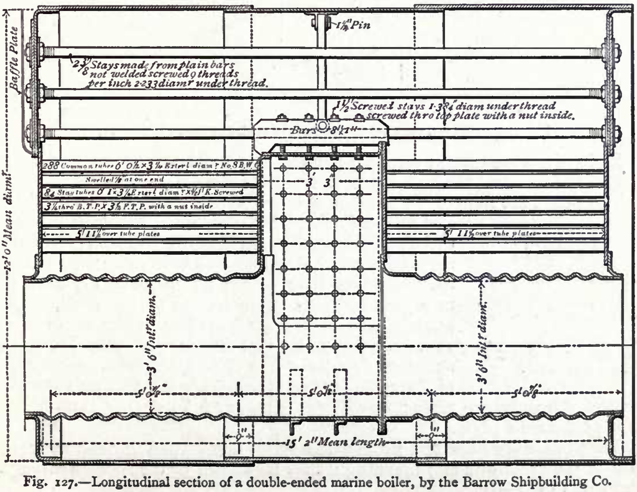


Fig. 127.—Longitudinal section of a double-ended marine boiler, by the Barrow Shipbuilding Co.

boiler is 12 feet mean diameter, and 15 feet 2 inches long, made of steel-plates $1\frac{1}{16}$ inch thick. It has 288 plain tubes 6 feet $\frac{1}{2}$ inch long, $3\frac{1}{4}$ external diameter, No. 8 BWG thick, swelled $\frac{1}{8}$ inch at one end; 84 stay-tubes, 6 feet 1 inch long, and $3\frac{1}{4}$ inches external diameter, $\frac{3}{8}$ inch thick. The length over the tube-plates is 5 feet $11\frac{1}{2}$ inches. The heating-surface of the tubes is 1893·4 square feet; of the furnaces, 162·4 square feet; and of the combustion-chambers, 146·9 square feet; total heating-surface, 2202·7 square feet; fire-grate area = 84 square feet.

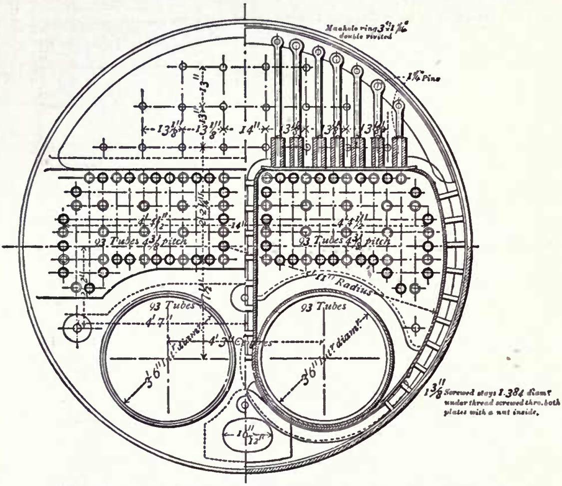


Fig. 128.—Half-end view and cross section of the boiler shown in Fig. 127.

Ratio of grate-surface to heating-surface = 1 : 26·2.

Ratio of flue-area through tubes, to fire-grate surface = 1 : 4·88.

The plates are flanged outwards at one end of the combustion-chambers and one end of the boiler-shell, to enable the boiler to be completed by machine riveting. The crown-stays of the combustion-chambers are supported at the middle of their length by sling-stays connected to a pair of angle irons riveted to the shell. The arrangement of the shell-riveting is shown in Fig. 129. The rivets are of mild steel $1\frac{3}{16}$ inch diameter, and the rivet-holes are drilled in place. The area of the plate between the rivet-holes, and the shearing area of the rivets are respectively 83 per cent., and 88·56 per cent. of the area of the solid plate.

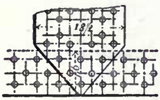


Fig. 129.—Riveted seams of the boiler shown in Figs. 127 and 128.

Fire-box-stays are best made with a small hole drilled through them, so that in the event of a stay breaking it may be known by the leakage through the hole in the stay.

The Marine-Boiler, shown in Figs. 130 and 131, has a shell 9 feet

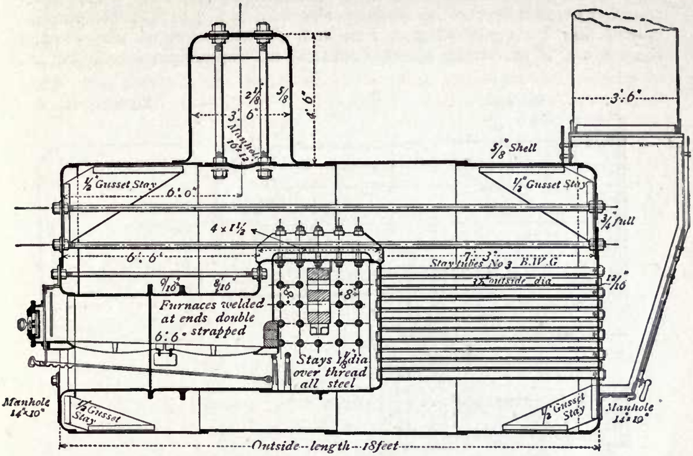


Fig. 130.—Longitudinal section of a marine boiler, by Simons & Co.

mean diameter, and 18 feet long, made of steel plates $\frac{5}{8}$ inch thick. The furnace-tubes have plain plates $\frac{1}{2}$ inch thick, and 3 feet 2 inches internal diameter. The end plates are $\frac{3}{4}$ inch thick, flanged inwards at each end to join the shell. A plan of the riveting of the shell is shown in Fig. 132. The working pressure is 90 lbs. per square inch.

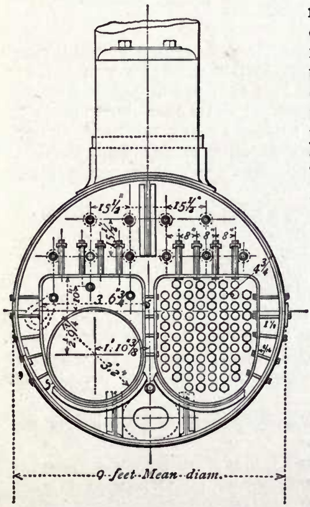


Fig. 131.—Transverse section of the boiler shown in Fig. 130.

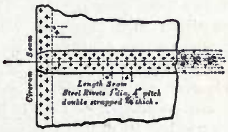


Fig. 132.—Plan of the riveting of the shell of the boiler shown in Figs. 130 and 131.

A Marine Boiler of a Modified Locomotive Type is shown in Figs. 133, 134, it differs from a locomotive boiler in the following respects. The fire-box is formed into a water-bottom from which a steady supply of water is afforded to the whole length of the surface forming the sides of the fire-box, where the evaporation is most intense. The combustion-chamber extends beyond the fire-bars, as shown in Fig 133, for the purpose of assisting the combustion of fuel and the utilisation of the products thereof, and of preventing intense heat from the furnace playing directly on

Fig. 133.

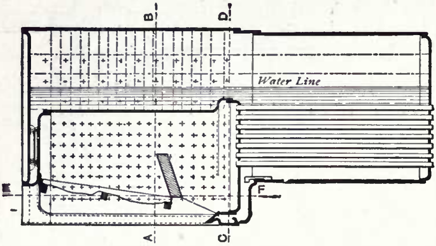
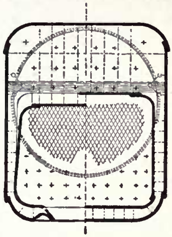


Fig. 134.



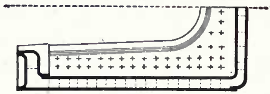


Fig. 135.

Figs. 133—135.—Marine boiler of modified locomotive type, by Hawthorn, Leslie, & Co., Newcastle-on-Tyne.

the tube-plate. Fig. 134 is a half cross section through A B, and through C D in Fig. 133, and Fig. 135 is a half-plan section through E F in Fig. 133.

Blow-off Cocks, shown in section in Fig. 136, and in plan in Fig. 137, should be fitted with a guard, as shown, in order that the spanner for turning the cock cannot be withdrawn without closing the blow-off cock. The elbow-pipe for connecting the blow-off cock to the boiler should be wrought-iron. Blow-off cocks should be taken apart, examined, adjusted, and greased when a boiler is cleaned.

Fusible-Plugs, shown in Fig. 138, are a safeguard against collapse of a furnace-top from over heating, when they are of reliable construction, and in good order and clean on both the fire side and water side of the plate. They require to be frequently renewed, as the nature of the alloy changes with long service. Fusible-plugs should be examined when a boiler is cleaned, and scraped clean on both the fire-side and water-side of the plate. Brass-cones filled with lead are sometimes used for this purpose, but they are not so reliable as those filled with fusible-metal. The following alloys for fusible plugs melt at the temperature of steam of the absolute pressures given below:—

2 Tin :	1 Lead :	melts at	340°F.=	temp.	of steam of	118	lbs.	per sq. in.
10½ ,,	4 ,,	,,	350°F.=	,,	,,	135	,,	,,
17 ,,	4 ,,	,,	370°F.=	,,	,,	174	,,	,,
4 ,,	5 ,,	,,	390°F.=	,,	,,	220	,,	,,
8 ,,	11 ,,	,,	400°F.=	,,	,,	248	,,	,,

Plugs filled with pure tin fuse at 446° Fahr., and those with lead at 620° Fahr.

In getting up Steam the fire should not be lighted until water is visible in the glass of the water-gauge, and the boiler should be gradually warmed. The fire should be fed regularly, and the grate-bars kept evenly covered with a thick fire if there be plenty of steam, and with a thin fire if short of steam. In emptying a boiler, water should not be run off under steam-pressure or when the boiler and seating are hot. In case the boiler should be found to be short of water, if the fire be thin and the furnace-crown is not overheated, the fire should be drawn quickly, beginning at the front, but if the fire be thick or the furnace-crown overheated, the fire should be smothered with either wet slack or wet ashes, and the damper closed.

Fig. 136.

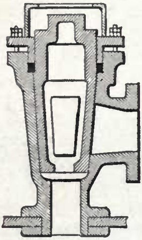
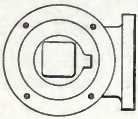


Fig. 137.



Figs. 136 and 137.—Blow-off cock.

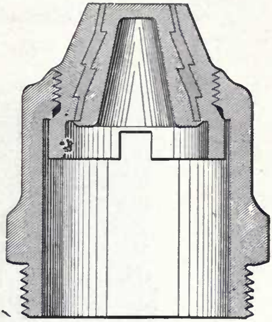


Fig. 138.—Fusible-plug by the National Boiler Insurance Co.

The Time required to Lower the Water in a Boiler, a certain distance when either a fusible-plug or a rivet is blown out, and the quantity of water blown out, may be found by the following *Rules:*

The quantity of water in cubic feet to be lowered or blown off = the surface of the water in feet × the depth in feet.

The quantity of water blown out in one minute = (diameter of hole in inches)^2 × $2\frac{1}{2}$ constant × $\sqrt[3]{}$ pressure.

The time in minutes required to lower the water $= \dfrac{\text{Quantity of water lowered}}{\text{Quantity of water blown out per minute.}}$

Example: A rivet $\frac{3}{4}$ inch diameter was blown out of a boiler in which the pressure of steam shown by the steam-gauge was 64 lbs. per square inch, the depth of water in sight in the water-gauge was 5 inches, the area of the surface of the water was 120 square feet: How long would it take to lower the water 5 inches?

Then the depth of water is 5 inches or $\frac{5}{12}$ feet, $\frac{120 \times 5}{12} = 50$ cubic feet of water to be blown out through the rivet-hole,

and $\cdot75 \times \cdot75$ inch $\times 2\frac{1}{2}$ constant $\times \sqrt[3]{64} = 11\cdot25$ cubic feet of water blown out in one minute.

Then $\dfrac{\text{50 cubic feet the quantity to be lowered or blown out}}{\text{11·25 cubic feet the quantity blown out per minute}} = 4$ minutes 45 seconds.

Water-Tube Boilers are those in which water is contained in a number of tubes of moderate size, the steam being generated from it by heat applied to the external surfaces of the tubes. It is essential that the tubes of such boilers be regularly supplied with good water, and there must be a rapid circulation of the water to carry off the heat absorbed by the tubes, the heating surfaces of which must be kept clean and free from deposits of sediment.

A Water-Tube Boiler of improved construction, with a plain furnace,

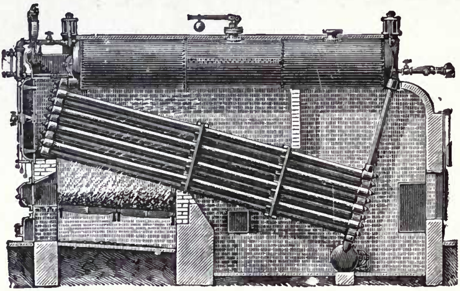


Fig. 139.—Babcock and Wilcox water-tube boiler.

is shown in Fig. 139. This boiler is essentially made of three parts, each connected with the others, viz.:—

(1st.) A series of inclined solid welded water-tubes over the furnace, in which the water being divided into small volumes is quickly raised to a high temperature, and rises through vertical connecting boxes or "headers" at the front end into—

(2nd.) A horizontal steam and water drum, where the steam separates from the water—the remaining body of water returning through the vertical tubes at the back end into the inclined water-tubes, where it is again heated and again passes into the steam and water-drum; thus a continuous and rapid circulation is kept up and a uniform temperature maintained throughout the boiler.

(3rd) A mud-collector is attached to the lowest point of the inclined water-tubes, and into this the matter held in suspension in the water is precipitated by its greater specific gravity as the water falls through the vertical tubes and tube boxes at the rear.

Construction.—The inclined water tubes are formed of solid lap-welded wrought-iron, expanded at each end into hollow vertical connecting boxes, each containing one zigzag row of pipes. These end connecting pieces are provided with hand-holes opposite each water tube for cleaning purposes—the hand-hole covers being faced metal to metal and secured by wrought-iron clamps, thus dispensing with all bolts and perishable material employed in ordinary joints. A large and continuous water-way through all the parts is secured by the end connecting boxes being attached to the horizontal steam and water drum by short tubes expanded into accurately bored holes.

The steam and water-drum is made of best flanging quality of iron or mild steel, and is double-riveted. The mud-collector is made of cast-iron—this metal being found to resist corrosion best—and is provided with ample facilities for cleaning. All the water circulates in one direction, no cross currents obstructing it.

Tests were made of one of these water-tube boilers, erected at Grant's Mills, Ramsbottom, the results of which are given in the following Table:—

Table 77.—Results of Tests of a Babcock and Wilcox Water-Tube Boiler of 136 Nominal Horse-Power.

Particulars of Tests.		Test with Plain Furnace.	Test with Regenerative Furnace.
Heating surface	sq. ft.	1563	1563
Grate surface (6 ft. long × 5 ft. wide)	sq. ft.	30	30
Ratio of heating to grate-surface		1÷52	1÷52
Duration of test		8 hours	8 hours
Average observed steam pressure	lbs.	85	90
Average temperature of water fed to the boiler by injector	deg.	210	208
Pounds of coal fired	lbs.	5264	5712
Pounds of refuse	lbs.	448	672
Pounds of combustible	lbs.	4816	5040
Per cent. of ashes	per cent.	8·4	11·7
Coal consumed per sq. ft. of grate per hour	lbs.	21·9	23·80
Total water evaporated	lbs.	48100	60900
Water evaporated per hour	lbs.	6125	7612
Water evaporated per sq. ft. heating surface per hour	lbs.	3·92	4·86
Water evaporated per lb. of coal—actual conditions	lbs.	9·137	10·661
Water evaporated per lb. of coal, assuming feed-water at 212°, and under atmospheric pressure	lbs.	9·397	11·168
Water evaporated per lb. of combustible—actual conditions	lbs.	9·987	12·083
Water evaporated per lb. of combustible, assuming feed-water at 212°, and under atmospheric pressure	lbs.	10·376	12·651
Quality of steam—percentage of moisture about		1	½
Nominal horse-power (1 H.-P. = 30 lbs. of water evaporated from 212°, at 70 lbs. pressure)	H.-P.	136	136
Horse-power developed, assuming feed-water at 212° and steam pressure at 70 lbs.	H.-P.	205·8	258
Per cent. *above* the nominal capacity	per cent.	51·3	89·7
Temperature of boiler-room	deg.	70°	75°
Temperature of flue gases	about deg.	500°	600°
Force of draught in inches of water	inches	½	½

Nut-coal, costing 8*s*. per ton, was used in the test with the plain furnace. The cost of evaporating 100 gallons of water to steam of 70 lbs. per square inch pressure, was 4·63 pence. A mixture of equal parts of dross, costing 4*s*. 9*d*. per ton, and slack, costing 5*s*. 3*d*. per ton, was used in the test with the regenerative furnace, the cost of evaporating 100 gallons of water to steam of 70 lbs. per square inch pressure was 2·56 pence. The saving, with the regenerative-furnace, over the test with the plain furnace, being $\frac{4{\cdot}63 - 2{\cdot}56 \times 100}{2{\cdot}56} = 81$ per cent The regenerative-furnace consists of a number of fire-clay-blocks, fitted between the water-tubes over the furnace, which absorb a considerable quantity of heat from the fire while it is bright, and part with a portion of such heat to effect ignition of the gases evolved immediately after firing.

CHIMNEYS FOR STEAM BOILERS.

A Chimney requires sufficient area to carry off the noxious gases of combustion, and sufficient height to produce a sufficient flow of air, or draught, to maintain steady and efficient combustion.

The velocity of the hot gases of combustion may be calculated by Peclet's formula, which is—

$$V = 8\left(\frac{461 + t}{523}\right)\sqrt{\frac{H_1}{1 + G\left(\frac{\frac{fl}{d}}{4} + N\right)(1 + at)^2}}$$

where V = the velocity of the hot air or gases in the chimney in feet per second;

l = the whole length of flues and height of chimney in feet;

f = a coefficient of friction equal ·012 for the passage of furnace gases over sooty surfaces;

d = the diameter of round, or side of square flues;

G = co-efficient of resistance for the passage of air through the fire-grate and the layer of fuel above it, which may be taken at 40 for ordinary boiler-furnaces;

N = the number of bends at right angles to the direction of the current.

The Actual Velocity of the Hot Gases in a Chimney is in practice very much less than the theoretical velocity, owing to the friction of winding flues, and to the cooling of a portion of the stream of gases in passing through the flues and chimney. In careful tests of a large number of chimneys, in various parts of this country, the highest velocity of the hot gases in factory chimneys was found to be 36 feet per second, the lowest 3 feet per second, and the mean velocity 12 feet per second. Therefore, in designing factory chimneys, the velocity of the current of hot gases in the chimney may be taken at 12 feet per second.

Size of Chimneys for Factory-Boilers.—The draught power of a chimney varies as the square root of the height. The retarding of the ascending gases by friction may be considered as equivalent to a diminution of the area of the chimney, or to a lining of the chimney by a layer of gas which has no velocity. The thickness of this lining may be assumed to be 2 inches for all chimneys, or to the diminution of the area by the product of the perimeter by 2 inches (neglecting the overlapping of the corners of the lining). In the following formulæ,* let D = diameter, A = area, E = effective area, then—

For square chimneys, $E = D - \frac{8}{12} D = A - \frac{2}{3} \sqrt{A}$.

For round chimneys, $E = \pi \left(D^2 - \frac{8}{12} D \right) = A - \cdot 592 \sqrt{A}$.

For simplifying calculations the coefficient of $\sqrt{A}$ may be taken as ·6 for both square and round chimneys, and the formula becomes $E = A - \cdot 6 \sqrt{A}$. The power varies directly as this effective area E. A chimney 80 ft. high, 42 in. diameter, has been found to be sufficient to cause a rate of combustion of 120 lb. of coal per hour per square foot of area of chimney, or if the grate area is to the chimney area as 8 to 1 a combustion of 15 lb. of coal per square foot of grate per hour. This is a fair practice for a boiler of modern type, in which the flues or tubes are of moderate diameter, gas passages circuitous, and the heating surface extensive in proportion to rate of combustion, so as to cool the chimney gases from 460° to 560°, and produce high economy. A chimney should be proportioned so as to be capable of giving sufficient draught to cause the boiler to develope much more than its nominal power in case of emergencies, or to cause the combustion of 5 lb. of fuel per nominal horse-power of boiler per hour. The 80 ft. by 42 in. chimney, having 9·62 square feet area will cause the combustion of $9 \cdot 62 \times 120 = 1154 \cdot 4$ lb. of coal per hour, or at 5 lb. of coal per horse-power per hour, is rightly proportioned for 231 horse-power of boilers. The power of the chimney varying directly as the effective area E, and as the square root of the height h, the formula for the horse-power of a boiler for a given size of chimney will take the form, H.-P. $= C\, E \sqrt{h}$, in which C is a constant.

For the 80 feet by 42 inch chimney, $E = A - \cdot 6 \sqrt{A} = 7 \cdot 76$ square feet.
$\sqrt{h} = 8 \cdot 944$ feet.

Substituting these values in the formula it becomes $231 = C \times 7 \cdot 76 \times 8 \cdot 944$. Whence, $C = 3 \cdot 33$, and the formula for horse-power is

H.-P. $= 3 \cdot 33\, E \sqrt[2]{h}$, or H.-P. $= 3 \cdot 33\, (A - \cdot 6 \sqrt{A}) \sqrt{h}$.

If the horse-power of boiler is given to find the size of chimney, the height being assumed, $E = \frac{\cdot 3}{\sqrt{h}}$ H.-P.

For round chimneys, diameter of chimney = diameter of E + 4 inches, for square chimney, size of chimney $= \sqrt{E} + 4$ inches.

No allowance has been made in these formulæ for the differences of friction and of rate of cooling of the gases in chimney. Therefore it will

* See a Paper by Mr. W. Kemp, in the "Mechanical Engineer."

be necessary to deduct 15 per cent. from the results obtained by these rules to obtain the correct area and power.

The Area at the Top of a Chimney for factory steam-boilers is frequently made equal to from $\frac{1}{10}$th to $\frac{1}{12}$th the area of fire-grate for chimneys not less than 90 feet high. When half a dozen boilers are working together with one high chimney, the area at the top of the chimney is frequently made equal to from $\frac{1}{15}$th to $\frac{1}{18}$th the total area of the fire-grates.

BOILER-EXPLOSIONS.

Much interesting information on the explosion of steam-boilers has been collected by the chief engineers of boiler assurance companies, whose reports are very valuable, as they reveal the weak points and defects in the construction of steam boilers, and contain practical data given by men of special training and large experience in unravelling the causes of boiler-explosions.

The following information on boiler-explosions comprises extracts from the Reports of Mr. E. B. Marten, chief engineer of the Midland Steam-Boiler Inspection and Assurance Company, and other matter; the appearance presented by the boilers after explosion is accurately shown by woodcuts which make the matter plain and shorten description. It appears from these reports that:—

Marine Boilers have exploded chiefly from corrosion, decay of stays, and accumulations of salt, and scale or incrustation.

Locomotive Boilers have exploded chiefly from grooving or furrowing, or from cracks caused by the movement of the shell, either from the motion of the boiler, or from the strains of varying pressure.

Lancashire Boilers have exploded chiefly from collapse of furnace-tubes, weakness, corrosion, overheating, and improper setting.

Cornish Boilers have exploded chiefly from weakness of the large single furnace-tube, corrosion, and improper setting.

Plain Cylindrical Egg-Ended Boilers have exploded chiefly from overwork, causing frequent repairs over the fire to be necessary. The enormous fire-grate of these boilers allows them to be forced to do twice the work they should do.

Portable-Engine Boilers have exploded chiefly from over-pressure in the hands of inexperienced attendants, and from corrosion.

Vertical Rastrick Furnace-Boilers have exploded chiefly from the injury to the plates by the fierce heat opposite the furnace necks, necessitating frequent repairs and patching: and from corrosion.

Water-Tube Boilers have exploded chiefly from overheating, caused by a deposit of sediment in the water-tubes.

Vertical Cross-Tube Boilers, and Vertical Tubular Boilers with vertical tubes, used for small engines and steam-cranes, usually burst from weakness, either original or from wasting of the plates by corrosion of the fire-box and of the plates underneath the fire-box. This type of boiler is the one which probably explodes the most, and with disastrous results; they require careful periodical inspection. The following particulars of explosions of this class of steam-boiler represent the usual manner of their failure. In an explosion of one of these boilers, shown in Fig. 140, the

Fig. 140.—Explosion of a vertical boiler from corrosion of fire-box.

Fig. 141. Explosion of a vertical boiler from corrosion of fire-box.

Fig. 142.—Explosion of a vertical boiler from corrosion of fire-box.

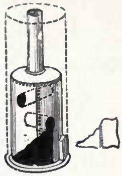


Fig. 143.—Explosion of a vertical boiler from over-pressure.

fire-box was very much corroded on the fire side and three pieces were blown out of the fire-box.

The vertical boiler shown in Fig. 141 exploded from corrosion. The fire-box was much corroded on the fire side and ruptured, a small piece of plate being bent inwards. In the explosion of the vertical boiler shown in Fig. 142 the fire-box was much corroded on the water side and collapsed, a piece being blown out of the fire-box. The explosion of the vertical boiler shown in Fig. 143 was caused by the safety-valves having been made fast, which allowed the pressure to accumulate to more than the boiler could

Fig. 144.

Fig. 145.

Explosion of a vertical boiler from over-pressure.

Fig. 146.—Collapse of the fire-box of a vertical boiler from shortness of water.

Fig. 147.—Collapse of the fire-box of a vertical boiler from corrosion.

bear, a piece being blown out of the fire-box. In the explosion of the vertical boiler shown in Figs. 144 and 145 the safety-valve was defective from the lever being bent, and the pressure accumulated to more than the boiler could bear, especially as the plates were thinned considerably by corrosion on the fire side. In the explosion of the vertical boiler shown in Fig. 146 the fire-box collapsed from shortness of water, the attendant being deceived by the water-gauge, the connecting pipes of which were choked. The explosion of the vertical boiler shown in Fig. 147 was caused by corrosion, the fire-box was much thinned by corrosion on the fire side and collapsed. Weakness, due to the plates of the fire-box being thinned on the fire side by corrosion, has been a fruitful cause of explosion of this type of boiler.

Boiler Explosions may be said to proceed chiefly from the following causes:—Corrosion; grooving or furrowing; overpressure; overheating; injury to the plates from the heat impinging too much on one place; weakness; deterioriation; defects in design by which undue strains are thrown upon the materials; improper setting; inefficient repairs; safety-valves sticking fast and allowing the pressure to accumulate beyond the working pressure; weakening effects of unequal expansion; weakness from wear and tear; defects of workmanship and materials.

Corrosion, both external and internal, by which the plates are gradually

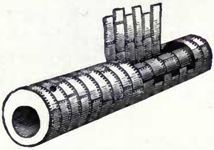


Fig. 148.—Explosion of a Cornish-boiler from corrosion of the shell.

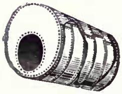


Fig. 149.—Explosion of a Cornish boiler from external corrosion.

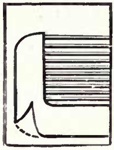


Fig. 150.—Explosion of a marine-boiler from corrosion of the combustion-chamber.

wasted away or thinned, either uniformly or in large or small patches, is a very prevalent cause of boiler explosions. It can only be detected by careful periodical inspection of boilers both inside and outside, and requires

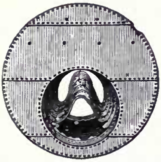


Fig. 151.—Exolosion of a Cornish-boiler from corrosion of the furnace-tube.

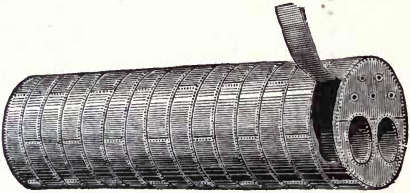


Fig. 152.—Explosion of a Lancashire-boiler from corrosion caused by contact with damp brickwork.

attention even if it be confined to a small patch, as the weakest part of a structure is the measure of its strength.

External Corrosion is frequently caused by leaking seams, as was the case in the boiler shown in Fig. 148, which was much corroded *outside* from leaking seams and *inside* from bad water; the rent ran along longitudinal seams, and the upper part of the shell opened out like a lid. A bad case of external corrosion is shown in Fig. 149, the strength of the shell at the bottom of the boiler had been so reduced by external corrosion, that it ruptured and the rents spread over the whole shell, which was torn into three main pieces. The explosion of the marine-boiler shown in Fig. 150

was caused by corrosion. The bottom of the combustion chamber was so much corroded on the fire side, that it ruptured and part was forced upwards.

External corrosion weakens the furnace-tubes of boilers. This was the cause of the explosion of the boiler shown in Fig. 151, the lower part of the furnace-tube of which was so much thinned and weakened by external corrosion, that it collapsed upwards.

External corrosion may arise from a damp situation. It is frequently caused by contact with damp brickwork. This was the cause of the explosion of the boiler shown in Fig. 152, which gave way at the back end on the right-hand side, where the last plate was very much corroded from damp brickwork, the rent extended across this plate, and then up the seams on each side, for a length of 4 feet 5 inches, so that the plate opened out like a trap-door. The front end-plate and bottom of the front end of the shell is frequently corroded from the injurious practice of slacking ashes on the floor-plates. The bottom of the shell is frequently corroded by leakage from the joint of the blow-off pipe, and the blow-off pipe sometimes fractures from being wasted away by corrosion.

Improper Setting frequently promotes external corrosion, as was the case with the exploded boiler shown in Fig. 153, which was externally

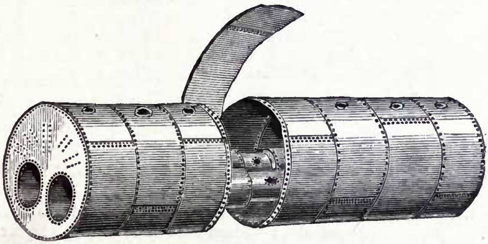


Fig. 153.—Explosion of a Lancashire-boiler from external corrosion, due to improper setting.

corroded where it rested on a centre wall or mid-feather. It gave way at the bottom, and the fifth belt of plates was nearly torn out. Boilers should not be set in this way, because water from leaking seams or other sources settles on the mid-feather, and rapidly corrodes the plates. Fractured plates and leaking seams have frequently been produced by strains caused by the weight of the boiler not being evenly distributed along its seating, due to part of the foundation settling, and from want of care in replacing the seating after repairs.

Internal Corrosion may be produced by bad feed-water, such as is obtained from wells in the neighbourhood of chemical works, or from mines or deep wells. These and other waters, and those impregnated with sewage, frequently contain free acids which rapidly corrode the boiler-plates. The action of these acids can in many cases be neutralized by the proper addition of soda. The feed-water used in the exploded boiler shown in

Fig. 154, was very corrosive, and the furnace-tube was rapidly reduced in strength by corrosion and collapsed downwards. In some cases the furnace-tube collapses upwards, as was the case in the boiler explosion shown in Fig. 155, which being much corroded internally, the back-end of the furnace-tube was so much weakened that it collapsed upwards for one-half of its length.

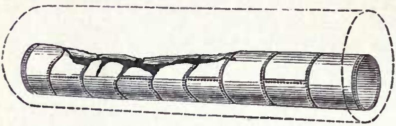


Fig. 154.—Explosion of a Cornish-boiler from collapse of furnace-tube, from corrosion due to bad feed-water.

Zinc is the Best Remedy for Internal Corrosion, a galvanic action being induced by the contact of zinc with the plates of the boiler. Zinc slabs, 12 inches long, 6 inches wide, and $\frac{1}{2}$ inch thick, are suspended in convenient parts of the boiler, one slab being used per 20 indicated horse-power, or 1 square foot of slab to 2 square feet of fire-grate surface. Philip's method of applying zinc for this purpose, consists of a number of

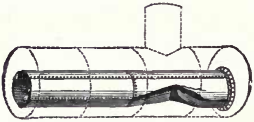


Fig. 155.—Explosion of a Cornish-boiler from collapse of the furnace-tube, due to internal corrosion.

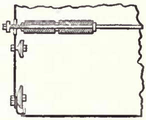


Fig. 156.—Philip's method of applying zinc for the prevention of corrosion in steam-boilers.

small discs of zinc fixed inside the boiler. Each disc or plate of zinc is attached to a stud about 4 inches long, projecting from the plates of the boiler, as shown in Fig. 156: zinc-sleeves are also attached to the longitudinal stays. The proportions are 1 square foot of zinc to 50 square feet of boiler-plate-surface below the water-level.

Zinc has also been used in steam-boilers as a disincrustant. Scrolls of sheet-zinc are said to be more efficacious than zinc-blocks. The sediment in boilers using zinc contains a considerable quantity of that metal. In one boiler employing this means of preventing incrustation, which was fed with water containing a large quantity of lime, the deposit collected when the boiler was cleaned was found to be of the following composition:—

Zinc-oxide	37·15
Peroxide of iron	·35
Lime	20·66
Magnesia	2·36
Sulphuric acid	31·38
Silica	1·65
Carbonic acid	6·45
	100

The Electrogen is said to be very effective in preventing internal corrosion. It consists of a ball of zinc with a copper conductor cast through its centre, the copper being so amalgamated with the zinc at the junction of the two metals as to form brass, so that corrosion cannot form between them to stop the galvanic current. The electrogen is fixed inside the boiler, a wire from each end of the conductor being brazed to the plates of the boiler, a constant galvanic current being kept up, by which the interior of the boiler is protected from corrosion, so long as the zinc lasts.

Pitted Plates should be scraped and cleaned with a strong solution of soda to remove grease and acids, and then covered with a thin coat of portland cement to fill up the pit-holes, by which means further wasting of the plate may generally be prevented.

Grooving or Furrowing is caused by the bending backwards and forwards of the plates of a boiler, either from expansion and contraction, or from alternations of temperature and pressure. It is often induced by caulking, which cuts through the skin of the plate, and promotes corrosion. When the end-plates of a cylindrical boiler are too rigidly stayed, it causes grooving either on the end-plates round the edges of the angle-iron, or flanges by which the furnace-tubes are attached to the end-plates, or at the root of the angle-iron, or corner of the flange if the furnace-tubes are flanged. When Cornish and Lancashire boilers have not been well set, or when the draught passed from the flue-tubes first along the side flues and then under the boiler, instead of passing first under the bottom of the boiler and last along the side-flues, the edges of the ring-seams at the lower part of the shell have been frequently found to be deeply grooved. Internal grooving is frequently found along the edge of longitudinal seams in the barrel of a locomotive-boiler, as shown in Fig. 157, when formed with an overlap joint, which is caused by the tendency of the barrel to assume a perfectly circular form under pressure. This defect may be prevented by forming the seams with double butt-joints, as shown in Fig. 158, which permit the barrel to be made perfectly circular. Many locomotive-boilers have exploded from corrosive grooving, as was the case with the boiler shown in Fig. 159, which gave way at a longitudinal seam, deeply grooved or furrowed from the strain of being worked habitually at more pressure than it was able to bear safely.

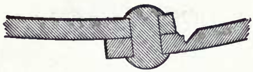


Fig. 157.—Corrosive grooving.

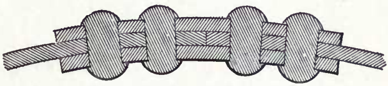


Fig. 158.—Double butt rivetted-joint.

The locomotive-boiler shown in Fig. 160, gave way where it was grooved, or channelled by internal corrosion or furrowing, resulting from corrosion in a line of strain from continued bending of the boiler-plate backwards and forwards.

Overheating may arise from shortness of water; defective circulation; a deposit of salt; attachment of scale, which is a bad conductor of heat and prevents the water reaching the plates to carry off the heat; a soapy deposit

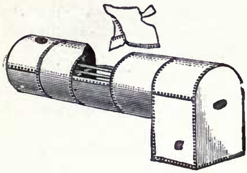


Fig. 159.—Explosion of a locomotive-boiler from corrosive-grooving.

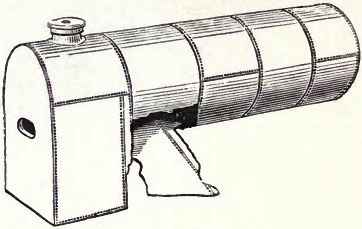


Fig. 160.—Explosion of a locomotive-boiler from corrosive-grooving.

of soda and grease; a greasy deposit from a boiler composition. It may be due to the use of excessively thick furnace-plates; badly arranged strengthening rings; and to the impingement of flame against a double thickness of plate. Many cases of overheating have been caused by grease in the feed-water mixing with deposit in the boiler, and so thickening the water as to offer great resistance to the transmission of heat and cause the furnace crown-plates to become hot, and bulge down over the fire. In many other cases the furnace crown-plates have become hot, strained and bulged down from an accumulation of muddy or greasy deposit from boiler compositions, the plates having become lined with a glutinous coating, which prevented the water reaching the plates to carry off the heat.

The Pouring of Cold Water into a Hot Boiler will not cause an explosion, because a large quantity of steam cannot be generated by throwing water on to a hot plate, owing to the low specific heat of the material, which cannot retain sufficient heat to generate much steam. Therefore the pressure of steam cannot be sufficiently augmented by turning the feed-water suddenly into an overheated boiler, to cause an explosion; but the plates may be seriously injured in this way, by the strain caused by sudden contraction after excessive expansion, and seam-rips are sometimes caused by the sudden contraction of the plates on filling the boiler with cold water while the bottom is hot after emptying. Explosions from overheating are generally caused by the plates softening and rupturing at the ordinary working-pressure of the steam.

When a Boiler Explodes from Overheating, the crown-plates of the furnace-tube, having become hot and weakened, are forced inwards towards the furnace, and they usually bulge down, collapse, and rupture over the fire, as shown in Fig. 161. In other cases the top of the furnace-tube has bulged down and collapsed nearly its entire length, as shown in Fig. 162, from the weakening of the plates through shortness of water. The explosion of a marine-boiler caused by a deposit of salt is shown in Fig. 163. The boiler was worked with salt water, and, for want of sufficient knowledge, it was not noticed that the salt had accumulated and caused collapse of the crown of the furnace-tube. These three examples present the usual appear-

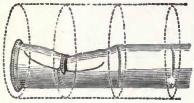


Fig. 161.—Explosion of a boiler from collapse of the furnace-tube from over-heating, due to shortness of water.

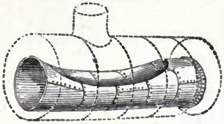


Fig. 162.—Explosion of a Cornish-boiler from collapse of the furnace-tube from overheating, due to shortness of water.

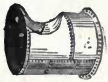


Fig. 163.—Collapsed furnace-tube of a marine-boiler which exploded from a deposit of salt.

ance of a boiler furnace-tube collapsed from overheating of the plates from deposit and shortness of water.

Furnace-Plates in Fracturing from Overheating assume certain colours which denote the temperatures at which they were fractured. The fracture of the overheated plates becomes the following colours at the respective temperatures given:—Bright yellow at 440° Fahr., orange at 470°, red at 510°, violet at 530°, blue at 560°, green at 630° Fahr., and a dull bluish-grey colour at higher temperatures. The fracture of iron at temperatures between 212° and 360° Fahr. is of a clear, bright, whitish-grey colour, but these temperatures do not weaken iron.

Shortness of Water may be due to inattention to water-gauges; to the feed-valve sticking fast or being out of order; to failure in the supply of feed-water; to excessive priming; to leakage from fractured plates and pipes; and to neglecting to shut the blow-off cock. Shortness of water was the cause of the explosion of the marine boiler shown in Fig. 164. The top of the combustion-chamber was so much weakened in this way, that it partially collapsed.

Weak Furnace-Tubes have been the cause of many boiler explosions. The furnace-tubes of most old boilers are weaker than the shells, their col-

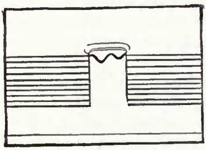


Fig. 164.—Explosion of a marine-boiler from collapse of combustion-chamber, due to shortness of water.

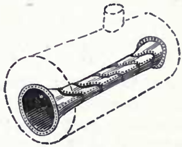


Fig. 165.—Explosion of a Cornish-boiler from a weak furnace-tube and overpressure.

lapsing pressure being generally much less than the bursting-pressure of the shells. A weak furnace-tube was the cause of the explosion of the Cornish boiler shown in Fig. 165, the furnace-tube of which, being without strengthening-rings, and much wasted by corrosion, collapsed in the manner shown, from over-pressure.

Over-pressure, relative to the strength of a boiler, may be due to either the safety-valve being overweighted, or to its sticking fast and being inoperative; to water freezing in the escape-pipe when the safety-valve is bonneted, or to an accumulation of water in the escape-pipe; to fixing the working-pressure of the steam too high for the condition of the boiler; to overrating the strength of the boiler; to ignorance of the quality of the plates; and to the omission of making proper allowance for diminution of the strength of the plates from wear and tear.

When a Weak Furnace-Tube fails from Over-pressure it usually collapses from end to end, and the bottom of the furnace-tube generally comes up to meet the top of the tube, that is, the tube collapses downwards at the crown and upwards at the underside at one and the same time. Weakness and over-pressure were the cause of an explosion of a Cornish boiler, the collapsed furnace-tube of which is shown in Fig. 166. The

Fig. 166.—Explosion of a Cornish-boiler from a weak furnace tube.

furnace-tube collapsed nearly from end to end, the bottom being forced upwards and ruptured. There were no strengthening rings, and the furnace-tube was so much reduced in thickness by corrosion that it was too weak to bear the ordinary pressure.

When a Weak Boiler-Shell gives way from Over-pressure a rent may be made in the plate, or a piece of the shell may be blown out. Many boilers have been weakened and injured by shocks from the sudden opening and shutting of stop-valves; from severe tensile strains from contraction caused by the sudden impingment of cold water and cold air against hot plates; by unequal contraction caused by introducing the feed-water at too low a point of the boiler; by the injudicious manner in which mountings are fixed on the boiler; by frequent repairs, careless patching, and excessive caulking; by emptying boilers under pressure and cooling them too hastily, and by emptying them while the surrounding brickwork is hot.

Weakness from Wear and Tear has been the cause of numerous boiler-explosions. The defective condition of the boilers being due to their continuing at work too long, and being so much weakened by wear and tear as to be unsuitable for their working pressure, and they burst simply because the plates had worn too thin to any longer sustain the ordinary pressure. The explosions might have been prevented by reducing the working pressure to suit the age and condition of the boilers. Weakness from wear and tear was the cause of the explosion of the boiler shown in Fig. 167, the furnace-tube of which was so much reduced in thickness by long wear, that it was unable to bear the ordinary working pressure any longer, and it collapsed from end to end.

Fractured Plates may be caused by overheating; weakness; brittle-ness; bad workmanship; fatigue from long exposure to strains, due to variations of pressure; unequal expansion of the material in different parts of the boiler; want of freedom to expand and contract; and by sudden contraction, due to the impingement of cold water or cold air on hot plates.

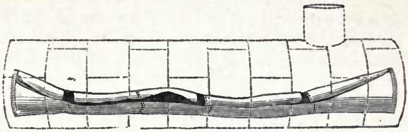


Fig. 167.—Explosion of a Cornish boiler from collapse of the furnace-tube from weakness, due to wear and tear.

Defects in the Design of Boilers which have led to explosions are:—Impeded convection from crowded tubes; defective circulation from crowded tubes; defective circulation from cramped water-spaces; insufficient inclination of water-tubes; omitting to provide holes in the shell, for manholes, domes, mudholes, and mountings, with strengthening rings; oval manholes placed with the longest diameter in the longitudinal direction of the boiler; stays cut away to clear obstructions; the use of cast-iron for mouthpieces and stand-pipes, or seatings for mountings, instead of wrought-iron or steel; omitting to provide for expansion and contraction of the metal by heat; errors in the arrangement of stays; imperfect staying; staying too rigidly; omitting to stay flat surfaces; stiffening flat end-plates instead of staying them; omitting to strengthen furnace-tubes with strengthening-rings and flanged seams; longitudinal seams placed in a continuous line from end to end, instead of being crossed.

Defective Workmanship may be:—Injuring, straining, or fracturing the metal in flanging, dishing, bending, hammering, or punching; burning plates or rivets; injuring or fracturing plates by drifting blind rivet-holes to force them in line; careless caulking; faulty riveting; rivet-holes not fair, causing distorted rivets; defective welding of stays and plates; imperfect threads on bolts and nuts, and in the holes for screwed stays; and rivets not fitting their holes.

Defective Materials may be:—Weak rivets; laminated, blistered or burnt plates; plates of brittle, inferior, or hard quality, so much deficient in ductility, and of such an unyielding nature that they soon suffer under strain, and break instead of elongating as tough plates would do. It is essential that boiler-plates should be ductile, because when a plate is hard and unyielding, each rivet, and the plate between the rivet-holes, has frequently to bear an undue strain, which would be spread over a longer line in a tough plate. A severe strain may be caused by the giving-way of a rivet, or by the plate tearing between the rivet-holes, when the strain would come on the adjacent parts with a jerk. All steam-boilers should be efficiently and periodically inspected by an independent authority.

A Boiler, when not in use, may be preserved from corrosion by drying it thoroughly with brasiers of burning charcoal, and placing pans of quicklime inside, and afterwards closing all openings to prevent the admission of atmospheric air. The quicklime will absorb any moisture remaining in the boiler after it is closed. The boiler should be opened and the quicklime renewed at least twice a year. Another method is to fill the boiler quite full of water, if the water be not of a corrosive nature, and close all openings to prevent the admission of air.

Method of Testing a Steam-Boiler under ordinary working conditions.—Connect the feed-pipe to a tank placed upon a weighing-machine, so that the water entering the boiler may be weighed. With the pressure of the steam and the level of the water in the boiler at the ordinary working-height, and the fires ready for stoking, clear out the clinkers and ashes, and note the condition and level of the fires and the level of the water in the water-gauge. Then commence the test:—stoke the fires from a weighed heap of coal, and shovel into the furnaces all small coals and cinders which fall through the fire-bars during the trial. Regulate the draught by the damper to maintain a nearly uniform working-pressure of steam, without blowing off at the safety valves. Note the pressure of the steam and the temperature of the feed-water every half-hour. Towards the end of the test, allow the fires to burn down until ready for stoking, clear out the clinkers and ashes, and leave the fires as nearly as possible in the same condition at the end of the test as they were at the commencement. Weigh the clinkers and ashes and find the average of the steam pressures, and that of the temperature of the feed-water, and note the quantity of coal and water consumed. The results of the test may be calculated as shown by the following example:—A Lancashire boiler 7 feet diameter, 28 feet long, with two furnace-tubes 33 inches diameter, having 33 square feet of fire-grate surface, was carefully tested. It had heating-surface=internal flues 398 square feet, Galloway tubes 56 square feet, external side flues 270 square feet, bottom 98 square feet, or a total of 822 square feet. The result being that, 2480 lbs. of good small coal evaporated 20832 lbs. of feed-water at 51° Fahr., to steam of 66 lbs. per square inch average pressure, in five hours $= 2480$ lbs. $\div 5 = 496$ lbs. of coal burnt per hour, and 496 lbs. $\div 33$ square feet $= 15$ lbs. of coal burnt per square foot of fire-grate surface per hour. The weight of coal burnt per square foot of boiler heating-surface was $= 496$ lbs. $\div 822$ square feet $= \cdot 63$ lb. per hour. The clinkers and ashes weighed 155 lbs. $= (155 \times 100) \div 2480$ lbs. $= 6 \cdot 25$ per cent. of the coal consumed. The water evaporated was 20832 lbs. $\div 5$ hours $= 4166 \cdot 4$ lbs. per hour $= 4166 \cdot 4$ lbs. $\div 33$ square feet $= 126 \cdot 25$ lbs., or $126 \cdot 25$ lbs, $\div$ 10 lbs. $= 12 \cdot 625$ gallons per square foot of fire-grate surface per hour; and $= 4166 \cdot 4$ lbs. $\div 822$ square feet of heating-surface $= 5 \cdot 07$ lbs. of water per square foot of boiler heating-surface per hour. Then 20832 lbs. of water $\div$ 2480 lbs. of coal $= 8 \cdot 4$ lbs. of water evaporated per lb. of coal. The total heat of steam of $66 + 15 = 81$ lbs. per square inch absolute pressure is, from table 79, $= 1178$ units, and the factor of evaporation is $= (1178 + 32) - 51°$ temperature of feed water $\div 966 = 1 \cdot 2$, and $8 \cdot 4$ lbs. of water $\times 1 \cdot 2$ $= 10 \cdot 08$ lbs. the equivalent quantity of water evaporated from and at 212° Fahr. The heat evolved during combustion was, $= 2480$ lbs. of coal $\times$ $10 \cdot 08$ lbs. of water $\times 966$ heat-units per lb. $= 24148454$ units, and assuming the calorific power of the coal at 14000 thermal units, the efficiency of this boiler is

$$= \frac{\text{Heat utilised}}{\text{Heat supplied}} = \frac{24148454 \text{ units.}}{2480 \text{ lbs.} \times 14000 \text{ units}} = \cdot 69,$$

that is, presuming no water was carried with the steam from priming. By the rule on page 178, the power of this boiler is $= (4166 \cdot 4$ lbs. per hour $\times 1 \cdot 2$ the factor of evaporation) $\div 34 \cdot 5$ lbs. $= 167$ actual horse-power. The nominal horse-power is $= (4166 \cdot 4 \times 1 \cdot 2) \div 62 \cdot 4$ lbs. per cubic foot $= 80$ nominal horse-power.

SECTION III.

STEAM, CONDENSATION, CONDENSERS, AIR-PUMPS, WATER-PUMPS; SLIDE-VALVES, PISTON-VALVES, CORLISS AND OTHER VALVES; LINK-MOTION AND OTHER VALVE-GEARS, ETC.

SECTION III.

STEAM, CONDENSATION, CONDENSERS, AIR-PUMPS, WATER-PUMPS; SLIDE-VALVES, PISTON-VALVES, CORLISS AND OTHER VALVES; LINK-MOTION AND OTHER VALVE-GEARS, ETC.

STEAM: CONDENSATION: CONDENSERS AND AIR-PUMPS: WATER-PUMPS AND TANKS.

The Pressure of Steam is equal in all directions, and it is usual to measure the pressure with reference to that of the atmosphere, which is equal to 14·7 lbs. per square inch of surface, and is the measure of one atmosphere of pressure. Vapours, of which steam is one, do not follow the law peculiar to permanent gases, according to which the volume of a given weight is inversely as the pressure. It has been demonstrated on the contrary, that there exists a constant relation between the pressure, the density, and the temperature of steam; such that the pressure cannot be raised above a given maximum, without, at the same time, a certain elevation of temperature.

Volume and Pressure of Steam.—If the volume be forcibly reduced, and the vapour compressed, without any change of temperature, the compression has not the effect of augmenting the pressure, as would happen if air was similarly treated; it only results in liquefying a portion of the steam, according as the volume is reduced, so that the volume, however reduced, will only contain so much proportionally the less of steam of the original pressure. In order to increase the pressure, the temperature must be raised.

Point of Saturation of Steam.—When the vapour has attained the limit of density and pressure, corresponding to the temperature, the steam is said to be saturated, and it is always in the state of saturation when in contact with water. For one pressure there is one density and one temperature; and the higher the pressure, the greater is the density and the higher is the temperature.

Expansion of Steam.—When a quantity of steam is placed out of contact with water, as in the cylinder of a steam-engine, it may be expanded, and again compressed up to the limit of saturation, and it will follow approximately, though not precisely, the law of Boyle or Mariotte; that is to say, the pressure is nearly in the inverse ratio of the volume, insomuch that when the volume is doubled, the pressure is reduced to about one-half, and when the volume is trebled, the pressure is reduced to about a third.

Superheated Steam.—Superheated steam is amenable to the laws of permanent gases, and behaves as one of them, expanding and contracting in the inverse ratio of the pressure, when the temperature is constant, without the condensation of any portion of it.

Density, Pressure, and Temperature of Steam.—It follows from the above: 1st. That one density and one pressure relative to one temperature are attained in a steam-boiler; these several qualities are in equilibrium, and the steam is in a state of saturation. 2nd. That so long as the state of saturation corresponding to a given temperature is not attained, evaporation continues; and when attained, evaporation ceases. 3rd. If the capacity of the boiler be increased, evaporation is resumed, until the state of saturation is again arrived at. Likewise, if the temperature be increased, evaporation is resumed, and continues till the steam again becomes saturated. 4th. If the temperature falls, the pressure and the density fall also. 5th. If the boiler be closed, and the steam remain at the same temperature, the conditions remain unchanged. But, if an opening be made for the outflow of steam, the pressure will fall, and evaporation will be recommenced, until saturation is re-established. This new generation of steam is very rapid, so much so that the pressure does not sensibly vary between and during the charges of steam taken from the boiler for each stroke of the piston.

Economy in the production of Steam-power increases with the pressure of steam, because the total heat required to generate steam being the same for all pressures, the same quantity of fuel is required to evaporate a given weight of water whether the pressure be high or low, and the higher the pressure the greater the power. The relative value of steam of different pressures when used without expansion varies as the pressure × by the volume. Take for instance steam of 50 lbs. and 150 lbs. per square inch absolute pressure, and assuming an evaporation of 10 lbs. of water per hour per lb. of coal, then 10 lbs. × 8·2 cubic feet per lb. the volume of the steam × 50 lbs. pressure = 4100, and 10 × 3 cubic feet per lb. the volume of the steam × 150 lbs. = 4500, or a gain of $\frac{4500 - 4100 \times 100}{4100}$ = say 10 per cent.: if the steam were used expansively the gain by the higher pressure would be considerably greater, as the available rate of expansion increases with the pressure, and the percentage of back-pressure is less as the total mean-pressure is greater.

Temperature of Saturated Steam.—Steam when in contact with the water producing it, is at the maximum density consistent with that temperature and pressure, and is then called saturated steam, and its temperature is called the maximum temperature of saturation at the given pressure. A certain pressure accompanies a fixed temperature of steam and *vice versâ*, so that one can not increase or decrease without a corresponding change in the other, as will be seen from Table 78.

Table 78.—Temperature and Volume of Saturated Steam from the Experiments of Regnault, Fairbairn, and Tate.

Total Pressure of the Steam in lbs. per Square Inch, including the Pressure of the Atmosphere.	Temperature in degrees, Fahrenheit.	Volume or Number of Cubic Feet of Steam from one Cubic Foot of Water. Water = 1 at 39°.	Total Pressure of the Steam in lbs. per Square Inch, including the Pressure of the Atmosphere.	Temperature in degrees, Fahrenheit.	Volume or Number of Cubic Feet of Steam from one Cubic Foot of Water. Water = 1 at 39°.
	°			°	
1	102	17,985	41	269	614
2	127	10,355	42	271	600
3	142	7,285	43	272	587
4	154	5,610	44	273	574
5	163	4,567	45	275	562
6	171	3,852	46	276	551
7	177	3,332	47	277	539
8	183	2,936	48	279	529
9	189	2,625	49	280	519
10	194	2,375	50	281	509
11	198	2,167	51	283	499
12	202	1,994	52	284	490
13	206	1,846	53	285	482
14	210	1,720	54	286	473
14·7	212	1,642	55	287	465
15	213	1,609	56	288	457
16	217	1,512	57	290	450
17	220	1,427	58	291	443
18	223	1,350	59	292	436
19	226	1,282	60	293	429
20	228	1,220	61	294	422
21	231	1,165	62	295	416
22	234	1,113	63	296	410
23	236	1,067	64	297	404
24	238	1,024	65	298	398
25	240	985	66	299	392
26	243	948	67	300	387
27	245	915	68	301	382
28	247	883	69	302	377
29	249	854	70	303	372
30	251	827	71	304	367
31	253	801	72	305	362
32	254	767	73	306	357
33	256	755	74	307	353
34	258	734	75	308	349
35	260	714	76	309	344
36	261	695	77	310	340
37	263	677	78	310·2	336
38	264	660	79	311	332
39	266	644	80	312	329
40	268	628	81	313	325

Table 78 *continued.*—Temperature and Volume of Saturated Steam.

Total Pressure of the Steam in lbs. per Square Inch, including the Pressure of the Atmosphere.	Temperature in degrees, Fahrenheit.	Volume or Number of Cubic Feet of Steam from one Cubic Foot of Water. Water=1 at 39°.	Total Pressure of the Steam in lbs. per Square Inch, including the Pressure of the Atmosphere.	Temperature in degrees, Fahrenheit.	Volume or Number of Cubic Feet of Steam from one Cubic Foot of Water. Water=1 at 39°.
82	314	321	125	345	220
83	315	318	130	348	212
84	316	314	135	350	206
85	316·2	311	140	353	198
86	317	308	145	356	193
87	318	304	150	359	185
88	319	301	155	362	180
89	320	298	160	364	175
90	320·2	295	165	367	170
91	321	292	170	369	165
92	322	289	175	371	160
93	323	286	180	373	156
94	323·3	284	185	376	152
95	324	281	190	378	149
96	325	278	195	380	145
97	326	276	200	382	142
98	327	273	210	387	136
99	327·2	271	220	390	130
100	328	268	230	394	124
105	332	257	240	398	120
110	335	246	250	402	115
115	338	238	275	410	109
120	342	228	300	418	97

The Properties of Saturated Steam are given in the following table.

Table 79.—Weight, Volume, Total Heat, and Latent Heat of Steam from the Experiments of Regnault, Fairbairn, and Tate.

Total Pressure.		Weight.	Volume.	Units of Heat.	Units of Heat Latent.
In lbs. per Square Inch.	In Inches of Mercury.	In lb. per Cubic Foot.	Cubic Feet per lb.	Total per lb. from 32° Fahr.	Latent Heat per lb.
1	2·037	·0034	289	1113	1044
2	4·074	·0060	166	1121	1027
3	6·111	0086	117	1126	1016
4	8·148	·0112	90	1129	1008
5	10·185	·0137	74	1132	1001
6	12·22	0162	62	1134	996
7	14·26	·0188	54	1136	991
8	16·29	·0213	47	1138	987

Table 79 *continued.*—WEIGHT, VOLUME, ETC., OF STEAM.

TOTAL PRESSURE.		WEIGHT.	VOLUME.	UNITS OF HEAT.	UNITS OF HEAT LATENT.
In lbs. per Square Inch.	In Inches of Mercury.	In lb. per Cubic Foot.	Cubic Feet per lb.	Total per lb. from 32° Fahr.	Latent Heat per lb.
9	18·33	·0238	42	1140	983
10	20·37	·0263	38	1141	979
11	22·41	·0288	35	1143	978
12	24·44	·0313	32	1144	973
13	26·48	·0338	30	1145	971
14	28·52	·0363	28	1146	968
14·7	29·92	·0380	27	1147	966
15	30·55	·0388	26	1147·5	965
16	32·59	·0413	25	1148	963
17	34·63	·0438	23	1149	961
18	36·67	·0463	22	1150	959
19	38·71	·0487	21	1151	958
20	40·74	·0512	20	1152	956
21	42·78	·0536	19	1153	953
22	44·82	·0561	18	1153·5	951
23	46·85	·0586	17	1154	950
24	48·89	·0610	16·5	1155	948
25	50·93	·0634	16	1156	946
26	52·97	·0659	15·2	1156·5	945
27	55·00	·0683	14·7	1157	943
28	57·04	·0707	14·2	1157·5	942
29	59·08	·0731	13·7	1158	940
30	61·11	·0755	13·3	1159	939
31	63·15	·0780	12·9	1159·5	938
32	65·19	·0804	12·5	1160	936
33	67·23	·0828	12·1	1160·5	935
34	69·26	·0850	11·8	1161	934
35	71·30	·0875	11·5	1161·5	933
36	73·34	·0899	11·2	1162	931
37	75·38	·0923	10·9	1162·5	930
38	77·41	·0947	10·6	1163	929
39	79·45	·0971	10·4	1163·5	928
40	81·49	·0994	10·1	1164	927
41	83·52	·1018	9·9	1164·5	926
42	85·56	·1041	9·7	1165	925
43	87·60	·1065	9·4	1165·5	924
44	89·64	·1088	9·2	1166	923
45	91·67	·1112	9·0	1166·5	922
46	93·71	·1135	8·9	1166·8	921
47	95·75	·1158	8·7	1167	920
48	97·98	·1182	8·5	1167·5	919
49	99·82	·1205	8·3	1167·8	918

Table 79 *continued.*—Weight, Volume, etc., of Steam.

Total Pressure.		Weight.	Volume.	Units of Heat.	Units of Heat Latent.
In lbs. per Square Inch.	In Inches of Mercury.	In lb. per Cubic Foot.	Cubic Feet per lb.	Total per lb. from 32° Fahr.	Latent Heat per lb.
50	101·86	·1228	8·2	1168	917
51	103·90	·1251	8·0	1168·5	916
52	105·93	·1274	7·9	1168·8	915
53	107·97	·1297	7·7	1169	914
54	110·01	·1320	7·6	1169·5	913
55	112·04	·1343	7·5	1169·8	912
56	114·08	·1366	7·3	1170	911·5
57	116·12	·1389	7·2	1170·5	911
58	118·16	·1412	7·1	1170·8	910·5
59	120·19	·1434	7·0	1171	910
60	122·23	·1457	6·9	1171·5	909·5
61	124·27	·1479	6·8	1171·8	909
62	126·30	·1502	6·7	1172	908
63	128·34	·1525	6·6	1172·5	907
64	130·38	·1547	6·5	1172·8	906·5
65	132·42	·1570	6·4	1173	906
66	134·45	·1592	6·3	1173·5	905
67	136·49	·1613	6·2	1173·8	904
68	138·53	·1637	6·1	1174	903
69	140·36	·1660	6·0	1174·5	902
70	142·60	·1682	5·95	1174·8	901
71	144·64	·1704	5·90	1175	900·5
72	146·68	·1726	5·81	1175·5	900
73	148·72	·1748	5·73	1175·8	899·5
74	150·75	·1770	5·66	1176	899
75	152·79	·1792	5·60	1176·5	898
76	154·83	·1814	5·52	1176·8	897·5
77	156·86	·1836	5·45	1177	897
78	158·90	·1858	5·40	1177·2	896·5
79	160·94	·1880	5·32	1177·6	896
80	162·98	·1902	5·26	1177·8	895
81	165·01	·1924	5·21	1178	894
82	167·05	·1946	5·15	1178·2	893
83	169·09	·1967	5·10	1178·6	892·5
84	171·12	·1989	5·03	1178·8	892
85	173·16	·2011	4·98	1179·2	891
86	175·20	·2032	4·93	1179·6	890·5
87	177·24	·2054	4·88	1179·8	889
88	179·27	·2075	4·82	1180	888·6
89	181·31	·2097	4·78	1180·2	888·2
90	183·35	·2119	4·73	1180·5	888
91	185·38	·2140	4·68	1180·7	887·5

Table 79 *continued.*—WEIGHT, VOLUME, ETC., OF STEAM.

TOTAL PRESSURE.		WEIGHT.	VOLUME.	UNITS OF HEAT.	UNITS OF HEAT LATENT.
In lbs. per Square Inch.	In Inches of Mercury.	In lb. per Cubic Foot.	Cubic Feet per lb.	Total per lb. from 32° Fahr.	Latent Heat per lb.
92	187·42	·2161	4·63	1180·9	887
93	189·46	·2182	4·59	1181	886·6
94	191·50	·2203	4·54	1181·2	886
95	193·53	·2225	4·50	1181·5	885·7
96	195·57	·2246	4·46	1181·7	885·3
97	197·61	·2268	4·42	1181·9	885
98	199·65	·2288	4·38	1182	884·6
99	201·68	·2309	4·34	1182·2	884
100	203·72	·2330	4·30	1182·5	883·7
101	205·76	·2351	4·26	1182·8	883
102	207·79	·2372	4·22	1183	882·4
103	209·83	·2393	4·18	1183·3	882
104	211·87	·2414	4·15	1183·6	881
105	213·91	·2434	4·11	1183·8	880
106	215·94	·2455	4·08	1184	879·6
107	217·98	·2475	4·04	1184·2	879·5
108	220·02	·2497	4·01	1184·5	879
109	222·05	·2517	3·98	1184·8	878·5
110	224·10	·2538	3·95	1185	878
111	226·13	·2559	3·91	1185·2	877·5
112	228·14	·2571	3·88	1185·5	877
113	230·20	·2600	3·85	1185·7	876·6
114	232·24	·2621	3·82	1185·9	876
115	234·28	·2640	3·79	1186	875·7
120	244·40	·2743	3·65	1186·7	874
125	254·60	·2843	3 52	1187	871
130	264·80	·2942	3·40	1188	869
135	275·00	·3041	3·29	1189	867
140	285·20	·3139	3·19	1190	865
145	295·40	·3236	3·10	1191	863
150	305·60	·3332	3·00	1192	861
155	315·73	·3450	2·91	1192·4	860
160	325·92	·3548	2·83	1193	858
165	336·10	·3674	2·75	1194	857
170	346·29	·3770	2·67	1194·7	855
175	356·48	·3876	2·60	1195	853
180	366·84	·3882	2·52	1196	852
185	376·85	·4092	2·46	1196·5	850
190	387·03	·4178	2·42	1197	849
195	397·22	·4285	2·36	1197·6	847
200	407·40	·4385	2·30	1198	846
210	427·77	·4590	2·20	1200	842

Table 79 *continued.*—WEIGHT, VOLUME, ETC., OF STEAM.

TOTAL PRESSURE.		WEIGHT.	VOLUME.	UNITS OF HEAT.	UNITS OF HEAT LATENT.
In lbs. per Square Inch.	In Inches of Mercury.	In lb. per Cubic Foot.	Cubic Feet per lb.	Total per lb. from 32° Fahr.	Latent Heat per lb.
220	448·14	·4800	2·10	1201	840
230	468·51	·5010	2·00	1202	837
240	488·88	·5200	1·92	1203	834
250	509·25	·5420	1·86	1204	832
275	560·18	·5926	1·68	1206	825
300	611·10	·6439	1·56	1210	820
350	712·95	·7452	1·35	1213	811
400	814·80	·8457	1·20	1218	801

Boiler-Pressure as shown by the Steam-gauge is the pressure of steam above the atmosphere.

The Absolute Pressure of Steam is the total pressure of steam in lbs. per square inch, reckoned from vacuum, or including atmospheric pressure, or the pressure shown by the steam-gauge plus the pressure of the atmosphere.

Initial Pressure is the pressure of steam when admitted to the cylinder at the beginning of the stroke.

Terminal Pressure is the pressure of steam when discharged from the cylinder.

The Flow of Steam through an Orifice with a square edge is 15 per cent. less than through an orifice with a rounded edge.

The Flow of Steam is neither increased nor diminished by reducing the outside pressure below about 58 per cent. of the absolute pressure in the boiler, for example the same weight of steam would flow from a boiler under 100 lbs. pressure into steam of 58 lbs. absolute pressure as into the atmosphere.

The Weight of Steam that will Flow in 70 seconds into a space in which the pressure does not exceed 58 per cent. of the gross pressure of the steam, is equal to the gross pressure of the steam upon an area equal to the area of the orifice. Hence the loss of steam and fuel due to a crack in a cylinder, may easily be calculated.

Example: Steam escapes through a crack, of 1·25 square inches area, in the metal between the exhaust port of the low pressure cylinder of a compound engine and the steam-jacket, which contains steam of 60 lbs. pressure by the gauge. If 1 lb. of coal produces 8 lbs. of steam, what is the loss of steam and coal in 24 hours due to the crack?

Then $60 + 15 = 75$ lbs. per square inch absolute pressure of the steam, and $\frac{75 \times 1·25 \times 60 \text{ minutes} \times 60 \text{ seconds}}{70 \text{ seconds}} = 4821·428$ lbs. of steam lost per hour; and $4821·428 \times 24$ hours $= 115714·272$ lbs. of steam, or $115714·272 \div 2240 = 51·658$ tons of steam lost in 24 hours, and $51·658 \div 8 = 6·457$ tons of coal lost in 24 hours.

The Pressure of Steam when an engine is working at less than its normal speed may be found by the following *Rule*:—

$$\frac{\text{normal pressure} \times (\text{reduced revolutions})^2}{(\text{normal revolutions})^2}$$

Example: An engine makes 62 revolutions, its normal speed when the pressure of steam is 70 lbs. per square inch, the speed becomes reduced to 50 revolutions, what is the pressure of the steam?

Then $\frac{70 \times 50 \times 50}{62 \times 62} = 45.57$ lbs. pressure per square inch.

Superheated Steam is steam which has been isolated from water, and further heated to form gaseous steam or steam-gas. Superheated steam is usually produced by means of a superheater, composed of tubes placed between the boiler and the chimney, and heated by the waste products of combustion which pass between and surround the tubes on their way to the chimney. The steam passes through the superheater before entering the cylinder.

Fig. 167A shows the plan of a superheater attached to the steam-pipes of four marine boilers: A is the superheater, over which stands the funnel; B the uptakes leading the smoke from the smoke-boxes to the chimney; the stop-valves are shown at C.

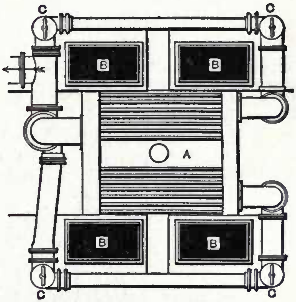


Fig. 167A.—Superheater.

The Limit of Temperature of Superheated Steam is about 380° Fahr.; at higher temperatures the packing of the glands becomes charred, the lubricants are destroyed, and the faces of the valves and cylinder become injured.

Superheated Steam expands nearly at the same rate as a perfect gas, according to the experiments of Fairbairn and Tate, who give the following formula for the expansion of superheated steam:—

Where V = the volume of steam at the temperature t.

V^1 = the volume of steam at the temperature t^1.

$$\frac{V}{V^1} = \frac{438 + t.}{438 + t^1}$$

This formula is a modification of that given below by the same authorities.

The rate of expansion of a perfect gas, R, or the fraction expressing the increment of volume for one degree of temperature Fahr., may be found by the following formula, where t is the temperature of the gas:—

$R = \frac{1}{459 + t}$, thus at 212° Fahr., the rate of expansion of a perfect gas is $\frac{1}{459 + 212^\circ} = \frac{1}{671}$, and at 320° Fahr., it is $\frac{1}{459 + 320} = \frac{1}{779}$.

Superheating prevents partial condensation in the cylinder, and conduces to economy in the consumption of fuel. Mr. Fairbairn* gives the following table of the expansion of superheated steam :—

Table 80.—EXPANSION OF SUPERHEATED STEAM FROM THE EXPERIMENTS OF FAIRBAIRN AND TATE.

Maximum Temperature of Saturation.	Temperatures between which the Expansion is taken.	Co-efficient of Expansion of Steam.	Co-efficient of Expansion of Air.
Degrees Fahr.			
136·77	140 and 170	$\frac{1}{593}$	$\frac{1}{599}$
155·33	160 " 190	$\frac{1}{556}$	$\frac{1}{619}$
159·36	159·36 " 170·2	$\frac{1}{150}$	$\frac{1}{618}$
159·36	170·20 " 209·9	$\frac{1}{624}$	$\frac{1}{629}$
171·48	171·48 " 180	$\frac{1}{200}$	$\frac{1}{630}$
171·48	180·00 " 200	$\frac{1}{604}$	$\frac{1}{639}$
174·92	174·92 " 186	$\frac{1}{190}$	$\frac{1}{634}$
174·92	180·00 " 200	$\frac{1}{637}$	$\frac{1}{639}$
182·30	182·30 " 186	$\frac{1}{230}$	$\frac{1}{641}$
182·30	186 " 209·5	$\frac{1}{630}$	$\frac{1}{645}$
188·30	191 " 211	$\frac{1}{604}$	$\frac{1}{650}$
242·90	243 " 249	$\frac{1}{517}$	$\frac{1}{702}$
255·50	257 " 259	$\frac{1}{392}$	$\frac{1}{716}$
255·50	257 " 264	$\frac{1}{600}$	$\frac{1}{716}$
267·21	268 " 271	$\frac{1}{210}$	$\frac{1}{727}$
267·21	271 " 279	$\frac{1}{640}$	$\frac{1}{730}$
269·20	271 " 273	$\frac{1}{232}$	$\frac{1}{730}$
269·20	273 " 279	$\frac{1}{551}$	$\frac{1}{733}$
279·42	283 " 285	$\frac{1}{298}$	$\frac{1}{742}$
279·42	285 " 289	$\frac{1}{533}$	$\frac{1}{744}$
292·53	297 " 299	$\frac{1}{281}$	$\frac{1}{756}$
292·53	299 " 302	$\frac{1}{638}$	$\frac{1}{758}$

Dr. Siemens found that steam of 212° Fahr., superheated but maintained at atmospheric pressure, augmented rapidly in volume until the temperature rose to 220°, and less rapidly up to 230°, or 18 degrees above saturation-point, from thence it behaved like a permanent gas. Ordinary saturated steam may be made gaseous by superheating it to from 10 to 25 degrees.

A Separator is an apparatus for depriving the steam of any water it may have carried with it from the boiler. In its simplest form, it consists of a cast-iron cylinder, as shown in Fig. 168, having a diaphragm, or partition-plate, extending from the top to a little more than half the depth of the casting. It is connected to the steam-pipe between the boiler and the engine; the steam enters at the top on one side and strikes against the partition-plate, which disengages the particles of water carried from the boiler with the steam which fall to the bottom of the separator. The steam passes under

* See "Mills and Millwork," published by Longmans & Co., London.

the partition-plate, and leaves the vessel at the opposite side to which it entered: a cock is fitted to the bottom of the separator for the purpose of drawing off the water, and a water-gauge is fitted to show the depth of

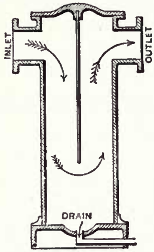


Fig. 168.—Separator.

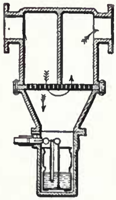


Fig. 169.—Separator.

water. Care must be taken to prevent the water rising too high and closing the passage for the steam.

A Separator of another form is shown in Fig. 169. It has a grating at the bottom, through which the water is drained into a Hanson's steam-trap, by which the water is carried off automatically.

A Separator of improved construction and very efficient in its action, is shown in Fig. 170. The steam traverses a spiral passage, which effects the disengagement of the particles of water carried with the steam. The water falls to the bottom of the vessel, whence it is drained off by a cock; or automatically by connecting the drain-pipe to a steam-trap. The depth of water is shown by a water-gauge.

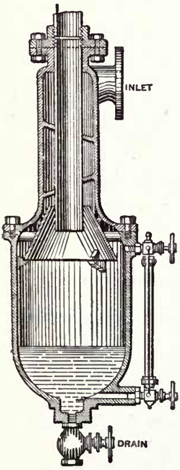


Fig. 170.—Improved separator.

A Steam-Drier, shown in Fig. 171, is an apparatus for evaporating the particles of water which the steam may have carried with it from the boiler. It dries and slightly superheats the steam, and prevents partial condensation in the cylinder. It is fixed between the boiler and the chimney, and is heated by the gaseous products of combustion, part of which pass through the centre tube of the drier. The steam passes through the drier on its way to the cylinder.

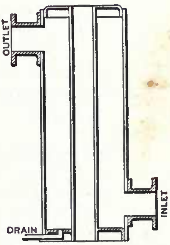


Fig. 171.—Steam-drier.

Condensation of Steam.—The weight of water to be mixed with steam to produce water of a certain temperature may be found as follows:—

Let S = lbs. of steam of a given pressure.
H = the number of units of heat in the steam from 32° Fahr.
W = the weight in lbs. of water per lb. of steam used to condense with.
t = the temperature of the water per lb. of steam used to condense with.
T = the temperature of the water produced.

The steam loses S (H—T) units of heat, and the water gains W (T—t) units of heat.

$$\text{Then } W = \frac{H-T}{T-t}.$$

Example: What weight of water at 50° Fahr. must be mixed with 10 lbs. of steam of atmospheric pressure to produce water at 100° Fahr. The number of units of heat in steam of atmospheric pressure = 1178·6.

Then $\frac{1178 \cdot 6-100}{100-50} = 21 \cdot 57$ lbs. of water for each lb. of steam, and

21.57 × 10 lbs. of steam = 215·7 lbs. total weight of water required to be mixed with that weight of steam to produce water at 100° Fahr.

The Temperature of the Condensed Water may be found as follows:—The notation being the same as in the previous formula. Then

$$T = \frac{1178 \cdot 6 + (W \times t)}{W + 1}.$$

Example: Required the temperature of the water produced from the mixture of water and steam given in last example.

Then $\frac{1178 \cdot 6 + (21 \cdot 57 \times 50^\circ)}{21 \cdot 57 + 1} = 100^\circ$ Fahr., the temperature of the water produced by that mixture of water and steam.

Condensation of Steam.—The following *Rule* is sometimes used for finding the quantity of water required to condense steam.

Let T = the temperature of the steam.
t = the temperature of the water to be produced.
t^1 = the temperature of the water used to condense with.
S = lbs. of steam of a given pressure.
W = the weight of water in lbs. per lb. of steam used to condense with.

$$\text{Then } W = \frac{1000 + (T-t)}{t-t^1}.$$

Example: Taking the particulars from the former example, the temperature of the steam = 212° Fahr. Then this *Rule* will give $\frac{1000 + (212^\circ-100)}{100-50}$ = 22·22 lbs. of water for each lb. of steam, and 22·22 × 10 lbs. of steam, 222·2 lbs. of water required to be mixed with the given weight of steam.

The Temperature of the Water produced may be found by the converse of the previous *Rule.*

Thus $t = \frac{1000 + W \times t + T}{W + 1}$. Taking the particulars from the former example, this *Rule* will give $\frac{1000 + (22{\cdot}22 \times 50) + 212^\circ}{22{\cdot}22 + 1} = 100$ Fahr., the temperature of the water produced by the mixture of water and steam.

Condensation of Steam.—The following *Rule* is sometimes used, although it is not strictly accurate, for finding the weight of water required to condense steam. One lb. of steam when it is condensed, is considered to be capable of raising 1000 lbs. of water 1° Fahr. Let T = the temperature of the water to be produced, t = the temperature of the water used to condense with, W = the weight in lbs. of water per lb. of steam required to condense the steam.

Then $W = \frac{1000}{T - t}$. Taking the particulars from the previous example, this *Rule* will give $\frac{1000}{100 - 50} = 20$ lbs. of water required per lb. of steam, and 20×10 lbs. of steam = 200 lbs. of water to be mixed with the given weight of steam.

The Temperature of the Water produced may be found by the converse of the previous *Rule.*

Thus $T = \frac{1000 + (W \times t)}{W}$. Taking the particulars from the previous example, this *Rule* will give $\frac{1000 + (20 \times 50)}{20} = 100^\circ$ Fahr., the temperature of the water produced by the mixture of water and steam.

Condensation of Steam.—The following formula is sometimes used for finding the weight of water required to condense steam: the notation being the same as in the previous formula.

$W = \frac{1150 - T}{T - t}$. Taking the particulars from the previous example, this *Rule* will give $\frac{1150 - 100}{100 - 50} = 21$ lbs. of water required per lb. of steam, and 21×10 lbs. of steam = 210 lbs. of water to be mixed with the given weight of steam.

The Temperature of the Water produced may be found by the converse of the previous *Rule.*

Thus $T = \frac{1150 + (W \times t)}{W + 1}$. Taking the particulars from the previous example, this *Rule* will give $T = \frac{1150 + (21 \times 50)}{21 + 1} = 100^\circ$ Fahr., the temperature of the water produced by the mixture of water and steam.

Efficiency of Steam in an Engine.—It is not possible to convert all the heat-energy initially in the steam into mechanical energy even in the cylinder of a perfect engine, if such could be made. In a perfect heat-engine, the heat would be supplied at the highest temperature and discharged at the lowest. The zero of Fahrenheit's thermometer is 461 degrees above the point of absolute cold or zero, at which point both the volume and pressure of a gas shrink to nothing. If the highest temperature, or that of the initial absolute pressure of the steam be denoted by T_1, and the lowest temperature, or that of the steam as it enters the condenser or atmosphere, by T_2, the efficiency, E, of a perfect heat-engine would be expressed by the following formula:

$E = \frac{T_1 - T_2}{T_1 + 461}$. This represents the greatest theoretical efficiency which can possibly be obtained in any heat-engine, and it is the standard by which the actual efficiency of an engine should be compared. As an *Example* of the above formula:—Required the efficiency of a perfect condensing steam-engine, using steam of 90 lbs. per square inch initial pressure, or 105 lbs. per square inch absolute pressure, the final pressure or pressure of the steam when it is discharged from the cylinder into the condenser being 2 lbs. per square inch, or 13 lbs. below atmospheric pressure.

Then the temperature of steam of 105 lbs. per square inch absolute pressure is from Table 78 = 332° Fahr.: the temperature of steam of 2 lbs. per square inch absolute pressure is = 127° Fahr., and $\frac{332^\circ - 127}{332 + 461} = \cdot 2585$, or say 26 per cent., the percentage of the total heat in the steam which could be converted into work. It will be seen from Table 79, that each lb. of steam of 105 lbs. absolute pressure per square inch contains 1183·8 units of heat, and as 26 per cent. of the heat would be utilised in a perfect engine, $1183 \cdot 8 \times \cdot 26 = 307 \cdot 788$ units of heat would be converted into work for each lb. of steam used. As one unit of heat is equal to 772 foot lbs. of mechanical energy, $772 \times 307 \cdot 788$ units $= 237612 \cdot 336$ foot lbs. would be obtained per lb. of steam used, equal to $\frac{237612 \cdot 336}{33000} = 7 \cdot 2$ Horse-power.

The Efficiency of an engine relatively to that of a perfect heat-engine, working between the same limits of temperature, may be found by this *Rule:*

$$\text{Efficiency of engine} = \frac{\text{Actual efficiency of the engine.}}{\text{Efficiency of a perfect heat-engine.}}$$

Final Temperature of Steam in a Cylinder or Condenser.—In a non-condensing engine, the exhaust port being open to the atmosphere, there is a back-pressure = 15 lbs. per square inch atmospheric pressure, plus 5 lbs. per square inch, the power necessary to drive the engine against its own friction, and to expel the exhaust steam from the cylinder; so that the lowest terminal absolute pressure is 20 lbs. per square inch, and the lowest temperature 228° Fahr. In a condensing engine there is always vapour in the condenser of at least 3 lbs. pressure per square inch, which, added to the power necessary to drive the engine against its own friction and to the resistance to the escape of the steam from the condenser due to the friction of the exhaust passages, equal to 5 lbs. per square inch, gives 8 lbs. per square inch, the lowest terminal absolute pressure and the lowest temperature is 183° Fahr. Therefore, if a non-condensing engine and a condensing engine were each worked with steam of 89 lbs. per square inch initial absolute pressure, the temperature of which is 320° Fahr., the non-condensing engine would work between the limits of temperature of 320° and 228° Fahr., and the condensing engine between the limits of temperature of 320° and 183° Fahr. The theoretical efficiency of the steam in the condensing engine would be $\frac{320-183}{320-228} = 1{\cdot}48$ times greater than that in the non-condensing engine.

A Vacuum is an empty space, or a space void of all pressure. A vacuum in the condenser of a steam-engine means absence of pressure. The vacuum is never perfect in the condenser, owing to the presence of a small quantity of air and vapour, and it is influenced by the temperature in the condenser. The pressure of vapour at 32° Fahr. is ·089 lb., therefore the vacuum is nearly perfect at that temperature, the usual temperature of the water in the condenser is 100° Fahr., at which the pressure of vapour is ·942 lb., or in round numbers 1 lb. Hence the degree of vacuum rapidly diminishes as the temperature of the water in the condenser is increased, and the temperature of the condenser will always show the state of the vacuum.

The Loss of Vacuum due to an Increase of Temperature in the Hot-well may be calculated by the following *Rule*, where T = the greater temperature and t = the less temperature.

$$\text{Decrease of vacuum} = \frac{(T-t) \times (T-50^\circ) \times (t-50).}{100000}$$

Example: The normal temperature of the water in the hot-well is 103° Fahr., and the vacuum in the condenser is 12 lbs., but owing to an accident the temperature of the water in the hot-well increased to 128° Fahr. What will the decrease of vacuum be and what will the vacuum be now?

Then $\frac{(128-103) \times (128-50) \times (103-50)}{100000} = 1{\cdot}033$ lb. decrease of vacuum, and the vacuum will now be $= 12 - 1{\cdot}033 = 10{\cdot}967$ lbs.

A Vacuum is produced by condensation of steam in the jet-condenser of a condensing engine, into which the steam passes from the engine-cylinder, where it comes in contact with a jet of cold water. The steam is condensed by the cold water, and falls in the form of hot water to the bottom of the condenser, and a vacuum is formed above the water. The water and any air it contains, or which may have entered with it, is pumped out of the condenser by the air-pump, leaving only a slight vapour in the condenser, say, of 3 lbs. pressure, which, deducted from the atmospheric pressure, leaves 12 lbs. as the pressure removed by the condenser, which opposed the advance of the piston, the resistance thus removed being equivalent to an equal amount of steam-pressure on the piston.

The Power obtained by using a Condenser may easily be calculated. For instance, if a non-condensing engine with cylinder 18 inches diameter, length of stroke 3 feet, number of revolutions per minute 50, were converted into a condensing-engine, the power gained by using a condenser would be as follows:—Say the pressure in the condenser is 3 lbs., then 15 lbs. atmospheric pressure $-$ 3 $=$ 12 lbs. vacuum, equivalent to 12 lbs. per square inch steam pressure, and

$$\frac{18 \times 18 \text{ inches} \times \cdot 7854 \times 12 \text{ lbs.} \times (3 \text{ feet} \times 2 \times 50)}{33000} = 27 \cdot 76 \text{ Horse-}$$

power gained by converting the non-condensing engine to a condensing engine.

The Temperature of the Water in the Condenser is usually about 100° Fahr., at this temperature the steam is sufficiently condensed, and a minimum quantity of condensing water is lifted by the air-pump. If the water enters the condenser at a temperature of 50° Fahr. and leaves it at 100° Fahr., out of every unit of water $100^\circ - 50^\circ = 50^\circ$ of cold are available for condensing the steam.

The Quantity of Water required for Condensation in Condensing-Engines, taking the above temperatures, may be found as follows:—Taking the total amount of heat in a given unit of steam at 1178° units Fahr. The heat imparted to each unit of water is $100^\circ - 50^\circ = 50$ units. Of the 1178 units of heat in each unit of steam, it must give up $1178 - 100 = 1078$ units, and the units of water required will be $1078 \div 50 = 21 \cdot 56$. A cubic foot of steam is produced by a cubic inch of water, therefore, each cubic foot of steam will require 21·56 cubic inches of water to condense it. And each cubic inch of water evaporated to steam will require 21·56, or, say, 22 cubic inches of water at 50° Fahr. for condensation, and will form altogether 23 cubic inches of water in the hot-well at 100° Fahr.; in practice rather more than this quantity is required, and it is usual to allow from 27 to 30 cubic inches of injection-water for every cubic inch of water evaporated to steam in the boiler.

The Weight of Injection-Water in lbs. required to Condense 1 lb. of Steam may be found by this *Rule*:—

Condensing water in lbs. $=$

$$\frac{1150 - \text{temperature of hot-well}}{\text{temperature of hot-well} - \text{temperature of injection-water}}.$$

Example: If the temperature of the injection-water be 52° Fahr. and that of the hot-well 100° Fahr., how many lbs. of injection-water will be required to condense 1 lb. of steam?

Then, $\dfrac{1150^\circ - 100^\circ}{100^\circ - 52^\circ} = 21 \cdot 87$ lbs. of injection-water.

The Quantity of Injection-Water in tons is frequently calculated by the following rule, where I. H.-P. $=$ Indicated horse-power—

Tons per day $=$

$$\frac{\text{I. H.-P.} \times \text{lbs. of steam per I. H.-P.} \times 1000 \times 10 \text{ hours.}}{2240 \times (\text{temperature of hot-well} - \text{temperature of injection-water})}.$$

Example: The indicated horse-power of an engine is 800, it uses 20 lbs. of steam per I. H.-P. per hour. If 1 lb. of steam on being condensed gives out sufficient heat to raise 1000 lbs. of water 1° Fahr., and if the temperature of the injection-water be 52° Fahr. and that of the discharge-water 104°, how many tons of injection-water are used in a day of 10 hours?

$$\text{Then, } \frac{800 \text{ H.-P.} \times 20 \text{ lbs. steam} \times 1000 \times 10}{2240 \times (104 - 52)} = 1374 \text{ tons.}$$

The Temperature of the Hot-Well may be calculated from the weight of injection-water used by the following *Rule*:—

$$\frac{1150 + (\text{lbs. of injection-water} \times \text{its temperature})}{\text{lbs. of injection-water} + 1}.$$

Example: If the temperature of the injection-water be 68° Fahr., what will the temperature of the hot-well be if 24 lbs. of water be required to condense 1 lb. of steam?

$$\text{Then, } \frac{1150 + (24 \text{ lbs.} \times 68^\circ)}{24 + 1} = 112^\circ \text{ Fahr., the temperature of the}$$

hot-well.

The Increase in Evaporation due to an Increase in Temperature of the Hot-Well may be found by the following *Rule*:—T = the greater temperature, and t the less temperature of the hot-well, and E the rate of evaporation.

$$\text{Increase of evaporation} = \frac{(1100 + (T - t)) \times E}{1100}.$$

Example: The evaporation was $8\frac{1}{2}$ lbs. of water per lb. of coal, the temperature of the hot-well being 104° Fahr. If the temperature of the hot-well increased to 138° Fahr., what would the evaporation be?

$$\text{Then, } \frac{(1100 + (138 - 104) \times 8{\cdot}5 \text{ lbs.}}{1100} = 8{\cdot}76 \text{ lbs., but the gain in eva-}$$

poration is counterbalanced by the loss of vacuum.

Jet-Condensers.—One method of effecting condensation to produce a vacuum in a condensing-engine is to discharge the exhaust steam from the cylinder into an air-tight vessel, or condenser, where it comes in contact with a jet of cold water, which absorbs the calorific properties of the steam.

A Vertical Jet-Condenser for a beam-engine is shown in Fig. 172. The exhaust-steam passes from the engine-cylinder through the pipe A, to the condenser B; C is a valve for the admission of cold water into the condenser, where it is discharged through a rose at the end of the injection pipe; D D are the foot-valves; E is the passage from the condenser to the air-pump F; G is the air-pump-bucket provided with valves; H is the hot-well into which the air-pump delivers its water and air, through the delivery-valves at the top of the air-pump; I is the suction-pipe of the boiler feed-pump, which takes its water from the hot-well; J is the cold-water pump which delivers cold-water through the pipe K, to the cold-water cistern L, in which the condenser and air-pump are placed. The air-pump is single-acting.

Horizontal Jet-Condensers.—In horizontal engines the condenser, air-pump and hot-well are usually contained in one casting, as shown in Fig. 183. The condensed steam from the jet-condenser is delivered by the air-pump into the hot-well, from which it is discharged into the condensation-reservoir, in land engines, or into the sea, in marine engines.

In marine engines using sea-water for injection, the contents of the hot-well will consist of a mixture of from 27 to 30 parts of sea-water to 1 of pure water derived from the condensed steam. As the heat in steam of high pressure rapidly deposits sulphate of lime, salt-water can only be used for

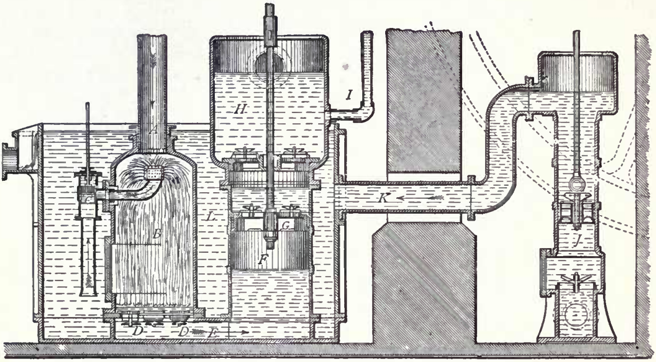


Fig. 172.—Vertical jet-condenser, with air-pump and cold-water pump, for a beam-engine.

producing steam of pressures under 40 lbs. per square inch; hence fresh-water is used for producing steam above that pressure. This is obtained by using surface condensers to condense the exhaust-steam without its coming in contact with water. The water derived from the condensed steam of salt-water contains no salt, and is returned to marine boilers to be re-evaporated to steam and used over and over again, the small quantity lost by leakage being replaced by fresh-water, obtained from distilled salt-water.

The Capacity of the Condenser should equal from 1 to $1\frac{1}{4}$ times that of the air-pump.

The Diameter of the Injection-Pipe may be equal to $\frac{1}{9}$th the diameter of the cylinder for high-speed engines, and $\frac{1}{15}$th the diameter of the cylinder for low speeds.

The Size of Injection Pipe necessary for the supply of a given quantity of condensation-water, making allowance for friction, may be found by this *Rule*:—Divide the quantity of water required in cubic feet per minute, by 1550, the quotient will be the area of the pipe in square feet.

Air-Valve for Condensing-Engines.—When the injection-cock is not closed properly on stopping a condensing-engine, the vacuum may cause water to be drawn into the cylinder and pipes, which would impede the re-starting of the engine. This may be prevented by using the air-valve shown in Fig. 173, the upper part of which, marked A, is connected to the steam-chest, and the lower part is connected to the exhaust-passages from the cylinder to the condenser. So long as the engine is working, the pressure of the steam forces the flexible plate-valve on to the top of the pipe c, and closes its opening. When the steam is shut off on stopping the engine, the pressure is relieved from the valve B, and it is forced open by the spring, the passage c being opened before the engine comes to a stand-still. Air is then admitted through the holes in the lower part of the valve-case to destroy the vacuum and prevent entrance of water.

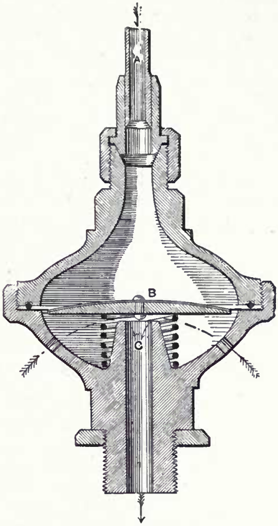


Fig. 173.—Air-valve for condensing-engines.

The Ejector-Condenser, shown in Fig. 174, is another form of jet-condenser. It is used for stationary engines and steam-pumps, and it dispenses with an air-pump. In starting this condenser the central spindle is raised by turning a hand-wheel, and a jet of injection-water is discharged through the centre of the current of exhaust-steam from the engine, which becomes condensed and produces a vacuum within the condenser. The water enters with a velocity due to the difference of pressure between the external atmosphere and the degree of vacuum maintained in the condenser plus the velocity due to the head of the injection-water. The water-jet rushes through the combining nozzle of the ejector and the discharge-tube, and issues into the atmosphere in a continuous stream, carrying with it any air that may be mixed with the exhaust-steam, the action of the ejector being similar to that of an injector. The ejector-condenser is very efficient and rapid in its action, the vacuum being produced within 4 or 5 seconds of the time of opening the injection-cock.

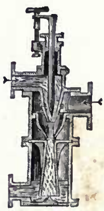


Fig. 174.—Morton's ejector-condenser.

Surface-Condensers.—In surface-condensers the steam is condensed by coming in contact with cold metallic surfaces without coming in contact or mixing with the water used for refrigeration. In some cases the water for refrigeration circulates through tubes, and the steam to be condensed is on the outside: in other cases, the steam passes through tubes, and the water for refrigeration circulates on the outside.

When the steam passes through the tubes any grease which the steam may carry with it from the engine-cylinder is deposited on the front tube-plate and inside the tubes, whence it can be removed by using suitable brushes. When the steam is outside the tubes the grease is deposited on the tubes and casing, and it may be removed by using a solution of soda, or it may be detached by working the condenser at a good heat and washing it out with a jet of water.

The Circulation of the Water through Tubes to condense the steam on the outside is the most efficient way of condensing steam in surface-condensers, because the largest amount of cooling surface is exposed to the hot steam, and the tubes being capable of resisting a greater internal than external pressure, can be made as thin as possible; no scale is deposited on the outside of the tubes which would prevent their being easily drawn when required, and if india-rubber packing be used for the ends of the tubes it is not liable to be injured by grease in the steam. Surface-condensers are frequently fitted with an injection valve and rose, so that in case of accident they may be worked as jet-condensers.

The Weight of Circulating Water passing through a Surface Condenser may be found by this *Rule:*—

$$\text{Weight of water in lbs. passing through the condenser} = \frac{\text{lbs. of water raised 1 degree in temperature}}{\text{temperature of discharged water} - \text{temperature of circulating water}}.$$

Example: If 1 lb. of steam raise the temperature of 1000 lbs. of water 1 degree, and the temperature of the circulating water is 52° Fahr., and that of the discharged water 104° Fahr., how many lbs. of circulating water pass through the condenser per lb. of steam?

Then, $1000 \text{ lbs.} \div (104^\circ - 52^\circ) = 19{\cdot}24$ lbs. of circulating water.

Cooling Surface of Surface-Condensers.—The rapidity of condensation by cold metallic-surfaces depends upon the efficiency and rapidity of the circulation of the cooling fluid. Peclet found that water was about 10 times more efficient as a cooling fluid than air, and that with refrigerating water at an initial temperature of from 68° to 77° Fahr. one square foot of copper-plate would condense $21\frac{1}{2}$ lbs. of steam per hour. Joule found that with a rapid circulation of the cooling water 100 lbs. of steam might be condensed per hour per square foot of surface.

The cooling surface in square feet required per pound of steam condensed per hour, may be found by dividing the weight of steam condensed per hour by 11. For instance, a steam engine of 1400 indicated horse-power using 17 lbs. of steam per indicated horse-power per hour, requires a surface-condenser with $(17 \text{ lbs.} \div 11) \times 1400 = 2163$ square feet of cooling surface.

The Quantity of Steam Condensed by coming in contact with the cold surface of flat plates of metal $\frac{1}{8}$ inch thick, kept cool by flowing water at a temperature of 50° Fahr., expressed in pounds of steam per square foot of surface, has been found by experiment to be as follows:—

	lbs.		lbs.
Cast-iron plates condense	41	Gun-metal plates condense	26
Wrought-iron plates condense	39	Phosphor-bronze plates condense	25
Steel-plates condense	38	Copper-plates condense	24
White-metal plates condense	32	Tin-plates condense	22
Brass plates condense	27	Glass-plates condense	40
Bell-metal plates condense	26	Tiles condense	38

The temperature of the steam was 228° Fahr. corresponding to 20 lbs.

pressure per square inch. Each pound of steam condensed by the above plates represents 1152 thermal units.

In a test of the Kirkcaldy Surface-Condenser 128·34 lbs. of steam were condensed per square foot of tube-surface per hour, the temperature of the circulating water was 40° Fahr., temperature of steam, 295° Fahr., and the temperature of the water from the condensed steam was 70° Fahr. The tubes were corrugated in the direction of their length, they were measured as if plain and circular; this makes a difference of 10 per cent., so that the actual quantity of water condensed per square foot of surface was 115 lbs.

The weight of steam condensed in surface-condensers depends upon the efficiency of the condenser. It considerably increases with the velocity of the water circulating through the tubes. The rate of condensation in some surface-condensers is from 10 to 12 pounds; but in others it is only from $3\frac{1}{2}$ to 5 lbs. of steam per square foot of the cooling-surface of the tubes per hour.

Horizontal surface-condensers are generally from 20 to 25 per cent. more efficient than vertical surface-condensers. Surface-condensers are generally so arranged that the hottest water meets the hottest steam.

The Relation of the Cooling-surface of the Surface-condensers of Marine Engines to the Heating-surface of the Steam Boilers varies considerably in practice, as will be seen from the following table.

TABLE 81.—SHOWING THE COOLING-SURFACE OF THE SURFACE-CONDENSER OF TRIPLE EXPANSION ENGINES AND THE HEATING-SURFACE OF THE BOILERS OF A NUMBER OF STEAMSHIPS.

Indicated Horse-power of the Engines.	Cooling-surface of Surface Condenser.	Heating-surface of the Steam Boilers	Diameter of the Cylinders of the Triple Expansion Engines.			Length of Stroke.
I.H.P.	Square feet.	Square feet.	Inches.	Inches.	Inches.	Inches.
130	209	350	8 +	13 +	21	15
500	1,016	1,318	14 +	22 +	36	24
645	1,360	3,160	22 +	34 +	57	39
1,000	1,280	2,800	19 +	30 +	49	33
1,100	1,700	3,000	22 +	35 +	57	54
1,300	1,977	4,320	23 +	37 +	60	42
1,500	1,600	4,125	20 +	31 +	52	36
1,700	2,800	6,160	27 +	45 +	71	48
2,000	2,552	8,418	24 +	37 +	55	19
2,500	4,000	6,250	22 +	34 +	51	21
2,700	4,300	7,715	29 +	46 +	74	51
4,000	6,045	11,380	36 +	56 +	90	60
5,000	8,000	11,100	31 +	45 +	68	33
6,000	6,750	9,900	40 +	59 +	88	51
7,000	9,500	13,800	42 +	59 +	92	42
7,500	11,550	17,650	40 +	66 +	100	72
9,000	14,500	20,174	40 +	59 +	88	51
10,000	14,600	25,920	42 +	62 +	92	46
11,000	18,500	31,025	43 +	62 +	96	51
16,000	10,300	47,000	44 +	67 +	106	63
20,000	33,000	50,250	45 +	71 +	113	60

The Cooling-Surface per Indicated Horse-power usually provided in surface-condensers is from 2 to $2\frac{1}{2}$ square feet of tube-surface; but it is frequently less, and varies considerably in practice, as may be seen from Table 82, page 271, which contains the cooling-surface, and the quantity of cooling water provided in a number of marine engines.

The Quantity of Water required for Efficient Cooling in Surface-Condensers is one-half more than that required for injection water in jet condensers: the quantity of water required may be found as follows:—The heat given up by each pound of steam condensed was found in the case of jet condensers to be 1078 units. If the temperature of the circulating water be increased 21 degrees in passing through the condenser, then $1078 \div 21 = 51{\cdot}3$ lbs. of circulating water will be required per lb. of steam. It is usual to provide from 40 to 50 lbs. of cooling water per pound of steam supplied to surface-condensers.

The Velocity of the Passage of the Cooling Water through the Surface-Condenser may be found by the following formula, where

V = the velocity of the cooling water in feet per minute.
L = the length of the tubes in feet.
T = number of times the water circulates through the tubes.
P = pounds of cold water per indicated horse-power.
D = the internal diameter of the tubes in inches.
S = square feet of tube-surface per indicated horse-power.

$$V = \frac{L\,T\,P}{80\,D\,S}.$$

Example: The internal diameter of the tubes of a surface-condenser is $\frac{3}{4}$ inch: the tubes are 6 feet long, there are $2\frac{1}{2}$ square feet of tube-surface per indicated horse-power, the water circulates three times through the tubes, the weight of cooling water per indicated horse-power is 800 lbs. Required the velocity of the passage of the cooling-water—

$$\text{Then } \frac{6 \text{ feet} \times 3 \text{ times} \times 800 \text{ lbs.}}{80 \times {\cdot}75 \times 2{\cdot}5} = 96 \text{ feet per minute.}$$

The Capacity of a Circulating-Pump for circulating the water in a surface-condenser is generally equal to from ·5 to ·67 of the capacity of the air-pump when single-acting, and one-half that size when double-acting. A circulating-pump, together with air-pump and feed-pump, is shown in Fig. 175: the barrel is lined with gun-metal, and the pump-rod is of iron cased with brass; the plunger and valve-seats are of brass; the valves are of india-rubber, made of best caoutchouc, with no other ingredients than sulphur and white oxide of zinc, and capable of enduring a test with dry-heat of 270° F., and with moist-heat of 320° F.

Condenser-Tubes are made as small as possible in diameter, and they are placed as closely together as possible in order to obtain the greatest amount of cooling surface in the smallest space. The external diameter of the tubes is usually $\frac{1}{2}$ inch for very small condensers: $\frac{5}{8}$ inch for small: $\frac{3}{4}$ inch for medium size: and 1 inch for large condensers. The thickness of the tubes varies from 20 to 16 I W G, or from ·036 to ·064 inch. The unsupported length of tube should not much exceed 100 times the external diameter of the tube: a supporting-plate for the tubes is frequently fitted in the middle of the condenser.

The Pitch of Surface-Condenser Tubes is generally from $1\frac{1}{2}$ to $1\frac{3}{4}$ times the external diameter of the tube; the tubes are usually placed horizontally, and arranged in zigzag rows, so that the tubes in one row divide the spaces in the next row, as shown in Figs. 177 and 181.

Brass Tubes are the best for Condensers, copper tubes do not answer for this purpose, because the acids produced by grease, carried with the steam from the cylinders, dissolve copper and produce soluble salts of that metal which are carried into the boiler with the feed-water and injure the boiler-plates. The tubes are sometimes tinned as a protection against corrosion. The condenser tube-plates and all bolts and screws should be made of brass containing not less than 70 per cent. of copper in its composition: all the joints between the cylinder, condenser, air-pump, and framing, should be faced and firmly secured. Directing-plates should be fitted to cause the circulating-water to pass over the whole of the tube-surface, and provision should be made for examining and cleaning the interior of the condenser. The thickness of the tube-plates varies from 1·25 to 1·75 times the external diameter of the tube, according to the description of packing used for the tubes.

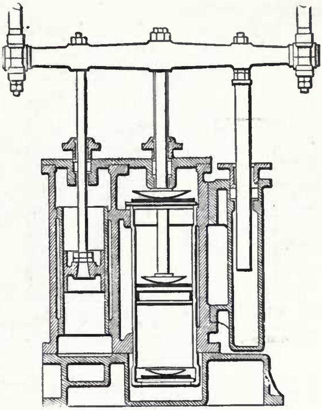


Fig. 175.—Vertical circulating-pump, air-pump, and feed-pump.

Condenser - Tubes are usually composed of 70 per cent. of best copper and 30 per cent. of fine spelter.

Packings for Condenser-Tubes. — Wood-ferrules, shown in Figs. 176 and 177, form cheap and efficient packings for condenser-tubes. They are made of well-seasoned soft wood, such as willow or pine, and are made about an eighth of an inch larger in diameter than the holes in which they have to fit, and are compressed in a die to the proper size for driving into their holes. When wet they expand and fit tightly round the tube and in the tube-plate holes.

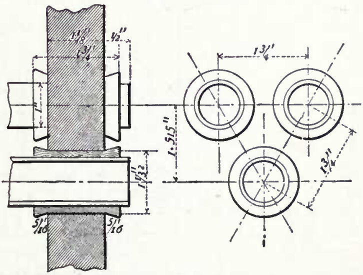


Fig. 176. Fig. 177.
Condenser-tubes packed with wood-ferrules.

A hydraulic-leather packing for tubes, shown in Fig. 178, can only be used when the tubes are placed horizontally, and where the water-pressure is inside the tubes: the pressure of the water expands the leather-washer and makes a tight joint.

An indiarubber packing for condenser-tubes is shown in Fig. 179. The rings are driven into a recess in the tube-plate, this arrangement may be used when the water circulates through the tubes, as the packing-rings are then pressed into their place by the water, and they are not exposed to the action of grease.

A stuffing-box and gland packed with tape or cord for each tube,

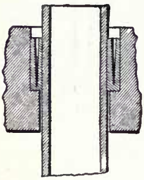


Fig. 178.—Hydraulic-leather packing for condenser-tubes.

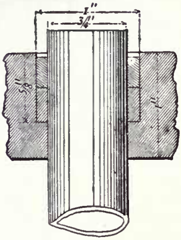


Fig. 179.—Indiarubber-packing for condenser-tubes.

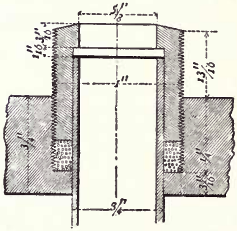


Fig. 180.—Stuffing-box and gland packing for condenser-tubes.

shown in Fig. 180, is the best method of packing condenser-tubes: the water may circulate either inside or outside the tubes, and the packing is not affected by heat. A plan of the tube-plate showing the arrangement of the tubes with these stuffing-boxes is shown in Fig. 181.

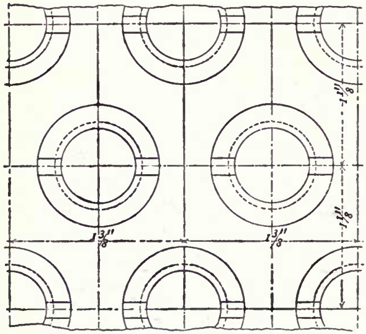


Fig. 181.—Tube-plate with stuffing-box and gland-packing.

A Single-acting Vertical Air-pump draws the air and water from the condenser only in its upward stroke, and the air and water pass through valves in the pump-bucket during its downward stroke, and are discharged into the hot-well.

Vertical air-pumps are usually single-acting, having valves in the buckets, and they are more efficient than horizontal air-pumps with solid plungers. A vertical air-pump and surface-condenser of a marine-engine are shown in Fig. 182.

A Double-acting Air-pump draws the air and water from the condenser at each stroke, and has no valve in its plunger or piston. It is usually placed horizontally, and has a set of inlet and delivery-valves at each end of the pump-barrel; a double-acting air-pump for a stationary engine is shown at Fig. 183. The direction of the flow of water is con-

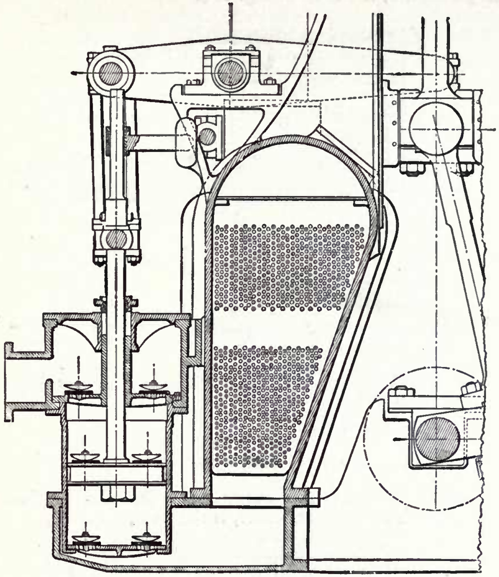


Fig. 182.—Vertical air-pump and surface-condenser of a marine engine.

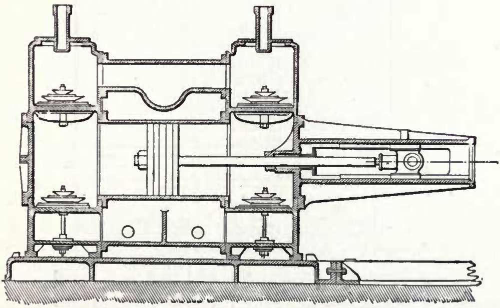


Fig. 183.—Double-acting horizontal air-pump.

tinually changing in this class of pump, which impairs its efficiency, for which reason horizontal air-pumps are frequently made single-acting.

A Horizontal Plunger-Air-pump, with Jet-Condenser, for a Stationary Engine, is shown in Fig. 184.

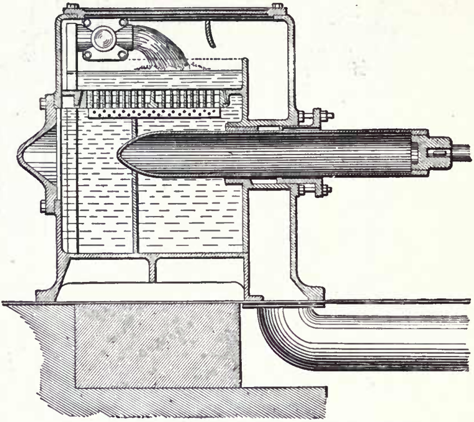


Fig. 184.—Horizontal plunger-air-pump for a stationary engine.

A Horizontal Plunger-Air-pump for a Marine-Engine, together with feed-pump and surface-condenser, is shown in Fig. 185.

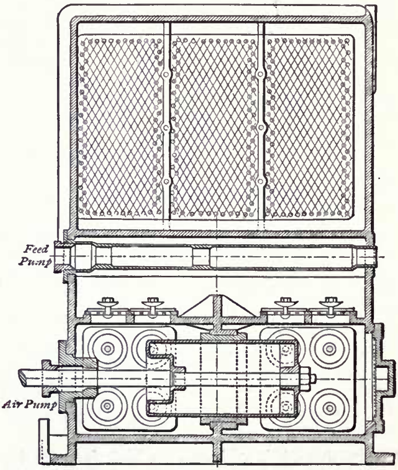


Fig. 185.—Horizontal plung-r-air-pump with surface-condenser for a marine engine.

A Trunk Air-pump has a hollow cylinder attached to the bucket, which passes through a stuffing-box in the air-pump cover, as shown in Fig. 186. It is used in confined places where the space does not admit of a cross-head or guides. In the down-stroke the trunk displaces a portion of the water contained above the bucket, equal in volume to the displacement of the trunk.

The Difference between a Bucket Air-pump, a Piston Air-pump, and a Vertical Plunger Air-pump, is :—A bucket air-pump

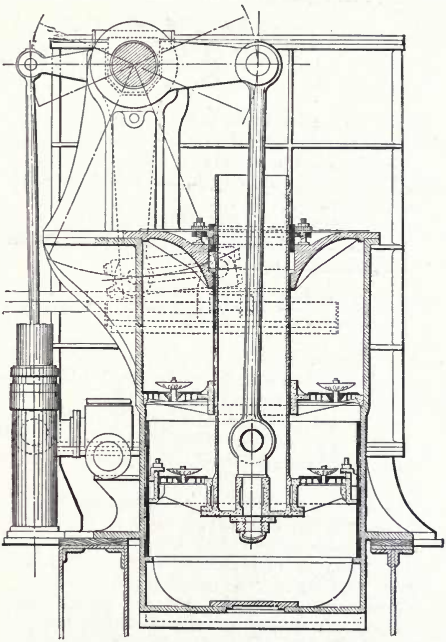


Fig. 186.—Trunk air-pump for a marine engine.

is single-acting, and is fitted with foot and head-valves. The valves in the bucket open on the down-stroke to admit water through the bucket, and close on the up-stroke, the theoretical quantity of water lifted being equal to the capacity of the air-pump for one revolution of the engine.

A piston air-pump may either be made single or double-acting as required. The piston is solid, without valves ; suction and delivery-valves are fitted in the casing. When made single-acting it discharges water at each double stroke of the engine, and when double-acting it discharges at each single stroke of the engine. The vertical plunger air-pump is a double-acting pump, the bucket being attached to a plunger instead of a rod, there are no head-valves; the plunger displaces water contained above the bucket on its down-stroke, and water is discharged at both the up and down-strokes, whereas the bucket air-pump discharges only on its up-stroke.

The Capacity of the Air-pump for a Jet-Condenser may be equal to from $\frac{1}{5}$ to $\frac{1}{8}$ the capacity of the low-pressure cylinder when single-acting, and one-half that capacity when double-acting.

Air-pump capacity $\frac{1}{8}$ that of cylinder = diam. of cylinder	$\times \cdot 5$	When the stroke is one-half that of the piston.
„ „ $\frac{1}{6}$ „ „ „	$\times \cdot 575$	
„ „ $\frac{1}{5}$ „ „ „	$\times \cdot 67$	

When the stroke of the bucket is either more or less than one-half the stroke of the piston, the diameter of the air-pump may be found by this *Rule:* Square the given diameter, multiply by the length of stroke, and divide the product by the proposed length, the square root of the quotient will be the diameter of the air-pump.

Example: An engine with a 4 feet stroke has an air-pump 28 inches diameter, and 2 feet stroke. Required the diameter of the air-pump if the length of stroke be increased to 30 inches.

Then $\frac{28 \times 28 \times 24 \text{ inches}}{30 \text{ inches}} = \sqrt[2]{627} = 25$ inches, the diameter of the air-pump required.

When the capacity of the air-pump is to be equal to $\frac{1}{8}$th of the capacity of the cylinder, it may be calculated as follows:—

Example: The diameter of the cylinder of an engine is 56 inches. The length of stroke is 42 inches. The air-pump is 28 inches diameter, and 18 inches stroke. Is the air-pump large enough to comply with the previous *Rule?*

Then 56×56 inches $\times \cdot 7854 \times 42$ inches $= 103446$ cubic inches, the capacity of the cylinder, and $103446 \div 8 = 12930 \cdot 75$ cubic inches, the capacity the air-pump should be; but the air-pump capacity is $= 28 \times 28$ inches $\times \cdot 7854 \times 18$ inches $= 110844$ cubic inches, therefore it is of less capacity than the *Rule* in question gives.

Air-pump Lever.—An air-pump for a vertical marine-engine is worked by a lever, shown in Fig. 182, connected to the crosshead of the piston-rod. The weight on the centre-bearing of the lever may be calculated by this *Rule:*

Weight on centre-bearing =

$$\frac{\text{Weight on pump-end of lever} \times \text{its distance from centre-bearing}}{\text{Length of crosshead-end of lever from centre-bearing}} +$$

load on lever.

Example: An air-pump lever sustains a weight of 5 tons at the pump-end of the lever which is 3 feet 10 inches from the centre-bearing, the crosshead-end of the lever is 5 feet 9 inches from the centre-bearing.

What will be the weight on the centre-bearing, exclusive of friction and of the weight of the lever and its connections?

Then $\frac{5 \text{ tons} \times 46 \text{ inches}}{69 \text{ inches.}} = 3{\cdot}33$ tons $+$ 5 tons load on the lever $=$ 8·33 tons, the weight on the centre-bearing.

In Jet-condensers, the air-pump discharges the condensation-water, the water from the condensed-steam and the air. In surface-condensers, the air-pump discharges only the water from the condensed-steam and the air; therefore a smaller pump is required in the latter than in the former case. Surface-condensers are sometimes arranged to work also as jet-condensers in cases of accident, then the capacity of the air-pump is made the same as for a jet-condenser.

The Passages and Valves of an air-pump should not be less than equal to one-fourth the area of the air-pump.

The Velocity of the Water through the Passages and Valves of an Air-pump should not exceed 500 feet per minute.

The Capacity of the Air-pump for a Surface-condenser, may be equal to $\frac{1}{24}$ the capacity of the low-pressure cylinder. The capacity of the air-pump, the cooling surface, and also the quantity of injection water provided, varies considerably in practice, as will be seen from the following Table collated from recent practice:—

TABLE 82.—AIR-PUMP CAPACITY, COOLING SURFACE, AND QUANTITY OF CIRCULATING WATER PROVIDED FOR THE SURFACE-CONDENSERS OF MARINE-ENGINES.

Indicated Horse-power of the Marine Engines.	Cooling Surface of the Condenser, in Square Feet.	Relative Capacity of the Air-pump to that of the Low-pressure Cylinder.	Quantity of Circulating Water provided, per Square Foot of Condensing Surface.	
			Cubic Feet.	Lbs. of Water.
625	800	1 : 40·04	·152	9·51
860	1695	1 : 54·83	·167	10·42
1025	1980	1 : 20·47	·151	9·50
1200	3000	1 : 24·13	·241	15·00
1400	3400	1 : 24·00	·226	14·14
1550	3865	1 : 22·86	·208	12·94
1600	1930	1 : 46·28	·142	8·92
1980	4786	1 : 22·23	·192	11·94
2285	7810	1 : 36·47	·191	12·00
2350	4850	1 : 25·00	·187	11·75
2525	2940	1 : 23·83	·161	10·10
2650	5280	1 : 24·36	·079	4·94
3400	5500	1 : 27·50	·121	7·75
3800	5670	1 : 26·10	·196	12·00
4550	5500	1 : 25·20	·128	8·00
5600	11610	1 : 27·25	·215	13·25
8010	11687	1 : 20·37	·222	13·80
10000	25000	1 : 24·00	·248	15·50

Air-pump-Valves.—Indiarubber valves are suitable for moderate temperatures, they should be made of the best caoutchouc, with not more than 3 per cent. of sulphur, and 67 per cent. white oxide of zinc. They should not be employed where mineral oil is used for lubricating the cylinder, as it decomposes them. Several small valves are more effective than one large one. Vulcanised-fibre valves are more suitable for high temperatures than indiarubber, besides being more durable, and not liable to injury from the action of lubricants, and they can be made much thinner than indiarubber. The lift of indiarubber and vulcanised-fibre valves is controlled by the perforated guard, shown in Fig. 187, in which, for comparison, an indiarubber valve is shown on the left-hand side and a vulcanised-fibre valve on the right hand side.

Flexible Metallic Valves for Air-pumps, shown in Fig. 188, are

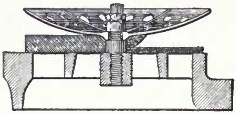


Fig. 187.—Air-pump valve.

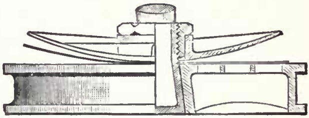


Fig. 188.—Kinghorn's Metallic valves.

made of a thin and elastic sheet of phosphor-bronze. They are neither affected by the hottest water, nor by the action of lubricants, and are efficient and durable. The valve in lifting bends gradually against the curved guard by which the lift of the valve is controlled.

Rigid Metallic Air-pump Valves, shown in Fig. 186, consist of light castings of bronze; the valve is rigid, and slides up the spindle when it lifts from its seat.

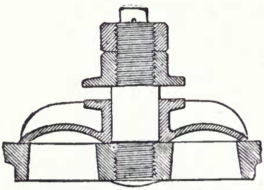


Fig. 189.—Thomson's metallic valves.

Heat Carried off in Condensing-Water.—The efficiency of a steam-engine is the ratio of the heat converted into work to the heat expended. Only a small portion of the heat supplied to an engine is converted into work, the remainder is lost by conduction and radiation, and by passing into the condensing-water of a condensing-engine, or by being discharged in the exhaust steam of a non-condensing engine. Therefore by measuring the quantity of heat carried off by the exhaust steam, the loss of heat, per horse-power developed, may be accurately ascertained. It is evident that of two engines receiving per minute an equal quantity of heat from the boilers, that one is the better of the two which converts the larger portion of this heat into work, and discharges the least into the condensing-water; hence by a measurement of the quantity of heat carried off by the condensing-water, a comparative estimate can be formed of the performance of the two engines. This method of measuring the condensing-water is adopted in engine-tests, to obtain the number of thermal-units or pound-degrees for each indicated horse-power exerted in any condensing-engine. It is a very ready method of testing the consumption of steam where the measurement of the feed-water is difficult or impossible, as steam-boilers are frequently used for other purposes than power. The quantity of the waste-water from the condenser and its increase of temperature simply are taken.

Method of Measuring the Quantity of Water coming from the Air-pump of a Condensing-engine.—It is important to measure this

Fig. 190.

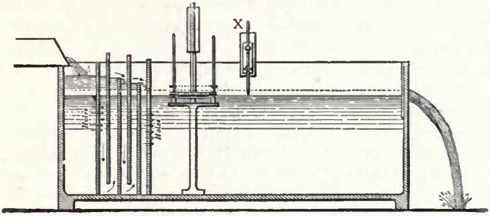


Fig. 191.

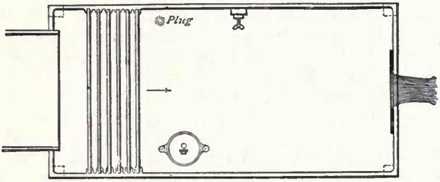
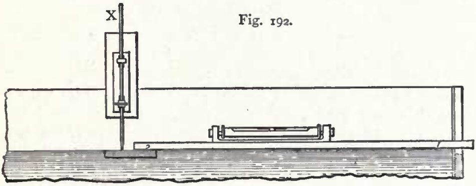


Fig. 192.



Figs. 190—192.—Apparatus for measuring waste-water from condensing-engines.

water in engine-tests, because it contains nearly all the heat and steam that passes through a condensing-engine. It may be measured by a simple apparatus shown in Figs. 190–192, which may be briefly described as follows *:—

The water is led into a tank or reservoir, having a series of transverse divisions or baffle-plates. By passing over some, under others, and

* The Author is indebted to Messrs. B. Donkin & Co., engineers, Bermondsey, London, the makers of the apparatus, for this information, which was originally given in a paper read before the Institution of Civil Engineers, and also in "Engineering."

through the last of these plates, the water can be made to flow very steadily and without undulations. Beyond these divisions the water should be perfectly smooth. Under the above conditions the quantity of water flowing over a weir is easily ascertained by the usual formula, $Q = C\,L\,\frac{2}{3}\sqrt{2\,g\,h^3}$, in which Q is the actual quantity, C the coefficient of discharge, or the percentage of the actual over the theoretical quantity, L the length of the weir, and h the head of water on the weir taken some way back. The above formula becomes Q, cubic feet per second $= C\,L\,5\cdot3472\,\sqrt{h^3}$, putting the values for g and h in feet. When Q is required in lbs. per minute, L and h being taken in inches, the above formula becomes Q lbs. per minute $= C\,L\,40\cdot082\sqrt{h^3}$. Taking the weight of a cubic foot of water at 62·31963 lbs. at 62°, with 0·62 for C, and L (the width of bay) 6 inches, the formula becomes Q lbs. per minute $= 149\cdot105\sqrt{h^3}$. From the above formula Table 83 was calculated.

It is convenient to be able to read on a scale constantly the height of water flowing over the weir, and this is effected in the following way:—A float is fixed near the last baffle-board (perforated with holes) where the water is quite smooth and quiet. The float, generally made of thin sheet brass, and about six inches in diameter, is allowed to move up and down freely, guided by glass rods. It carries an index which rises and falls in front of a fixed scale. The scale is adjusted by a screw, so that the index denotes zero when the water is level with the sill of the weir. To overcome the difficulty of capillary attraction if the tank were simply filled with water to the level of the sill of the weir, a brass rod X, Figs. 190 and 192, with a sharp point at the lower end, is attached to the side of the tank, provided with a vertical screw adjustment. A special straight-edge, Fig. 192, is used, and laid with the end marked 1 on the sill of the weir, and the end marked 2 held under point X. On this straight-edge is laid a spirit-level, and the end of the pointer adjusted until level with the sill of the weir. The straight-edge being removed, the tank is filled with water very slowly until the surface of the water touches the pointer. The water is now level with the end of the pointer and the sill of the weir. The scale should then be adjusted to read zero. If the water is now allowed to run through the tank, any rise of the float will indicate the height of the water over the weir, and the readings can be noted as often as desired. The zero of the scale should be tested before and after each trial. This method is very ready and accurate. The temperature of the hot water should be taken as near to the air-pump as practicable, but not before it is well mixed. In some cases a pencil has been attached to the float, which marked continuously the height of the water above the sill of the weir, on a band of paper moved by clockwork. The temperature of the water can be recorded automatically in a similar manner by means of a Bourdon metallic thermometer. The water as it leaves the weir should fall quite clear of any obstruction. From the quantity, and rise in temperature, of the condensing-water of any engine the heat rejected can of course be readily calculated.

The coefficient for the actual discharge is ·62 of the theoretical discharge; the correctness of this coefficient has been many times practically tested. The following Table saves the trouble of making calculations in connection with the above described apparatus. It gives the weight of water in lbs. discharged per minute for different heads of water flowing over a bay or weir 6 inches wide, such as is shown in Fig. 190.

Table 83.—Weight of Water that will flow over a Bay or Weir 6 inches wide at heights from 1 inch to $5\frac{15}{16}$ inches.

Inches over Bay.	Pounds of Water per Minute.	Inches over Bay.	Pounds of Water per Minute.	Inches over Bay.	Pounds of Water per Minute.	Inches over Bay.	Pounds of Water per Minute.	Inches over Bay.	Pounds of Water per Minute.
1	149	2	422	3	775	4	1193	5	1667
$1\frac{1}{16}$	163	$2\frac{1}{16}$	442	$3\frac{1}{16}$	799	$4\frac{1}{16}$	1221	$5\frac{1}{16}$	1698
$1\frac{1}{8}$	178	$2\frac{1}{8}$	462	$3\frac{1}{8}$	824	$4\frac{1}{8}$	1249	$5\frac{1}{8}$	1730
$1\frac{3}{16}$	193	$2\frac{3}{16}$	482	$3\frac{3}{16}$	849	$4\frac{3}{16}$	1278	$5\frac{3}{16}$	1762
$1\frac{1}{4}$	208	$2\frac{1}{4}$	503	$3\frac{1}{4}$	874	$4\frac{1}{4}$	1306	$5\frac{1}{4}$	1794
$1\frac{5}{16}$	224	$2\frac{5}{16}$	524	$3\frac{5}{16}$	899	$4\frac{5}{16}$	1335	$5\frac{5}{16}$	1826
$1\frac{3}{8}$	240	$2\frac{3}{8}$	546	$3\frac{3}{8}$	924	$4\frac{3}{8}$	1364	$5\frac{3}{8}$	1858
$1\frac{7}{16}$	257	$2\frac{7}{16}$	567	$3\frac{7}{16}$	950	$4\frac{7}{16}$	1394	$5\frac{7}{16}$	1891
$1\frac{1}{2}$	274	$2\frac{1}{2}$	589	$3\frac{1}{2}$	976	$4\frac{1}{2}$	1423	$5\frac{1}{2}$	1923
$1\frac{9}{16}$	291	$2\frac{9}{16}$	612	$3\frac{9}{16}$	1003	$4\frac{9}{16}$	1453	$5\frac{9}{16}$	1956
$1\frac{5}{8}$	309	$2\frac{5}{8}$	634	$3\frac{5}{8}$	1029	$4\frac{5}{8}$	1483	$5\frac{5}{8}$	1989
$1\frac{11}{16}$	327	$2\frac{11}{16}$	657	$3\frac{11}{16}$	1056	$4\frac{11}{16}$	1513	$5\frac{11}{16}$	2022
$1\frac{3}{4}$	345	$2\frac{3}{4}$	680	$3\frac{3}{4}$	1083	$4\frac{3}{4}$	1544	$5\frac{3}{4}$	2056
$1\frac{13}{16}$	364	$2\frac{13}{16}$	703	$3\frac{13}{16}$	1110	$4\frac{13}{16}$	1574	$5\frac{13}{16}$	2089
$1\frac{7}{8}$	383	$2\frac{7}{8}$	727	$3\frac{7}{8}$	1137	$4\frac{7}{8}$	1605	$5\frac{7}{8}$	2123
$1\frac{15}{16}$	402	$2\frac{15}{16}$	751	$3\frac{15}{16}$	1165	$4\frac{15}{16}$	1636	$5\frac{15}{16}$	2157

The above weights are calculated on the assumption that the water be allowed to enter the reservoir and leave the bay under the conditions stated.

The above Table can be used for bays or weirs of other widths by multiplying the tabular quantities by the actual width of the bay or weir and dividing the product by 6.

In using this apparatus, the tank should be so placed that the water flowing over the bay has a clear fall of 12 or 18 inches. The water from the hot-well is delivered into the tank at the end near which the transverse partitions are fixed, and it flows over, under, and through the latter, as shown by the arrows. By this means the current is steadied so that the water over the bay may be accurately measured. The temperature of the condensing-water should be measured, first, before entering the condenser, and, secondly, at its escape from the measuring-tank; the best mode of taking the latter reading being to allow the overflowing water to fall on the thermometer. By doing this, it is secured that the reading is taken at a place where the water is thoroughly mixed, and its temperature is therefore fairly uniform throughout.

The Mode of carrying out an Engine-Test on the above System is as follows: Indicators should be fitted to the engines, and indicator diagrams should be taken simultaneously from both ends of the engine-cylinder, at regular intervals, say of ten minutes and at the same time, the initial and final temperatures of the condensing-water should be taken, and the head of discharge should be noted. Besides these observations, the speed of the engine should be recorded, if possible, by a counter. The more irregular the power developed by the engine, the more frequent should be the intervals at which the observations are taken, while in all cases these intervals should be regular, as in dealing with considerable variations of power, an irregularity in the intervals between the observations, may lead to erroneous results.

From the data obtained in an engine-test carried out in the manner above explained, the following facts can be derived:—

(1.) The mean indicated horse-power.
(2.) The mean quantity of water discharged per minute from the condenser, and
(3.) The mean rise in temperature of the condensing-water.

If the number of pounds of water discharged per minute be multiplied by the mean rise in temperature, and the product divided by the mean indicated horse-power, the quotient will be the number of pound-degrees, or units of heat discharged per minute per indicated horse-power, and this number forms a constant by which the performance of the engine can be judged, or from which the quantity of steam used by it can be calculated.

To make this perfectly clear it may be illustrated by an example. Suppose the observations made on an engine show that the mean indicated horse-power developed is 130 horse-power; the average quantity of water discharged per minute from the condenser is 1250 lbs.; and that the mean rise in temperature of this water is 40° Fahr., then the number of units of heat discharged from the condenser per minute is $1250 \times 40 = 50000$ pound-degrees, and the engine "constant" or number of units of heat discharged per horse-power per minute is $\frac{50000}{130} = 384 \cdot 6$. This "constant" forms a means by which the performance of the engine can be compared with that of any other engine tested on the same system; the lower this constant is, the higher, of course, being the efficiency of the engine.

The manner in which the quantity of steam used per horse-power per hour can be obtained from the "constant" can be readily explained by continuing the consideration of the example above given. Suppose, for instance, that the engine in question is supplied with steam of 50 lbs. pressure, and that the temperatures of the injection-water and of the discharge from the condenser are 55° and 95° respectively. The amount of steam used will be made up of three quantities, namely, first, that to supply the heat converted into work; second, that to supply the heat imparted to the condensing-water; and third, that to supply the losses by radiation, &c. For each horse-power developed there are converted into work $\frac{33000 \times 60}{772} = 2564 \cdot 5$ units of heat per hour, and as the total heat of steam of 50 lbs. pressure is 1207·8°, and the discharge from the condenser takes place at 95°, it gives in this case $\frac{2564 \cdot 5}{1207 \cdot 8 - 95} = 2 \cdot 306$ lbs. of steam per hour required to account for the heat converted into work. Again, the discharge from the condenser, as shown by the "constant," carries off $384 \cdot 6 \times 60 = 23076$ units of heat per hour, and to furnish this $\frac{23076}{1207 \cdot 8^\circ - 55^\circ} = 20 \cdot 01$ lbs. of steam have to be provided. It will be noticed that in this last calculation the divisor is the total heat of the steam less the temperature of the injection-water, and not of the discharge from the condenser. This divisor must be taken, because the whole discharge from the condenser does not consist of heated injection-water, but of that water mixed with the steam which has been condensed. This is allowed for by taking the divisor as given. Thus 2·306 + 20·01 = 22·316 lbs. of steam per hour are accounted for, and to this amount an allowance must be added for losses by radiation, &c., which may be taken at 5 per cent., making the total amount of steam required per horse-power per hour = 23·48 lbs.

The quantity of steam required calculated as above must be true steam. If the boiler primes to the extent of 10 per cent.—a by no means uncommon occurrence—then the *apparent* evaporation from the boiler working the engine we have taken as our example, would be greater, or probably about $25\frac{1}{2}$ lbs. of steam per horse-power per hour. It will be seen from the above, that when used in conjunction with a boiler trial, this system of testing affords a clear indication of the steam supplied to the engine, while it is evident that when used alone it gives results which are practically (although not absolutely) independent of priming, and which afford therefore a better criterion of the performance of the engine itself than can be obtained by any other mode of testing.

WATER-PUMPS AND TANKS.

Pumps.—All pumps lift considerably less water than the theoretical quantity due to their size and speed. The efficiency or quantity of work realized in proportion to that applied, is called the modulus of a machine. Thus, if a pump only lifts one-half the quantity of water it is theoretically capable of lifting, its modulus will be ·5.

The Modulus or Efficiency of Pumps averages as follows:—

Centrifugal pumps, low lift	·50
Common lifting pumps	·50
Ordinary lifting and force-pumps	·66
Air-pumps	·56 to ·56

In high-class pumps, such as are used for water-works, the quantity of water actually pumped is frequently from 80 to 94 per cent. of the theoretical contents of the pumps.

Example 1: How many cubic feet of water is a 6 horse-power steam-pump capable of lifting per hour to a height of 20 feet?

Then, the work applied per hour = 6 × 33000 × 60 minutes.

The work done = 6 × 33000 × 60 × ·66 modulus.

The work in raising 1 cubic foot of water = 62·5 lbs. × 20 feet.

The number of cubic feet of water $= \dfrac{6 \times 33000 \times 60 \times \cdot 66}{62\cdot 5 \times 20} = 6273$ cubic feet per hour.

Example 2: A 6 horse-power steam-pump lifts 6273 cubic feet of water per hour to a height of 20 feet, What is the modulus or efficiency of the pump?

The work done per hour = 6 × 33000 × 60 minutes.

Total work done = 6273 × 62·5 × 20 feet.

The efficiency or modulus =

$$\frac{\text{Work done}}{\text{Work applied}} = \frac{6 \times 33000 \times 60}{6273 \times 62\cdot 5 \times 20} = \cdot 66 \text{ modulus.}$$

⁂ For much useful information on Pumps and Pumping see the author's work "The Works' Manager's Handbook," published by Crosby Lockwood & Son, London.

Example 3: Required the indicated horse-power of an engine to pump 6273 cubic feet of water per hour from a depth of 20 feet.

Then, work in raising the water per hour $= 6273 \times 62\cdot5$ lbs. $\times 20$ feet.

Effective work of one horse-power per hour $= 33000 \times 60$ minutes $\times \cdot66$ modulus, and the power of the engine $= \dfrac{6273 \times 62\cdot5 \times 20}{33000 \times 60 \times \cdot66} = 6$ horse-power.

Example 4: A tank is 12 feet long, 6 feet wide and 6 feet deep; the height from the water in the well to the bottom of the tank is 70 feet. In what time can the tank be filled with water by a hand-pump, if the man performs 2600 units of work per minute and the modulus of the pump is ·66?

Then, weight of water the tank will hold $= 12 \times 6 \times 5$ feet $\times 62\cdot5$ lbs. $= 27000$ lbs.

Height to which the centre of gravity of the water is raised $= \frac{6}{2} + 70 = 73$ feet.

Work $= 27000$ lbs. $\times 73$ feet $= 1971000$.

Effective work of the man per hour 2600 units $\times 60$ minutes $\times \cdot66$ modulus $= 102960$.

Then, $\dfrac{1971000 \text{ work}}{102960 \text{ effective work}} = 19\frac{1}{4}$ hours.

A Vertical Pump, with ram, is shown in Fig. 193. The valves and seats are of gun-metal, and the casing of cast-iron.

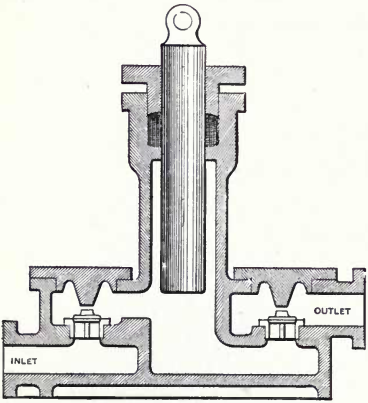


Fig. 193.—Vertical pump.

The Area of a Pump-Barrel is $=$ diameter $^2 \times \cdot7854$.

The Capacity of a Pump-Barrel is $=$ the area in square inches $\times$ the length of stroke in inches.

The Quantity of Water in Cubic Inches delivered per Stroke is $=$ the capacity $\times$ the fraction of the pump's efficiency.

The Quantity in Cubic Inches delivered per Hour is $=$ the net quantity of water delivered per stroke $\times$ the number of strokes per minute and by 60.

The Quantity in Cubic Feet delivered per Hour is $=$ the quantity in cubic inches divided by 1728.

Example: How many cubic feet of water will be delivered per hour by a single-acting pump, 4 inches diameter, 12 inches stroke, number of strokes per minute $= 20$, the pump being two-thirds full at each stroke?

Then $4^2 \times \cdot 7854 = 12\cdot 566$ the area of the pump-barrel in square inches.

$12\cdot 566 \times 12$ inches stroke $= 150\cdot 792$ the capacity of the pump-barrel in cubic inches.

$150\cdot 792 \times \frac{2}{3} = 100\cdot 528$ the quantity of water delivered per stroke in cubic inches.

$100\cdot 528 \times 20$ strokes $\times 60 = 120633\cdot 6$ the quantity of water delivered per hour in cubic inches.

$120633\cdot 6 \div 1728 = 69\cdot 81$ the quantity of water delivered per hour in cubic feet.

The Diameter and Length of Stroke of Pumps, the number of strokes per minute, and the quantity of water delivered, may be obtained by the following formulæ :—

Let Q = the quantity of water in cubic feet delivered per hour.
D = the diameter of the pump-barrel in inches.
L = the length of stroke in inches.
N = the number of strokes per minute.
M = the modulus of the pump = in most cases $\frac{2}{3}$ or ·66.
P = the quantity of fresh-water in tons.
S = the quantity of sea-water in tons.

$$Q = \frac{\cdot 7854\ D^2\ L\ N\ M \times 60}{1728}.$$

$$D = \sqrt[2]{\left(\frac{1728\ Q}{\cdot 7854\ L\ N\ M \times 60}\right)}.$$

$$L = \frac{1728\ Q}{\cdot 7854\ D^2\ N\ M \times 60}.$$

$$N = \frac{1728\ Q}{\cdot 7854\ D^2\ L\ M \times 60}.$$

$$M = \frac{1728\ Q}{\cdot 7854\ D^2\ L\ N \times 60}.$$

$$P = \frac{\cdot 7854\ D^2\ L\ N\ M \times 60}{1728 \times 35\cdot 9}.$$

$$S = \frac{\cdot 7854\ D^2\ L\ N\ M \times 60}{1728 \times 35}.$$

Example: Required the diameter of a pump to deliver 69·81 cubic feet of water per hour. Length of stroke, 12 inches; number of strokes per minute, 20.

Then $\frac{1728 \times 69\cdot 81}{\cdot 7854 \times 12 \times 20 \times \frac{2}{3} \times 60} = 16$ and $\sqrt[2]{16} = 4$ inches diameter.

A vertical pump with an escape-valve attached to the delivery side of the pump is shown in Fig. 194. The escape-valve relieves the pipes of any strain due to the accidental obstruction of the delivery pipes : it is loaded with a spring, which may be adjusted to suit the pressure required. The water which passes the escape-valve, is discharged into the suction-pipe.

The Time required to Pump a given Quantity of Water may be found by dividing the quantity of water in cubic feet to be pumped, by the number of cubic feet of water discharged per hour by the pump.

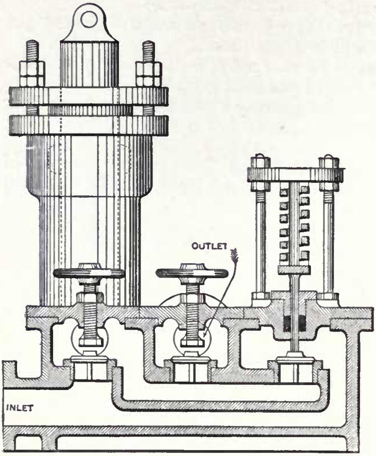


Fig. 194.—Vertical pump, with escape-valve.

Example: What length of time will be required to empty a tank containing 100 tons of sea-water, with a single-acting pump, 4 inches diameter, 12 inches stroke, making 20 strokes per minute, the pump being two-thirds full at each stroke.

Then the quantity of water in the tank will $= 100 \times 35$ cubic feet per ton $= 3500$ cubic feet. The quantity of water the pump is capable of discharging per hour as given in the previous example is 69·81, or say 70 cubic feet per hour, and $3500 \div 70 = 50$ hours. If the pump had been double-acting, only one-half the time, or 25 hours, would be required to empty the tank.

The Time required to Pump a given Quantity of Water, when two or more pumps of different size are employed to empty a tank, may be found by the following :—

Rule : Divide the product of the time in which each pump will separately empty the tank, by the sum of the times required by each pump to separately empty the tank.

Example: The water-ballast pump of a steamship will empty a tank in 3 hours, and the boiler pump will empty it in 10 hours: In what time can the tank be emptied by both pumps working together?

Then $\dfrac{3 \times 10}{3 + 10} = 2$ hours, 18 minutes.

The Quantity of Water required to be Pumped into a Boiler to raise the water-level a given height may be found by this *Rule:* Multiply the superficial area of the water-level by the required height and divide by the quantity of water.

Example: The surface-area of water in a boiler at the water-level is $= 6$ feet $\times$ 10 feet: How many tons of sea-water are required to be pumped into the boiler to raise the water-level in the water-gauge glass to the extent of 6 inches?

Then $\dfrac{6 \times 10 \times \cdot 5 \text{ feet}}{35 \text{ cubic feet}} = 17 \cdot 14$ cwt.; or if the dimensions be taken in inches, then $\dfrac{72 \times 120 \times 6 \text{ inches}}{35 \text{ cubic feet} \times 1728 \text{ inches}} = 17 \cdot 14$ cwt.

For fresh water, use 35·9 as a divisor, instead of 35 in the above example.

The Velocity of Water in a Pump in feet per minute, may be found by multiplying the length of stroke in feet by the number of strokes per minute. The velocity of water in pipes is in inverse proportion to their areas, or to the squares of their diameters.

Example: A pump makes 50 strokes per minute. Length of stroke, 2 feet; diameter of ram, 6 inches; diameter of delivery pipe, 3 inches. Required the velocity of the water in the pump-barrel and in the delivery pipe in feet per minute.

Then 50 strokes × 2 feet = 100 feet, the velocity of the water in the pump-barrel and $\frac{6 \text{ inches} \times 6 \text{ inches} \times 100}{3 \text{ inches} \times 3 \text{ inches}} = 400$ feet, the velocity of the water in the delivery-pipe.

In Calculations for Pumps, where accuracy is required, it is necessary, in addition to the load on the pump, to make allowance for the following resistances:—

1st. The friction of the ram in the gland, and of the bucket or piston against the sides of the pump.
2nd. The friction of the water in the pipes.
3rd. The resistance of the water in passing through the pump-valves.
4th. The weight of the valves.
5th. The force necessary to work the ram or bucket by itself.

In well-proportioned vertical-pumps, the load to be overcome in raising the pump-bucket or ram, including these resistances, independent of the weight of the ram or bucket and rod, may be found by the following formulæ:—

When the force necessary to work the bucket is not taken into account, the load to be overcome in raising the ram or bucket is equal to $62{\cdot}5 \text{ lbs.} \times D^2 \times {\cdot}7854 \times L \times R$.

Where 62·5 = the weight in lbs. of a cubic foot of water.
D = the diameter in feet of the pump-barrel.
H = the height in feet from the level of the water from which the pump draws its supply, to the point of delivery above.
R = a co-efficient to provide for the above resistances = 1·1 in most cases.

Then $62{\cdot}5 \times {\cdot}7854 \times 1{\cdot}1 = 54$, and the formula for the load to be overcome on the pump is simply $54\, D^2 \times H$.

When the force necessary to work the ram or bucket by itself is not known, it may in most cases be provided for by adding ·06 to 1·1 given above, and the co-efficient of the resistances becomes 1·16; then $62{\cdot}5 \times {\cdot}7854 \times 1{\cdot}16 = 57$, and the formula in such cases for the load to be overcome on the pump is $57\, D^2 \times H$.

In each case the weight of the pump-ram, or bucket and rod, must be added to the result obtained, in order to find the total load to be overcome in raising the ram or bucket in a vertical pump. As the weight of the ram or bucket assists its depression, less power is required to depress it than that necessary to raise it.

Example: Required the load in lbs. to be overcome in raising a pump-bucket, 6 inches diameter, to lift water a height of 50 feet, from the water in the well to the point of delivery in a tank, the weight of the bucket and rod not being taken into account.

Then $57 \times \cdot 5 \times \cdot 5 \times 50 = 712 \cdot 5$ lbs.

The Horse-power required to Work a Pump may be found by adapting the above formula as follows, where V is the velocity of the pump in feet per minute.

$$\text{Horse-power of a pump} = \frac{57\, D^2 \times H \times V}{33{,}000}.$$

Example: Required the horse-power of the pump described in the last example, if the velocity of the pump-bucket is 60 feet per minute.

$$\text{Then } \frac{57 \times \cdot 5 \times \cdot 5 \times 50 \times 60}{33{,}000} = 1 \cdot 29 \text{ horse-power.}$$

The Quantity of Water to be Pumped out of a Ship, flowing in through a hole in the bottom, may be found by the following formula, where Q = the quantity of water in cubic feet flowing in per second, and H = the depth in feet of the hole below the surface of the water.

$$Q = \frac{\text{Area of hole in square inches} \times 8\sqrt{H}}{144}$$

The Quantity of Water a Tank will contain may be found by the following *Rule*: Multiply the length, width, and depth together and multiply the product by 6·25, the product will be the number of gallons of fresh water the tank will hold.

Example: How many gallons of water will a tank 4 feet long, 4 feet wide, and 3 feet deep contain?

Then $4 \times 4 \times 3 \times 6 \cdot 25 = 300$ gallons, or 100 gallons per foot in depth.

The Weight of Water a Tank will contain may be found by the following *Rule:* Multiply the contents of the tank in cubic feet by 62·5 for the weight in lbs. of fresh water, or by 64 for the weight in lbs. of sea-water.

Example: Required the weight of fresh water that the tank named in the previous example will contain.

Then $4 \times 4 \times 3 = 48 \times 62 \cdot 5 = 3000$ lbs. or $48 \div 35 \cdot 9 = 1$ ton 6 cwt. 0 qrs. 20 lbs. This tank would hold $48 \div 35 = 1$ ton. 7 cwt. 1 qr. 19 lbs. of sea-water.

The Size of Tank to contain a given Number of Gallons may be found by the following formulæ: the dimensions being expressed in feet.

$$\text{The depth of a tank} = \frac{\text{Number of gallons}}{\text{Length} \times \text{width} \times 6 \cdot 25}.$$

$$\text{The width of a tank} = \frac{\text{Number of gallons}}{\text{Length} \times \text{depth} \times 6 \cdot 25}.$$

$$\text{The length of a tank} = \frac{\text{Number of gallons}}{\text{Width} \times \text{depth} \times 6 \cdot 25}.$$

Example 1: A tank 4 feet long, 4 feet wide, and 3 feet deep, contains 200 gallons of water: What is the depth of the water, and what is the distance of the surface of the water from the top of the tank?

Then $\frac{200 \text{ gallons}}{4 \text{ feet long} \times 4 \text{ feet wide} \times 6 \cdot 25} = 2$ feet depth of water in the tank, and the distance of the surface of the water from the top of the tank is $3-2=1$ foot.

Or 200 gallons $\div 6 \cdot 25 = 32$ cubic feet of water in the tank.

4 feet long $\times$ 4 feet wide $= 16$ square feet surface of water.

$32 \div 16 = 2$ feet depth of water in the tank.

And 3 feet depth of tank—2 feet $= 1$ foot depth of the surface of the water from the top of the tank.

Example 2: A tank 4 feet long, 4 feet wide, and 4 feet deep is full of olive oil: required the consumption per day, if in 10 days the depth of oil is lowered 18 inches; and also the weight of oil in the tank.

Then 4 feet $\times$ 4 feet $\times$ 4 feet $= 64$ cubic feet, the contents of the tank, and 4 feet wide $\times$ 4 feet long $\times$ 1·5 foot $= 24$ cubic feet, the quantity of oil used, $24 \times 6 \cdot 25 = 150$ gallons, $150 \div 10 = 15$ gallons consumption per day.

The specific gravity of olive oil is ·915 and 62·5 lbs. $\times$ ·915 $= 57 \cdot 19$ lbs., the weight of a cubic foot of oil, then $57 \cdot 19 \times 64$ cubic feet the capacity of the tank $= 3660$ lbs. the weight of oil.

The Diameter of a Tank to contain a given number of Gallons, being given to find the depth, or the depth being given to find the diameter, all the dimensions being expressed in feet.

The height of a circular tank $= \frac{\text{Number of gallons}}{\text{Diameter}^2 \times \cdot 7854 \times 6 \cdot 25}$.

The area of a circular tank $= \frac{\text{Number of gallons}}{\text{Height} \times 6 \cdot 25}$.

The diameter of a tank $= \sqrt[2]{\text{area} \div \cdot 7854}$.

Example 1: Required the height of a tank 4 feet 6 inches diameter, to hold 400 gallons of oil.

$$\text{Then } \frac{400 \text{ gallons}}{4 \cdot 5 \times 4 \cdot 5 \times \cdot 7854 \times 6 \cdot 25} = 4 \cdot 02 \text{ feet high.}$$

Example 2: Required the diameter of a circular tank 4·02 feet high, to contain 400 gallons of oil.

Then $\frac{400 \text{ gallons}}{4 \cdot 02 \times 6 \cdot 25} = 15 \cdot 9$ area in square feet and $15 \cdot 9 \div \cdot 7854 = \sqrt[2]{20 \cdot 24}$ $= 4$ feet 6 inches diameter.

The Time a Tank will take in Filling, when a given quantity of water is going in and a given quantity going out in a given time, may be found by this *Rule:* Divide the contents of the tank, in gallons, by the difference of the quantity going in and that going out of the tank, and the quotient will be the time in hours that the tank will take in filling.

Example: If 60 gallons per hour be pumped into, and 40 gallons per hour be pumped out of a tank capable of holding 400 gallons, in what time will the tank be filled?

$$\text{Then } \frac{400}{60-40} = 20 \text{ hours.}$$

SLIDE-VALVES, PISTON-VALVES, CORLISS-VALVES, LINK-MOTION AND OTHER VALVE-GEARS.

The Slide-valve regulates the admission, expansion, and exhaust of the steam in the cylinder. The action of the slide-valve may be clearly comprehended by constructing, with thin strips of wood, a working model of a slide-valve, arranged so that the valve may be moved over the ports of the cylinder, to show how communication is established alternately between each steam port and the exhaust port, and the position of the valve at various points of the stroke of the piston.

A model of an ordinary single-ported slide-valve is shown in Fig. 195,

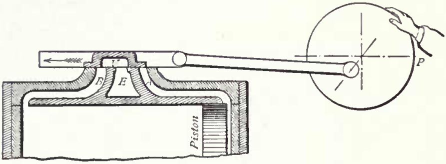


Fig. 195.—Model of an ordinary slide-valve.

in which the slide-valve V is shown in the position it would occupy when the piston has moved in the cylinder to the end of one stroke, and is about to commence the return stroke, A being the port through which steam is entering the cylinder, and B the port through which the steam, which propelled the piston on its previous stroke, is escaping into the exhaust port. The valve V is moving in the direction of the arrow, and opening the port, A, for the admission of the steam, and a communication is at the same time established between the port B, through which the steam is exhausting, and the exhaust port E. When the piston arrives at the other end of the cylinder the valve will have moved back again, and these conditions will be reversed; the steam will be admitted to the cylinder through the port B, and it will escape or exhaust from the cylinder through the port A. The valve moves from its middle position to the end of its travel, and back again to its middle position, while the piston moves from one end to the other end of the cylinder.

A similar model of an expansion slide-valve is shown in Fig. 196. The

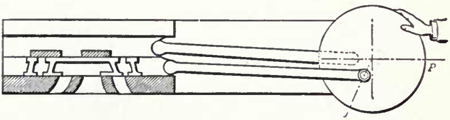


Fig. 196.—Model of an expansion slide-valve.

circle in each case represents the motion of the crank-pin, and P is a pointer to show the position of the crank when the valve is set to give the required lead.

When the terms lap and lead are used, they are understood to refer to outside lap and lead only, unless otherwise expressed.

Lead of a Valve is the width the port is open to steam at the beginning of the stroke, as shown in Fig. 197, in order to bring the full pressure of the steam on the piston as it commences its stroke. The steam thus admitted acts as a cushion upon the piston, and enables it to reverse its motion without shock, and prevents the jerk and strain which would come on the crank-pin if the piston were thrown upon the crank with full force at its dead points ; it also ensures the quick getting-away of the piston from the end of the stroke.

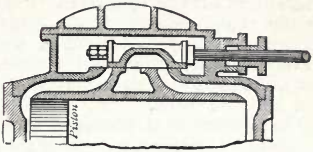


Fig. 197.—Section of cylinder-ports and slide-valve.

Lead is effected by fixing the eccentric a little more than 90° in advance of the crank, as shown in Fig. 198, the throw-line A of the eccentric being placed in advance of a line B, at right angles to the centre-line of the crank C, the arrow denoting the direction in which the engine is to run.

The Radius of the Eccentric is equal to one-half the travel of the valve.

The angular advance of the eccentric θ, may be found approximately by this *Rule* :—

$$\text{Sin } \theta = \frac{\text{outside lap} + \text{outside lead}}{\text{half-travel of the valve.}}$$

The Angle of Advance of the Eccentric is the value of θ, found by the last rule, added to 90°.

The positions of the eccentrics on the crank-axle of a locomotive engine

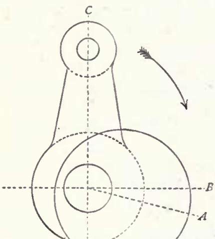


Fig. 198.—Crank and eccentric.

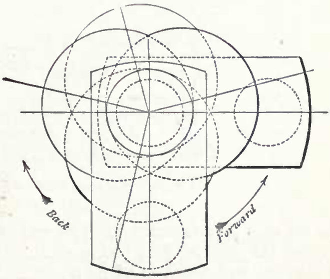


Fig. 199.—Locomotive crank-axle and eccentrics.

are shown in Fig. 199, in which the angular advance of each eccentric is clearly shown. The eccentric is made in halves, united upon the axle by bolts : there are two eccentrics to each cylinder—the forward and the backward.

Deficient and Excessive Lead of the Valve.—If the lead of the valve is deficient or altogether wanting, the maximum pressure of the steam in the cylinder is not attained until after a portion of the stroke is traversed by the piston. When the lead of the valve is excessive, the steam is admitted so readily as to be momentarily compressed and to cause in some cases an unfavourable pulsatory action of the steam. The total absence of lead of the valve likewise occasions an unsteady pulsatory action of steam in the cylinder, as steam can neither enter nor leave the cylinder when the crank is on its dead centres. High-speed engines require more lead than engines of moderate speed.

Compression of Steam or Cushioning, by giving the valve the proper amount of lead, and by closing the exhaust port a little before the termination of the return stroke, enables the momentum of the piston to be efficiently balanced, and the play of the bushes or bearings to be taken up, by which means thumping and hot bearings are prevented. The compressed steam saves steam which would otherwise be wasted in filling the clearance and waste-room in ports and passages. The rise in temperature of the compressed steam re-evaporates any water condensed on the surfaces of the metal, and sufficient steam may be locked in the cylinder to raise the pressure at the end of the stroke to nearly that of the steam entering from the steam-chest. Excessive compression pulls up, or retards the working of, the engine, and deficient compression leads to sudden shocks on the admission of steam, and irregularity in the working of the engine.

Lap of a Valve is the amount the valve extends beyond the edges of each of the steam-ports, when the valve is in the position of half-travel, as shown in Fig. 200, and its effect is to cut off the supply of steam when the piston has only travelled a portion of its stroke, the remainder of the stroke being performed by the expansive force of the

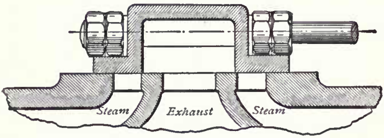


Fig. 200.—Section of cylinder-ports and slide-valve.

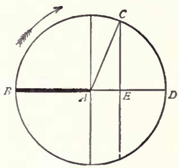


Fig. 201.—Diagram showing lap of slide-valve.

steam shut into the cylinder until the valve opens for exhaust. If the valve opens the exhaust-port too soon, the steam is released before it is fully expanded and deprived of its force, and if the valve opens the exhaust-port too late, it will cause back-pressure, and the engine will not run smoothly. High-speed engines require large measures of expansion, and an early release of the steam, to prevent back-pressure.

The Lap of the Valve and opening of the steam-port may be ascertained approximately by drawing a diagram like Fig. 201, in which A B is the position of the crank, at the beginning of the stroke. The circle represents the path of the centre of the eccentric, which moves the slide-valve. The diameter B D is the whole travel of the valve, C is the point which the centre of the eccentric occupies when the piston is at the end of the stroke. Draw the line E perpendicular to B D from the point C. Then, neglecting the obliquity of the eccentric-rod,

A E = the lap of the valve.
E D = the opening of the steam-port.

Table 84.—LAP OF VALVE REQUIRED FOR VARIOUS GRADES OF EXPANSION OF STEAM.

Distance of the piston from the termination of its stroke, in parts of the length of its stroke.	$\frac{8}{24}$ or $\frac{1}{3}$	$\frac{7}{24}$	$\frac{6}{24}$ or $\frac{1}{4}$	$\frac{5}{24}$	$\frac{4}{24}$ or $\frac{1}{6}$	$\frac{3}{24}$ or $\frac{1}{8}$	$\frac{2}{24}$ or $\frac{1}{12}$	$\frac{1}{24}$
Lap on the steam side of the valve, in parts of the length of the stroke of the valve.	·289	·270	·250	·228	·204	·177	·144	·103

The Lap, Lead, and Travel of Locomotive-Engine Slide-valves, and Size of Ports, are frequently as follows :—

Lap of slide-valve, either . . $\frac{7}{8}$ inch, 1 inch, or $1\frac{1}{8}$ inch.
Lead of slide-valve, either . $\frac{1}{16}$ inch, $\frac{3}{32}$ inch, or $\frac{3}{16}$ inch.
Maximum travel, either . . $3\frac{3}{4}$ in., 4 in., $4\frac{1}{2}$ in., or 5 in.
Steam-port, either . . . . . $15 \times 1\frac{1}{4}$ in., or $15 \times 1\frac{1}{2}$ in.
Exhaust-port, either . . . 15×3 in., or $15 \times 3\frac{1}{2}$ in.
Thickness of bridge . . . 1 in.

Exhaust-lap or inside-lap is the amount which the inside edge of the

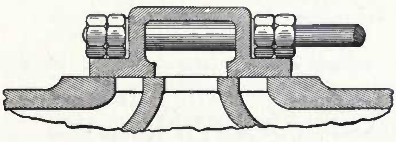


Fig. 202.—Slide-valve with exhaust-lap.

slide-valve covers or overlaps the steam-port when the valve is at half-stroke, as shown in Fig. 202. Its effect is to delay the release of the steam or period of exhaust, and to increase the amount of cushioning; it is sometimes applied to the slide-valves of quick-speed engines.

Negative Exhaust-lap, or inside clearance, is the amount which the inside edge of the valve clears the edge of steam-port, or is open to the exhaust when the valve is at half-stroke, as shown in Fig. 203. Its effect is to hasten the release of the steam, or period of exhaust, and to decrease the amount of cushioning.

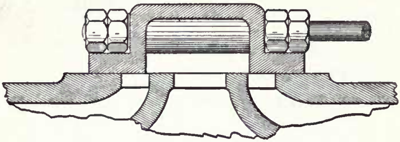


Fig. 203.—Slide-valve with negative lap.

The closing of the Exhaust-port too soon, is followed by excessive compression of the steam, or cushioning; if it be closed too late, it is followed by back-pressure and deficient cushioning.

The Point of Cut-off is the position of the piston at the instant the slide-valve has closed the steam-port, and cut off the admission of steam to the cylinder. The most economical point of cut-off is, theoretically, that which would leave the steam at the instant of its release from the cylinder no capacity for developing further power. The pressure of the steam would then have fallen during expansion to that of the back-pressure, and the release of the steam would take place when the final pressure equals one atmosphere in a non-condensing engine, and when it equals that of the condenser in a condensing-engine. In such cases the ratio of expansion is equal to the pressure of the steam on its admission to the cylinder, divided by the back-pressure, according to Marriotte's law of expansion, and the initial pressure multiplied by the initial volume will equal the final pressure multiplied by the final volume.

$$\text{And } \frac{\text{initial pressure}}{\text{final pressure}} = \frac{\text{final volume}}{\text{initial volume}} = \text{ratio of expansion.}$$

This is theoretically the best adjustment of the point of cut-off, but in practice allowance must be made for loss of power by radiation of heat, cooling by expansion, and for friction, which necessitates a different adjustment of the point of cut-off.

The Point of the Stroke at which the Slide-valve cuts off the Steam may be found by this *Rule*:—

$$\text{Point of cut-off} = 1 - \left(\frac{\text{lap} \times 2}{\text{travel of valve}}\right)^2$$

Example: The travel of a slide-valve is 6 inches, the lap of the valve is $1\frac{1}{4}$ inches, at what part of the stroke will the slide-valve cut off the steam?

$$\text{Then } 1 - \left(\frac{1{\cdot}25 \times 2}{6}\right)^2 = {\cdot}8265 \text{ of the stroke.}$$

The Distance of the Piston from the end of the Stroke when the Steam is cut-off may be found by this *Rule*:—

$$\left(\frac{\text{lap} \times 2 + \text{lead}}{\text{travel of valve}}\right)^2 \times \text{length of stroke.}$$

Example: The travel of a slide-valve is 8 inches, the lap is $1\frac{3}{4}$ inch, the lead is $\frac{1}{16}$ inch, and the length of stroke of the piston is 40 inches. How far is the piston from the end of the stroke when the steam is cut off?

Then $\left(\frac{1{\cdot}75 \times 2 + {\cdot}0625}{8 \text{ inches}}\right)^2 \times 40$ inches $= 7{\cdot}942$ inches, the distance of the piston from the end of the stroke when the steam is cut off.

The Travel of a Slide-valve necessary to open the steam-port its full width, is equal to twice the sum of the width of port and the lap of the valve. Thus, for a steam-port $1\frac{1}{2}$ inch wide, and 1 inch lap of valve, the travel of the valve will be $1\frac{1}{2} + 1 \times 2 = 5$ inches; and if the length of port be 16 inches, the greatest area the valve is open for steam is $16 \times 1\frac{1}{2} = 24$ square inches.

The area of Opening given by a Slide-valve for the admission of steam when the travel of the valve is not sufficient to open the steam-port its full width, may be found by this *Rule:* Deduct the lap from one-half the travel of the valve, which will give the width the steam-port is opened by the valve, multiply by the length of the port, and the product will be the area of the opening.

Example: The steam-port of a cylinder is 24 inches long and $2\frac{1}{2}$ inches wide; the lap of the valve is $1\frac{5}{8}$ inch, the travel of the valve is $6\frac{3}{4}$ inches: What is the greatest area of opening given by the valve for the admission of steam?

Then $6\frac{3}{4} \div 2 = 3\frac{3}{8}$ inches, one-half the travel of the valve, $3\frac{3}{8} - 1\frac{5}{8} = 1\frac{3}{4}$ inches, the width the valve opens, and $1{\cdot}75 \times 24$ inches $= 42$ square inches, the area of the opening given by the valve for the admission of steam.

A Trick Slide-valve, shown in Figs. 204 and 205, is a valve with a

Fig. 204.

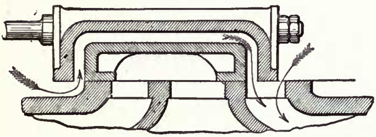
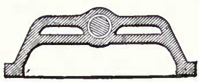


Fig. 205.



Figs. 204 and 205.—Sections of a trick-valve.

steam-passage through it, the steam being admitted from both ends of the steam-chest into the same port. By this means a large opening for the admission of steam is obtained with a small travel of the valve.

A Double-ported Slide-valve admits steam to the cylinder through two steam-ports at each end of the valve-seating, as shown in Fig. 206, thus enabling the same quantity of steam to be admitted to the cylinder with one-half less travel of the valve than that required for an ordinary valve and a single port at each end of the valve-seating. This form of valve is adapted for large engines, and is largely used for marine engines.

A Gridiron Expansion-valve has a number of bars which slide over corresponding openings in the valve-face, as shown in Fig. 207, a large area of opening for the admission of steam being obtained by this arrangement from a small travel of the valve. The travel of the valve is regulated by a link attached to the valve-spindle, worked by an eccentric from the crank-shaft as shown; the travel of the valve being altered by moving the block in the link up or down, by which means the point of cut-off may be varied as required. This form of valve is usually employed when the expansion-valve is required to work on an independent face, and it is also sometimes arranged to work on the back of a main slide-valve.

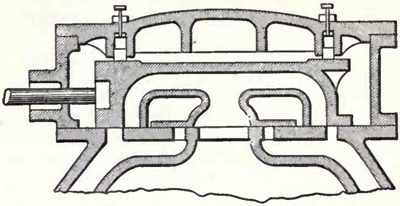


Fig. 206.—Double-ported slide-valve.

A separate Expansion-valve works separately from, and is independent of, the main slide-valve. It is used in cases where an earlier cut-off of steam is required than can be obtained by lap on the ordinary slide-valve, or where variable expansion is required. It is worked by an independent eccentric and rod, and the cut-off can be arranged to suit any required portion of the stroke of the piston.

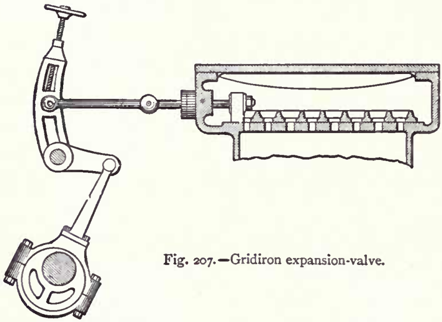


Fig. 207.—Gridiron expansion-valve.

A variable expansion-valve working on the back of a main slide-valve is shown in Fig. 208, in which the cut-off plates are placed upon a spindle having a right- and left-hand screw, by which their position on the back of the main slide-valve can be adjusted to regulate the point of cut-off at any point of the stroke of the piston. An expansion-valve prevents the expeditious starting or reversing of an engine.

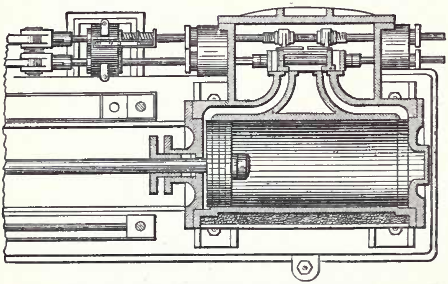


Fig. 208.—Cylinder with separate expansion-valve.

An Expansion-valve with a fixed Cut-off is shown in Fig. 209. This arrangement gives a fixed rate of expansion, and only one cut-off plate is required, which works upon the back of the main slide-valve as shown.

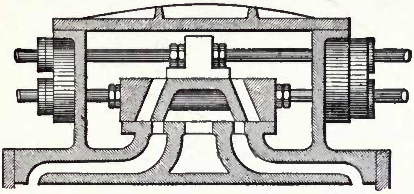


Fig. 209.—Expansion-valve with a fixed cut-off.

The action of cut-off plate can be regulated by altering the position of the eccentric on the shaft.

The Principal Points in the Motion of a Slide-Valve are four in number, viz. :—

The Admission of Steam for a certain period.

The Cut-off in the supply of steam to allow the steam already in the cylinder to propel the piston by its expansive force.

The Release of steam or opening to the exhaust.

The Compression of steam, or closing of the exhaust before the end of the stroke to cushion the steam behind the piston.

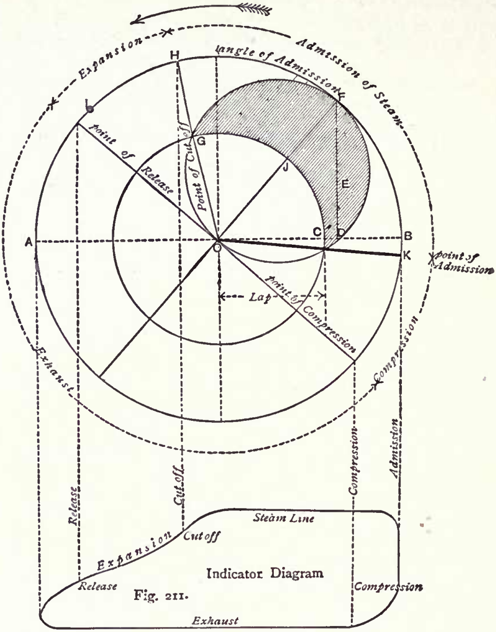
The Motion of the Slide-valve may be illustrated by a diagram* of the slide-valve shown in Fig. 210.

To find the Position of the Eccentric, the Points of Admission, Cut-off, release and compression: the travel of the valve, its lap, and lead being given.

Draw the centre-line A B, in Fig. 210; from the point O describe a circle equal in diameter to the travel of the valve; from the centre O mark the lap C and the lead D; at D erect a line E, perpendicular to A B, cutting the circle at F; join O F, and on this line draw a circle cutting the points F O. Draw a circle from O of a radius equal to the lap of the valve; draw a line from O through the point of intersection of the two circles at G to the point H in the large circle. Draw a line from O to K, which will represent the position of the crank, when the valve begins to open the port to admit steam. Then O H is the position of the cut-off, and the points of admission, release and compression are as shown. The position of the eccentric is not actually at O F., when the engine is running, as shown by the direction of the arrow, but is at an equal angular advance on the other side of the vertical centre-line, at the point I. The shaded portion of the diagram F J represents the amount of opening of the port from the point of the admission of the steam to the point of cut-off. An indicator diagram similar to that which this valve would produce is

* The above and following Diagrams of slide-valves are constructed according to Zeuner's method, described in his work, "Die Schiebersteuerungen."

Fig. 210.



Figs. 210 and 211.—Diagram of a slide-valve.

shown underneath the valve-diagram in Fig. 211, the points of admission, cut-off, release, and compression being projected from the valve diagram.

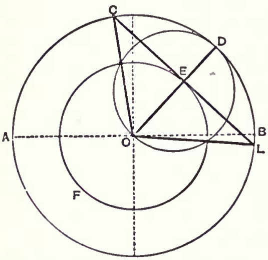


Fig. 212 —Diagram showing lap of valve and position of eccentric.

To Find the Lap of the Valve and the Position of the Eccentric: the travel of the valve, the point of cut-off and the lead being given.

Draw the centre-line A B, Fig. 212, and from the point O describe a circle equal in diameter to the travel of the valve; from B mark the lead L; mark the point of cut-off C; bisect the angle between L and C by line O D, on which draw a circle cutting the points O D, join C L. Then O E is the amount of lap of the valve required, E F is the lap-circle, and A O D is the angle between the crank and the eccentric which will give the required cut-off.

The Motion of an Expansion Slide-valve working on the back of a main slide-valve, having two cut-off plates adjustable by a right-hand and left-hand screw, for cutting off the steam at any required portion of the stroke, as shown in Fig. 213, may be illustrated by the following diagram, Fig. 214.

Diagram of an Expansion Slide-valve working on the back of a main slide-valve as shown in Fig. 213.

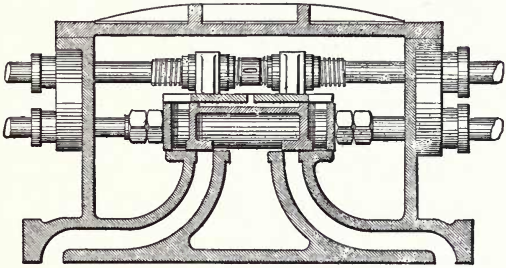


Fig. 213.—Expansion slide-valve, with variable cut-off.

From the point O on the line A H in Fig. 214, describe a circle equal in diameter to the travel of the main slide-valve, on the centre line of which at the point A describe a small circle equal in radius to the lead of the main slide-valve. Let the line A O be the position of the crank when on its dead centre, and the line O B its position at the point where the steam

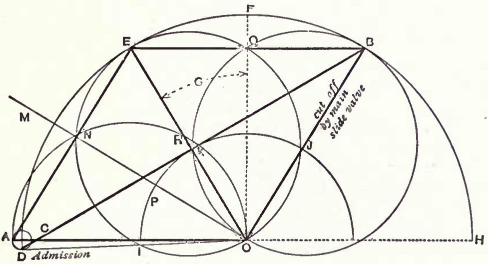


Fig. 214.—Diagram of an expansion slide-valve.

is cut off by the main slide-valve. Draw the line C from B to D, and join O D, the line O D will represent the position of the crank when the main slide-valve is on the point of admission of the steam to the cylinder. Bisect the arc A B at E and join E O, then the angle E O F = G, and the centre line of the throw of the eccentric of the main valve will have an angular advance of the crank equal to H O E or $= 90° + G$. At the centre of the line O E describe a circle equal in diameter to one-half the travel of the valve, which will intersect the lines O D at I, and the line O B at J, and it will touch the line C at K. O I and O J will equal O K, which will represent the amount of lap on the steam side of the valve required to cut off the steam when the crank is in the position O B. A circle drawn from the point O will intersect the points I J K, which is called the lap-circle. To show the travel of the valve, suppose a radial arm or line to revolve round the centre O, then the amount of the travel of the valve from its middle position will equal the length of the radial line contained in the circle described on E O. For instance, if the crank moves from A to M the valve will move from its middle position a distance equal to the length intercepted O N, part of this length O P lies within the lap-circle, and the remainder or length N P represents the width of opening of the steam-port when the crank is in the position O M. When the crank is in the position O E the port is wide open, when it is at F it is closed to the extent of F Q, and when it is at O B the valve is closed and the steam is cut off.

The Cut-off Valve is moved by an eccentric similar to the main slide, and in considering its action, the main slide may be supposed to be stationary, and the cut-off valve only movable.

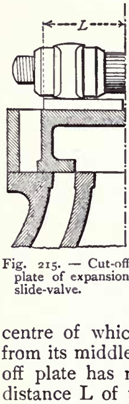


Fig. 215. — Cut-off plate of expansion slide-valve.

To find the travel and angular advance of the cut-off eccentric, and the position of the cut-off plates or length L in Fig. 215.

On the centre-line O B in Fig. 214 describe a circle equal in diameter to one-half the travel of the main slide-valve. Draw from E the line E A parallel to O B, join E B and draw from O the line O A parallel to E B intersecting the line E A at A. Then O A represents in position and length the radius of the eccentricity of the eccentric for the cut-off slide-valve, and the centre line O A of the cut-off eccentric leads that of the main eccentric O E by the angle E O A. On the line O A as a diameter describe a circle cutting the points O A as shown, which will be the cut-off-valve circle, round the centre of which a revolving arm or line will show the travel of the valve from its middle position. Thus at the position O E of the crank, the cut-off plate has moved from its middle position a distance = O R. The distance L of the cut-off-plates from the middle position, as shown in Fig. 215, will be equal to the distance E K in Fig. 214, if it were required to cut off the steam at the position O E of the crank.

The Friction of a Slide-Valve is very considerable, owing to the pressure of the steam acting on the back of the valve. The force required to move a slide-valve has, in some cases, been found to equal from $\frac{1}{4}$th to $\frac{1}{3}$rd the total pressure on the valve.

The Friction of a Slide-Valve may be reduced by reducing the area of the back of the valve exposed to the pressure of the steam in the steam chest, by means of *an equilibrating-ring* recessed into the cover of the steam-chest, having springs adjustable by set-screws, as shown in

Fig. 216. The back of the valve works steam-tight against the ring, and the space inside the ring is connected by a pipe to the condenser, so that a vacuum is maintained within the ring, and the pressure is considerably relieved from the back of the valve. The springs on the back of the valve permit the valve to leave its seat to allow the escape of water from the cylinder, in case of priming. The area of the equilibrating-ring is generally made equal to the area of the exhaust-port.

A Relief-Frame, shown in Fig. 217, is another method of relieving the pressure of the steam from the back of the valve, in order to reduce the friction. It consists of a cast-iron frame, attached to the back of the valve by means of a spring-steel diaphragm-plate; the space within the

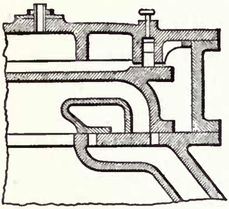


Fig. 216.—Showing equilibrating-ring on the back of a slide-valve.

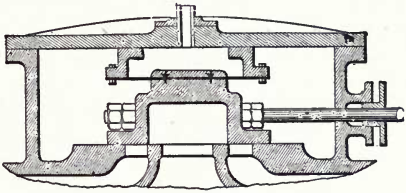


Fig. 217.—Dawe's slide-valve relief-frame.

frame at the back of the valve is connected by a pipe to the condenser, so that a vacuum is maintained within the frame, and the pressure on the back of the valve is reduced; the elastic diaphragm compensates for the wear of the rubbing surfaces.

The Relief-Frame is frequently placed on the steam-chest cover, as shown in Fig. 218, instead of on the back of the slide-valve. The frame

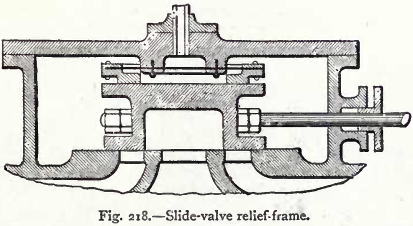


Fig. 218.—Slide-valve relief-frame.

is pressed against the back of the valve by a spring-steel diaphragm, the back of the valve working steam-tight against the frame. An enlarged view of a similar arrangement attached to a door in the cover of the valve-chest is shown in Fig. 219.

A Slide-Valve Relief-Frame of another kind is shown in Fig. 220. The valve has a circular spigot cast on the back, fitted with rings like a

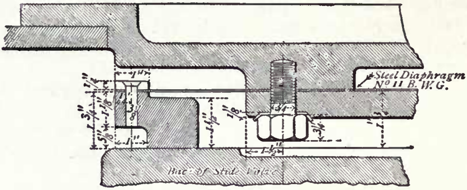


Fig. 219.—Slide-valve relief-frame on steam-chest cover.

piston to keep it steam-tight, which fits into a socket attached by a spring, in the form of a cross, to the back of the slide-valve. The spring presses the socket against the steam-chest cover, and the steam-pressure is partly relieved from the back of the valve, which is thereby nearly balanced.

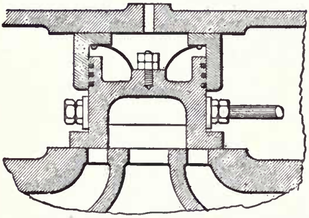


Fig. 220.—Slide-valve relief-frame on back of slide-valve.

As devices of this kind are never perfect, slide-valves are never perfectly balanced, but are only relieved of the greater portion of the pressure of the steam on their backs.

A Balanced Slide-Valve of another kind is shown in Fig. 221. It consists of a relief-ring fitted into a short cylinder attached to the cover of the valve-chest. The flange outside the relief-ring forms the bottom of the recess B, which is packed with gasket, pressed down a junk-ring by means of the screws C, which pass

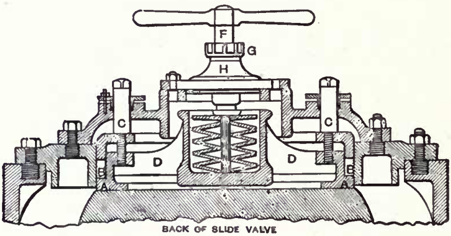


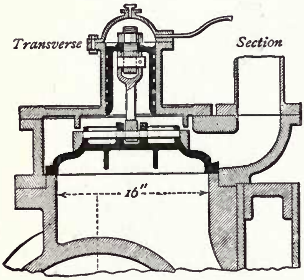
Fig. 221.—Claparede's balanced slide-valve.

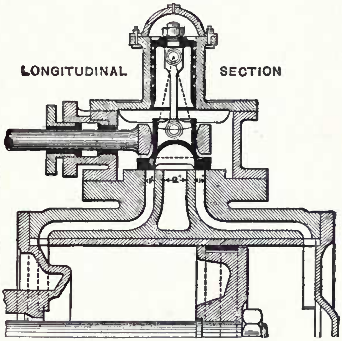
through stuffing-boxes in the cover of the valve-chest, so that they can be tightened when the engine is working. The balancing space D, or that to which steam cannot get access, is placed in communication with the atmosphere by a small cock. The valve is kept to its seat by the spring E, the tension of which can be regulated by the screw F, which passes through a stuffing-box G, on the lid H.

A Balanced Slide-Valve, applied to a locomotive engine, is shown in Figs. 222 and 223. The slide-valve is attached by a link to a balance-

Fig. 223.

Fig. 222.





Figs. 222 and 223.—Urquhart's balanced slide-valve applied to a locomotive-engine.

piston fitted with piston-rings to make it steam-tight, and a pipe is fixed to the balance-cylinder to carry off any steam which may escape past the piston.

A Balanced Slide-Valve of another kind, for locomotive engines, is shown in Figs. 224 and 225. It consists of a rectangular frame, formed of

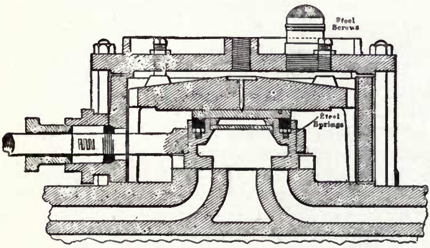


Fig. 225.
Section of valve.

Fig. 224.—Delancey's balanced slide-valve.

strips of metal $\frac{3}{4}$ inch square, fitted into grooves in the back of the valve; the strips are halved on to each other at the corners, and rest on springs fixed to the bottom of the grooves. A crown-plate, forming the top of the valve, rests on the top of the strips. The crown-plate works against an adjustable frame fixed in the steam-chest; the frame rests on feet at its four corners, therefore the steam-pressure on its top does not affect the slide-valve. The small holes through the top of the slide-valve, shown in Fig. 225, are to allow any leakage of steam past the strips to escape to the exhaust cavity of the valve.

The Air or Relief-Valve shown on the steam-chest in Fig. 224 is shown in enlarged views in Figs. 226 and 227. It opens inwards, and

Fig. 226.

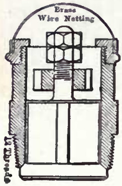
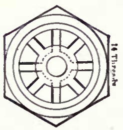


Fig. 227.



Figs. 226 and 227.
Air-valve for steam-chest.

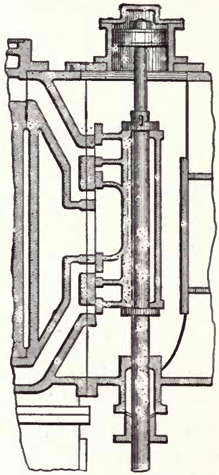


Fig. 228.—Balanced slide-valve of a marine engine.

admits the atmosphere when a vacuum is formed in the steam-chest, as it is found that, when the regulator is closed to shut off the steam, dust and ashes are liable to be sucked down the exhaust-pipe and are likely to cut the face of the slide-valve.

The Slide-Valves of Vertical Engines are usually balanced by a piston fixed on the top of the valve-spindle, working in a small cylinder on the top of the steam-chest, as shown in Fig. 228, to which steam is supplied from the steam-chest; the pressure of the steam on the balance-piston balances the valve, the area of the balance-cylinder being arranged to suit the weight of the slide-valve, rods, and gear.

The Strain on a Valve-Spindle in moving a Slide-Valve depends upon the force with which the valve is pressed against the face of the cylinder by the steam in the steam-chest. This force is equal to the product of the pressure of the steam by the area of the back of the valve exposed to the pressure of the steam.

Example: The size of the back of the valve exposed to the pressure of steam of 60 lbs. per square inch, is 11 inches by 16 inches. Required the strain on the valve-spindle due to the pressure of steam on the back of the valve.

Then $11 \times 16 = 176$ square inches, the area of the back of valve, and $60 \times 176 = 10560$ lbs., the total pressure on the back of the valve. Taking the frictional resistance at 10 per cent. of the pressure, it will give $\frac{10560 \times 10}{100} =$ 1056 lbs. frictional resistance, or strain on the valve-spindle.

Where the friction due to either a large size of slide-valve, or to a high pressure of steam upon it, would be great, it is better to adopt a piston-valve, instead of a slide-valve with a relief-frame.

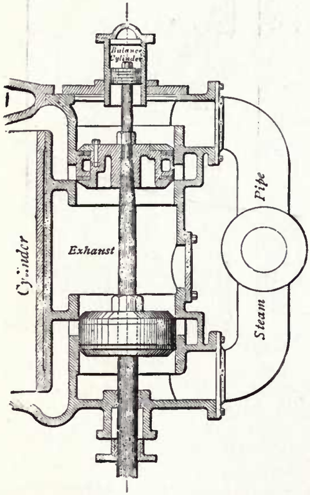


Fig. 229.—Piston-valve of a marine-engine.

Piston-Valves work in equilibrium, as the pressure of steam cannot force the valves against the sides of the casing in which they work. A piston-valve consists of a spindle or socket having a piston at each end; steam is admitted from the outside of the valve, and it exhausts into the space between the pistons. Small piston-valves are fixed on a solid spindle, and the ends of the valve-chest are connected by a pipe, as shown in Fig. 229. Large piston-valves are attached to a hollow socket, through which the steam can pass from end to end of the valve-chest. The pistons are fitted with rings of cast-iron or gun-metal, like an ordinary piston; a simple arrangement of rings is shown in Fig. 230, in which there is one outside ring and one inside spring-ring of cast-iron. The rings are prevented from springing into the ports by diagonal bars placed across the ports, as shown in Figs. 231 and 232. The diameter of a piston-valve is frequently made equal to one-half the diameter of the steam-cylinder.

Fig. 230. Fig. 231.

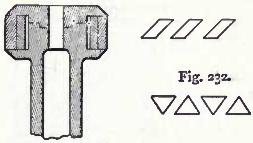


Fig. 232.

Figs. 230—232.—Piston of piston-valve: and cylinder-ports.

Thom's Piston-Valve, shown in Figs. 233—235, differs from ordinary piston-valves in that its working face is provided with a passage similar to the passage in a trick-valve, but having positive and negative exhaust-lap at the top and bottom of the cylinders, so that the negative exhaust-lap forms a communication between the opposite ends of the cylinders just before exhausting to the condenser, and so that steam at its terminal pressure is transferred from one side of the cylinder-piston to the other through the passage, such steam being then compressed nearly up to the initial

Fig. 233.

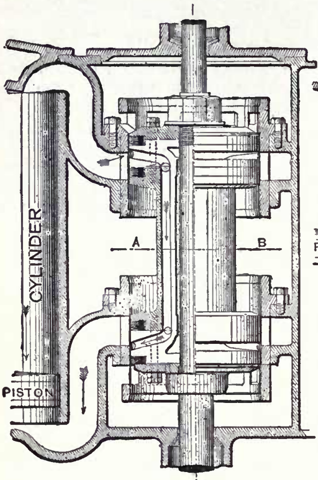


Fig. 234.

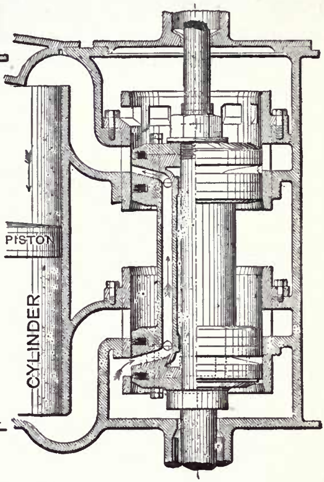
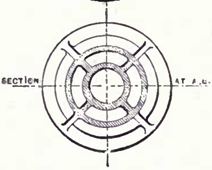


Fig. 235.



Figs. 233—235.—Thom's piston-valves applied to a marine-engine.

pressure and used over again on the return stroke of the engine, and causing the engines at the same time to turn the centres, especially that of the low-pressure cylinder of compound engines, with less shock, due to having steam in the cylinder to compress.

In Fig. 233 the valve is shown in the position in which a communication is formed between the two ends of the cylinder for a very brief period, the result being that a portion of the exhaust steam first released passes to the other end of the cylinder and assists in cushioning. Fig. 234 shows how the connecting-passage also serves to afford a double inlet for the steam, in the same manner as in the trick-valve. Fig. 235 is a section through the valve at A B in Fig. 233.

Double-beat Valves, shown in Fig. 236, work nearly in equilibrium, as the pressure of the steam is nearly equal on both the inside and outside of the valve. They are frequently used in place of slide-valves in low-speed engines, but they are difficult to keep steam-tight on account of the unequal expansion of the valves and seatings. This kind of valve is fully open when it is lifted a distance equal to one-fourth the diameter of the valve. Double-beat valves are also made in the form of two mushroom-shaped valves of different diameters attached to a spindle. If D be the difference of the areas of the valve-seats of such a valve, then, $\frac{\text{load in lbs. on the valve}}{\text{D}} =$ the pressure on the valves in lbs. per square inch, and D × pressure in lbs. per square inch = the load in lbs. on the valve.

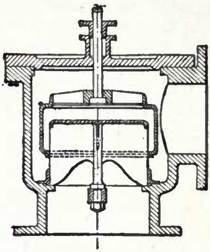


Fig. 236.—Double-beat valve.

Example: In a double-beat valve the internal diameters of the two valve-seats are 6 inches and $4\frac{1}{2}$ inches respectively. The load on the valve, including the weight of the valve, is 108 lbs. What is the pressure on the valve? Then, the difference of the areas of the valves is $(6^2 - 4\frac{1}{2}^2) \times \cdot 7854 = 12\cdot37$ square inches, and $108 \div 12\cdot37 = 8\cdot73$ lbs. per square inch, the pressure on the valve.

Corliss-Valves, shown on page 409, are disc-shaped. The valve-gear of a Corliss-engine, having a steam-jacketed cylinder 20 inches diameter, and 4 feet length of stroke, is shown in Fig. 237, which exhibits the novel feature of opening the port wide during the first tenth of the stroke, and keeping it full open until it is tripped. Thus, for all grades of expansion beyond and including one-tenth, there can be no wire-drawing whatever due to a contracted port. The valve-gear is driven by a side-shaft actuated by a bevel wheel on the crank-shaft. On this side-shaft there are two cams, one for each steam-valve. On the top of each cam there rides an arm or bracket projecting from a casting which carries the tripping-gear, and slides upon a vertical rod connected to the valve. This casting rises and falls with the cam, the upward motion commencing when the engine is upon the centre, and being completed by the time the piston has

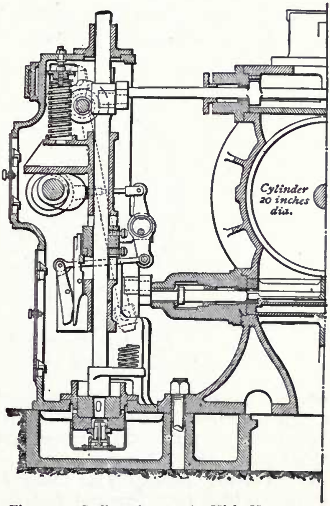


Fig. 237.—Corliss valve-gear by Hick, Hargreaves & Co., Bolton.

finished one-tenth of the stroke. There is then a "dwell" or pause due to a circular portion of the cam, and the casting remains elevated until it is again lowered by the cam. But in the meantime the trip-gear comes into action, and breaking the connection between the casting and the vertical rod upon which it slides, allows the valve to close with the suddenness peculiar to the Corliss valve-gear. The tripping-motion is worked by an eccentric on a side-shaft through two links and a rocking-lever. It consists of a block or catch on the vertical rod, a catch-lever which engages with this block, and a second lever which trips the first. The point of cut-off is determined by the governor, which turns a small rocking-shaft as it rises and falls. The rocking-lever, referred to above as forming part of the tripping-gear, is mounted upon this shaft, but not directly, an eccentric-bush being first keyed upon the shaft. The result of this arrangement is that when the governor rises, the centre of the lever is moved in such a way that the acting end of the tripping-lever is brought nearer to its work, and comes into contact with its companion-lever sooner than before, cutting off the steam at an earlier point. The vertical rod is connected to the lever of the steam-valve by a block working in a slotted crosshead, and is moved to close the valve by a spring, a dash-pot preventing any concussion. The exhaust-valves are also worked from the side-shaft by eccentrics.

The Motion of an Engine may be reversed by altering the position of the eccentric on the crank-shaft. This may be effected by attaching the

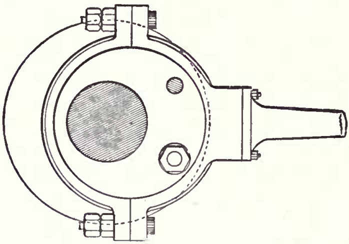


Fig. 238.—Reversing-plate for eccentric.

eccentric to a plate fixed on the crank-shaft, as shown in Fig. 238. The eccentric is loose on the shaft, and is bolted to the reversing-plate with a bolt, sliding in a slot, thus enabling the position of the eccentric to be shifted to a similar position on the opposite side of the crank, thereby reversing the motion of the engine. This arrangement is used for portable and other engines which only required to be occasionally reversed. It necessitates stopping the engine for reversing, and it is not so convenient as a link-motion.

Link-motion is used to effect the reversal of the motion of an engine. A link-motion with a shifting link is shown in Fig. 239. The crank is shown in its position when the piston is at the end of the stroke. Two eccentrics are keyed on the crank-shaft, the top one being the forward eccentric, and the bottom one the backward eccentric. The bottom of the link is connected to a lever, by means of which the link is moved up or down as required. When the link is lowered so as to bring the forward

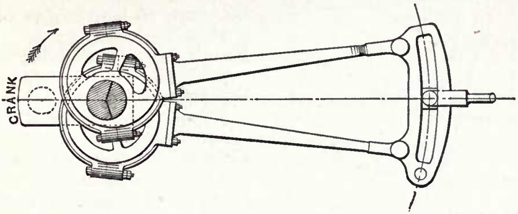


Fig. 239.—Link-motion of a horizontal-engine.

eccentric-rod in line with the valve-spindle, the motion of the slide-valve is governed by the forward eccentric, and the engine goes forward. On the contrary, when the link is raised so as to bring the backward eccentric-rod in line with the valve-spindle, the motion of the slide-valve is governed by the backward eccentric, and the motion of the engine will be reversed. Only one eccentric works the valve at a time, and when the middle of the link is in line with the valve-spindle, the motion of neither eccentric is communicated to the slide-valve. As the link is moved from the central position it causes more and more steam to be admitted to the cylinder, and when the link is shifted to bring the block up to the end of the link, full steam is admitted. The distance between the centres of the eccentric-rod-pins of the link is usually about three times the maximum travel of the valve, and the radius of the link is struck from the centre of the crank-shaft.

The Expansion-Link of Locomotive Engines is generally either supported at the centre as shown in Fig. 240, or at the top from the pin of

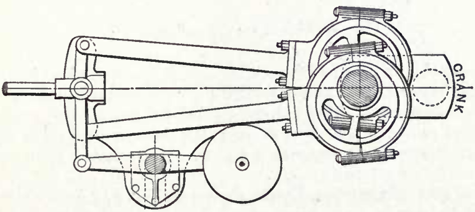


Fig. 240.—Link-motion of a locomotive-engine.

the forward eccentric-rod, as shown in Fig. 241, the reversing shaft being below the motion and behind the link. The usual proportions of the valve-motion of locomotives are as follows :—

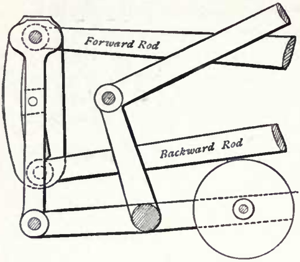


Fig. 241.—Link-motion of a locomotive-engine.

Length of expansion-link from centre to centre of pins = 17 inches.
Thickness of expansion-link = $2\frac{1}{4}$ inches.
Diameter of motion-pins = $1\frac{1}{2}$ or $1\frac{3}{4}$ inch.
Radius of expansion-link = 4 feet, 7 inches.
Diameter of reversing-shaft = $3\frac{1}{2}$ inches at the centre and 3 inches at the bearings.
Length of eccentric rod = 4 feet, 7 inches, centres.
Diameter of eccentric-sheaves = 16 inches.
Width of eccentric-sheaves = $2\frac{7}{8}$ inches.
Throw of eccentrics = $6\frac{1}{2}$ inches.
Angle of forward eccentric = $103\frac{1}{2}°$.
Angle of backward eccentric = $104°$.
Lead of slide-valve = $\frac{3}{32}$ inch back and $\frac{5}{32}$ inch front.
Lap of slide-valve = 1 inch.
Travel of slide-valve = $4\frac{5}{16}$ inches.

The Stationary Expansion-Link, shown in Fig. 242, is another form of link-motion : the link is supported in a fixed position as shown, the changing of the gear being effected by moving up and down the expansion-

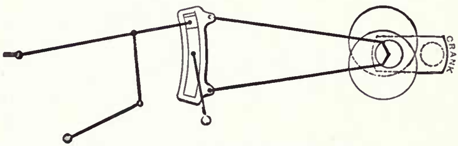


Fig. 242.—Link-motion with stationary-link.

link a radius-rod connected to the sliding block at one end and to the valve-spindle at the other end. The curvature of the expansion-link is in the reverse direction to the shifting expansion-link previously described, to adapt it to the radial movement of the radius-rod of the valve-spindle ; the radius-rod of the valve-spindle is supported and moved by levers on the reversing shaft in front of the expansion-link.

The Straight Expansion-Link, shown on Fig. 243, has parallel instead of curved sides, and is something like a combination of the shifting-link and the stationary link. The link and the radius-rod, connecting the link-block and the valve-spindle, are supported by links connected to the ends of arms of unequal length on the reversing-shaft as shown. The expansion-link and the radius-rod are shifted in opposite directions, one being moved upwards and the other downwards, by one movement of the revers-

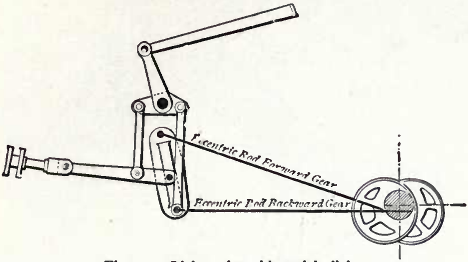


Fig. 243.—Link-motion with straight link.

ing shaft. The advantage of this form of link-motion is, that by moving two pieces of the link-motion, the reversing of the engine may be effected with only one-half the vertical movement necessary when the movement is applied to only one of the pieces, as in the case of the ordinary link-motion shown in Fig. 239.

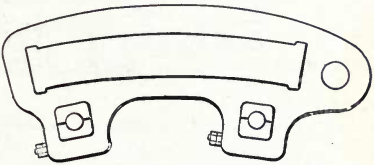


Fig. 244.—Expansion-link of a marine-engine.

Expansion - Links of Marine-Engines, when a single link is used, are generally provided with adjustable bushes, as shown in Fig. 244, and the sliding block is also bushed, to facilitate repairs. Expansion-links of marine-engines are frequently made double, as shown in Fig. 245. The sliding-block moves between two radius-links; the ends of the eccentric-rods are forked to span the link; and all the bearings are provided with adjustable bushes.

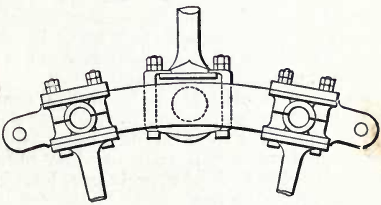


Fig. 245.—Expansion-link of a marine-engine.

Linking-up means altering the working position of the link, whereby the point of cut-off is altered and the steam is worked expansively. When the position of the link-block is moved from the extremity towards the centre or dead-point of the link, the travel of the valve is shortened, and it is equivalent to diminishing the throw of the eccentric and increasing its angular position, so that the points of lead and cut-off are earlier.

Eccentrics and Eccentric-Straps for working the Slide-Valve of a Locomotive Engine, by means of the link-motion previously described, are shown in Figs. 246 and 247. The eccentric-sheaves are made in two

Fig. 246.

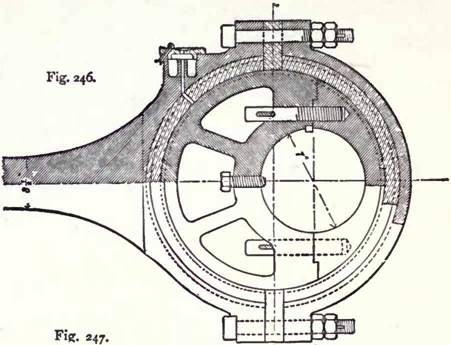
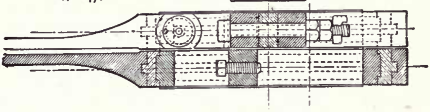


Fig. 247.



Figs. 246 and 247.—Eccentrics and eccentric-straps of a locomotive-engine.

pieces let into each other and bolted together as shown; the larger piece of the sheave is generally made of cast-iron and the small piece of wrought-iron. The eccentric-strap is made of wrought-iron, lined with gun-metal tongued and grooved into the eccentric-strap. The eccentric has a projection on its circumference which works between two flanges on the lining of the eccentric-strap. The eccentric-rod is flat in section and is forged solid with one-half of the eccentric-strap.

An eccentric-strap for a locomotive engine, made of cast-iron,—the pattern used on the London, Brighton and South Coast Railway—is shown in Fig. 248. The sheaf is of cast-iron, and the wear and tear is very slight, as cast-iron works well upon cast-iron when efficiently lubricated. The eccentric-rod is made of wrought-iron, bolted to the strap by studs screwed into the strap as shown.

An Eccentric and Eccentric-strap for a Stationary Engine is shown in Figs. 249—252. The eccentric is either made solid as shown in Fig. 249, or in two pieces let into each other and bolted together as shown in Fig. 251, the larger piece of the sheave being made of cast-iron and the smaller piece of wrought-iron. A number of plugs of anti-friction metal are let into the circumference of the wrought-iron piece for the purpose of reducing friction and assisting lubrication. The eccentric-strap is made of cast-iron, and is kept in its place on the sheave, by

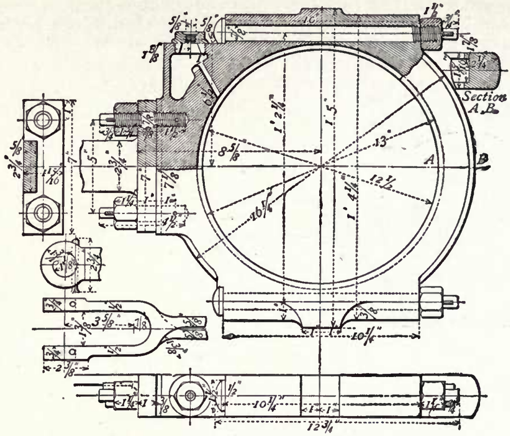


Fig. 248.—Eccentric-strap of a locomotive-engine.

a projection on the sheave which works in a groove in the eccentric-strap. The eccentric-rod is made of wrought-iron, bolted into a recess on the side of the strap, as shown.

An eccentric and eccentric-strap for a stationary engine in which the rod is provided with a flange and bolted to the end of the strap is shown in Figs. 253 and 254; the strap is made of cast-iron, lined with gun-metal; the eccentric-sheaf is

Fig. 249.

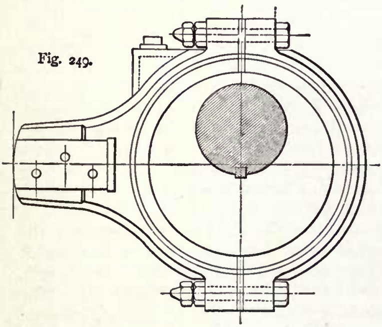


Fig. 250.

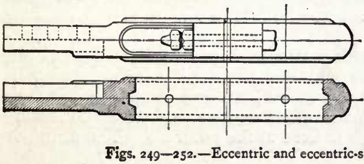


Fig. 251.

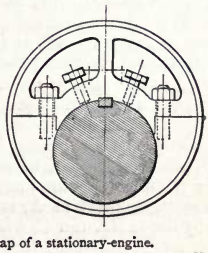


Fig. 252.

Figs. 249—252.—Eccentric and eccentric-strap of a stationary-engine.

recessed on each side to receive the flanges of the eccentric-strap as shown.

A Set of Eccentrics and Eccentric-Straps for a Marine-Engine is shown in Fig. 255; the sheaves are of solid cast-iron, but they are fre-

Fig. 253.

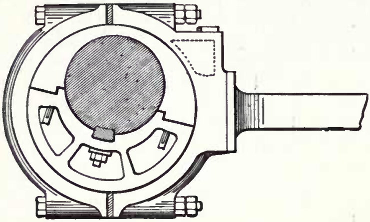


Fig. 254.

Figs. 253 and 254.—Eccentric and eccentric-strap of a stationary-engine.

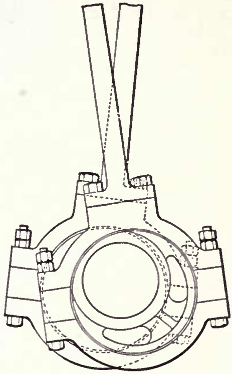


Fig. 255.—Eccentrics and eccentric-straps of a marine-engine.

quently made in halves and bolted together, as previously described for stationary engines. The eccentric-straps are of solid gun-metal, or of wrought-iron or steel lined with gun-metal.

Joy's Reversion and Expansion Valve-Gear.—The essential feature of this valve-gear is that the movement for the valve is produced by a combination of two motions at right angles to each other, and by the various proportions in which these are combined; and by the positions in which the moving parts are set with regard to each other, it gives both the reversal of motion and the various degrees of expansion required. Eccentrics are dispensed with; the motion is taken direct from the connecting-rod, and by utilising independently the backward and forward action of the rod, due to the reciprocation of the piston, and combining this with the vibrating action of the rod, a movement results which is suitable to work the valves of engines allowing the use of any proportions of lap and lead desired, and giving an almost mathematically correct "cut-off" for both sides of the piston and for all points of expansion intermediately, as well as a much quicker action at the points of "cut-off" and "release" than is given by a link gear.

The machinery for accomplishing this is less complicated than the ordinary link-motion, and is shown in elevation in Fig. 256. Here E is the main valve-lever, pinned at D to a link B, one end of which is fastened to the connecting-rod at A, and the other end maintained in about the vertical by the radius-rod C. which is fixed at the point C1. The centre or fulcrum F of the lever E, partaking of the vibrating movement of the connecting-rod at the point A, is carried in a curved slide J, the radius of which is equal to the length of the link G, and the centre of which is fixed to be concentric with the fulcrum F of the lever when the piston is at either extreme end of its stroke. From the upper end of the lever E the motion is carried direct to the valve by the rod G. It will be evident thus that by one revolution of the crank the lower end of the lever E will have imparted to it two different movements, one along the longer axis of the ellipse, travelled by the point A, and one through its minor axis up and down, these movements differing as to time, and corresponding with the part of the movement of the valve required for lap and lead, and that part constituting the port opening for admission of steam.

The former of these is constant and unalterable, the latter is controllable by the angle at which the curved slide J may be set with the vertical.

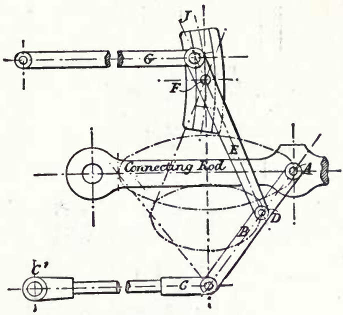


Fig. 256.—Joy's valve-gear.

It will further be evident that if the lever E were pinned direct to the connecting-rod at the point A, which passes through a practically true ellipse, it would vibrate its fulcrum F unequally on either side of the centre of the curved slide J by the amount of the versed line of the arc of the lever E from F D; it is to correct this error that the lever E is pinned at the point D to a parallel motion formed by the parts B and C, the point D performing a figure which is equal to an ellipse, with the error to be eliminated added, so neutralising its effect on the motion of the fulcrum F.

Thus the "lap" and "lead" are opened by the action of the valve-lever acting as a lever, and the port-opening is given by the incline of the curved slide in which the centre of that lever slides, and the amount of this opening depends upon the angle given to that incline. Consequently, when these two actions are in unison, the motion of the valve is very rapid, and this occurs when the steam is being admitted. Then follows a period of opposition of these motions, during which time the valve pauses momentarily, this corresponding to the time when the port is fully open. Further periods of unison follow, at which time the sharp "cut-off" is obtained.

The "compression" resulting with this gear is also reduced to a minimum, owing to the peculiar movement given to the valves (*i.e.*, the series of acclerations and retardations referred to), as while the "lead" is obtained later and quicker, the port is also shut for "compression" later and quicker, doing away with the necessity for a special expansion-valve, and allowing the main valve to be used for expansion, as the "compression" is not of an injurious amount, even with a "cut-off" reduced to 15 per cent., or about $\frac{1}{6}$th of the stroke.

Joy's Valve-Gear for Triple-Expansion Engines, is shown in

Figs. 257 and 258, in which, instead of employing three distinct sets of valve-gears, one for each cylinder, a valve-gear is fitted to each outer cylinder, and these two are united by a floating lever, motion being taken for the valve of the third cylinder from about the middle of this lever, this motion being found to be correct in its character, for the purpose, and is under the same control for reversing and for expansion as the other two

Fig. 257.

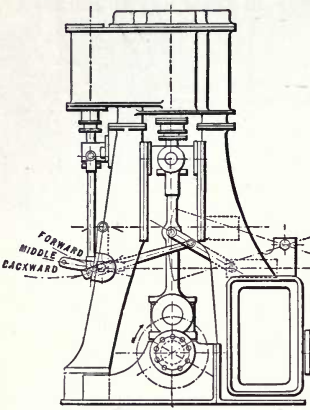
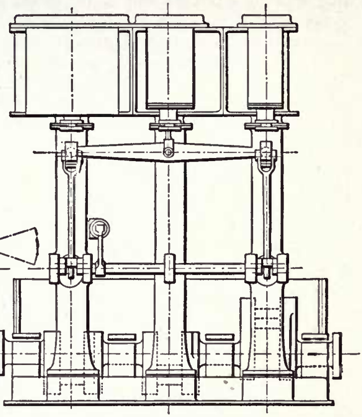


Fig. 258.



Figs 257 and 258.—Triple-expansion engines with Joy's valve-gear.

valve-motions. Being, moreover, the resultant of the other two motions, it is a mean between them, so that the high-pressure cylinder may be set with a cut-off of, say, ·65, and the low-pressure at ·55; the resultant motion for the medium-pressure cylinder will be ·60. These proportions may be varied to suit circumstances, or all the valves may be set to cut-off alike, and within considerable limits also the cut-offs of the high and low pressure cylinders may be varied independently of each other to equalise the strains on the cranks, without affecting appreciably the cut-off of the intermediate cylinder. Thus a considerable saving in complication of the triple-cylinder engine is effected, a complete set of valve-gear being replaced by a single lever.

Bremme's Valve-gear, shown in Figs. 259—261, is worked from only one eccentric, and it has consequently fewer working joints than a link-motion.

Figs. 259 and 260 illustrate the mechanism of this valve-gear in its most elementary form. GH is a swinging link or radius-rod, pivoted at G to a bracket, and at H to an arm projecting from the hoop of the single eccentric. The bracket itself is centred at F, in the same horizontal plane as the centre line of the shaft C, and can be partially rotated around that centre by a worm or worm-segment (Fig. 260). The two extreme positions of the bracket are shown by dotted lines, and correspond to the two directions of motion of the engine.

A modification in the arrangement of this valve-gear is shown in Fig. 261, which represents the engines of a steam-yacht, in which, instead of the toothed-bracket FG, a double bar is used, which is carried by a lever at one end and by a curved slot at the other—the radius of each of these being the same as FG or GH (Figs. 259 and 260). The centre, G, of the swinging link or radius-rod GH, is thus, in reversing or varying the expansion, carried in the arc of a circle around the centre F, with a radius equal to FG or GH. When the crank is on the dead point at top and bottom, then F and H coincide, so that shifting the bracket does not then move the valve.

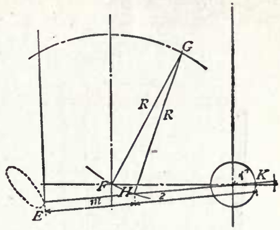


Fig. 259.—Diagram of Bremme's valve-gear.

The Bremme valve-gear gives at all grades of cut-off a uniform lead.

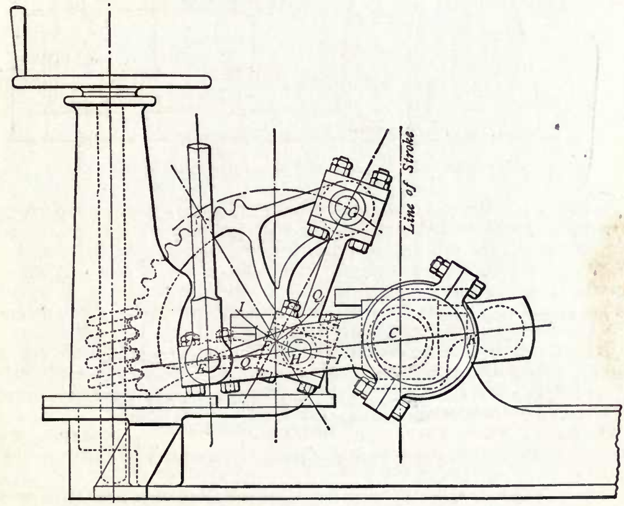


Fig. 260.—Bremme's valve-gear.

In reversing or altering the expansion there is no side-thrust put on the valve-spindle. It is simple in its action, and has very few working parts.

A well-made link-motion for one cylinder has two eccentrics, four double joints, and one single joint constantly working—or eleven parts in all;

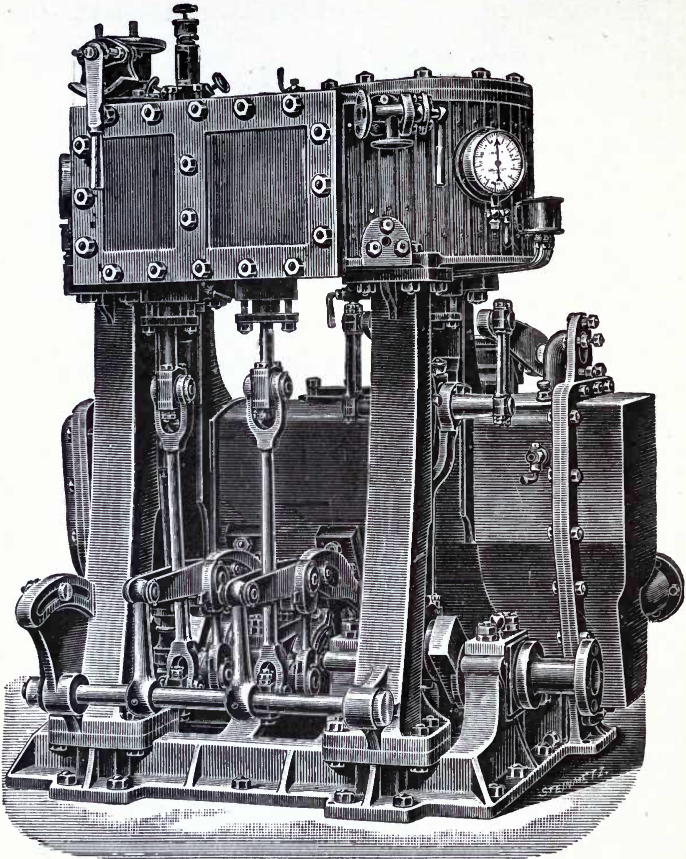


Fig. 261.—Yacht-engines with Bremme's valve-gear, by Ross and Duncan, Glasgow.

whereas a Bremme-gear for one cylinder, when constructed in its best form, as shown in Fig. 261, has one eccentric, and four single joints constantly working—or five in all.

SECTION IV.

CONNECTING-RODS, COUPLING-RODS, CRANK-SHAFTS, CRANK-AXLES, SCREW-PROPELLER-SHAFTING AND BEARINGS, SCREW-PROPELLERS, PADDLE-WHEELS, AND JET-PROPELLERS, ETC.

SECTION IV.

CONNECTING-RODS, COUPLING-RODS, CRANK-SHAFTS, CRANK-AXLES, SCREW-PROPELLER-SHAFTING AND BEARINGS, SCREW-PROPELLERS, PADDLE-WHEELS, AND JET-PROPELLERS, ETC.

CONNECTING-RODS, COUPLING-RODS, CRANK-SHAFTS, SCREW-PROPELLER SHAFTS, AND HOLLOW SHAFTING.

Connecting-Rods for Marine-Engines are usually made equal in length to from twice to three times the length of stroke of the engine; the usual forms of the ends of the rod are shown in Figs. 262—265. The

Fig. 262.

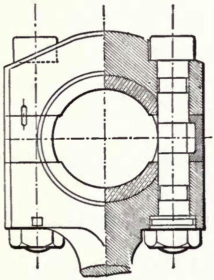
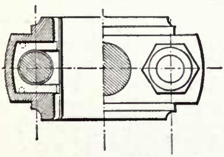


Fig. 263.



Figs. 262 and 263.—Marine-engine connecting-rod.

Fig. 264.

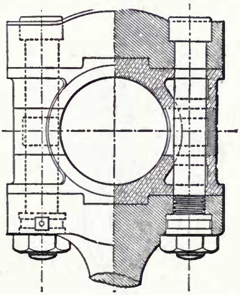
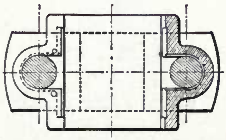
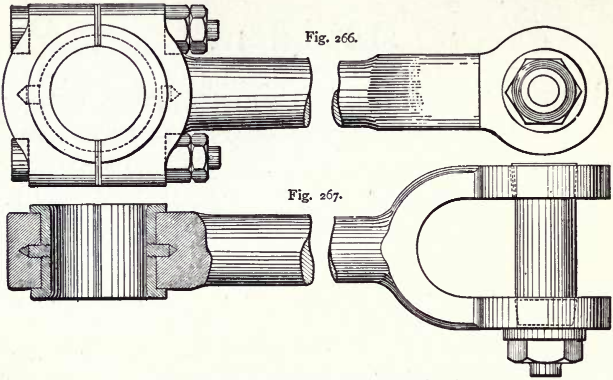


Fig. 265.



Figs. 264 and 265.—Marine-engine connecting-rod.

strongest is that shown in Fig. 262, in which one-half of the end is forged solid with the rod, which forms a firm support for the cap-bolts. Bushes,



Figs. 266 and 267. – Connecting-rod with fork-end.

having packing-pieces, or liners for adjustment, are fitted and secured by a cap bolted to the solid end of the rod; the bushes are sometimes lined with anti-friction metal. The connecting-rod shown in Fig. 264 terminates in a T-piece, into which a gun-metal block, forming the bushes, is partly recessed; the bushes are retained by the cap and bolts as shown. The nuts on the bolts are each provided with a collar, recessed into the metal as shown, and a set-screw is fitted into a groove in the collar, by which the nut can be secured and prevented from working loose.

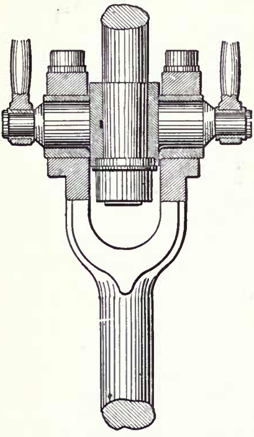


Fig. 268. Fork-end of a connecting-rod.

A Connecting-Rod with a Fork-end is shown in Figs. 266 and 267, the end being forked to span the cross-head; the large end of the rod has adjustable bushes with caps and bolts. The fork-end carries the pin for the crosshead; the pin is fitted into tapered holes in the jaws of the fork-end. Another arrangement of the fork-end of a connecting-rod is shown in Fig. 268; it is provided with caps, bolts, and adjustable bushes, the ends of the cross-head pin are fitted with links for working the air-pump lever of a marine-engine.

The Diameter of the Connecting-Rod at the Centre may be found by the following formula:—

Let D = the diameter of the cylinder in inches.
P = the initial absolute pressure of the steam in lbs. per square inch.
C = the diameter of the connecting-rod at the centre, in inches.

$$\text{Then } C = \frac{D}{55}\sqrt[2]{P}$$

Example: Required the diameter of the connecting-rod at the centre, for a marine-engine with a cylinder 42 inches diameter; the initial absolute pressure of the steam being 90 lbs. per square inch.

Then $42 \div 55 = \cdot 76$, and $\sqrt[2]{90} = 9\cdot 49 \times \cdot 76 = 7\cdot 22$ inches, the diameter at the centre of the connecting-rod.

The Diameter of the Connecting-Rod at the End may be found by the following formula, where the notation is the same as in the previous formula.

Let E = the diameter at each end of the connecting-rod.

$$\text{Then } E = \frac{D}{60}\sqrt{P}$$

Example: Required the diameter of the ends of the connecting-rod given in the previous example.

Then $42 \div 60 = \cdot 7$ and $\sqrt[2]{90} = 9\cdot 49 \times \cdot 7 = 6\cdot 65$ inches, the diameter of each end of the connecting-rod.

The Diameter of the Cap-Bolts for the Connecting-Rod shown in Figs. 262—266 may be found approximately by the following formula, where the notation is the same as in the two previous examples.

Let B = the diameter in inches of each cap-bolt.

$$\text{Then } B = \frac{D}{125}\sqrt[2]{P}$$

Example: Required the diameter of the bolts for the connecting-rod given in the previous example.

Then $42 \div 125 = \cdot 336$ and $\sqrt[2]{90} = 9\cdot 49 \times \cdot 336 = 3\cdot 18$ inches, the diameter of each bolt. The diameter of the bolt-head should be one-half larger than the bolt.

The Centres of the Cap-Bolts of the Connecting-Rod may be = the diameter of the neck plus from 1·25 to 1·4 diameter of the bolt; they are generally placed as closely as possible.

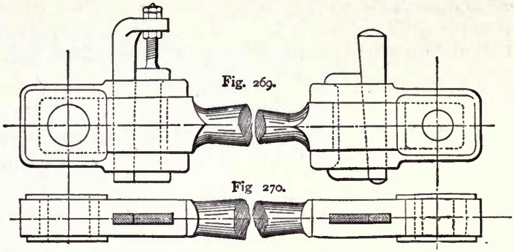
The Thickness of the Cap of the Connecting-Rod may be equal to one-half the diameter of the bearing.

A Connecting-Rod for a Stationary Engine, the ends having a strap to secure the bushes, and gib and cotter, are shown in Figs. 269 and 270.

When the cotter is tightened to take up the wear of the brasses, the con-

Fig. 269.

Fig 270.



Figs. 269 and 270.—Connecting-rod of a stationary-engine.

necting-rod is shortened in length. The taper of the cotter is usually from $\frac{3}{8}$ to $\frac{1}{2}$ inch per foot. In the large end of the connecting-rod shown in

Fig. 271.

Fig. 272.

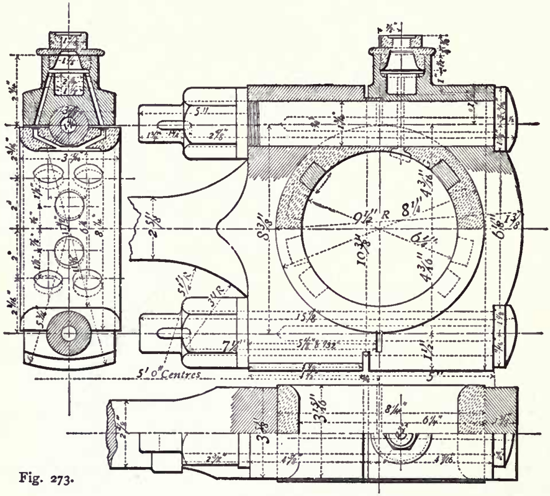


Fig. 273.

Figs. 271—273.—Large end of the connecting-rod of a locomotive-engine.

Fig. 269 the cotter is locked by nuts on a screw forged on the head of the gib as shown, and in the small end the cotter is locked by a steel set-screw.

The number of turns required to be given to the nut on the screw of the cotter to draw the brasses a given degree closer, may be ascertained as follows:—Supposing the screw to have 14 threads per inch, and the taper of the cotter = $\frac{1}{2}$ inch per foot, it is required to find the number of turns to be given to the hexagon nut to draw the brasses $\frac{1}{16}$ inch closer.

Then $\dfrac{\cdot 504 \text{ inch taper of cotter}}{14 \text{ threads} \times 12 \text{ inches}} = \cdot 003$ inch for one turn of the nut, the required adjustment of brasses $= \frac{1}{16}$ or $\cdot 0625$ inch, and $\dfrac{\cdot 0625}{\cdot 003} = 20\cdot 8$ turns. As the nut has six cants, it will require 20 turns and $\cdot 8 \times 6 = 4\cdot 8$ cants, or say 5 cants, to draw the brasses $\frac{1}{16}$ inch closer.

A Locomotive-Engine Connecting-rod, the pattern used on the Brighton Railway, is shown in Figs. 271—273. The large end of the connecting-rod is fitted with a cap and bolts. The bolts are bored up for the greater part of their length so as to reduce their sectional area to that of the screwed portion, and thus secure equal elasticity. With these long bolts it is not found necessary to employ lock-nuts; a small cotter is fitted in the ends of the bolts.

The small end of the connecting-rod, shown in Figs. 274 and 275, has a strap secured with a cotter and bolt, the butt-end of the rod is recessed to receive the brass-bush as shown. The taper of the cotter is 1 in 6.

A locomotive engine connecting-rod of another design, the pattern used on the London and North-Western Railway, is shown in Figs. 276—279. The large end of the connecting-rod, shown in enlarged view, Fig. 278, is open-ended, and secured at the end with a bolt passing through a block, which forms an abutment for the bush in front of it. The brasses are adjustable by a cotter having double nuts at each end. Strips of anti-friction-metal are let into the brasses, as shown in Fig. 281. The middle of the connecting-rod is flat-sided, as shown in Fig. 277; a boss is formed on the rod, which is bushed to receive a pin for coupling a link to work a rod connected to Joy's valve-gear. The small end of the rod is solid, without strap, and is bushed, as shown in Fig. 280.

Fig. 274.

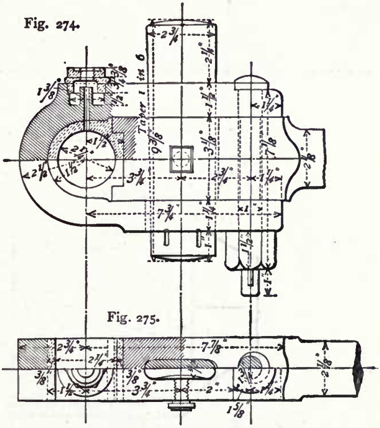
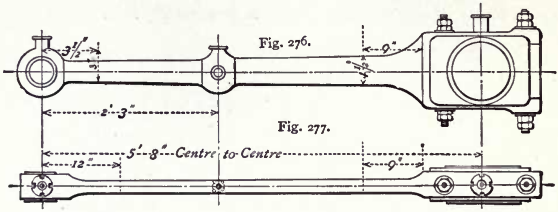


Fig. 275.

Figs. 274 and 275.—Small end of the connecting-rod of a locomotive-engine.

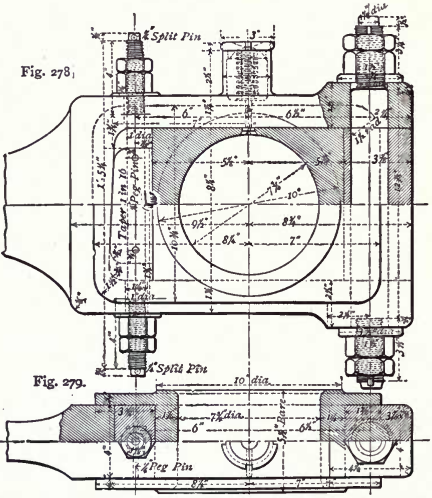
Locomotive Engine Coupling-rods or Side-rods with adjustable bushes, are shown in Figs. 282—285. The large end is solid, without straps, and is provided with a cotter for adjusting the bushes. The small end is forked and fitted with an end-block, as shown, and a cotter for adjusting the brasses. The brasses of the small end are formed with a cap to cover in



Figs. 276 and 277.—Connecting-rod of a locomotive-engine.

the end of the pin to protect it from dust, as shown in Figs. 283 and 284. A cross section of the large end of the rod is shown in Fig. 285.

The ends of a set of coupling-rods or side-rods of another design, for a locomotive engine, are shown in Figs. 286 and 287. They are fitted with solid



Figs. 278 and 279.—Large end of the connecting-rod shown in Figs. 276 and 277.

bushes, restrained from turning round by taper-pins. The ends of the rods are hardened, and the bushes are forced into place by hydraulic pressure.

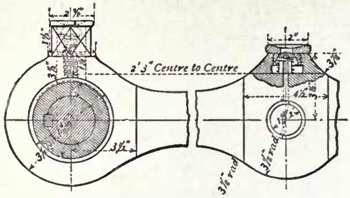


Fig. 280.—Small end of the connecting-rod shown in Figs. 276 and 277.

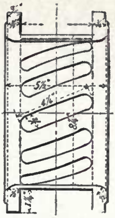
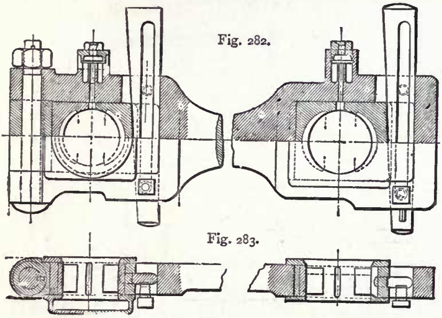


Fig. 281.—Bush of connecting-rod.

Fig. 282.

Fig. 283.



Figs. 282 and 283.—Locomotive-engine side-rods.

Fig. 284.

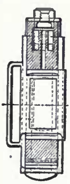
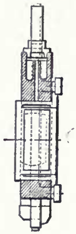


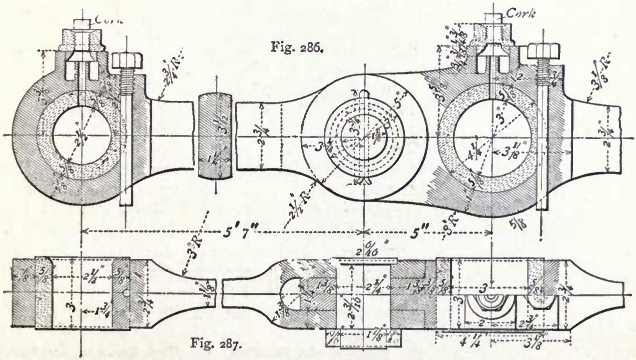
Fig. 285.



Figs. 284 and 285.—Cross-section of the ends of the side-rods shown in Figs. 282 and 283.

Fig. 286.

Fig. 287.



Figs. 286 and 287.—Locomotive-engine side-rods.

*** For some of the above, and several other illustrations in this work, the author is indebted to "Engineering."

The bushes are bored $\frac{1}{32}$ inch larger than their pins to allow for the play of the axle due to inequalities of the road. The middle or plain part of the rod is flat-sided.

Crank-shafts of Marine-Engines are generally constructed in two or

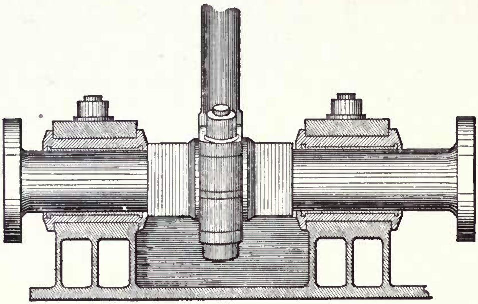


Fig. 288.—Crank-shaft of a marine-engine.

more parts, which are interchangeable, and connected together by flanged

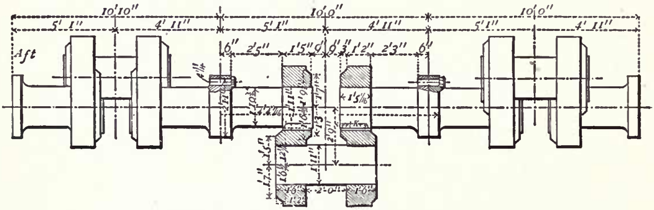


Fig. 289.—Built-up crank-shaft of a marine-engine.

couplings, a separate part being provided for each cylinder, as shown in Fig. 288; each part is a duplicate of the other, so that in the event of a break-down only one part has to be renewed.

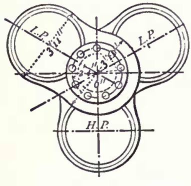


Fig. 290.—End-view of the crank-shaft shown in Fig. 289.

The Marine-Engine Crank-shaft, shown in Figs. 289 and 290, is built of separate pieces; the two pieces of each part of the shaft, and also the cranks, are forged out of the best scrap-iron. A forging of cast-steel is used for the crank-pin. In this method of building up the shaft in pieces, the forgings for engines of the largest size are of the simplest kind, and of such a size as to insure their being sound.

A Crank-shaft for a Triple-expansion Engine, built up on a very simple and effective plan (J. P. Hall's patent), is shown in Fig. 291, the novel feature of

The flange of the crank-pin is secured to each web of crank by four bolts, although two only are shown in this engraving.

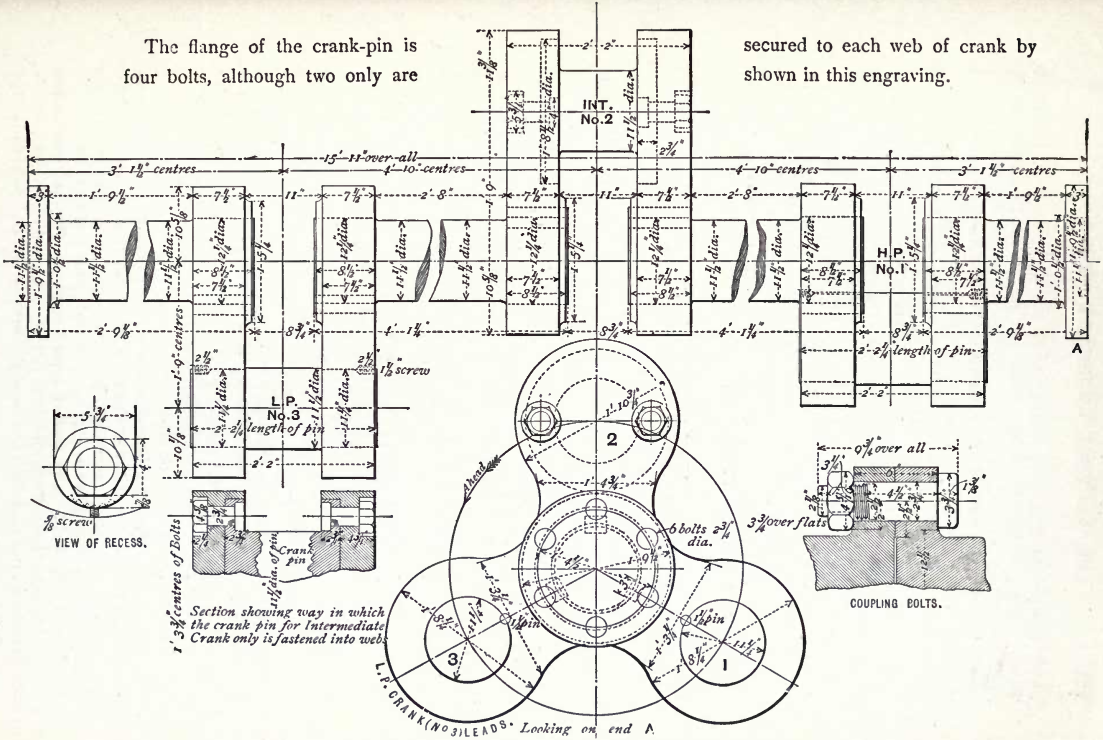


Fig. 201.—Crank-shaft of a triple-expansion marine-engine, built up on Mr. J. P. Hall's system.

which consists in bolting the centre crank-pin to the webs of the crank. The crank-pin is flanged at both ends, the flanges are recessed into the crank-webs and secured by bolts, as clearly shown in the sectional view. This arrangement is particularly adapted for the three-throw crank-shafts of triple-expansion engines, as the shaft then only consists of two pieces and a crank-pin, consequently only one-half a shaft need be carried as spare-gear, or provision for a break-down on a voyage.

A shaft constructed on this principle is more flexible than a solid shaft, and it possesses all the advantages of a shaft made in three pieces and joined by flanged couplings, without losing the space occupied by such couplings. The application of this system to one of the cranks of a built-up shaft for marine engines of 1200 indicated horse-power, is shown in Fig. 291, but it may be applied to all the cranks of a similar shaft, and also to cranks having webs solid with the shaft. A number of crank-shafts are working with very satisfactory results, constructed on this principle, the proportions of a few of which are given in the following Table:—

Table 85.—Proportions and Finished Weights of Crank-shafts built on Mr. J. P. Hall's system; the Shafts being made of Wrought-iron and the Crank-webs and Crank-pins of Forged Steel.

Name of Vessel.	Length of Stroke.	Diam. of Shaft.	Length Over All.		Diam. of Coupling		Length of Crank-Webs				Thickness of Webs.	Weight of Shaft.	
							For H. P. & L. P.		For I. P.				
	Inches.	Inches.	Ft.	In.	Ft.	In.	Ft.	In.	Ft.	In.	Inches	Tns	Cwt
Earnwell .	42	$11\frac{1}{2}$	15	8	1	$9\frac{1}{2}$	3	$5\frac{7}{8}$	3	7	$7\frac{1}{2}$	6	5
Earnmoor .	42	$11\frac{1}{2}$	15	8	1	$9\frac{1}{2}$	3	$5\frac{7}{8}$	3	7	$7\frac{1}{2}$	6	5
Era . .	39	$10\frac{3}{4}$	16	2	1	9	3	$3\frac{3}{8}$	3	5	$7\frac{3}{4}$	6	1
Hopetoun .	39	$11\frac{1}{4}$	15	11	1	$9\frac{1}{2}$	3	$4\frac{1}{8}$	3	$5\frac{1}{4}$	$7\frac{1}{4}$	6	2
Starling . .	33	$9\frac{3}{4}$	13	6	1	8	2	$10\frac{5}{8}$	2	$11\frac{1}{2}$	$6\frac{1}{2}$	4	5
Alice Depeaux	36	$10\frac{1}{4}$	14	2	1	9	3	$1\frac{1}{8}$	3	$1\frac{3}{4}$	$6\frac{3}{4}$	4	18
Cairnryan . .	36	10	14	2	1	8	3	$1\frac{1}{8}$	3	$1\frac{3}{8}$	$6\frac{3}{4}$	4	15
Drever . .	36	$10\frac{1}{2}$	14	7	1	$8\frac{1}{2}$	3	$1\frac{5}{8}$	3	$2\frac{1}{8}$	$7\frac{1}{4}$	5	5
Glanystwyth .	36	$10\frac{3}{8}$	14	7	1	$8\frac{1}{2}$	3	$1\frac{5}{8}$	3	$2\frac{1}{8}$	$7\frac{1}{4}$	5	8
Northwood .	36	10	14	2	1	8	3	$1\frac{1}{8}$	3	$1\frac{5}{8}$	$6\frac{3}{4}$	4	15
Cairntoul . .	36	$10\frac{1}{2}$	14	7	1	$8\frac{1}{2}$	3	$1\frac{5}{8}$	3	$2\frac{1}{8}$	$7\frac{1}{4}$	5	5

The Diameter of a Wrought-iron Crank-shaft for a Marine Engine may be found by the following formula:—

Let IHP = the indicated horse-power of the engines; N = the number of revolutions per minute; D = the diameter of the shaft in inches.

Then $D = 4{\cdot}5 \sqrt[3]{(IHP \div N)}$.

Example: Required the diameter of a wrought-iron crank-shaft, also the dimensions of the web of the crank for a marine engine of 2,400 indicated horse-power, making 60 revolutions per minute.

Then $2400 \div 60 = \sqrt[3]{40} = 3{\cdot}42 \times 4{\cdot}5 = 15{\cdot}39$ inches, or say $15\frac{1}{2}$ inches diameter of the shaft. The area of each web of the crank should at least be equal to that of the shaft, which is $15{\cdot}5^2 \times {\cdot}7854 = 188{\cdot}7$ square inches; taking the width of each web at two-thirds the diameter of the shaft, or say

10 inches, the depth of each web will be $188{\cdot}7 \div 10 = 18{\cdot}87$, or say 19 inches. Hence the area of the section of each web of the crank will be $10 \times 19 = 190$ square inches; the diameter and length of the crank-pin

Fig. 292.

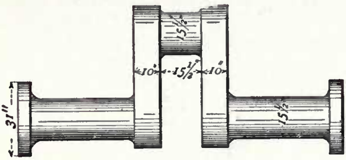


Fig. 293.

Figs. 292 and 293.—Crank-shaft of a marine-engine.

may each be equal to the diameter of the crank-shaft, and with these proportions the crank-shaft shown in Figs. 292 and 293 is obtained.

The Web of a Crank may be proportioned by the following rule, which gives nearly the same results as those obtained by the previous method. If w = the width and d = the depth of each crank-web, wd^2 should be equal to the cube of the diameter of the crank-shaft. *Example:* Required the dimensions of the web of the crank-shaft given in the previous example, $15\frac{1}{2}$ inches diameter.

Then the cube of the diameter of the crank-shaft $(15\frac{1}{2}{}^3) = 3724$.

If the width of the web be two-thirds the diameter of the shaft, or say 10 inches, it will give $3724 \div 10 = 372{\cdot}4$ and $\sqrt[2]{372{\cdot}4} = 19{\cdot}3$ inches, the depth, and $10 \times 19{\cdot}3^2 = 3724$ as required by the rule, which makes the depth of the crank ·3 inch more than that obtained by the previous method.

The Diameter of a Solid Steel Crank-shaft for a Marine-engine may be found by the following formula:—

Let IHP = the indicated horse-power of the engine.
N = the number of revolutions per minute.
D = the diameter of the shaft in inches.

$$\text{Then } D = 4{\cdot}3\sqrt[3]{\frac{IHP}{N}}$$

Example: Required the diameter of a solid steel crank-shaft for a marine-engine of 2400 indicated horse-power, making 60 revolutions per minute.

Then 2400 horse-power $\div 60 = 40$ and $\sqrt[3]{40} = 3{\cdot}42 \times 4{\cdot}3 = 14{\cdot}7$, or say $14\frac{3}{4}$ inches, being $15{\cdot}5^3 \div 14{\cdot}75^3 = 1{\cdot}15$ or 15 per cent. less in size than the wrought-iron crank-shaft found for the same power in the previous example.

Journals and Couplings of Crank-shafts of Marine-engines are usually of the following proportions:—

Length of each journal or bearing of crank-shaft = diameter of crank-shaft $\times$ $1{\cdot}25$ to $2{\cdot}5$; the higher the speed the longer should the journal be.

Diameter of the coupling of crank-shaft = diameter of crank-shaft × 1·8 to 2.

Thickness of each flange of the coupling = diameter of crank-shaft × ·25 to ·3.

The Length of Bearings or Journals of a Crank-shaft suitable for a given pressure may be found by this *Rule:* Divide the initial pressure of the steam on the piston in lbs. per square inch of its area, by the product of the given pressure in lbs. on the bearing and the diameter of the bearing in inches.

Example: The piston of an engine is 60 inches diameter, the initial pressure of the steam is 50 lbs. per square inch, the diameter of each bearing of the crank-shaft is 14 inches: what should the length of each bearing be, if the pressure on the bearing is not to exceed 600 lbs. per square inch?

Then $\frac{60 \times 60 \text{ inches} \times \cdot 7854 \times 50 \text{ lbs. pressure}}{600 \text{ lbs.} \times 14 \text{ inches diameter}} = 16 \cdot 82$ inches, the length of each bearing required.

The Bushes for the Journals of Crank-shafts are best made of good tough bronze or gun-metal, with strips of anti-friction metal let into the bearing-surfaces.

The gun-metal for the bushes may be composed of:—

88 parts of copper
10 parts of tin
2 parts of zinc.

The strips of anti-friction metal may be composed of:—

85 parts of tin
5 parts of copper
10 parts of antimony.

The strips may be let into the bearing-surfaces in the manner shown in Fig. 281.

The Proportions of the Crank of a Marine-engine are usually as follows:—

Width of each web of crank = diameter of crank-shaft × ·67.
Depth of each web of crank = diameter of crank-shaft × 1·25.
Diameter of crank-pin = diameter of crank-shaft × 1 to 1·1.
Length of crank-pin = diameter of crank-shaft × 1 to 1·25.

The area of bearing of each crank-pin should not be less than = ·2 square inch per indicated horse-power developed by the corresponding cylinder.

The Strain or Pressure on a Crank-pin may be found by this *Rule:*—

$$\frac{\text{Area of cylinder in ins.} \times \text{maximum pressure of the steam in lbs per sq. in.}}{\text{diametrical sectional area of the crank-pin in square inches}}$$

The diametrical sectional area of the crank-pin is the product of the diameter of the crank-pin in inches by its length in inches.

Example: The diameter of the low-pressure cylinder of a compound engine is 60 inches, the areas of the two cylinders are to each other as 1 is to 3·7, the effective pressure of the steam on the high-pressure piston is 85 lbs. per square inch at the beginning of its stroke, and on the low-pressure piston the effective pressure of the steam is 23 lbs. per square inch at the beginning of its stroke, the crank-pin is 14 inches diameter and 15 inches long. Required the diametral sectional area of the crank-pin in square inches; and also the pressure in lbs. per square inch on the surface of each of the crank-pins?

Then, the diametral sectional area of the crank-pin is 14×15 inches $= 210$ square inches.

The area of the low-pressure cylinder is $60 \times 60 \times \cdot 7854 = 2827 \cdot 44$ square inches.

The area of the high-pressure cylinder is $2827 \cdot 44 \div 3 \cdot 7 = 764 \cdot 17$ square inches.

The pressure on the low-pressure crank-pin is $\dfrac{2827 \cdot 44 \times 23 \text{ lbs.}}{210 \text{ area of crank-pin}} =$ 309·67 lbs. per square inch.

The pressure on the high-pressure crank-pin is $\dfrac{764 \cdot 17 \times 85 \text{ lbs.}}{210 \text{ area of crank-pin}} =$ 309·30 lbs. per square inch.

The Weight of the Crank and of half the Length of the Connecting-rod next the Crank-pin of high-speed engines should be accurately balanced by means of a counter-balance-weight revolving opposite to the crank, so that both may revolve in the same plane of revolution. The counter-weight may either be solid with the crank, or bolted on. When it is bolted on to the crank, the diameter of the bolts required may be found by the following formulæ:—

Let R = the number of revolutions of the crank-shaft per minute.
r = the effective radius of the balance-weight in feet.
N = the number of times the weight is increased by centrifugal force.

$$N = \frac{R^2 \times r}{2935}$$

$$\text{Area of bolt in sq. ins.} = \frac{N \times \text{weight of balance-weight in lbs.}}{\text{number of bolts} \times \text{working stress on the bolt in lbs.}}$$

Example: The counter-balance-weight of a crank weighs 1480 lbs., its effective radius being 1·6 feet, the shaft makes 65 revolutions per minute. What should the diameter of each bolt be, if two bolts be used to bolt it to the crank, and the strain on the bolts is not to exceed 5000 lbs. per square inch of section?

Then $\dfrac{65 \times 65 \times 1 \cdot 6}{2935} = 2 \cdot 3$, the number of times the weight is increased by the centrifugal force, which multiplied by the weight of the counter-weight in lbs., will give the stress upon the bolts, and

$$\frac{2 \cdot 3 \times 1480 \text{ lbs. weight of counter-balance}}{2 \text{ bolts} \times 5000 \text{ lbs. the strain allowed}}$$

$= 3 \cdot 4$ square inches the area of each bolt required, then $\sqrt[2]{3 \cdot 4 \div \cdot 7854} =$ 2·08 inches, the diameter of each bolt.

The Boiler-Pressure of Steam suitable for a given Size of Shaft is sometimes found by the following *Rule* :—

Let D = the diameter of the low-pressure cylinder in inches.
H = the diameter of the high-pressure cylinder in inches.
L = the length of stroke in inches.
d = the diameter of the shaft in inches.
f = a constant = 4936 for wrought-iron crank-shafts.
f = a constant = 5760 for wrought-iron tunnel or propeller-shafts.
B = Boiler-pressure of steam in lbs. per square inch.

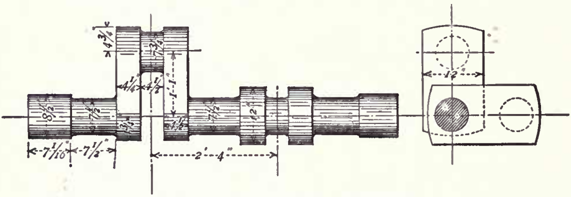
$$B = \frac{f d^3 - 15 \text{ LD}^2}{\text{L} \times \text{H}^2}$$

Example: What boiler-pressure of steam may be used for a compound engine having a wrought-iron crank-shaft 11 inches diameter; diameter of the low-pressure cylinder 60 inches; diameter of high pressure cylinder 30 inches; length of stroke 36 inches.

Then $\frac{4936 \times 11^3 - 60^2 \times 36 \times 15}{36 \text{ inches} \times 30^2 \text{ inches}} = 142$ lbs. per square inch, the boiler-pressure of steam that may be used for that crank-shaft.

A Crank-Axle for a Locomotive Engine with inside cylinders, single frames and single bearings, is shown in Figs. 294 and 295. The cylinders

Fig. 294. Fig. 295.



Figs. 294 and 295.—Locomotive crank-axle.

are 17 inches diameter, and the length of stroke is 26 inches; the journals are $7\frac{1}{2}$ inches diameter and $7\frac{1}{2}$ inches long. The crank-pins are each $7\frac{3}{4}$ inches diameter and $4\frac{1}{2}$ inches long; their diametrical sectional area being $7\frac{3}{4}$ inches $\times$ $4\frac{1}{2}$ inches=34·88 square inches. Taking the maximum mean effective pressure of the steam in the cylinder at 100 lbs. per square inch, the area of the cylinder being $17^2 \times \cdot 7854 = 227$ square inches, the total pressure that may be delivered on each crank-pin would equal 227 square inches $\times$ 100 lbs. = 22700 lbs. or a little more than 10 tons being at the rate of 22700 lbs. $\div$ 34·88 = 623 lbs. per square inch of diametrical section of the crank-pin. A pressure of 10 tons on each crank-pin would, when the cranks are in certain positions, result in a combined stress of 20 tons on the axle. The outside or thinner webs of the crank are $4\frac{1}{4}$ inches thick and 12 inches broad, the sectional area of each web being $12 \times 4\frac{1}{4} = 51$ square inches, the sectional area of each crank-pin is $7\frac{3}{4}^2 \times$

$\cdot 7854 = 47\cdot 17$; the sectional area of each journal is $7\frac{1}{2}^2 \times \cdot 7854 = 44\cdot 17$.

Hence the thinner webs of the crank are of sufficient area, being a little greater than that of the axle.

A Crank-Axle for a Locomotive Engine with inside cylinders, double

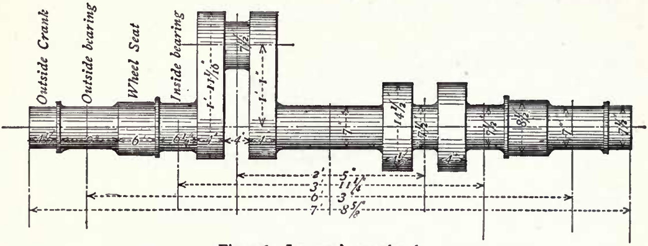


Fig. 296.—Locomotive crank-axle.

frames, and both inside and outside bearings, is shown in Fig. 296; outside cranks are provided for the coupling-rods or side-rods.

Crank-Shafts and Crank-Axles frequently fail at the angle between the web of the crank and the crank-pin as shown in Fig. 297, and they sometimes fail by a ball-and-socket shaped fracture at the crank-pin which separates in this form from one of the crank-arms. The cause of failure in many cases being overstraining of the crank beyond the limit of its elastic strength.

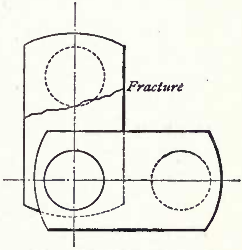


Fig. 297.—Broken crank-web.

Crank-Axles of Locomotive Engines frequently fail from being kept running too long a time. It is considered that the maximum mileage of iron crank-axles should not exceed 200,000 miles, and of steel 180,000 miles. In order that in the event of a breakage, the crank may hold together until the train is stopped or a station reached, each web of the crank is frequently hooped, and a hole is bored through the centre of the crank-pin into which a bolt is fitted to hold the webs together.

A Crank-Shaft may be strained excessively by its bearings being out of line; by the bearings giving or springing; by slackness of the brasses; by want of rigidity in the bed-plate; or by the presence of water in the cylinder: and its tensile strength may be considerably diminished by long-continued excessive straining. When the bearings are so much out of line as to cause the necks or journals to become hot, the strain on the shaft is seldom less than one-third greater than that due to ordinary working in true bearings, it frequently is considerably more, and it may be from 50 to 100 per cent. greater than when the bearings are true. When the shaft is fractured by being out of line, it is frequently due to the strain produced by the opening and closing of the jaws of the crank at each revolution of the crank-shaft.

The slackness of the brasses, if slight, may be taken up gradually by cushioning, but where cushioning is defective, or absent, the pressure of the steam comes on the piston suddenly and causes an impulsive strain on the shaft which varies in intensity with the amount of slackness of the brasses. The presence of water in the cylinder may cause a severe bending strain on the crank-shaft.

Steel Crank-Axles and Crank-Shafts should be made of mild, homogeneous *forged-steel*, of which the following is an average analysis :—

Carbon ·200 ; manganese ·600 ; silicon ·015 ; sulphur ·060 ; phosphorus ·050. The tensile strength should be about 28 tons per square inch ; elongation 25 per cent. ; contraction 40 per cent. Crank-shafts are sometimes made as steel-castings, but they are generally inferior to, and not so reliable as, forged-steel shafts.

Bent Cranks, made from round bars of iron or steel, shown in Fig. 298, are used for portable-engines, pumps, thrashing-machines, looms, &c. ; the small sizes are bent by special machines, and the larger sizes by hydraulic pressure in suitable presses, the ends being forced inwards while the throws are being formed, so as to complete the crank with as few heats as possible and prevent deterioration of the material.

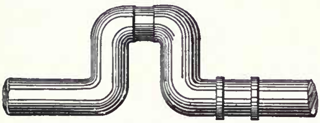


Fig. 298.—Crank-shaft of a portable-engine.

This is the cheapest method of making a crank-shaft, and it is also the strongest, because the fibre of the material runs along the arms and round the throw of the crank, whereas when the crank is forged in a solid block, the fibre runs across the webs and is severed by cutting out the throw. When a bent crank is required to work in a limited space, the webs, instead of being round are slightly flattened, or made oval-shaped.

Three-Throw Cranks for Large Pumps are frequently made of cast-iron, and when made of good tough metal they work satisfactorily. The crank-pins or sling-bearings are usually made one-fifth larger in diameter than the main bearings, to compensate for the torsional strain being greater on the sling-bearings than on the main-bearings, owing to the peculiar form of this kind of crank. A cast-iron crank-shaft will only bear two-thirds the torsional strain of that of a wrought crank-shaft of the same proportions. Thus if a wrought-iron crank-shaft be suitable for a strain of 30 horse-power, a cast-iron shaft of the same proportions would only bear a strain of 20 horse-power.

Cast-iron Cranks for Large Pumps should be cast from good tough metal. The following is a good mixture of metal for this purpose :—

Scotch mixed brands, 5 cwt.
Weardale, 7 cwt.
Good clean scrap, 8 cwt.

A test-bar of cast-iron cast from this mixture, 1 inch square, placed upon supports three feet apart, should bear a gradually applied weight of about $7\frac{1}{2}$ cwt. with a deflection of about $\frac{5}{8}$ inch.

A Screw-Propeller Shaft is connected to the crank-shaft by several lengths of shafting with couplings forged solid with the shaft, as shown in Fig. 299. The foremost length of shafting has a journal fitted with thrust-

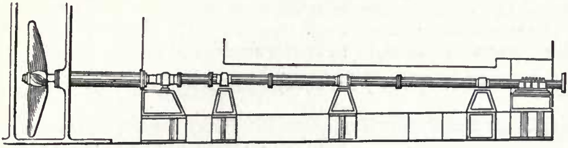


Fig. 299.—Screw-propeller-shafting.

collars which work in recesses in a thrust-bearing, which receives the thrust of the propeller.

The Diameter of a Solid, Wrought-Iron Propeller-Shaft, or screw-shaft, may be found by the following formula :—

Let I H P = the indicated horse-power of the engines.
N = the number of revolutions per minute.
D = the diameter of the screw-shaft.

$$\text{Then } D = 4\sqrt[3]{\frac{I\,H\,P}{N}}$$

Example: Required the diameter of solid wrought-iron screw-propeller shafting for a pair of marine engines of 1500 indicated horse-power, making 62 revolutions.

Then $1500 \div 62 = \sqrt[3]{25} = 2{\cdot}93 \times 4 = 11{\cdot}72$, or say $11\frac{3}{4}$ inches diameter.

The size of shaft obtained by this rule, although it agrees fairly with practice, does not allow a sufficient margin of strength for contingencies, the number of failures of these shafts prove that they are frequently made too light. They frequently fail from want of stiffness to resist the strain due to the continual bending and unbending action of the overhung weight of the propeller.

Screw-Propeller-Shafting is exposed to an end-thrusting strain due to propulsion, a twisting strain from the engine, and a bending strain due to the weight of the propeller. The diameter of the shafting may be found by the following *Rule*, which provides for all these strains and allows a margin of strength for contingencies, to obviate fracture.

$$D = \sqrt[3]{\frac{(I\,H\,P \times 90)}{N}}.$$

Example: Required the diameter of the solid wrought-iron propeller-shaft, given in the previous example.

Then $\frac{1500 \times 90}{62} = \sqrt[3]{2016} = 12{\cdot}96$, or say 13 inches diameter of shaft, being $13^3 \div 11\frac{3}{4}{}^3 = 1{\cdot}35$, or 35 per cent. larger than that obtained by the previous *Rule*.

Screw-Propeller Shafts of Large Size are best made hollow of Whitworth's Compressed Steel, as the weight of the shaft may thus be considerably reduced without diminishing the strength.

Hollow-Shafting of Whitworth's Compressed-Steel for screw-propellers should be calculated, for the external diameter, by the last *Rule* given above for wrought-iron propeller-shafts, and the internal diameter should = the external diameter of the shaft × ·56.

Example: Required the internal diameter of a propeller-shaft of Whitworth's Compressed Steel, of 14 inches external diameter.

Then $14 \times \cdot 56 = 7 \cdot 84$, or say $7\frac{7}{8}$ inches internal diameter.

Couplings for Screw-Propeller Shafting consist of flanges forged solid with the shafts, one flange having a projection to fit into a recess in the other flange, in order to keep the shafts central, as shown in Fig. 300. The diameter of the flange or coupling = the diameter of the shaft × 1·8 to 2 : the thickness of each flange = $\frac{1}{4}$th the diameter of the shaft.

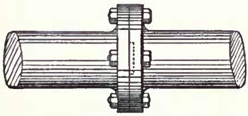


Fig. 300.—Coupling of propeller-shafting.

Bolts for the Couplings of Screw-Propeller Shafting are exposed principally to a shearing strain. The diameter of the bolts, D, may be found by the following formula, where N = the number of the bolts.

$$D = \frac{\cdot 55}{\sqrt[2]{N}} \times \text{diameter of the shaft.}$$

Example: Required the diameter of the bolts for the couplings of a screw-propeller shaft of 12 inches diameter, the number of bolts to be six.

Then $\sqrt[2]{6} = 2 \cdot 45$ and $\cdot 55 \div 2 \cdot 45 = \cdot 224 \times 12 = 2 \cdot 69$ inches, the diameter of the bolts required. If the number of bolts were eight, their diameter would be $= \sqrt[2]{8} = 2 \cdot 83$ and $\cdot 55 \div 2 \cdot 83 = \cdot 194 \times 12 = 2 \cdot 33$ inches.

The Shearing-Strain on the Bolts of the Couplings of Shafting may be found by dividing the power in foot pounds transmitted, by the distance travelled by the bolts in one minute.

Example: Required the shearing-strain on the bolts of the couplings of the shafting of a screw-propeller transmitting 1000 indicated horse-power, making 60 revolutions per minute, the diameter of the circle of the centres of the bolts is 18 inches, and there are 8 bolts, 2 inches diameter, in each coupling.

Then 1000 horse-power × 33000 lbs. = 33,000,000 foot lbs. transmitted and 1·5 feet pitch-circle of bolts × 3·1416 × 60 revolutions = 282·744, the distance travelled by the bolts in one minute. The strain on all the bolts of the coupling will be = 33,000,000 ÷ 282·744 = 116,713·31 lbs., the strain on each bolt will be = 116713·31 ÷ 8 = 14,590 lbs., the area of each bolt is $2^2 \times \cdot 7854 = 3 \cdot 1416$ square inches, and the strain on each bolt is 14590 ÷ 3·1416 = 4644 lbs. per square inch of the sectional area of the bolt.

The Strain on the Shaft of a Screw-Propeller due to the Overhang of a Propeller may be found by the following formula:

Let W = the weight of the propeller in tons.
„ D = the distance in inches of the centre of the weight from the point of support.
„ S = the maximum strain in lbs. per square inch.

$$\text{Then } S = \frac{10{\cdot}2 \times W \times 2240 \times D}{(\text{Diameter of the shaft in inches})^3}.$$

Example: Required the maximum strain in lbs. per square inch on the shaft of a screw-propeller due to the overhang of a propeller weighing 5 tons, the centre of weight being 25 inches from the point of support and the shaft being 11 inches diameter.

$$\text{Then } \frac{10{\cdot}2 \times 5 \text{ tons} \times 2240 \times 25 \text{ inches}}{11 \times 11 \times 11 \text{ inches}} = 2145{\cdot}8 \text{ lbs. per square inch.}$$

The Strength of a Shaft or Shafting, to resist Bending, is only equal to one-half of its strength to resist torsion or twisting.

The Strength of Round Bars or Shafts to resist a Torsional or Twisting Strain is in proportion to the cubes of their diameters. A bar of wrought-iron, 1 inch diameter, is twisted asunder by a weight or force of 800 lbs. applied at the end of a lever 12 inches long, measured from the centre of the bar, and the relative resistance of different bars is usually expressed by stating the weight which twists them asunder when applied in this manner. Hence if the torsional strength of a bar 1 inch diameter be known, the strength of bars of other dimensions of the same material may be calculated from it.

Example: As a bar of wrought-iron 1 inch diameter is twisted asunder by a force of 800 lbs., applied at the end of a lever 12 inches long from the centre of the bar, what force will be required to break a $\frac{3}{4}$ inch diameter wrought-iron stud, applied at the end of a lever 16 inches long?

Then 16 inches : 12 : : 800 lbs. = 600 lbs. the force required to break a stud 1 inch diameter applied at the end of a lever 16 inches long.

And $1^3 : 75^3 :: 600 = 253{\cdot}125$ lbs. the force required to break the $\frac{3}{4}$ inch diameter wrought-iron stud at a leverage of 16 inches.

The Torsional Strain on Shafting may be found by the following formula:—

Let F = the force in lbs. applied to the shaft.
L = the leverage in inches through which the force acts.
D = the diameter of the shaft in inches.
S = the maximum strain per square inch of section of the shaft.

$$\text{Then } S = \frac{F \times L \times 5{\cdot}1 \text{ constant}}{D^3}.$$

Example: A shaft 10 inches diameter is subjected to a torsional strain by a force of 50,000 lbs., acting with a leverage of 20 inches. Required the maximum stress per square inch of section of the shaft.

$$\text{Then } \frac{50{,}000 \text{lbs.} \times 20 \text{inches} \times 5{\cdot}1}{10 \times 10 \times 10 \text{ inches}} = 5{,}100 \text{ lbs. strain per square inch of}$$

section of the shaft.

The maximum strain, S, per square inch on the transverse section of the shaft should not exceed the following:—

For wrought-iron crank-shafts, 5,000 lbs.
For mild steel crank-shafts, 5,500 lbs.
For all other shafts and shafting of wrought-iron, 9,000 lbs.
For all other shafts and shafting of mild steel, 10,000 lbs.

The Force, acting with a given Leverage, which may be applied to a Shaft may be found by the following formula, the notation being the same as in the previous formula:—

$$F = \frac{D^3 \times S}{L \times 5{\cdot}1}.$$

Example 1: Required the force in lbs. which may be applied, at a leverage of 20 inches, to a shaft 10 inches diameter, which is capable of bearing a maximum strain of 5,100 lbs. per square inch of section.

Then $\frac{10^3 \times 5{,}100 \text{ lbs.}}{20 \times 5{\cdot}1} = 50{,}000$ lbs., the force which may be applied to the shaft at that leverage.

Example 2: If the maximum strain to be allowed on wrought-iron be taken at 9,000 lb. per square inch, what pressure may be applied at the end of a crank 20 inches long, to be transmitted by a shaft $10\frac{1}{2}$ inches diameter.

$$\text{Then } \frac{10{\cdot}5^3 \times 9{,}000}{20 \times 5{\cdot}1} = 102{,}163 \text{ lbs.}$$

The Diameter of a Shaft, subject only to a Torsional or Twisting Strain, suitable for a given force and leverage, may be found by the following formula, the notation being the same as in the two previous *Examples*:

$$D = \sqrt[3]{\frac{F \times L \times 5{\cdot}1}{S}}.$$

Example: Required the diameter of a shaft suitable for a force of 50,000 lbs. applied at a leverage of 20 inches, the material of which the shaft is made being capable of bearing a strain of 5,100 lbs. per square inch of cross section.

$$\text{Then } \sqrt[3]{\frac{50{,}000 \times 20 \times 5{\cdot}1}{5{,}100}} = 10 \text{ inches diameter.}$$

The Torsional Strain on a Crank Shaft may be found by the following formula:—

Let F = the force at leverage = the effective pressure of the steam in lbs. × length of crank in inches.
A = the area of the cylinder in square inches.
D = the diameter of the crank-shaft in inches.
S = the torsional strain per square inch of section of the shaft.

$$S = \frac{F \times A \times 5{\cdot}1}{D^3}.$$

Example: The diameter of the cylinder of a marine engine is 42 inches, the length of stroke is 48 inches, the effective pressure of the steam is 60 lbs. per square inch, and the diameter of the crank-shaft is 13 inches. Required the torsional strain on the crank-shaft per square inch of section.

Then the area of the cylinder is $48^2 \times \cdot 7854 = 1385 \cdot 45$ square inches, and $\frac{1385 \cdot 45 \text{ area} \times 60 \text{ lbs. pressure} \times 24 \text{ inches crank} \times 5 \cdot 1}{13 \times 13 \times 13 \text{ inches}} = 4632$ lbs., the strain per square inch of section of the shaft.

The End-thrust on the Shafting of a Screw-propeller, if the whole of the power were utilised by the propeller, may be ascertained by the following formula:—

Let IHP = the indicated horse-power of the engines.
N = the number of revolutions per minute.
S = the diameter of the screw-propeller in feet.
T = the mean thrust in lbs. on the propeller-shaft.

$$\text{Then } T = \frac{\text{IHP} \times 33000 \times 12}{N \times S \times 12}$$

Example: Required the end-thrust on the shafting driving a screw-propeller of 10 feet diameter, the indicated horse-power of the engines being 1000, and the number of revolutions per minute = 50.

$$\text{Then } \frac{1000 \times 33000 \times 12}{50 \times 10 \times 12} = 66000 \text{ lbs., the end-thrust.}$$

Main Tunnel and Propeller-shafts will not be passed by the Board of Trade if found less in diameter than that found by the two following rules; but first-class makers generally put in larger shafts than those found by the following formulæ:—

Where S = diameter of shaft in inches, d^2 = square of diameter of high pressure cylinder in inches, or sum of squares of diameters when there are two or more high pressure cylinders, D^2 = square of diameter of low pressure cylinder in inches, or sum of squares of diameters when there are two or more low pressure cylinders, P = absolute pressure in lbs. per square inch, that is boiler pressure plus 15 lbs. C = length of crank in inches, f constant from the following Table.

For compound condensing engines with two or more cylinders, when the cranks are not overhung:—

$$S = \sqrt[3]{\frac{C \times P \times D^2}{f\left(2 + \frac{D^2}{d^2}\right)}}, \qquad P = \frac{f \times S^3}{C \times D^2}\left(2 + \frac{D^2}{d^2}\right).$$

For ordinary condensing engines with one, two, or more cylinders, when the cranks are not overhung:—

$$S = \sqrt[3]{\frac{C \times P \times D^2}{3 \times f}} \qquad P = \frac{3 \times f \times S^3}{C \times D^2}.$$

Table 86.—Constants for Board of Trade Rules for Shafting.

For Two Cranks, Angle between Cranks.	For Crank and Propeller Shafts. *f*.	For Tunnel Shaft. *f*.
90°	1047	1221
100°	966	1128
110°	904	1055
120°	855	997
130°	817	953
140°	788	919
150°	766	894
160°	751	877
170°	743	867
180°	740	864
For Three Cranks. 120°	1110	1295

(Printed sideways in the "For Crank and Propeller Shafts" column: For paddle engines of ordinary type, multiply the constant in this column suitable for the angle of cranks by 1·4.)

Note.—When there is only one crank the constants applicable are those in the table opposite 180.

The four previous formulæ are the only Board of Trade Rules for shafts.

The Stern-shaft, or aftermost length of the propeller-shafting, marked A in Fig. 301, passes through the stern of the ship and drives the screw-propeller. In order to preserve the shaft from oxidation it is always covered or cased at the bearings with gun-metal, of from ½ inch to ¾ inch thick, and in some cases it is cased from end to end. When the casing is in several lengths, they are filletted into each other at the joints to prevent leakage. The casing is either cast on the shaft, or cast separate, and bored to fit the shaft tightly, on which it is forced by hydraulic pressure.

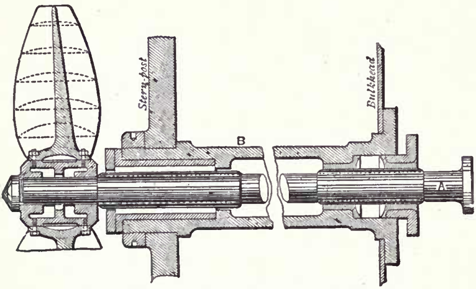


Fig. 301.—Stern-tube and stern-shaft

The Stern-tube, marked B in Fig. 301, carries the bearings on which the stern-shaft revolves. It consists of a strong tube of cast-iron or gun-metal, having a flange on the forward end, or end of the tube inside the ship, by which it is bolted to the after-bulkhead; this end is fitted with a stuffing-box and gland to prevent water entering the ship. The length of the stern-tube is determined by the distance between the after-bulkhead and the stern-post. The after or outer end of the stern-tube passes through the stern-post, to which it is secured on the outside by a strong nut screwed on the end of the tube. The bearing at the outside or after-end of the stern-tube, and sometimes at both ends, is fitted with strips of lignum-vitæ recessed into the bush, as shown in Fig. 302, the strips being secured by a check-plate to prevent them working out.

A pipe with cock is fitted for drawing water from the stern-tube to ascertain if the bearings are working cool.

Lignum-Vitæ Bearings, consisting of strips of lignum-vitæ recessed into a bush as shown in Fig. 302, are the best and most durable bearings for the journals of shafts working in water; the space between the strips form watercourses by which the water circulates and efficiently lubricates the journals of the shafts. These bearings will stand a working pressure of as much as 2000 lbs., being five times as much as can in many cases be safely carried by metal bearings. They are used for the outer-bearings of paddle-shafts and stern-shafts.

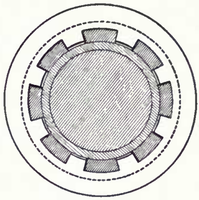


Fig. 302.—Section of Lignum-vitæ bearing.

Plummer-blocks for screw-propeller-shafting are usually of the form shown in Fig. 303. They are made of cast-iron, and lined with anti-friction-metal or white-metal. The plummer-blocks should be spaced as widely apart as possible, consistent with freedom from the shafts' sagging, in order to allow the shafting to yield to the strains in the hull due to the rolling of the ship, to prevent friction and diminish liability to fracture.

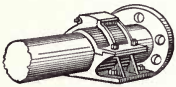


Fig. 303.—Plummer-block.

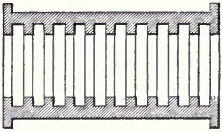


Fig. 304.—Thrust-bearing.

A Thrust-bearing is shown in Fig. 304. It consists of a bush of either gun-metal or white-metal, made in halves; it is fitted into a cast-iron thrust-block. The bush is provided with a number of thrust-rings, which fit between corresponding collars on the thrust-journal; the rings are bored a little larger than the collars, so that the circumferences of the collars on the thrust-journal sufficiently clear the bottom of the recesses in the bush to prevent the weight of the shaft coming on the thrust-block, which should only take the end-pressure due to the thrust of the propeller. The wear takes place on the sides or fore and aft surfaces of the thrust-rings. An oil-tube is provided for lubricating each thrust-collar. An ordinary plummer-block is fixed close to the thrust-block to relieve it of the weight of the shafting and prevent vibration.

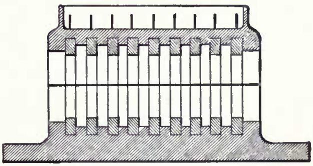


Fig. 305.—Thrust-bearing.

A Thrust-bearing composed of separate Thrust-rings recessed into the Thrust-block is shown in Fig. 305.

A Thrust-block provided with adjustable Thrust-surfaces is shown in Fig. 306. The thrust both ahead and astern is taken up as a

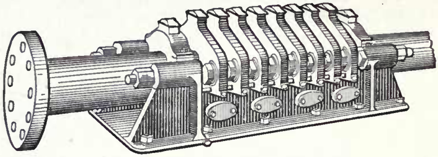


Fig. 306.—Thrust-bearing with horse-shoe rings.

tensile strain by two strong bars screwed from end to end; these bars are fitted at each end with nuts for taking the total strain against the bosses on the thrust-block, the thrust-surfaces of the shaft bear against horse-shoe-shaped cast-iron rings, which are faced with white-metal, each ring being adjusted and secured in position along the screwed bars by means of nuts and check-nuts.

SCREW-PROPELLERS, PADDLE-WHEELS, AND JET-PROPELLERS.

Screw-propellers are made with two, three and four blades. Propellers with four blades cause less vibration, and are more efficient in a rough sea than those having a less number of blades. The blades are frequently cast separately and bolted to the boss of the propeller by a flange, as shown in Fig. 307. The bolt-holes are elongated to allow the position of the blades to be adjusted when it is necessary to alter the pitch of the propeller. The flanges of the blades are frequently coated with cement, rounded off to follow the outline of the boss, to cover in the bolts and reduce the resistance of their surfaces to the water.

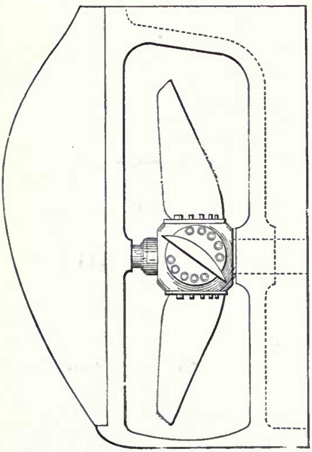


Fig. 307.—Screw-propeller with blades bolted to the Boss.

Screw-propellers made of Cast-iron should be made from good tough metal, twice run. The following is a good mixture of metal for this purpose, which should be melted and cast into pigs in order to mix properly:—

Hematite, No. 4	7 cwt.
Glengarnock, No. 3	3 "
Gartsherrie, No. 3	3 "
Clyde, No. 3	3 "
Weardale, No. 3	2 "
Monkland, No. 3	2 "

A test-bar, cast from this mixture of cast-iron, 1 inch square, placed upon supports 3 feet apart, should bear a gradually applied weight of $8\frac{1}{2}$ cwt., with a deflection of about $\frac{3}{4}$ inch.

Screw-propellers made of Gun-metal can be made much thinner and lighter than when made of cast-iron. The following mixture makes a tough and durable gun-metal for this purpose :—

Copper 88 parts, tin 10, and zinc 2 parts.

Another mixture for the same purpose is copper 87·7 parts, tin, 8·3 and Silesian spelter 4 parts.

Delineation of a Screw-Propeller. A well-proportioned and very efficient screw-propeller, 15 feet 4 inches diameter, is shown in Figs. 308—310; it has four blades cast solid with the boss. Fig. 309 is a view looking forward in line with the axis of the propeller-shaft, the full line representing the projected shape of the propeller-blade as it will then appear, and the dotted line the shape of blade when developed or laid flat. Fig. 310 is a plan of the propeller-blade looking in direction of line A B (Fig. 309). Fig. 308 is a view of the propeller-blade in direction C D (Fig. 310); that is, a side view of blade. This propeller was fitted to s.s. "Flamboro."

After drawing an outline of the boss, the centre lines, and diameter of the propeller, the developed shape of the propeller-blade, as shown in Fig. 310, is usually determined next, and for the screw-propeller illustrated, the following is a general outline of the mode of procedure.

In Fig. 309 draw any convenient number of arcs from the centre E with radii E F, E G, E H, etc., and E A, E F being the same as or slightly more than that of the boss, and E A the full radius of the propeller, the intermediate axes being equally spaced, or nearly so.

In Fig. 310 draw the line M N parallel to D C at a distance corresponding with the aftermost part of the blade. On any convenient scale make O P = pitch, and P Q and P R = circumference of circles of radii E F and E A (Fig. 309), respectively. Draw O Q and O R. These will give the developed angles of blade near the boss and at periphery respectively.

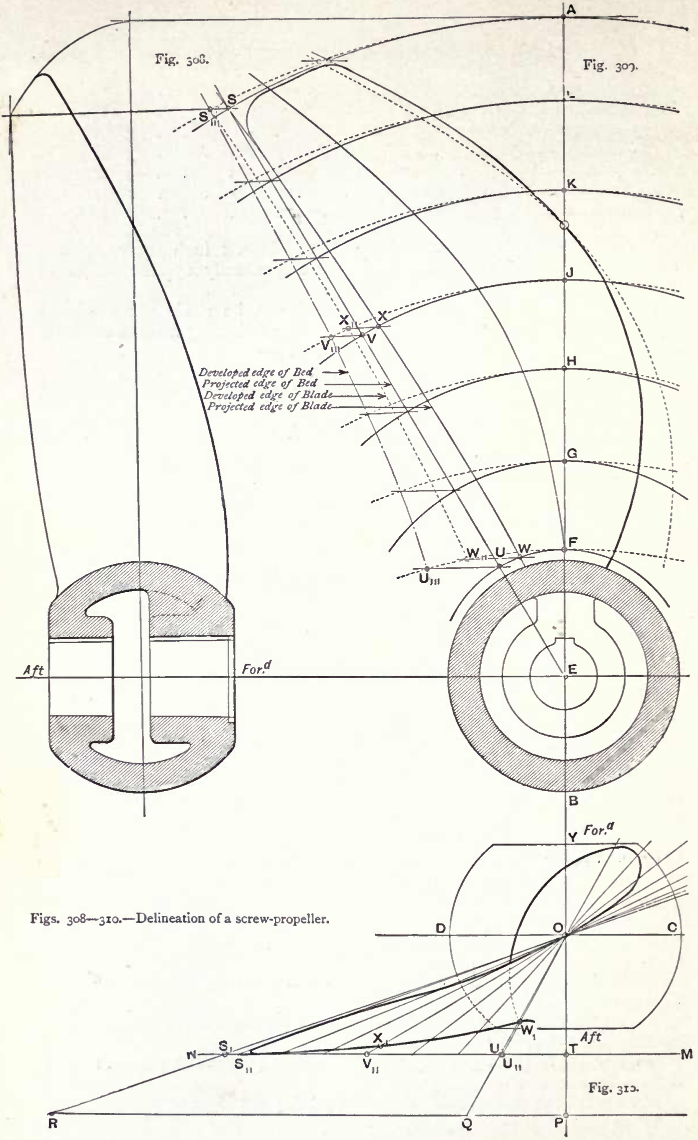
In Fig. 309 lay off round full arc at A a distance A S = T $S_{\prime}$ (Fig. 310), and round full arc at F lay off F U = T $U_{\prime}$ (Fig. 310), join S U cutting the full arcs in V, etc. This line in a propeller of uniform pitch will pass through the centre of the shaft if produced. On line M N (Fig. 310) lay off T $S_{\prime\prime}$= perpendicular distance of S from A B (Fig. 309).

Horizontally in line with S (Fig. 309) mark point $S_{\prime\prime\prime}$ at a perpendicular distance from A B = O $S_{\prime\prime}$ (Fig. 310).

With centre somewhere on line A B or A B produced, draw *dotted* arc passing through point A and $S_{\prime\prime\prime}$.

On line M N (Fig. 310) lay off T $U_{\prime\prime}$ = perpendicular distance of U from A B (Fig. 309).

Horizontally in line with U (Fig. 309) mark point $U_{\prime\prime\prime}$ at a perpendicular distance from A B = O $U_{\prime\prime}$ (Fig. 310).



Figs. 308—310.—Delineation of a screw-propeller.

With centre somewhere on line A B or A B produced, draw *dotted* arc passing through points F and $U_{///}$.

In Fig. 310 divide $S_{//} U_{//}$ into parts by points $V_{//}$ etc., corresponding with points V etc. (Fig. 309) join O $V_{//}$ etc. (Fig. 310).

Horizontally in line with V (Fig. 309) mark point $V_{///}$ at a proper distance from A B = O $V_{//}$ (Fig. 310).

With centre in A B or A B produced, draw dotted arc through J $V_{///}$.

Draw the other dotted arcs in the same manner and then draw $S_{///} U_{///}$ through the points $S_{///}$ $V_{///}$ etc., and $U_{///}$.

All the other arcs should be continued on right side of A B as shown.

These arcs correspond with those that will eventually be drawn on "bed" of the propeller-mould in the foundry, the projected edge of "bed" being represented by the full line S U, and the developed edge by the dotted line $S_{///} U_{///}$.

The dotted arcs are approximately the development of parts of the helices wound round cylinders of radius E A and E F respectively, with pitch equal to that of the propeller.

The figure A $S_{///}$ $U_{///}$ F, and a similar figure on other side of A B show the limits with which the blade must be drawn.

On dotted curve F (Fig. 309) mark point $W_{//}$ at a perpendicular distance from A B = O $W_{/}$ (Fig. 310), this distance O $W_{/}$ being limited by the length of boss.

Starting from point $W_{//}$ (Fig. 309), and keeping within the prescribed limits, draw the outline of the blade as in dotted lines of such a shape and area as may be thought best.

The projected outline (Fig. 309), may now be drawn from the development obtained, for example, mark point W on full arc F (Fig. 309) at a perpendicular distance from A B = perpendicular distance of $W_{/}$ from Y P (Fig. 310). This point W is the first point in projected outline.

Then on line O $V_{//}$ (Fig. 310) mark point $X_{/}$ at a distance from O = perpendicular distance of $X_{//}$ from A B (Fig. 309), then mark X (Fig. 309) on full curve at a perpendicular distance from A B = perpendicular distance of $X_{/}$ from Y P (Fig. 310), or simply draw horizontal line $X_{//}$ X, cutting full curve in X.

All the other points for the projected outline may be obtained from the developed outline in the same manner, and the curve drawn through them.

The plan may be drawn through the points $X_{/}$ etc., previously obtained from Fig. 309.

Fig. 308 may be drawn through points obtained from Fig. 309 and 310 together.

This method, in which a series of helices of different radii are developed, does not give the length of the edge $S_{///}$ $U_{///}$ of the blade exactly, but the error is trifling, and by neglecting it the method is simplified.

If the projected view of the blade (Fig. 309) is to be determined first, then part of the operation will simply require to be reversed.

The blade is 8 inches thick at the root and tapers to $\frac{3}{4}$ inch thick at the tip; diameter of boss 2 feet 9 inches, depth of boss 2 feet 2 inches, taper of hole in boss 1 inch in 18 inches. Total developed surface 64·8 square feet.

The Thickness of the root of the Blade of a Screw-propeller per foot in diameter of the propeller may be = $\frac{1}{2}$ inch for cast-iron, $\frac{7}{16}$ inch for gun-metal, and $\frac{3}{8}$ inch for mild-steel. The entrance-edge of the blade should be as sharp as practicable.

The Velocity of the Blades of a Screw-propeller through the Water, consequent upon its revolution and the forward movement of the vessel, at different points from the centre of the boss: or the velocity or distance passed through the water each revolution by any point in the blade from the boss to the extremity may be found by this formula:

$$\sqrt{(2\pi r)^2+(P-S)^2}=D.$$

D being the distance passed through by the point each revolution, P the pitch of the propeller, and S the slip each revolution, or in other words: D is = hypotenuse of a right-angled triangle where the base is = the circumference of the circle of any diameter of the propeller, and the perpendicular is = the actual progress of the vessel each revolution of the propeller.

Example: Required the velocity of the blades of a screw-propeller through the water, consequent on one revolution and the forward movement of the vessel, at a point in the blade 3 feet 4 inches radius or 6 feet 8 inches diameter, the pitch of the screw-propeller being 27·77 feet and the slip 10 per cent.

Then $(3{\cdot}1416\times3{\cdot}333$ feet radius $\times\ 2)^2=439{\cdot}56$.

and $(27{\cdot}77-10$ per cent. slip$)^2=625$. Then $\sqrt[2]{(439{\cdot}56+625)}=32{\cdot}62$ feet the velocity for each revolution.

The Pitch of a Screw-Propeller is the distance, in the line of the shaft, from one convolution to the next, or the distance the propeller would advance in one revolution if it worked in an unyielding or solid substance.

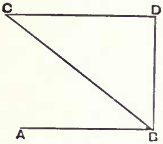


Fig. 311.—Diagram of pitch of a screw-propeller.

To Measure the Pitch of a Screw-Propeller. Let A B in Fig. 311 = the line of shafting, draw the angle A B C that the blade makes with the shaft at N feet from the centre of the shaft. Draw the line B D perpendicular to A B, and make B D equal in length to $2\ N\times3{\cdot}1416$. From D draw the line C D parallel to A B, then C D = the pitch of the screw-propeller.

The Pitch of a Screw-Propeller necessary to maintain a given speed of Screw with a given number of revolutions may be found by the following formula:—

Let K = the speed of the screw-propeller in knots per hour; L = the length of a knot in feet = 6080 feet; R = the number of revolutions; P = the pitch of the screw-propeller in feet.

$$\text{Then } P=\frac{K\times L}{R}.$$

Example 1: Required the pitch of a screw-propeller necessary to maintain a progression of 10 knots an hour, if there were no slip, with 3000 revolutions.

Then (10 knots × 6080 feet) ÷ 3000 = 20·266 feet pitch.

Example 2: A ship is required to steam 10 knots per hour when the screw-propeller is making 58 revolutions per minute, what must be the pitch of the screw?

Then (10 × 6080) ÷ (58 × 60) = 17·47 feet.

When the Pitch of a Screw-Propeller is altered, if the mean effective pressure of the indicator diagram remains unaltered, the product of the pitch by the square of the knots per hour is not altered.

The Speed of a Steamship, if the Pitch of the Screw-Propeller be altered, the indicator diagram of the engine remaining the same, may be found by this *Rule*:—

$$\frac{\text{Old pitch} \times \text{old knots}^2}{\text{New pitch}} = \text{new knots}^2, \text{ and } \sqrt[2]{\text{new knots}^2} = \text{knots.}$$

Example: The pitch of a screw-propeller is 20 feet, and the speed of the ship is 10 knots an hour. Required the speed, S, if the pitch be altered to 17 feet, the indicator diagram remaining the same.

Then $S^2 \times 17$ feet the new pitch $= 10$ knots$^2 \times 20$ feet, the old pitch,

And $\frac{20 \times 10^2}{17} = \sqrt[2]{117 \cdot 64} = 10 \cdot 82$ knots.

When both the Pitch and the Speed of a Screw-Propeller are altered the speed of the ship may be found by the following *Rule*:—

$$\frac{\text{New pitch} \times \text{new revolutions} \times \text{old knots}}{\text{Old pitch} \times \text{old revolutions}} = \text{new speed in knots.}$$

Example: The pitch of a screw-propeller is 20 feet, the number of revolutions per minute 60, and the speed of the ship 10 knots an hour. What would be the speed of the ship if the pitch were altered to 22 feet and the number of revolutions increased to 75 per minute, the slip of the screw remaining the same.

Then $\frac{22 \times 75 \times 10}{20 \times 60} = 13 \cdot 75$ the speed of the ship after altering the pitch and increasing the speed of the propeller.

The Slip of a Screw-Propeller is the difference between the actual advance of the ship and that due to the speed of the propeller, or the amount of work lost from the screw working in a yielding substance. Slip varies from 10 to 25 per cent.

Let v = the speed of the screw-propeller, and V = the speed of the ship.

Then $v - V$ = the slip of the screw, and $\frac{v-V}{v}$, the slip of the screw expressed as a fraction.

$\therefore \frac{v-V}{v} \times 100$ = the slip expressed as a percentage of the speed.

The Slip is calculated on the Speed of the Screw-Propeller.

Example 1: A ship is required to steam at the rate of 12 knots, and the crank-shaft of the engine is to make 76 revolutions; what must be the pitch of the screw-propeller if 20 per cent. be allowed for slip?

Then if 20 per cent. be deducted from the speed it leaves $\frac{76 \times 20}{100} = 15 \cdot 20$, $76 - 15 \cdot 20 = 60 \cdot 8$ knots as the effective speed of the propeller, and $\frac{12 \times 6080}{60 \cdot 8 \times 60} = 20$ feet, the pitch of the screw-propeller required.

Example 2: A steamship has a screw-propeller making 63 revolutions per minute, the pitch of the screw is 18 feet, and the slip 15 per cent. Required the speed of the ship in knots per hour?

Then the speed of the screw-propeller per hour in feet $=18\times63\times60=$ 68240 feet.

The speed of the screw-propeller, if there were no slip, would be in knots per hour$=\dfrac{18\times63\times60}{6080 \text{ feet}}=11{\cdot}22$ knots. Taking off 15 per cent. slip, leaves $\dfrac{85}{100}$, and the speed of the ship will be $=\dfrac{18\times63\times60\times85}{6080\times100}$ $=9{\cdot}54$ knots per hour.

The Slip of a Screw-Propeller may be ascertained by comparing the distance steamed by the ship with the progression of the screw-propeller, or distance run by the engines.

Example: A ship steamed a distance of 27 knots, the screw-propeller made 9120 revolutions, the pitch of the screw was 20 feet. Required the slip of the screw-propeller?

$$\text{Then } \frac{9120 \text{ revolutions}\times20 \text{ feet pitch}}{6080 \text{ feet}}=30 \text{ knots.}$$

The difference between the distance steamed by the ship and the progression of the screw-propeller is $30-27=3$ knots, and the slip of the screw-propeller is $\dfrac{3\times100}{30}=10$ per cent.

Negative Slip of a Screw-Propeller is the difference between the velocity of the screw-propeller and that of the ship, when a ship is propelled at a faster speed than would be obtained if the screw-propeller worked in an unyielding substance. Negative slip, when it exists, is probably due to the following cause:—The propeller throws the water outwards and backwards in the form of a hollow cone, this hollow is filled up partly by the stream following the ship, and partly by water which returns to fill up the vacuum caused by the centrifugal action of the screw-propeller: the water coming in these two directions impinges upon the screw-propeller and causes additional thrust.

The Angle of a Screw-Propeller is the inclination of the thread of the screw to the horizon. The angle may be found from the tangent of the angle, which is obtained as follows:—

$$\text{Tangent of the angle of the screw}=\frac{\text{Pitch}}{\text{Circumference}}.$$

Example: The diameter of a screw-propeller is $11{\cdot}026$ feet, the pitch is 20 feet. Required the angle of the screw.

Then $11{\cdot}026\times3{\cdot}1416=34{\cdot}641016$ feet, the circumference of the screw, and $\dfrac{20 \text{ feet pitch}}{34{\cdot}641016}={\cdot}57735$ tangent $\therefore$ angle$=30°$.

The Pitch of a Screw-Propeller, when the angle and the diameter are given, may be found by the following *Rule:*—

Pitch of screw$=$tangent of angle$\times$circumference of screw.

Example: Required the pitch of screw-propeller in feet, when the angle is 30°, and the diameter 11·026 feet.

The tangent of angle 30°=·57735.

Then $11·026 \times 3·1416 = 34·641016 \times ·57735 = 20$ feet, the pitch of the screw-propeller.

The Thread of a Screw-Propeller is the distance along the edge of the blade.

The Length of a Screw-Propeller is the fraction of the pitch actually used.

The Diameter of a Screw-Propeller is the diameter of the circle described by the tips of the blades.

The Area of the Blade of a Screw-Propeller is the surface of the blade.

The Disc-Area of a Screw-Propeller is the area of the circle described by the tips of the blades.

The Projected-Area of the Blade of a Screw-Propeller is the area that would be traced by a pencil held parallel to the propeller-shaft, traversing the edge of the blade of the propeller, on a plane at right angles to the shaft.

The Expanded-Area of the Blade of a Screw-Propeller is the same as that of a sheet of paper which would cover the blade.

The Boss of a Screw-Propeller is generally bored to a taper of 1 inch in 18 inches, in propellers of moderate size, and to a taper of 1 inch per foot in large propellers. The boss is keyed on the shaft with one key, sunk into the shaft, for moderate-sized propellers, and with two keys for large propellers, as shown in Fig. 312, which represents a boss 3 feet $1\frac{1}{2}$ inch

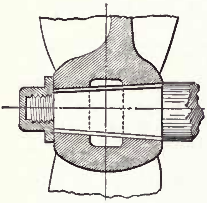


Fig. 312.—Boss of a s rew-propeller.

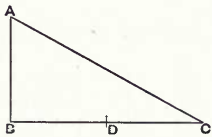


Fig. 313.—Diagram of the surface of a screw-propeller-blade.

diameter, and 3 feet long, of a screw-propeller 18 feet diameter. The diameter of the large end of the tapered hole is $15\frac{1}{2}$ inches, and the taper being 1 inch per foot, the diameter of the small end of the tapered hole will be $15\frac{1}{2} - \frac{(36 \text{ inches} \times 1 \text{ inch taper})}{12 \text{ inches}} = 12\frac{1}{2}$ inches. A nut is screwed on the end of the shaft to tighten against the boss, and a check-plate is fitted to prevent the nut working loose.

The width of each Key of a Screw-Propeller, when fixed on the shaft, as shown in Fig. 312, should = diameter of shaft × ·226, and the key should enter the boss to a depth, measured at the side of the key = diameter of shaft × ·08. Hence each key of the above propeller should be

$15{\cdot}5 \times {\cdot}226 = 3\frac{1}{2}$ inches wide, and the depth of each keyway in the boss should be $15{\cdot}5 \times {\cdot}08 = 1\frac{1}{4}$ inch.

The Surface of a Blade of a Screw-Propeller may be found approximately as follows:—

In Fig. 313, let AB be the pitch, BC the circumference, and AC the thread of the screw-propeller: then, if the blade be considered to be made up of a large number of triangles placed side by side, and if the part ABC be taken away, leaving AC to form the blade, then if the lengths of all the lines corresponding to AC were found, their sum divided by their number would give an average length, which, multiplied by the radius, will give the approximate area of the blade. In practice it is sufficiently correct in most cases to find only three of these lines, then by dividing by three and multiplying by the radius, the area of the blade is obtained.

In the above figure AB may be considered as the length of the screw to obtain the approximation. If BC be bisected in D, and AD joined, then AC represents the longest line on the surface of the blade, AB the shortest, and AD an intermediate one.

Then the area of the blade of the screw-propeller $= \dfrac{AB + AD + AC}{3} \times$ radius.

Example: The diameter of a screw-propeller with two blades is 15 feet, the pitch is 20 feet, and the length 3 feet. Required the area of the complete screw, the surface of one blade, and the surface of two blades.

$$
\begin{aligned}
\text{Then } BC &= 15 \text{ feet diameter} \times 3{\cdot}1416 = 47{\cdot}12 \text{ feet circumference.}\\
BD &= \text{one-half of } 47{\cdot}12 = 23{\cdot}56 \text{ feet.}\\
AB &= 20 \text{ feet pitch.}\\
AC^2 &= AB^2 + BC^2 = 400 + 2220{\cdot}2944 = 2620{\cdot}2944.\\
&\therefore AC = 51.\\
AD^2 &= AB^2 + BD^2 = 400 + \tfrac{1}{4}\text{th of } 2220{\cdot}2944 = 955{\cdot}736.\\
&\therefore AD = 31.
\end{aligned}
$$

$$
\begin{aligned}
\text{and}\quad AB &= 20.\\
AC &= 51.\\
AD &= \frac{31}{102} \div 3 = 34 \times 7{\cdot}5 \text{ radius} = 255 \text{ square feet, the area of the}
\end{aligned}
$$

complete screw.

The length being 3 feet, the area of one blade will $= \frac{3}{20}$ of the area of complete screw.

The area of the complete screw is $= 255$ square feet.

The area of one blade $= \dfrac{255 \times 3}{20} = 38{\cdot}25$ square feet.

The area of two blades $= 38{\cdot}25 \times 2 = 76{\cdot}50$ square feet.

The Resistance of a Steamship to Propulsion by a Screw-Propeller may be found by this *Rule*:—

$$\text{Resistance in lbs.} = \frac{\text{indicated horse-power} \times {\cdot}6 \times 33000}{\text{pitch of screw} \times \text{revolutions per minute}}.$$

Example: Required the resistance of a steamship to propulsion by a screw, driven by engines of 800 indicated horse-power, pitch of screw-propeller 18 feet, number of revolutions per minute = 56.

$$\text{Then } \frac{800 \times \cdot 6 \times 33000}{18 \text{ feet pitch} \times 56 \text{ revolutions}} = 15714 \text{ lbs.}$$

The Indicated Thrust of a Screw-Propeller may be found by this *Rule*:—

Thrust = pressure in foot-lbs. on the thrust × pitch × number of revolutions.

$$\therefore \text{Thrust} = \frac{\text{indicated horse-power} \times 33000}{\text{pitch} \times \text{revolutions}}.$$

Example: Required the thrust of a screw-propeller 20 feet pitch, driven by engines of 1000 indicated horse-power at 60 revolutions per minute, slip 10 per cent.

Then $\frac{1000 \times 33000}{20 \times 60} = 27500$ lbs. thrust, being the total power exerted by the engines, but in practice only about 38 per cent. of the indicated horse-power is utilised or effectively applied in the thrusting-force of a screw in propelling a ship.

And the horse-power at the thrust-block will be . . $\left.\right\}$ $\frac{1000 \times 33000 \times 38}{20 \times 60 \times 100} = 10450$ lbs. thrust.

The loss by slip is $\frac{10450 \times 10}{100}$ . . . . = 1045 lbs.

The actual power on the thrust . . . 9405 lbs.

The Indicated Horse-Power required to exert a given Thrust or power on a thrust-block, may be found by the following *Rule*:—

$$\text{Indicated horse-power} = \frac{\text{pressure in lbs. on the thrust-block} \times \text{pitch} \times \text{revolutions}}{33000}.$$

Example: What indicated horse-power would be required to exert a force or thrust of 27500 lbs. on a thrust-block, with a screw-propeller of 20 feet pitch, making 60 revolutions per minute?

Then $\frac{27500 \text{ lbs. thrust} \times 20 \text{ pitch} \times 60 \text{ revolutions}}{33000} = 1000$ indicated horse-power.

If the slip be 10 per cent., the power lost will $= \frac{1000 \times 10}{100} = 100$ horse-power, and the power actually employed will = 1000 − 100 = 900 indicated horse-power.

The Percentage of the Whole Power which is applied to the Thrust-Block to push the Vessel ahead may be found as shown by the following example.

Example: The power on a thrust-block is 16500 lbs., the speed of the ship is 10·84 knots per hour, with a screw-propeller 20 feet pitch, making 60 revolutions per minute, the engines developing 1000 indicated horse-power. What percentage of the power as actually applied at the thrust-block in pushing the ship ahead, including the power lost as slip? What power is applied at the thrust-block, and what power is lost by slip?

Then, the knots run by the engine are $= \frac{20 \times 60 \times 60}{6080 \text{ feet}} = 11{\cdot}84$ knots.

The percentage of speed lost by slip is $= \frac{(11{\cdot}84 - 10{\cdot}84) \times 100}{11{\cdot}84} = 8{\cdot}5$ per cent.

The power used, in percentage of the available power, is $= \frac{16500 \times 20 \times 60}{1000 \times 33000} \times 100 = 60$ per cent.

The loss by slip $= \frac{60 \times 8{\cdot}5}{100} = 5{\cdot}1$ per cent.

The actual power at the thrust-block, in percentage $= 60 - 5{\cdot}1 = 54{\cdot}9$ per cent.

The actual power applied at the thrust-block $= \frac{1000 \times 54{\cdot}9}{100} = 549$ horse-power.

The power lost by slip $= \frac{1000 \times 5{\cdot}1}{100} = 51$ horse-power.

The Surface of a Screw-Propeller being oblique to the direction of the thrust, the area of the screw-disc is less effective for thrust than an equal area of feathering float with the same slip. The following formula, given in another form by Rankine, gives approximately the area, A, of an equivalent pair of paddle-floats, which will be equal to the effective area of the screw-disc of a given diameter, when the pitch is between ·9 D and 2·1 D.

Let P = the pitch of the screw-propeller.
D = the diameter of the screw-propeller.
B = the diameter of the boss.

$$\text{Then } A = \left({\cdot}7854 - \frac{P}{5\,D}\right) \times (D^2 - B^2);$$

$$\text{or } A = \left({\cdot}7854 - \frac{P}{5\,D}\right)(D + B)(D - B).$$

Example: The diameter of a screw-propeller is 20 feet, and the pitch is 21 feet, the diameter of the boss is 4 feet: what is the equivalent area?

Then $\cdot 7854 - \frac{21}{5 \times 20} = \cdot 5754$ and $20^2 - 4^2 = 384$, and $\cdot 5754 \times 384 = 220{\cdot}9536$.

The Thrust in lbs. of a Screw-Propeller, T, may be found by this *Rule:*—

T = 5·66 × the slip in knots per hour × speed of the propeller in knots per hour × the equivalent area of stream or column of water sent astern, which may be found by the previous rule.

Example: The screw-propeller described in the previous example makes

60 revolutions per minute, the speed of the ship is 10 knots an hour, and the equivalent area of stream is 220·95, as found in the previous example: what is the thrust in lbs?

Then $\frac{21 \text{ pitch} \times 60 \text{ revolutions} \times 60}{6080} = 12{\cdot}41$, the speed of the propeller in knots per hour.

And 12·41 knots − 10 knots = 2·41, the slip in knots per hour.

Then 5·66 × 2·41 slip × 12·41 speed of propeller × 220·95 equivalent area = 37468·46 lbs. thrust.

If fresh water, use a constant of 5·5 in the above rule instead of 5·66, which is for sea-water.

Horse-Power lost by slip of a Screw-Propeller.—In addition to other losses in effecting the propulsion of a screw-steamer, the yielding of the column of water on which the screw-blades act, is a loss of power equal to the product of the thrust by the difference between the advance of the ship and the speed of the screw. Applying this to the previous example, the loss of work by slip in actual horse-power is as follows:—

$$\frac{2{\cdot}41 \text{ slip} \times 6080 \text{ feet}}{60 \text{ minutes}} = 244{\cdot}21, \text{ the slip in feet per minute,}$$

$$\text{and } \frac{37468{\cdot}46 \text{ lbs. thrust} \times 244{\cdot}21}{33000} = 277{\cdot}27 \text{ horse-power lost by slip.}$$

Horse-power required to drive a Screw-Propeller.— Effective horse-power is the proportion of the indicated horse-power available for performing useful work after deducting the power absorbed by friction. In the case of steamships it is found that only about 38 per cent. of the indicated horse-power is utilised or effectively applied in the thrusting-force of a screw-propeller in propelling a ship. The effective horse-power required to drive the propeller described in the two previous examples may be found as follows:—

$\frac{21 \text{ feet pitch} \times 60 \text{ revolutions} \times 37468{\cdot}46 \text{ lbs. thrust}}{33000} = 1430{\cdot}64$ nett, or effective horse-power required to drive the screw-propeller; and the total or gross indicated horse-power required will be $= 1430{\cdot}64 \times \frac{100}{38} = 3764{\cdot}84$ horse-power.

Horse-Power lost in propelling a Ship with a Screw-Propeller.—The difference between the effective horse-power and the indicated horse-power found in the previous example = 3764·84 − 1430·64 = 2334·2 horse-power represents the loss of power which takes place in propelling a screw-steamer though the water, equal to $\frac{1430{\cdot}64 \times 100}{2334{\cdot}2} = 62$ per cent. of the total power expended.

The Resistance of a Ship's Model determines the resistance of the ship and the horse-power required to propel it. Mr. Froude found that for two vessels of similar forms, as, for instance, a ship and her model, the corresponding speeds are to one another as the square roots of the similar dimensions; and at these corresponding speeds the resistances of the two vessels are to one another as the cubes of the similar dimensions, or as their volumes. The resistances may be calculated as shown by the following example.

Example: Required the resistance and horse-power necessary to propel a ship 360 feet long, at a speed of 15 knots per hour. Then, if the model of the ship be, say, 12 feet long, its speed corresponding to that of the ship will be $= 15 \text{ knots} \times \sqrt{\dfrac{12 \text{ feet length of model}}{360 \text{ feet length of ship}}} = \cdot 28$ knot per hour. If the resistance of the model at that speed = say 4·25 lbs. the resistance of the ship at 15 knots per hour will be $= \left(\dfrac{360 \text{ feet length of ship}}{12 \text{ feet length of model}}\right)^3 \times$ 4·25 lbs. = 114750 lbs.;

or, $360 \div 12 = 30$, and $\dfrac{30 \times 30 \times 30 \times 4{\cdot}25 \text{ lbs.} \times 15 \text{ knots} \times 6080 \text{ feet}}{33000 \times 60} =$ 5281 effective horse-power.

The Speed of a Steam-ship in Knots per Hour, due to a given Pitch and Speed of Screw-Propeller may be found by the following *Rule:*—

Multiply the pitch of the propeller in feet by the number of revolutions per minute and by 60, and divide the product by 6080.

Example: The pitch of a screw-propeller is 20 feet, and its speed, or number of revolutions, is 62 per minute, how many knots will the ship go per hour, making no allowance for slip?

The ship advances at each revolution of the screw 20 feet.

The ship advances in one minute $20 \times 62 = 1240$ feet.

The ship advances in one hour $20 \times 62 \times 60 = 74400$ feet.

The speed in knots per hour $= \dfrac{\text{speed in feet per minute} \times 60}{\text{length of a knot in feet}} =$

$\dfrac{20 \text{ feet} \times 62 \text{ revolutions} \times 60}{6080 \text{ feet.}} = 12{\cdot}23$ knots per hour.

The Life of a Screw-propeller depends greatly upon its resistance to corrosion and corrosive-pitting, caused by air drawn down to the back of the propeller, the backs of the blades being most affected. Corrosion blunts the blades, increases friction, and causes loss of power. Gun-metal has great resistance to corrosion, and a gun-metal propeller may, excepting accidents, last as long as the ship. Cast-iron blades are liable to corrosive-pitting, and require to be renewed on an average every six years. Steel blades are more liable to corrosive-pitting than cast-iron; and are less true to form, owing to being liable to twist both in casting and annealing; they require renewal on an average every three and one-half years.

The Resistance of a Screw-propeller to Friction decreases as the sectional area of the blade decreases, and as the smoothness and accuracy of the blade increases.

Froude's Formulæ for Screw-Propellers are as follows:—

Indicated thrust.

Let I = indicated thrust; M = mean piston-pressure; T = total piston-travel per revolution; P = pitch of screw-propeller; N = number of revolutions; IHP = indicated horse-power.

$$\mathrm{I} = \frac{\mathrm{M} \times \mathrm{T}}{\mathrm{P}} = \frac{33000 \times \mathrm{IHP}}{\mathrm{P} \times \mathrm{N}}.$$

Indicated thrust is resolved into the following six elements:—

No. 1. The ship's nett resistance, or useful thrust.

No. 2. Augment of resistance due to negative pressure created about the ship's stern by the action of the screw. This is nearly proportional to the useful thrust.

No. 3. Water friction of screw. This is also nearly proportional to the useful thrust.

No. 4. Constant friction, or friction of engine without external load. This may also be taken as nearly proportional to the useful thrust.

No. 5. Friction due to external load. This may be taken as constant at all speeds.

No. 6. Air-pump and feed-pump resistance. This may be taken as nearly proportional to the square of the number of revolutions.

The above six elements are force-factors, and when multiplied by $\frac{\text{the speed of the ship in feet per minute}}{33000}$ constitute the ship's horse-power as fundamentally due to the progress.

Let EHP = effective horse-power—that is, the power due to the nett resistance of the ship; SHP=ship's horse-power; IHP=indicated horse-power.

Then the ship's horse-power due to the several elements is as follows:—

Ship's horse-power due to

No. 1 = EHP.	No. 4 = ·143 SHP.
No. 2 = ·4 EHP.	No. 5 = ·143 SHP.
No. 3 = ·1 EHP.	No. 6 = ·075 SHP.

Or in combination SHP=1·5 EHP + ·361 SHP. So that ·639 SHP = 1.5 EHP;

$$\text{or,}\quad \text{SHP} = \frac{1\cdot5}{\cdot639}\ \text{EHP} = 2\cdot347\ \text{EHP};$$

To this must be added—Slip = ·1 SHP., making IHP = 1·1 SHP.

$$\text{Thus IHP} = 2\cdot582\ \text{EHP} = \frac{100}{38\cdot7}\ \text{EHP}; \text{ or, EHP} = \cdot387\ \text{IHP}.$$

To convert the formula from one adapted to high speed only to one adapted to all speeds, it is necessary to keep the term involving constant friction separate from the rest, for it represents simply the effect of a constant resistance operating with the existing speed of the engine.

In shaping the formula the co-efficient 2·7, derived from experience, will be adhered to, instead of the co-efficient 2·582, as the latter is built up from somewhat hypothetical data, assuming, however, that the constant friction is equal throughout to one-seventh of the maximum load.

Of the 2·7 EHP which make up the IHP at the maximum speed V, one-seventh part, or ·385, is the part due to constant friction, leaving 2·315 as due to the other sources of expenditure of power. And to express the IHP due to constant friction at any other speed v, the coefficient must be altered in the direct ratio of the speed, so that the term becomes $\frac{v}{V} \times \cdot385 \times$ EHP at designed maximum speed. Thus the formula for IHP at any speed v is as follows :—

$$\text{IHP} = 2{\cdot}315 \text{ EHP} + {\cdot}385 \frac{v}{\text{V}} \times (\text{EHP due to v});$$

or, if the useful is finally severed from the collateral expenditure of power, it stands thus :—

$$\text{IHP} = \text{EHP} + 1{\cdot}315\text{EHP} + {\cdot}385 \frac{v}{\text{V}} \times (\text{EHP due to v}).$$

Thom's Formulæ for Screw-Propellers,* given below, are based upon the assumption that the area of propeller-disc should be proportional to the indicated horse-power, divided by the cube of the speed, and the same rule applies to the projected area of the propeller and also to the surface. In this way constants can be obtained for the disc-area, the projected-area and the expanded area as follows :—

$$\text{Disc-area constant} = \frac{\text{Area of propeller disc} \times \text{speed of ship in knots}^3}{\text{Indicated horse-power}}.$$

$$\text{Projected-area constant} = \frac{\text{Projected-area of propeller} \times \text{speed of ship in knots}^3}{\text{Indicated horse-power}}.$$

$$\left.\begin{array}{l}\text{Expanded area}\\ \text{constant}\end{array}\right\} = \frac{\text{Expanded-area of propeller} \times \text{speed of ship in knots}^3}{\text{Indicated horse-power}}.$$

Table 87.—PARTICULARS OF SCREW-PROPELLERS AND CONSTANTS.

Ship.	Length of Ship.	Disc Constant.	Projecting Surface Constant.	Feet per Sec. Speed per Tips.	Remarks.
City of Rome .	542	220	69	4715	Single Screw-propeller.
Normandie .	459	250	66	4099	
Furnessia . .	445	223	69	3654	
Eden . .	300	211	64	3080	
Yorouba . .	270	213	63	3202	
Taygete . .	260	238	56	3166	
Kow-shing . .	250	171	69	3369	
S. Y. Monarch	152	221	65	4040	
,, Aries .	138	179	56	2986	
Twin-screw Fenella.	200	244	64	2890	
Twin-screw H.M.S. Fearless.	220	277	67	5022	Estimated with a speed of 17·5 knots, and 3370 I.H.P.

The Speed of a Steam-ship may be found by the following formulæ which are based on the assumption that the resistance offered by the water to the motion of a ship varies as the square of the speed, and that the power required to overcome this resistance varies as the cube of the speed, which is not quite correct, because the indicated power varies as a higher

* See a Paper on "Atlantic Steamers" read by Mr. Wm. John before the Institution of Naval Architects.

power than the cube of the speed, but the *Rules* are useful for arriving at an approximation of the speed.

$$\text{Speed}^3 = \frac{\text{I H P} \times \text{constant}}{\text{Displacement}^{\frac{2}{3}}}.$$

$$\text{Speed}^3 = \frac{\text{I H P} \times \text{constant}}{\text{Area of midship section}}.$$

$$\text{Constant} = \frac{\text{Speed}^3 \times \text{displacement}^{\frac{2}{3}}}{\text{I H P}}.$$

$$\text{Constant} = \frac{\text{Speed}^3 \times \text{area of midship section}}{\text{I H P}}$$

Where I H P = the indicated horse-power:—

$\text{Displacement}^{\frac{2}{3}} = \sqrt[3]{(\text{displacement})^2}$.

Example: Required the constants for a steam-yacht, fitted with engines of 800 indicated horse-power, having a speed of 12 knots an hour; a displacement of 850 tons; the area of immersed midship section being 273 square feet.

Then constant $= \frac{12^3 \text{ knots} \times 850^{\frac{2}{3}}}{800} = \frac{1728 \times \sqrt[3]{850 \times 850}}{800 \text{ I H P}} = \frac{1728 \times 90}{800} =$ 194 constant.

And constant $= \frac{12^3 \text{ knots} \times 273 \text{ square feet}}{800 \text{ I H P}} = \frac{1728 \times 273}{800} = 590$ constant.

The Speed may be found with the above constants as follows:—

$\text{Speed}^3 = \frac{800 \text{ I H P} \times 194 \text{ constant}}{850^{\frac{2}{3}} \text{ displacement}} = \frac{155200}{\sqrt[3]{850 \times 850}} = \frac{155200}{90} = \sqrt[3]{1725}$ $= 12$ knots.

$\text{Speed}^3 = \frac{800 \text{ I H P} \times 590}{273 \text{ square feet}} = \frac{472000}{273} = \sqrt[3]{1728} = 12$ knots.

The Indicated Horse-Power required to Propel a Ship may be found by the converse of the previous formulæ:—

$$\text{Indicated horse-power of engines} = \frac{\text{Speed}^3 \times \text{displacement}^{\frac{2}{3}}}{\text{Constant}}.$$

$$\text{Indicated horse-power of engines} = \frac{\text{Speed}^3 \times \text{area of midship section}}{\text{Constant}}.$$

Taking the particulars from the previous example, the horse-power of the engines required to propel that steam yacht, will be =

$\frac{12^3 \times 850^{\frac{2}{3}}}{194} = \frac{1728 \times \sqrt[3]{850 \times 850}}{194} = \frac{1728 \times 90}{194} = 800$ indicated horse-power.

Or, $\frac{12^3 \times 273 \text{ square feet}}{590} = \frac{1728 \times 273}{590} = 800$ indicated horse-power.

The Indicated Horse-Power and Displacement of a number of Modern Steamships is given in Table 88*, which also gives the midship-area, the speed on trial trips, the co-efficients for the lines both from the block or parallelopipedon, and also from the midship section prism, together with the length and angle of entrance obtained by Kirk's rule, the Admiralty displacement co-efficient, and the coal-consumption per day and per indicated horse-power per hour. This table contains particulars of some of the most important of the Atlantic steamers, and also of a

* The Author is indebted for this Table to a paper on "Atlantic Steamers," read by Mr. Wm. John before the Institution of Naval Architects.

Table 88.—Dimensions, Moulded Draught, Midship-Area, Displacement, Indicated Horse-Power, Speed, Block Co-efficient, Midship Section Co-efficient, and other useful Data of Modern Steam-Ships.

Name.	Length.		Breadth.		Moulded draught.		Midship-area.	Displacement.	Indicated H.P.	Speed.	Block co-efficient.	Midship section co-efficient.
	ft.	in.	ft.	in.	ft.	in.						
City of Rome	542	6	52	0	21	$5\frac{1}{2}$	1031	11,230	11,890	18·235	·649	·925
Normandie	459	4	49	11	19	$9\frac{3}{4}$	892	7,975	6,959	16·66	·614	·901
Furnessia	445	0	44	6	22	$2\frac{1}{2}$	893	8,578	4,045	14	·682	·903
Arizona	450	0	45	$1\frac{1}{2}$	18	9	758	6,415	6,300	17	·589	·895
Orient	445	0	46	0	21	$4\frac{1}{2}$	904	7,770	5,433	15·538	·621	·919
Stirling Castle	420	0	50	0	22	3	990	7,600	8,396	18·4	·569	·889
Elbe	420	0	44	9	20	0	807	6,350	5,665	16·571	·591	·901
Pembroke Castle	400	0	42	0	17	0	648	5,130	2,435	13·25	·623	·623
Umbria and Etruria	500	0	57	0	22	6	1090	9,860	14,321	20·18	·538	·896
Aurania	470	0	57	0	20	0	1020	8,800	8,500	*17·5	·575	·942
America	441	8	51	3	—		—	6,500	—	*17·8	—	—
Oregon	501	0	54	2	23	8	1150	11,000	13,300	18·3	·599	·849
Servia	515	0	52	0	23	$3\frac{1}{2}$	1046	10,960	10,300	*16·9	·610	·862
Scotia, P.S.	369	0	47	6	19	9	867	6,000	4,632	*14·31	·605	·92
Alaska	500	0	50	0	21	0	949	9,210	—	—	·614	·904
Aller	438	0	48	0	21	0	907	7,447	7,974	17·9	·590	·899
Ems	430	0	46	10	20	$7\frac{1}{2}$	877	7,030	7,251	17·55	·593	·907

Name.	Prismatic midship section co-efficient.	$\frac{D^{\frac{2}{3}} \times S^3}{\text{I.H.P.}}$	$\frac{D^{\frac{2}{3}} \times S^3}{\text{I.H.P.} \times \sqrt{\text{ent.}}}$ 10.	Kirk's system.		Coal consumption.		Cylinders.		Boilers.		Working pressure.
				Length of entrance.	Angle.	Per day.	Per I.H.P.	Diameter.	Stroke.	Heating surface.	Bar surface.	
								Inches.	Ins.			lbs.
City of Rome	·702	255	201·3	161·27	8° 29′	185	2·2	3 @ 46 3 @ 86	72	29,286	1398	90
Normandie	·681	265	219·5	146·41	8° 44′	148	2	3 @ $35\frac{7}{16}$ 3 @ $74\frac{7}{8}$	67	21,404	756	85·2
Furnessia	·755	284	273	108·7	10° 28′	97	2·2	49—100	66	10,396	440	90
Arizona	·658	269·2	217	153·79	7° 30′	—	—	1 @ 62 2 @ 90	66	—	—	90
Orient	·676	270·8	225	144·17	8° 21′	—	—	1 @ 60 2 @ 85	60	—	—	75
Stirling Castle	·639	286·8	233·7	151·3	8° 22′	—	—	1 @ 62 2 @ 90	66	21,161	787	100
Elbe	·655	275·5	229	144·6	7° 56′	—	—	1 @ 60 2 @ 85	60	—	—	—
Pembroke Castle	·692	284	258	122·9	8° 49′	44	1·7	43 and 86	57	7,896	288	99
Umbria and Etruria	·637	260	191·8	184	6° 52′	315	2·1	1 @ 71 2 @ 105	72	38,817	1606	110
Aurania	·632	266	204·6	170	8° 38′	215	2·2	1 @ 68 2 @ 01	72	23,284	1001	—
America	—	—	—	—	—	185	—	1 @ 63 2 @ 91	66	—	882	—
Oregon	·67	227·9	190	164·3	9° 39′	310	2·2	1 @ 70 2 @ 104	72	38,047	1428	110
Servia	·71	231	192	145·3	10° 42′	205	2	1 @ 72 2 @ 100	78	27,483	1014	—
Scotia, P.S.	·65	208·9	186	126·8	13° 21′	168	3·4		—	—	—	—
Alaska	·679	—	—	160·23	8° 2′	—	—	1 @ 68 2 @ 100	72	—	—	100
Aller	·656	277	225	150·6	8° 10′	—	—	1 @ 44 1 @ 70 1 @ 100	72	22,630	799	150
Ems	·652	273	223	149·4	8° 40′	—	—	1 @ 62 2 86	60	19,700	780	100

* Mean speed of a voyage across the Atlantic Ocean.

number of other typical ships. The co-efficient given in the fourth column of the bottom half of the table, viz.:—

$$\frac{\text{Displacement}^{\frac{2}{3}} \times \text{speed}^3}{\text{Indicated horse-power} \times \dfrac{\sqrt{\text{entrance}}}{10}}$$

generally comes out for ships of similar type more nearly a constant in the true sense of the word than the corresponding Admiralty constant.

The Admiralty Knot = 6080 feet : 1 statute mile = 5280 feet. The length of a knot is sometimes taken at 6082·66 feet, and also at 6086 feet.

The Speed of a Ship in knots × 1·15 = miles.

The speed of a ship in miles × ·87 = knots.

The speed of a ship in feet per minute × ·01 = knots per hour.

One knot per hour = 1·688 foot per second.

One knot per hour = ·5144 metre per second.

One metre per second = 1·944 knot per hour.

One foot per second = ·592 knot per hour

One foot per second = ·682 mile per hour.

The most Economical Speed for a Steamer going up a River down which the tide is coming, is half as fast again as the tide. The progress made by the ship per hour will be equal to the difference between the speed of the ship per hour and the velocity of the tide or current per hour.

Example: A steamer is going up a river, down which the tide is coming at 3 miles an hour: how fast must she steam and what progress will she make per hour?

Then $3 \times \frac{3}{2} = 4 \cdot 5$ miles per hour, the speed required, and the progress will be $4 \cdot 5 - 3 = 1\frac{1}{2}$ mile per hour.

The Number of Revolutions of a Shaft vary as the Cube of the Speed. The following examples show the application of this *Rule*, which applies to all similar questions:—

Example 1: The number of revolutions of the crank-shaft of a marine-engine are 40, and the speed of a ship is 8 knots per hour: what will the speed be if the number of revolutions be increased to 50?

Let V be the required speed. Then $\frac{V^3}{8} = \frac{50}{40}$.

$$\therefore V^3 = \frac{50}{40} \times 8^3 = 640.$$

$$\therefore V = \sqrt[3]{640} = 8 \cdot 617 \text{ knots.}$$

Example 2: The horse-power of a pair of marine-engines is 500, and the speed of the ship is 9 knots per hour; it is required to increase the speed to 12 knots: what power of engines will the ship require?

$$\text{Then } \frac{500 \times 12^3}{9^3} = 1185 \text{ horse-power.}$$

Example 3: With a pair of marine-engines of 1185 horse-power a steamship has a speed of 12 knots an hour: what will be the speed of the ship when the engines are developing only 500 horse-power?

$$\text{Then } \frac{500 \times 12^3}{1185} = \sqrt[3]{729} = 9 \text{ knots per hour.}$$

Example 4: If a pair of marine-engines of 900 horse-power give a ship a speed of 9 knots an hour, what speed will be obtained with new engines of 1400 horse-power?

$$\text{Then } \frac{1400 \times 9^3}{900} = \sqrt[3]{1134} = 10{\cdot}428 \text{ knots per hour.}$$

Example 5: A steamship has four boilers; when two of them are in use the speed of the ship is 8 knots an hour: what will be the speed when three boilers are used, and also when all the four boilers are used?

Then $\frac{3 \times 8^3}{2} = \sqrt[3]{768} = 9{\cdot}158$ knots per hour, the speed with three boilers.

And $\frac{4 \times 8^3}{2} = \sqrt[3]{1024} = 10{\cdot}079$ knots per hour, the speed with four boilers.

The Number of Revolutions of a Screw-Propeller per Knot, if there were no slip, would be $= \frac{6080 \text{ feet}}{\text{Pitch in feet}}$.

The Number of Revolutions of a Screw-Propeller per Hour, if there were no slip, would be $= \frac{6080 \text{ feet}}{\text{Pitch in feet}} \times$ number of knots per hour.

The Number of Revolutions of a Screw-Propeller per Hour including Slip $= \frac{6080 \times (100 + \text{slip})}{\text{Pitch in feet} \times 100} \times$ number of knots per hour.

Example: A steamship is to have a speed of 14 knots per hour; the pitch of the screw-propeller is 20 feet: at what speed must her engines run, assuming a slip of 15 per cent.?

Then, if there were no slip, the number of revolutions of the screw-propeller would be $= \frac{6080}{20} = 304$ revolutions per knot.

Assuming 15 per cent. slip the number of revolutions of the screw-propeller will be $= \frac{6080 \times 115}{20 \times 100} = 349{\cdot}6$ revolutions per knot.

And $349{\cdot}6$ revolutions per knot $\times$ 14 knots $= 4894{\cdot}4$ revolutions of the screw-propeller per hour, or $4894{\cdot}4 \div 60 = 81{\cdot}57$ revolutions per minute, the speed at which the engines must run.

Twin Screw-Propellers, shown in Fig. 314, are used when the draught of water is not sufficient to permit a single screw of sufficiently large diameter to be used. One advantage of twin-screws is, that in case of accident to one of them, the other screw can be used for propelling the ship at a reduced speed: another is, that by working one of the screws ahead, and the other astern, the ship can be turned in her own length, and in case of loss of the rudder the ship can be steered by the engines.

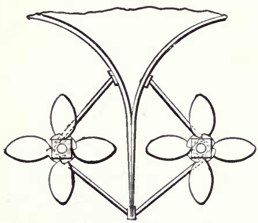


Fig. 314.—Twin screw-propellers.

Feathering Screw-Propellers are so arranged that the blades can be moved to a nearly fore and aft position, without stopping the ship, in order to remove the resistance which the screw-propeller would offer to the advance of the ship when under sail. The boss of the propeller, and also the propeller-shaft, is made hollow. Each blade is fitted to the boss with a round shank, to which a lever is fixed, and connected to a shaft inside the screw-propeller shaft; the forward end of the shaft is connected to a collar, which can be moved along the shaft by a nut screwed on the outside of the propeller-shaft.

A Pitchometer, shown in Fig. 315, is an instrument for measuring the

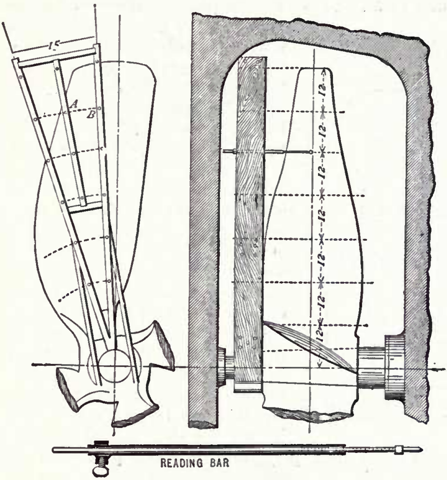


Fig. 315.—Pitchometer for screw-propellers.

pitch of a screw-propeller in its place: it enables the pitch to be quickly ascertained without calculation.

It is based on the ordinary rule used in measuring screw-propellers, that is:—

$$\frac{\text{Fraction of Pitch}}{\text{Whole Pitch}} = \frac{\text{Fraction of Circumference}}{\text{Whole Circumference}}.$$

If then, the angle between the legs of the instrument be made, a certain definite fraction of the whole 360°, the fractional pitch, as measured by inserting the reading bar at A and B, and taking the difference of the distances to the face of the propeller, will bear the same ratio to the whole pitch.

In the device, as actually constructed, the angles are $7\frac{1}{2}°$ and $15°$, or $\frac{1}{48}$th and $\frac{1}{24}$th of the whole circumference, so that the fractional pitch measured will be $\frac{1}{48}$th or $\frac{1}{24}$th of the whole pitch. Since this is always the case, if the reading-bars be graduated to scales of $\frac{1}{4}$in. = 1 foot, and $\frac{1}{2}$ inch = 1 foot, the whole pitch in feet and inches can at once be read off.

For example, suppose the readings at A and B are actually 6 inches, and

$12\frac{1}{4}$ inches, the difference is $6\frac{1}{4}$ inches; and as the ratio here is $\frac{1}{48}$, the whole pitch = 25 feet. With graduated scale of $\frac{1}{4}$ inch = 1 foot, the readings would have been 24 feet, and 49 feet, or pitch = 25 feet.

This instrument can be used for screws of uniform pitch, or pitch expanding from the boss to the periphery, but not for those where it expands from the entering to the leaving edge of the blades.

A Screw-Steering Propeller or Rudder-Screw is shown in Fig. 316. It consists of a small screw-propeller placed exterior to and aft the rudder. The shaft of the rudder-screw is carried in a bearing fixed on the rudder; it is cased with gun-metal and runs in lignum-vitæ bushes. The thrust of the screw is taken, through gun-metal collars and lignum-vitæ bearings, by the rudder. The rudder-screw makes revolution for revolution with the screw-propeller of the ship, the shaft of each propeller being coupled by a universal-joint, which enables the axis of the rudder-screw to be placed, by a force acting on the rudder, at any angle less than 55 degrees with the axis of the screw-propeller of the ship. A rudder-screw enables a ship to be turned quickly and to make a complete circle in about one-half the diameter required with a rudder alone. A rudder acts by resistance, and only while the ship is in motion: a rudder-screw is a motive power in turning the hull the instant the shaft revolves, and before the ship attains the speed necessary to bring the rudder into play, and it gives great manœuvring power to a ship.

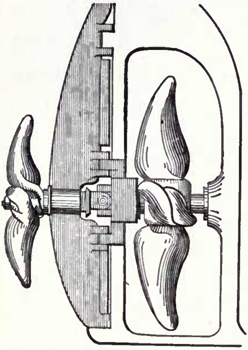


Fig. 316.—Kunstadter's screw-steering propeller, or rudder-screw.

Radial Paddle-Wheels have floats fixed rigidly to their arms. The resistance offered by radial floats on entering and leaving the water is considerable: the whole of their faces is presented to it, and they lift a quantity of water; and as each float can only give a direct sternward motion to the water when in a vertical position, their propulsive efficiency is very small. If the water did not yield to the pressure of the floats, the distance the ship would be propelled at each revolution of the paddle-wheel would be equal to the diameter of the circle of the centre of pressure of the floats $\times 3{\cdot}1416$: the point of maximum pressure or centre of pressure is at one-third the depth of the float from the outer edge. The slip of radial paddle-wheels is from 20 to 25 per cent., and they are not nearly so efficient as feathering paddle-wheels.

Feathering Paddle-Wheels have the floats arranged to move so that they may enter and leave the water in a nearly vertical position. The floats are hung, and turn on pins fixed on the side of the wheel, to which arms are attached, connected by radius-rods to an eccentric-strap, which works loose on a feathering-stud fixed in a position eccentric to the wheel, on the outer sponson on the side of the ship. All the radius-rods are jointed by pins to the boss except one, the driving eccentric-rod, which is bolted

Fig. 317.

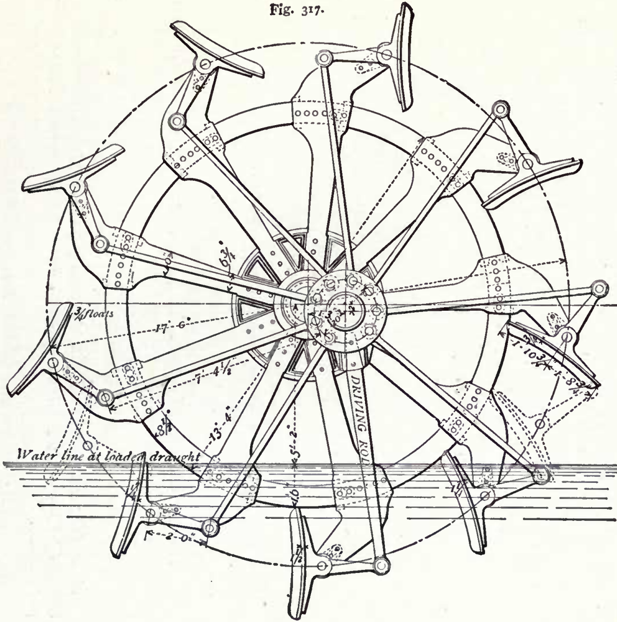
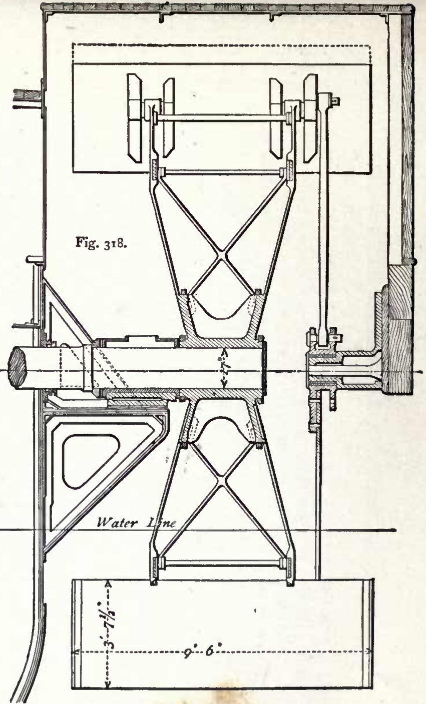


Fig. 318.



Figs. 317 and 318.—Feathering paddle-wheel designed by Mr. Stroudley rigidly to the eccentric-strap and drives the strap round the feathering-stud. By this arrangement the floats are so governed by the eccentric that their faces, while immersed, are nearly at right angles to the surface of the water, and very little power is lost, as the force applied to the float is nearly all expended in direct fore- and aft-thrust on the water.

The centre of the eccentric-strap and the feathering-stud is placed forward, or in advance of the paddle-shaft, and a little below a horizontal line with the centre of the paddle-shaft, as shown in Fig. 319, in which the feathering-stud is two inches below the line of the paddle-shaft.

A Feathering Paddle-wheel, 17 feet diameter, is shown in elevation in Fig. 317, and in cross-section in Fig. 318. The arms carrying the floats are forged in dies in one solid piece of iron without weld, and are machined all over the bearing-surfaces; they are also planed parallel where they fit into the boss, the grooves of the boss being slightly tapered and the edges of the arms correspondingly bevelled, so that as each arm is drawn into its place by the bolts it is fixed very securely. The bolts are relieved from strain by joggles forged on the arms, which clip the ring. The arms are $1\frac{3}{4}$ inch thick, and the ring is $8\frac{1}{2}$ inches wide and $1\frac{1}{4}$ inch thick. The floats, which are 9 feet 6 inches long and 3 feet $7\frac{1}{2}$ inches wide, are curved to the same radius as the wheel, and are provided at each end with an angle-iron to prevent the escape of the water laterally. The upper edge of each float, in its lowest position, is immersed 18 inches with the steamer at load-draught, and there is a space of 12 inches between the inner edges of the floats and the sides of the vessel. There are 9 floats to each wheel, six of which are $\frac{3}{4}$ inch thick; while of the other three, the centre one is $1\frac{1}{2}$ inch thick, and the two others $1\frac{1}{4}$ inch thick, the thick floats being used to balance the engines. The feathering-stud is fixed two inches below the line of the paddle-shaft, and 15 inches in advance. The joints of the feathering-gear are all bushed with lignum-vitæ. The curvature given to the floats enables them to enter at a better angle and tends to prevent the back-action on leaving the water, reducing the evil effects due to the fact that every part of the wheels below the centre-line of the shaft has a different forward velocity. These paddle-wheels are fitted to several steamers noted for their high speed, they do not lift the water at the back when running at full speed, and if the door at the side of the paddle-box is opened it is found that the box only contains a slight mist.

The Diameter of a Feathering Paddle-wheel, as shown in Fig. 162, may be found by the following formula:—

Let S = the speed of the ship in knots per hour.
R = the number of revolutions per minute.
D = the diameter in feet of the paddle-wheel at the centres of the axes of the floats.

$$\text{Then } D = \frac{S \times 32}{R}.$$

Example: Required the diameter of a feathering paddle-wheel for a steamer to make $17\frac{1}{2}$ knots an hour, with engines making 33 revolutions per minute.

$$\text{Then } \frac{17{\cdot}5 \times 32}{33} = 17 \text{ feet diameter.}$$

The Number of Floats may be found by dividing the circumference of the wheel in feet by 6. Hence the above wheel will require $\frac{17 \text{ feet} \times 3{\cdot}1416}{6}$ $=8{\cdot}9$, or say 9 floats.

The Length of each Float is usually equal to about $\frac{1}{3}$ the breadth of the steamer; if the latter be 28 feet 6 inches the length of float for the above wheel will be $28{\cdot}5 \div 3 = 9$ feet 6 inches. The width of float may be equal to the length of float $\times {\cdot}382$, and the float for the above wheel will be $= 9{\cdot}5 \times {\cdot}382 = 3{\cdot}629$ feet, or say 3 feet $7\frac{1}{2}$ inches wide.

The Number of Revolutions of a Paddle-wheel required per minute for a given speed of a steamer, may be found by this *Rule*:—

$$\text{Revolutions per minute} = \frac{\text{Knots per hour} \times 6080 \text{ feet}}{\text{Circumference of wheel in feet} \times 60 \text{ minutes}}.$$

Example: Required the number of revolutions per minute of the engines of a steamer, with paddle-wheels 17 feet diameter, to make $17\frac{1}{2}$ knots per hour.

$$\text{Then } \frac{17{\cdot}5 \times 6080}{17 \times 3{\cdot}1416 \times 60} = 33 \text{ revolutions per minute.}$$

The Speed of a Paddle-steamer in knots per hour with a given diameter of paddle-wheel and a given number of revolutions per minute, may be found by this *Rule*:—

$$\left.\begin{array}{l}\text{Speed in knots}\\ \text{per hour}\end{array}\right\} = \frac{\text{Circumference in feet of paddle-wheel} \times \text{revolutions} \times 60}{6080 \text{ feet in a knot}}.$$

Example: Required the speed in knots per hour of a steamer with paddle-wheels 17 feet diameter making 33 revolutions per minute.

$$\text{Then } \frac{17 \times 3{\cdot}1416 \times 33 \times 60}{6080 \text{ feet in a knot}} = 17{\cdot}5 \text{ knots per hour.}$$

The Speed of a Paddle-Steamer in knots per hour may be found approximately, by multiplying the circumference of the paddle-wheel in feet by the number of revolutions per minute and dividing the product by 100.

Example: Required the speed of the paddle-steamer given in the previous example?

$$\text{Then } \frac{17 \times 3{\cdot}1416 \times 33}{100} = 17{\cdot}63 \text{ knots per hour.}$$

Jet-Propellers: The propelling power in hydraulic propulsion is derived from the reaction of jets of water. The water is drawn through orifices at the bottom of the vessel by a centrifugal pump, and is discharged through orifices either at the sides or stern of the vessel. The quantity of water projected astern should be as large as possible and its velocity should be as small as possible, in order to obtain the utmost efficiency.

In a jet-propeller considerable power is lost in imparting to the water which enters the pump, a velocity equal to that of the ship: and also from friction.

A trial of hydraulic propulsion was made with the French boat "Nautilus," the arrangement of the machinery of which is shown in Figs. 319 and 320: the following were the leading dimensions:—

The length of boat=14 metres, width of beam 1m. 80, immersed section ·78 sq. m., mean draft ·54m. The propeller was a Maginott's pump, P, 60 centimetres at its largest diameter, driven by an inverted cylinder engine with a single cylinder 20cm. diameter and 15cm. stroke. The suction takes place through the pipe A, and the delivery at the stern through a conical

Fig. 319.

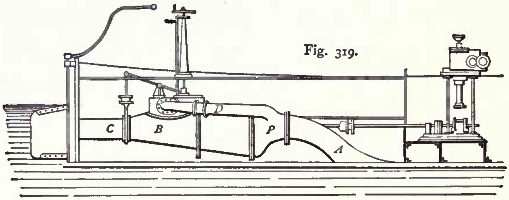
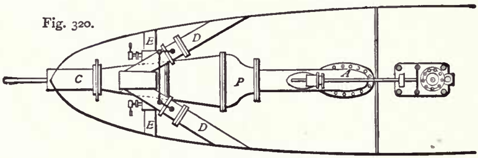


Fig. 320.



Figs. 319 and 320.—Jet-propeller.

copper pipe, B, ending in the mouthpiece, C, which has a bore of 241 millimetres. The pipes D are auxiliary pipes for going astern, and two other pipes E are for turning the vessel on the spot; valves being used for changing the flow of the water. The complete turn by 180 degrees was effected in 45 seconds, and a complete stoppage in a distance of about 14 metres when the boat was going at the rate of 14 kilometres per hour. The results of the trial are as follows:—

Horse-power developed by the engine	3·09	10·15	11·18	12·27
Speed of jet, metres per second	4·40	6·55	7·80	9·55
Speed of boat	2·46	3·30	3·83	4·22
Ratio	1·8	·2	2·04	2·28
Efficiency of pump	·60	·60	·66	·80
Efficiency of jet	·715	·66	·66	·613
Total efficiency	·425	·40	·435	·49

The total efficiency reached as high as 49 per cent., or nearly the same as that usually assumed for screw-propulsion. The efficiency of the pump increased considerably with an increased speed, but that of the jet decreased greatly at the same time.

SECTION V.

HORSE-POWER; EXPANSION OF STEAM IN A CYLINDER; PROPORTIONS OF THE CYLINDERS OF STEAM-ENGINES; PISTONS, PISTON-RODS, AND CROSS-HEADS; MILL-ENGINES; DOUBLE-EXPANSION, TRIPLE-EXPANSION, AND QUADRUPLE-EXPANSION MARINE-ENGINES, ETC.

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POWER OF STEAM AND HORSE-POWER OF STEAM-ENGINES.

Heat is the Source of the Power of Steam, and the elastic force of steam is due to its inherent heat-energy. Steam in performing work in the cylinder of an engine gives up heat, and the difference in the economy of fuel between any two engines is measured by the quantity of heat turned into work. If loss from radiation and condensation be not taken into account, the heat which leaves the cylinder in the exhaust-steam is less than that which entered it, in the proportion of one unit for every 772 units of work done by the engine. The quantity of heat theoretically capable of being transferred to motion, or the maximum amount of work that can be got in a heat-engine out of any gas or vapour, such as steam, is expressed by the following formula :—

Let M = the mechanical value of heat.
T = the highest absolute temperature of the gas or steam.
t = the lowest absolute temperature of the gas or steam.

$$M = 772 \frac{T - t}{T}$$

The zero of Fahrenheit's thermometer being 461° above the point of absolute cold or zero, t will for condensing engines = 100° Fahr., the temperature of the hot-well, + 461° = 561° Fahr., and for non-condensing engines t will = 212° Fahr., the temperature of steam of the pressure of the atmosphere, + 461° = 673° Fahr. Taking the maximum temperature of the combustion of coal at 3000° Fahr., if the whole of the available heat were absorbed by the furnace-plates and transferred to the water in a boiler, it would give $772 \times \frac{3000 - 561}{3000} = 772 \times {\cdot}801 = 618{\cdot}37$ units of work, or foot-pounds, the value or mechanical equivalent of the heat; but there are no means available for using such a high temperature in a steam-engine. The highest satisfactory working temperature is probably 400° Fahr., the mechanical value of which is theoretically $= 772 \frac{(400 + 461) - 561}{(400 + 461)} = 772 \times \cdot 348 = 268\cdot 65$ foot-pounds.

By comparing the theoretical value or efficiency of the heat with the efficiency actually obtained with engines in practice, the mechanical equivalent of the heat used may be accurately estimated. For instance, if an engine use say 20 lbs. of steam per indicated horse-power per hour at the highest feasible temperature, or 400° Fahr., the total heat of steam of this temperature is $(400° \times \cdot 305) + 1082 = 1204$ units per lb. of steam, $= 1204 \times 20$ lbs. $= 24080$ units in 20 lbs. of steam, and $24080 \times 268\cdot 65$ foot-pounds, the theoretical value of the heat, $= 6469092$, the units of work theoretically available in 20 lbs. of steam.

Then as one indicated horse-power is the expression of 33000 foot-pounds of work done per minute, $33000 \times 60 = 1980000$ units of work per hour, and $\frac{1980000 \text{ units of work per hour}}{6469092 \text{ units of work available}} = \cdot 306$, the efficiency of the engine, or less than one-third the theoretical efficiency.

One Horse-power is equivalent to $\frac{33000 \text{ foot-lbs.}}{772 \text{ foot-lbs.}} = 42\cdot 746$ thermal units per minute, which multiplied by the number of indicated horse-power of an engine will give the thermal equivalent of the work done by that engine.

Example: Required the number of thermal units in a marine-engine with cylinder 60 inches diameter, length of stroke 3 feet, number of revolutions per minute 70, mean pressure of steam 34 lbs. per square inch: and what would be the coal-consumption per hour and per day per indicated horse-power.

Then $\frac{60 \times 60 \text{ inches} \times \cdot 7854 \times 34 \text{ lbs.} \times (3 \text{ feet} \times 2 \times 70)}{772 \text{ foot-lbs.}} = 52300\cdot 3$ thermal units per minute, or thermal equivalent of the work done by that engine;

or $\frac{60 \times 60 \times \cdot 7854 \times 34 \text{ lbs.} \times (3 \text{ feet} \times 2 \times 70)}{33000 \text{ lbs.}} = 1223\cdot 514$ indicated horse-power $\times 42\cdot 746 = 52300\cdot 3$ thermal units per minute, the thermal equivalent of the work done by that engine.

If 1 lb. of good coal give out at the piston, say 1200 thermal units, the consumption will be $= \frac{52300\cdot 3}{1200} = 43\cdot 58$ lbs. of coal burnt per minute, or $43\cdot 58 \times 60 = 2614\cdot 8$ lbs. per hour, and $\frac{2614\cdot 8}{1223\cdot 514} = 2\cdot 13$ lbs. of coal per indicated horse-power per hour, or $2\cdot 13 \times 24 = 51\cdot 12$ lbs. of coal per indicated horse-power per day of 24 hours.

Nominal Horse-power is a commercial term for denoting the size of an engine without regard to the actual power it will develop. Marine-engines work up to from 4 to 8 times their nominal horse-power; the average actual or indicated horse-power in merchant-steamers being probably 4 times the nominal.

The Nominal Horse-power of ordinary Condensing Steam-engines of the low-pressure type, in which the mean effective pressure of the steam in the cylinder is assumed to be 7 lbs. per square inch, and the speed of the piston is either 128 times the cube root of the length of stroke, or 220 feet per minute, may be calculated by the following *Rules* :—

(1.) Nominal horse-power =

$$\frac{\text{area of piston in square inches} \times 7 \text{ lbs.} \times (128 \times \sqrt[3]{\text{stroke in feet}})}{33000 \text{ lbs.}}$$

(2.) Nominal horse-power =

$$\frac{\text{diameter of cylinder in inches}^2 \times \sqrt[3]{\text{stroke in feet}}}{47}$$

(3.) Nominal horse-power $= \dfrac{\text{diameter of cylinder in inches}^2}{30}$; this Rule gives 30 circular inches of area of piston per nominal horse-power.

(4.) Nominal horse-power =

$$\frac{\text{area of piston in square inches} \times 7 \text{ lbs.} \times 220 \text{ feet piston speed}}{33000 \text{ lbs.}}$$

(5.) Nominal horse-power $= \dfrac{\text{diameter of cylinder}^2}{28}$

(6.) Nominal horse-power =

$$\frac{\text{diameter of cylinder}^2 \times \text{speed of piston in feet per minute}}{6000}$$

These Rules may be illustrated by applying them to the following *Examples.* Required the nominal horse-power of a low-pressure condensing steam-engine, with cylinder 30 inches diameter, and length of stroke 3 feet.

Then by Rule (1). $\dfrac{30 \times 30 \times \cdot 7854 \times 7 \times (128 \times \sqrt[3]{3} = \text{the piston speed})}{33000}$

$= 27 \cdot 68$ nominal horse-power.

Rule (2). $\dfrac{30 \times 30 \times \sqrt[3]{3}}{47} = 27 \cdot 61$ nominal horse-power.

Rule (3). $\dfrac{30 \times 30}{30} = 30$ nominal horse-power.

Rule (4). $\dfrac{30 \times 30 \times \cdot 7854 \times 7 \times 220}{33000} = 32 \cdot 98$ nominal horse-power.

Rule (5). $\dfrac{30 \times 30}{28} = 32 \cdot 14$ nominal horse-power.

Rule (6). Assuming the number of revolutions per minute of this engine to be 37, $\dfrac{30 \times 30 \times (3 \text{ ft. stroke} \times 2 \times 37)}{6000}$

$= 33 \cdot 3$ nominal horse-power.

The Speed of a Piston in feet per minute = twice the length of stroke in feet multiplied by the number of revolutions per minute.

The Admiralty Rule for finding the horse-power of a low-pressure type of steam-engine is that given above in Rule (6), and to which it still applies. The divisor 6000 is obtained as follows :—

$$\frac{33000 \text{ lbs.}}{\cdot 7854 \times 7 \text{ lbs.}} = \text{in round numbers } 6000.$$

Nominal Horse-power of modern Condensing Steam-engines.—The following Rules are suitable for the high pressures of steam and increased piston speed now used.

(7.) Nominal horse-power =

$$\frac{\text{diameter of cylinder in inches}^2 \times \sqrt[3]{\text{stroke in feet}}}{28}$$

(8.) Nominal horse-power =

$$\frac{\text{diameter of cylinder in inches}^2 \times \sqrt[3]{\text{stroke in feet}}}{24}$$

Applying these Rules to the previous examples, it gives :—

by Rule (7). $\frac{30 \times 30 \times \sqrt[3]{3}}{28} = 46{\cdot}3$ nominal horse-power.

Rule (8), $\frac{30 \times 30 \times \sqrt[3]{3}}{24} = 54$ nominal horse-power.

Nominal Horse-power of Non-condensing Steam-engines.—The following are a few of the Rules used in different districts for calculating the nominal horse-power of non-condensing steam-engines.

Nominal horse-power of small steam-engines $= \frac{\text{area of cylinder in sq. ins.}}{8}$

This Rule only applies to steam-engines having a cylinder-area of not exceeding 100 square inches, and a length of stroke = diameter of cylinder $\times$ 1·4 to 1·5.

Example: Required the nominal horse-power of steam-engines with cylinders of $6\frac{3}{8}$ inches, $7\frac{3}{4}$ inches, 9 inches, $10\frac{1}{8}$ inches, and $10\frac{3}{4}$ inches diameter.

Engine with a cylinder $6\frac{3}{8}$ inches diameter $= \frac{6{\cdot}375 \times 6{\cdot}375 \times {\cdot}7854}{8} =$ 4 nominal horse-power.

Engine with a cylinder $7\frac{3}{4}$ inches diameter $= \frac{7{\cdot}75 \times 7{\cdot}75 \times {\cdot}7854}{8} =$ 6 nominal horse-power.

Engine with a cylinder 9 inches diameter $= \frac{9 \times 9 \times {\cdot}7854}{8} = 8$ nominal horse-power.

Engine with a cylinder $10\frac{1}{8}$ inches diameter $= \frac{10{\cdot}125 \times 10{\cdot}125 \times {\cdot}7854}{8}$ = 10 nominal horse-power.

Engine with a cylinder $10\frac{3}{4}$ inches diameter $= \frac{10{\cdot}75 \times 10{\cdot}75 \times {\cdot}7854}{8} =$ 11 nominal horse-power.

For steam-engines having a cylinder-area larger than 100 square inches, the following are a few of the Rules used by makers of engines.

(1.) Nominal horse-power $= \frac{\text{diameter of cylinder in inches}^2}{10}$, this Rule gives 10 circular inches of area of piston per nominal horse-power.

(2.) Nominal horse-power $= \frac{\text{diameter of cylinder in inches}^2}{12}$, this Rule gives 12 circular inches of area of piston per nominal horse-power.

(3.) Nominal horse-power =

$$\frac{\text{area of cylinder in square inches} \times \sqrt[3]{\text{stroke in feet}}}{12}$$

(4.) Nominal horse-power =

$$\frac{\text{area of cylinder in square inches} \times \sqrt[3]{\text{stroke in feet}}}{14}$$

(5.) Nominal horse-power =

$$\frac{\text{area of cylinder in square inches} \times \sqrt[3]{\text{stroke in inches}}}{27}$$

(6.) Nominal horse-power =

$$\frac{\text{area of cylinder in square inches} \times \sqrt[3]{\text{stroke in inches}}}{30}$$

(7.) Nominal horse-power =

$$\frac{\text{area of cylinder in square inches} \times \sqrt[3]{\text{stroke in inches}}}{40}$$

(8.) Nominal horse-power =

$$\frac{\text{area of cylinder in square inches} \times \sqrt[3]{\text{stroke in inches}}}{46}$$

(9.) Nominal horse-power =

$$\frac{\text{area of cylinder in square inches} \times \sqrt[3]{\text{stroke in feet}}}{17{\cdot}5}$$

(10.) Nominal horse-power =

$$\frac{\text{area of cylinder in square inches} \times \sqrt[3]{\text{stroke in feet}}}{20}$$

The Nominal Horse-power of Modern Non-condensing Engines may be found by the first rule given above when the engines are small, with cylinder-area not exceeding 100 square inches, and for large engines by Rule (3). These two Rules represent probably the average modern practice of engine-makers for the high pressures of steam and high speed of piston now used.

These Rules may be illustrated by applying them to the following:—

Example: Required the nominal horse-power of a steam-engine with a cylinder 12 inches diameter, length of stroke 2 feet.

Then by Rule (1) $\frac{12 \times 12}{10} = 14 \cdot 4$ nominal horse-power.

Rule (2) $\frac{12 \times 12}{12} = 12$ nominal horse-power.

Rule (3) $\frac{12 \times 12 \times \cdot 7854 \times \sqrt[3]{2 \text{ feet}}}{12} = 11 \cdot 8$ nominal horse-power.

Rule (4) $\frac{12 \times 12 \times \cdot 7854 \times \sqrt[3]{2 \text{ feet}}}{14} = 10 \cdot 1$ nominal horse-power.

Rule (5) $\frac{12 \times 12 \times \cdot 7854 \times \sqrt[3]{24 \text{ inches}}}{27} = 12$ nominal horse-power.

Rule (6) $\frac{12 \times 12 \times \cdot 7854 \times \sqrt[3]{24 \text{ inches}}}{30} = 10 \cdot 8$ nominal horse-power.

Rule (7) $\frac{12 \times 12 \times \cdot 7854 \times \sqrt[3]{24 \text{ inches}}}{40} = 8 \cdot 1$ nominal horse-power.

Rule (8) $\frac{12 \times 12 \times \cdot 7854 \times \sqrt[3]{24 \text{ inches}}}{46} = 7 \cdot 1$ nominal horse-power.

Rule (9) $\frac{12 \times 12 \times \cdot 7854 \times \sqrt[3]{2 \text{ feet}}}{17 \cdot 5} = 8 \cdot 1$ nominal horse-power.

Rule (10) $\frac{12 \times 12 \times \cdot 7854 \times \sqrt[3]{2 \text{ feet}}}{20} = 7 \cdot 1$ nominal horse-power.

The Effective Mean Pressure of Steam Required for a given Nominal Horse-power may be found by this *Rule*:—

Effective mean pressure in lbs. per square inch =

$$\frac{\text{Number of foot-pounds}}{\text{Cylinder area in square inches per nominal horse-power} \times \text{speed of piston.}}$$

Example:—If 30 circular inches be considered equal to one nominal horse-power, what is the effective mean pressure of the steam required to indicate four times the nominal horse-power with a piston speed of 420 feet per minute.

Then $30 \times \cdot 7854 = 23 \cdot 562$ square inches of cylinder area per nominal horse-power and $\frac{33000 \times 4}{23 \cdot 562 \times 420 \text{ feet}} = 13.338$ lbs. the effective mean pressure per square inch.

The Number of times the Indicated Horse-power of an Engine is larger than the nominal horse-power may be found by the converse of the above *Rule*.

Example: The speed of the piston of an engine is 420 feet per minute, with an effective mean pressure of steam of 13.338 lbs. per square inch: 30 circular inches have to be given per nominal horse-power. How many times will the nominal horse-power be greater than the indicated horse-power.

Then 30 circular inches × ·7854 = 23·562 square inches per nominal horse-power, and

$$\frac{23{\cdot}562 \text{ square inches} \times 13{\cdot}338 \text{ lbs. mean pressure} \times 420 \text{ feet}}{33000 \text{ lbs.}} = 4,$$

the number of times the indicated horse-power is greater than the nominal horse-power.

The Speed of an Engine required to develop a given Number of horse-power may be found by this *Rule*:—

Number of revolutions of an engine per minute =

$$\frac{\text{Number of foot-pounds}}{\text{Area of cylinder in inches} \times \text{pressure in lbs.} \times \text{twice length of stroke in feet.}}$$

Example 1: An engine was specified to have 30 circular inches of cylinder per nominal horse-power, and to indicate five times the nominal horse-power. Length of stroke 39 inches. Effective mean pressure of the steam 30 lbs. per square inch. How many revolutions per minute must it make to fulfil the contract.

Then 30 circular inches × ·7854 = 23·562 square inches, area of cylinder.

$$\frac{33000 \times 5}{23{\cdot}562 \text{ square inches} \times 30 \text{ lbs. pressure} \times 3{\cdot}25 \text{ feet} \times 2} = 35{\cdot}91 \text{ revolutions per minute.}$$

Example 2. An engine with a cylinder 41 inches diameter and 3 feet 6 inches length of stroke, is required to develop one thousand horse-power, with steam of 50 lbs. per square inch effective mean pressure, how many revolutions must it make per minute?

Then $\dfrac{33000 \times 1000}{41 \times 41 \text{ inches} \times {\cdot}7854 \times 50 \text{ lbs.} \times 3{\cdot}5 \text{ feet} \times 2} = 71{\cdot}41$ revolutions per minute.

Nominal Horse-power of Compound or Double-expansion Marine Steam-engines. A number of rules are used for calculating the nominal horse-power of marine-engines, the results obtained from which vary considerably, and are not proportional to the actual or indicated horse-power.

The following are a few of the rules used by makers of engines to determine the nominal horse-power of compound, or double-expansion, condensing marine-engines: where HP=high pressure cylinder, and LP= low pressure cylinder.

(1) Nominal horse-power =

$$\frac{(\text{Diameter of HP cylinder})^2 + (\text{diameter of LP cylinder})^2}{25 \text{ circular inches}}$$

(2) Nominal horse-power =

$$\frac{(\text{Diameter of HP cylinder})^2 + (\text{diameter of LP cylinder})^2 \times \sqrt[3]{\text{stroke in feet}}}{38}$$

(3) Nominal horse-power =

$$\frac{(\text{Diameter of HP cylinder})^2 + (\text{diameter of LP cylinder})^2 \times \sqrt[3]{\text{stroke in inches}}}{87}$$

(4) Nominal horse-power $= \dfrac{(\text{Diameter of HP cylinder})^2 + (\text{diameter of LP cylinder})^2}{30 \text{ circular inches}}$. This rule gives 30 *circular inches* of area of piston per nominal horse-power.

These rules may be illustrated by applying them to the following example :—

Example :—Required the nominal horse-power of a compound condensing-engine with a high-pressure cylinder 30 inches diameter, and a low-pressure cylinder 60 inches diameter. Length of stroke 3 feet.

Then by Rule (1) $\dfrac{(30 \times 30) + (60 \times 60)}{25 \text{ circular inches}} = 180$ nominal horse-power.

Rule (2) $\dfrac{(30 \times 30) + (60 \times 60) \times \sqrt[3]{3}}{38} = 171$ nominal horse-power.

Rule (3) $\dfrac{(30 \times 30) + (60 \times 60) \times \sqrt{36}}{87} = 171$ nominal horse-power.

Rule (4) $\dfrac{(30 \times 30) + (60 \times 60)}{30 \text{ circular inches}} = 150$ nominal horse-power.

The Number of Circular Inches of Cylinder-Area per Nominal Horse-power of compound, or double-expansion condensing-engines, may be found by dividing the sum of the squares of the diameters of the cylinders by the nominal horse-power

Example: The cylinders of a compound engine are 41 inches diameter, and 70 inches diameter; the nominal horse-power is 250. How many circular inches is that per nominal horse-power, and what is the ratio of the cylinders?

Then $\dfrac{(41 \times 41) + (70 \times 70)}{250 \text{ nominal horse-power}} = 26{\cdot}32$ circular inches of cylinder-area per nominal horse-power.

The proportion of circles to one another is as the squares of their diameters. And $41^2 : 70^2 :: 1 : 3{\cdot}915$ the ratio of the cylinders.

The cylinder-area of triple-expansion engines may be found as follows:—

Example: The cylinders of a set of triple-expansion engines are 21 inches, 35 inches, and 57 inches diameter; the nominal horse-power is 224: How many circular inches of cylinder-area is that per nominal horse-power, and what is the ratio of the cylinders?

Then $\dfrac{(21 \times 21) + (35 \times 35) + (57 \times 57)}{224} = 21{\cdot}94$ circular inches of cylinder-area per nominal horse-power.

And $21^2 : 57^2 :: 1 : 6{\cdot}859$ the ratio of the high-pressure cylinder to the low-pressure cylinder.

$35^2 : 57^2 :: 1 : 2{\cdot}469$ the ratio of the intermediate cylinder to the low-pressure cylinder.

Mr. Parker and Mr. Milton, engineer-surveyors of Lloyd's, in their report on nominal horse-power for the information of the Committee of Lloyd's, in order to show the great diversity of practice, prepared the following table, which contains a number of *Rules* used for calculating the nominal horse-power of double-expansion marine-engines, and the nominal powers of three double-expansion engines of different sizes calculated by them.

Table 89.—CONTAINING 18 DIFFERENT RULES USED IN PRACTICE FOR OBTAINING THE NOMINAL HORSE-POWER OF DOUBLE-EXPANSION MARINE STEAM-ENGINES, AND GIVING THE NOMINAL POWERS OF 3 DIFFERENT SIZES OF DOUBLE-EXPANSION ENGINES CALCULATED BY THEM.

No.	Formulæ for Nominal Horse-power of Compound, or Double-Expansion, Engines.	Engines of 41 & 70 Cylinder 42 in. Stroke. A	Engines of 33 & 62 Cylinder 39 in. Stroke. B	Engines of 33 & 62 Cylinder 45 in. Stroke. C	Remarks.
1	$(D^2 + d^2) \div 33 =$ . . .	199	149	149	
2	$(D^2 + d^2) \div 32 =$ . . .	205	154	154	
3	$(D^2 + d^2) \div 30 =$ . .	219	164	164	Rule of makers of Engine B.
4	$(D^2 + d^2) \div 28 =$ . . .	235	176	176	
5	$(D^2 + d^2) \div 27 =$ . .	243	182	182	
6	$(D^2 + d^2) \div 26 =$ . . .	253	190	190	
7	$\frac{(D^2 + d^2) \times \sqrt[3]{S,\text{ inches}}}{100} =$.	229	167	175	
8	$\frac{(D^2 + d^2) \times \sqrt[3]{S,\text{ inches}}}{90} =$.	254	185	195	
9	$((D^2 + d^2) \times \sqrt[3]{S,\text{ feet}}) \div 40$	250	180	191	
10	$(D^2 \times \sqrt[3]{S,\text{ feet}}) \div 33 =$.	222	172	181	
11	840 cubic inches of cylinder = .	258	180	207	
12	800 to 700 cubic inches of cylinder = . . .	271 to 310	189 to 216	218 to 249	
13	Area of H.P. cylinder ÷ 5 = .	262	171	171	
14	$((D^2 + d^2) \times S,\text{ feet}) \div 90 =$	256	178	204	
15	$\frac{(D^2 + d^2)}{2000} \times (S + 30) =$.	236	170	185	Rule of makers of Engine C.
16	$\frac{(D^2 + d^2)}{2016} \times (S + 30) =$.	235	168	183	
17	$\frac{(D^2 + d^2) \times \text{speed of piston}}{6000} =$	462	288	328	Rule given in proposed Merchant Shipping Act, 1871.
18	Actual registered horse-power =	256	160	200	

D = Diameter of Low-pressure Cylinder, and d = Diameter of the High-pressure Cylinder.
S = Length of Stroke. H.-P. = High-pressure Cylinder.

The Nominal Horse-Power of Triple-Expansion Engines may be found by the following formula :—

Let D = the diameter of each cylinder in inches ; S = length of stroke in inches ; N = nominal horse-power of the engines.

$$\text{Then } N = \frac{D^2 + D^2 + D^2 \times \sqrt[3]{S}}{90}$$

Example: Required the nominal horse-power of a set of triple-expansion engines with cylinders 33, 54, and 86 inches diameter. Length of stroke 60 inches.

$$\text{Then } \frac{33^2 + 54^2 + 86^2 \times \sqrt[3]{60}}{90} = 496 \text{ nominal horse-power.}$$

The Nominal Horse-Power of Triple-Expansion Surface-Condensing Marine-Engines may be found by the following formula, which assumes a boiler-pressure of 150 lbs. per square inch above the atmosphere:—

Let H P = the diameter of the high-pressure cylinder in inches.
M P = the diameter of the intermediate cylinder in inches.
L P = the diameter of the low-pressure cylinder in inches.

$$\text{Then nominal horse-power} = \frac{\text{H P}^2 + \text{M P}^2 + \text{L P}^2}{5{\cdot}5 \times x}.$$

Where x = the number of times the indicated horse-power is required to be greater than the nominal horse-power.

Example: In a set of triple-expansion engines the high-pressure cylinder is 21 inches diameter, the intermediate cylinder is 35 inches diameter, and the low-pressure cylinder is 55 inches diameter; the engines were guaranteed to work up to, and indicate, four times their nominal horse-power: What is the nominal horse-power of the engines, and what power should they indicate?

Then $x=4$

And $\dfrac{(21\times 21)+(35\times 35)+(55\times 55)}{5{\cdot}5\times 4}=213{\cdot}22$ nominal horse-power, and these engines should work up to, and indicate $213{\cdot}22\times 4=852{\cdot}88$, or say 853 horse-power.

If the nominal horse-power of the above engines were calculated at 30 circular inches of area of piston per nominal horse-power, their power would only be $= \dfrac{(21\times 21)+(35\times 35)+(55\times 55)}{30 \text{ circular inches}} = 156{\cdot}36$ nominal horse-power, and they would require to work up to $\dfrac{853}{156{\cdot}36} = 5{\cdot}45$ times the nominal horse-power, to develop 853 indicated horse-power.

The Nominal Horse-Power of Quadruple-Expansion Surface-Condensing Marine-Engines may be found by the following formula, which assumes a boiler-pressure of 170 lbs. per square inch above the atmosphere.

Let L P = the area of the low-pressure cylinder in square inches.
H P = the area of the high-pressure cylinder in square inches.
A = the area of the *first* intermediate cylinder in square inches.
B = the area of the *second* intermediate cylinder in square inches.

$$\text{Then, nominal horse-power} = \frac{\text{L P}^2 + \text{H P}^2 + \text{A}^2 + \text{B}^2}{4{\cdot}5 \times x}$$

Where x = the number of times the indicated horse-power is required to be greater than the nominal horse-power.

Example: In a set of quadruple expansion-engines the high-pressure cylinder is $16\frac{1}{2}$ inches diameter, the first intermediate cylinder is $23\frac{1}{2}$ inches diameter, the second intermediate cylinder is 33 inches diameter, and the low-pressure cylinder is 45 inches diameter; the engines were guaranteed to work up to and indicate four times their nominal horse-power: What is the nominal horse-power of the engines?

Then $x = 4$.

And $\frac{(16\cdot5 \times 16\cdot5)+(23\cdot5 \times 23\cdot5)+(33 \times 33)+(45 \times 45)}{4\cdot5 \times 4} = 218\cdot8$ nominal horse-power, and these engines should work up to and indicate $213\cdot8 \times 4 = 875\cdot2$, or say 876 horse-power.

If the nominal horse-power of the above engines were calculated at 30 circular inches of area of piston per nominal horse-power, their power would only be $= \frac{(16\cdot5 \times 16\cdot5)+(23\cdot5 \times 23\cdot5)+(33 \times 33)+(45 \times 45)}{30 \text{ circular inches}} = 131\cdot28$ nominal horse-power, and they would require to work up to $\frac{876}{131\cdot28} = 6\cdot67$ times the nominal horse-power to develop 876 indicated horse-power.

The Indicated Horse-Power of an engine is the effective power calculated from an indicator diagram, or the power imparted by the steam in the cylinder to the piston.

Actual or Indicated Horse-Power.—One actual horse-power is equal to 33000 lbs. raised one foot high in one minute, or 550 lbs. raised one foot high in a second.

The power imparted to the piston by the steam in a cylinder, or the actual horse-power, is the quotient of the effective work done in the cylinder in foot-pounds per minute divided by 33000. The effective work is found by multiplying the area of the piston in square inches by the effective mean-pressure in lbs. per square inch, by twice the length of stroke and by the number of revolutions per minute of the crank-shaft. The following formulæ are used for calculating the actual or indicated horse-power, the diameter of cylinder, the effective mean-pressure, and speed of steam-engines; the working friction of the engine not being taken into account.

Let = IHP = the indicated horse-power of the engine.
D = the diameter of the cylinder in inches.
P = the effective mean pressure of the steam in lbs. per square inch.
L = the length of stroke in feet.
R = the number of revolutions of the crank-shaft per minute.

Then, area of piston $= D^2 \times \cdot7854$.

Pressure on the piston $= D^2 \times \cdot7854 \times P$.

Number of units of work in one revolution $= D^2 \times \cdot7854 \times P \times 2L$.

$\left.\begin{array}{l}\text{Number of lbs. lifted one foot high per minute} \\ \text{Or number of units of work done per minute}\end{array}\right\} D^2 \times \cdot7854 \times P \times 2L \times R.$

(1.) Indicated horse-power, $I\,H\,P = \frac{\text{Units of work done per minute}}{33000}$.

(2.) The speed of the piston in feet per minute $= L \times 2 \times R$.

(3.) $\left.\begin{array}{c}\text{Indicated horse-} \\ \text{power, } I\,H\,P\end{array}\right\} = \frac{D^2 \times \cdot7854 \times P \times \text{speed of piston in ft. per minute}}{33000}$

(4.) Indicated horse-power, $I\,H\,P = \dfrac{D^2 \times \cdot 7854 \times P \times (L \times 2 \times R)}{33000}$.

(5.) Diameter of cylinder in inches, $D = \sqrt[2]{\dfrac{33000\,I\,H\,P}{\cdot 7854 \times P \times (L \times 2 \times R)}}$.

(6.) Effective mean-pressure, $P = \dfrac{33000 \times I\,H\,P}{D^2 \times \cdot 7854 \times (L \times 2 \times R)}$.

(7.) Length of stroke in feet, $L = \dfrac{33000\,I\,H\,P}{D^2 \times \cdot 7854 \times P \times (2\,R)}$.

(8.) Number of revolutions per minute, $R = \dfrac{33000\,I\,H\,P}{D^2 \times \cdot 7854 \times P \times (2\,L)}$.

These *Rules* may be illustrated by applying them to the following example :—

Example: Required the actual or indicated horse-power of a non-condensing steam-engine, with cylinder 16 inches diameter, length of stroke 2 feet, number of revolutions per minute 80, mean pressure on the piston 50 lbs. per square inch.

Then by Rule (1.) $IHP = \dfrac{\text{units of work done per minute}}{33000}$

$\therefore \dfrac{16 \times 16 \times \cdot 7854 \times 50 \times 2 \times 2 \times 80}{33000} =$ 97·454 indicated horse-power.

Rule (2.) 2 feet stroke × 2 × 80 revolutions per minute = 320, the speed of piston in feet per minute.

Rule (3.) $IHP = \dfrac{16 \times 16 \times \cdot 7854 \times 50 \times 320}{33000} = 97\cdot 454$ indicated horse-power.

Rule (4.) $IHP = \dfrac{16 \times 16 \times \cdot 7854 \times 50 \times 2 \times 2 \times 80}{33000} =$ 97·454 indicated horse-power.

Rule (5.) $D = \dfrac{33000 \times 97\cdot 454}{\cdot 7854 \times 50 \times 2 \times 2 \times 80} = \sqrt[2]{256} = 16$ inches diameter of cylinder.

Rule (6.) $P = \dfrac{33000 \times 97\cdot 454}{16 \times 16 \times \cdot 7854 \times 2 \times 2 \times 80} = 50$ lbs. per square inch mean pressure of steam on the piston.

Rule (7.) $L = \dfrac{33000 \times 97\cdot 454}{16 \times 16 \times \cdot 7854 \times 50 \times 2 \times 80} = 2$ feet, length of stroke.

Rule (8.) $R = \dfrac{33000 \times 97\cdot 454}{16 \times 16 \times \cdot 7854 \times 50 \times 2 \times 2} = 80$ revolutions per minute.

Friction of Steam-Engines.—The power absorbed by friction depends upon the load on the journals and guides; the alignment and finish of the rolling and sliding surfaces, and the efficiency of their lubrication. It averages from 10 to 20 per cent. of the total power developed by an engine when fully loaded. No allowance is made for friction in the above Rules for finding the actual or indicated horse-power of steam-engines. It will be necessary to deduct from the horse-power obtained by these Rules 12 per cent. for a non-condensing engine, and 25 per cent. for a condensing engine, to allow for the power absorbed in driving the engine when fully loaded, against the resistance due to its own friction, and in working the pumps, the remainder will be the actual or available motive power, or the effective horse-power of the engine. For the friction of an engine without pumps, a deduction may be made from the effective mean pressure of $1\frac{1}{2}$ lb. per square inch for large engines, 2 lbs. for medium sized, and 3 lbs. for small engines.

EXPANSION OF STEAM PROPORTIONS OF THE CYLINDERS, PISTONS, PISTON-RODS, AND CROSS-HEADS, OF STEAM-ENGINES.

Expansion of Steam in a Cylinder.—When steam is cut-off in a cylinder after the piston has travelled a portion of its stroke, the steam expands for the remainder of the stroke, the piston being driven by the expansive force of the steam, and its pressure at any part of its stroke during expansion is nearly in inverse proportion to its volume. For instance, if steam of 100 lbs. pressure per square inch be admitted to a cylinder, and cut off when the piston has travelled one-fourth the length of its stroke, the remaining three-fourths of the stroke will be performed by the expansive force of the steam, which will increase in volume to four times its initial volume and be reduced to about one-fourth its initial pressure, or to about 25 lbs. per square inch. It is therefore usual to assume the pressure of expanding steam to be inversely as its volume. The terminal pressure of the steam is equal to the product of the initial pressure by the grade of expansion. The method of calculating the pressure by these Rules is shown in the following example:—

Example: Steam of 66 lbs. pressure per square inch is admitted to the cylinder of a steam-engine having a 6 feet stroke, the steam is cut off after the piston has travelled 2 feet of its stroke. Required the terminal pressure, the average pressure throughout the stroke, and the saving effected by working the steam expansively.

The terminal pressure of the steam is:—

The grade of expansion is $\frac{2 \text{ feet admission of steam}}{6 \text{ feet length of stroke}} = \frac{1}{3}$ and $66 \times \frac{1}{3} =$ 22 lbs. per square inch, the terminal pressure of the steam;

$$\text{or, } \frac{\text{initial pressure}}{\text{terminal pressure}} = \frac{\text{whole stroke}}{\text{part of stroke}}$$

$\therefore \frac{66}{x} = \frac{6}{2} \therefore x = 22$ lbs. per square inch, the terminal pressure.

The average pressure of the steam throughout the stroke is :—

Pressure of the steam during the first foot in length of the stroke					$=66$
,,	,,	second	,,		$=66$
,,	,,	third	,,	,,	$=66\times\frac{2}{3}=44$
,,	,,	fourth	,,	,,	$=66\times\frac{2}{4}=33$
,,	,,	fifth	,,	,,	$=66\times\frac{2}{5}=26{\cdot}4$
,,	,,	sixth	,,	,,	$=66\times\frac{2}{6}=22$
					$257{\cdot}4$

the total pressure, and $257{\cdot}4 \div 6 = 42{\cdot}9$ lbs. per square inch, the average pressure of the steam throughout the stroke.

The saving effected by working expansively is :—

If the steam had been admitted throughout the whole length of stroke, the average pressure would have been 66 lbs. per square inch; by working expansively the average pressure was $42{\cdot}9$ lbs. per square inch, showing a saving of $66 - 42{\cdot}9 = 23{\cdot}1$ lbs. per square inch $= \frac{23{\cdot}1 \times 100}{66 \text{ lbs.}} = 35$ per cent. effected by working expansively.

The average pressure of the steam throughout the stroke may also be found by constructing a diagram as shown in Fig. 321, which represents the cylinder described in the previous example. AB represents the initial pressure of the steam = 66 lbs. per square inch, and BD the length of stroke, which is divided into six equal parts by the lines *e*, *f*, *g*, *h*, *i*, equal in depth to the pressure of the steam at these points of the stroke, as obtained in the previous example. For instance, the depth of the line, *g*, is $\frac{2}{3}$rds that of AB, because the steam is cut off at *f*, and when the piston arrives at the position marked *g* the pressure of the expanding steam will be reduced to $66 \times \frac{2}{3} = 44$ lbs. per square inch. The curve drawn through the points thus found is a hyperbola: it represents the falling pressure of the steam during expansion. The average pressure of the steam throughout the stroke may be calculated approximately by Simson's *Rule* :—

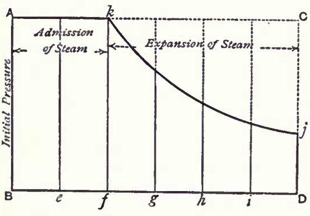


Fig. 321.—Diagram of steam-pressure.

To the sum of the extreme ordinates add four times the sum of the even ordinates, and twice the sum of the odd ordinates. This sum, multiplied by one-third the common distance between the ordinates, will give the area of that part of the diagram marked *k*, *j*, D, *f*, and therefore the steam pressure.

Then the area $= \frac{1}{3}\{66 + 22 + 4\,(44 + 26{\cdot}4) + 2 \times 33\} = 145{\cdot}2$ lbs.

The work done before the steam was cut off is $66 \times 2 = 132$ lbs., the total work done is $145{\cdot}2 + 132 = 277{\cdot}2$ lbs., and the average pressure throughout the stroke is $277{\cdot}2 \div 6 = 46{\cdot}2$ lbs. per square inch, or $46{\cdot}2$ —

$42.9 = 3.3$ lbs. per square inch more than the pressure found by the previous rule.

The Maximum Economy of Steam is obtained, theoretically, when the whole of the work is done in the cylinder by the elastic force proceeding from the expansion of steam. Therefore, to obtain all its useful work, steam should be expanded as many times as will reduce its final pressure to the lowest point consistent with doing useful work. When expansion ceases to do useful work it ceases to be of any practical value.

As the temperature changes with the pressure of steam, the reduction of pressure at the end of the stroke is accompanied by loss of heat and cooling of the cylinder, which causes the condensation of a portion of the steam which enters at the following stroke. This loss of heat limits the number of times steam can be economically expanded in one cylinder; and wide variation of pressure in a cylinder produces irregular rotational force on the crank-pin.

The less the difference between the highest and lowest temperature in a cylinder, the less is the loss of heat and power from the cooling of the cylinder at the end of the stroke. Consequently, when large measures of expansion are used, the highest economy, derived from the expansion of steam, will be obtained by expanding the steam in two or more cylinders, according to the grade of expansion used, by which means the range between the highest and lowest pressure in each cylinder is diminished as much as possible, and loss of heat and power is prevented.

A Steam-Jacket keeps the metal of the cylinder at a uniform temperature, and prevents initial condensation of the steam. It also prevents condensation during expansion by imparting heat to the expanding steam. The heat abstracted from the steam during expansion is in proportion to the work performed and a part of the steam is condensed; the greater the measure of expansion the greater is the loss of heat and the greater the necessity for supplying the expanding steam with the heat lost in the performance of work. The heat corresponding to the steam condensed in an efficient steam-jacket, is—according to Mr. Anderson's theory—approximately equal to the amount of heat converted into work. If H=the total heat of the absolute pressure of the steam in the jacket, from Table 79, and T = its temperature, from Table 78, the heat-units, U, converted into work per minute are, $U = (\text{IHP} \times 33000) \div 772$, and the weight of water in lbs., W, from the steam condensed in the jacket per hour, is approximately, $W = \dfrac{U \times 60 \text{ minutes.}}{(H + 32^\circ) - T}$ For instance, if the pressure of the steam in the jackets of a steam-engine of 1200 indicated horse-power be 150 lbs., or $150 + 15 = 165$ lbs. per square inch absolute pressure, the heat converted into work per minute is $= (1200 \times 33000) \div 772 = 51295$ units, and the quantity of water from the steam condensed in the jackets is $= \dfrac{51295 \times 60 \text{ minutes}}{(1194 + 32^\circ) - 367^\circ} = \dfrac{3077700}{859} = 3583 \text{ lbs.} \div 10 \text{ lbs.} = 358.3$ galls. per hour; or $= \dfrac{3583 \text{ lbs.} \times 859 \text{ units}}{966 \text{ units}} = 3186 \text{ lbs.} \div 10 \text{ lbs.} = 318.6$ gallons per hour, when reduced to the standard temperature, from and at 212° Fahr.; and $= 3186 \text{ lbs.} \div 1200 = 2.66$ lbs. of water per indicated horse-power per hour.

The Pressure of Steam in a Cylinder when it opens to the Exhaust may be found by this *Rule:*—

$$\text{Terminal pressure} = \frac{\text{initial absolute pressure} \times \text{length of steam-stroke}}{\text{distance travelled by the piston when the exhaust opens.}}$$

Example: The length of stroke of an engine is 6 feet, initial pressure 65 lbs. per square inch, steam is cut off when the piston has travelled 1 foot of its stroke. Required the pressure in the cylinder when it opens to the exhaust, which is 3 inches before the piston arrives at the end of its stroke.

Then $\frac{65 + 15 \text{ lbs.} \times 1 \text{ foot}}{5{\cdot}75 \text{ feet}} = 14$ lbs., the terminal absolute pressure.

The Absolute Mean Pressure of Steam in a Cylinder and the Final Pressure may be found by the graphical method shown in Fig. 322.

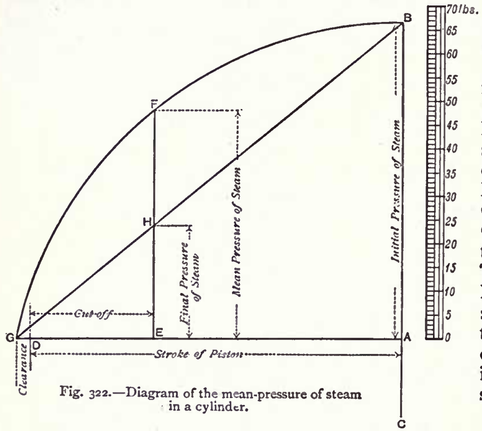


Fig. 322.—Diagram of the mean-pressure of steam in a cylinder.

Draw two lines, CB and AG, at right angles. Using any convenient scale, make the length AB = the initial absolute pressure of the steam, and AC = one-fourth the length of AB. With C as a centre and CB as a radius, draw the line B, F, G. The length of the line AG will represent the stroke of the piston, plus the clearance measured in terms of the stroke.

Find a point D, on the line AG, so that $\frac{DG}{DA} = \frac{\text{clearance in terms of the stroke}}{\text{length of stroke.}}$

In DA take E, so that $\frac{DE}{DA}$ = cut-off, and draw the perpendicular line EF. The length EF, measured on the same scale as AB, will equal the absolute mean pressure of the steam, from which deduct 15 lbs. in non-condensing-engines and 3 lbs. in condensing engines, and the remainder will be the effective mean pressure. Join the corners BG by a straight line, then the length from E to where the diagonal line cuts EF at H, measured on the same scale as the other pressures, will equal the final absolute pressure of the steam. Make the distance from G to D so that

$$\frac{DG}{DA} = \frac{\text{volume of clearance}}{\text{volume of cylinder swept by the piston in one stroke.}}$$

The initial absolute pressure of the steam shown in the diagram Fig. 322, is 67 lbs. per square inch, the steam is cut-off at one-third of the stroke.

The volume of one steam-port plus the clearance space between the piston and the cylinder-cover when the piston is at the beginning of its stroke, is equal to $\frac{1}{24}$th the stroke of the piston, therefore GD is $\frac{1}{24}$th part of DA. The diagram gives a mean absolute pressure of $48\frac{1}{2}$ lbs. per square inch, and a final absolute pressure of 24 lbs. per square inch.

The Effective Mean Pressure of Steam on the Piston may be found approximately without the aid of hyperbolic logarithms by the following formula :—

Let P = the absolute pressure of the steam in lbs. at the beginning of the stroke.
R = the number of times the steam is expanded.

Then, the effective mean pressure of the steam

$$= P \times \left(\frac{18 - R}{40} + \frac{\cdot 85}{R}\right) - \text{back-pressure.}$$

Example : Required the actual or indicated horse-power of an engine with cylinder 62 inches diameter, length of stroke 39 inches, pressure of steam in the cylinder at the beginning of the stroke 55 lbs. per square inch, cut off at 13 inches of the stroke, number of revolutions per minute 64, back-pressure 3·06 lbs.

Then, the steam is expanded $\frac{\text{39 inches stroke}}{\text{13 inches point of cut-off}} = 3$ times, the absolute pressure of the steam is 55 lbs. + 15 lbs. = 70 lbs. per square inch, and $\left(\frac{18-3}{40} + \frac{\cdot 85}{3}\right) = \cdot 658 \times 70$ lbs. $= 46\cdot 06 - 3\cdot 06$ back-pressure = 43 lbs. per square inch, the effective mean pressure of the steam on the piston, and $\frac{62 \times 62 \text{ inches diameter} \times \cdot 7854 \times 43 \text{ lbs.} \times 3\cdot 25 \text{ feet} \times 2 \times 64}{33000 \text{ lbs.}}$ = 1636·55 actual or indicated horse-power.

The Effective Mean Pressure of the Steam on the Piston throughout the Stroke may be found by this *Rule :* To the hyperbolic logarithm of the total number of expansions add 1, divide the sum by the total number of expansions, and multiply the quotient by the initial absolute pressure of the steam—that is, the boiler pressure shown by the steam-gauge plus 15 lbs.—which will give the average pressure of the steam expanded the given number of times, from which deduct the back-pressure, and the remainder will be the mean effective pressure on the piston throughout the stroke.

Example : Required the mean pressure of the steam in a compound engine, the initial pressure of the steam in the high-pressure cylinder being 69 lbs. per square inch, and the final pressure in the low-pressure cylinder 12 lbs. per square inch.

Then the initial absolute pressure of the steam is 69 + 15 = 84 lbs. per square inch, and $\frac{\text{84 lbs. initial absolute pressure}}{\text{12 lbs. final pressure}} = 7$, the total number of expansions, or number of times the steam is expanded.

The hyperbolic logarithm of 7 is from Table 90, $= 1{\cdot}9459$, which added to 1 $= 2{\cdot}9459$, then $\frac{2{\cdot}9459}{7 \text{ expansions}} = {\cdot}4208 \times 84$ lbs. absolute pressure $=$ 35·34 lbs. per square inch, the average pressure of steam of 84 lbs. absolute pressure expanded 7 times, and if 3·34 lbs. be deducted for back-pressure, it leaves 32 lbs. per square inch, the effective mean pressure.

Table 90.—Hyperbolic Logarithms.

Number.	Logarithm.	Number.	Logarithm.	Number.	Logarithm.	Number.	Logarithm.
$1\frac{1}{4}$	·2231	6	1·7918	$10\frac{3}{4}$	2·3749	16	2·7726
$1\frac{1}{2}$	·4054	$6\frac{1}{4}$	1·8326	11	2·3979	$16\frac{1}{2}$	2·8034
$1\frac{3}{4}$	·5596	$6\frac{1}{2}$	1·8718	$11\frac{1}{4}$	2·4201	17	2·8332
2	·6931	$6\frac{3}{4}$	1·9095	$11\frac{1}{2}$	2·4430	$17\frac{1}{2}$	2·8621
$2\frac{1}{4}$	·8109	7	1·9459	$11\frac{3}{4}$	2·4636	18	2·8904
$2\frac{1}{2}$	·9162	$7\frac{1}{4}$	1·9810	12	2·4849	$18\frac{1}{2}$	2·9173
$2\frac{3}{4}$	1·0116	$7\frac{1}{2}$	2·0149	$12\frac{1}{4}$	2·5052	19	2·9444
3	1·0986	$7\frac{3}{4}$	2·0477	$12\frac{1}{2}$	2·5262	$19\frac{1}{2}$	2·9703
$3\frac{1}{4}$	1·1787	8	2·0794	$12\frac{3}{4}$	2·5455	20	2·9957
$3\frac{1}{2}$	1·2528	$8\frac{1}{4}$	2·1102	13	2·5649	21	3·0445
$3\frac{3}{4}$	1·3217	$8\frac{1}{2}$	2·1401	$13\frac{1}{4}$	2·5840	22	3·0911
4	1·3862	$8\frac{3}{4}$	2·1691	$13\frac{1}{2}$	2·6027	23	3·1355
$4\frac{1}{4}$	1·4469	9	2·1972	$13\frac{3}{4}$	2·6211	24	3·1781
$4\frac{1}{2}$	1·5040	$9\frac{1}{4}$	2·2246	14	2·6391	25	3·2189
$4\frac{3}{4}$	1·5581	$9\frac{1}{2}$	2·2513	$14\frac{1}{4}$	2·6567	26	3·2581
5	1·6094	$9\frac{3}{4}$	2·2773	$14\frac{1}{2}$	2·6740	27	3·2958
$5\frac{1}{4}$	1·6582	10	2·3026	$14\frac{3}{4}$	2·6913	28	3·3322
$5\frac{1}{2}$	1·7047	$10\frac{1}{4}$	2·3279	15	2·7081	29	3·3673
$5\frac{3}{4}$	1·7492	$10\frac{1}{2}$	2·3513	$15\frac{1}{2}$	2·7408	30	3·4012

Mean Pressure of Steam in the Cylinder.—The lowest economical final pressure in a low-pressure cylinder is in practice from 10 to 12 lbs. per square inch. The mean pressure of steam in the cylinders of steam-engines worked expansively, with various initial pressures of steam worked down to a final pressure of 12 lbs. per square inch, is given in Table 91, which has been calculated by the previous rule with hyperbolic logarithms, but no allowance has been made for back-pressure, which must be deducted to obtain the effective mean pressure throughout the stroke. When the back-pressure is not known it will be sufficiently near in most cases to assume it to be 3 lbs. per square inch.

The Table shows the pressure of steam required for a given number of expansions, the point of cut-off required, and the mean pressure of the steam throughout the stroke. For instance, if it be required to expand steam 10 times, steam would be required of 120 lbs. initial absolute pressure, the point of cut-off from the commencement of the stroke would be at $\frac{1}{10}$th of the length of stroke, and the mean pressure of the steam would be 39·62 lbs. per square inch.

Table 91.—Showing the Mean Pressure of Steam in the Cylinders of Steam-Engines worked Expansively with various Initial Pressures of Steam worked down to a Final Pressure of 12 lbs. per Square Inch.

Number of Times the Steam is Expanded.	Point of Cut-off in Fractions of the Length of Stroke, from its Commencement.	Mean Pressure throughout the Stroke, the Initial Pressure being = 1.	Absolute Pressure. Initial Pressure in lbs. per Square Inch.	Absolute Pressure. Mean Pressure in lbs. per Square Inch.	Number of Times the Steam is Expanded.	Point of Cut-off in Fractions of the Length of Stroke, from its Commencement.	Mean Pressure throughout the Stroke, the Initial Pressure being = 1.	Absolute Pressure. Initial Pressure in lbs. per Square Inch.	Absolute Pressure. Mean Pressure in lbs. per Square Inch.
			lbs.	lbs.				lbs.	lbs.
1 1/8	7/8	·9933	13·5	13·40	10	1/10	·3302	120	39·62
1 1/4	4/5	·9875	15	14·81	10 1/4	4/41	·3246	123	49·92
1 1/3	3/4	·9641	16	15·42	10 1/2	2/21	·3191	126	40·20
1 3/7	7/10	·9488	17·1	16·22	10 3/4	4/43	·3139	129	40·49
1 1/2	2/3	·9369	18	16·88	11	1/11	·3089	132	40·77
1 2/3	6/10	·9090	20	18·18	11 1/4	4/45	·3040	135	41·04
1 3/4	4/7	·8912	21	18·71	11 1/2	2/23	·2993	138	41·30
2	1/2	·8465	24	20·31	11 3/4	4/47	·2947	141	41·55
2 1/4	4/9	·8048	27	21·73	12	1/12	·2904	144	41·81
2 3/11	11/25	·8010	26·18	20·95	12 1/4	4/49	·2861	147	42·05
2 1/2	2/5	·7664	30	22·99	12 1/2	2/25	·2821	150	42·31
2 3/4	4/11	·7311	33	24·12	12 3/4	4/51	·2780	153	42·53
3	1/3	·6995	36	25·18	13	1/13	·2742	156	42·77
3 1/4	4/13	·6703	39	26·14	13 1/4	4/53	·2704	159	42·99
3 1/3	3/10	·6618	40	26·47	13 1/2	2/27	·2668	162	43·22
3 1/2	2/7	·6436	42	27·03	13 3/4	4/55	·2633	165	43·44
3 3/4	4/15	·6191	45	27·85	14	1/14	·2599	168	43·66
4	1/4	·5965	48	28·63	14 1/4	4/57	·2566	171	43·87
4 1/4	4/5	·5757	51	29·36	14 1/2	2/29	·2533	174	44·07
4 1/2	2/9	·5463	54	29·50	14 3/4	4/59	·2502	177	44·28
4 3/4	4/19	·5385	57	30·69	15	1/15	·2472	180	44·49
5	1/5	·5218	60	31·30	15 1/2	2/31	·2412	186	44·86
5 1/4	4/21	·5063	63	31·89	16	1/16	·2358	192	45·37
5 1/2	2/11	·4917	66	32·45	16 1/2	2/33	·2305	198	45·63
5 3/4	4/23	·4781	69	32·98	17	1/17	·2255	204	46·02
6	1/6	·4653	72	33·50	17 1/2	2/35	·2206	210	46·32
6 1/4	4/23	·4532	75	33·99	18	1/18	·2160	216	46·65
6 1/2	2/13	·4418	78	34·46	18 1/2	2/37	·2112	222	46·88
6 3/4	4/27	·4310	81	34·91	19	1/19	·2076	228	47·33
7	1/7	·4208	84	35·34	19 1/2	2/39	·2036	234	47·64
7 1/4	4/29	·4111	85	34·94	20	1/20	·1998	240	47·95
7 1/2	2/15	·4002	90	36·01	21	1/21	·1926	252	48·53
7 3/4	4/31	·3932	93	36·56	22	1/22	·1860	264	49·10
8	1/8	·3849	96	36·95	23	1/23	·1800	276	49·68
8 1/4	4/33	·3779	99	37·32	24	1/24	·1741	288	50·14
8 1/2	2/17	·3694	102	37·67	25	1/25	·1687	300	50·61
8 3/4	4/35	·3621	105	38·02	26	1/26	·1638	312	51·10
9	1/9	·3552	108	38·36	27	1/27	·1591	324	51·54
9 1/4	4/37	·3486	111	38·69	28	1/28	·1547	336	51·97
9 1/2	2/19	·3422	114	39·01	29	1/29	·1506	348	52·40
9 3/4	4/39	·3361	117	39·32	30	1/30	·1467	360	52·81

The Back-Pressure, usually 3 lbs. per square inch, must be deducted from the mean pressure of the steam given in the above Table, and the remainder will be the effective mean pressure throughout the stroke.

Number of Times the Steam is Expanded.—If the number of expansions be carried out in two or more cylinders, as in double-expansion, triple-expansion, and quadruple-expansion engines, the final pressure of the steam is the same as in a single cylinder engine, and this table applies to all steam-engines in which the steam is worked expansively. The total expansion is equal to the number of cubic feet of space behind the low-pressure piston when the exhaust opens, divided by the space behind the high-pressure piston at the point of cut-off; or=final volume÷initial volume.

The Diameter of the Cylinder of a Steam-Engine may be found from Table 91, as shown by the following example—

Example: Required the diameter of a cylinder for an engine of 800 indicated horse-power, with an initial pressure of steam of 90 lbs. per square inch, above the atmosphere, with a speed of piston of 420 feet per minute, back-pressure 3·02 lbs. per square inch, the steam to be expanded $8\frac{3}{4}$ times?

Then the initial absolute pressure of the steam will be $= 90 + 15$ lbs.$=$ 105 lbs. per square inch, the mean pressure of which with $8\frac{3}{4}$ expansions is according to the Table$=38{\cdot}02$ lbs. per square inch, from which deduct the back-pressure, and it leaves $38{\cdot}02 - 3{\cdot}02 = 35$ lbs., the effective mean pressure of the steam throughout the stroke.

The total pressure on the piston to develop 800 horse-power will be $= \frac{33000 \times 800 \text{ IHP}}{420 \text{ feet per minute}} = 62762$ lbs., and $\frac{62762 \text{ lbs.}}{35 \text{ lbs. pressure}} = 1793$ square inches area of the cylinder, and $\sqrt[2]{\frac{1793}{{\cdot}7854}}$=say 48 inches diameter.

The Diameter of a Cylinder suitable for a given size of Crank-Shaft, for a low-pressure steam-engine using steam of under 35 lbs. initial pressure per square inch, may be found by the following formula:—

Let D $=$ the diameter of cylinder in inches; $d =$ the diameter of the crank-shaft in inches; L $=$ the length of stroke in feet.

$$\text{Then } D = \sqrt[2]{\frac{6{\cdot}55\, d^3}{L}}$$

Example: What diameter of cylinder is suitable for an engine using steam of 30 lbs. per square inch initial pressure, the diameter of the crank-shaft being $13\frac{1}{2}$ inches, and the length of stroke 4 feet?

Then $\frac{6{\cdot}55 \times 13{\cdot}5 \times 13{\cdot}5 \times 13{\cdot}5 \text{ inches}}{4 \text{ feet length of stroke}} = \sqrt[2]{4026} = 64$ inches, the diameter of cylinder required.

Diameter of Cylinder of Non-Condensing Steam-Engines.—The area of the piston should be in proportion to the pressure of the steam and the speed of the piston. The piston-area per nominal horse-power may be 8 square inches for very high pressures and speed, $9\frac{1}{2}$ to 10 square inches for high pressures and speed, and 11 square inches for moderate pressures and speeds. Diameter $= \sqrt[2]{}$(area in square inches $\div$ 7854).

Diameter of Cylinder of Condensing Steam-Engines.—The area of the piston per nominal horse-power may be from 13 to 16 square inches for very high pressures of steam, from 18 to 20 square inches for high pressures, and from 22 to 25 square inches for low pressure steam.

Diameter of Cylinder, per Nominal Horse-Power, of Double Expansion Compound Steam-Engines, with one high-pressure cylinder and one low-pressure cylinder.

Diameter of high-pressure cylinder $= \sqrt[2]{\dfrac{\text{nominal horse-power} \times 26}{\text{ratio of the areas of cylinders.}}}$

Diameter of low-pressure cylinder = diameter of high-pressure cylinder $\times \sqrt[2]{\text{ratio of cylinder-capacity.}}$

Example: Required the diameter of the cylinders of a compound steam-engine of 250 nominal horse-power, the ratio of the cylinder areas being 4 to 1.

Then $\dfrac{250 \times 26}{4} = \sqrt[2]{1625} = 40{\cdot}32$ inches, diameter of high-pressure cylinder;

and $40{\cdot}32 \times \sqrt[2]{4} = 80{\cdot}64$ inches, diameter of low-pressure cylinder.

The Diameter of Cylinder required for a given Actual or Indicated Horse-power may be found by the following *Rule:*—

Diameter of cylinder =

$$\sqrt[2]{\frac{33000 \times \text{number of indicated horse-power}}{{\cdot}7854 \times \text{mean pressure of steam in lbs.} \times \text{speed of piston in feet.}}}$$

Example: Required the diameter of cylinder for an engine with steam of 42 lbs. per square inch mean pressure, length of stroke 2 feet, to develop 25 indicated horse-power when making 70 revolutions per minute.

Then $\sqrt[2]{\dfrac{33000 \times 25 \text{ horse-power}}{{\cdot}7854 \times 42 \text{ lbs.} \times (2 \text{ feet} \times 2 \times 70)}} = 9\tfrac{1}{2}$ inches, the diameter of cylinder required for that engine.

In Compound Engines the steam, after performing work in one cylinder, is exhausted into and performs work in another, or in several cylinders. The expansion of the steam being carried out in two or more cylinders, the loss from cooling produced by extreme variation of temperature is diminished as much as possible by dividing the extreme range of temperature of the steam between several cylinders. The economy effected by using a double expansion compound engine, with one high-pressure and one low-pressure cylinder, instead of a simple engine, is about 25 per cent. of fuel. The expansion of the steam is carried out in two stages in two cylinders in double-expansion compound engines, in three stages in three cylinders in triple-expansion engines, and in four stages in four cylinders in quadruple-expansion engines.

The Ratio of the Capacities of the Cylinders of Compound or Double-expansion Engines depends upon the initial pressure of the steam in the high-pressure cylinder. The area of the low-pressure cylinder of a compound engine is calculated as if all the power were to be developed in that cylinder. When the diameter of the low-pressure cylinder would be excessive, the capacity of the low-pressure cylinder is divided between two cylinders, and the cylinders are arranged to act on a three-throw crank, with crank-pins placed at equal angles of 120° apart, in order to secure equality of strains and power and uniformity of turning-effort. The diameters of the cylinders of a double-expansion compound condensing engine, with one high-pressure cylinder and one low-pressure cylinder, for a given actual or indicated horse-power may be found by the following *Rule*:—

Let IHP = the indicated or actual horse-power of the engines.
E = the effective mean pressure of the steam in lbs. per square inch.
S = the length of stroke in feet.
R = the number of revolutions of the crank-shaft per minute.
LP = the area of the low-pressure cylinder in square inches.
HP = the area of the high-pressure cylinder in square inches.

$$\text{Then LP} = \frac{33000 \times \text{IHP}}{\text{E} \times (\text{S} \times 2 \times \text{R})}$$

$$\text{HP} = \frac{\text{LP}}{\text{initial absolute working-pressure} \times \cdot 042}.$$

Example: Required the diameters of the two cylinders of a marine compound condensing engine to indicate 1400 horse-power, length of stroke 3 feet 9 inches, number of revolutions per minute 56, the working-pressure of the steam by the steam-gauge being 90 lbs. per square inch, and the mean pressure 37 lbs. per square inch.

Then the absolute pressure of the steam will be $90 + 15 = 105$ lbs. per square inch.

and $\dfrac{33000 \times 1400}{37 \text{ lbs.} \times (3{\cdot}75 \times 2 \times 56)} = 2973$ square inches, the area of the low-pressure cylinder.

$\dfrac{2973}{105 \text{ lbs.} \times \cdot 042} = 675$ square inches, the area of the high-pressure cylinder.

Then $\sqrt[2]{\dfrac{2973}{\cdot 7854}} =$ say 62 inches, the diameter of the low-pressure cylinder.

$\sqrt[2]{\dfrac{675}{\cdot 7854}} =$ say $29\frac{1}{2}$ inches, the diameter of the high-pressure cylinder.

In order to provide for the loss due to the fall in pressure of the steam in passing from one cylinder to the other, their areas found by the above Rules should be increased to the extent of from 10 to 20 per cent.

The Proportion of the Cylinders of Triple-Expansion Engines depends upon the initial pressure of the steam in the high-pressure cylinder. The diameters of the cylinders of triple-expansion surface-condensing marine engines may be calculated by the following formulæ:—

Let IHP = the indicated horse-power of the engines.
E = the effective mean pressure of the steam in lbs. per square inch.
S = the length of stroke in feet.
R = the number of revolutions of the crank-shaft per minute.
LP = the area of the low-pressure cylinder in square inches.

$HP =$ the area of the high-pressure cylinder in square inches.
$M =$ the area of the intermediate cylinder, in square inches.

$$\text{Then } LP = \frac{33000 \times IHP}{E \times (S \times 2 \times R)}$$

$$HP = \frac{LP}{\text{initial absolute working-pressure} \times \cdot 042}$$

$$M = HP \times 2\cdot 5.$$

Example: Required the diameters of the three cylinders of a triple expansion engine to indicate 1000 horse-power, length of stroke 3 feet 6 inches, number of revolutions per minute 60, the working-pressure of the steam by the steam-gauge being 150 lbs. per square inch, and the mean pressure of the steam 30 lbs. per square inch.

Then the absolute pressure of the steam will be 150 + 15 lbs. = 165 lbs. per square inch,

and $\frac{33000 \times 1000}{30 \times (3\cdot 5 \times 2 \times 60)} = 2620$ square inches, the area of the low-pressure cylinder,

$\frac{2620 \text{ square inches}}{165 \text{ lbs.} \times \cdot 042} = 379$ square inches, the area of the high-pressure cylinder,

and 379 square inches $\times\ 2\cdot 5 = 948$ square inches, the area of the intermediate cylinder.

Then $\sqrt[2]{\frac{2620}{\cdot 7854}} = 58$ inches, the diameter of the low-pressure cylinder,

$\sqrt[2]{\frac{948}{\cdot 7854}} = 35$ inches, the diameter of the intermediate cylinder,

$\sqrt[2]{\frac{379}{\cdot 7854}} = 22$ inches, the diameter of the high-pressure cylinder.

The proportion of these cylinders is in round numbers $1 : 2\frac{1}{2} : 7$, which is correct for this initial pressure of steam. All the cylinders, or at least the high-pressure cylinder, should be steam-jacketed to obtain the utmost economy.

The best arrangement of the Cranks to ensure uniformity of rotative pressure on the crank-shaft is to place them at equal angles of 120° apart; and to make the crank of the high-pressure cylinder the leading crank, the low-pressure crank should follow and the intermediate crank should be last. The economy effected by using a triple-expansion engine instead of a double-expansion or compound engine averages 25 per cent. in the best engines; in some cases as much as 33 per cent. saving of fuel has been effected.

The Economy of Triple-Expansion and Quadruple-Expansion Engines is due, partly to the utilization of the principle of expansion in using steam of the highest pressure and expanding it as many times as possible in the most efficient manner, the extreme range of temperature being divided between several cylinders; and partly to the means afforded for the effectual re-evaporation of the initial condensation of steam, the steam condensed in the small cylinder being re-evaporated to steam of a lower pressure in each larger cylinder and used expansively upon its piston.

The Diameters of the Cylinders of Quadruple-Expansion Surface-Condensing Marine-Engines may be found by the following formula:—

Let IHP = the indicated horse-power of the engines.
E = the effective mean pressure of the steam in lbs. per square inch.
S = the length of stroke in feet.
R = the number of revolutions of the crank-shaft per minute
LP = the area of the low-pressure cylinder in square inches.
HP = the area of the high-pressure cylinder in square inches.
A = the area of the *first* intermediate cylinder in square inches.
B = the area of the *second* intermediate cylinder in square inches.

$$\text{Then LP} = \frac{33000 \times \text{I H P}}{\text{E} \times (\text{S} \times 2 \times \text{R})}.$$

$$\text{HP} = \frac{\text{L P}}{\text{Initial absolute working pressure} \times \cdot 042}.$$

$$\text{A} = \text{HP} \times 2.$$

$$\text{B} = \text{A} \times 2.$$

Example: Required the diameters of the four cylinders of a quadruple-expansion engine to indicate 500 horse-power. Length of stroke, 2 feet; number of revolutions, 105; the working pressure of the steam by the steam-gauge being 165 lbs. per square inch, the steam to be expanded 15 times.

Then the absolute of the steam is $165 + 15 = 180$ lbs. per square inch; the mean pressure of which, when expanded 15 times, is, according to Table 91, = 44·49 lbs., and if 3·09 lbs. be deducted for back-pressure, it leaves $44{\cdot}49 - 3{\cdot}09 = 41{\cdot}4$ lbs., the effective mean pressure.

And $\frac{33000 \text{ lbs.} \times 500}{41{\cdot}4(2 \times 2 \times 105)} = 948{\cdot}93$ square inches, the area of the low-pressure cylinder.

$\frac{948{\cdot}93}{180 \times \cdot 042} = 125{\cdot}52$ square inches, the area of the high-pressure cylinder.

$125{\cdot}52 \times 2 = 251{\cdot}04$ square inches, the area of the *first* intermediate cylinder.

$251{\cdot}04 \times 2 = 502{\cdot}08$ square inches, the area of the *second* intermediate cylinder.

Then $\sqrt[2]{\frac{948{\cdot}93}{\cdot 7854}} = 34\frac{3}{4}$, or say 35 inches, the diameter of the low-pressure cylinder.

$\sqrt[2]{\frac{502{\cdot}8}{\cdot 7854}} = 25\frac{3}{8}$, or say $25\frac{1}{2}$ inches, the diameter of the *second* intermediate cylinder.

$\sqrt[2]{\frac{251{\cdot}04}{\cdot 7854}} = 17\frac{7}{8}$, or say 18 inches, the diameter of the *first* intermediate cylinder.

$\sqrt[2]{\frac{125 \cdot 52}{\cdot 7854}} = 12\frac{5}{8}$, or say $12\frac{3}{4}$ inches, the diameter of the high-pressure cylinder.

The proportion of these cylinders is in round numbers, 1 : 2 : 4 : $7\frac{1}{2}$, which is correct for this initial pressure of steam.

Cylinders should be cast from hard close-grained cast-iron, perfectly free from honey-comb. The cylinders of marine-engines are usually fitted with a liner of either Whitworth's compressed steel, or hard cast-iron, having a flange at one end by which it is bolted to the cylinder, as shown in Fig. 323. The space between the liner and the cylinder-casting is from 1 inch to $1\frac{1}{2}$ inches, forming a jacket which can be filled with steam from the boiler to prevent condensation.

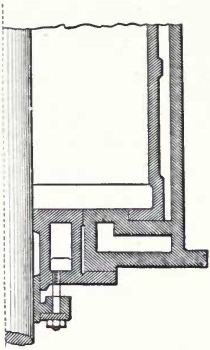


Fig. 323.—Cylinder-liner.

Cast-Iron Liners for Cylinders should be cast from tough, hard, close-grained metal; the following is a good mixture of metal for this purpose—

White	3 cwt.
Summerlee	3 ,,
Weardale	2 ,,
Scotch Mixed Brands, No. 3 . . .	6 ,,
Good Clean Scrap	6 ,,

melted and cast into pigs in order to be properly mixed. A test-bar of cast-iron cast from this mixture, 1 inch square, placed upon supports 3 feet apart, should bear a gradually applied weight of about $7\frac{3}{4}$ cwt., with a deflection of about $\frac{11}{16}$ inch.

The Thickness of a Cast-Iron Liner for a Cylinder may be found by the following formula:—

Let D = the diameter of the cylinder in inches.
P = the initial pressure of the steam in lbs. per square inch.
C = a constant divisor = 2400 for cast-iron.
T = the thickness of the liner in inches.

$$\text{Then } T = \frac{D \times P}{C}.$$

Example: Required the thickness of a cast-iron liner for a cylinder 60 inches diameter; initial pressure of the steam 70 lbs. per square inch.

$$\text{Then } \frac{60 \times 70}{2400} = 1 \cdot 75 \text{ inch.}$$

The Thickness of a Steel Liner for a Cylinder may be found by the above Rule by using a constant divisor, C, of 3500.

Example: Required the thickness of a steel liner for a cylinder 40 inches diameter; initial pressure of the steam 75 lbs. per square inch.

$$\text{Then } \frac{40 \times 75}{3500} = \cdot 86 \text{ inch.}$$

The Thickness of Metal for Marine-Engine Cylinders may be found by the following formula:—

Let D = the diameter of the cylinder in inches.
P = the initial pressure of the steam in lbs. per square inch.
T = the thickness of the cylinder in inches.

$$\text{Then } T = \frac{D \times P}{3000} + \cdot 6.$$

Example: Required the thickness of a marine-engine cylinder 50 inches diameter; initial pressure of the steam 70 lbs. per square inch.

$$\text{Then } \frac{50 \times 70}{3000} = 1\cdot 17 + \cdot 6 = 1\cdot 77.$$

Locomotive-Engine Cylinders are usually $\frac{7}{8}$ inch thick when 17 inches diameter, 1 inch thick when 18 inches diameter, and $1\frac{1}{8}$ inch thick

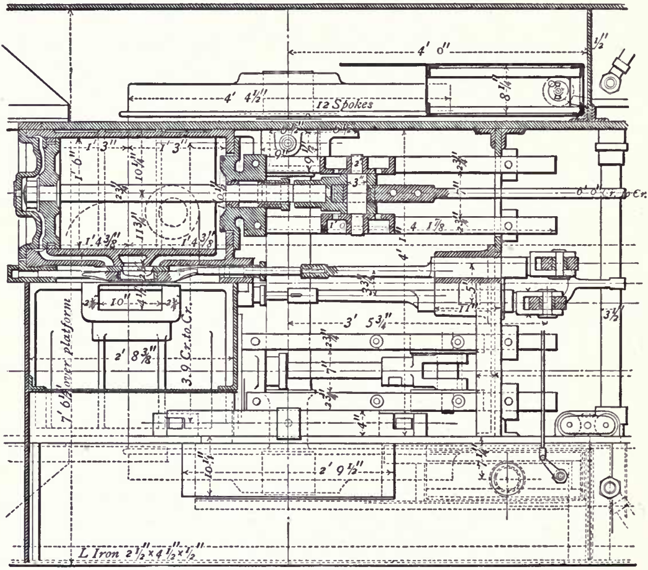


Fig. 324.—Plan of the cylinders of a locomotive-engine.

when 19 inches diameter. In locomotive cylinders a considerable allowance is necessary for re-boring, to provide for wear and tear due to high piston speed, and for the liability of the cylinders to become scored by ashes drawn into the cylinder through the exhaust-passages, when the engine is running with the steam shut off. A plan of the cylinders of a passenger locomotive engine is shown in Fig. 324. The cylinders are made of the best close-grained cast-iron, twice run, as hard as can be worked, and free from honey-comb. The front cover of each cylinder is dished to correspond to the piston, and the back cover is provided with lugs for carrying the front ends of the slide-bars. The cylinders are 18 inches diameter, and 2 feet $4\frac{1}{2}$ inches centre to centre, length of stroke 26 inches: they are attached to the frames by flanges, and secured by turned bolts, driven into rose-bitted holes. The tops of the cylinders are generally covered with thin fire-brick or cement. The pistons are of tough cast-iron, the packing rings are of cast-iron, turned $\frac{1}{2}$ an inch larger than the cylinder, and then cut and sprung into their places. The piston-rods are made of mild steel $2\frac{3}{4}$ inches diameter. The steam-ports are $1\frac{1}{2}$ inches wide and 15 inches long, and the exhaust-port is $3\frac{1}{4}$ inches wide; thickness of bridges, 1 inch. The slide-valves are made of hard gun-metal, the slide-valve spindles are of best Yorkshire iron. The slide-bars are made of mild cast-steel, and the slide-blocks are of chilled cast-iron; the cross-head and gudgeons are of best Yorkshire iron; the gudgeons are case-hardened, and forced into the cross-heads by hydraulic pressure. The taper of the cone in the cross-head is 1 in 16, and in the piston 1 in 6: the number of threads per inch of the screwed end of the piston-rod = 6.

Locomotive-engine cylinders should be cast from metal of the following mixture, or equal quality:—

Pontypool Cold Blast,	No. 4	10 cwt.
Clyde	No. 4	7 „
Coltness	No. 4	3 „

melted and cast into pigs, in order to be properly mixed.

The Cylinder of a Horizontal Stationary-Engine is shown in Fig.

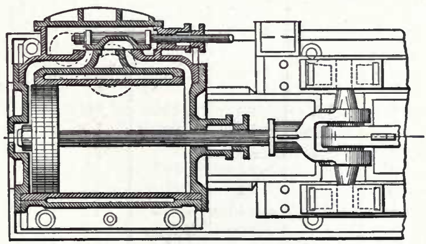


Fig. 325.—Cylinder of a horizontal stationary-engine.

325. It is provided with a liner of cast-iron. Stationary-engine cylinders should be cast from metal of the following mixture, or equal quality:—

Pontypool Cold Blast,	No. 4	8 cwt.
Monkland	No. 4	6 „
Clyde	No. 4	6 „

melted and cast into pigs, in order to be properly mixed.

Table 92.—PROPORTIONS OF THE CYLINDERS OF HORIZONTAL AND VERTICAL NON-CONDENSING SATIONARY-ENGINES.

LIGHT ENGINE WITH SHORT STROKE.				STRONG ENGINE WITH LONG STROKE.			
Nominal Horse-power.	Diameter of Cylinder.	Length of Stroke.	Speed in Revolutions per Minute.	Nominal Horse-power.	Diameter of Cylinder.	Length of Stroke.	Approximate Weight.
	Inches.	Inches.			Inches.	Inches.	Cwts.
$1\frac{1}{2}$	$4\frac{1}{4}$	7	215	3	5	10	13
2	$4\frac{3}{4}$	8	180	4	6	12	16
$2\frac{1}{2}$	$5\frac{1}{4}$	8	180	5	7	14	23
3	$6\frac{1}{4}$	8	180	6	8	16	30
4	$6\frac{3}{4}$	10	150	7	9	18	41
5	$7\frac{3}{8}$	12	125	8	10	20	62
6	$8\frac{1}{2}$	12	125	10	11	22	72
7	$8\frac{7}{8}$	12	125	12	12	24	83
8	$9\frac{1}{2}$	12	125	14	13	26	92
9	$10\frac{1}{8}$	12	125	16	14	28	100
10	$10\frac{1}{2}$	14	110	18	15	30	116
12	12	14	110	20	16	32	132
14	$12\frac{1}{2}$	16	100	25	18	36	174
16	$13\frac{1}{2}$	20	80	30	20	40	210

Table 93.—PROPORTIONS OF THE CYLINDERS OF COMPOUND, OR DOUBLE-EXPANSION, CONDENSING MARINE-ENGINES.

Nominal Horse-power.	Diameter of High-pressure Cylinder.	Diameter of Low-pressure Cylinder.	Length of Stroke.	Nominal Horse-power.	Diameter of High-pressure Cylinder.	Diameter of Low-pressure Cylinder.	Length of Stroke.
	Inches.	Inches.	Inches.		Inches.	Inches.	Inches.
8	7	13	9	120	28	52	36
10	8	14	12	135	29	56	36
25	12	24	14	140	30	57	36
40	16	30	22	155	32	60	39
50	17	34	24	175	36	62	42
55	18	36	24	220	38	70	45
70	21	40	24	280	45	78	48
85	23	44	30	320	48	84	48
100	25	48	30	360	50	90	60
110	27	50	33	480	56	105	72

The Cylinders of a Set of Triple-Expansion Surface-Condensing Marine-Engines are shown in Fig. 326. The high-pressure cylinder is 19 inches diameter, the intermediate cylinder is 30 inches diameter, and the low-pressure cylinder is 50 inches diameter, length of stroke 36 inches. The high-pressure cylinder is fitted with a piston-valve 9 inches diameter, the intermediate and low-pressure cylinders are fitted with double-ported slide-valves. The weight of the slide-valve of the low-pressure cylinder is counterbalanced by a balance-cylinder at the upper end of the valve-spindle. These engines indicated 800 horse-power, the boiler-pressure of the steam being 150 lbs. per square inch.

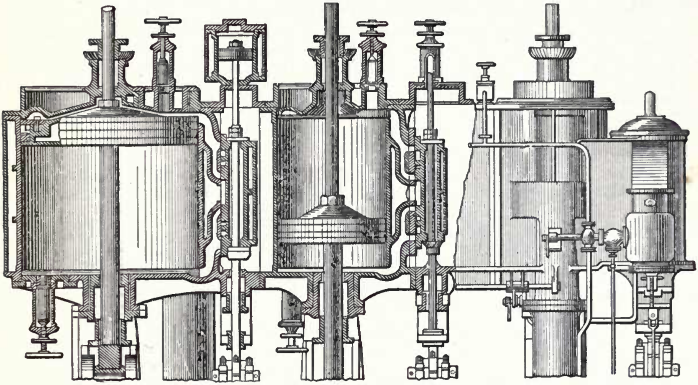


Fig. 326.—Cylinders of a set of triple-expansion marine-engines.

Marine-engine cylinders should be cast from metal of the following mixture, or equal quality—

Gartsherrie, No. 3	3 cwt.
Monkland No. 3	3 „
Coltness No. 3	3 „
Summerlee, No. 3	3 „
Clyde No. 3	3 „
Good clean scrap	5 „

melted and cast into pigs, in order to be properly mixed.

A test-bar cast from any of the above cylinder-mixtures of cast-iron, 1 inch square, placed upon supports 3 feet apart, should bear a gradually-applied weight of from $7\frac{3}{4}$ cwt. to 8 cwt., with a deflection of from $\frac{5}{8}$ inch to $\frac{3}{4}$ inch.

The Velocity of Steam through the Cylinders of Triple-Expansion Engines should not exceed 300 feet per second, and the velocity of the exhaust-steam should not exceed 170 feet per second. The velocity of the steam in feet per second may be found by this *Rule*:—

$$\text{Velocity} = \frac{\text{area of cylinder} \times \text{piston-speed in feet per second}}{\text{area of the opening of the port.}}$$

The initial velocities of the steam in triple expansion engines are frequently as follows:—

		High-pressure Cylinder.	Intermediate Cylinder.	Low-pressure Cylinder.
Initial velocity of steam per second	=	100 ft.	200 ft.	250 ft.
Initial velocity of exhaust-steam per second . .	=	90 ft.	120 ft.	140 ft.

In some cases the velocity of the exhaust-steam in the low-pressure cylinder is as high as 160 feet per second without detrimental effect.

Table 94.—Proportions of the Cylinders of Triple-Expansion Surface-Condensing Marine-Engines, collated from Recent Practice. Working-Pressure of Steam, from 150 to 160 lbs. per square inch.

Name of Vessel.	Diameter of High-pressure Cylinder.	Diameter of Intermediate Cylinder.	Diameter of Low-pressure Cylinder.	Length of Stroke.
	Inches.	Inches.	Inches.	Inches.
Roseland	6	9	16	12
Cassandra	9	14	22	15
Condor	11	16½	30	21
Somalie	13½	21	34	24
Elgiva	15	24	40	33
Warrior	16	24	40	24
J. Joicey	16½	26	43	36
Charles	17½	29	47	33
Teresa	18	30	48	36
Fijian	18½	31	49	36
Cairnryan	18½	31	51	36
Gloamin	19	30	50	42
Mandalay	20	33	54	36
Drever	20	33	54	36
Bleville	21	33	52	42
Era	21	34	57	39
Obeona	21	35	57	39
Thames	21½	33	54	36
Loch Etive	21½	34	56	42
Northenden	21½	35	57	39
Indian Prince	21½	37	58	39
Flamboro	22	35	58	42
Clitus	23½	37	61	42
Rosemorran	23½	38	62	42
Saint Oswald	24	39	64	42
Euterpe	24	42	69	48
Dunbrodie	24½	39	62	42
Chingtua	25	40	62	48
Anglian	26	42	69	42
Argus	26	43	70	45
Methley Hall	27	44	71	48
Iran	28	43	77	51
Port Pirie	29	44	74	48
Locksley Hall	29	47	76	51
Monmouthshir	30	47	70	51
Buffalo	33	54	86	60
Lusitania	36	60	96	48
Orizaba	40	66	100	72
Orinoco	42	66	96	66
Ormuz	46	73	112	72

The indicated horse-power of the engines given in the previous Table may be calculated approximately by taking the average piston-speed and mean-pressure of the steam in the cylinders of triple-expansion engines in merchant-steamers. It may be assumed that the effective mean-pressure of the steam of the three cylinders reduced to the low-pressure cylinder, averages one-fifth of the initial absolute pressure, and that the average piston-speed is 500 feet per minute for engines of moderate speed, and 770 feet per minute for high-speed engines: the latter being the piston-speed of the engines of some Atlantic steamers. Take for instance the engines of the steamer *Thames:* the working-pressure of the steam is 150 lbs. per square inch $= 150 + 15 = 165$ lbs. per square inch absolute pressure, and $165 \times \frac{1}{5} = 33$ lbs., the assumed effective mean-pressure, the low-pressure cylinder is 54 inches diameter, then, if working at the moderate piston-speed assumed, the engines would develop =

$$\frac{54 \times 54 \text{ inches} \times \cdot 7854 \times 33 \text{ lbs.} \times 500 \text{ feet piston-speed}}{33000} = 1145, \text{ indi-}$$

cated horse-power.

Again, the engines of the large steamer *Orizaba*, if working at the highest piston-speed assumed, would develop =

$$\frac{100 \times 100 \text{ inches} \times \cdot 7854 \times 33 \text{ lbs.} \times 770 \text{ feet piston-speed}}{33000} = 6000 \text{ indi-}$$

cated horse-power.

Cylinder Escape-Valves, shown in Fig. 327, are fitted to cylinder-covers to allow the escape of water from condensation or priming, which, in case there were no relief-valves and the drain-cocks were closed, could only escape from the cylinder by forcing the slide-valve from its seat. The diameter of the escape-valve may be equal to one-sixteenth the diameter of the cylinder.

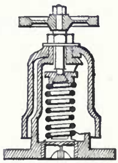


Fig. 327.—Cylinder escape-valve.

Bolts of Cylinder-Covers.—The working-strain on the bolts of cylinder-covers should not exceed 2000 lbs. per square inch of section.

The number of bolts required for a cylinder-cover may be found by this *Rule:*—

$$\text{Number of bolts} = \frac{\text{area of cylinder} \times \text{pressure of steam}}{\text{area of one bolt} \times \text{working-strain on bolts.}}$$

Example: How many bolts $1\frac{1}{2}$ inches diameter are required for the cover of a cylinder of 56 inches diameter, the initial pressure of the steam being 90 lbs. per square inch, the working-strain on the bolts not to exceed 2000 lbs. per square inch of section of the bolts?

$$\text{Then } \frac{56 \times 56 \text{ inches} \times \cdot 7854 \times 90 \text{ lbs. pressure}}{1 \cdot 5 \times 1 \cdot 5 \text{ inch} \times \cdot 7854 \times 2000 \text{ lbs. strain}} = 62 \cdot 15, \text{ or say } 63 \text{ bolts}$$

will be required for that cylinder-cover.

The number of bolts when the diameter and pitch are given, may be found by the following *Rule:*—

$$\text{Number of bolts} = \frac{\text{diameter of pitch circle of bolts} \times 3 \cdot 1416}{\text{pitch of the bolts}}$$

Example: The diameter of a cylinder-cover is 33 inches, the centres of the bolts are $1\frac{1}{2}$ inches from the edge, the pitch of the bolts is $3\frac{1}{4}$ inches. How many bolts should there be in the cover?

Then $33 - (1\frac{1}{2} \times 2) = 30$ inches, diameter of pitch circle of bolts.

$$\frac{30 \times 3{\cdot}1416}{3{\cdot}25 \text{ inch pitch}} = 28{\cdot}99, \text{ or say } 29 \text{ bolts.}$$

The Most Economical Piston-Speed is attained when an engine is run at as high a speed as its design and condition will permit with freedom from vibration and heated bearings, because a piston with a given pressure upon it will exert power in proportion to its speed. The speed of a piston in feet per minute is found by multiplying twice the length of stroke in feet by the number of revolutions of the crank-shaft per minute. Piston-speeds vary considerably in practice, they range from 300 to 1200 feet per minute.

The piston-speed of double expansion compound-engines averages 420 feet per minute, triple expansion engines of moderate speed average 500 feet per minute, and of high-speed from 750 feet to 800 feet per minute, and torpedo-boat engines average from 1000 feet to 1200 feet per minute. The highest piston-speed is obtainable with the least wear and tear where the rotational force is most uniform; a nearly uniform turning-effort is obtained by dividing the application of the power between three cranks placed at equal angles of 120° apart on the same shaft, with an equal initial stress on each crank.

Piston-Displacement is the space swept through by the piston in a single stroke of an engine. It may be found by multiplying the area of the piston by the length of stroke.

Example: Required the piston-displacement of an engine with cylinder 30 inches diameter, length of stroke 3 feet, making 70 revolutions per minute.

Then $2{\cdot}5 \times 2{\cdot}5$ feet diameter $\times {\cdot}7854 \times 3$ feet $= 14{\cdot}726$ cubic feet piston-displacement; or, $2{\cdot}5 \times 2{\cdot}5$ feet diameter $\times {\cdot}7854 \times 3$ feet stroke $\times 2 \times 70 = 2061{\cdot}64$ cubic feet piston-displacement per minute.

The Pressure on a Piston may be found by this *Rule*:—

$$\text{Pressure on a piston in lbs. per square inch} = \frac{33000 \times \text{indicated horse-power}}{\text{area of cylinder} \times \text{speed of piston.}}$$

Example: The indicated horse-power of an engine is 180, the diameter of the cylinder is 17 inches, the length of stroke is 2 feet, and the number of revolutions per minute of the crank-shaft is 80. What is the pressure on the piston in lbs. per square inch?

Then $\dfrac{33000 \times 180 \text{ IHP}}{17 \times 17 \times {\cdot}7854 \times (2 \times 2 \times 80)} = 81{\cdot}8$ lbs. per square inch, pressure on the piston.

The Diameter of a Piston may be found, when the total pressure and the effective pressure per square inch is known, by the following *Rule*:—

$$\text{Diameter of piston in inches} = \sqrt[2]{\frac{\text{total pressure in lbs. on the piston}}{\text{effective pressure in lbs. per square inch} \times {\cdot}7854}}$$

Example: The effective pressure per square inch on the surface of a piston is 60 lbs., the total pressure is 7 tons. What is the diameter of the piston?

$$\text{Then } \sqrt[2]{\frac{7 \text{ tons total pressure} \times 2240 \text{ lbs.}}{60 \text{ lbs. effective pressure} \times \cdot 7854}} = 18 \cdot 248 \text{ inches diameter.}$$

Pistons.—The simplest form of piston is one fitted with Ramsbottom's spring-rings or packing-rings, as shown in Fig. 328. Three separate grooves are turned in the circumference of the piston, into which elastic rings of from $\frac{1}{4}$ to $\frac{3}{8}$ inch square steel or gun-metal are fitted. Each packing-ring is turned to a diameter a little larger than that of the cylinder, a short piece is cut out and the rings are sprung over the piston into the grooves, which are slightly deeper than the rings. The rings break joint, and the elasticity of the packing-rings maintains a nearly uniform outward pressure against the sides of the cylinder and renders the piston steam-tight.

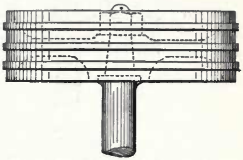
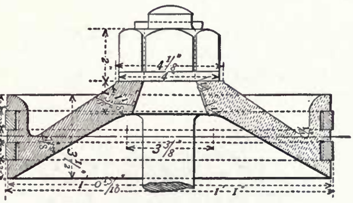


Fig. 328.—Piston with Ramsbottom's rings.

A Piston for a Locomotive-Engine Cylinder.—The pattern used on the London, Brighton and South Coast Railway is shown in Fig. 330. The body of the piston is made of gun-metal, and is cone-shaped, in

Fig. 329.

Fig. 330.



Figs. 329 and 330.—Piston of a locomotive-engine.

order to obtain the necessary strength with a light section. It is fitted with two cast-iron packing-rings, each $\frac{3}{4}$ inch wide, placed in separate grooves spaced $\frac{1}{2}$ inch apart. The piston fits on a steep cone, forged on the end of the piston-rod, shown in Fig. 329, so that it can be easily removed from the rod when required.

A Piston fitted with one broad packing-ring pressed outwards by spiral springs placed in holes radiating to the centre of the piston is shown in Figs. 331 and 332. The packing-ring shown in Fig. 332 is sprung over the body of the piston into its place.

A Piston, in which the packing-rings are expanded by a steel coil, is shown in Fig. 333. There are two packing-rings as shown, each having a flange; the flanges form a recess in which the coil is placed, and also act as abutments for the coil-ends. The packing-rings are pressed against the flanges of the piston, and also against the sides of the cylinder by the elasticity of the spring-coil, which is shown in Fig. 334.

A Piston provided with three packing-rings, two narrow outside rings

Fig. 331.

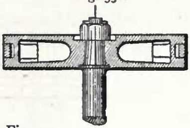
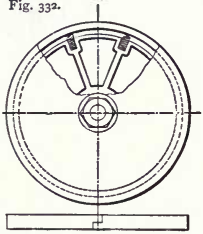


Fig. 332.



Figs. 331 and 332.—Piston with ring expanded by spiral-springs.

Fig. 333.

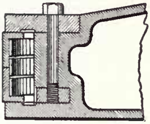


Fig. 333.—Section of piston, with Mather's steel-coil spring.

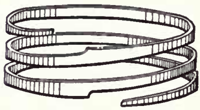


Fig. 334.—Piston-spring.

which fit the cylinder, and one broad inside ring, is shown in Fig. 335. A number of springs, shown in Fig. 336, are placed round the circumference of the piston, which abut on the body of the piston and press against the

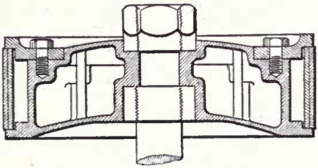


Fig. 335.—Piston of a stationary-engine.

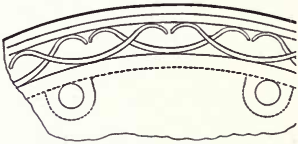


Fig. 336.—Piston-rings and springs.

inside ring, which expands and presses the outside rings against the sides of the cylinder.

A Piston provided with adjustable packing-rings is shown in Figs. 337 and 338. It is fitted with three packing-rings, two outside rings which fit the cylinder and break-joint, and one inside ring, which receives the pressure of a number of springs placed round the piston; each spring has a tongue-piece at the back, which fits into a recess, as shown. The springs are adjusted or set out to expand the packing-rings, by a gun-metal key driven into the recess behind the tongue-piece of each spring, which provides a solid abutment for the back of the spring, so that the pressure of the springs cannot vary when the piston is working. When this piston is used for a horizontal-engine, the bottom spring is removed and replaced by a metal block, which carries the weight of the piston. A guard-ring is fitted between the heads of the bolts, as shown, to prevent their working loose; the nuts of the bolts are gun-metal, let into the casting.

Fig. 337.

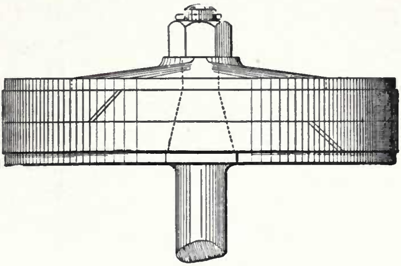
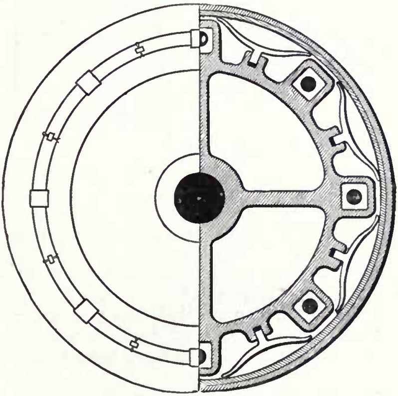


Fig. 338.



Figs. 337 and 338.—Piston with adjustable rings.

A Piston in which the packing-rings are expanded by a spiral-spring, which acts against two packing-rings, each having a flange, the spring being placed in a recess formed by the flange of each packing-ring, is shown in Fig. 339. The packing-rings are pressed against the sides of the cylinder, and also against each flange of the piston, by the elasticity of the spring, which is clearly shown in Figs. 340 and 341. The bolts of the piston-cover are screwed into gun-metal nuts let into the casting, and a guard-ring is fitted between the heads of the bolts to prevent their working loose.

To prevent Steam escaping past the joint of Packing-Rings a plate is inserted as shown in Fig. 342, the joint being covered by another, or inside ring. Another way of preventing the escape of steam, suitable for single rings, is shown in Fig. 343, the joint being covered inside the ring by a plate having a tongue-piece which fits into a slot in the piston-ring as shown, the screw-holes are slotted on one side of the joint to allow the ring to expand.

Packing-Rings for Pistons are turned to a diameter equal to $\frac{1}{8}$ inch larger than the cylinder for every foot in diameter of the cylinder, this being necessary in order to make them spring out and fill the cylinder, a piece being afterwards cut out of the ring to make it the proper size. The ring is cut in an oblique direction, as shown in Fig. 342, in order to prevent the ends, or joint, scoring the cylinder. The length of piece required to

Fig. 339.

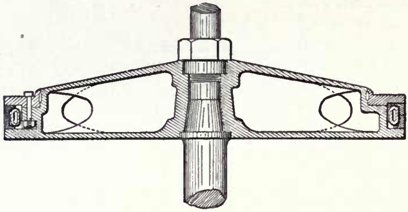


Fig. 340.

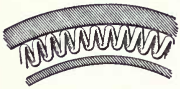
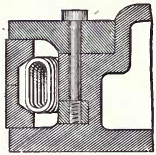


Fig. 341.



Figs. 339—341.—Piston with Buckley's spring.

Fig. 342

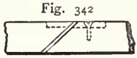
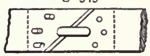


Fig. 343.



Figs. 342 and 343.—Joints of piston-rings.

be cut out of the ring, is equal to the difference between the external circumference of the ring and the circumference of the cylinder.

Example: A packing-ring for a piston is $60\frac{5}{8}$ inches external diameter before being cut, the diameter of the cylinder is 60 inches. How much should be cut out of the circumference of the ring to make it exactly fit the cylinder when sprung in?

Then $(60{\cdot}625 \times 3{\cdot}1416) - (60 \times 3{\cdot}1416) = 1{\cdot}9635$ inches.

Or $60{\cdot}625 - 60 = {\cdot}625 \times 3{\cdot}1416 = 1{\cdot}9635$ inches.

The Diameter of a Piston-Rod may be found by the following formulæ:—

Let D = the diameter of cylinder in inches.

P = $\left\{\begin{array}{l}\text{the initial absolute pressure of the steam in lbs. per square}\\ \text{inch, or the pressure shown by the steam-gauge plus 15 lbs.}\end{array}\right.$

d = the diameter of the piston-rod in inches.

s = the diameter of the screwed end of the piston-rod.

To find the diameter of the piston-rod, d:—

$$\text{Single piston-rods, } d = \frac{D}{60}\sqrt[2]{P}.$$

$$\text{Double piston-rods, } d = \frac{D}{80}\sqrt[2]{P}.$$

To find the diameter of the screwed end of the piston-rod, s:—

$$\text{Single piston-rods, } s = \frac{D}{85} \sqrt[2]{P.}$$

$$\text{Double piston-rods, } s = \frac{D}{120} \sqrt[2]{P.}$$

Example: Required the diameter of a piston-rod for a cylinder 60 inches diameter, the initial pressure of steam in the cylinders being 65 lbs. per square inch. If the piston be fitted with two piston-rods, what diameter should they be?

Then 65 lbs. initial pressure + 15 lbs. = 80 lbs. absolute pressure of the steam.

And $\frac{60 \text{ inches diameter}}{60 \text{ constant}} \times \sqrt[2]{80 \text{ lbs.}} = 8{\cdot}944$ inches diameter of single piston-rod.

$\frac{60 \text{ inches diameter}}{85} \times \sqrt[2]{80 \text{ lbs.}} = 6{\cdot}413$ inches diameter of the screwed-end of single piston-rod.

If the piston be fitted with two piston-rods, the diameter will be:—

$\frac{60 \text{ inches diameter}}{80} \times \sqrt[2]{80 \text{ lbs.}} = 6{\cdot}708$ inches diameter of each double piston-rod.

$\frac{60 \text{ inches diameter}}{120} \times \sqrt[2]{80 \text{ lbs.}} = 4{\cdot}472$ inches diameter of the screwed end of each double piston-rod.

The Taper of the Cone on the Piston-rod, on which the piston is fitted, should not be less than 1 inch per foot, and there should be either a shoulder or collar on the rod at the bottom of the cone, to prevent splitting of the piston, and ensure its easy removal when required.

To Prevent the Nut which secures the Piston on the Piston-rod working loose, it may be fitted in an octagonal recess in the piston, equal in depth to one-fourth that of the nut, having the space between the sides of the nut and of the recess filled up with white-metal.

The Strain on a Piston-Rod may be found by dividing the maximum strain on the piston per square inch of its area, by the sectional area of the piston-rod, in inches.

Example 1: The diameter of a cylinder is 33 inches. The diameter of the piston-rod is 5·75 inches. The maximum working-pressure of the steam on the piston is 90 lbs per square inch. Required the tensile strain per square inch on the piston-rod.

Then 33×33 inches $\times {\cdot}7854 \times 90$ lbs. = 7697·7 lbs. the strain per square inch of the area of the piston?

And $\frac{7697{\cdot}7 \text{ lbs. strain on the piston}}{5{\cdot}75 \times 5{\cdot}75 \text{ inches} \times {\cdot}7854} = 2964{\cdot}3$ lbs. per square inch, the strain on the piston-rod.

Example 2: The diameter of a piston-rod at the smallest part is $\frac{1}{12}$ of the diameter of the piston. What is the tensile strain per square inch on the piston-rod when the pressure on the piston is 60 lbs., the back-pressure is $3\frac{1}{2}$ lbs., and the diameter of the body of the piston-rod is $\frac{1}{8}$ the diameter of the piston?

Then, if the diameter of the piston be supposed to = 12 inches, the least diameter of the piston-rod will = 1, and the diameter of the body of the piston-rod = $\frac{1}{8}$ of 12 inches = $1\frac{1}{2}$ inches.

And $\dfrac{12^2 - 1{\cdot}5^2 \times {\cdot}7854 \times (60 - 3{\cdot}5 \text{ lbs.})}{1^2 \times {\cdot}7854} = 800{\cdot}87$ lbs. per square inch, the tensile strain on the piston-rod.

Metallic-Packing for Piston-Rod and other Glands.—The glands of triple-expansion and other engines using steam of very high pressure

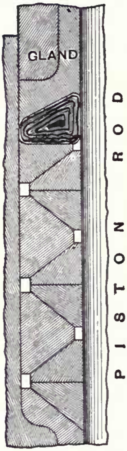


Fig. 344.—Metallic-packing.

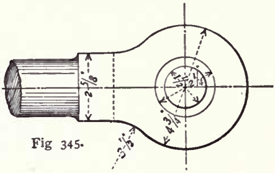


Fig 345.

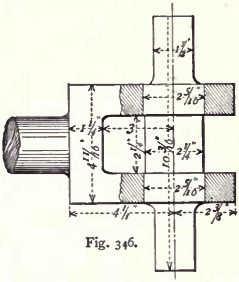


Fig. 346.

Fig. 345 and 346. —Locomotive-engine crosshead.

require to be packed with metallic-packing, as ordinary packing rapidly becomes charred by the high temperature of the steam. A simple and very efficient arrangement of metallic-packing is shown in Fig. 344. It consists of a number of V-shaped rings of anti-friction metal, in two pieces, arranged one above the other as shown, and tightened by a gland fitted to the stuffing-box with studs and nuts in the same way as for hemp packing.

The following is a good mixture of anti-friction metal for the packing-rings.

Tin	82	parts.
Lead	14	"
Copper	4	"

A piece of round canvas-packing, lubricated with a mixture of plumbago and tallow, is placed above the metallic-packing at the top of the stuffing-box, in order to prevent the escape of any steam which might leak through the joints of the metallic-rings, and also to give elasticity to the packing.

The Crosshead of a Locomotive-Engine is either forged on the piston-rod, as shown in Figs. 345 and 346, or the end of the piston-rod is tapered into, and secured to the boss of a separate crosshead with a cotter, as shewn in Fig. 347.

The boss of the crosshead, in Fig. 347, is bored conically to a depth of

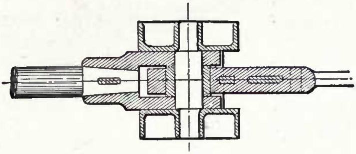


Fig. 347.—Crosshead and slide-blocks of a locomotive-engine.

$5\frac{3}{4}$ inches, to fit the tapered end of the piston-rod, the large end of the hole is $2\frac{7}{8}$ inches diameter, and the taper being 1 inch in 11 inches, the diameter of the small end, or bottom of the tapered hole, is $2 \cdot 875 - \frac{5 \cdot 75 \times 1}{11 \text{ inches}} = 2 \cdot 353$, or say $2\frac{3}{8}$ inches.

The Strength of the Cotter of the Crosshead shown in Fig. 347, may be ascertained as follows. The cotter is made of steel, $\frac{5}{8}$ inch thick, and $2\frac{1}{4}$ inches wide at the middle, and being in double-shear it has a shearing section of $\cdot 625$ inch thick $\times\ 2 \cdot 25$ inches wide $\times\ 2 = 2 \cdot 8125$ square inches for resisting a maximum working pressure of steam of, say, 140 lbs. per square inch on the piston. If the diameter of the piston be 17 inches, the area will be $17 \times 17 \times \cdot 7854 = 226 \cdot 98$ square inches, and the maximum strain will be $\frac{226 \cdot 98 \text{ square inches} \times 140 \text{ lbs.}}{2240 \text{ lbs.}} = 14 \cdot 186$ tons per square inch, equal to $14 \cdot 186 \div 2 \cdot 8125$ square inches, the shearing section of the cotter, $= 5 \cdot 008$, or say 5 tons strain per square inch of section of the cotter. Taking the ultimate shearing strength of the steel at 30 tons per square inch, the strength of the cotter will be $= 30 \div 5 = 6$ times as great as the maximum working stress upon it. The piston-rod is $2\frac{7}{8}$ inches diameter, made of steel, and it has to resist a maximum working strain of

$$\frac{14 \cdot 186 \text{ tons}}{2 \cdot 875 \times 2 \cdot 875 \times \cdot 7854} = 2 \cdot 185 \text{ tons per square inch.}$$

The Strength of the Crosshead-Pin may be ascertained in a similar way to the above. The crosshead-pin shown in Fig. 347 is $2\frac{3}{4}$ inches diameter between the jaws of the crosshead, at the bearing for the connecting-rod, and the length of the bearing is $2\frac{7}{8}$ inches. The diametral sectional-area being $2 \cdot 75$ inches diameter $\times\ 2 \cdot 875$ inches length $= 7 \cdot 857$ square inches and the pressure on the crosshead-pin is $14 \cdot 186$ tons $\div\ 7 \cdot 857$ square inches $= 1 \cdot 806$ tons per square inch of diametral section of the crosshead-pin bearing.

A Crosshead for a Horizontal Stationary-Engine is shown in Fig.

348, the pin projects on each side of the crosshead and carries the slide-blocks, which work between flat slide-bars, the wear being taken up by adjusting the pillars between the ends of the bars.

A Crosshead for a stationary-engine of another design, is shown in Figs. 349–351. The slide-bars are concave, the sliding surfaces of the slide-blocks being convex. A cotter is fitted lengthways under the slide-block, as shown, adjustable by nuts to take up the wear.

A Crosshead for a stationary-engine of another

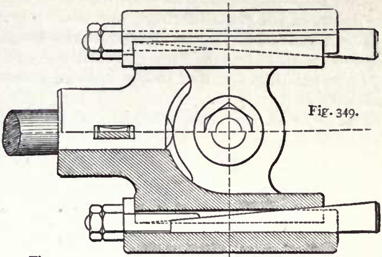


Fig. 349.

Fig. 350.

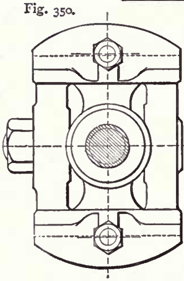


Fig. 351

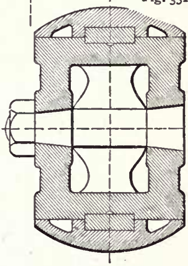


Fig. 348.

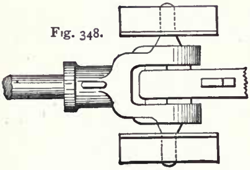


Fig. 348.

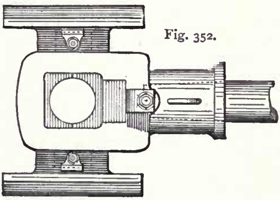


Fig. 352.

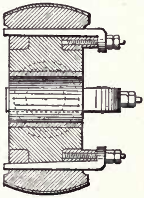


Fig. 353.

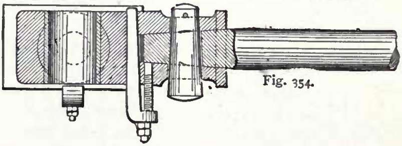


Fig. 354.

Figs. 348—354.—Crossheads of stationary-engines.

design is shown in Figs. 352–354. The crosshead-pin is fixed in the forked-end of the connecting-rod, and the pin works in bushes in the crosshead, which are adjustable by a cotter, the sliding surfaces of the slide-blocks are convex, and they work between concave slide-bars, each slide-block being adjustable by a cotter fitted crossways as shown.

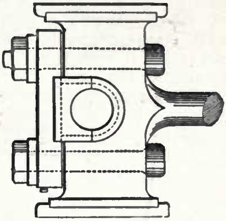


Fig. 355.—Crosshead of a marine-engine.

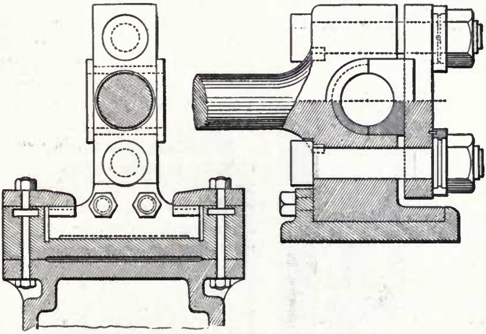
A Crosshead for a Marine-Engine, forged solid on the end of the piston-rod, is shown in Fig. 355. The crosshead-pin is fixed in the forked-end of the connecting-rod, and works in bushes in the crosshead, which are secured by a cap and bolts, as shown.

The diameter of the crosshead-pin may be = diameter of piston-rod × 1·18 to 1·25, and its length may be = diameter of piston-rod × 1·5 to 1·75.

The diameter of each bolt for the cap may = diameter of piston-rod × ·55 to ·6. Thus for a piston-rod 7 inches diameter, the diameter of each bolt for the cap would be = 7 × ·57 = 4 inches: the ratio of the area of the cap-bolts to the area of the piston-rod would be = (7 × 7 inches) ÷ (4 × 4 × 2 bolts) = 1·531 and the ratio of the area of the piston-rod to that of the bolts would be (4 × 4 × 2 bolts) ÷ (7 × 7 inches) = ·653.

Marine-Engine Crossheads vary considerably in design, a few of the most generally used designs are given in Figs. 356–361, in which different

Fig. 356. Fig. 357.



Figs. 356,—357.—Crosshead of a marine-engine.

methods are shown of attaching the shoe to the crosshead. The area of the shoe is frequently made = one-tenth that of the piston.

Then 60 × 60 × ·7854 = 2827·44 square inches, the area of the piston.

2827·44 ÷ 10 = 282·744 square inches, the area of the shoe.

282·744 ÷ 15 inches = 18·85 inches, the length of the shoe.

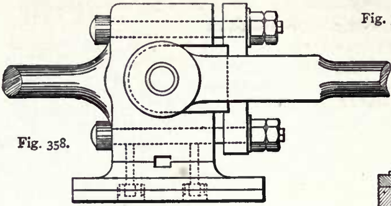


Fig. 358.

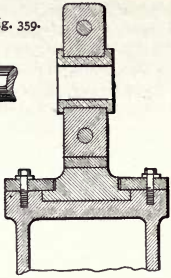


Fig. 359.

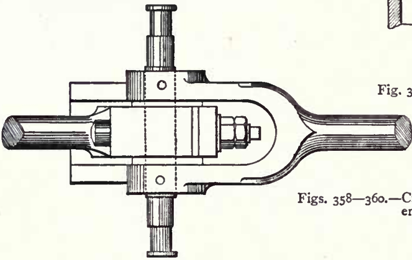


Fig. 360.

Figs. 358—360.—Crosshead of a marine-engine.

The Shoes of the Crossheads of high-speed engines should have strips of anti-friction metal let into their sliding surfaces. A good anti-friction metal for this purpose is composed of tin 16 parts; copper $1\frac{1}{2}$ parts; antimony 2 parts.

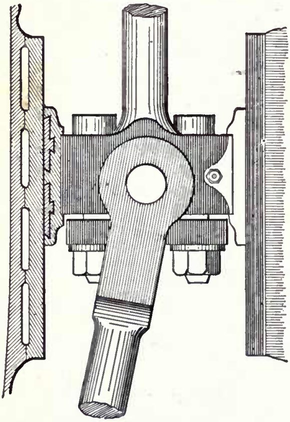


Fig. 361.—Crosshead of a marine-engine.

Mill-Engines driving spinning machinery require to run with the utmost uniformity of velocity, because both the quality and quantity of the products of a factory are considerably increased by an exact and unvarying engine-speed, therefore uniformity of turning-effort is the first consideration, and economy of fuel the second. The variation in the speed of best modern mill-engines does not exceed from one quarter to one half per cent., and the consumption of coal averages from $1\frac{3}{4}$ to 2 lbs. per indicated horse-power per hour of actual work, and from $2\frac{1}{2}$ lbs. to $2\frac{3}{4}$ lbs. per indicated horse-power per hour, including the coal used in getting up steam, in keeping it up during meal-times, and in heating the mill when required.

The types of steam-engines used for driving modern factories are —The horizontal or vertical simple engine of high speed, using steam of from 60 to 80 lbs. pressure per square inch. The horizontal or vertical compound condensing engine, with a high-pressure cylinder and a low-pressure cylinder coupled to one shaft with their cranks at right angles, so that when one crank is on its dead-centre, the other crank is in the best position for rotating the shaft. The horizontal or vertical single compound condensing tandem-engine, that is, with the pistons of both cylinders attached to one piston-rod. The horizontal or vertical double compound tandem-engine, that is, a pair of single compound tandem-engines coupled to one shaft with their cranks at right angles.

A Horizontal Stationary Non-Condensing Simple Steam-Engine is shown in Fig. 362. It has a high-speed governor, an equilibrium throttle-

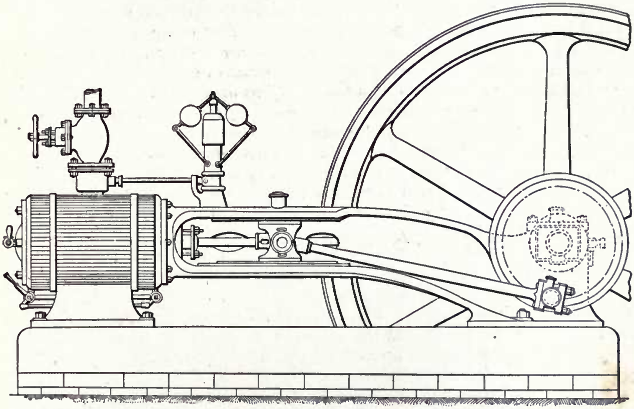


Fig. 362.—Horizontal stationary non-condensing engine.

valve, and wide bearings; the frame is girder-shaped of a depth equal to the diameter of the cylinder. The crank is cast on a disc having a counter-weight opposite to the crank, to counter-balance the rotating parts, the weight of the counter-weight is equal to the sum of the weights of the crank and pin, and the weight of half the length of the connecting-rod next the crank-pin. The momentum of the piston and other reciprocating parts is balanced by compression of steam in the cylinder.

An engine balanced in this manner may be run at a high speed without vibration, and a high speed is conducive to uniformity of velocity, as the efficiency of a fly-wheel in producing regularity of motion varies, other things being equal, as the square of its angular velocity. An engine fitted with a slide-valve and controlled by a throttle-valve, works with considerable variation in speed, and is not so suitable for driving spinning-machinery as an engine fitted with Corliss valve-gear, in which the steam is controlled automatically by a high-speed governor, with the slightest possible variation in speed.

Corliss Engines are largely used as mill-engines. The cylinder of a horizontal Corliss engine is shown in Fig. 363. It is a built-up-cylinder on a simple and effective plan, with separate end-valve chambers, separate internal cylinder-liner and jacket-casing. The Corliss-valves are shown in position; the two top valves are the steam admission-valves; the steam ports and passages are made very short and direct, to avoid undue clearance-spaces. The lower valves are the exhaust-valves, and, as they are placed below the cylinder they drain it thoroughly, which is an advantage of considerable importance. This method of building a cylinder enables the cylinder-liner to be made a simple casting of sound, hard, close metal. The steam and exhaust-valves are shown separately in section, with their spindles. The piston, piston-rod and crosshead are shown in their place. The engine-frame is bolted to the front of the cylinder, so that it takes the strain direct and in the best possible manner. The valves are actuated by a central disc or wrist-plate, the steam-valves are arranged with liberating gear to cut off the steam instantaneously, the point of cut-off being controlled by the main-governor. The engine is also fitted with one of Knowles' supplementary governors, by which an almost uniform speed can be maintained, even with considerable variation in the steam-pressure, or in the load, an important advantage in spinning mills. This governor has a friction-pulley, fitted between two flanges on its socket. At the normal speed of the engine neither of the flanges touches the friction-pulley, but should any increase or decrease take place, one or other of the flanges presses upon and causes the friction-pulley to rotate in one direction or the other, which operates by means of cord-pulleys, one of which is fixed on a nut which moves a vertical rod, in two pieces, one piece being screwed with a right-hand and the other with a left-handed thread, attached to the forked-lever of the main-governor. The clearance between the flanges and the friction-pulley is so small, that the slightest variation in speed brings the supplementary-governor into operation, which is thus continuously resisting, automatically, any tendency to change of speed in the engine. The consumption of steam in Corliss-engines is usually from 17 to 18 lbs. per indicated horse-power, corresponding to about 2 lbs. of coal per indicated horse-power per hour. In a test of one of these engines, having a single-cylinder of 52 inches diameter, 6 feet stroke, making 60 revolutions per minute, or a piston-speed of 720 feet per minute, with a vertical air-pump 36 inches diameter, and 30 inches stroke and condenser: it drove by belts from the fly-wheel about 1800 indicated horse-power. The steam used per indicated horse-power was 17·45 lbs., and the consumption of coal was 2·14 lbs. per indicated horse-power per hour.

Automatic Barring-Engine.—Mill-engines frequently stop in such a position that it is necessary to turn them partly round by means of a bar before they can be re-started, and it is sometimes necessary to bar them round slowly when putting on belts or doing repairs in the factory. It being difficult to bar large engines by hand-power, a barring-engine, shown in Fig. 364, is used for this purpose, which is provided with a simple arrangement for automatic disengagement when the engine begins to run

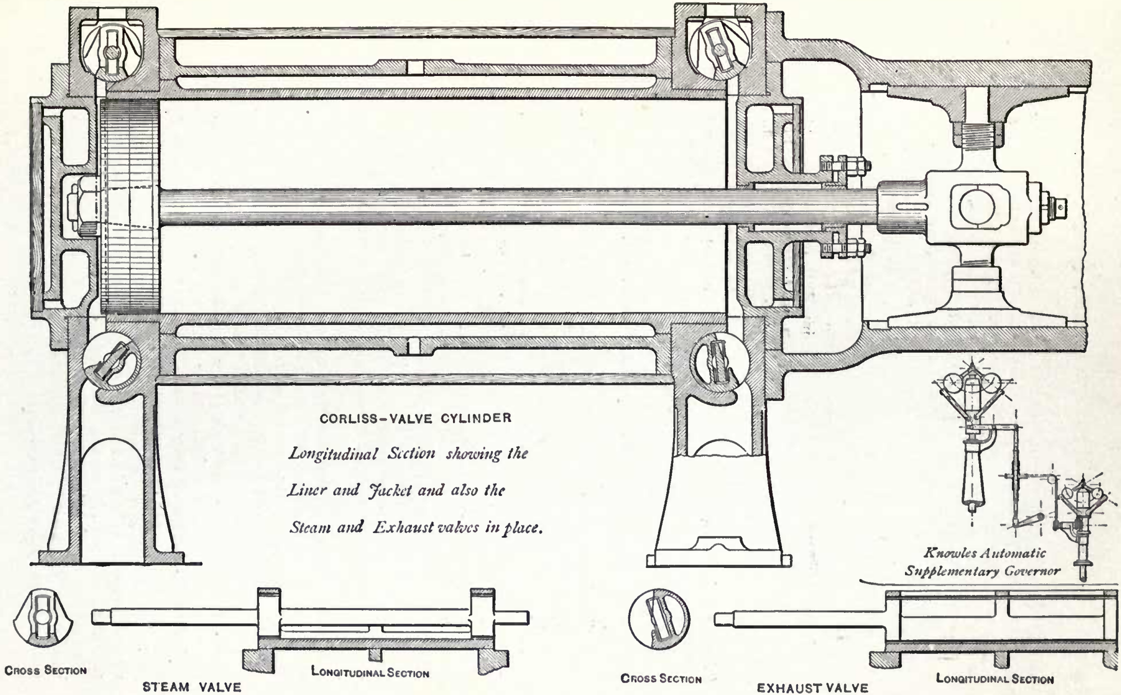


Fig. 363.—Cylinder and valves of a Corliss-engine, by Hick, Hargreaves and Co., Bolton.

quicker than the barring-engine. The pinion which acts as driver to the fly-wheel is shaped as a worm-wheel on one side of the tooth and as a spur-wheel on the other, to gear with an internal toothed-wheel on the fly-wheel, the motion being communicated to it by a worm fixed on the crank-shaft of the barring-engine. The pinion is keyed on to a shaft which works in a slot, thus allowing a certain lateral motion. When it is desired to throw the pinion into gear a brake is held against the bottom edge by a lever, and the result is that the pinion is thrown into gear by the action of the worm, the point where the brake is applied becoming the fulcrum. As soon, however, as the fly-wheel becomes the driver the action is reversed, the worm becomes the fulcrum and the pinion falls back automatically, its axis moving along the slot referred to, and the wheel is retained in its out-of-gear position by a spring. This is a perfectly safe and automatic arrangement, and is a compact and well-contrived device with a small amount of mechanism.

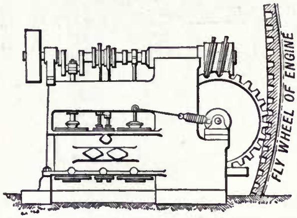


Fig. 364.—Barring-engine, by Hick, Hargreaves and Co., Bolton.

A Vertical Tandem Compound Mill-Engine is shown in Fig. 365. The high-pressure cylinder is 14 inches diameter, the low-pressure cylinder is 24 inches diameter, and the length of stroke is 2 feet, when running at 80 revolutions per minute it develops 120 indicated horse-power with steam of 100 lbs. per square inch working-pressure. The high-pressure cylinder is rigidly supported by two wrought-iron pillars and the rectangular pipe connected to the low-pressure cylinder. The high-pressure cylinder is fitted with Corliss-valves and the low-pressure cylinder with a slide-valve, the spindle of which is carried through the upper end of the casing to work the Corliss exhaust-valves. The air-pump and feed-pump are worked by a lever coupled to the crosshead. One of the frames acts as a condenser: an air space is left between the frame and the slide-plate to prevent the slide-block heating.

Non-Condensing Simple Engines frequently discharge the exhaust-steam into the atmosphere before it has been thoroughly deprived of all its available heat and power. In order to expand the steam in the most economical manner and leave it no capacity for developing further power, and also to equalise the strains and secure greater uniformity of turning-effort, non-condensing engines are sometimes compounded, and have their cranks placed at right angles.

A Non-Condensing Compound Engine, compact in design, accessible to inspection, occupying small space, and requiring only a slight foundation, is shown in Fig. 366. The frame or bed of the engine is formed of two strong wrought-iron girders, firmly braced together by cross-stays. The cylinders are bolted, at the bottom to the frame, and at the top to the smoke-box of the boiler. The crank-shaft is of steel, bent from a single bar; it is balanced, and carries a heavy fly-wheel. The cylinders are each formed by a separate barrel of hard metal forced into the main casing,

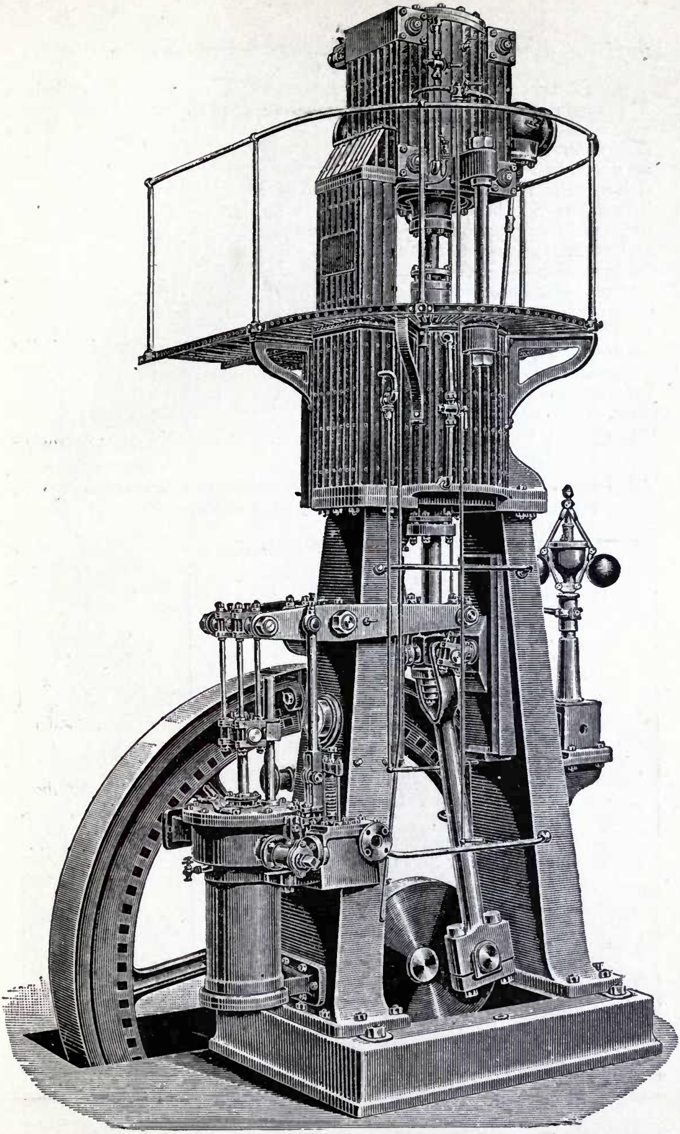


Fig. 365.—Vertical tandem compound mill-engine, by Douglas and Grant, Kirkcaldy.

a space between the barrels forms jackets which surround the cylinders. The steam is admitted to the high-pressure cylinder by a double-ported expansion-valve working on the back of the main slide-valve, and automatically controlled by the governor, so as to vary the admission from any point up to one half the stroke, according to the power required. The expansion-valve is very prompt in action, and is actuated in a simple manner by a radius-rod working in a rocking slot-link driven by a separate eccentric, the raising or lowering of the governor-balls, by changing the position of the radius-rod in the slot-link, shortens or lengthens the travel of the expansion-valve, causing it to cut-off steam earlier or later as required. The governor shown in Fig. 367 is of the crossed-arm type with heavy balls and is very sensitive. An oil-cylinder is provided to prevent any tendency to "dance," the piston of which has a simple arrangement for regulating the flow of oil, and consequent promptness of movement.

The steam is usually cut-off in the high-pressure cylinder when the piston has travelled nearly one half its stroke, and is expanded a little more than twice, it then passes into the low-pressure cylinder, which has two and one-half times the capacity of the high-pressure cylinder, where it is expanded

Table 95.—Results of Trials of a 12 Nominal Horse-Power Non-Condensing Compound Engine.

	1st Trial.	2nd Trial.	3rd Trial.
Working boiler-pressure maintained	120 lbs.	120 lbs.	120 lbs.
Average revolutions per minute of the engine	129	130·8	129·7
Average piston speed per minute	301 ft.	305·2 ft.	302·6 ft.
Effective horse-power on brake	29·64	30·26	29·92
Kind of coal used	Welsh.	Welsh.	Welsh.
Amount consumed during the trial	168 lbs.	300 lbs.	300 lbs.
Amount consumed per brake horse-power per hour	2·54 lbs.	2·63 lbs.	2·7 lbs.
Amount of feedwater supplied	1330 lbs.	2328 lbs.	2408 lbs.
Amount of feedwater per brake horse-power per hour	20·09 lbs.	20·46 lbs.	21·63 lbs.
Temperature of water as supplied	56° F.	57° F.	55·4° F.
Temperature of water forced into the boiler	155° F.	151° F.	152·8° F
Increase of temperature from water-heater	99° F.	94° F.	97·4° F.
Amount of exhaust-steam condensed per hour per horse-power	1·97 lbs.	1·93 lbs.	2·12 lbs.
Steam (exclusive of that condensed in the jackets) used per hour per brake horse-power	22·06 lbs.	22·49 lbs.	23·75 lbs.
Water evaporated per hour per pound of coal from feed temperature	8·68 lbs.	8·49 lbs.	8·69 lbs.

nearly four times, and finally escapes to the chimney, reduced by a six-fold expansion to a pressure of 8 lbs. per square inch above the atmosphere, or at a final absolute pressure of 23 lbs. per square inch.

Several trials were made of one of these engines, of 12 nominal horse-power,

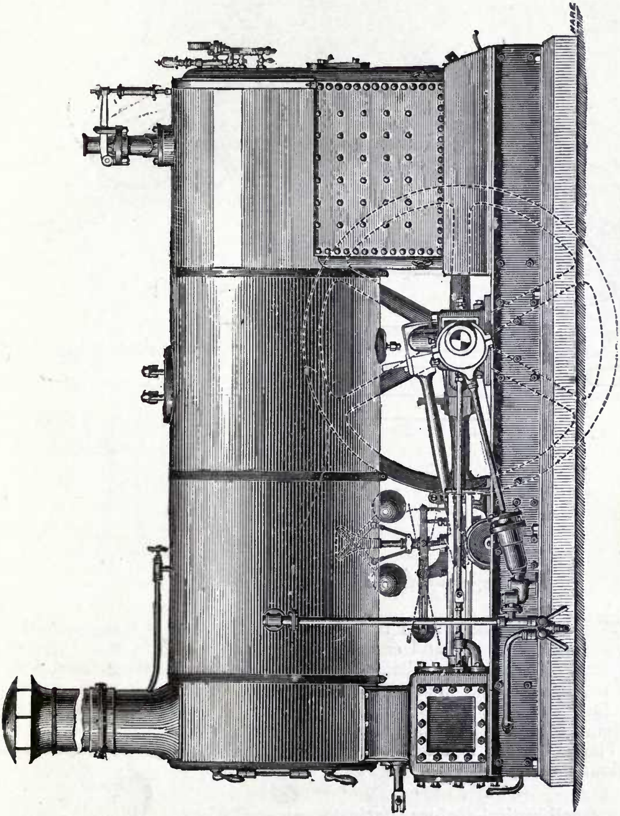


Fig. 366.—Non-condensing compound-engine, by Ruston, Proctor and Co., Lincoln.

with a high-pressure cylinder 7 inches diameter, and a low-pressure cylinder 11 inches diameter, length of stroke of both cylinders 14 inches. The results of the trials are given in Table 95.

The consumption of 2·63 lbs. of coal per effective horse-power per hour on the second trial was equal to about 2·4 lbs. per indicated horse-power per hour, an extremely low consumption for a small non-condensing steam-engine; the saving effected being about 40 per cent. over a non-condensing simple engine.

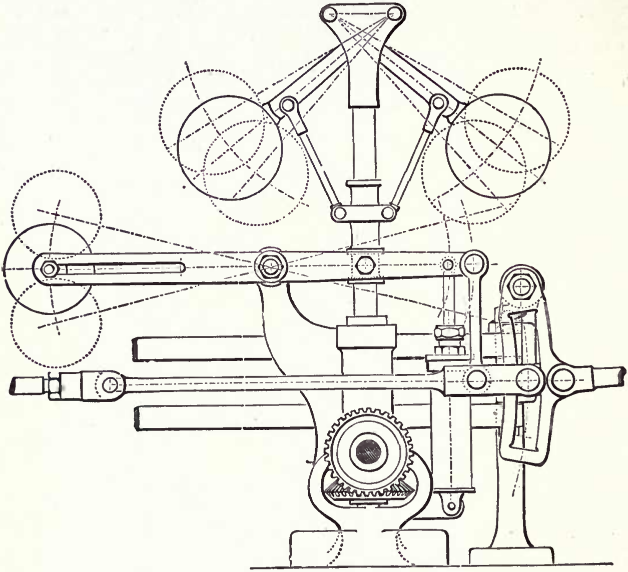


Fig. 367.—Automatic expansion-gear of the engine shown in Fig. 366.

Marine-Engines are of two classes, engines driving screw-propellers called screw-engines, and engines driving paddle-wheels, called paddle-engines. The chief consideration in a marine-engine is, the greatest power on the least weight, in the smallest space, from the smallest consumption of fuel. The weight of steam-engines in comparison with the power developed varies considerably in different classes of engines.

The weight of steam-engines per indicated horse-power averages as follows:—

	Weight per indicated horse-power in lbs.
Small vertical-engines attached to vertical boilers	800
Stationary-engines, condensing	750
Stationary-engines, non-condensing, strong	700
Portable-engines	620

	Weight per indicated horse-power in lbs.
Stationary-engines, non-condensing, light	560
Compound-engines of merchant-steamers	480
Triple-expansion-engines of merchant-steamers . . .	450
Royal navy engines, with natural draught	360
Special engines for light-draught vessels	280
Royal navy despatch vessels, *Surprise* and *Alacrity* class .	240
Royal navy engines, with forced draught	200
Locomotive-engines, heavy class	200
Locomotive-engines, light class	150
Torpedo-boat engines	60

These weights include that of pipes, fittings, and water in the boiler, that is, all in working condition.

A Pair of Compound Surface-Condensing Screw-Engines of 35 nominal horse-power, of a type much used for yachts, cargo-vessels, and tugs, is shown in Fig. 368. The high-pressure cylinder is $14\frac{1}{2}$ inches diameter, the low-pressure cylinder is 28 inches diameter, and the length of stroke of both pistons is 22 inches. The surface-condenser forms the base of the back-columns, being cast in one piece with them. The air-pump, circulating-pump and bilge-pumps are worked by rocking-levers from the cross-head of the low-pressure cylinder. The reversing-gear is link-motion, and being of a size easily handled, the position of the link is controlled by a hand-lever. The design of the engine is compact and substantial, and it admits of easy access to all the working parts. The vessel for which the engines were made is 95 feet long, 18 feet beam, and 9 feet moulded depth.

A Set of Triple-Expansion Surface-Condensing Screw-Engines of 1200 indicated horse-power is shown in the frontispiece. The cylinders are three in number, of hard close-grained cast-iron. The first, or high-pressure cylinder, is 22 inches diameter, the second, or intermediate cylinder, 35 inches diameter, and the third, or low-pressure cylinder, is 58 inches in diameter, all with 42 inches stroke, and fitted with escape-valves, indicator-cocks, and drain-cocks, with handles to starting-platform. The high-pressure cylinder is steam-jacketed by means of a liner, and all cylinders are neatly covered with felt and teak secured by brass screws. The first and second cylinder valve-chests are in line with the cylinders. The low-pressure cylinder valve-chest is in front of the engines. The first, or high-pressure cylinder, has piston-valves fitted with loose liners, the second and third cylinders have ordinary D valves. A valve is fitted so as to admit steam to the steam-chest of the second and third cylinders, its handle being reached from the starting-platform. A relief-valve is fitted to the steam-chests of the second and third cylinders to prevent the possibility of high-pressure steam being let into these cylinders.

The valves are of hard cast-iron. A piston-valve is fitted to the first cylinder, a single ported-slide-valve to the second cylinder, and a double ported-slide-valve to the third cylinder.

The pistons are of cast-iron. The piston of the first cylinder is fitted with three cast-iron Ramsbottom-rings. The pistons of the second and third cylinders have ordinary cast-iron packing-rings, with junk-rings of cast-iron, held down by wrought-iron T-headed bolts and brass nuts, so arranged that the bolts and nuts can be easily taken out without drawing the piston; the junk-ring bolts have patent guards.

The piston-rods are of hammered-iron secured to the pistons by nuts, with solid forged-heads fitted with adjustable cast-iron shoes for working on the guides. The recesses in the piston-rod-heads for the gudgeon-brasses are flat-bottomed for convenience in lining-up; the piston-rods are duplicates of each other.

The guides are bolted to columns on the condenser, and are provided with oil-boxes and lubricators.

The connecting-rods are forged of best scrap-iron with solid double-eye and gudgeon of large size at the top end, and are fitted with flat brasses at the bottom end. The brasses are lined with white-brass. The length of the connecting-rod is equal to $2\frac{1}{4}$ times the length of the stroke.

The crank-shaft makes 70 revolutions per minute. It is $11\frac{1}{2}$ inches diameter, built with three double-cranks of iron, with forged-steel pins set at angles of 120°, constructed in two pieces coupled in the centre-crank, and fitted with the pin bolted to each part. The shaft is interchangeable.

The shafting is of the best hammered-iron, having solid couplings forged upon each piece, and properly jointed with turned bolts. The intermediate lengths are $10\frac{3}{4}$ inches diameter in the body, and 11 inches diameter in the bearings. The propeller-shaft is $11\frac{1}{2}$ inches diameter, lined-up with brass in way of lignum-vitæ bush and stuffing-box.

The valves are worked by link-motion, of which all the principal bearings are made adjustable for wear.

The eccentric-sheaves are of cast-iron, and the eccentric-straps are of solid gun-metal. The studs for securing the eccentric-rods to the eccentric-straps are left $\frac{3}{4}$ inch above the top nut to allow for adjustment of the valves.

The starting and reversing-gear is very powerful, and is conveniently arranged with regard to the working-platform, which is on a level with the engine bedplate.

The air-pump is single acting, 18 inches diameter and 18 inches length of stroke, of solid brass fixed in a cast-iron casing. The bucket, foot and delivery valve-seats are of brass, fitted with metallic-valves. The pump-rod is of iron cased with brass.

The circulating-pump is double-acting, 12 inches diameter, 18 inches length of stroke, of solid brass fixed in a cast-iron casing, and arranged to force the water through the condenser. The bucket, foot-valve and delivery-valve seats are of brass with india-rubber valves. The pump-rod is of iron cased with brass. The pump is arranged to draw from the sea or bilge.

The air-pump and circulating-pump-levers are of forged-iron fitted to a weigh-shaft working in pillar-blocks having adjustable brasses. The levers are worked off the after-engine, and are fitted to the connecting-rod-gudgeon and pump-crosshead by links having adjustable brasses.

The feed-pumps are two in number, each $3\frac{1}{2}$ inches diameter and 18 inches length of stroke, of cast-iron with brass rams, worked from the air and circulating-pump crosshead, each pump is capable of supplying the boiler with water when the engines are working at full speed. They are fitted with brass suction and delivery-valves, and escape-valves of brass with steel spiral springs; each pump is capable of being worked independently of the other.

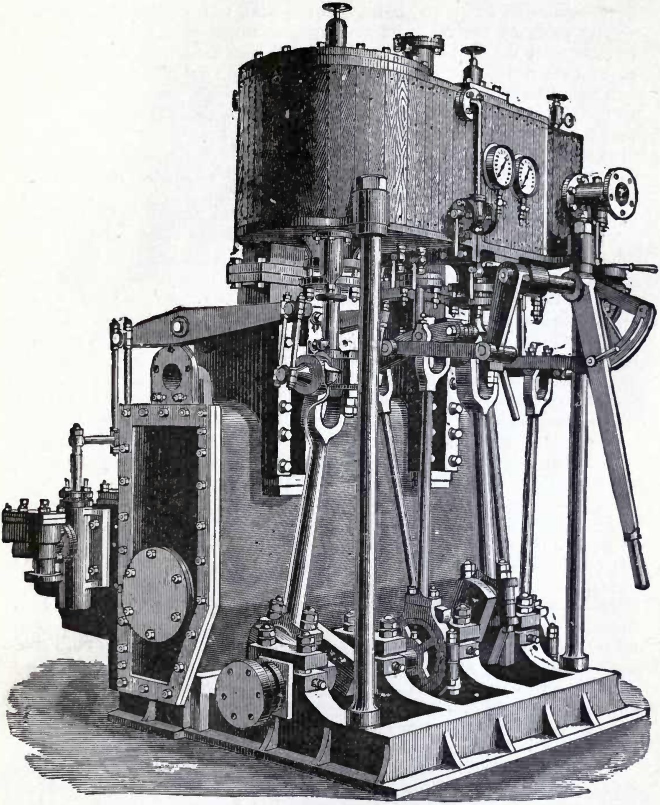


Fig. 368.—Compound surface-condensing marine-engines, by Ross and Duncan, Glasgow.

The bilge-pumps are worked from the air-pump and circulating-pump crosshead, and are similar in construction to the feed-pumps, but larger in size, being $4\frac{1}{2}$ inches diameter and 18 inches length of stroke, and are fitted with cast-iron rams and brass suction and delivery-valves and seats.

All the steam-pipes, and pipes under pressure, are made of copper, of thickness varying from 3 to 14 wire-gauge. The bilge-pipes are of lead.

The bed-plate is of ample strength, and is provided with recesses having square bottoms to receive the crank-shaft brasses; the brasses are held down by wrought-iron keeps and bolts, and are fitted with syphon-lubricators.

The columns are made of cast-iron, of ample strength to support the cylinders on the starboard side; there are also hollow cast-iron standards cast on the condenser to support the cylinders on the port side.

The surface-condenser is placed on the port side; it has a cooling-surface of 1500 square feet in 582 brass-tubes of $\frac{3}{4}$ inch external diameter; all the joints between the cylinders, framing, condenser, air-pumps and bed-plate are carefully faced and firmly secured. The tubes are placed horizontally; they are packed with wood ferrules in brass tube-plates. A soda-cock is fitted. The engines are provided with a governor.

The steam drives the pistons of the first and second cylinders during 60 per cent. of the stroke, and the piston of the third cylinder during 55 per cent. of the stroke.

The engines are supplied with steam of 150 lbs. pressure per square inch by two single-ended boilers, 13 feet diameter and 10 feet 6 inches long, with six furnaces 3 feet 2 inches diameter. Smoke-tubes $3\frac{1}{4}$ inches diameter outside and 7 feet long and No. 8 W. G. thick. The boilers are made entirely of Siemens-Martin steel except the stays and tubes, which are of wrought-iron. The total heating-surface is 3310 square feet, and the total fire-grate surface 97·5 square feet. Diameter of funnel 6 feet. The consumption of coal is $1\frac{1}{2}$ lbs. per indicated horse-power per hour.

The vessel in which these engines are fitted is 265 feet in length, 39 feet 6 inches in breadth, and 24 feet 7 inches in moulded depth: gross tonnage 1936, net register tonnage 1261.

Conversion of Compound Engines to Triple-Expansion Engines.—The economy derived from triple-expansion warrants the conversion of existing compound or double-expansion engines to triple-expansion engines. This may be effected, in inverted-cylinder engines, in three different ways:—

(1.) By adding another cylinder, tandem-fashion, to the top of either the present existing high-pressure cylinder, or low-pressure cylinder. This is the simplest way, but it has the objection of causing an unequal initial stress on each crank, and want of uniformity of turning-effort on the crank-shaft.

(2.) By placing another cylinder on the top of both the high-pressure cylinder and the low-pressure cylinder, and carrying out one of the three stages of expansion in two cylinders instead of in one cylinder. This tandem-arrangement permits the attainment of a nearly equal stress on each crank, and approximate uniformity of turning-effort.

(3.) By lengthening the bed-plate of the engine, and adding another crank and cylinder. This arrangement of cylinders on three cranks increases the length of the engine-room, but it ensures equality of initial stress on each crank, and greater uniformity of turning-effort and steadiness than either of the previous methods of conversion, besides minimising wear and tear. Space may be economised by placing the steam-chests at the side, which enables the cylinders to be placed close together, and the total length of the engine becomes no greater than that of a compound-engine.

Triple-Expansion Engines, with Cylinders arranged on Two Cranks, do not run so sweetly as those arranged on three cranks, and are

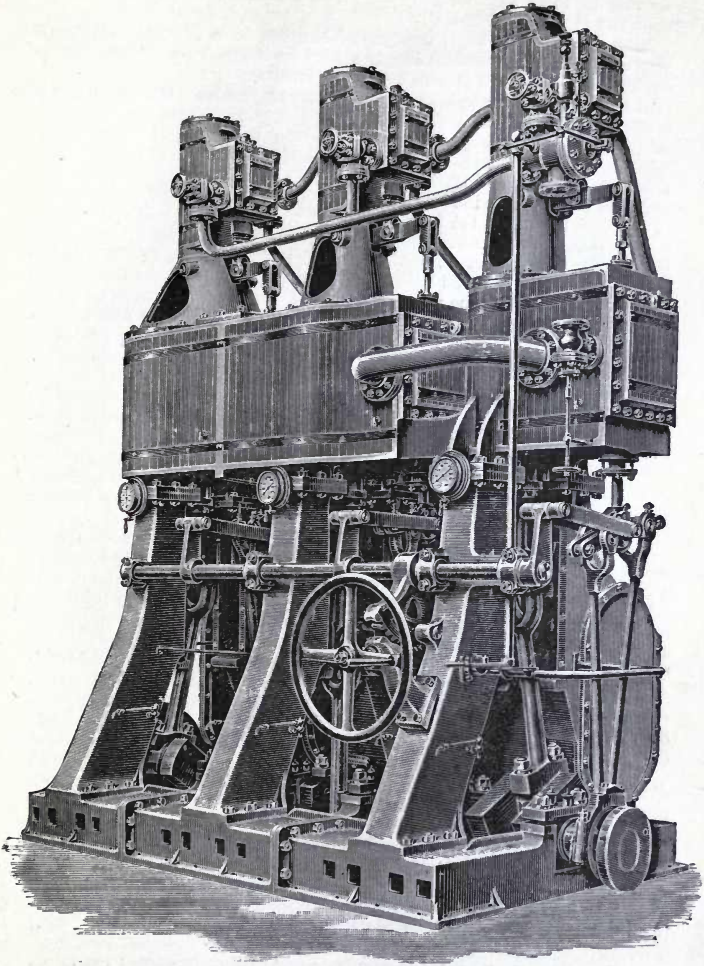


Fig. 369.—Quadruple-expansion surface-condensing marine-engines, by Rankin and Blackmore, Greenock

inferior to them in uniformity of motion, freedom from vibration, and economy in wear and tear.

The Conversion of a Compound-Engine to a Two-Crank Quadruple-Expansion Tandem-Engine may be effected by adding another cylinder to each of the present existing cylinders.

A Set of Quadruple-Expansion "Disconnective" Screw-Engines of 528 indicated horse-power is shown in Fig. 369. There are three high-pressure cylinders, placed tandem-fashion over the first and second intermediate and low-pressure cylinders; the respective diameters being 7 inches, 7 inches, 7 inches, 16 inches, 22 inches, and 34 inches: and the length of stroke of pistons is 24 inches. The reason why six cylinders were adopted in this case instead of the four-cylinder arrangement was, that the engines were required to run sometimes very slowly, or not exceeding 15 revolutions per minute. But the capacity of the six cylinders was made the same as would have been required with the four-cylinder arrangement. Another motive for distributing the power equally over three cranks was to make the engines work as sweetly as possible, this being a matter of the first importance in a yacht. Again, by admitting steam to the three high-pressure cylinders simultaneously, prompt handling is insured and starting-valves are dispensed with, as the three cranks are set at angles of 120° apart.

Further, this combination of cylinders enables the so-called "disconnective" arrangement to be applied in a singularly efficient way, as each high-pressure cylinder forms a natural starting-point for the three principal subdivisions of the engine when working single tandem, for which purpose auxiliary exhaust-pipes have been provided. The high-pressure cylinders are also utilised for heating-up the lower cylinders in a very simple manner by allowing the hot water and steam to drain into them instead of into the bilges as usual. The chief objection to this type of engine as compared with the ordinary triple-expansion working on three cranks is the increased friction of the additional cylinders; but there is not so much in this as might be supposed at first, as owing to the number of stages, the high-pressure pistons (which with their rods form guides for the larger pistons in a heavy sea-way), and indeed the others, also, can be made so easy a fit that no oil need be used unless just before stopping the engines, as the steam itself will do all the necessary lubrication, and any portion which may escape will be worked up in the next stage. The "disconnective"-gear affords security against a complete breakdown, or in the event of any part requiring to be overhauled; say, for example, if the white-metal often employed for crank-pin bushes should give out, it would only be the work of a few minutes to uncouple the connecting-rod and set the remaining two-thirds, or one-third if need be, of the engine to work, thus allowing ample time for refilling the bushes at leisure.

The high-pressure cylinders are provided with liners of hard cast-iron, and their pistons are fitted with Ramsbottom packing-rings, while the intermediate and low-pressure pistons have Buckley's rings and springs. The high-pressure and first intermediate piston-rods and valve-spindles have their stuffing-boxes filled with metallic packing, ordinary packing being used for the others. The valve-gear is of the ordinary link-motion type, with all the working parts made very large and easily adjustable; the valves are all of the common locomotive description.

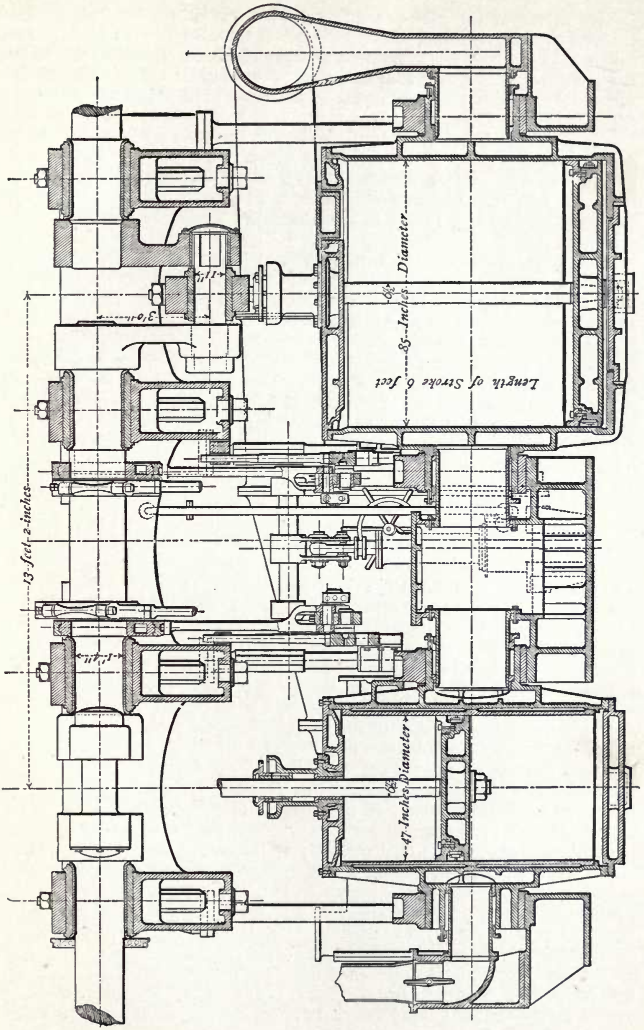


Fig. 370.—Oscillating marine-engine, by J. Jack and Co., Liverpool.

The valves of the high-pressure cylinders are driven by their spindles coupled to rocking-levers, one end of each of these levers working on a fixed fulcrum, while the other is coupled to a prolongation of the corresponding low-pressure (or intermediate cylinder) valve-spindle. The high-pressure valves are thus given a travel equal to half that of the valves of the larger cylinders below.

The air-pump, circulating-pump, feed-pump, and bilge-pumps are worked from the after division of the engine by levers.

Steam of 180 lbs. pressure is supplied to the engines by a return-tube boiler 11 feet 6 inches in diameter by 9 feet 10 inches long, having two corrugated furnaces 3 feet 5 inches internal diameter, and firebars 5 feet 9 inches long.

A large stop-valve is fitted to the forward high-pressure cylinder, to the bottom branch of which is bolted the main steam-pipe connected to the boiler, and on its upper branch is placed a small stop-valve for the admission of steam to the forward high-pressure cylinder. Similar small stop-valves are also attached to the other two high-pressure cylinders, so that, in the event of the vessel running short of coal, one or two of the high-pressure cylinders could have the steam shut off. The first stage of the expansion of the steam is carried out in the three high-pressure cylinders, which is virtually one cylinder subdivided into three. Steam is admitted to the high-pressure cylinders simultaneously, through a pipe with branches which connect these cylinders together, and it impels their pistons during thirteen-sixteenths of the stroke, when the engines are working up to their full power of 528 indicated horse-power. Steam is exhausted through a horizontal curved pipe placed at the back of the engines, and connected to each high-pressure cylinder by branches, which conducts the steam to the first intermediate cylinder beneath, in which the second stage of expansion is carried out, from which it is exhausted into the second intermediate cylinder in which the third stage of expansion is conducted, and whence it is exhausted into the low-pressure cylinder, where the fourth and last stage of expansion is conducted, and it is finally discharged into the condenser. The steam drives the pistons of each intermediate cylinder, and of the low-pressure cylinder during three-fourths of the stroke. The engines when tested developed 528 indicated horse-power, when making 113 revolutions per minute, and expanding the steam twelve times, the boiler-pressure of the steam being 170 lbs. per square inch. When the engines made 103 revolutions per minute, with steam cut off in the high-pressure cylinders at three-fourths of the stroke, the valves of the first and second intermediate, and low-pressure cylinder cutting off the steam at eleven-sixteenths of the stroke, they developed 412 indicated horse-power, and the consumption of hand-picked Penrikyber Welsh coal was only $1\frac{1}{8}$ lb. per indicated horse-power.

An Oscillating Compound Surface-Condensing Paddle-Engine of 2000 indicated horse-power is shown in Fig. 370. The upper-end of each piston-rod is fitted with brasses, which work on the crank-pin. The cylinders are supported by, and oscillate on, trunion-bearings, which enable the piston-rods to accommodate themselves to the motion of the crank. Steam is admitted to, and exhausted from the cylinders through the hollow trunions. The high-pressure cylinder is 47 inches diameter, the low-pressure cylinder is 85 inches diameter, and the length of stroke is 6 feet.

The crank-shaft, crank-pins, and piston-rods are forged from ingots of Siemens-Martin steel. The crank-shaft bearings are 16 inches diameter, and 25 inches long, the crank-pin bearing is 11 inches diameter, and 18 inches long, and the brasses are lined with antifriction-metal. The paddle-shaft-bearing is $16\frac{1}{2}$ inches diameter, and 2 feet 9 inches long, the paddle-wheels are of the feathering type, the floats are made of iron, and are curved on the face. The area of the tube-surface of the surface-condenser is 3980 square feet. The air-pump, centrifugal circulating-pump, feed-pumps and bilge-pumps are worked by a pair of small independent compound engines. This arrangement permits the main engines to be run as slow as six revolutions per minute when required, and makes their prompt handling certain under any conditions. The links are reversed by one of Brown's patent engines, and the motion of both reversing and telegraph-handles coincides in direction with that of the ship. Steam of 85 lbs. per square inch working-pressure is supplied to the engines by two double-ended steel boilers 14 feet 3 inches mean diameter, and 16 feet 1 inch long. having three furnaces at each end, or twelve furnaces in all, each pair of fore and aft furnaces opening into one combustion-chamber; the diameter of each furnace is 3 feet 4 inches, and its length is 6 feet 6 inches. The fire-grate surface is 200 square feet, the heating-surface of the tubes is 5662 square feet, and the heating surface of the fire-boxes is 966 square feet. The total heating-surface being 6628 square feet. The boilers produce an ample supply of steam with easy firing. The vessel in which these engines are fitted is 270 feet long over all, 31 feet 3 inches broad, and 15 feet 6 inches deep to main deck, the height from the main-deck to the promenade-deck being 7 feet 9 inches.

Marine-Engine Governors are used to prevent the racing of the engines when the sea is rough; a very efficient and sensitive governor of this description is shown in Fig. 371. The performance of this governor is limited to working the slide-valve of a small steam-cylinder, the piston-rod of which is connected to, and moves the throttle-valve of the marine-engine, from which the governor is driven by a band, so that both work in unison. The governor consists of a small fly-wheel, with two weighted arms, hung diametrically opposite, and geared to a bevel-pinion cast on the driving-pulley. The inertia of the fly-wheel and arms allows the driving-pulley to overrun them on any increase of speed, and the centrifugal force of the weights keeps the position; the motion so attained is conveyed through a sleeve and levers to a valve on the governor's steam-cylinder which admits and exhausts the steam, the piston in the cylinder moving in unison with the movement of the valve. This is accomplished by attaching the piston-rod end to the end of the valve-spindle by a link, so that the motion of the piston moves the small valve laterally and shuts off the steam. Thus the angular motion of the valve, derived from the governor, opens the steam and exhaust-ports, and

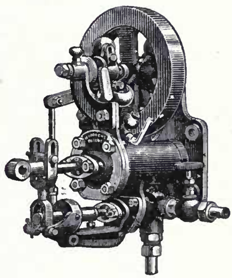


Fig. 371.—Murdoch's marine-engine governor.

the lateral motion of the piston closes the ports again. The governor is connected to the main engines by a band, it runs in unison with them, and should any increase of speed take place, produced by the sea leaving the propeller of the ship, it causes the fly-wheel to be overrun by the driving-pulley of the governor, and the weighted arms expand and move the sleeve along the spindle, and the small valve is turned in its seat, thereby opening the steam and exhaust ports in the governor's steam-cylinder and causing the piston to move *inwards*, and as the clutch on the end of the piston-rod is connected by a rod to the steam throttle-valve of the main engines, the throttle-valve is thereby closed. The opening action is the reverse of that of closing, that is, the piston moves outwards.

The great sensitiveness of this governor prevents racing by closing the throttle-valve on any sudden increase of speed, and prevents the sudden shocks caused by the immersion of the propeller after it has been out or partly out of the water, by opening the valve again so quickly that practically a uniform speed is maintained in the heaviest seas.

The loss of efficiency of the Steam Engine is due principally to the small range of temperature through which it works, to heat lost by radiation, and to condensation of steam during its admission to the cylinder.

The loss from initial condensation of Steam averages from 15 to 45 per cent., being proportional to the area of metal exposed to the steam up to the point of cut-off, and also to the difference of temperature of the steam at the point of cut-off, and that of the exhaust-steam, hence the earlier the cut-off the greater the loss. Therefore, the quantity of steam shown by an indicator diagram is generally less by at least 15 per cent. than that actually used, and the steam thus lost should be added to the weight of steam deduced from a diagram, to obtain the quantity of steam actually used by the engine. This may be illustrated by an example :—An engine of 900 indicated horse-power with a high-pressure cylinder 22 inches diameter, length of stroke 42 inches, cutting off steam at two-thirds the length of stroke, makes 63 revolutions per minute; temperature of feed-water 104° Fahr., pressure of steam shown by the indicator diagram 90 lbs. per square inch. Required the consumption of steam per indicated horse-power per hour, and the quantity of water evaporated from and at 212° Fahr. per lb. of fuel?

Then $\frac{22 \times 22 \times \cdot 7854 \times 42 \times 2 \times 63 \times 60 \text{ minutes} \times 2}{1728 \times 3 \text{ denominator of cut-off}} = 46566 \cdot 37$ cubic feet of steam used per hour. The weight of one cubic foot of steam of $90 + 15 = 105$ lbs. per square inch absolute pressure is, from Table 79, $= \cdot 2434$ lb. and $(46566 \cdot 37 \times \cdot 2434) \div 900$ I.H.P. $= 12 \cdot 56$ lbs. of steam shown by the diagram per indicated horse-power per hour. The total heat in the steam is, from Table $79 = 1183 \cdot 8$ units, and $\frac{(1183 \cdot 8 + 32) - 104^\circ}{966} = 1 \cdot 15$, the factor of evaporation, $\times 12 \cdot 56$ lbs. $= 14 \cdot 444$ lbs., the equivalent evaporation from and at 212° Fahr. Assuming the steam lost by initial condensation to be 16 per cent., then $\frac{14 \cdot 444 \times 100}{84} = 17 \cdot 2$ lbs., the weight of steam actually used by this engine per indicated horse-power per hour.

SECTION VI.

STRENGTH AND SPECIFIC GRAVITY OF STEEL AND WROUGHT-IRON PLATES AND BARS; CAST-IRON, GUN-METAL, BRASS, AND OTHER ALLOYS; TIMBER AND OTHER MATERIALS, ETC.

SECTION VI.

STRENGTH AND SPECIFIC GRAVITY OF STEEL AND WROUGHT-IRON PLATES AND BARS; CAST-IRON, GUN-METAL, BRASS, AND OTHER ALLOYS; TIMBER AND OTHER MATERIALS, ETC.

STRENGTH AND SPECIFIC GRAVITY OF MATERIALS.

The Strength of Materials is measured by the resistance they oppose to alteration of form and rupture when subjected to strain or load.

Tensile Strength or Tensile Stress is the resistance offered by a body to being pulled or drawn asunder. It produces elongation.

Shearing Strength or Shearing Stress is the resistance offered by a body to being severed or cut through. It produces deflection and elongation.

Compressive Strength or Compressive Stress is the resistance offered by a body to being crushed. Its effect is to compress, shorten, and produce lateral deflection.

Transverse Strength or Transverse Stress is the resistance offered by a body to a lateral pressure tending to bend and break it across. It produces lateral deflection.

Torsional Strength or Torsional Stress is the resistance offered by a body to being twisted asunder. It produces angular deflection.

Working-Strain or Working-Stress is the utmost strain or stress to which it is considered safe to subject a body, during its ordinary use as part of a machine or structure.

Ultimate Strain is the utmost strain or stress, or alteration of shape, which a body can bear without breaking.

Working-Load is the load which produces the working-stress.

Proof-Strength, Proof Strain, or Proof Stress, is the utmost strain or stress which a body can bear without suffering any diminution of stiffness or strength.

Proof-Load is the load which produces the proof stress.

Dead-Load, means one that is put on by degrees and remains steady.

Live-Load, means one that is put on suddenly and accompanied with vibration. The effect of a live-load upon a structure is much more injurious than that of a dead-load.

Set is the permanent strain or alteration of shape of an imperfectly elastic body which remains after a load has been removed.

Stiffness is measured by the intensity of the stress required to produce a certain fixed quantity of strain.

Pliability is the inverse of stiffness, and is measured by the quantity of the strain produced by a certain fixed stress.

Modulus of Elasticity is the reciprocal of the direct pliability when the stress does not exceed the proof-strength.

The modulus of elasticity is the measure of the elastic force of any substance. It may be expressed as the force in lbs. required to stretch a bar to double its length, if its elasticity remained perfect. A stretching-force was applied, in an experiment, to a bar of good wrought-iron, 10 feet long, which produced an extension in inches equal to $\frac{1}{233500}$th part of the weight or force in lbs. required to stretch it, and the modulus of the elasticity of this iron is $= 233500 \times 120$ inches length of bar $= 28020000$ lbs. per square inch. The modulus of elasticity, or resistance to stretching, of metals and woods, averages as follows:—

Cast-steel, hardened	37500000	Teak	2167000
Cast-steel, not hardened	33000000	Oak	1714500
Mild steel	30000000	Ebony	1610000
Wrought-iron	28020000	Birch	1600000
Iron-wire	25000000	Mahogany, H.	1593000
Homogeneous metal	23830000	Ash	1525000
Platinum-wire	22000000	Pine	1500000
Cast-iron	18000000	Box	1421000
Copper-wire	14500000	Elm	1343000
Cast-iron, weak	14000000	Beech	1316000
Phosphor-bronze	13500000	Poplar, white	1134000
Brass-wire	13500000	Alder	1086750
Gun-metal	9500000	Sycamore	1036000
Brass	9000000	Chestnut	924750
Tin	4500000	Walnut	837000
Lead, sheet	720000	Blue-gum	800000

The Stress or Pull in lbs. per Square Inch required to Elongate a Bar may be found by multiplying the strain by the modulus of elasticity.

The Strain produced by a given Direct Stress may be found by dividing the stress, or pull in lbs. per square inch required to elongate a bar, by the modulus of elasticity.

Spring or Resilience, is the greatest quantity of work which a body can bear in the form of a blow or shock without injury. It is equal to one-half of the product of the proof strength of the body by its proof strain.

Direct Extensibility, or compressibility, is the amount of direct strain produced by each pound on the square inch of direct stress.

Elastic Strength or the Elastic Limit is the utmost amount of stress which a body can bear without set. The elastic limit of good mild steel-plates averages 17 tons per square inch either along or across the grain, and of good wrought iron-plates it averages 13 tons per square inch either along or across the grain. The elastic limit of iron and steel-bars and plates may in a general way be taken at one-half the breaking weight. A stress of one ton per square inch applied to a bar of wrought-iron will produce an elongation of approximately $\frac{1}{10000}$th part of its length, and each additional ton of strain applied will stretch the bar another $\frac{1}{10000}$th part of its length, until the limit of its elasticity is reached.

The Fatigue of a Metal is the disturbance of its component particles under strain or stress within the limits of its elastic strength.

The Refreshment of a Metal is the re-adjustment of its component particles after fatigue, or the restoration of the metal to its original state.

The Patience of a Metal is the time required for its restoration after fatigue.

The Endurance of a Metal is its power of resisting a prolonged strain or stress. The endurance of a metal has no fixed relation to its tensile strength, or its power of resisting a tensile strain for a short period.

The Factor of Safety is the ratio in which the breaking-strain on a piece of material exceeds the working-strain.

$$\text{Factor of safety} = \frac{\text{breaking-strain}}{\text{working-strain}}.$$

$$\text{Breaking-strain} = \text{working-strain} \times \text{factor of safety}.$$

The factor of safety for a *live-load* is usually 6 in metals, 8 in masonry, and 10 in timber. The factor of safety for a *dead-load* is usually one-half that required for a live-load.

A Test-Strip of Steel, about 1 inch wide and from 6 to 9 inches long, cut either lengthways or crossways of the plate or bar, after being heated to a low cherry-red and cooled in water at a temperature of 82° Fahr., should stand bending double round a curve of which the diameter is not more than three times the thickness of the piece to be tested, without fracture. The fracture of a broken test-strip of steel should be silky.

A Test-Strip of Wrought-Iron 2 feet long and 2 square inches of sectional area of the iron used for crank-shafts, should not fracture until twisted at least five complete turns, and the fracture should be fibrous. It should also double up cold, quite close, without fracture. The edges of the test-strip should be planed in the direction of the grain: when planed across the grain it is liable to fracture in the tool-grooves.

A good Steel Casting should bend through a right angle before breaking, and it should be composed of ·28 carbon, ·3 silicon, and ·69 manganese.

Steel-Plates, Wrought-iron Plates, and Bars, should be well and cleanly rolled, and free from scales, blisters, laminations, cracked edges, or other defects. They should be of such quality and strength as to be equal to the tensional strains given in the following table, and to indicate the per-centages of elongation and of contraction of the area at the point of fracture therein given.

Table 96.—TENSILE STRENGTH OF STEEL AND IRON PLATES AND BARS.

Description of Plates and Bars.	Tensional Breaking Strain per Square Inch in Tons.	Percentage of Contraction of Area of Fracture.
	Tons.	Per cent.
Siemens-Martin mild steel boiler-plates, either along or across the grain. With an elongation of 20 per cent. in 8 inches; not to be less than . . .	26	20
Or more than	30	30
Steel-rods for making rivets; not to be less than .	25	40
Or more than	28	40
Siemens-Martin mild steel-plates for girders, bridge-plates, channel, angle or flat bars, either along or across the grain, with an elongation of 20 per cent. in 8 inches; not to be less than . . .	27	20
Or more than	31	20
Mild hoop-steel	33	12
Mild steel for piston-rods and valve-spindles . .	28	30
Mild steel plates for fire-boxes	25	50
Steel-castings, with an elongation in 10 inches of not less than 18 per cent.; not to be more than .	30	45
Best Yorkshire wrought-iron plates, along the grain	24	15
Best Yorkshire wrought-iron plates, across the grain	22	12
Wrought-iron bolts, nuts and rivets	23	25
Wrought-iron boiler-plates, along the grain . .	21	10
Wrought-iron boiler-plates, across the grain . .	18	5
Wrought-iron ship-plates and bridge-plates, along the grain	20	8
Wrought-iron ship-plates and bridge-plates, across the grain	17	4
Wrought-iron round and square bars, and flat bars under 6 inches wide	24	20
Wrought-iron angle, channel, T bars, and flat bars 6 inches wide and upwards	22	15
Hoop-iron	22	8
Wrought-iron rolled-joists	24	15
Wrought-iron crank-shafts	25	42

Mild Steel for Fire-box Plates should have very little phosphorous and no sulphur in its composition, as they reduce the heat-enduring power of steel.

Table 97.—AVERAGE STRENGTH OF IRON AND STEEL BARS AND PLATES, CULLED FROM THE TEST-BOOKS OF SEVERAL NOTED MANUFACTURERS.

Description.			Tensional Breaking Strain per Square Inch in Tons.	Percentage of Contraction of Area of Fracture.
			Tons.	Per cent.
Wrought-iron bars made at the Earl of Dudley's Round Oak Iron Works, Dudley :—				
L. W. R. O. bars, elongation	28·8	per cent. in 10 inches.	24·94	48·2
L. W. R. O. bars, elongation	27·5	per cent. in 10 inches.	26·57	44
Best bars, elongation	25·4	per cent. in 10 inches.	24·67	45·3
Best, best, bars, elongation	29·7	per cent. in 10 inches.	23·35	45·2
Best, best, best, bars, elongation	30·7	per cent. in 10 inches.	23·60	46·9
Best, best, best, C bars, elongation	27·5	per cent. in 10 inches.	26·42	47·9
Best rivet-iron bars, elongation	26·6	per cent. in 10 inches.	24·75	45·7
Best, best, rivet-iron bars, elongation	28·4	per cent. in 10 inches.	24·75	47·2
Best, best, best, rivet-iron bars, elongation	27·4	per cent. in 10 inches.	24·26	47·2
Best, best, best, rivet-iron special bars, elongation	28·8	per cent. in 10 inches.	24·40	47·3
Best cable-iron bars, elongation	25·9	per cent. in 10 inches.	24·28	45·3
Best, best cable-iron bars, elongation	29·1	per cent. in 10 inches.	23·25	49·1
Best, best, best, cable-iron bars, elongation	29·7	per cent. in 10 inches.	23·94	47·3
The average tensile breaking-strain of the above bars, per square inch of fractured area was 46·2 tons.				
Wrought-iron plates and bars made by the Shelton Iron and Steel Co., Limited, Stoke-on-Trent :—				
Best boiler-plates $\frac{1}{2}$ inch thick, lengthways			22·3	10·3
" " " " crossways			18·7	4·6
Elongation in 12 inches = $\frac{7}{8}$ inch lengthways, and $\frac{1}{2}$ inch crossways.				

Table 97 *continued.*—Average Strength of Iron and Steel Bars and Plates, by Several Noted Manufacturers.

Description.	Tensional Breaking Strain per Square Inch in Tons.	Percentage of Contraction of Area of Fracture.
	Tons.	Per cent.
Wrought-iron plates and bars made by the Shelton Iron and Steel Co., Limited:—		
Best, best, boiler-plates $\frac{7}{16}$ inch thick, lengthways	23·6	16·2
,, ,, ,, ,, ,, crossways .	20·16	10·4
Elongation in 12 inches = $1\frac{1}{16}$ inch lengthways, and $\frac{5}{8}$ inch crossways.		
Best, best, boiler-plates $\frac{11}{16}$ inch thick, lengthways	23·11	14·3
,, ,, ,, ,, ,, crossways .	20·6	10·5
Elongation in 12 inches = 1 inch lengthways, and $\frac{11}{16}$ inch crossways.		
Rivet-iron. Elongation in 12 inches = $3\frac{1}{4}$ inches, or 27 per cent.	25	40
Bolt-iron. Elongation in 12 inches = $3\frac{5}{8}$ inches, or 30·2 per cent.	23·14	47
Angle-iron. Elongation in 12 inches = $3\frac{1}{4}$ inches, or 27 per cent.	26·5	34·1
Cable or chain-iron. Elongation 31·8 per cent. .	24·5	40·6
Wrought-iron bars and plates made by the Butterley Company, Alfreton:—		
Best round and square bars	23·9	
Best, best, round and square bars	25·1	
Best, flat, angle, and T-bars	21·6	
Best, best, flat, angle, and T-bars	24·5	
Best plates, lengthways	20·63	
,, ,, crossways	16·17	
,, best plates, lengthways	22·83	
,, ,, ,, crossways	19·26	
Rolled-joists	24·92	
Wrought-iron and steel-plates and bars made by W. Beardmore & Co., Parkhead Works, Glasgow:—		
Best, best, wrought-plates, elongation . . . 10 per cent. in 8 inches.	21	
Best triple-crown wrought-iron plates, elongation . . 12 per cent. in 8 inches.	$22\frac{1}{2}$	

Table 97 *continued.*—AVERAGE STRENGTH OF IRON AND STEEL BARS AND PLATES, BY SEVERAL NOTED MANUFACTURERS.

Description.	Tensional Breaking Strain per Square Inch in Tons.	Percentage of Contraction of Area of Fracture.
	Tons.	Per cent.
Wrought-iron and steel-plates and bars made by W. Beardmore and Co., Parkhead Works, Glasgow:—		
Best wrought-iron bars, elongation . . . . . 20 per cent. in 8 inches.	23	25
Best, best, wrought-iron bars, elongation . . . . . 25 per cent. in 8 inches.	$24\frac{1}{4}$	32
Angle-iron, elongation . . 23 per cent. in 8 inches.	$24\frac{1}{2}$	
Rivet-iron, elongation . . $25\frac{3}{4}$ per cent. in 8 inches.	$24\frac{1}{4}$	
Iron-forgings, elongation . $12\frac{1}{2}$ per cent. in 8 inches.	$22\frac{1}{2}$	
Steel boiler shell-plates, elongation . . . . . $22\frac{1}{2}$ per cent. in 8 inches.	29	
Steel internal boiler-plates, elongation . . . $24\frac{1}{4}$ per cent. in 8 inches.	$27\frac{1}{2}$	
Steel ship-plates, elongation $18\frac{1}{4}$ per cent. in 8 inches.	$30\frac{1}{2}$	
Steel-angles and bars, elongation . . . . $17\frac{1}{2}$ per cent. in 8 inches.	31	
Rivet-steel, elongation . . 33 per cent. in 8 inches.	$27\frac{1}{2}$	
Steel bridge-plates, elongation . . . . . 23 per cent. in 8 inches.	29	
Steel-forgings, elongation . $23\frac{1}{2}$ per cent. in 8 inches.	$29\frac{1}{2}$	
Wrought-iron and steel-plates and bars made by John Brown and Co., Limited, Sheffield:—		
Iron boiler-plates, best, best, lengthways . .	20	
Iron boiler-plates „ crossways . . .	17	
Iron boiler-plates, best, best, best, lengthways .	22	
Iron boiler-plates „ crossways . .	18	
Steel-plates and bars; not less than . . .	26	
Or more than	30	
Rivet-steel and boiler-stay bars; not less than .	26	
Or more than	30	
Steel-shafts and axles; not less than . . .	30	
Or more than	35	

Table 97 *continued.*—Average Strength of Iron and Steel Bars and Plates, by Several Noted Manufacturers.

Description.	Tensional Breaking Strain per Square Inch in Tons.	Percentage of Contraction of Area of Fracture.
	Tons.	Per cent.
Wrought-iron and steel-plates and bars made by John Brown and Co., Limited, Sheffield :—		
Steel-tires; not less than	26	
Or more than	50	
Wrought-iron bars made by N. Hingley and Sons, Netherton Iron Works, Dudley :—		
Netherton crown-best bar-iron, elongation 30 per cent.	22·6	45
Netherton crown-best bar-iron, elongation 19 per cent.	22·9	46
Netherton crown-best bar-iron, elongation 22 per cent.	23·2	44
Netherton crown-best bar-iron, elongation 24 per cent.	23·8	35
Netherton crown-best rivet-iron, elongation 20 per cent.	23·5	50
Netherton crown-special best, best, cable-iron, elongation 23 per cent.	24	45
Netherton crown-special best, best, cable-iron, elongation $23\frac{1}{2}$ per cent.	25·5	45
Netherton crown-special best, best, $2\frac{9}{16}$ inch cable-iron, elongation 14 per cent. . . .	25·7	27
Steel-plates, bars, &c., made by the Bolton Iron and Steel Company, Limited, Bolton :—		
Steel-plates and bars for boilers, elongation . . 25 per cent. in 8 inches.	28	45
Steel bridge-plates, elongation . . . . . . 20 per cent. in 8 inches.	$29\frac{1}{2}$	40
Steel-angles, tees, bulb-beams, &c., for bridge and ship-building, elongation . . 20 per cent. in 8 inches.	$29\frac{3}{4}$	40
Rivet-steel, elongation . . 30 per cent. in 8 inches.	27	50
Steel locomotive-crank-axles and straight axles, elongation . . . . . . 30 per cent. in 2 inches.	28 to 30	50

Table 98.—BREAKING STRENGTH OF STEEL-PLATES AND BARS, AND OF STEEL-WIRE IN LBS. PER SQUARE INCH.

Description.	Tenacity in lbs. per Square Inch.		Authority.
	Lengthway.	Crossway.	
Siemens-Martin mild-steel boiler-plates, highest average . .	64000	64000	—
Siemens-Martin mild-steel boiler-plates	60000	60000	—
Siemens-Martin mild-steel boiler-plates	58150	58150	—
Siemens-Martin mild-steel boiler-plates	57850	57850	—
Siemens-Martin mild-steel boiler-plates, lowest average . .	53000	53000	—
The above steel-plates will probably average an elastic limit of 40000 lbs. per square inch tension, with an elongation of 20 per cent. in 8 inches, and a contraction of area of fracture of 20 per cent. in 8 inches.			
Mild-steel plates and bars containing ·13 per cent. of carbon with an elongation of 26 per cent. in 8 inches	62000	62000	—
Mild-steel plates, stamping grade .	58240	58240	—
,, bars, average of number	59000	59000	—
,, bolts	80640	80640	—
,, rivets	62720	62720	—
Cast-steel castings	65000	65000	—
Hadfield's manganese-steel, hammered	94080	...	—
Homogeneous metal-bars . .	100990	...	W. H. Barlow.
,, ,, . .	93000	...	Fairbairn.
,, ,, . .	90647	...	Kirkaldy.
,, ,, . .	89720	...	,,
Bessemer-steel bars, rolled and forged	152912	...	Wilmot.
Bessemer-steel bars, rolled and forged	111460	...	Kirkaldy.
Cast-steel bars, rolled and forged .	134000	...	Rennie.
,, ,, ,, .	132900	...	Kirkaldy.
,, ,, ,, .	92015	...	,,
Shear-steel bars, rolled and forged	118468	...	,,
Soft-steel plates, tempered . .	121700	...	Tresca.
,, ,, not tempered .	81700	...	,,

Table 98 *continued.*—BREAKING STRENGTH OF STEEL-PLATES AND BARS, AND OF STEEL-WIRE IN LBS. PER SQUARE INCH.

Description.	Tenacity in lbs. per Square Inch.		Authority.
	Lengthway.	Crossway.	
Hard-steel plates, tempered . .	103000	...	Tresca.
Hard-steel plates, not tempered .	74300	...	"
Puddled-steel bars, rolled & forged	116330	...	W. H. Barlow.
" " " "	95233	...	"
" " " "	94760	...	Mallet.
" " " "	90000	...	Fairbairn.
" " " "	71480	...	Kirkaldy.
" " " "	62760	...	"
Spring-steel bars " "	72529	...	"
Whitworth's compressed-steel .	89600	...	Sir J. Whitworth.
" " " . .	152320	...	"
Whitworth's steel, tempered in oil	107968	...	"
" " " "	93000	...	Fairbairn.
" " " "	90647	...	Kirkaldy.
" " " "	89724	...	"
Cast-steel plates	96280	97150	"
" "	75590	69082	"
" " hard . . .	102900	...	Fairbairn.
" " soft . . .	85400	...	"
Puddled-steel plates . . .	102590	85360	Kirkaldy.
" " . . .	71530	67685	"
Soft cast-steel for guns, not tempered	77930	...	Anderson.
Cast-steel tempered in oil . .	120467	...	"
Krupp's cast-steel bolts . .	91840	...	Kirkaldy.
" steel crank-shaft . .	93640	...	—
Cast-steel for drifts . . .	116480	...	Kirkaldy.
" for taps	103040	...	"
" for tools . . .	134400	...	"
Large crank-shafts, steel-castings, not forged	92600	...	—
Large crank-shafts, steel-castings, not forged	72300	...	—
Manganese-steel	64300	...	—
Mild hoop-steel	95000	...	—
" "	82000	...	—
" "	74000	...	—
Bessemer-steel tyres, hammered .	78400	...	Steel Committee.
" axles, " .	74816	...	"
" rails, " .	74368	...	"
Crucible-steel tyres, hammered .	79520	...	"
" axles, " .	91616	...	"

Table 98 *continued*.—Breaking Strength of Steel-Plates and Bars, and of Steel-Wire in lbs. per Square Inch.

Description.	Tenacity in lbs. per Square Inch. Lengthway.	Crossway.	Authority.
Crucible-steel rails, hammered	85430	...	Steel Committee.
Cast-steels for chisels	118182	...	,,
Siemens-steel plates, not annealed	69880	69440	Kirkaldy.
,, ,, annealed	64520	64300	,,
,, ,, annealed and hardened	64650	...	,,

Description.	Diameter. Inch.	Tenacity in lbs. per Square Inch. Lengthway.	Crossway.	Authority.
Steel pianoforte-wire	·035	268800	...	Dr. Percy.
Steel-wire	·019	358400	...	,,
Steel-wire	·030	360416	...	,,
Steel-wire, John Fowler & Co.'s special	·093	344960	...	,,
Steel-wire, John Fowler & Co.'s special	·132	257600	...	,,
Steel-wire, John Fowler & Co.'s special	·159	224000	...	,,
Steel-wire, John Fowler & Co.'s special	·191	201600	...	,,

Fowler's special steel-wire is hard, tough, and rigid. It is used for making steel-wire-ropes for ploughing-tackle, and is composed of ·828 per cent. of carbon, ·587 per cent. manganese, ·143 per cent. silicon, ·009 per cent. sulphur, and of ·030 per cent. of copper.

Table 99.—Breaking Strength of Wrought-Iron Bars and Plates, and of Iron-Wire in lbs. per Square Inch.

Description.	Tenacity in lbs. per Square Inch. Lengthway.	Crossway.	Authority.
Iron-wire, very strong	114000	...	Morin.
,, strong	100000	...	Gordon.
,, medium	86000	...	Telford.
,, weak	71000	...	Morin.
,, average	80640	...	Barlow.

Table 99 *continued.*—Breaking Strength of Wrought-Iron Bars and Plates, and of Iron-Wire in lbs. per Square Inch.

Description.	Tenacity in lbs. per Square Inch.		Authority.
	Lengthway.	Crossway.	
Iron-wire, Warrington, not annealed	80000	...	Barlow.
Iron-wire, Warrington, annealed	53000	...	,,
Good wrought-iron bars, average	60000	51000	,,
,, ,, plates, average	50000	42000	,,
Wrought-iron bars, rolled or forged, mean	57550	...	Kirkaldy.
Wrought-iron bars, Yorkshire	66390	...	,,
,, ,, ,,	60075	...	,,
Rivet-iron, Yorkshire and Staffordshire	59740	...	Fairbairn.
Charcoal bar-iron	63620	...	,,
Wrought-bars, Staffordshire, Lion Brand, best, best	53760	47000	—
Wrought-bars, Staffordshire, Lion Brand, best, best	50000	42600	—
Wrought-iron bars, Staffordshire, best, best, average	52267	45000	—
Wrought-iron bars, Staffordshire	62230	...	Kirkaldy.
,, ,, ,,	56715	...	,,
,, ,, Lancashire	60110	...	,,
,, ,, ,,	53775	...	,,
,, ,, Lanarkshire	64795	...	,,
,, ,, ,,	51327	...	,,
,, ,, Swedish	48933	...	,,
,, ,, ,,	41250	...	,,
,, ,, Russian	59096	...	,,
,, ,, ,,	49564	...	,,
,, ,, Staffordshire (S. C. Crown)	65400	...	T. Lloyd.
,, ,, Staffordshire (S. C. Crown)	53530	...	,,
Wrought-iron boiler-plates, Staffordshire	43904	37676	Edwin Clark.
Wrought-iron plates, average of a large number of tests	45000	38000	—
Wrought-iron plates for steam-boilers	47000	40000	Board of Trade estimate.
Best scrap rivet-iron $\frac{7}{8}$ inch diameter	53760	...	,,
Bushelled-iron from turnings	55878	...	Kirkaldy.
Hammered scrap-iron	53420	...	,,

Table 99 *continued.*—Breaking Strength of Wrought-Iron Bars and Plates, and Forgings in lbs. per Square Inch.

Description.	Tenacity in lbs. per Square Inch. Lengthway.	Crossway.	Authority.
Wrought-iron bars, Staffordshire (S. C. Crown)	50400	...	Steel Committee.
Wrought-iron bars, best Yorkshire	52200	...	,,
,, ,, Lowmoor . .	55550	...	,,
,, ,, ,, .	64750	...	,,
Angle-iron from various districts .	61260	...	Kirkaldy.
,, ,, ,, ,, .	50056	...	,,
,, Cleveland . . .	51800	...	—
Wrought-iron plates, mean . .	50737	46171	Kirkaldy.
,, ,, Lowmoor .	64200	62490	Fairbairn.
,, ,, Yorkshire .	58487	55033	Kirkaldy.
,, ,, ,, .	52000	46221	,,
Wrought-iron plates, Yorkshire bridge-iron	49930	43940	Fairbairn.
Wrought-iron bridge-plates, Cleveland	52810	41400	—
Wrought-iron bridge-plates, Cleveland	46860	40000	—
Wrought-iron plates, ordinary good Staffordshire	56996	51251	Kirkaldy.
Wrought-iron plates, ordinary good Staffordshire	46404	44760	,,
Wrought-iron plates, Staffordshire best, best	59820	54820	Fairbairn.
Wrought-iron plates, Staffordshire best, best	49945	46470	,,
Wrought-iron plates, Staffordshire best	61280	53820	,,
Wrought-iron plates, Staffordshire best, best, charcoal . . .	45010	41420	,,
Wrought-iron plates, Staffordshire good common	52825	50820	,,
Wrought-iron plates, Staffordshire bridge-iron	47600	44385	,,
Wrought-iron plates, Lancashire .	48865	45015	,,
,, ,, Lanarkshire	53849	48848	Kirkaldy.
,, ,, ,,	43433	39544	,,
,, ,, Durham .	51245	46712	,,
Pieces cut out of large wrought-iron forgings	47582	44578	,,
Pieces cut out of large wrought-iron forgings	43759	36824	,,

Table 99 *continued.*—BREAKING STRENGTH OF WROUGHT-IRON BARS AND PLATES, AND OF HOOP-IRON IN LBS. PER SQUARE INCH.

Description.	Tenacity in lbs. per Square Inch.		Authority.
	Lengthway.	Crossway.	
Effects of cold-rolling, black bar	58627	...	Fairbairn.
Effects of cold-rolling, black bar turned	60747	...	,,
Effects of cold-rolling, black bar cold rolled	88229	...	,,
Effects of cold-rolling, plate cold-rolled	114912	...	,,
Hoop-iron, best, best	64000	...	—
,, ,,	53000	...	—
,, average	48000	...	—

Description.	Authority.
Loss of strength in screwed and chased bolts, from $7\frac{1}{2}$ to $33\frac{1}{2}$ per cent.	Kirkaldy.
Loss of strength in welded-joints, from 15 to 30 per cent.	,,
Increase of strength of steel by hardening in oil, 12 to 79 per cent.	,,

Table 100.—ULTIMATE TENSILE AND COMPRESSIVE STRENGTH OF CAST-IRON FROM THE EXPERIMENTS OF FAIRBAIRN, HODGKINSON, AND OTHERS.

Description.	Tensile Strength in lbs. per Square Inch.	Compressive Strength in lbs. per Square Inch.	Authority.
Cast-iron, No. 1 cold-blast	12690	56450	Fairbairn and Hodgkinson.
,, ,, ,,	17460	80560	,,
,, No. 1 hot-blast	13430	72193	,,
,, ,, ,,	16125	88740	,,
,, No. 2 cold-blast	13345	68530	,,
,, ,, ,,	18855	102408	,,
,, ,, hot-blast	13500	82730	,,
,, ,, ,,	17800	102030	,,
,, No. 3 cold-blast	14200	76900	,,
,, ,, ,,	15500	115400	,,
,, ,, hot-blast	15278	101830	,,
,, ,, ,,	23468	104880	,,

Table 100 *continued.*—ULTIMATE TENSILE AND COMPRESSIVE STRENGTH OF CAST-IRON.

Description.	Tensile Strength in lbs. per Square Inch.	Compressive Strength in lbs. per Square Inch.	Authority.
Toughened cast-iron	23460	129870	Fairbairn and Hodgkinson.
,, ,, ...	25760	119457	,,
Cast-iron, No. 3 hot-blast after first melting	...	98500	,,
Cast-iron, No. 3 hot-blast after twelfth melting	...	163740	,,
Cast-iron, No. 3 hot-blast after eighteenth melting	...	197120	,,
Cast-iron, weak	13400	82000	Hodgkinson.
,, average	16500	112000	,,
,, strong	29000	145000	,,
,, very strong	34000	...	Anderson.
Pieces cut from cast-iron guns began to yield, or give way at .	...	35000	,,
.	...	40700	,,
Cast-iron, mean of 16 various sorts, 12 inches high	...	85547	Hodgkinson.
Cast-iron, Lowmoor, No. 3 C. B. .	14538	...	,,
,, Devon, No. 3 H. B. .	21683	...	,,
,, cold-blast average of various	16845	99232	,,
,, hot-blast average of various	15300	102502	,,
American iron, effect of re-melting and retaining the metal in a state of fusion for 4 hours.—Pigs .	13440	...	Major Wade.
1st melting	20870	...	,,
2nd ,, ...	24770	...	,,
3rd ,, ...	26790	...	,,
4th ,, ...	27888	...	,,
Samples from 100 gun-heads .	33376	...	,,
A lot of pig-iron in the crude state	12678	...	,,
Twenty-seven guns cast from this pig-iron, 3rd melting . . .	35280	...	,,
Stirling's metal, a mixture of cast-iron and wrought-iron, average .	24500	125000	Hodgkinson.
Cast-iron average strength of good	15680	107520	—
,, medium quality . .	13440	94080	—

A Test-Bar of good Cast-Iron, 3 feet 6 inches long and 2 inches by 1 inch in section, placed edgeways upon supports 3 feet apart, should not break with a less weight than 30 cwts. gradually applied in the middle of the bar.

Table 101.—Breaking Strength of Metals and Alloys in lbs. per Square Inch.

Description.	Tensile Strength in lbs. per Square Inch.	Compressive Strength in lbs. per Square Inch.	Authority.
Aluminium-bronze 90 copper 10 aluminium	73185	132000	Anderson.
	96320	...	"
Tin, cast	4600	...	Rennie.
Zinc, "	6900	...	—
" sheet	6100	...	—
Lead, cast	1814	...	Rennie.
" sheet	1920	...	—
" pipe	2240	...	Jardine.
Brass	1800	10300	Rennie.
Brass, fine, 2 copper: 1 zinc	2890	...	Anderson.
" 7 copper: 3½ zinc	28940	...	"
Alloy of copper 10: iron 10: zinc 80 parts	6988	...	"
Gun-metal or bronze 12 parts copper: 1 part tin	29000	...	"
Gun-metal or bronze 11 parts copper: 1 part tin	30700	...	"
Gun-metal or bronze 10 parts copper: 1 part tin	33000	...	"
Gun-metal or bronze 9 parts copper: 1 part tin	38000	...	"
Gun-metal or bronze 8 parts copper: 1 part tin	36000	...	Gordon.
Gun-metal or bronze 6 parts copper: 1 part tin	43800	...	—
Gun-metal, average strength of good bronze	33000	...	Anderson.
Gun-metal from American guns	23900	...	Major Wade.
" " " "	35480	...	"
Phosphor-bronze	56000	...	—
Muntz-metal	49300	...	—
Austrian sterro-metal	59900	...	Major Wade.
Sterro-metal, copper 60, zinc 39 Tin 1·5: iron 3, cast in sand	43120	...	"
	48160	...	"
Sterro-metal, copper 60, zinc 35, tin 2, iron 3	85120	...	"
Copper, wrought	33600	...	"
" cast	19000	...	"
" "	26000	...	"
" bolts	33000	...	"
" cast	19000	...	Rennie.
" sheet, rolled	30000	...	—

Table 101 *continued.*—BREAKING STRENGTH OF METALS AND ALLOYS IN LBS. PER SQUARE INCH.

Description.	Tensile Strength in lbs. per Square Inch.	Compressive Strength in lbs. per Square Inch.	Authority.
Copper, sheet, hammered . .	33600	...	Rennie.
,, bolts	36000	...	Fairbairn.
,, wire	60000	...	Anderson.
,, ,, ...	64200	...	Kirkaldy.
Phosphor-bronze wire . . .	147113	...	,,
Platinum-wire, hard . . .	265000	...	Wertheim.
Coleford gun-metal, strongest . .	160540	...	Fairbairn.
,, ,, mean of 10 sorts	137340	...	,,
Gold, pure	40320	...	—

A Test-Bar of Good Tough Gun-Metal, 1 inch square and 15 inches long, placed upon supports 12 inches apart, should not break with less than four blows from a monkey of 50 lbs. weight dropped from a height of 5 feet, which will cause a permanent deflection of from $2\frac{1}{4}$ inches to 3 inches.

Table 102.—TENSILE STRENGTH AND RESISTANCE TO TORSION OF VARIOUS WIRES, THE RESULTS OF EXPERIMENTS BY MR. KIRKALDY.

Description of Wire.	Pulling Stress, Wire as Drawn.				Twists in 5 In.	
	Diameter.	Area.	Stress. Total.	Stress. Per Square Inch.	As Drawn. Mean of Three.	Annealed. Mean of Three.
	Inch.	Sq. Inch.	lbs.	lbs.	Twists.	Twists.
Phosphor-bronze . .	·0655	·003367	340	100980	5·0	91
,, ,, . .	·0640	·003216	389	120957	22·3	52
,, ,, . .	·0600	·002827	352	124313	7·0	87
,, ,, . .	·0610	·002922	379	129705	8·3	98
,, ,, . .	·0595	·002778	336	120950	13·0	124
,, ,, . .	·0585	·002655	395	147113	7·5	97
,, ,, . .	·0640	·003216	513	159515	13·3	66
Copper	·0640	·003216	203	63122	86·7	96
Brass	·0605	·002871	233	81156	14·7	57
Steel, ordinary . .	·0600	·002827	342	120976	22·4	79
Iron galvanized, best-best C.	·0580	·002643	170	64321	26·0	44
Iron, galvanized, best Charcoal E. . .	·c580	·002643	174	65834	48·0	87

Table 103.—Tensile Strength and Conductivity of Silicium-Bronze and Phosphor-Bronze Wire, for Electric Appliances, Ropes, &c.

Description of Wire.	Tensile Strength in Tons per Square Inch.		Conductivity per Cent.	
	From	to	From	to
Silicium-bronze telegraph and electric-light wire. High conductivity	29	33	90	96
Silicium-bronze telephone-line wire for long distances	45	50	34	40
Silicium-bronze telephone (local) and spring-wire	50	65	20	30
Phosphor-bronze wire	40	65	20	30

Table 104.—Tensile Strength, Electrical Resistance and Relative Conductivity of Various Wires, the Diameter in each case being One Millimetre=·03937 Inch.

Description of Wire.	Tensile Strength in Tons per Square Inch.	Resistance per mile in Ohms.	Relative Conductivity.
Pure copper	17·78	33·1	100
Silicium-bronze (telegraph)	28·57	34·5	96
Silicium-bronze (telephone)	48·25	103	34
Phosphor-bronze (telephone)	45·71	124	26
Swedish galvanized-iron	22·86	216	16
Galvanized Bessemer-steel	25·40	249	13
Siemens-Martin steel	26·67	266	12

Table 105.—Breaking Strength of Timber from the Experiments of Bevan, Hodgkinson, Fowke, and others.

Description.	Tensile Strength in lbs. per Square Inch.	Compressive Strength in lbs. per Square Inch.	Authority.
Abele, European	7200	5120	Bevan and Hodgkinson.
Acacia „	16000	...	Bevan.
African oak, teak	6730	5270	—
Alder, European	14180	6895	—
Apple-tree, European	19500	6500	Bevan and Hodgkinson.
Ash, English	3640	2974	„

Table 105 *continued.*—Breaking Strength of Timber.

Description.	Tensile Strength in lbs. per Square Inch.	Compressive Strength in lbs. per Square Inch.	Authority.
Ash, European	17000	9000	Bevan and Hodgkinson.
Aspen	10180	...	—
Bamboo, Southern Asia	6300	...	Bevan.
Baracara	...	8818	Fowke.
Bartaballe	...	8818	,,
Baywood, Newfoundland	2800	6000	—
Beech	11500	9360	Bevan and Hodgkinson.
Birch	15100	6402	Hodgkinson.
,, American black	...	11660	,,
Bitter-wood, West Indian	...	5510	Fowke.
Blue-Mahoe ,,	...	8820	,,
Blue-gum, Australian	...	8820	,,
Box, European	20000	10300	Bevan and Hodgkinson.
,, Australian	...	8820	Fowke.
Broadleaf, West Indian	...	7720	,,
Brown-ebony, Guianaian	...	12566	,,
Buckati ,,	...	9920	,,
Buhuradda ,,	...	12120	,,
Bullet-tree, West Indian	...	14330	,,
,, bastard ,,	...	11020	,,
,, red ,,	...	9920	,,
Cabacalli, Guianaian	...	9920	,,
Cabbage-bark, West Indian	...	9920	,,
Calabash ,,	...	5510	,,
Cedar of Lebanon, North African	11400	5860	Bevan and Hodgkinson.
Chestnut, European	10500	...	Bevan.
Chow, West Indian	6780	12100	—
Cogwood, West Indian	...	12120	Fowke.
Crab-tree, English	19800	6980	—
Cypress, European	6000	...	—
Dantzic-fir ,,	3400	3000	—
Deal ,,	10000	5000	—
Dogwood, West Indian	...	11020	Fowke.
Ducaliballi, Guianaian	...	13220	,,
Ebony, West Indian	...	18960	,,
Elder, European	10230	8466	Hodgkinson.
Elm, English	5200	...	—
,, European	14400	10330	Bevan and Hodgkinson.

Table 105 *continued.*—Breaking Strength of Timber.

Description.	Tensile Strength in lbs. per Square Inch.	Compressive Strength in lbs. per Square Inch.	Authority.
Fiddle-wood, West Indian	...	6610	Fowke.
Fir, red-pine, Norway	14300	5375	Bevan and Hodgkinson.
,, Norway spruce	12400	...	Bevan.
,, American pitch-pine	7800	...	—
,, larch, Northern Europe	10000	5500	—
French oak	7700	8000	—
Fustic, West Indian	...	12120	Fowke.
Greenheart, yellow, Guianaian	8000	12120	,,
,, black ,,	...	15430	,,
Hawthorn, European	10500	...	Bevan.
Hazel ,,	18000	...	,,
Hickory, Australian	...	7050	Fowke.
Holly, European	16000	...	Bevan.
Hornbeam ,,	20240	8500	Bevan and others.
Iron-bark, Australian	...	9920	Fowke.
,, rough-leaved, Australian	...	13220	,,
Ironwood, West Indian	...	17630	,,
Kakaralli, Brazilian	...	13220	,,
Kauri, New Zealand	3900	11000	—
Laburnum, European	10500	...	Bevan.
Lancewood, West Indian	23400	6610	Bevan and Fowke.
Letterwood ,,	...	14100	Fowke.
Lignumvitæ ,,	11780	9860	—
Lime-tree, European	23500	...	Bevan.
Locust, North American	16000	...	,,
Mahogany, Spanish, West Indian	21000	8198	Bevan and Hodgkinson.
,, Honduras, Central America	2800	6000	—
Maple	5100	...	—
Mora	...	9920	Fowke.
Mountain-Ash, Australian	...	11010	,,
Oak, British, average	10000	10055	Bevan and Hodgkinson.
,, ,, very strong	19800	...	Bevan.
,, Dantzic	12650	7650	—
,, red, North American	10000	5800	—
Orange-tree, wild, West Indian	...	13230	Fowke.
Pear-tree, European	13000	...	Bevan.
Pine, pitch, European	3970	6000	—

Table 105 *continued.*—BREAKING STRENGTH OF TIMBER.

Description.	Tensile Strength in lbs. per Square Inch.	Compressive Strength in lbs. per Square Inch.	Authority.
Pine, yellow, European	2340	3860	—
Plane, occidental, North American	11700	...	Bevan.
„ common, European	...	...	—
Plum-tree „	11300	9300	—
Poplar „	7200	5124	Bevan and Hodgkinson.
Quassia, West Indian	...	5510	Fowke.
Rock-elm, American	8700	8200	—
Sabicu, West Indian	4980	7965	—
Satin-wood, West Indian	...	12560	Fowke.
Saul, Asian	9640	...	—
Silverballi, Guianaian	...	7716	Fowke.
Small-leaf, West Indian	...	15430	„
Snakewood „	...	14000	—
Spruce	3820	...	—
Sweetwood, West Indian	...	9920	Fowke.
Sycamore, European	...	...	—
Teak, South-Eastern Asian	15000	12100	Bevan and Hodgkinson.
Wallaba, Guianaian	...	6614	Fowke.
Walnut, Western Asian	8000	6600	—
Water-gum, Australian	...	11020	Fowke.
White-cedar, Guianaian	...	9920	„
Willow, European	...	...	
Yellow-sanders, West Indian	...	6614	Fowke.
Yew, European	8000	...	Bevan.

Table 106.—SPECIFIC GRAVITY OF TIMBER.

Description.	Specific Gravity. Water = 1.	Description.	Specific Gravity. Water = 1.
Abele	·515	Beech	·687
Acacia	·710	Birch	·710
African oak	·985	„ Canadian	·603
Alder	·558	Bitterwood	·557
Apple-tree	·790	Blackbutt, Australian	·780
Ash	·755	Blue-gum	·910
Aspen	·600	Blue-mahoe	·535
Bamboo	·400	Box	·963
Baracara	·808	Broadleaf	·770
Bartaballi	·642	Brown-ebony (Guianaian)	1·030
Baywood	·560	Buckati	·810

Table 106 *continued.*—SPECIFIC GRAVITY OF TIMBER.

Description.	Specific Gravity. Water = 1.	Description.	Specific Gravity. Water = 1.
Buhuradda	·815	Locust	·710
Bullet-tree	1·027	Mahogany, Spanish	·850
Cabacalli	·890	Maire, black	1·160
Cabbage-bark	·943	Manuka, New Zealand	·944
Calabash	·555	Maple	·670
Cedar of Lebanon	·487	Mora	·920
„ Canadian	·400	Moreton-Bay or hoop-pine	·470
Cherry-tree	·700	Mountain-ash, Australian	1·070
Chestnut	·535	Mulberry	·870
Chow	1·110	Oak, British, average	·935
Cogwood	·960	Olive-tree	·690
Cowrie, New Zealand	·580	Orange, wild, West Indian	·890
Crab-tree	·800	Pear-tree	·690
Cypress	·658	Pine, pitch	·663
Dantzic-fir	·590	„ red	·655
Deal, white	·460	„ white, Canadian	·446
„ yellow	·550	Plane	·642
Dogwood	·935	Plum-tree	·780
Ducaliballi	·907	Poplar	·515
Ebony, West Indian	1·190	Quassia	·557
Elder	·700	Rata, ironwood	1·046
Elm	·550	Rimu, red-pine, New Zealand	·564
Fiddle-wood	·710	Rock-elm	·750
Fir, red pine	·680	Sabicu	·912
„ American pitch-pine	·700	Saul	·950
„ red, Canadian	·546	Silverballi	·550
„ larch	·530	Small-leaf, West Indian	1·165
Flooded-gum	·680	Spruce	·482
Fustic	·960	„ Canadian	·416
Greenheart, yellow	1·050	Stringy-bark, Australian	·860
Hawthorn	·912	Sweet-wood	·968
Hazel	·863	Sycamore	·595
Hemlock, Canadian	·504	Tallow-wood, Australian	·790
Hickory	·750	Teak, Indian	·770
Holly	·760	„ African	·985
Hornbeam	·765	Totara, New Zealand	·560
Iron-bark	1·030	Wallaba	1·030
Iron-wood	·985	Walnut	·675
Kauri	·557	Water-gum	·998
Kowhai, New Zealand	·885	White-cedar	·510
Laburnum	·924	Willow	·400
Lancewood	·950	Yellow-Sanders, W. Ind.	·865
Letterwood	·994	Yew	·800
Lignumvitæ	1·327		
Lime-tree	·765		

Table 107.—Specific Gravity of Cast-iron, Wrought-iron, Steel, Bronze, Brass, Tin, Lead, Zinc, and Other Metals.

Description.	Specific Gravity. Water = 1.	Description.	Specific Gravity. Water = 1.
Lithium	·58	Alloy, copper 1, tin 1	8·00
Potassium	·84	Manganese	8·04
Sodium	·95	Pure iron by electro-deposit (Dr. Percy)	8·14
Rubidium	1·50	Bell-metal (small bells)	7·50
Calcium	1·60	White-metal	7·74
Magnesium	1·75	Muntz-metal	8·22
Glucinum	2·15	Yellow brass	8·33
Strontium	2·50	Alloy, copper 2, zinc 1	8·34
Aluminium, cast	2·60	Brass, cast, good	8·40
„ wrought	2·70	Brass-wire	8·55
Titanium	5·35	Cobalt	8·55
Arsenic	5·84	Nickel, cast	8·30
Chromium	6·06	„ hammered	8·70
Tellurium	6·15	Ruthenium	8·64
Antimony	6·76	Gun-metal	8·65
Zinc, cast	6·90	Bronze	8·78
„ sheet	7·20	Copper, cast	8·70
Alloy, tin 3, zinc 1	7·22	„ sheet	8·80
Cast-iron, weak	6·95	„ hammered bars	8·88
„ average	7·12	„ wire	8·92
„ strong	7·30	Molybdenum	8·66
„ white	7·50	Cadmium	8·74
Pewter	7·32	Bismuth	9·86
Tin	7·40	Osmium	10·10
Speculum-metal	7·45	Alloy, copper 1, lead 1	10·35
Aluminium-bronze	7·67	Silver, cast	10·48
Wrought-iron, common bars	7·50	„ hammered	10·58
Wrought-iron, puddled slab	7·56	„ wire	10·65
Wrought-iron rails	7·60	Rhodium	10·68
„ bars, average good	7·70	Lead, cast	11·30
Wrought-iron, Lowmoor	7·80	„ pipe	11·36
Steel, common	7·30	„ sheet	11·42
„ blister	7·80	Palladium	11·87
„ compressed	7·84	Thallium	12·00
„ Bessemer	7·85	Mercury	13·58
„ Siemens-Martin	7·85	Tungsten	17·70
„ crucible	7·86	Uranium	18·50
„ cast	7·86	Iridium	18·75
Homogeneous metal	7.89	Gold, cast	19·25
		„ hammered	19·37
		Platinum	21·00

Table 108.—Specific Gravity of various Materials and Liquids.

Description.	Specific Gravity. Water = 1.	Description.	Specific Gravity. Water = 1.
Red iron-ore	5·21	Lancashire coal, Haydock Rushy Park	1·32
Magnetic iron-ore	5·00	„ Wigan cannel	1·23
Basalt . . 2·47 to	3·00	„ Wigan 4 feet	1·20
Granite	2·95	Newcastle coals, West Hartley main	1·30
Mica	2·85	„ Broomhill	1·25
Puzzolano	2·84	Derbyshire coal, Staveley	1·27
Limestone	2·84	„ Parkgate	1·30
Slate	2·83	Welsh coal, Merthyr	1·30
White Parian-marble	2·83	„ Ebbw Vale	1·27
Green-marble	2·75	„ Watney's anthracite	1·39
White Carrara-marble	2·73	Scotch coal, Eglington	1·25
Florentine-marble	2·50	„ Dalkeith	1·27
Jasper	2·75	„ Kilmarnock	1·24
Alabaster	2·75	Patent coal, Warlich's	1·15
Plate-glass	2·70	„ Livingstone's	1·14
Pure rock-crystal	2·66	„ Wylam's	1·10
Quartz	2·65	Average of a number of samples of Forest of Dean coal	1·29
Purbeck-stone	2·60	Average of a number of coals from various districts	1·31
Flint	2·59	Mud-deposit from harbour	1·50
Portland-stone	2·58	Plaster, cast	1·29
Chalk	2·55	Wine	·99
Crown-glass	2·50	Resin-oil	·98
Paving quartz	2·50	Castor-oil	·96
Yorkshire paving-stone	2·47	Wax	·95
Common glass	2·45	Gunpowder	·94
Lias	2·40	Whalebone	·94
Rock-salt	2·22	Ice	·93
Bath-stone	2·20	Linseed-oil	·93
Graphite	2·16	Sweet almond-oil	·93
Sandstone	2·12	Cotton-seed oil	·92
Potash	2·08	Whale-oil	·92
Sulphur	2·00	Lard-oil	·91
Clay	1·91	Neatsfoot-oil	·91
Ivory	1·91	Nut-oil	·91
Common salt, solid	1·90		
Sand, pure	1·90		
Tiles	1·84		
Stone, mean of various	2·00		
Brick	2·00		
Brickwork	1·77		
Mortar	1·70		
Lancashire coal, Ince Hall Py	1·34		

Table 108 *continued.*—SPECIFIC GRAVITY OF VARIOUS MATERIALS AND LIQUIDS.

Description.	Specific Gravity. Water = 1.	Description.	Specific Gravity. Water = 1.
Olive-oil	·91	Sea-water, ordinary . .	1·026
Rape-seed oil . . .	·91	Nitric-acid . . .	1·271
Petroleum	·89	Sulphuric-acid . . .	1·840
Sperm-oil . . .	·88	Muriatic-acid . .	1·200
Naphtha	·85	Bromine	2·970
Benzine	·83	Fluoric-acid . . .	1·060
Wood-spirit . . .	·80	Citric-acid . . .	1·034
Coke, hard . . .	·90	Essential oil of cinnamon	1·043
Peat, hard . . .	·80	„ of lavender .	·894
„ soft	·50	„ of turpentine	·870
Wood-charcoal . .	·24	„ of amber .	·868
Water of the Baltic . .	1·015	Alcohol . . .	·835
Water of the Dead Sea .	1·24	Ether, nitric . . .	·908
Milk	1·032	Proof spirit . . .	·922
Cider	1·018	Vinegar	1·009

Table 109.—SPECIFIC GRAVITY OF GASES AND VAPOURS AT 32° FAHR. UNDER ONE ATMOSPHERE OF PRESSURE.

Description.	Specific Gravity. Air = 1.	Description.	Specific Gravity. Air = 1.
Vapour mercury, ideal .	6·974	Carbonic-acid gas . .	1·527
Vapour of bromine . .	5·542	Nitrous-oxide gas . .	1·527
Chloroform . . .	5·297	Oxygen	1·106
Vapour of turpentine .	4·694	Air	1·000
Hydriodic-acid gas .	4·340	Nitrogen	·974
Acetic-ether . . .	3·000	Carbonic-oxide . .	·968
Vapour of benzine .	2·695	Olefiant-gas . . .	·967
Vapour of sulphuric ether	2·579	Prussic-acid gas . .	·937
		Steam of water . . .	·623
Vapour of ether . .	2·557	Ammoniacal-gas . .	·590
Chlorine-acid gas . .	2·500	Light carburetted hydrogen	·553
Sulphurous-acid gas .	2·500	Coal-gas	·438
Cyanogen . . .	1·805	Hydrogen . . .	·069
Alcohol	1·613		

The Specific Gravity of a Body may be found by dividing the weight in lbs. of a cubic foot of the body by 62·355, the weight in lbs. of a cubic foot of pure water at 62° Fahr.

The Specific Gravity of a Solid Body heavier than Water, may be found by weighing it in the air, and again immersed in pure water at

62° Fahr., and dividing the weight in the air by its buoyancy, or the loss of weight when immersed in the water. The quotient is the specific gravity.

The Specific Gravity of a Solid Body lighter than Water may be found as follows:—Weigh the body in the air, then load it with a body heavier than water, and large enough to sink the light body, and weigh them together in water; also weigh the heavy body separately in air and in water. Subtract the buoyancy of the heavy body from that of the two bodies together, the remainder will be the buoyancy of the light body singly, by which its weight in air is to be divided. The quotient is the specific gravity.

The Specific Gravity of a Solid Body which is soluble in Water may be found as follows:—Weigh it in a liquid in which it is not soluble, divide its weight in air by the loss of weight in the liquid, and multiply the quotient by the specific gravity of the liquid. The product is the specific gravity required.

The Specific Gravity of a Liquid may be found approximately as follows:—Weigh a solid body in the given liquid, also in air and in water, divide the buoyancy, or loss of weight in the liquid, by the buoyancy or loss of weight in water. The quotient is the specific gravity.

The Weight of a Cubic Foot of a Body may be found by multiplying its specific gravity by 62·355.

Table 110.—WEIGHT IN LBS. OF ONE CUBIC FOOT OF VARIOUS METALS.

Description.	Weight of One Cubic Foot.	Description.	Weight of One Cubic Foot.
	lbs.		lbs.
Lithium	36	Steel-plates	489
Potassium	54	Steel, average	490
Sodium	61	Homogeneous metal	492
Rubidium	94	Manganese	498
Calcium	99	Pure iron, by electro deposit (Dr. Percy)	508
Magnesium	109	Bell-metal (small bells)	509
Glucirum	130	White-metal	511
Strontium	159	Muntz-metal	512
Aluminium, cast	159	Yellow brass	520
„ wrought	168	Brass, cast	527
Titanium	332	Brass-wire	531
Arsenic	362	Cobalt	532
Chromium	376	Nickel, cast	515
Tellurium	380	„ hammered	540
Antimony	420	Ruthenium	535
Zinc, cast	428	Gun-metal	540
„ sheet	450	Bronze	545
Cast-iron	450	Copper, cast	538
Pewter	454	„ sheet	548
Tin	456	„ hammered bars	555
Speculum-metal	464	„ wire	554
Aluminium-bronze	480	Molybdenum	535
Wrought-iron	480		

Table 110 *cont.*—WEIGHT IN LBS. OF ONE CUBIC FOOT OF VARIOUS METALS.

Description.	Weight of One Cubic Foot.	Description.	Weight of One Cubic Foot.
	lbs.		lbs.
Cadmium	542	Thallinum	743
Bismuth	617	Mercury, fluid	848
Osmium	624	,, solid	977
Type-metal	654	Tungsten	1096
Silver, cast	656	Uranium	1146
,, standard	658·4	Iridium	1166
,, hammered	660	Gold, standard	1102·9
Rhodium	661	Gold	1203
Lead	708	Gold, hammered	1210
Palladium	736	Platinum	1340

Aluminium is a remarkably light metal, its specific gravity being only about one-third that of steel. It is strong, ductile, easily wrought, and not liable to rust. The addition of a small quantity of aluminium to gun-metal and steel, greatly increases their strength. The tenacity of aluminium-bronze depends upon the purity of its copper; it averages from 30 to 36 tons per square inch with 5 per cent. of aluminium, and from 40 to 48 tons per square inch with 10 per cent. of aluminium.

Table 111.—WEIGHT IN LBS. OF ONE CUBIC FOOT OF STONE AND MINERAL SUBSTANCES.

Description.	Weight of One Cubic Foot Solid.	Description.	Weight of One Cubic Foot Solid.
	lbs.		lbs.
Coal, bituminous	80	Flint and felspar	165
,, anthracite	100	Rock-crystal	166
Earth and plaster	90	Quartz and talc	169
Sand, dry	90	Glass, crown	156
,, damp	120	,, plate	169
Portland cement	92	,, flint	187
Mud	105	Marble, Egyptian green	167
Mortar	109	,, Carrara	170
Brickwork	112	Porphyry and trap	170
Marl and clay	119	Limestone	170
Sulphur and tiles	125	,, magnesian	179
Concrete	128	Granite	170
Potash	130	,, grey	190
Rock-salt	136	Slate	178
Masonry	140	Serpentine	176
Sandstone	157	Basalt	188
Chalk	159	Iron ore, brown	245
Shale	162	,, red	328

Table 112.—WEIGHT AND BULK OF STONE.

Description.	Weight of One Cubic Yard.	Number of Cubic Feet in One Ton.
	lbs.	
Chalk	3170	$19\frac{1}{4}$
,,	4750	13
Sandstone	3900	16
,,	2200	14
Shale	4300	$13\frac{1}{2}$
Felspar	4370	$13\frac{3}{4}$
Flint	4450	$13\frac{1}{2}$
Limestone	4560	$13\frac{1}{4}$
,,	4730	$12\frac{3}{4}$
,, Magnesian	4800	$12\frac{1}{2}$
Quartz	4450	$13\frac{1}{2}$
Granite	4450	$13\frac{3}{4}$
,,	4650	13
Basalt	5070	12

Table 113.—WEIGHT IN LBS. OF ONE CUBIC FOOT OF DRY TIMBER.

Description.	Weight of One Cubic Foot.	Description.	Weight of One Cubic Foot.
	lbs.		lbs.
Elder-pith	5	Sycamore	35
Charcoal from pine	13	Silverballi	35
Charcoal from oak	15	Calabash	35
Cork	15	Bitterwood	35
Poplar, Lombardy	24	Alder	35
Bamboo	25	White-deal, English	35
Willow	26	Mahogany, Cuban	35
Pinaster	27	Red-pine, Norway	36
Poplar, Abele	27	Totara, New Zealand	36
Lime-tree	28	Cedar, Jamaica	37
Yellow-pine, American	29	Cowrie	37
Weymouth-pine	29	Rimu, New Zealand	37
White-spruce, American	29	Crabwood, Guiana	38
Red-pine, Russia	30	Vine-tree	38
Larch	31	Aspen-tree	38
Cedar of Lebanon	31	Spanish-chestnut	38
Norway-spruce	32	Poon	40
Chestnut	34	Swamp-oak	40
Blue-mahoe	34	Bartaballi	40
Elm	34	Walnut, American	40
Memel-fir	34	Jungle-teak, Indian	40
Baywood	35	Yacca	40

Table 113 *cont.*—Weight in lbs. of One Cubic Foot of Dry Timber.

Description.	Weight of One Cubic Foot.	Description.	Weight of One Cubic Foot.
	lbs.		lbs.
Plane-tree, occidental	40	Buckati	51
,, common	41	Yew	51
Maple	41	American white-oak	52
Cypress	41	Spanish-mahogany	53
Green - Mahogany, Ja'	42	Beech, Jamaica	53
Jarrow	42	Burute	53
Kullum	42	Hazel	54
Pitch-pine	42	Yellow-sanders, Jamaica	54
Riga-masts	42	Prune-tree ,,	54
Service-tree	42	Stringy-bark	54
Walnut, brown	43	Oak, American	54
Lancewood	43	Bibla	54
Locust-tree	43	Kaieeriballi	54
Purple-heart	43	Sissoo	54
Olive-tree	43	Tamarind, Jamaica	55
Prickle, yellow	43	Teak, Indian	55
Cos	43	Cashaw	55
Beech	44	Blue-gum	55
Lemon-tree	44	Blackbutt, Australian	56
Cartan	44	Kowhai, New Zealand	56
White-cedar	44	Mulberry	56
Orange-tree	45	Blackwood	56
Fiddle-wood	45	Ducaballi	56
Locust-tree, Guiana	45	Gymp	56
Birch	45	Ironbark, rough, Aust'l'n	57
Teak, China	45	Cabacalli	57
Cherry-tree	45	Tallow-wood	57
Pear-tree	46	Wild orange-tree, Jamaica	57
Apple-tree	46	Hawthorn	57
Crab-tree	47	Gray-gum	57
Hickory	47	Mora	57
Fir, Norway pine	47	Laburnum	58
Dantzic-oak	47	Oak, English	58
Oak sap-wood	47	Walnut, green	58
Hornbeam	48	Monkey-pot	58
Holly	48	Mahogany, Australian	58
Green-elm	48	Manuka, New Zealand	59
Moreton Bay, Aust'l'n	48	Dogwood	59
Matai, New Zealand	49	Flooded-gum, Aust'l'n	60
Sweet orange-tree	50	Puriri, New Zealand	60
Rock-elm	50	Sabicu	60
Baracara	50	Satin-wood	60
Acacia	51		

Table 113 *cont.*—WEIGHT IN LBS. OF ONE CUBIC FOOT OF DRY TIMBER.

Description.	Weight of One Cubic Foot.	Description.	Weight of One Cubic Foot.
	lbs.		lbs.
Cog-wood	60	Paloo	66
Mee	60	Rata, New Zealand	66
Sal	60	Greenheart	67
True-box	60	Kakaralli	68
Sweet-wood, Jamaica	61	Bastard-box	70
Fustic ,,	61	Mountain-ash, Australian	70
Oak, European	61	Ebony	71
Tonka	61	Caluvere	72
Erroul	61	Small-leaf	72
Iron-wood	62	Lignumvitæ, Jamaica	73
Teak, African	62	Ironbark, white, Aust'l'n	73
Woolly-butt	62	Box, Ilwarra	73
Letter-wood, Guiana	63	Maire, black	73
Lance-wood . 42 to	63	Oak, heart of	73
Water-gum	64	Oak, black	75
Wallaba	64	Ebony, black	75
Brown ebony, Guiana	65	,, green	76
Boxwood	65	Boxwood of Holland	83
Rosewood	65	Lignumvitæ . 41 to	83
Bullet-tree	65	Pomegranate	85

Table 114.—WEIGHT OF LIQUIDS AT 32° FAHR.

Description.	Weight of One Cubic Foot.	Weight of One Gallon.
	lbs.	lbs.
Ether, sulphuric	44·9	7·2
Alcohol, pure	49·3	7·9
,, proof spirit	57·4	9·2
Wood, spirit	49·9	8·0
Oil, linseed	58·7	9·4
,, olive	57·1	9·15
,, whale	57·4	9·2
,, turpentine	54·3	8·7
Naphtha	53·1	8·5
Petroleum	54·9	8·8
Wine, Burgundy	61·9	9·9
Sea-water, ordinary	64·05	10·3
Milk	64·30	10·3
Acetic acid, maximum concentration	67·4	10·8
Nitric acid	76·2	12·2
Water of the Dead Sea	77·4	12·4
Sulphuric acid, maximum concentration	114·9	18·4
Mercury	848·75	136·0

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