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The Powering of Ships.

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I. INTRODUCTION.

ONE of the principal features of the author's work as the Superintendent Engineer of new construction of a large shipping company is concerned with the powering of new vessels and the design and construction of the main propelling and auxiliary machinery.

The Institute of Marine Engineers is publishing a series of articles by the author describing the method adopted for determining the shaft horsepower required to meet certain specified conditions and also the design of a high-pressure single-reduction geared turbine installation and its auxiliaries.

When a new vessel is proposed it is necessary for specification purposes to define the powers, speeds, steam and fuel consumptions on a given displacement, which the various builders who are asked to tender will be required to guarantee.

The first part of the present article will be devoted to the formulæ required for determining the resistance, effective horsepower and shaft horsepower, and the latter part to the application of the method to a proposed vessel.

The writer is a strong advocate of tank tests and the specification should always require the approved builders to have E.H.P. and self-propulsion tank tests carried out.

II. THE POWERING OF SHIPS.

(2) On the Resistance and Propulsion of Ships.

The two principal elements in the resistance of a ship are the *frictional resistance* and the *residuary or wave and eddy making resistance.*

The frictional resistance may be determined from the law put forward by the late William Froude, *viz.,*

$$R_F = f.S.V^n \quad \dots \quad (1)$$

where, R_F = frictional resistance in pounds.

f = coefficient of friction in salt water, which is dependent upon length.

S = wetted surface of hull in square feet.

V = speed of vessel in knots.

n = an index determined from Froude's experiments as 1·83.

TABLE 1.

From Froude's experiments on planks corrected for salt water *"f"* may be taken as follows :—

Length in feet	*"f"*	*Length in feet*	*"f"*
100	·009207	450	·008802
120	·009135	500	·008776
140	·009085	550	·008750
160	·009046	600	·008726
180	·009016	700	·008680
200	·008992	800	·008639
250	·008923	900	·008608
300	·008902	1000	·008574
350	·008867	1100	·008548
400	·008832	1200	·008524

The wetted surface *"S"* can only be accurately calculated when the lines of the vessel are available but a near approximation of the surface may be made from the formulæ given below.

Denny's formula $S=1{\cdot}7\,L.d.+\dfrac{V}{d}$ ... (2)

D. W. Taylor's formula $S=15{\cdot}4\sqrt{D.L.}$ ... (3)

R. E. Froude's formula $S=V^{\frac{2}{3}}\left(3{\cdot}4+\dfrac{L}{2.V^{\frac{1}{3}}}\right)$ (4)

where, D = displacement in tons; V = volume of displacement in cubic feet; L = length of ship on load water line in feet.
d = moulded draft in feet.

The constants given in the formulæ above are as given by their respective authors but the writer has found in modern practice that the surfaces obtained are too low.

From a number of recent vessels, the mean constants are as follows :—

Denny's formula $S=1{\cdot}86\,L.d.+\dfrac{V}{d}$

Taylor's ,, $S=16{\cdot}25\sqrt{D.L.}$

Froude's ,, $S=V^{\frac{2}{3}}\left(3{\cdot}7+\dfrac{L}{2.V^{\frac{1}{3}}}\right)$

(3) Residuary Resistance.

Residuary resistance does not lend itself to formulæ calculation and recourse has to be made to model experiments. In this connection Admiral D. W. Taylor, of the United States Navy, has carried out a classic series of *experiments on 80 models having the following characteristics.

$$\frac{\text{Moulded beam}}{\text{Moulded draft}}=2{\cdot}25 \text{ and } 3{\cdot}75$$

$$\frac{\text{Displacement in tons}}{\left(\dfrac{\text{Length on waterline in feet}}{100}\right)^{3}}=26{\cdot}6,\ 53{\cdot}2,\ 79{\cdot}81,\ 133{\cdot}02 \text{ and } 199{\cdot}52$$

$$\text{Prismatic coefficient}=\frac{\text{Volume of displacement in cubic feet}}{\left(\begin{array}{c}\text{Area of midship}\\ \text{section in sq. ft.}\end{array}\right)\left(\begin{array}{c}\text{Length in feet}\\ \text{between perpendiculars}\end{array}\right)}$$

$=·48,\ ·52,\ ·56,\ ·60,\ ·64,\ ·68,\ ·74$ and $·80$.

*"Speed and Power of Ships", by D. W. Taylor.

For each beam-draft ratio and prismatic coefficient a series of residuary resistance curves per ton of displacement were plotted for speed-length ratios from 0·6 to 2·0. The speed-length ratio being,

$$\frac{\text{Speed in knots}}{\sqrt{\text{Length on waterline in feet}}}=\frac{V}{\sqrt{L}}\ \dots\ \dots\ (5)$$

The residuary resistance in Taylor's series is plotted per ton of displacement and to obtain the total resistance per ton of displacement R_F equation (1) above has to be divided by the displacement. Hence,

$$R_t=R_f+R_r\ \dots\ \dots\ \dots\ \dots\ \dots\ (6)$$

where, $R_f=\dfrac{R_F}{D}$

Having found the total resistance per ton of displacement we are in a position to calculate the effective horsepower or tow rope horsepower of the bare hull without appendages, as follows :—

$$\text{E.H.P.}=\frac{R_t\times D\times V\times 6{,}080}{60\times 33{,}000}=·0030707\,R_t\,D.V.\quad(7)$$

where, D = displacement in tons; V = speed of ship in knots.
1 knot = 6,080ft.

The E.H.P. is calculated for the full range of speeds on the estimated trial, service and load displacements.

Since the block coefficient and prismatic coefficient are required to enable the residuary resistance to be assessed, it is necessary to approximate these for the reduced displacements.

(4) On The Block Coefficient at reduced Displacements.

From the author's investigations the block coefficient at reduced displacements does not appear to have been given by any of the authorities consulted and the writer therefore puts forward the following.

Over 40 ships of various types were taken for this investigation and the actual block coefficients at various displacements were calculated and plotted on a base of displacement. In every case the curve obtained was an inclined straight line obeying the law

$$\beta=A+b.D.\ \dots\ \dots\ \dots\ \dots\ \dots\ (8)$$

where, β = block coefficient, A = a constant and D = displacement in tons.

A and b were found to vary for block coefficients, between 0·56 and 0·73, with the length and breadth of the vessel. For block coefficients of cargo ships of ·76 to ·77, A was found to be constant at ·693 whilst b varied between 3·34 and 5·9.

The following are the results obtained.

Class (1) : Length 400ft.; Breadth 58ft.; Load block coefft. = ·56

$$\beta=·4259+\frac{22{\cdot}2\,D}{1{,}000{,}000}$$

Class (2) : L = 520ft.; B = 74ft.; Load block coefft. = ·5725

$$\beta=·440+\frac{8{\cdot}25\,D}{1{,}000{,}000}$$

Class (3) : L = 530ft.; B = 73ft.; Load block coefft. = ·646

$$\beta=·507+\frac{6{\cdot}6\,D}{1{,}000{,}000}$$

Class (4) : L=630ft.; B=82ft.; Load block coefft.=·646

$$\beta = \cdot 484 + \frac{5\cdot 7\, D}{1{,}000{,}000}$$

Class (5) : L=585ft.; B=76ft.; Load block coefft.=·654

$$\beta = \cdot 5216 + \frac{5\cdot 7\, D}{1{,}000{,}000}$$

Class (6) : L=518·5ft.; B=71ft.; Load blk. coefft.=·6575

$$\beta = \cdot 527 + \frac{6\cdot 5\, D}{1{,}000{,}000}$$

Class (7) : L=530ft.; B=60ft.; Load block coefft.=·685

$$\beta = \cdot 574 + \frac{6\cdot 225\, D}{1{,}000{,}000}$$

Class (8) : L=580ft.; B=67ft.; Load block coefft.=·6875

$$\beta = \cdot 547 + \frac{6\cdot 35\, D}{1{,}000{,}000}$$

Class (9) : L=547ft.; B=71ft.; Load block coefft.=·70

$$\beta = \cdot 59 + \frac{5\cdot 1\, D}{1{,}000{,}000}$$

Class (10) : L=525ft.; B=70ft.; Load block coefft.=·71

$$\beta = \cdot 60 + \frac{5\cdot 15\, D}{1{,}000{,}000}$$

Class (11) : L=600ft.; B=73ft.; Load block coefft.=·73

$$\beta = \cdot 607 + \frac{4\cdot 5\, D}{1{,}000{,}000}$$

Class (12) : L=450ft.; B=58ft.; Load block coefft.=·74

$$\beta = \cdot 63 + \frac{6\cdot 9\, D}{1{,}000{,}000}$$

Class (13) : L=400ft.; B=52ft.; Load block coefft.=·76

$$\beta = \cdot 693 + \frac{5\cdot 9\, D}{1{,}000{,}000}$$

Class (14) : L=480ft.; B=58ft.; Load block coefft.=·77

$$\beta = \cdot 693 + \frac{4\cdot 535\, D}{1{,}000{,}000}$$

Class (15) : L=520ft.; B=64ft.; Load block coefft.=·77

$$\beta = \cdot 693 + \frac{3\cdot 34\, D}{1{,}000{,}000}$$

In view of the straight line law for the block coefficient, A and b were assumed to vary as a straight line law for ships of similar load block coefficient but of different lengths and breadths. The ratio of the beam to the load draft for the vessels mentioned above varied from 2·0 to 2·8 but the draft could not be included in the formulæ as there would have been two unknowns. It was thought that, for approximating the block coefficient at reduced displacements, only the length and breadth should be taken into account. The following are the formulæ obtained.

For ships 400ft. by 58ft. to 520ft. by 74ft. Load β 0·56 to 0·57

$$\beta = \cdot 4044 + \frac{\cdot 926\, L.B.}{1{,}000{,}000} + \frac{(43\cdot 5 - \cdot 00092\, L.B.)\, D}{1{,}000{,}000} \qquad (1\ \&\ 2)$$

530ft. by 73ft. to 630ft. by 82ft. Load block coefft. 0·64 to 0·65

$$\beta = \cdot 576 - \frac{1\cdot 78\, L.B.}{1{,}000.000} + \frac{(9\cdot 3 - \cdot 00007\, L.B.)\, D}{1{,}000{,}000} \qquad (3\ \&\ 4)$$

520ft. by 71ft. to 600ft. by 76ft. Load block coefft. 0·65 to 0·66

$$\beta = \cdot 5528 - \frac{0\cdot 7\, L.B.}{1{,}000.000} + \frac{(10\cdot 325 - \cdot 000104\, L.B.)\, D}{1{,}000{,}000} \qquad (5\ \&\ 6)$$

520ft. by 60ft. to 600ft. by 70ft. Load block coefft. 0·68 to 0·69

$$\beta = \cdot 693 - \frac{3\cdot 75\, L.B.}{1{,}000.000} + \frac{(5\cdot 7375 - \cdot 000015\, L.B.)\, D}{1.000{,}000} \qquad (7\ \&\ 8)$$

520ft. by 70ft. to 550ft. by 71ft. Load block coefft. 0·70 to 0·71

$$\beta = \cdot 776 - \frac{4\cdot 79\, L.B.}{1{,}000.000} + \frac{(6\cdot 03 - \cdot 000024\, L.B.)\, D}{1{,}000{,}000} \qquad (9\ \&\ 10)$$

450ft. by 58ft. to 600ft. by 73ft. Load block coefft. 0·73 to 0·74

$$\beta = \cdot 664 - \frac{1\cdot 3\, L.B.}{1{,}000.000} + \frac{(10\cdot 45 - \cdot 000136\, L.B.)\, D}{1{,}000{,}000} \qquad (11\ \&\ 12)$$

400ft. by 52ft. to 520ft. by 64ft. Load block coefft. 0·76 to 0·77

$$\beta = \cdot 693 + \frac{(10\cdot 19 - \cdot 000206\, L.B.)\, D}{1{,}000{,}000} \quad \ldots \quad \ldots \qquad (13\text{-}15)$$

where, β=block coefficient; L=length between perpendiculars; B=moulded breadth; D=displacement in tons.

(5) Definition of Ships' Coefficients.

$$\text{Block coefficient} = \frac{\text{Volume of displacement in cubic feet}}{L \times B \times d} \qquad (9)$$

$$\text{Prismatic coefficient} = \frac{\text{Volume of displacement in cubic feet}}{(\text{Area of midship section to waterline in square feet}) \times L} \qquad (10)$$

$$\text{Mid area coefficient} = \frac{\text{Area of midship section to waterline in square feet}}{B \times d} \qquad (11)$$

$$= \frac{\text{Block coefft.}}{\text{Prismatic coefft.}} \quad \ldots \quad \ldots (12)$$

where, L=length in feet between perpendiculars
B=moulded breadth in feet
d=draught in feet

To approximate the prismatic coefficient at reduced displacements, it is sufficiently near to assume the mid area coefficient constant, since the variation of this coefficient is small in normal ship forms over a wide range of displacement.

(6) Determination of Shaft Horsepower.

The S.H.P. required to propel the bare hull at any given speed is found by dividing the E.H.P. by the propulsive efficiency.

$$\text{Propulsive efficiency} = \frac{\text{E.H.P. bare hull}}{\text{S.H.P.}} \quad \ldots \quad \ldots (13)$$

$$= \frac{\text{Resistance in pounds} \times V \times \frac{6.080}{60}}{\text{Torque in lb. ft.} \times 2\pi \times \text{revs. per minute}} \quad \ldots (14)$$

This may be written as follows :—

$$\frac{\text{E.H.P.}}{\text{S.H.P.}} = \frac{(R)}{(T)}\,\frac{(V)}{(V_1)}\,\frac{(T.V_1)}{(S_1)}\,\frac{(S_1)}{(S)}\,\frac{(101\cdot 33)}{(33{,}000)} \quad \ldots (15)$$

$$\text{Now, } \frac{R.V}{T.V_1} = \frac{E.H.P.}{T.H.P.} = \text{hull efficiency} = n_H \quad \ldots \quad \ldots (16)$$

$$\frac{101\cdot 33.T.V_1}{33{,}000.S_1} = \frac{\text{Thrust horsepower}}{\text{S.H.P. in open water}} = \text{Screw efficiency in open water} = n_p \quad (17)$$

$$\frac{S_1}{S} = \frac{\text{S.H.P. in open water}}{\text{S.H.P. behind}} = \text{relative rotative efficiency} = n_r \quad (18)$$

$\therefore$ Propulsive efficiency $= n_H . n_p . n_r$ ... ...(19)

Where, R = resistance of bare hull in pounds
T = thrust of propellers in pounds
V = speed of ship in knots
V_1 = speed of advance of propellers in knots
S_1 = shaft horsepower at propellers when running in open water
S = shaft horsepower at propellers when running behind ship.

From the above, the S.H.P. required at the propellers for the bare hull without appendages is,

$$\text{S.H.P.} = \frac{\text{E.H.P.}}{n_p . n_r . n_H} \quad \ldots \ldots \ldots \ldots (20)$$

The hull efficiency (n_H) may be written as follows :—
Suppose
$T - R = t . T$
where t is a coefficient called the "thrust deduction factor". Then,

$$\frac{R}{T} = (1 - t) \quad \ldots \ldots \ldots \ldots \ldots (21)$$

Again,

$$\text{Froude's wake factor } w = \frac{V - V_1}{V_1}$$

whence,

$$\frac{V}{V_1} = (1 + w) \quad \ldots \ldots \ldots \ldots \ldots (22)$$

$$\frac{R.V}{T.V_1} = (1 - t)\,(1 + w) \quad \ldots \ldots \ldots \ldots (23)$$

This fraction is generally in the region of unity, seldom below, and for practical estimating purposes, before tank experiments, may be taken as unity. Also, from a number of tank experiments of liner forms, n_r, the relative rotative efficiency, may be assumed as unity so that the product ($n_H . n_r$) called the "qualified hull efficiency" becomes unity. Equation (20) therefore reduces to,

$$\text{S.H.P. bare hull} = \frac{\text{E.H.P.}}{n_p} \quad \ldots \ldots \ldots (24)$$

To obtain the estimated S.H.P. required at the thrust block to propel the ship at any given speed, an allowance has to be made for the following :—

(*a*) Appendages—bossing, rudder and bilge keels.
(*b*) Wind resistance.
(*c*) Weather conditions.
(*d*) Friction of shafting.

An allowance on the bare hull E.H.P. of 5 per cent. will cover (*a*) above, and under trial trip conditions an additional 10 per cent. will cover (*b*) and (*c*).

For round voyage conditions, for Eastern-going liners, a total allowance for (*a*) (*b*) and (*c*) of 29 per cent. has been found in practice to be adequate.

In modern liners, self-lubricated tunnel shafting bearings are installed which reduce the frictional losses between the thrust block and propellers to about 1 per cent. Hence, for estimating purposes,

$$\text{Trial S.H.P.} = \frac{\text{E.H.P. bare hull} + 15\%}{\cdot 99 n_p} \quad \ldots (25)$$

$$\text{Voyage S.H.P.} = \frac{\text{E.H.P. bare hull} + 29\%}{\cdot 99 n_p} \quad \ldots (26)$$

where "E.H.P. bare hull" applies to the respective trial and voyage displacements at any given speed.

(7) On the Estimation of "n_p", the Open Water Efficiency of a Screw Propeller.

A complete investigation into the experimental work of Froude, Taylor, Durand and Schaffron on propellers was made by Admiral D. W. Taylor of the United States Navy in 1924 and the series of curves shown in *Figs. 1 and 1A were put forward embodying the work of these four great authorities.

The data is given in the form of dimensionless constants, the abscissa being a constant B_p based upon three factors which are always known in the early stages of the design, namely,

Propeller revolutions per minute ... N
Shaft horsepower at propeller ... ... S
Speed of advance through wake water... V_1

$$\text{Power coefficient } B_p = \frac{N . S^{\frac{1}{2}}}{V_1^{2.5}} \quad \ldots \ldots \ldots (27)$$

The curves show the efficiency, pitch ratio and diameter constant δ, where,

$$\delta = \frac{N.D.}{V_1}$$

a function of pitch ratio and real slip.

$$\text{Real slip } (s) = \frac{p.N - V_1 \times 101{\cdot}33}{p.N.}$$

whence,

$$N = \frac{V_1 \times 101{\cdot}33}{p\,(1 - s)}$$

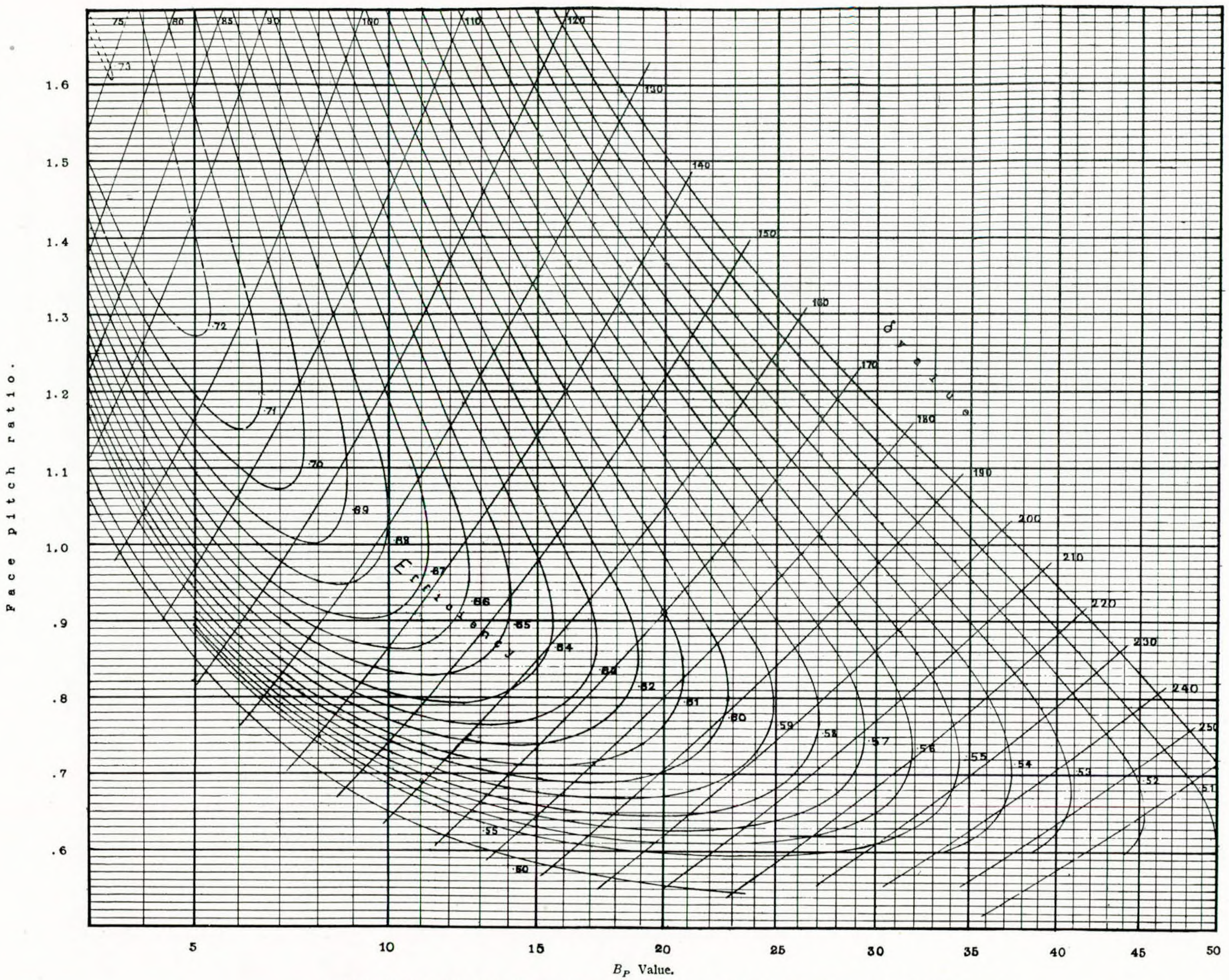
$$\delta = \frac{V_1 \times 101{\cdot}33\, D}{p.V_1\,(1 - s)} = \frac{101{\cdot}33}{\frac{p}{D}\,(1 - s)} \quad \ldots (28)$$

where, $\frac{p}{D}$ = pitch ratio.

Before Figs. 1 or 1a can be used for designing the propeller and obtaining its open water efficiency, it is necessary to approximate the S.H.P. at the required speed. The method is as follows :—

(*a*) Calculate the E.H.P. on the given displacement and speed.
(*b*) Assume $n_H = 1{\cdot}0$ and $n_r = 1{\cdot}0$.
(*c*) From similar types of twin screw ships, assume n_p at say 66%.
(*d*) V_1 from similar ships may be taken as $\frac{V}{1{\cdot}20}$
(*e*) B_p can now be calculated.
(*f*) Lift off from diagram 1 the efficiency and pitch ratio at various values of δ.
(*g*) Draw curves of δ and efficiency on a base of pitch ratio.
(*h*) Select the best propeller and lift off efficiency from curve.
(*i*) S.H.P. on trial $= \frac{\text{E.H.P.} + 15\%}{\cdot 99\, n_p}$

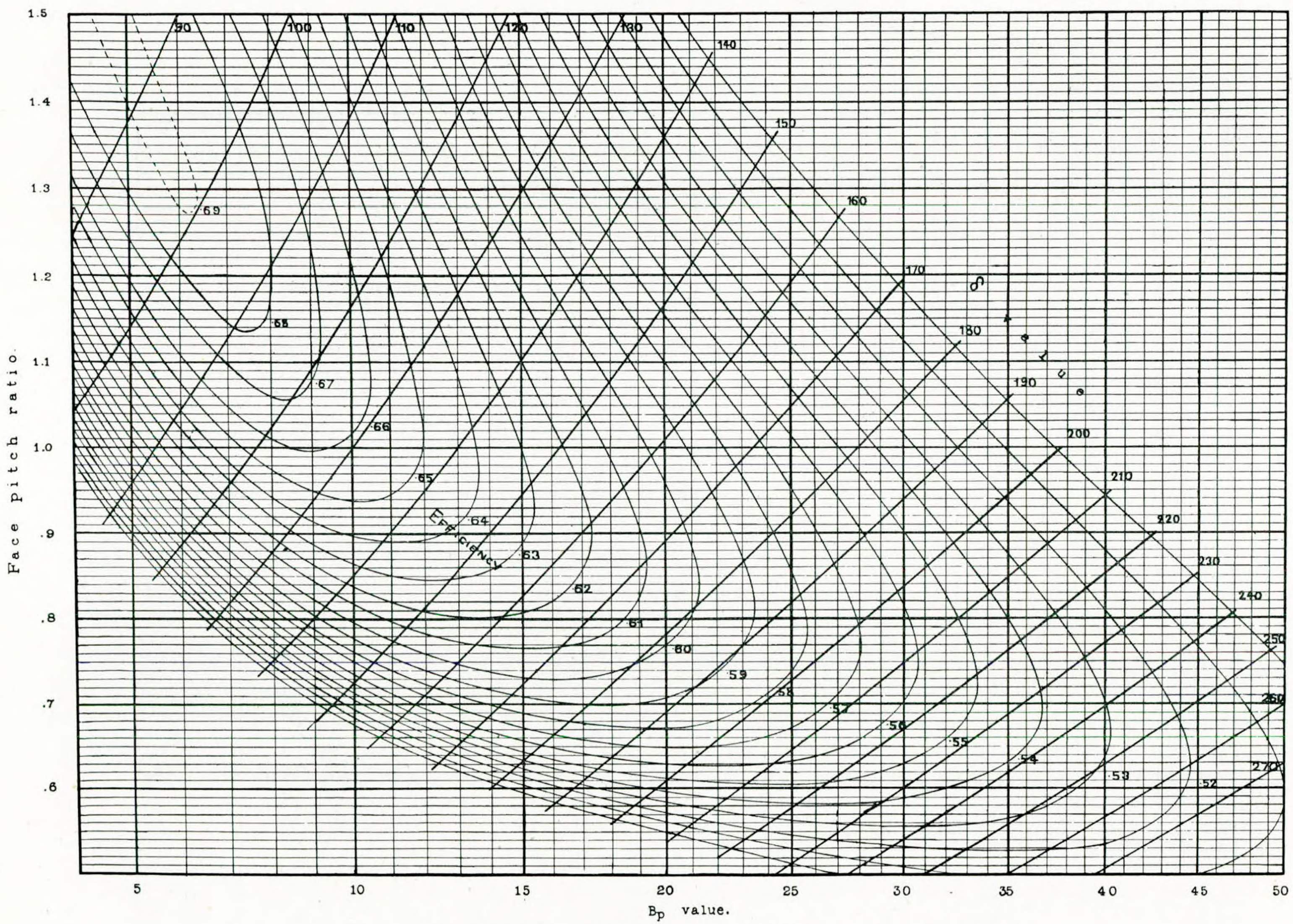
*Figs. 1 and 1A reproduced by kind permission of Dr. G. S Baker from his book "Ship Design, Resistance and Screw Propulsion".



—Taylor's B_P Diagram for Three-bladed Screws. Circular back, b.t.f. = ·05, m.w.r. = ·25. Elliptic outline.

$$B_P = \frac{N\,S^{0.5}}{V_1^{2.5}} \qquad \delta = \frac{N\,D}{V_1} \qquad \text{where} \begin{cases} S = \text{shaft horse-power}, & V_1 = \text{wake speed in knots} = \left(\dfrac{V}{1+w}\right) \\ N = \text{revs. per minute}, & D = \text{diameter in feet.} \end{cases}$$

Fig. 1.



—TAYLOR'S B_P DIAGRAM FOR FOUR-BLADED SCREWS. Circular back. b.t.f. = ·05. n.w.r. = ·25. Elliptic outline.

$$B_P = \frac{N\,S^{0.5}}{V_1^{2.5}} \qquad \delta = \frac{N\,D}{V_1} \qquad \text{where} \begin{cases} S = \text{shaft horse-power}, & V_1 = \text{wake speed in knots} = \left(\dfrac{V}{1+w}\right) \\ N = \text{revs. per minute}, & D = \text{diameter in feet} \end{cases}$$

FIG. 1A.

(j) S.H.P. at sea $= \dfrac{\text{E.H.P.} + 29\%}{\cdot 99\, n_p}$

From the above basic discussion, the E.H.P. and S.H.P. will be calculated at the estimated trial, service and load displacements for a vessel having the following dimensions.

(8) Ship Dimensions.

Length between perpendiculars	530ft. 0in.
,, on waterline	550ft. 0in.
Moulded beam	73ft. 0in.
Load draught	29ft. 6in.
Load block coefficient	0·650
,, prismatic ,, ...	0·670
Sea speed	19¼ knots on load displacement

$$\text{Load displacement} = \frac{530 \times 73{\cdot}0 \times 29{\cdot}5 \times {\cdot}65}{35} = 21{,}190 \text{ tons}$$

$$\text{Beam draft ratio} = \frac{73}{29{\cdot}5} = 2{\cdot}474$$

$$\text{Displacement length ratio} = \frac{21{,}190}{\left(\frac{550}{100}\right)^3} = 127{\cdot}3$$

Taylor's formula

$$S = 16{\cdot}25\sqrt{21{,}190 \times 550} = 55{,}500 \text{ sq ft.}$$

Denny's formula

$$S = 1{\cdot}86 \times 550 \times 29{\cdot}5 + \frac{21{,}190 \times 35}{29{\cdot}5} = 55{,}300 \text{ sq. ft.}$$

Froude's formula

$$S = (21{,}190 \times 35)^{\frac{2}{3}}\left(3{\cdot}7 + \frac{550}{2 \times (21{,}190 \times 35)^{\frac{1}{3}}}\right)$$
$$= 8{,}200 \times 6{\cdot}74$$
$$= 55{,}270 \text{ sq. ft.}$$

The mean of the three calculations may be taken as 55,400 sq. ft. and the resistance will be calculated on this.

"f" from Table 1 above $= {\cdot}00875$

(9) Calculation of E.H.P. on Load Displacement.

The following table gives the residuary resistance in pounds per ton of displacement for the complete speed range of the vessel. Columns 3 and 4 are taken direct from D. W. Taylor's book "Speed and Power of Ships" and column 5 shows the correction for the actual beam-draft ratio.

TABLE 2.

$\frac{V}{\sqrt{L}}$	V Knots	*Beam draft* 3·75 R_r	*Beam draft* 2·25 R_r	*Beam draft* 2·474 R_r
·6	14·04	1·005	0·61	0·6685
·65	15·22	1·280	0·85	0·9136
·70	16·40	1·630	1·155	1·2253
·75	17·55	2·110	1·70	1·7607
·80	18·73	2·720	2·40	2·4474
·85	19·92	3·800	3·15	3·2462
·90	21·08	5·45	4·80	4·8962
·95	22·25	8·35	8·35	8·35

The frictional resistance per ton of displacement is worked out for each of the above speeds and added to column 5 above to obtain R_t, the total resistance per ton of displacement from which the E.H.P. is at once determined from equation 7 above.

$\frac{V}{\sqrt{L}} = {\cdot}6.\quad R_f = \dfrac{f.S.V^{1\cdot83}}{D} = \dfrac{{\cdot}00875 \times 55{,}400 \times 14{\cdot}04^{1\cdot83}}{21{,}190}$

$= {\cdot}022876 \times 14{\cdot}04^{1\cdot83} = 2{\cdot}91$ lb./ton

R_r ... ... $= 0{\cdot}6685$,,

R_t ... ... $= 3{\cdot}5785$,,

E.H.P. $= {\cdot}0030707 \times 21{,}190 \times 3{\cdot}5785 \times 14{\cdot}04 = 3{,}260$

$\frac{V}{\sqrt{L}} = {\cdot}65.\quad R_f = {\cdot}022876 \times 15{\cdot}22^{1\cdot83}$ ... $= 3{\cdot}343$ lb./ton

R_r ... ... $= 0{\cdot}9136$,,

R_t ... ... $= 4{\cdot}2566$,,

E.H.P. $= {\cdot}0030707 \times 21{,}190 \times 4{\cdot}2566 \times 15{\cdot}22 = 4{,}215$

$\frac{V}{\sqrt{L}} = {\cdot}70.\quad R_f = {\cdot}022876 \times 16{\cdot}40^{1\cdot83}$ ... $= 3{\cdot}82$ lb./ton

R_r ... ... $= 1{\cdot}2253$,,

R_t ... ... $= 5{\cdot}0453$,,

E.H.P. $= {\cdot}0030707 \times 21{,}190 \times 5{\cdot}0453 \times 16{\cdot}40 = 5{,}375$

$\frac{V}{\sqrt{L}} = {\cdot}75.\quad R_f = {\cdot}022876 \times 17{\cdot}55^{1\cdot83}$ ... $= 4{\cdot}4$ lb./ton

R_r ... ... $= 1{\cdot}7607$,,

R_t ... ... $= 6{\cdot}1607$,,

E.H.P. $= {\cdot}0030707 \times 21{,}190 \times 6{\cdot}1607 \times 17{\cdot}55 = 7{,}030$

$\frac{V}{\sqrt{L}} = {\cdot}80.\quad R_f = {\cdot}022876 \times 18{\cdot}73^{1\cdot83}$ ... $= 4{\cdot}925$ lb./ton

R_r ... ... $= 2{\cdot}4474$,,

R_t ... ... $= 7{\cdot}3724$,,

E.H.P. $= {\cdot}0030707 \times 21{,}190 \times 7{\cdot}3724 \times 18{\cdot}73 = 8{,}990$

$\frac{V}{\sqrt{L}} = {\cdot}85.\quad R_f = {\cdot}022876 \times 19{\cdot}92^{1\cdot83}$ ... $= 5{\cdot}500$ lb./ton

R_r ... ... $= 3{\cdot}2462$,,

R_t ... ... $= 8{\cdot}7462$,,

E.H.P. $= {\cdot}0030707 \times 21{,}190 \times 8{\cdot}7462 \times 19{\cdot}92 = 11{,}330$

$\frac{V}{\sqrt{L}} = {\cdot}90.\quad R_f = {\cdot}022876 \times 21{\cdot}08^{1\cdot83}$ ... $= 6{\cdot}070$ lb./ton

R_r ... ... $= 4{\cdot}8962$,,

R_t ... ... $= 10{\cdot}9662$,,

E.H.P. $= {\cdot}0030707 \times 21{,}190 \times 10{\cdot}9662 \times 21{\cdot}08 = 15{,}020$

(10) Calculation of E.H.P. on Average Estimated Voyage Displacement of 16,500 tons.

(a) Estimated block coefficient at 16,500 tons displacement from formulæ (3-4) above.

$$\beta = {\cdot}576 - \frac{1{\cdot}78 \times 530 \times 73}{1{,}000{,}000} + \frac{(9{\cdot}3 - {\cdot}00007 \times 530 \times 73)\ 16{,}500}{1{,}000{,}000}$$
$$= {\cdot}5071 + {\cdot}1087$$
$$= {\cdot}6158, \text{ say } {\cdot}616$$

(b) Estimated draught on 16,500 tons displacement.

$$\text{Draft} = \frac{35 \times 16{,}500}{530 \times 73 \times {\cdot}616} = 24{\cdot}22 \text{ feet.}$$

(c) Beam draft ratio $= \dfrac{73}{24{\cdot}22} = 3{\cdot}008$

prismatic coefft. $\dfrac{{\cdot}616}{{\cdot}97} = {\cdot}635$

(d) Displacement/length coefft. $= 99{\cdot}5$

TABLE 3.

$\frac{V}{\sqrt{L}}$	V Knots	Beam draft (3·75) R_r	Beam draft 2·25 R_r	Beam draft 3·008 R_r
·60	14·04	·900	·54	0·722
·65	15·22	1·120	·745	0·9345
·70	16·40	1·370	·980	1·177
·75	17·55	1·73	1·37	1·552
·80	18·73	2·22	1·88	2·052
·85	19·92	2·92	2·48	2·702
·90	21·08	4·25	3·67	3·963

Wetted surface $=16{\cdot}25\sqrt{16{,}500\times 550}=48{,}900$ sq. ft.

Denny's formula :

$$S=1{\cdot}86\times 550\times 24{\cdot}22+\frac{16{,}500\times 35}{24{\cdot}22}=48{,}600 \text{ sq. ft.}$$

Froude's formula :

$$S=(16{,}500\times 35)^{\frac{2}{3}}\left(3{\cdot}7+\frac{550}{2(16{,}500\times 35)^{\frac{1}{3}}}\right)=48{,}500 \text{ sq. ft.}$$

Taking the mean of these three,

wetted surface $=48{,}667$ sq. ft.

$\frac{V}{\sqrt{L}}={\cdot}6.\quad R_f=\frac{f.S.V^{1\cdot 83}}{D}=\frac{{\cdot}00875\times 48{,}667\times V^{1\cdot 83}}{16{,}500}={\cdot}0258V^{1\cdot 83}$

$R_f={\cdot}0258\times 14{\cdot}04^{1\cdot 83}$ ... $=3{\cdot}28$ lb./ton

R_r ... ... $={\cdot}722$ „

R_t ... ... $=4{\cdot}002$ „

E.H.P. $={\cdot}0030707\times 16{,}500\times 4{\cdot}002\times 14{\cdot}04=2{,}853$

$\frac{V}{\sqrt{L}}={\cdot}65.\quad R_f={\cdot}0258\times 15{\cdot}22^{1\cdot 83}$ ... $=3{\cdot}765$ lb./ton

R_r ... ... $={\cdot}9345$ „

R_t ... ... $=4{\cdot}6995$ „

E.H.P. $={\cdot}0030707\times 16{,}500\times 4{\cdot}6995\times 15{\cdot}22=3{,}622$

$\frac{V}{\sqrt{L}}={\cdot}70.\quad R_f={\cdot}0258\times 16{\cdot}4^{1\cdot 83}$ ... $=4{\cdot}31$ lb./ton

R_r ... ... $=1{\cdot}177$ „

R_t ... ... $=5{\cdot}487$ „

E.H.P. $={\cdot}0030707\times 16{,}500\times 5{\cdot}487\times 16{\cdot}4=4{,}560$

$\frac{V}{\sqrt{L}}={\cdot}75.\quad R_f={\cdot}0258\times 17{\cdot}55^{1\cdot 83}$ ... $=4{\cdot}950$ lb./ton

R_r ... ... $=1{\cdot}552$ „

R_t ... ... $=6{\cdot}502$ „

E.H.P. $={\cdot}0030707\times 16{,}500\times 6{\cdot}502\times 17{\cdot}55=5{,}790$

$\frac{V}{\sqrt{L}}={\cdot}80.\quad R_f={\cdot}0258\times 18{\cdot}73^{1\cdot 83}$ ... $=5{\cdot}55$ lb./ton

R_r ... ... $=2{\cdot}052$ „

R_t ... ... $=7{\cdot}602$ „

E.H.P. $={\cdot}0030707\times 16{,}500\times 7{\cdot}602\times 18{\cdot}73=7{,}220$

$\frac{V}{\sqrt{L}}={\cdot}85.\quad R_f={\cdot}0258\times 19{\cdot}92^{1\cdot 83}$ ... $=6{\cdot}200$ lb./ton

R_r ... ... $=2{\cdot}702$ „

R_t ... ... $=8{\cdot}902$ „

E.H.P. $={\cdot}0030707\times 16{,}500\times 8{\cdot}902\times 19{\cdot}92=9{,}000$

$\frac{V}{\sqrt{L}}={\cdot}90.\quad R_f={\cdot}0258\times 21{\cdot}08^{1\cdot 83}$ ... $=6{\cdot}845$ lb./ton

R_r ... ... $=3{\cdot}963$ „

R_t ... ... $=10{\cdot}808$ „

E.H.P. $={\cdot}0030707\times 16{,}500\times 10{\cdot}808\times 21{\cdot}08=11{,}560$

(11) Calculation of E.H P. on Estimated Trial Displacement of 15,000 tons.

Block coefft.

$$={\cdot}576-\frac{1{\cdot}78\times 530\times 73}{1{,}000{,}000}+\frac{(9{\cdot}3-{\cdot}00007\times 530\times 73)\ 15{,}000}{1{,}000{,}000}$$

$={\cdot}5071+{\cdot}0987$

$={\cdot}6058$, say ${\cdot}606$

Prismatic coefft. $=\frac{{\cdot}606}{{\cdot}97}={\cdot}625$

Draft at 15,000 tons displt. $=\frac{35\times 15{,}000}{530\times 73\times {\cdot}606}=22{\cdot}4$ feet

Beam-draft ratio $=\frac{73}{22{\cdot}4}=3{\cdot}26$

Displacement length coefficient $=\frac{D}{\left(\frac{L}{100}\right)^3}=\frac{15{,}000}{(5{\cdot}5)^3}=90{\cdot}4$

TABLE 4.

$\frac{V}{\sqrt{L}}$	V Knots	Beam draft 3·75 R_r	Beam draft 2·25 R_r	Beam draft 3·26 R_r
·60	14·04	·87	·51	·750
·65	15·22	1·07	·70	·946
·70	16·40	1·30	·94	1·180
·75	17·55	1·62	1·30	1·513
·80	18·73	2·125	1·73	1·993
·85	19·92	2·75	2·30	2·600
·90	21·08	3·95	3·40	3·766

Wetted Surface.

Taylor's formula

$=16{\cdot}25\sqrt{15{,}000\times 550}=46{,}600$ sq. ft.

Denny's formula

$$=1{\cdot}86\times 550\times 22{\cdot}4+\frac{15{,}000\times 35}{22{\cdot}4}=46{,}320 \text{ sq. ft.}$$

Froude's formula

$$=(15{,}000\times 35)^{\frac{2}{3}}\left(3{\cdot}7+\frac{550}{2\ (15{,}000\times 35)^{\frac{1}{3}}}\right)=46{,}250 \text{ sq. ft.}$$

Mean wetted surface $=46{,}390$ sq. ft.

$\frac{V}{\sqrt{L}}={\cdot}60.\quad R_f=\frac{{\cdot}00875\times 46{,}390V^{1\cdot 83}}{15{,}000}={\cdot}02702V^{1\cdot 83}$

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$R_f={\cdot}02702\times 14{\cdot}04^{1\cdot 83}$ ... $=3{\cdot}44$ lb./ton

R_r ... ... $={\cdot}75$ „

R_t ... ... $=4{\cdot}19$ „

E.H.P. $={\cdot}0030707\times 15{,}000\times 4{\cdot}19\times 14{\cdot}04=2{,}710$

$\frac{V}{\sqrt{L}}=\cdot65.$ $R_f=\cdot02702\times15{\cdot}22^{1\cdot83}$ ... $=3{\cdot}945$ lb./ton

R_r ... ... $=\ \cdot946$,,

R_t ... ... $=4{\cdot}891$,,

E.H.P. $=\cdot0030707\times15{,}000\times4{\cdot}891\times15{\cdot}22=3{,}425$

$\frac{V}{\sqrt{L}}=\cdot70.$ $R_f=\cdot02702\times16{\cdot}40^{1\cdot83}$ ... $=4{\cdot}51$ lb./ton

R_r ... ... $=1{\cdot}18$,,

R_t ... ... $=5{\cdot}69$,,

E.H.P. $=\cdot0030707\times15{,}000\times5{\cdot}69\times16{\cdot}40=4{,}300$

$\frac{V}{\sqrt{L}}=\cdot75.$ $R_f=\cdot02702\times17{\cdot}55^{1\cdot83}$ ... $=5{\cdot}195$ lb./ton

R_r ... ... $=1{\cdot}513$,,

R_t ... ... $=6{\cdot}708$,,

E.H.P. $=\cdot0030707\times15{,}000\times6{\cdot}708\times17{\cdot}55=5{,}440$

$\frac{V}{\sqrt{L}}=\cdot80.$ $R_f=\cdot02702\times18{\cdot}73^{1\cdot83}$ ... $=5{\cdot}810$ lb./ton

R_r ... ... $=1{\cdot}993$,,

R_t ... ... $=7{\cdot}803$,,

E.H.P. $=\cdot0030707\times15{,}000\times7{\cdot}803\times18{\cdot}73=6{,}750$

$\frac{V}{\sqrt{L}}=\cdot85.$ $R_f=\cdot02702\times19{\cdot}92^{1\cdot83}$ ... $=6{\cdot}495$ lb./ton

R_r ... ... $=2{\cdot}600$,,

R_t ... ... $=9{\cdot}095$,,

E.H.P. $=\cdot0030707\times15{,}000\times9{\cdot}095\times19{\cdot}92=8{,}360$

$\frac{V}{\sqrt{L}}=\cdot90.$ $R_f=\cdot02702\times21{\cdot}08^{1\cdot83}$ ... $=7{\cdot}170$ lb./ton

R_r ... ... $=3{\cdot}766$,,

R_t ... ... $=10{\cdot}936$,,

E.H.P. $=\cdot0030707\times15{,}000\times10{\cdot}936\times21{\cdot}08=10{,}630$

(12) Summary of E.H.P. Values.

Tables 5 and 6 below give the values of the E.H.P. for the three displacements.

TABLE 5.

$\frac{V}{\sqrt{L}}$	V knots	E.H.P. Displacement tons. 15,000	16,500	21,190
·60	14·04	2,710	2,853	3,260
·65	15·22	3,425	3,622	4,215
·70	16·40	4,300	4,560	5,375
·75	17·55	5,440	5,790	7,030
·80	18·73	6,750	7,220	8,990
·85	19·92	8,360	9,000	11,330
·90	21·08	10,630	11,560	15,020

The results given in Table 5 are plotted in Fig. 2 from which the E.H.P. at each half knot has been lifted and is given in Table 6.

TABLE 6.

$\frac{V}{\sqrt{L}}$	V *knots*	E.H.P. Displacement tons 15,000	16,500	21,190
0·5975	14	2,700	2,800	3,220
0·619	14·5	3,000	3,150	3,620
0·640	15	3,300	3,490	4,030
0·662	15·5	3,640	3,850	4,450
0·683	16	4,000	4,230	4,940
0·7045	16·5	4,380	4,640	5,500
0·725	17	4,860	5,150	6,160
0·747	17·5	5,350	5,690	6,910
0·769	18	5,860	6,260	7,740
0·790	18·5	6,530	6,870	8,580
0·810	19	7,100	7,530	9,450
0·820	19·25	7,450		9,900
0·8315	19·5	7,800	8,270	10,420
0·854	20·0	8,550	9,150	11,550
0·875	20·5	9,430	10,150	13,000
0·895	21·0	10,450	11,330	14,880
0·917	21·5	11,700	12,870	17,250

(13) Calculation of S.H.P. on Load Displacement.

As pointed out earlier, before Figs. 1 or 1A can be used for designing the propeller, an estimate of the shaft horsepower to be absorbed has to be made as outlined under the foregoing headings (*a*) to (*j*).

Assuming from known data of existing twin-screw vessels an open water propeller efficiency of 66 per cent., the estimated shaft horsepower required at sea to maintain 19¼ knots on load displacement is,

$$\text{S.H.P.}=\frac{9{,}900\times1{\cdot}29}{\cdot66\times\cdot99}=19{,}750$$

$= 9{,}875$ S.H.P. each propeller

Propeller revolutions per minute $=125$

Speed of advance $V_1=\frac{19{\cdot}25}{1{\cdot}20}=16{\cdot}04$ knots

Power coefficient $B_p=\frac{N.S^{\frac{1}{2}}}{V_1^{\,2\cdot5}}=\frac{125\times9{,}875^{\frac{1}{2}}}{16{\cdot}04^{2\cdot5}}=11{\cdot}93$

From Fig. 1, the following have been lifted :—

δ	*Pitch ratio.*	*Efficiency* %
167	·625	50
160	·700	58·4
150	·808	64·3
140	·940	66·7
130	1·115	66·2
120	1·323	63·5
110	1·600	59·0

From the curve of δ and efficiency plotted on pitch ratio, Fig. 1B, an efficiency of 66·9% is shown for a pitch ratio of ·98 and δ value of 137·5,

Hence,

$$\text{Diameter}=\frac{\delta.V_1}{N}=\frac{137{\cdot}5\times16{\cdot}04}{125}=17{\cdot}63\text{ ft.}$$

Face pitch $=\cdot98\times17{\cdot}63=17{\cdot}30$ ft.

Developed surface $=0{\cdot}4\times\cdot7854\times17{\cdot}63^2=97{\cdot}8$ sq. ft.

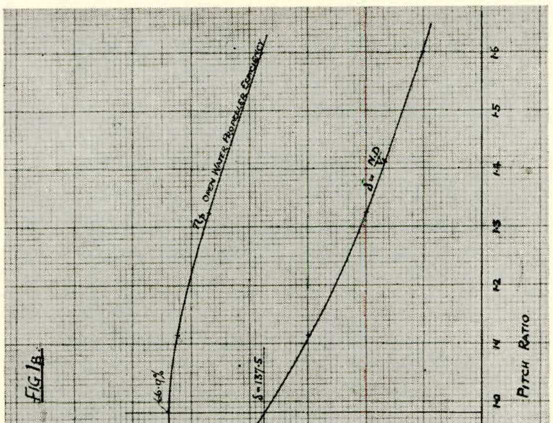


Fig. 1B.

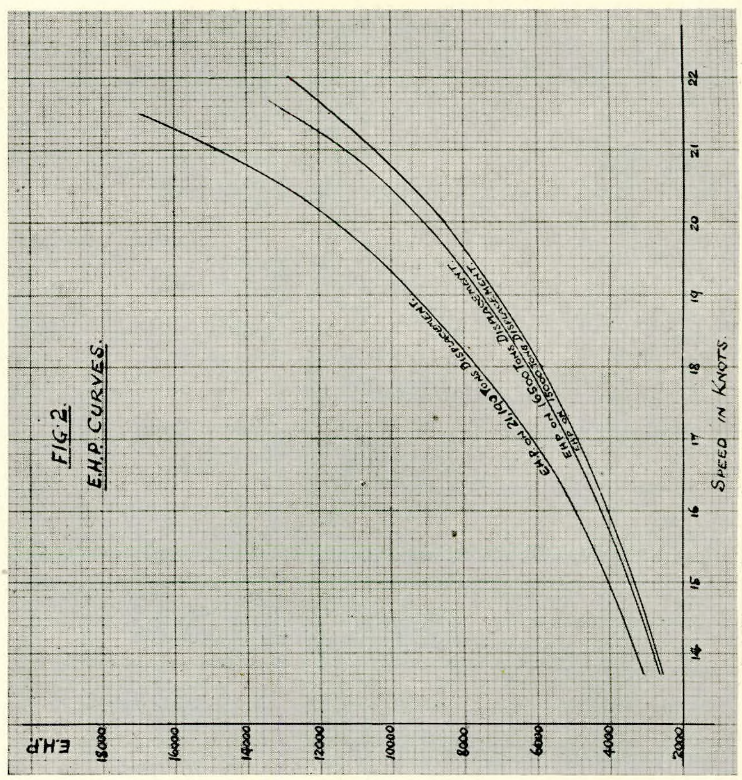


Fig. 2.

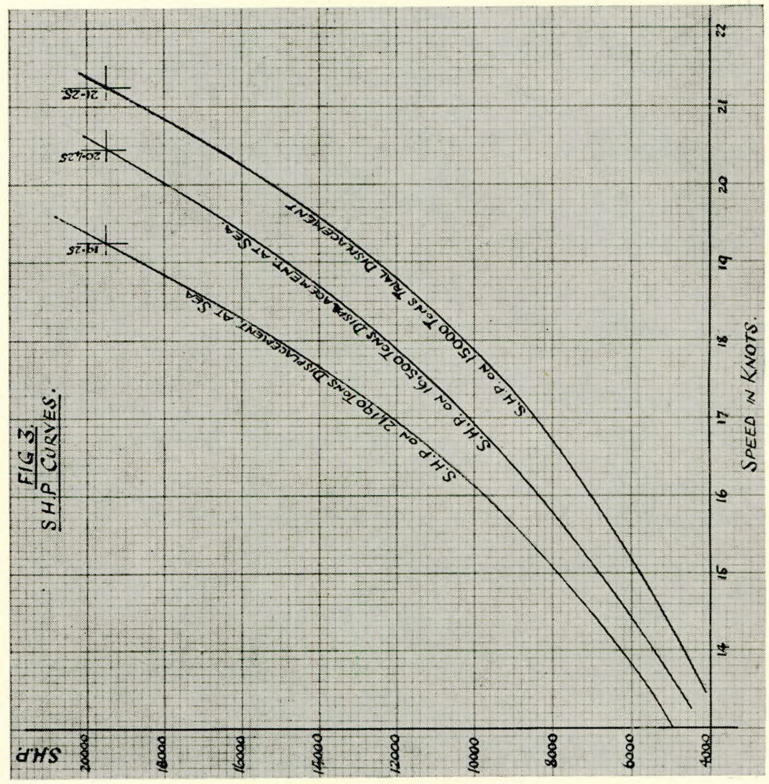


Fig. 3.

Dimensions of Propeller made as follows.

Diameter...	17′ - 9″
Mean pitch	17·4ft.
Developed surface	99 sq. ft.
Number of blades	3
Pitch ratio	0·98
Disc area ratio ...	0·40
Open water efficiency ...	$66{\cdot}9\% = n_p$

S.H.P. each screw at 19¼ knots on load displacement

$$= \frac{9{,}900 \times 1{\cdot}29}{{\cdot}669 \times {\cdot}99 \times 2} = 9{,}650$$

Total S.H.P. $= 2 \times 9{,}650 = 19{,}300$

Design machinery for 19,500 S.H.P.

Estimated Maximum Trial Speed.

Assuming that the propeller efficiency remains constant, the E.H.P. may be calculated and the speed lifted off the E.H.P. curve. Thus

E.H.P. on 15,000 tons trial conditions

$$= \frac{{\cdot}669 \times {\cdot}99 \times 19{,}500}{1{\cdot}15} = 11{,}090$$

From the curve of E.H.P. max. trial speed = 21¼ knots.

Estimated Maximum Sea Speed.

E.H.P. on 16,500 tons sea conditions

$$= \frac{{\cdot}669 \times {\cdot}99 \times 19{,}500}{1{\cdot}29} = 10{,}000$$

From E.H.P. curve 10,000 E.H.P. corresponds to 20·425 knots.

(14) Summary of S.H.P. Values.

Table 7 below gives the values of the shaft horse-power for the three displacements.

TABLE 7.

V knots	Estimated s.h.p.		
	15,000 tons trial	16,500 tons at sea	21,190 tons at sea
14	4,750	5,510	6,350
14½	5,280	6,205	7,130
15	5,805	6,880	7,940
15½	6,400	7,590	8,750
16	7,050	8,340	9,730
16½	7,710	9,150	10,810
17	8,555	10,150	12,130
17½	9,400	11,190	13,610
18	10,310	12,340	15,230
18½	11,500	13,540	16,890
19	12,500	14,820	18,600
19¼			19,500
19½	13,710	16,300	
20	15,050	18,000	
20·425		19,500	
20½	16,600		
21	18,400		
21¼	19,500		

Fig. 3 shows the shaft horse-powers at the three displacements.

REFERENCES.

"Speed and Power of Ships", by D. W. Taylor, U.S.N. Published by Chapman & Hall, Ltd.
"Ship Design, Resistance and Screw Propulsion", by Dr. G. S. Baker. Published by Charles Birchall & Sons, Ltd.

ELECTION OF MEMBERS.

List of those elected at Council Meeting held on Tuesday, 7th May, 1940.

Members.

Alfred Davis.
Francis Gerard D'aranjo.
Ian Munro Fraser.
Donald Fraser Gow.
Cecil Irving Hinchcliffe.
Irling Kielland Riple.

Associate Member.

Thomas Bell.

Associates.

Joseph John Bruce.
John Joseph Fowler.
Thomas Stanley Hodgson.
John McDonald Paterson.
James D. Linklater.
Hunter T. McMichael.
Albert Roy Norgate.
William Shearman.
David Simpson.

ADDITIONS TO THE LIBRARY.

Purchased.

Reports of the Progress of Applied Chemistry, Vol. XXIV, 1939. Society of Chemical Industry. Containing the following:

"General, Plant, and Machinery", by Donald.
"Fuel", by Hodsman, Claydon and Forsyth.
"Gas, Destructive Distillation, Tar and Tar Products", by Hollings and Voss.
"Mineral Oils", by Goulston.
"Intermediates and Dyes", by Rodd, Lapworth and Irving.
"Fibres, Textiles and Cellulose" ("The Protein Fibres", by Whewell and Speakman and "Cellulose Textile Chemistry", by Marsden).
"Pulp and Paper", by Ainslie.
"Acids, Alkalis, and Salts, etc.", by Parrish and Snelling.
"Glass", by Cousen.
"Refractories, Ceramics, and Cement", by Sugden.
"Iron and Steel", by Kennett.
"Non-Ferrous Metals", by Powell.
"Electrochemical and Electrometallurgical Industries", by Cuthbertson.
"Oils, Fats and Waxes", by Hilditch.
"Plastics", by Members of the Plastics Group.
"Resins, Drying Oils, Varnishes and Paints", by Members of the Oil and Colour Chemists' Association.
"Rubber", by Garner.
"Soils and Fertilisers", by Jacks.
"Sugars", by Lever.
"The Fermentation Industries", by Comrie.
"Foods", by Moran.

"Fine Chemicals, Medicinal Substances, and Essential Oils", by Paget and Sharp.
"Photographic Materials and Processes", by Batley.
"Sanitation, Water Purification, etc." by Coste.
"Leather and Glue".

Presented by the Publishers.

Regulations for the Electrical Equipment of Ships. The Institution of Electrical Engineers, 3rd edn., 3s. net cloth bound, 2s. net paper covered.

The National Physical Laboratory, Report for the Year 1939. H.M. Stationery Office, 2s. 6d. net.

"Welding as a Substitute for Casting". Paper by S. F. Dorey, D.Sc., M.I.Mar.E.

The following British Standard Specifications:—

No. 883-1940. Cables and Flexible Cords for Electrical Equipment of Ships (including Electric Propulsion).
No. 891-1940. Direct Reading Hardness Testing (Rockwell Principle).
No. 538-1940. Metal Arc Welding in Mild Steel as applied to General Building Construction.
No. 752-1940. Test Code for Acceptance Tests for Steam Turbines.
No. 893-1940. The Method of Testing Dust Extraction Plant and the Emission of Solids from Chimneys of Electric Power Stations

Vapor Charts and Special Tables for Turbine Calculations. By Frank O. Ellenwood and Charles O. Mackey. Chapman & Hall, Ltd., 43pp., copiously illus., 15s. net.

The authors are to be congratulated on the enormous amount of work involved in the preparation of the above book of charts.

These charts enable the enthalpy (total heat), volume per pound, entropy and degree of superheat to be read from extremely well scaled charts for steam pressures from 5,500 pounds per sq. in. absolute down to 0·18in. mercury (0·088lb./sq. in. abs.). The adiabatic heat drop may be obtained by following the constant entropy line from the initial to the required pressure line, the temperature or, below the saturation line, the percentage of moisture being immediately obtained.

In addition to the steam charts there are charts showing the thermodynamic properties of water, ammonia, freon and mixtures of air and water vapour.

The diagrams are published in book form and although this necessitates in certain operations turning from page to page to obtain the information one may require, the authors quite rightly assert that in this form the charts are much more immune from damage than if published in a single large folded sheet.

Most engineers are familiar with the Mollier diagram for working out thermodynamic steam problems but facility in the manipulation of the present charts will present little difficulty.

The steam diagrams are based on the steam tables of Keenan and Keyes and those for ammonia, freon and air and water on the work of leading American authorities.

Altogether a most valuable set of charts which should be of considerable use to steam, refrigerating and air conditioning plant designers.

Steam, Air, and Gas Power. By W. H. Severns, M.S. and H. E. Degler, M.E., M.S. Chapman & Hall, Ltd., 3rd edn., 511pp., 262 illus., 24s. net.

Recent progress in heat-power engineering has necessitated numerous changes in the third edition of this book. The chapter on principles of thermodynamics has been rewritten and augmented; tables showing the properties of saturated and superheated steam, abridged from *Thermodynamic Properties of Steam* published recently by Professors J. H. Keenan and F. G. Keyes, are included in the appendix; the chapters dealing with fuels and their combustion have been combined and made more concise; equations for the chemical reactions involved in the treatment of feedwaters have been added; additional information in regard to fan performances is supplied; and the material on steam engines has been rearranged and reduced in length.

Further, particular attention has been given to the recent advances in steam generating equipment, and to the current developments in steam turbines and internal-combustion engines. As in the previous editions, the authors' aim has been to present illustrations, descriptions, and underlying theory of construction, application and performance of modern heat power plants and their correlated equipment.

The scientific and engineering symbols and abbreviations that appear in this book are in general accordance with the American Tentative Standards approved by the American Standards Association. The omission of the period in abbreviations is also in accordance with these recommendations.

The book contains chapters on principles of thermodynamics, heat power plants, steam and steam calorimetry, steam fuels and combustion, steam generators or boilers, steam generator auxiliaries, feedwater treatment and feedwater heaters, draught apparatus (chimneys and fans), reciprocating steam engines, steam engine power and economy, steam turbines, steam condensing equipment, pumps, compressed air, and internal-combustion engines.

The Switchgear Handbook, Vol. II—Application. Edited by W. A. Coates, M.I.E.E. and H. Pearce, B.Sc., M.I.E.E. Sir Isaac Pitman & Sons, Ltd., 267pp., 131 illus., 21s. net.

This is the second volume of a very useful reference book. The first volume covered apparatus—circuit breakers, switches, instruments—and the present volume deals with their application to various forms of control gear.

The title page carries the monogram of a well known electrical company, so it is not surprising to find many parts of the text disclose a particular school of thought and practice in design and construction. The editors claim the advantage of a number of specialized contributors, and the only criticism of this is the resultant lack of continuity and the variance in the assumed degree of knowledge possessed by the reader. This is not a serious drawback if the volume is regarded as a number of separate papers, collected for easy reference.

The book is essentially concerned with land practice, and therefore mainly with alternating current. The first chapter deals with the calculation of short circuit currents, and most subsequent chapters with the application of specialized types of switch control gear to indoor, outdoor and mining duties.

Chapter four is perhaps the most useful, giving circuit equipments for a.c. and d.c. generators, motors, rotary converters and rectifiers. Each section mentions the principal features of the apparatus to be controlled in order that the means of control may be more clearly understood. This chapter includes many clear and helpful diagrams—indeed the diagrams are an excellent feature of the whole book.

Finally (Chapter XIII) there are brief notes on testing and maintenance, and the general advice on cleanliness and regular inspection could be effectively followed by all sea-going electrical engineers.

In the main, however, from the marine engineer's point of view, the book is one—and a very good one—for the student, but from this same point of view its price is rather disappointingly high.

First Year Engineering Science—Mechanical and Electrical. By G. W. Bird, Wh.Ex., B.Sc.; revised by B. J. Tams, M.Sc. Sir Isaac Pitman & Sons, Ltd., 3rd edn., 136pp., 120 illus., 8s. 6d. net.

This attractively produced book, which is now in its third edition and must be fairly well known, provides a preliminary course in general engineering science of first year standard. It consists of three sections, two of approximately equal length on applied mechanics and electrotechnology, and a brief one on heat and its effects. The ground covered in each subject is necessarily of limited extent, but the course is continued and developed in a second year book by the same author. The subject matter is clearly presented and illustrated with numerous excellent diagrams and half-tone blocks, and there is an ample supply of exercises with answers. Teachers who prefer to adopt a comprehensive first year course of this proportionately rudimentary type will welcome the book, and it can be confidently recommended.

Ships, Boats and Craft. By S. E. Beck. Herbert Jenkins, Ltd., 144pp., copiously illus., 5s. net.

Here is a book that is unusual both in scope and treatment. It contains black and white sketches of over 400 different types of ships and boats, each one accompanied by a brief description of the ship, its purpose and main characteristics. The work is divided into seven principal sections: historical, large sailing vessels and auxiliary, small sailing vessels, large steam and motor vessels, small power vessels, warships, and miscellaneous craft. The types in each section are arranged alphabetically for the sake of convenience. Technical terms have been avoided as far as possible, but a short glossary has been provided for quick reference; a full index completes the work.

The book will have special appeal to the general reader and to young members of The Institute embarking upon their careers and as yet unfamiliar with all the different types of ships. Even more mature readers will find the sections on sailing vessels of considerable interest.

Electrical Technology for Beginners. By G. W. Stubbings, B.Sc. E. & F. N. Spon, Ltd., 156pp., 48 illus., 5s. net.

This book is intended to assist those who have had no regular training in physical science and mathematics to obtain a knowledge of the principles underlying the production and utilisation of electrical energy for domestic and industrial purposes, such as will enable them to understand the technical implications of the I.E.E. Wiring Regulations. Mathematical symbolism has been almost entirely avoided, and the fundamental principles of electrical technology have been explained verbally.

The treatment of the subject-matter is based on the idea of the objective nature of electrical energy as something that is bought and sold, and the opening chapter explains this idea and the various units of measurement used in electrical technology. Thereafter the treatment follows more or less conventional lines, and it includes in chapter IV a rudimentary and non-mathematical explanation of alternating currents. The final chapters deal, in a little greater detail, with electrical installations and simple electrical testing. A few simple numerical questions follow each chapter, and the answers to these questions are fully worked out in the Appendix.

The book is primarily intended to meet the needs of students preparing for the elementary electrical engineering examinations of the Chartered Insurance Institute and of apprentices preparing for the City and Guilds of London Institute in Electrical Installation Work. It is hoped that the book will also be found useful by workers in non-electrical trades and industries for whom a knowledge of simple electrical technology is desirable.

JUNIOR SECTION.

The Slide Valve. By F. W. LUDLAM, B.Sc., A.M.I.Mar.E.

1. *The Cylinder.*

Fig. 1 illustrates (more or less diagramatically) a simple

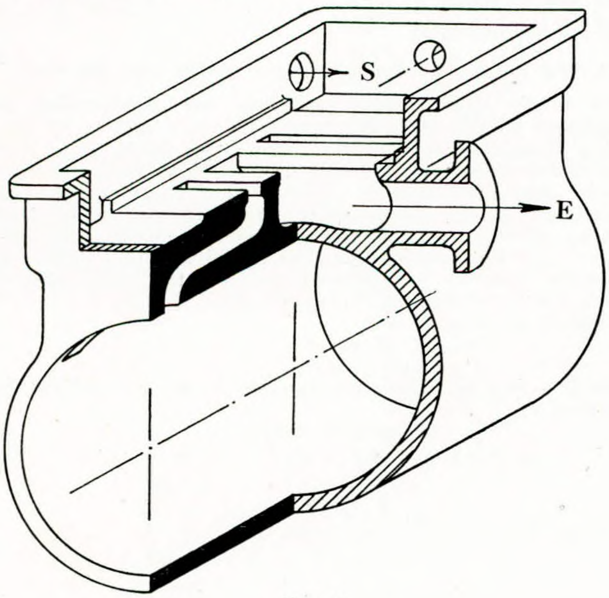


Fig. 1.

steam engine cylinder, in part section and with covers removed to show the essential features. The upper part is the valve chest, in which the valve, to be described presently, slides to and fro. In the planed face on which the valve slides are three rectangular slots or ports; the end ports communicate with the interior of the cylinder, and the middle one with the atmosphere, condenser, or receiver. Steam from the boiler enters the valve chest at *S* and is admitted by the valve to opposite sides of the piston alternately. When its work is done, the steam flows away from the cylinder by the same ports, but is conducted by the valve to the central cavity, and so to exhaust, *E*.

2. *The Valve.*

The general form of the valve will be clear from Fig. 2.

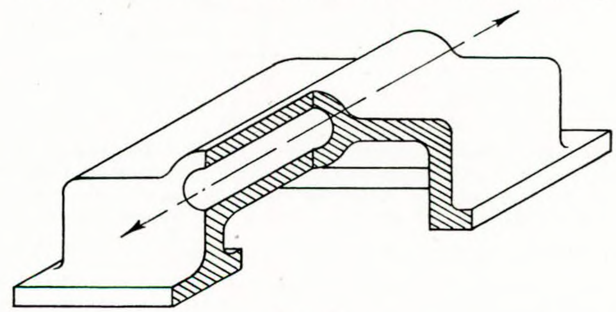


Fig. 2.

It is a shallow rectangular box with a brim, the face of which is accurately planed to make steamtight contact with the seat on which it slides. The arrow-heads indicate the direction of motion. The valve is shown in longitudinal section in Fig. 3,

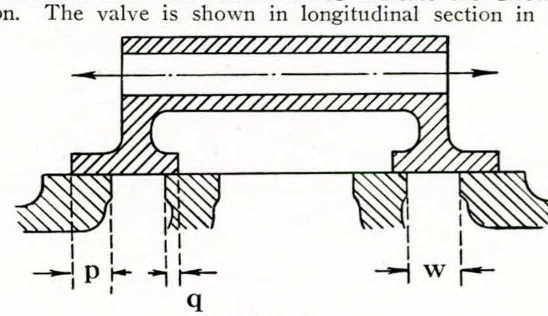


Fig. 3.

standing in mid-position over the steam ports. It will be noticed that each end of the valve overlaps the steam port, both inside and outside.

p is termed the outside, or *steam lap*.

q is termed the inside, or *exhaust lap*.

In the valve drawn, the steam laps at opposite ends are equal, and so are the exhaust laps, but this need not be the case. In fact the exhaust lap at one end may be *negative, i.e.* the steam port at that end may be open to exhaust when the valve is in mid-position—*by which we mean mid-way between its extreme positions.* The actual laps and their relative amounts depend upon the particular manner in which it is desired to distribute the steam to opposite sides of the piston, and regulate its inlet and outlet.

3. *Valve Displacement.*

The valve is driven to and fro by an eccentric, Fig. 4, which

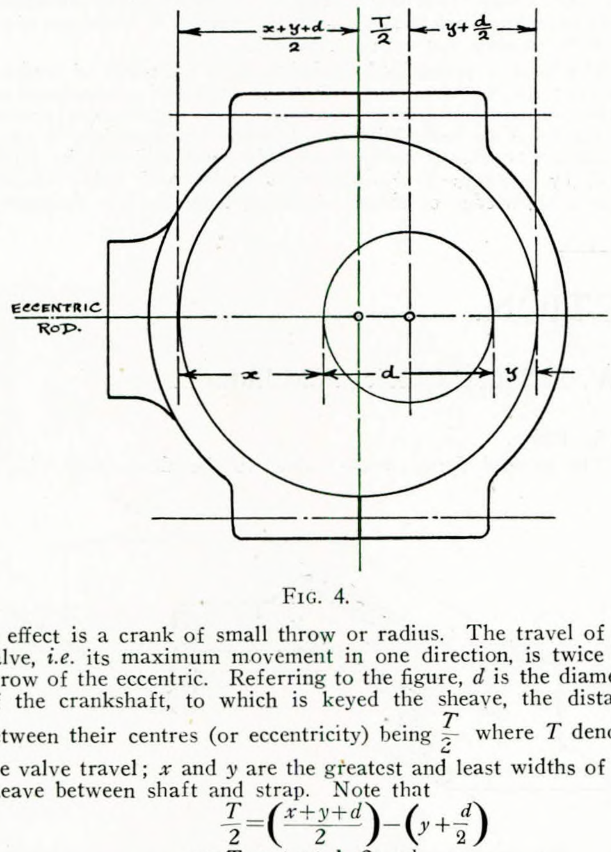


FIG. 4.

in effect is a crank of small throw or radius. The travel of the valve, *i.e.* its maximum movement in one direction, is twice the throw of the eccentric. Referring to the figure, d is the diameter of the crankshaft, to which is keyed the sheave, the distance between their centres (or eccentricity) being $\frac{T}{2}$ where T denotes the valve travel; x and y are the greatest and least widths of the sheave between shaft and strap. Note that

$$\frac{T}{2}=\left(\frac{x+y+d}{2}\right)-\left(y+\frac{d}{2}\right)$$

$$\text{or } T=x+y+d-2y-d$$

$$\therefore T=x-y$$

This affords a convenient practical means of measuring the valve travel produced by a given eccentric.

In Fig. 5, R represents the throw of the eccentric, or equiva-

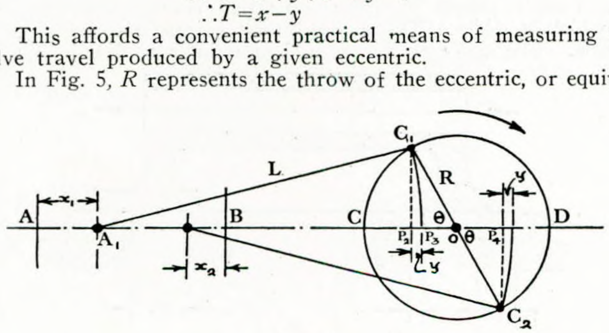


FIG. 5.

lent crank: we shall speak of this simply as the eccentric. L is the eccentric rod, drawn very short for convenience. AB is the valve travel and so also is CD. $AC=L$ and $BD=L$. In what follows we shall speak in terms of a vertical engine; A is the top position of the valve and B the bottom. Consider the downward stroke of the valve, from A to B.

x_1 is the displacement of the valve from A when the eccentric makes an angle θ with its initial position on the line of centres. The pin which connects the eccentric rod to the valve rod is at A_1 and the eccentric centre is at C_1. From C_1 drop a perpendicular C_1P_2 on to the line of stroke; then evidently CP_2 will be approximately equal to the valve displacement x_1. With centre A_1 and radius L strike the arc C_1P_3. Then CP_3 is the valve displacement accurately, because $AC=L$ and $A_1P_3=L$. Denote the small distance P_2P_3 by y. Then, for the downward stroke,

$$\text{valve displacement}=CP_2 \text{ approximately}$$

$$x_1=CP_2+y \text{ accurately.}$$

The same figure shows the eccentric in the equivalent position on the upward stroke. The eccentric centre has moved from D to C_2, the radius R having described an angle θ. The displacement of the valve from its initial position at the bottom is x_2, and it is at once noticeable that x_2 is less than x_1.

$$\text{The valve displacement}=DP_4 \text{ approximately}$$

$$x_2=DP_4-y \text{ accurately.}$$

$DP_4=CP_2$, so x_2 is less than x_1 by the amount $2y$.

For given values of L and R, or ratio $\frac{L}{R}$, the correction y will depend upon the angle θ, being greatest when the rod L occupies its most oblique position, *i.e.* when R is perpendicular to the line of stroke. In the diagram a rather low value of the ratio $\frac{L}{R}$ has been chosen, namely, 3·5. Eccentric rods, however, are very long in comparison with the eccentricity, and the maximum obliquity of the rod is so small that the correction y is negligible. To find the displacement of the valve it is sufficient to drop a perpendicular from the eccentric centre on to the line of stroke and measure the distance of its foot from the extreme position. Thus, in particular, we shall assume that *the valve is in mid-position when the eccentric is in a position perpendicular to the line of stroke.*

4. *Piston Displacement.*

In the crank and connecting rod mechanism the ratio $\frac{L}{R}$ is not nearly large enough to permit our disregarding the obliquity of the rod. The difference between x_1 and x_2 for equal angular displacements θ of the crank is so large as to affect considerably the relative amounts of work done by the steam on opposite sides of the piston. Given L and R we may find the displacement of the piston corresponding to any given crank angle by drawing a scale figure, and provided one has the ability to draw well this graphical method is satisfactory. Piston displacements may be obtained more accurately, and quite as quickly, by calculation as explained below.

Referring again to Fig. 5,

$$CP_2=R-R\cos\theta\ [=OC-OP_2]$$

$$=R(1-\cos\theta)$$

which is a first approximation for the piston or crosshead displacement x_1.

$$y=A_1P_3-A_1P_2.$$

$$A_1P_3=L \text{ and } A_1P_2=\sqrt{L^2-(C_1P_2)^2}$$

$$=\sqrt{L^2-(R\sin\theta)^2}$$

$$\therefore y=L-\sqrt{L^2-R^2\sin^2\theta}$$

Thus, for the downward stroke AB,

$$x_1=R(1-\cos\theta)+L-\sqrt{L^2-R^2\sin^2\theta}$$

$$\therefore \frac{x_1}{R}=1-\cos\theta+\frac{L}{R}-\sqrt{\frac{L^2-R^2\sin^2\theta}{R^2}}$$

$$=1-\cos\theta+\frac{L}{R}-\sqrt{\left(\frac{L}{R}\right)^2-\sin^2\theta}.$$

and denoting the ratio $\frac{L}{R}$ by N,

$$\frac{x_1}{R}=1-\cos\theta+N-\sqrt{N^2-\sin^2\theta}.$$

This expression may be simplified by making use of the following well-known approximation.

If a is small compared with 1, then

$$\sqrt{1\pm a}=1\pm\frac{1a}{2}$$

To prove this, square both sides: then

$$1 \pm a = 1 \pm a + \frac{a^2}{2},$$

and the error is $\frac{a^2}{2}$, which will be negligibly small when a is a small fraction, say less than 0·1.

$$\text{Now } \sqrt{N^2 - \sin^2\theta} = \sqrt{N^2\left(1 - \frac{\sin^2\theta}{N^2}\right)}$$

$$= N\sqrt{1 - \frac{\sin^2\theta}{N^2}}$$

$\sin^2\theta$ cannot be greater than 1, hence in all our problems $\frac{\sin^2\theta}{N^2}$ will usually be less than 0·1.

$$\therefore N\sqrt{1 - \frac{\sin^2\theta}{N^2}} = N\left(1 - \frac{\sin^2\theta}{2N^2}\right) \text{ approx.}$$

$$= N - \frac{\sin^2\theta}{2N}$$

Substituting this approximation in the above formula for $\frac{x_1}{R}$, we have

$$\frac{x_1}{R} = 1 - \cos\theta + \frac{\sin^2\theta}{2N}$$

Denoting the term $\frac{\sin^2\theta}{2N}$ by K, then

$$\frac{x_1}{R} = 1 - \cos\theta + K.$$

On the upward stroke, to find x_2 we must subtract y, as we have seen.

Hence we have the formulæ:—

$$\left.\begin{array}{ll}\text{Downstroke} & \frac{x_1}{R} = (1 - \cos\theta) + K \\ \text{Upstroke} & \frac{x_2}{R} = (1 - \cos\theta) - K\end{array}\right\} K = \frac{\sin^2\theta}{2N}$$

Example 1.

Connecting rod 8ft. long, crank 2ft. Find the piston displacement when the crank has made 30° (*a*) on the downstroke (*b*) on the upstroke.

Here N=4.

First approximation.

$$\frac{x_1}{R} = 1 - \cos 30°$$

$$\therefore \frac{x_1}{2} = 1 - 0·866 = 0·134$$

$$\therefore x_1 = 0·268\text{ft.}$$

$$\text{and } x_2 = 0·268\text{ft.}$$

Second approximation.

$$K = \frac{\sin^2\theta}{2N} = \frac{0·5^2}{8} = \frac{0·25}{8} = 0·031$$

$$\therefore \frac{x_1}{2} = 1 - 0·866 + 0·031 = 0·165$$

$$\therefore x_1 = 0·33\text{ft.} = 3·96\text{in.}$$

This is greater than the first approximation by 0·062ft., or 0·744in.

On the upstroke

$$\frac{x_2}{2} = 1 - 0·866 - 0·031 = 0·103$$

$$\therefore x_2 = 0·206\text{ft.} = 2·472\text{in.}$$

Note the difference between x_1 and x_2.

When θ lies between 90° and 180°, we must remember that $\cos\theta$ is negative, and that we take the supplement of the angle, *i.e.* the acute angle made by the crank with the line of centres.

Example 2.

For the engine of Example I find x_1 and x_2 when $\theta = 120°$

$$\cos 120° = -\cos 60° = -0·5$$

$$\sin 120° = \sin 60° = +0·866$$

$$\therefore \frac{x_1}{2} = 1 + 0·5 + \frac{0·866^2}{8}$$

$$= 1 + 0·5 + 0·094 = 1·594$$

$$\therefore x_1 = 3·188\text{ft.} = 38·26\text{in.}$$

$$\frac{x_2}{2} = 1 + 0·5 - 0·094$$

$$= 1·406$$

$$\therefore x_2 = 2·812\text{ft.} = 33·74\text{in.}$$

Note again the large difference between the downstroke and upstroke displacements.

The student will probably consider these calculations tedious, and prefer to draw scale diagrams. He can, however, go through the tedium once and for all by plotting the graphs of $1 - \cos\theta \pm K$ for all values of θ from 0° to 180°, and for various values of N: one curve for each value of N, all curves being drawn on the same sheet of squared paper. Or he may plot such curves for the correction term $K = \frac{\sin^2\theta}{2N}$ only, taking values of θ from 0° to 90°. By its aid, displacement problems can be solved in a few seconds, as accurately as by full calculation. *See* Fig. 6. Only three curves are shown here, but those for $N = 3\frac{1}{2}$, $4\frac{1}{2}$, $5\frac{1}{2}$, and 6 should be included.

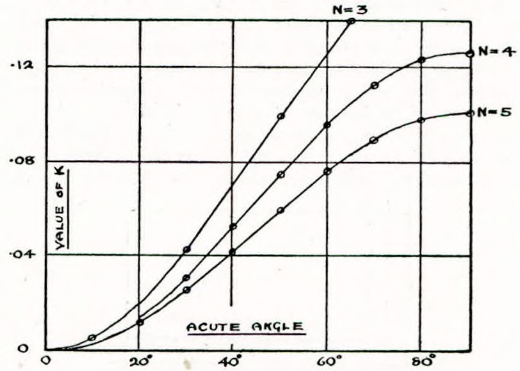


FIG. 6.

Example 3.

$L = 7$ft. 6in., $R = 2$ft. Find the upstroke displacement when $\theta = 140°$.

$$\text{Here } N = \frac{7·5}{2} = 3·75.$$

The acute angle is 180° − 140° = 40°. From the graph, reading between the curves $N = 3\frac{1}{2}$ and $N = 4$, $K = 0·055$.

$$\therefore \frac{x_2}{2} = 1 - \cos 140° - 0·055$$

$$= 1 + 0·766 - 0·055 = 1·711$$

$$\therefore x_2 = 3·422\text{ft.}$$

The student should verify this by calculation in full, and by scale drawing.

5. *The Simple Slide Valve.*

In Fig. 7 is shown a valve which has neither steam nor exhaust laps. Standing in mid-position as shown it just closes the steam ports. Any movement in the direction of the arrow will permit steam to enter the cylinder at the top and leave the cylinder at the bottom. When the valve is in this position the

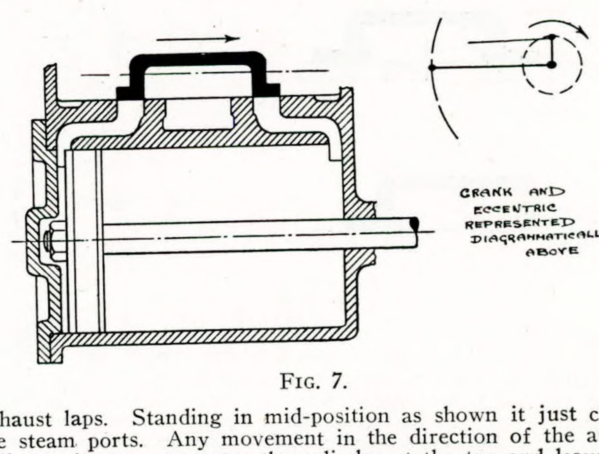


FIG. 7.

piston must be ready to commence its down-stroke towards the crankshaft, and the crank must lie along the line of centres. Moreover, the valve being in mid-position, the eccentric must be perpendicular to the line of centres: above that line (as in the figure) if clockwise rotation of the shaft is desired, below if anti-clockwise. *In whichever direction the shaft is to revolve, the eccentric must be 90° ahead of the crank.*

Now suppose one quarter revolution to take place. The eccentric will then be on the line of centres, and the valve will be at the bottom of its travel. Both steam ports will be fully open, the top to steam and the bottom to exhaust. The crank will be perpendicular to the line of stroke, and the piston will be rather more than half way along or down the cylinder. During the next quarter revolution the valve moves back to mid-position again, closing both ports just as the piston arrives at the bottom of its stroke. Thus steam has been admitted to the top of the piston continuously for the whole stroke, while the bottom has been continuously open to exhaust. During the next half-revolution the valve admits steam below the piston, and permits exhaust of the steam above it. With this simple lapless valve expansive working is not secured. On each side of the piston a volume of steam equal to the stroke volume is admitted during one stroke, and exhausted during the next. The

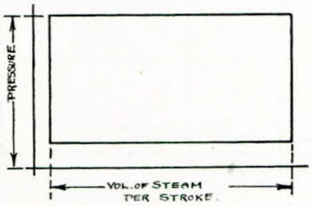


FIG. 8.

theoretical indicator diagram is rectangular, Fig. 8. Each stroke is a powerful, but not economical one.

Note that if we wish the steam port, of width w, to be opened fully, the eccentric throw must be at least equal to w.

6. *Angle of Lap, Lead, and Advance.*

In Fig. 9 is shown a valve with steam and exhaust laps.

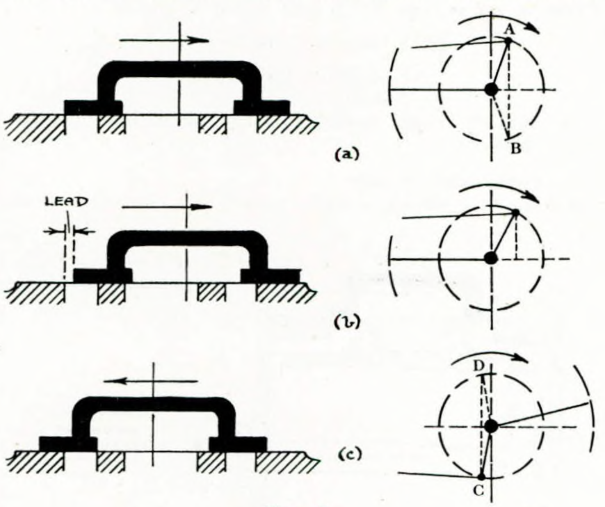


FIG. 9.

Supposing the piston and crank to occupy the positions of Fig. 7, admission must be about to occur to the top of the cylinder. The valve, therefore, must be in the admission position as shown in Fig. 9a, that is, it must be *below* the mid-position (Fig. 3) by an amount equal to the steam lap. Hence the eccentric must be ahead of the perpendicular position by an angle ϕ, termed the *angle of steam lap*. The eccentric must be $90°+\phi$ ahead of the crank. Half a revolution later, when the piston is at the other end of its stroke, the valve must be about to admit steam to the bottom of the cylinder. A moment's consideration will show this to be the case, for the eccentric will have advanced with the crank through half a revolution to the diametrically opposite position and the valve will therefore be *above* the mid-position by an amount equal to the steam lap.

A bad feature of the ordinary slide valve is the narrowness of the opening of the port to steam during the early part of the admission period, just when a full unrestricted flow of steam is especially desirable, and it is found to be advantageous to arrange for the steam port to be already open by a small amount when the piston is about to commence its stroke. This initial opening to steam is termed *lead*. It is illustrated in Fig. 9b, in which crank and piston are supposed to be positioned as in Fig. 7, and the eccentric is advanced a little more in order to provide the lead shown. The eccentric is now ahead of the crank by an angle $90°+\theta$, and θ is termed the *angle of advance*. θ is greater than ϕ, the angle of steam lap, by the small angle e, termed the *angle of lead*.

$$\theta=\phi+e.$$

If we provide lead at one end by advancing the eccentric, we automatically provide the same amount of lead at the other end (neglecting the obliquity of the eccentric rod), as the student will easily see by considering the position of the eccentric half a revolution later. It probably occurred to the student that the lead of Fig. 9b could have been obtained by moving the valve along the valve rod instead of advancing the eccentric. So it could, but then lead would not be obtained at the other end. Indeed the reverse would be the case: there would be *negative* lead. This moving the valve along the rod, or, what comes to the same thing, altering the effective length of the valve rod itself, is, however, useful in practice when it is desired to re-adjust the relative amounts of top and bottom lead, reducing one and increasing the other. Suppose, for example, the lead at the top in Fig. 9b is $\frac{1}{8}$in., obtained by advancing the eccentric. Then the lead at the bottom will be $\frac{1}{8}$in. also, if the top and bottom steam laps are equal. If now the effective length of the valve rod be increased $\frac{1}{16}$in. by the use of a $\frac{1}{16}$in. liner, the valve will be displaced $\frac{1}{16}$in. upwards, the top lead will be reduced to $\frac{1}{8}\text{in.}-\frac{1}{16}\text{in.}=\frac{1}{16}\text{in.}$ and the bottom lead increased to $\frac{1}{8}\text{in.}+\frac{1}{16}\text{in.}=\frac{3}{16}\text{in.}$ In vertical engines it is an advantage to have more bottom lead than top, for the following reason. The provision of lead means that boiler steam enters the cylinder before the piston arrives at the end of the stroke, and so the heavy reciprocating parts are retarded or "cushioned" over the reversal of their motion. The need for this cushioning effect is greater at the bottom than at the top of the stroke because the moving parts are descending, falling. Moreover, the increased lead results in a freer entry of steam at the commencement of the upstroke when the heavy parts have to be lifted against gravity.

To calculate the angles of steam lap, advance, and lead.

In Fig. 10, R is the eccentricity, or half travel, p the steam lap, and z the lead.

$$\text{Angle of steam lap } \sin\phi=\frac{p}{R}$$

$$\text{Angle of advance } \sin\theta=\frac{p+z}{R}$$

$$\text{Angle of lead}=\theta-\phi.$$

Example 4.

Travel 6in., steam lap 2in., lead $\frac{1}{4}$in.

Here $R=3,\ p=2,\ z=0{\cdot}25$

$\sin\phi=\frac{2}{3}=0{\cdot}6667$

$\therefore$Angle of steam lap $\phi=41°\ 49'$, by reference to tables.

$$\sin\theta=\frac{2{\cdot}25}{3}=0{\cdot}75.$$

$\therefore$Angle of advance $\theta=48°\ 36'$

$\therefore$Angle of lead $e=6°\ 47'$, by subtraction.

The same results may be obtained by scale drawing, but perhaps not as accurately. The student should try.

7. *Effect of Steam and Exhaust Laps.*

We have now to enquire into the effect the laps have upon the valve action, and it will be as well to work out in detail some actual examples. The four "events" of the cycle of operations on each side of the piston are:—

A. Admission, the moment when steam is admitted to the cylinder. Valve position as Fig. 9a. Eccentric at *A*.

B. Cut-off, the moment when the valve cuts off the supply of steam. Valve position again as Fig. 9a, but valve moving in opposite direction. Eccentric at *B*.

C. Release, the moment when the steam is allowed to escape to exhaust. Valve position as Fig. 9c, above mid-position by an amount equal to the exhaust lap. Eccentric at *C*.

D. Compression, the moment when further escape of steam to exhaust is prevented, by the valve having returned to position of Fig. 9c, travelling downwards. Eccentric at *D*.

The valve and eccentric positions of Fig. 9, referred to above, are of course the positions for the various events at the top of the cylinder. It is important to understand what is happening in the cylinder, not merely on one side of the piston, but on both sides simultaneously. Looking again at Fig. 9a, note that when the top port is about to open to steam, the bottom port is already well open to exhaust. In Fig. 9b the opening to exhaust has been further increased by the provision of lead to the other side of the piston, and had no exhaust lap been provided the exhaust opening at the commencement of the stroke would have been even greater. Thus we see that steam lap and lead, provided primarily to affect the admission of steam to one

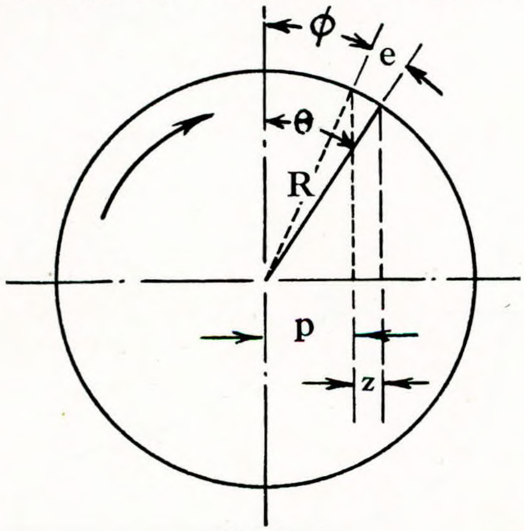


FIG. 10.

side of the piston, also affect the exhaust of steam from the cylinder, on the other side of the piston.

Example 5.

Travel 6in. Steam lap 2in., top and bottom. No lead or exhaust laps.

(The student will find it helpful to cut out stiff cardboard models of this and other valves discussed, and move them to and fro over a drawing of the ports).

$$\text{Angle of advance } \sin\theta = \frac{\text{steam lap}}{\text{half travel}}$$
$$= \tfrac{2}{3} = 0{\cdot}6667$$
$$\therefore \theta = 41^\circ\ 49'$$

$\therefore$Crank must be $90^\circ + 41^\circ\ 49' = 131^\circ\ 49'$ behind eccentric.

Eccentric positions at the four events:—

There being no lead, admission occurs where the crank is on the line of centres, which we shall call the 0° position. Therefore the eccentric is at *A*, 131° 49′. Fig. 11a. Cut-off occurs when the valve is in the same position on the return, *i.e.* when the eccentric is in a position *B*, immediately below *A*.

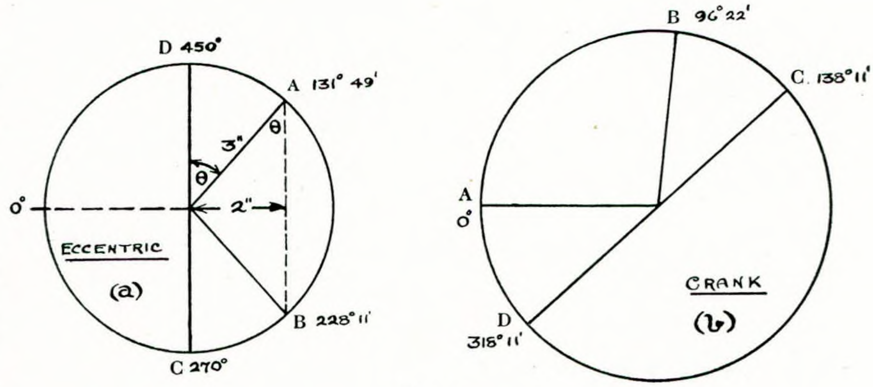


FIG. 11.

The angle is $270^\circ - 41^\circ\ 49' = 228^\circ\ 11'$. As there is no inside or exhaust lap, release occurs when the valve is in mid-position, because then inside edges of valve and port coincide. The eccentric is now at *C*, 270°.

The fourth event, compression, occurs when the valve has returned to the mid-position and just closed the port.

The eccentric is at *D*, 450° $(=270^\circ + 180^\circ)$.

Crank positions:—

These we find by simply subtracting 131° 49′ from each of the eccentric positions, so obtaining Fig. 11b.

Piston positions:—

Let us suppose the connecting rod to be 40in. long, crank 10in. long, that is, $N=4$

(*a*) Top. $\frac{x_1}{10} = 1 - \cos\theta + K.$

K will be read from the graph, Fig. 6.

$$\text{Admission } x_1 = 0''$$
$$\text{Cut-off } x_1 = 10\ (1 - \cos 96^\circ\ 22' + 0{\cdot}125)$$
$$= 10\ (1 + 0{\cdot}1109 + 0{\cdot}125)$$
$$= 12{\cdot}36''$$
$$\text{Release } x_1 = 10\ (1 - \cos 138^\circ\ 11' + K)$$
$$= 10\ (1 + 0{\cdot}7453 + 0{\cdot}056)$$
$$= 18{\cdot}01''$$
$$\text{Compression } x_2 = 10\ (1 + \cos 318^\circ\ 11' - K)$$
$$= 10\ (1 + {\cdot}7453 - {\cdot}056)$$
$$= 16{\cdot}89''.$$

(*b*) Bottom. Reverse the sign of *K* in each of the above calculations.

Admission	0″	Student should verify these.
Cut-off	9·86″	
Release	16·89″	
Compression	18·01″	

We may now assume convenient steam and exhaust pressures, and draw the theoretical indicator diagrams. They are shown in Fig. 12, drawn the same way to make it easier to compare them, and to show the effect of the obliquity of the connecting rod. It is clear that the addition of steam lap to the valve has resulted in expansive working; from *B* to *C* the piston is driven forward by the expansion of the steam.

Steam lap has also resulted in release before the end of the stroke, and compression before the end of the return stroke. Our valve had the same amount of steam lap at each end; but on the upstroke the cut-off is considerably earlier, due to the obliquity of the connecting rod. Release at the top of the stroke is also earlier; but there is later compression at the bottom and therefore less cushioning, whereas we should prefer to have more cushioning at the bottom than at the top in vertical engines. The upstroke diagram has a lesser area than the downstroke diagram, and here again we should prefer the reverse to be the case, because on the upstroke the heavy reciprocating parts have to be lifted.

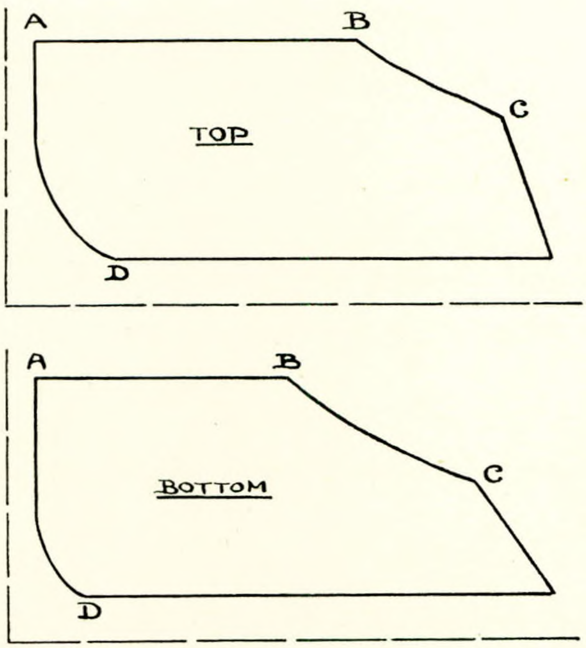


FIG. 12.

What will be the effect of increasing the steam lap? The student will perhaps feel able to answer this question at once, but he should work through the following exercise.

Example 6.

Problem as previous one, but steam lap to be $2\frac{1}{4}$in. You need work for the top of the piston only.

The answers are :—

Angle of advance 56° 26′

Crank positions at four events are

0°, 67° 8′, 123° 34′, 303° 34′

Piston positions are

0, 7·19, 16·4, and 14·66″.

The indicator diagram is as Fig. 13. The increased lap has

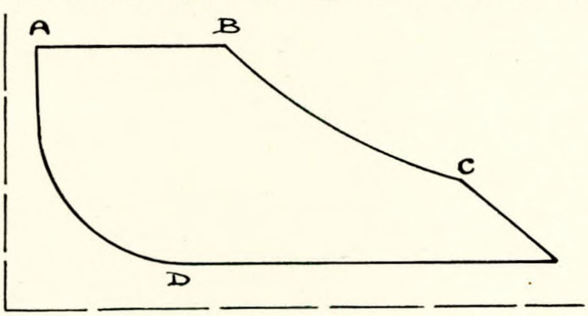


FIG. 13.

resulted in earlier cut-off, earlier release, and earlier compression. There is more expansion and more cushioning, in fact the latter is tending to become excessive.

Now let us try the effect of adding exhaust lap.

Example 7.

Same problem as preceding one, but with $\frac{1}{2}$″ exhaust lap. The steam lap is the same, so the angle of advance, admission, and cut-off are unaltered. But the eccentric positions at release and compression are now as in Fig. 14. The angle α we might call the angle of exhaust lap.

$$\tan \alpha = \frac{\frac{1}{2}}{3} = \frac{1}{6} = 0{\cdot}1667$$

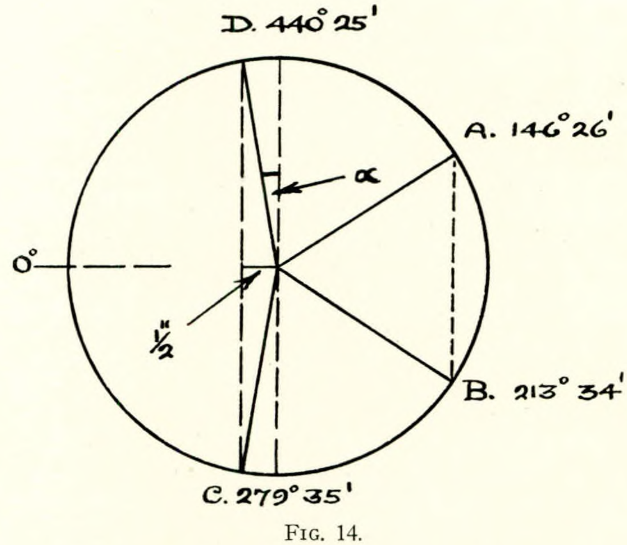


FIG. 14.

$$\therefore \alpha = 9°35'.$$

Thus release angle $= 270° + 9°\ 35' = 279°\ 35'$
and compression angle $= 450° - 9°\ 35' = 440°\ 25'$.

The crank angles at the four events we now find by subtracting 146° 26′. They are, in order, 0°, 67° 8′, 133° 9′, and 293° 59′.

The piston position at cut-off is the same as before, 7·19″.

Release. $\quad \frac{x_1}{10} = 1 - \cos 133°\ 9'' + K$

$$= 1 + 0{\cdot}6839 + 0{\cdot}067$$

$$\therefore x_1 = 17{\cdot}51''$$

Compression.

$$-\frac{x_2}{10} = 1 - \cos 133°\ 9' + K$$

(The angle 113° 59′ is the angle from the bottom position, $= 293°\ 59' - 180°$).

$$\therefore \frac{x_2}{10} = 1 + 0{\cdot}4064 - 0{\cdot}106$$

$$\therefore x_2 = 13''.$$

The indicator diagram may now be drawn, Fig. 15. Inside

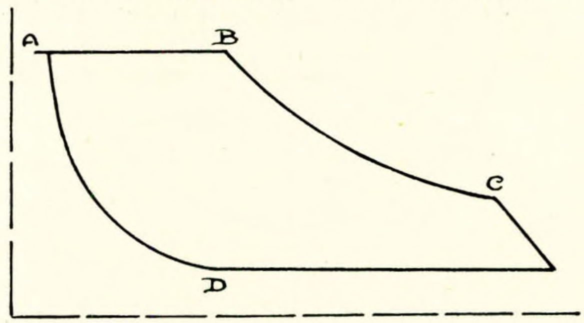


FIG. 15.

lap has retarded the moment of release, which is an improvement; but it has increased the amount of cushioning, which is now altogether excessive.

Example 8.

A slide valve for a vertical engine, Fig. 16, has the following particulars :—

Steam laps : $2\frac{3}{16}$″ top, $2\frac{1}{16}$″ bottom.
Exhaust laps : $-\frac{1}{4}$″ top, $+\frac{1}{4}$″ bottom.

The lead at the top is to be $\frac{1}{8}$″. Travel $7\frac{1}{2}$″. Piston stroke is 42″, length of connecting rod 7′ 6″. What will the bottom lead be? Calculate the angle of advance, and angles of steam lap and lead (top and bottom); also the piston displacements at

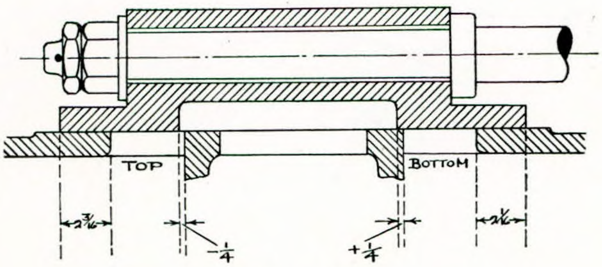


FIG. 16.

the four events, top and bottom. Draw the theoretical indicator diagrams, assuming steam and exhaust pressures of 60 and 15lb./sq. in. (gauge) respectively.

When the piston is on top dead centre, the valve must be below the mid-position of Fig. 16 by an amount

$$\text{top steam lap}+\text{top lead}$$
$$=2\tfrac{3}{16}+\tfrac{1}{8}=2\tfrac{5}{16}''$$

The angle of advance must be sufficient to secure this displacement; and then, when the piston is on bottom dead centre, the valve will be above mid-position by $2\frac{5}{16}''$; and as the bottom steam lap is only $2\frac{1}{16}''$, the bottom lead will be $2\frac{5}{16}''-2\frac{1}{16}=\frac{1}{4}''$.

Note carefully that, always,

steam lap+lead (bottom)=steam lap+lead (top).

$$\text{Angle of advance } \sin\theta=\frac{2\frac{5}{16}}{3\frac{3}{4}}=\frac{37}{60}=0{\cdot}6167$$
$$\therefore\theta=38^\circ\ 5'.$$

$\therefore$Crank must always be $90^\circ+38^\circ\ 5'=128^\circ\ 5'$ behind eccentric.

Top of cylinder :—

$$\text{Angle of steam lap } \sin\phi=\frac{2\frac{3}{16}}{3\frac{3}{4}}=\frac{7}{12}=0{\cdot}5833$$
$$\therefore\phi=35^\circ\ 41'$$
$$\therefore\text{Angle of lead}=\theta-\phi=2^\circ\ 24'$$

Eccentric positions :—

Admission. 90°+angle of steam lap$=90^\circ+35^\circ 41'=125^\circ\ 41'$.

Cut-off. $270^\circ-35^\circ\ 41'=234^\circ\ 19'$.

Release. Here note carefully that the exhaust lap is negative, and that release occurs *before* the valve gets back to mid-position. The angle of exhaust lap is $-\alpha$.

$$\sin\alpha=\frac{\frac{1}{4}}{3\frac{3}{4}}=\frac{1}{15}=0{\cdot}06667$$
$$\therefore\alpha=3^\circ\ 49',\text{ and eccentric angle is}$$
$$270^\circ-3^\circ\ 49'=266^\circ\ 11'.$$

Compression. $450^\circ+3^\circ\ 49'=453^\circ\ 49'$.

These eccentric positions are shown in Fig. 17a.

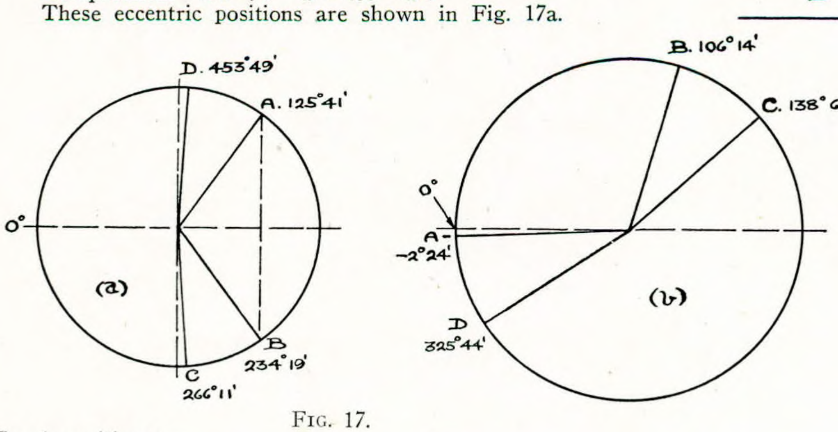


FIG. 17.

Crank positions :—

These we obtain by subtracting $128^\circ\ 5'$ from each of the eccentric positions, Fig. 17b.

Piston positions :—

$$\text{Admission.}\quad \frac{x_2}{21}=1-\cos\theta-K.$$

The value of N is $\frac{90}{21}=4{\cdot}3$ nearly.

K, from the graph,=a negligible quantity for the small angle $2^\circ\ 24'$.

$$\therefore\frac{x_2}{21}=1+\cos 2^\circ\ 24'$$
$$=1+0{\cdot}999'$$
$$\therefore x_2=41{\cdot}98'',\text{ practically on top dead centre.}$$

$$\text{Cut-off}\quad \frac{x_1}{21}=1-\cos 106^\circ\ 14'+K$$
$$=1+0{\cdot}2796+0{\cdot}108$$
$$\therefore x_1=29{\cdot}14''.$$

$$\text{Release}\quad \frac{x_1}{21}=1-\cos 138^\circ\ 6'+K$$
$$=1+0{\cdot}7443+0{\cdot}052$$
$$\therefore x_1=37{\cdot}72''.$$

$$\text{Compression}\quad \frac{x_2}{21}=1-\cos 145^\circ\ 44'-K$$
$$=1+{\cdot}8268-0{\cdot}036$$
$$\therefore x_2=37{\cdot}61''.$$

Bottom of cylinder. This may be left as an exercise for the student. Note that the exhaust lap is positive here.

The answers are: Angle of steam lap $33^\circ\ 22'$.

Eccentric positions.

$123^\circ\ 22'$, $236^\circ\ 38'$, $273^\circ\ 49'$, $446^\circ\ 11'$.

Crank positions.

$-4^\circ\ 43'$, $108^\circ\ 33'$, $145^\circ\ 44'$, $318^\circ\ 6'$.

Piston positions.

0 (practically), 25·5, 37·6″, and 37·72″ return stroke.

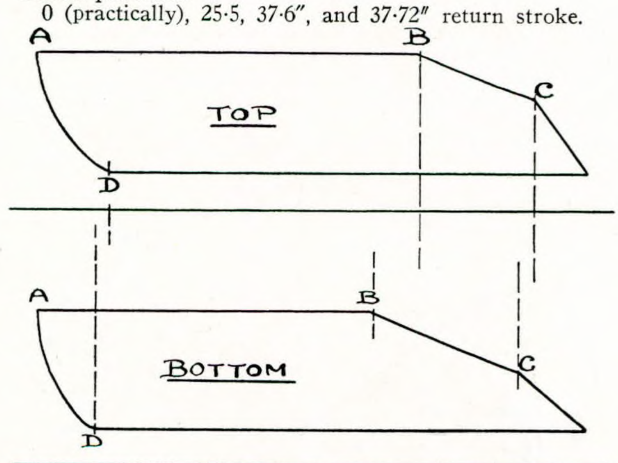


FIG. 18.

The indicator diagrams are shown in Fig. 18.

Reuleaux Valve Diagram.

Various methods of solving valve problems graphically have been devised. For ordinary problems the Reuleaux diagram is perhaps the simplest of all. Fig. 19 illustrates its basic

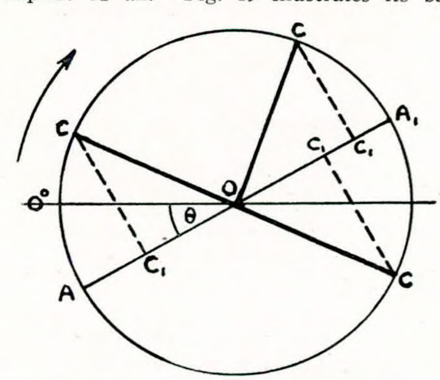


FIG. 19.

principle. On the crank-pin circle a diameter AOA_1 is set back at an angle θ from the line of centres. OC being any position of the crank, the perpendicular CC_1 on to AOA_1 represents to scale the displacement of the valve from mid-position. If OC is drawn equal to the half-travel of the valve, then CC_1 is the valve displacement, to the right or left of mid-position according as CC_1 is above or below AOA_1.

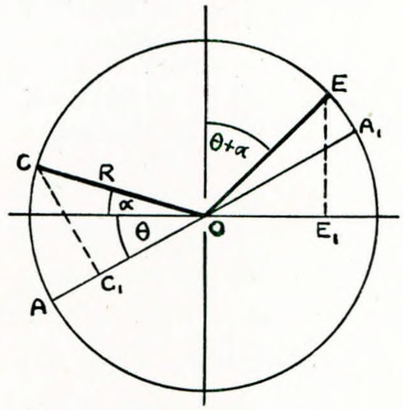


FIG. 20.

The proof is simple. Referring to Fig. 20, the crank OC is shown at an angle α from inner dead centre. The eccentric will also have moved through an angle α, and is at E. The valve displacement is OE_1, and this is equal to CC_1, for each $= R \sin(\theta+\alpha)$.

If now a line 1–2 is drawn parallel to AA_1 at a distance

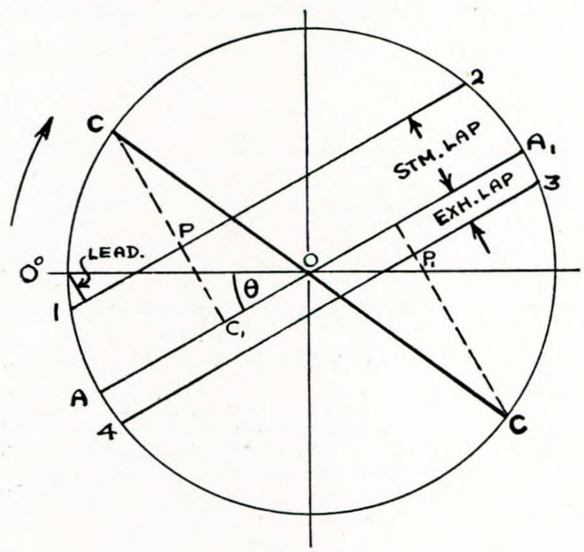


FIG. 21.

from it equal to the steam lap (Fig. 21), it subtracts this lap from the displacement CC_1 and so gives CP, the port opening to steam. On the other side of AA_1 the exhaust lap line 4–3 is drawn, so giving CP_1, the port opening to exhaust when the crank occupies a position below AA_1. The points 1, 2, 3, 4 are the crank positions at admission, cut-off, release, and compression respectively. A perpendicular from the 0° crank position (dead centre) on to the steam lap line 1–2 is the lead. A very useful feature of the valve diagram is that it shows at a glance the effect produced by altering one or more of the various characteristics of the valve: for example, other things remaining the same, a reduction of the steam lap will increase the lead, retard the moment of release, and give a larger maximum port opening to steam.

The diagram may be extended to give piston positions on

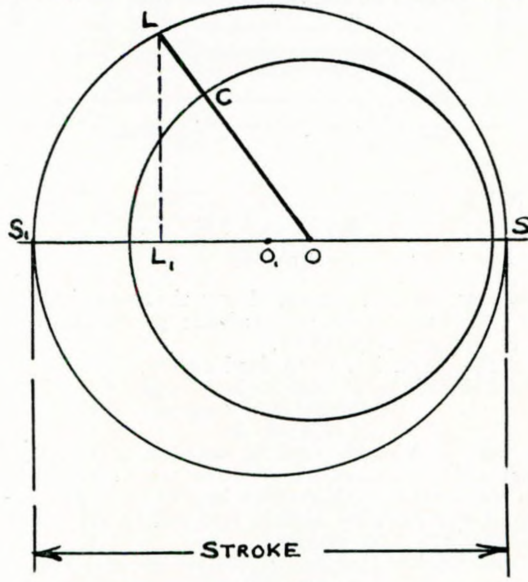


FIG. 22.

the stroke, allowing for obliquity. With centre O_1 (Fig. 22) and any convenient radius O_1S_1, a circle S_1LS_2 is drawn outside the Reuleaux valve circle. Make $OO_1 = \frac{OS_1}{2N}$, N being the connecting rod to crank ratio. Then, OC being any position of the crank, produce OC to L on the outer circle, and draw LL_1 perpendicular to S_1S_2. If S_1S_2 represents the stroke, L_1 represents the piston position; or, the fraction of the stroke completed $= \frac{S_1L_1}{S_1S_2}$. Evidently it will be a convenience to take S_1S_2 equal to 10in.,

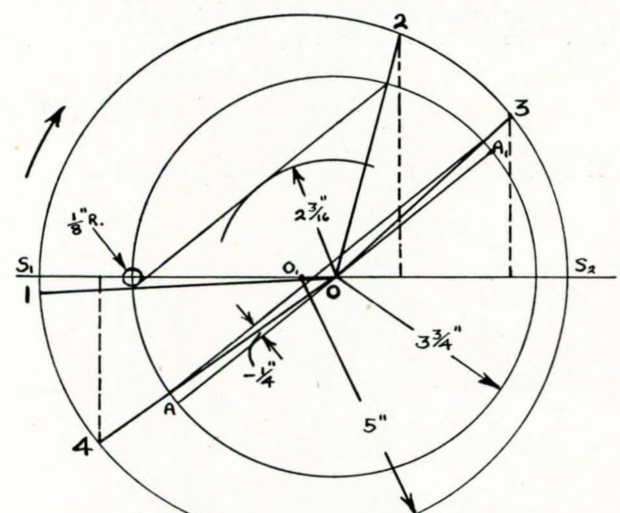


FIG. 23.

or if this is too large, say 5in. In Fig. 23 is shown the solution, by this graphic method, of Example 8 (top). The student should draw this diagram for himself, making the valve circle 7½in. dia., and the obliquity circle 10in. dia., and so verify the answers already obtained by calculation. The diagram is constructed as follows: draw the valve circle, radius 3¾in., and the lead circle, radius ⅛in. With radius $2\frac{3}{16}$in. (steam lap) strike an arc, and then draw the steam-lap line tangential to this arc and the lead circle. Through the centre of the valve circle draw a diameter parallel to the steam-lap line, and then draw the exhaust lap line at a distance $-\frac{3}{4}$in. from this diameter. The crank positions at the four events are thus determined. Choosing a radius 5in. for the obliquity circle, the centre of this will be to the left of the valve circle centre by a distance $=\frac{R}{2N}=\frac{5}{2\times\frac{90}{21}}=\frac{7}{12}$in. The crank positions are then produced to the outer circle, and vertical projectors on to the 10in. dia. stroke line determine the required piston positions. The diagram for the bottom end of the valve is left as an exercise for the student.

8. *Piston Valves.*

If the back of the ordinary slide-valve of Fig. 2 is exposed to steam, the resultant pressure between the face of the valve and its seat is much greater than is necessary to prevent leakage of steam, and work is absorbed in overcoming the consequent excessive friction. To reduce this loss of power, and the corresponding wear, means have been devised to shield part of the back of the valve from high steam pressure. Friction is eliminated, however, more effectively by the use of piston valves, of which a simple solid "dumb-bell" type is illustrated in Fig. 24.

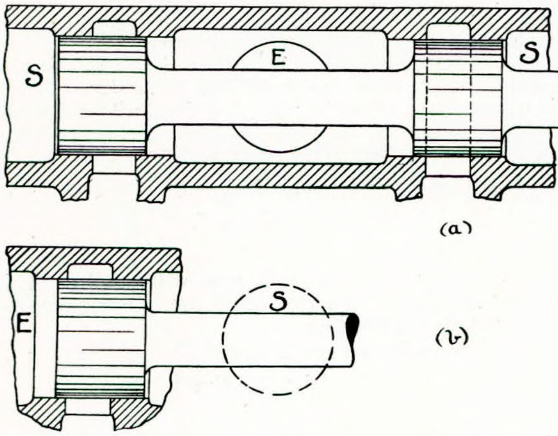


FIG. 24.

Piston valves slide to and fro just as ordinary slide valves do, and are provided with steam and exhaust laps in the same way. The valve of Fig. 24(a) is arranged to admit steam from the outside and exhaust on the inside, in the usual manner. In Fig. 24(b) is shown one end of a piston valve which admits steam on the inside, between the pistons, and exhausts on the outside, an arrangement which ensures that the valve chest stuffing box shall be subject to low pressure exhaust steam only. The steam lap is now on the inside, and exhaust lap on the *outside;* and this requires a different setting of the eccentric.

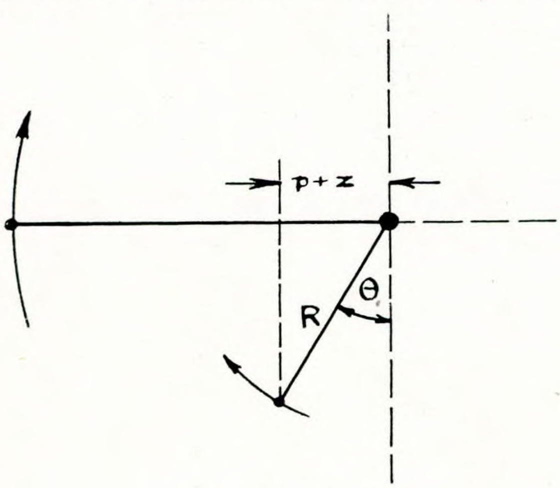


FIG. 25.

Fig. 24(b) shows the top end of the valve in mid-position. When the piston is on top dead centre, the valve must be above mid-position by an amount=steam lap+lead, and be moving upwards so as to be opening the port to steam. Evidently the relative positions of crank and eccentric must be as in Fig. 25. Thus *for the inside steam piston valve, the eccentric must be behind the crank by an angle* $(90-\theta)$, instead of ahead of it by an angle $90+\theta$ as for the outside steam valve of Fig. 24(a).

In each case, $\sin\theta=\frac{\text{steam lap+lead}}{R}$.

Example 9.

The piston valve for the H.P. cylinder has a travel of 7″. The steam lap is $1\frac{15}{16}$″ (bottom) $2\frac{1}{16}$″ (top), top lead $\frac{1}{4}$″. The stroke is 50″, ratio $\frac{\text{con. rod}}{\text{crank}}=4\frac{1}{2}$. Find (i) the bottom lead, (ii) maximum port openings, top and bottom, (iii) angle of keying for inside steam admission, (iv) position of piston on downstroke at cut off.

(i) $1\frac{15}{16}+\text{bottom lead}=2\frac{1}{16}+\frac{1}{4}$

$\therefore$bottom lead $=\frac{1}{8}+\frac{1}{4}=\frac{3}{8}$″.

(ii) Maximum port opening

$=$Half travel$-$steam lap

$=3\frac{1}{2}-2\frac{1}{16}=1\frac{7}{16}$″ (top).

and $=3\frac{1}{2}-1\frac{15}{16}=1\frac{9}{16}$″ (bottom).

(iii) $\sin\theta=\frac{2\frac{1}{16}+\frac{1}{4}}{3\frac{1}{2}}=0{\cdot}6607$

$\therefore\theta=41°\ 21'$.

$\therefore$Angle of keying$=90-\theta=48°\ 39'$.

Eccentric must be $48°\ 39'$ behind crank.

(iv) Angle of steam lap.

$\sin\phi=\frac{2\frac{1}{16}}{3\frac{1}{2}}=0{\cdot}5893\ \therefore\phi=36°\ 6'$.

Cut off occurs when the eccentric makes $90°-36°\ 6'=53°\ 54'$ with line of centres on return downstroke of valve. The crank angle will then be

$53°\ 54'+48°\ 39'=102°\ 33'$ from top dead centre.

Length of crank$=25''=R$.

$$\therefore\frac{x}{25}=1-\cos 102°\ 33'+K$$
$$=1+0{\cdot}2173+0{\cdot}107$$
$$=1{\cdot}3243$$
$$\therefore x=33{\cdot}11''.$$

Example 10.

For student. Find piston position at cut off on upstroke. Answer, 28·86″.

9. *Readjustment of Lead.*

Advancing the eccentric increases the lead equally at both ends, and the *mean lead is increased by the same amount.*

Advancing the valve on its spindle (or adjusting the effective length of the valve rod) by an amount x, *increases the lead by an amount x at one end, and reduces it by the same amount at the other; the mean lead is unaltered.*

Example 10.

(Outside-steam valves).

(i) Present leads. $\left.\begin{matrix}\text{Bottom } \frac{1}{4}'' \\ \text{Top } \frac{1}{8}''\end{matrix}\right\}\text{Mean}=\frac{3}{16}''$

Required leads. $\left.\begin{matrix}\text{Bottom } \frac{5}{16}'' \\ \text{Top } \frac{3}{16}''\end{matrix}\right\}\text{Mean}=\frac{4}{16}''$

The mean lead must be increased by $\frac{1}{16}$″, and so the eccentric must be advanced by this amount, *i.e.* by an angle sufficient to produce it. Adding this $\frac{1}{16}$″ to each of the present leads, the required leads result, and so no alteration of position of the valve on its spindle is necessary.

(ii) Present leads. $\left.\begin{matrix}\text{Top } \frac{1}{8}'' \\ \text{Bottom } \frac{1}{4}''\end{matrix}\right\}\text{Mean}=\frac{3}{16}''.$

Required leads. $\left.\begin{matrix}\text{Top } \frac{3}{16}'' \\ \text{Bottom } \frac{3}{16}''\end{matrix}\right\}\text{Mean}=\frac{3}{16}''.$

The eccentric setting must not be altered, but the valve must be lowered $\frac{1}{16}$″. The required leads then result.

(iii) Present leads. $\left.\begin{array}{l}\text{Top } \frac{1}{8}'' \\ \text{Bottom } \frac{1}{4}''\end{array}\right\}\text{Mean}=\frac{3}{16}''$

Required leads. $\left.\begin{array}{l}\text{Top } \frac{1}{16}'' \\ \text{Bottom } \frac{1}{8}''\end{array}\right\}\text{Mean}=\frac{1\frac{1}{2}''}{16.}$

The mean lead must be reduced by $\frac{1\frac{1}{2}}{16}$, and so the eccentric must be set back by this amount, which will give leads

$$\text{Top } \tfrac{1}{8}-\frac{1\frac{1}{2}}{16}=\frac{\frac{1}{2}}{16}''$$

$$\text{Bottom } \tfrac{1}{4}-\frac{1\frac{1}{2}}{16}=\frac{2\frac{1}{2}''}{16}$$

To obtain the required leads we must now evidently lower the value $\frac{\frac{1}{2}}{16}=\frac{1}{32}''$.

(iv) Present leads. $\left.\begin{array}{l}\text{Top } \frac{1}{8}'' \\ \text{Bottom } \frac{1}{4}''\end{array}\right\}\text{Mean}=\frac{3}{16}''$

Required leads. $\left.\begin{array}{l}\text{Top } \frac{1}{4}'' \\ \text{Bottom } \frac{1}{2}''\end{array}\right\}\text{Mean}=\frac{3}{8}''.$

Advancing the eccentric the necessary $\frac{3}{16}''$, the leads become:

$$\text{Top} \quad \tfrac{1}{8}+\tfrac{3}{16}=\tfrac{5}{16}''$$
$$\text{Bottom } \tfrac{1}{4}+\tfrac{3}{16}=\tfrac{7}{16}''.$$

Clearly the valve must now be raised $\frac{1}{16}''$.

(v) Present leads. $\left.\begin{array}{l}\text{Bottom } +\frac{1}{2}'' \\ \text{Top } -\frac{1}{4}''\end{array}\right\}\text{Mean}=\frac{+\frac{1}{2}-\frac{1}{4}}{2}=\frac{1}{8}''.$

Required leads. $\left.\begin{array}{l}\text{Top } +\frac{1}{8}'' \\ \text{Bottom } +\frac{3}{8}''\end{array}\right\}\text{Mean}=\frac{2''}{8}$

Eccentric must be advanced $\frac{1}{8}''$. The leads become

$$-\tfrac{1}{4}+\tfrac{1}{8}=-\frac{2}{16}''$$

$$\text{and } +\tfrac{1}{2}+\tfrac{1}{8}=+\frac{10''}{16}$$

Lowering the value by $\frac{4}{16}''$ then gives the required leads, because

$$-\frac{2}{16}+\frac{4}{16}=\tfrac{1}{8}''$$
$$+\frac{10}{16}-\frac{4}{16}=\tfrac{3}{8}''.$$

Thus the eccentric must be advanced $\frac{1}{8}''$ and the valve lowered $\frac{1}{4}''$.

(v) Owing to slipping of the eccentric sheave on shaft, and of valve on spindle, the leads are found to be: top $-\frac{3}{16}''$, bottom $-\frac{1}{4}''$. What adjustments must be made to give leads of $+\frac{3}{16}''$ top, $+\frac{3}{8}''$ bottom?

$$\text{The mean present lead}=\frac{-\frac{3}{16}-\frac{4}{16}}{2}=\frac{-3\frac{1}{2}}{16}$$

$$\text{The mean required lead}=\frac{+\frac{3}{16}+\frac{3}{8}}{2}=+\frac{4\frac{1}{2}}{16}$$

$\therefore$ the eccentric must be advanced $\frac{8}{16}=\frac{1}{2}''$

The leads then become

$$\text{Top} \quad -\tfrac{3}{16}+\tfrac{8}{16}=\tfrac{5}{16}''$$
$$\text{Bottom} -\tfrac{1}{4}+\tfrac{8}{16}=\tfrac{4}{16}''$$

Raising the value $\frac{1}{8}''$ will now give the required leads, because

$$\tfrac{5}{16}-\tfrac{2}{16}=\tfrac{3}{16}''$$
$$\text{and } \tfrac{4}{16}+\tfrac{2}{16}=\tfrac{3}{8}''$$

If these are the original leads, the eccentric must have worked back on the shaft $\frac{1}{2}''$, and the valve dropped down $\frac{1}{8}''$.

(vi) (For student). Find how much the eccentric must have slipped back on the shaft, and the valve dropped, if the leads are found to be, top $-\frac{3}{4}''$, bottom $-1\frac{1}{8}''$, the original leads having been, top $\frac{1}{4}''$, bottom $\frac{1}{8}''$.

Answer, eccentric worked back $1\frac{5}{16}''$, valve dropped $\frac{5}{16}''$.

With inside steam piston valves, to increase the lead equally at both ends, the eccentric must be advanced just as for outside steam valves. But to increase the top lead and reduce the bottom lead it is necessary to raise the valve, not lower it as for outside steam valves. Thus, if the data of Example 10 referred to inside steam piston valves, the answer to (iii) would be

$$\text{Set back eccentric by } \frac{1\frac{1}{2}}{16}=\tfrac{3}{32}''$$

$$\text{Raise the valve } \tfrac{1}{32}''.$$

Example 11.

(For student). Inside-steam piston valve. Present leads are, top $\frac{1}{4}''$, bottom $\frac{1}{8}''$. Required leads are, top $\frac{1}{8}''$, bottom $\frac{3}{8}''$. Find the necessary adjustments.

Answer, advance eccentric by $\frac{1}{16}''$.
lower valve by $\frac{3}{16}''$.

AKROYD STUART AWARD PAPER, 1938/39.

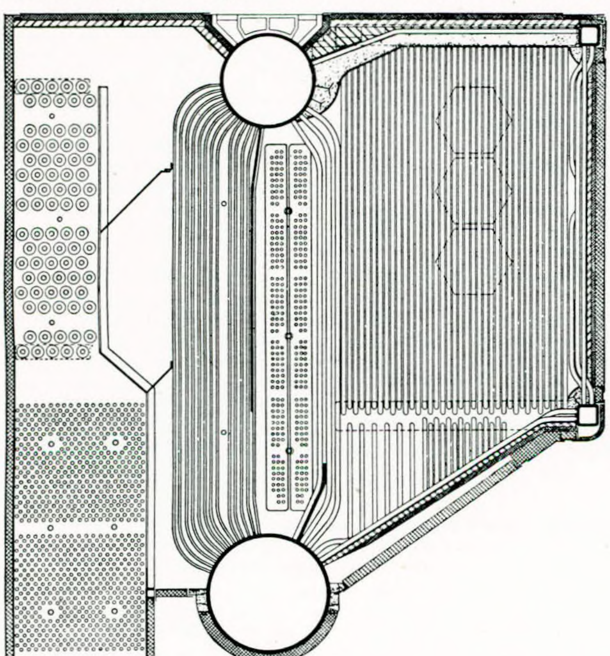
The attention of the Council having been drawn to the fact that an appreciable portion of the above *paper was a reprint, without comment or analysis, from two publications of another author in the Proceedings of the Institution of Mechanical Engineers for 1932 and 1939, the matter was referred to Mr. Mukerji, who regrets that he omitted to include an explicit statement to this effect in his essay and now makes full acknowledgment.

*"The Origin and Development of the Heavy-oil Engine", Vol. LI, Part 10 of TRANSACTIONS, November, 1939, pp. 299-318.

Abstracts of the Technical Press

Foster Wheeler Boilers of American C-3 Class Cargo Steamship "Sea Fox".

The cargo steamer "Sea Fox", launched at Kearny, N.J., on the 27th January, is the first of the six steam-driven vessels of the C-3 class to be built. The ship is of the standard C-3 design, with a single screw driven by a set of De Laval turbines of 8,500 s.h.p. designed to give her a speed of 16½ knots and fitted with double-reduction gearing by means of which the speed of the propeller is limited to 85 r.p.m. at normal full power. Steam at a pressure of 450lb./in.^{2} and superheat temperature of 750° F. is supplied by two oil-fired Foster Wheeler water-tube boilers arranged in a single casing. These boilers are of the so-called "D"-type, this designation originating from the general shape of a cross-section through the boiler. The 42-in. diameter steam-drum of each boiler is placed directly over the 30-in. water drum, the distance between centres being about 16ft. and the drums being connected by boiler tubes which form the vertical stroke of a capital "D". The furnace is built to one

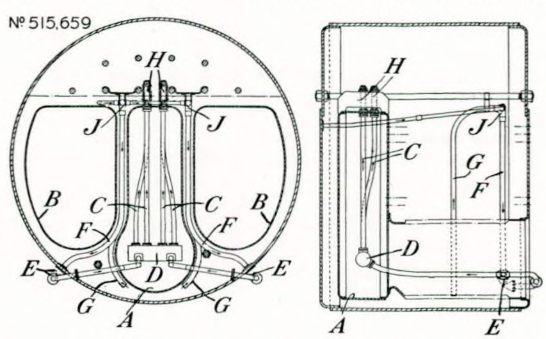


A cross-section diagram of the Foster Wheeler boiler installed in the "Sea Fox".

side, occupying a space approximately the shape of the area within the curved line of the capital "D" and being completely water-cooled at both sides, front and back by closely spaced 2-in. boiler tubes. Three oil fuel burners, arranged in a single vertical row, extend into the furnace parallel to the steam and water drums. The gases from the furnace pass across the first bank of boiler heating surface consisting of three rows of widely spaced 2-in. tubes, and then through a convection superheater. The latter is made up of 1¼-in. tubing of the horizontal "U"-bend type and is self-draining. The gases from the superheater enter the last bank of boiler heating surface near the top and pass downward, lengthwise of the 14 rows of 1¼-in. tubes, under a baffle and then up through the economiser and air heater to the boiler uptake. The economiser is of the marine, extended-surface type, providing six times the heat-exchange surface of bare tubing. The air heater is of the tubular type with 2-in. tubes 9ft. long, arranged horizontally to provide two passes for the air inside the tubes, while the gases flow round the outside. Four mechanical soot blowers are fitted to the air heater. The furnace walls composed of water tubes are backed with several layers of insulation which reduce radiation losses through the steel casing to a minimum. All the boiler tubes are readily accessible for cleaning through two manholes, one in the steam drum and the other in the water drum, while the water tubes can be cleaned and examined internally by opening handhole plugs opposite the tube connections. The furnace volume is 1,040 cu. ft. and being of approximately square cross-section, provides ample space and time for thorough combustion of the fuel, this process being carried out under pressure, as the air from the air heaters is forced into the furnace by blowers. This arrangement eliminates the complications of a closed stokehold and gives maximum control of firing conditions with the result that the overall efficiency of the boiler in service is over 88 per cent. The normal evaporative capacity of each boiler is 16·7 tons/hr. at 465lb./in.^{2} pressure at the superheater outlet and 765° F. final steam temperature, but the maximum evaporative capacity is just over 25 tons/hr. at a steam pressure of 525lb./in.^{2}. The heating surface of each boiler, including that of the water walls, is 4,110ft.^{2}, that of the economiser being 3,204ft.^{2} and that of the air heater 2,807ft.^{2}. The boilers are claimed to be exceptionally simple to maintain and operate in spite of the relatively high steam pressure. The accompanying diagram shows a cross-section through one of the boilers of the "Sea Fox".—*"The Nautical Magazine", Vol. 130, No. 2, February, 1940, pp. 10-13 and 25.*

Promoting Water Circulation in Steam Generators.

A recently-published British patent bearing the above title in the specification, applies to multitubular Scotch boilers. As shown in the accompanying drawings, the boiler has a central combustion chamber *A* and two wing combustion chambers *B*. Within the former are steam-generating tubes *C* which ascend from a common header *D* at about bridge level within the lower part of the combustion chamber and are connected to the bottom of the boiler's water space by means of two pipes led through the interior of the respective furnaces, below the level of the grate (not shown), to isolating valves *E* fitted to the underside



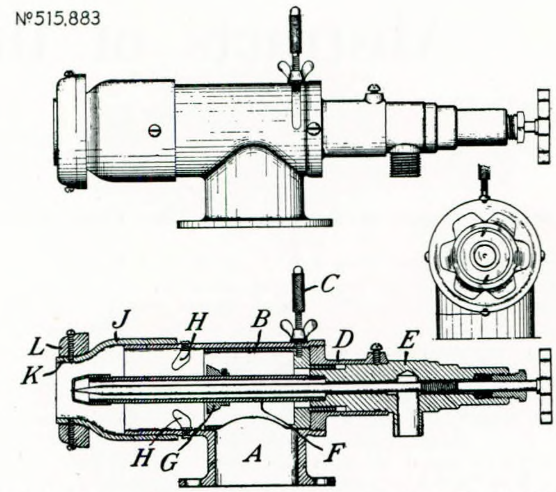
of the shell and each communicating with internal pipes *F* and *G*. The tubes *C* connect at their upper ends with conduit elements constituted by ported tubular bolts *H* fitted to the combustion chamber crown, which serve as tie-bolts for the crown-supporting girders. Each of the tubular tie-bolts houses a non-return valve element and is provided with lateral outlet ports below the low water level, being vented at its upper end of the steam space. The arrangement and disposition of the tubes enable them to function as steam-generating and water-circulating tubes. Feed water is supplied to the boiler through two injector nozzles each housed in a branch-piece *J* just above the level of the crowns of the combustion chambers, *i.e.*, at the top of the water space, and interposed between the upper ends of the internal pipes *F* and *G*, of which the pipes *G* lead water from the bottom of the water space to the branch-piece inlets, while the pipes *F* connect the branch-piece outlets to the isolating valves *E*. Feed water is supplied to the injector nozzles through an internal feel pipe from a check valve (not shown) at the back of the boiler.—*"The Engineer", Vol. CLXIX, No. 4,388, 16th February, 1940, p. 171.*

The First Twenty Years of Screw Propulsion.

The paper is the second of two dealing with the development of the screw propeller from 1838 to 1858. The author describes the difficulties experienced by the advocates of this form of propulsion in overcoming the attitude of the Admiralty authorities who chose to regard steam merely in the light of an auxiliary to sails. By the beginning of 1854 there were 200 commercial screw ships registered in Great Britain, but screw ships on long routes had failed to realise the results anticipated. The screw had no easy triumph over sails or paddle wheels and it was not until the success achieved by the s.s. "Archimedes", built at Millwall in 1838 for the special purpose of demonstrating Smith's screw, which led the Admiralty to fit a screw propeller to the sloop "Rattler", that the screw was introduced into the Royal Navy. The author gives a list of 43 screw ships included in the *Navy List* of January, 1850, and refers to some of the problems involved in the use of screws for propelling the very full or bluff-sterned hulls of sailing ships. Furthermore, the strengthening of wooden ships to withstand the vibration of the machinery and propeller was never successfully accomplished. With iron ships, of course, the problem presented no difficulty. Most of the early naval ships with screw propellers had horizontal engines with two or four cylinders driving a common crankshaft, half-a-dozen had oscillating engines and three had geared engines. Trunk engines quickly became very popular, and soon after 1850, Maudslays introduced the return connecting-rod engine, a form of steeple engine laid on its side. The brass linings of stern-tube bearings in early ships gave endless trouble, which was overcome by the introduction of lignum-vitæ bearings by John Penn. Lifting screws were fitted in naval ships over a long period, the lifting and lowering of a screw in a man-of-war being regarded as an evolution. In the course of a discussion on an engineering paper read by Robinson in 1855, it was stated that in the 24-gun frigate "Dauntless", the screw could be lifted by 30 men in nine minutes in any kind of weather and during any rate of speed without reducing canvas, and that it could be lowered again in six minutes. The 91-gun two-decker "Agamemnon" was the first line-of-battleship to be designed as a screw ship, and soon after the Crimean War the Royal Navy became for all practical purposes a screw navy.—*Paper read by Engineer Captain Edgar C. Smith, O.B.E., R.N., at a meeting of the Newcomen Society, on the 17th January, 1940.*

New Oil Fuel Burner.

A recently-published British patent concerns an improved type of oil fuel burner, the general construction of which is shown in the accompanying illustrations. The body of the burner is cylindrical, and has an opening *A* at one side for the admission of primary air under pressure, the rate at which the air enters being regulated by a barrel valve *B* which can be moved angularly by means of a handle *C* projecting through an arcuate slot in the body. A sleeve *D* at the rear end of the body receives the fuel supply tube assembly which consists of a cylindrical fitting *E* to which fuel is supplied through a laterally-extending union, a tube *F* extending through the body of the burner, an apertured nozzle at the front end of the tube, and a needle valve which controls the effective size of the aperture. An annular baffle or deflector *G* having a coned or chamfered peripheral surface of a diameter considerably less than the



internal diameter of the body is fitted over the tube *F* within the burner body and is axially adjustable along the tube. Projecting inwardly and forwardly from the burner body in front of the baffle are a number of pear-shaped blades *H* mounted on stems which project through the body and rotable within the latter, so that each blade may be adjusted angularly from outside. A sleeve *J* slides over the front end of the burner, its forward end being reduced in diameter to form a nozzle *K*. A ring *L* on the outside of the nozzle has in its inner edge a series of openings through which secondary air can pass to mix with the air and fuel issuing from the nozzle. When fuel and air are supplied to the burner the fuel is discharged from the jet and the air passing through the body is first deflected outwardly by the baffle *G* and then given a swirling motion by the blades *H*. A certain proportion of the air stream is deflected inwardly by the converging walls of the nozzle *K* and cuts across the fuel issuing from the jet and across the remainder of the air travelling along the axially adjacent tube *F*. The primary air and fuel leaving the nozzle are further mixed with secondary air drawn through the openings in the ring *L*. The position of the fuel jet relative to the nozzle can be varied by axial adjustment of the sleeve *J*, the baffle being likewise adjustable axially. As the blades are also individually adjustable angularly, the burner can be set to give good combustion with any fuel and any air pressure within reasonable limits.—*"The Engineer", Vol. CLXIX, No. 4,389, 23rd February, 1940, p. 195.*

A 27-knot Cross-Channel Motorship.

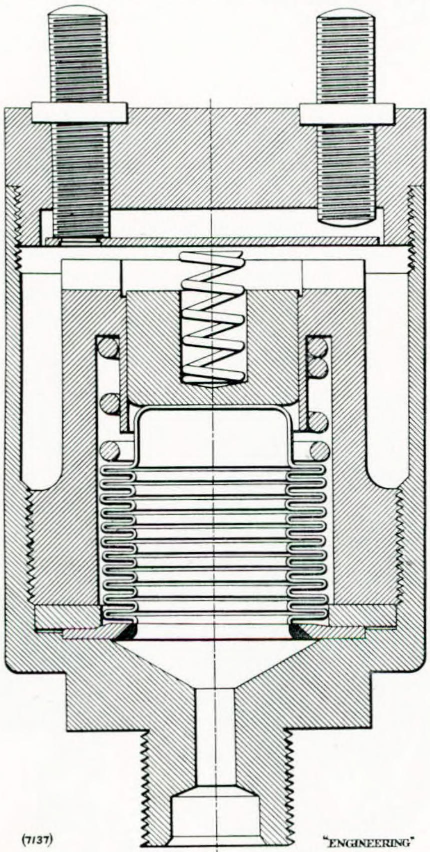
A memorandum submitted to the Royal Netherlands Society by Mr. H. W. Van Tijen, chief engineer of the De Schelde Shipbuilding Yard, concerns the design of a cross-Channel motorship of 2,000 tons displacement having a speed of 27½ knots. The propelling machinery would comprise six 2-stroke Diesel engines developing a total of 20,000 b.h.p. and driving two propeller shafts at 400 r.p.m. through electro-magnetic couplings and helical gearing. The Diesel engines would each have 11 cylinders of 420 mm. diameter with a piston stroke of 500 mm. and there would be two main engine-rooms, the forward one containing two and the after one four main engines. The forward engine room would also contain two 6-cylinder 4-stroke Diesel engines, each developing 240 b.h.p. at 500 r.p.m., and each driving a 140-kW. generator and a starting-air compressor, in addition to a fresh-water cooling pump and a forced-lubrication pump.—*"Journal de la Marine Marchande", Vol. 22, No. 1,088, 8th February, 1940, pp. 148-149.*

Japanese Motorship "Brazil Maru".

The 13,000-ton passenger vessel "Brazil Maru" recently completed at the Mitsubishi Dockyard for the Osaka Shosen Kaisha, is a sister ship of the "Argentina Maru" placed in service by that company last summer. She has a length of 544ft. 6in., a beam of 68ft. 9in., and a designed speed of 21 knots. The propelling machinery consists of two single-acting 2-stroke Sulzer-type engines developing a total of 16,500 b.h.p. Each engine has 11 cylinders of 720 mm. diameter, with a piston stroke of 1,250 mm., and runs at 140 r.p.m. The vessel has accommodation for 101 first-class and about 800 third-class passengers.—*"The Motor Ship", Vol. XX, No. 201, February, 1940, p. 408.*

Electrical Oil-pressure Indicator.

An appliance known as the "R. & S. Electrical Oil Indicator", recently placed on the market by a Coventry engineering firm, is an improved form of electrical switch arranged to break the circuit if the oil pressure fails, and thus give audible or visual warning of the fault, say by de-energising a solenoid, this operation, in turn, completing the bell or visual indicator circuit. Alternatively, in the case of an internal-combustion engine, the indicator can be arranged to short-circuit the ignition, also



through a solenoid, and thus stop the engine. The construction of the indicator is shown in the accompanying illustration. It comprises a brass body into which a brass sleeve is screwed, the sleeve containing a bellows and forming a guide for a plunger. The bellows are of metal, and are sealed by a leather washer clamped between the sleeve and the indicator body. The oil has access to the interior of the bellows through a hole in the bottom of the body, and when the pressure is normal, the bellows are expanded against a return spring which is shown encircling the plunger. The latter, which is of vulcanite, is thus lifted, and the small spring recessed in the plunger is then forced against the contact plate above it, closing the contacts and completing the solenoid circuit. The terminal plate is of moulded vulcanite, and the terminals are of brass. Should the oil pressure fail, the return spring presses the bellows, and thus allows the contact plate to break the circuit.—*"Engineering", Vol. 149, No. 3,867, 23rd February, 1940, p. 213.*

An Automatic Hot- and Cold-water Mixing Valve.

The accompanying sectional drawings (Fig. 5) illustrate the construction of the Leonard thermostatic hot and cold-water mixing valve which, it is claimed, will give a continuous supply of blended water at any desired temperature and pressure of the hot and cold-water supplies. A temperature-adjusting handle is provided for automatic regulation, the temperature in all cases

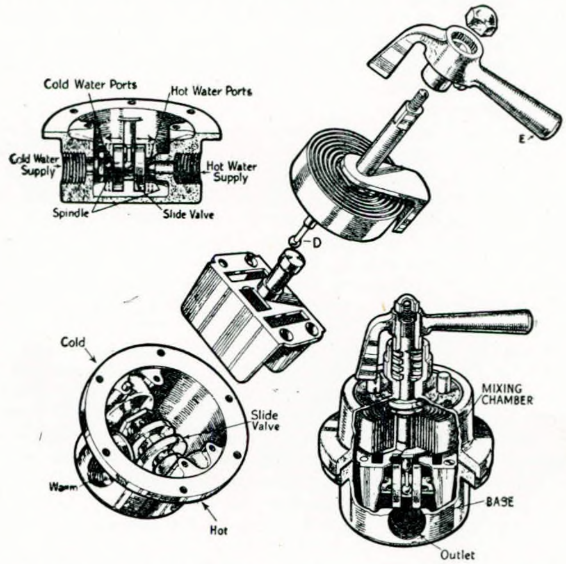


FIG. 5.—*Leonard thermostatic water-mixing valve, type "T".*

being maintained within about 2° F. of the desired range, say 95° to 100° F. for showers. The hot and cold water enters at opposite sides of the bottom casing and flows into the upper half of the casing, which forms a mixing chamber. The latter also houses a bi-metal strip coil, which is extremely sensitive to temperature changes. The coil has a projecting driving pin, which engages with a circular slide valve fitted in the hot and cold-water inlet ports. The movement of the coil causes a movement of the slide valve, and results in the admission of more or less hot and cold water. The larger types of this thermostatic valve have a vertical type of rotating slide valve, while another series of valves is available for mixing live steam and water. The complete arrangement has been approved by the Board of Trade, and it is stated that some 250 Leonard thermostatic hot and cold-water mixing valves have been fitted in the passenger quarters of the R.M.S. "Andes". Several other liners have been similarly equipped, and the valve has also found wide application in hospitals, public baths, factories, etc.——*"The Shipbuilder", Vol. XLVII, No. 365, February, 1940, p. 72.*

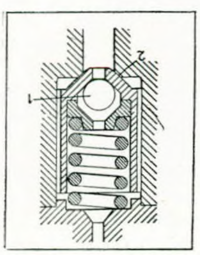
New U.S. Lake Tanker.

The latest addition to the Great Lakes fleet of the Socony-Vacuum Oil Company is the twin-screw motor tanker "Traverse City Socony", built by the Manitowoc Shipbuilding Company, Manitowoc (Wis.). The gross tonnage of the tanker is 2,242 tons, the main dimensions being $290 \times 49\frac{1}{2} \times 20\frac{1}{2}$ft. and the d.w. capacity 4,390 tons on a load draught of 17ft. 3in. She is of the single-deck type, with straight raked stem and cruiser stern, the propelling machinery being aft. The hull is built on the bracketed longitudinal system, the bulkheads and decks dividing it into 22 watertight compartments, including 12 cargo tanks, two ballast tanks, two fuel tanks, pump room, engine room, after peak, fore peak, dry cargo hold and cofferdam. The entire structure in way of the cargo tanks, bunkers and decks was welded, the bulkheads, shell and deck being pre-fabricated in the shop, while the deckhouses were arc-welded and assembled in place as units. The propelling machinery consists of two 6-cylinder 4-stroke Nordberg Diesel engines of the totally-enclosed trunk-piston type, each devoloping 750 h.p. at 300 r.p.m.

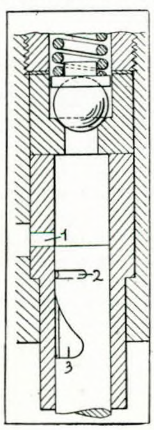
and having cylinders of 16-in. bore and 22-in. stroke. The control gear of the engines is said to be unique, each unit being controlled by a single hand-lever mounted on a common central-control platform placed forward between the engines. When this lever is at its central position the engine is in its "stop" position, but when moved either ahead or astern it starts the engine in that direction and controls the engine speed in accordance with the amount of movement given to the lever in either direction. When the lever is in either of the extreme positions of travel, the engine will run at its maximum speed of 300 r.p.m. in that direction. The reversing gear is actuated by an air-operated reversing cylinder in tandem with an oil cylinder which acts as a governor for the air cylinder so that the reversing of the camshaft may be effected at an even speed. This oil cylinder is also arranged to serve as a reversing cylinder in the event of no air being available. The movement of the air-operated reverse piston rod is transmitted through a rack which engages gears and turns the manœuvring shaft one revolution. When the shaft is thus turned, the camshaft is shifted laterally for the running position in the new direction. This control is positive and silent, reversing the engine from full speed ahead to full speed astern, or *vice versa,* in from 4 to 6 sec., depending merely on the quickness of the operator. Automatic interlocks are provided in the enclosed control boxes and no wrong movement can be made by the operator with the single control lever. While the speed of the engines within the normal range of operation is controlled by the position of the control levers, each is also equipped with an overspeed governor which keeps the engine speed within maximum limits. The service speed of the tanker is $12\frac{1}{2}$ knots on a fuel consumption of 83 gall./hr. The ship's cargo-handling equipment consists of two Diesel-driven pump generator units located at the forward end of the engine room, one at the outboard side of each main engine. The 310-ton rotary cargo pumps are driven through the bulkhead between the engine room and the pump by two Fairbanks-Morse 8-cylinder $5\frac{1}{2}$-in. by $7\frac{1}{2}$-in. air-starting Diesel engines each developing 120 b.h.p. at 900 r.p.m. These engines also drive two 75-kW. 125-volt d.c. generators at their after end. There are two 10-in. main suction lines, one at either side of the centre-line bulkhead. The starboard line draws from Nos. 1, 2 and 3 cargo tanks with suction through the centre-line bulkhead to the corresponding spaces on the port side, whilst the port line draws from Nos. 4, 5 and 6 tanks and then to the other side.—*"The Syren", Vol. CLXXIV, No. 2,267, 7th February, 1940, pp. 291-295.*

New Patents in Oil Engine Fuel Injection Arrangements.

A patent recently granted to Rudolf L'Orange, of Stuttgart, Germany covers a simple means whereby injection can be produced as a preliminary spurt, followed by the main charge after a momentary delay. Referring to the accompanying drawing, the necessary apparatus is contained in the delivery valve of the pump, the valve being fitted with the usual conical seating (2) loaded by a stiff spring. The body of the moving portion is, however, hollow, and houses a loose ball (1) having upper and lower seatings. The fuel first lifts the ball to its upper position and, in doing so, a small quantity is passed. The ball having seated, the fuel flow is momentarily interrupted until the rest of the charge lifts the valve as a whole. After injection, the ball on its return performs the additional duty of relieving the pipe-line pressure. Another method of obtaining two-spurt injection forms the subject of a patent granted to a Manchester engineering firm. In this scheme, the injection is interrupted by the momentary uncovering of an additional release groove in the plunger of the pump. The latter—known as the Gardner injection pump plunger—is shown in the accompanying illustration with an extra channel (2) connected with the usual recess (3). During the plunger ascent, injection commences as soon as the inlet port (1) is covered, but ceases for a moment while the groove (2) passes the port. Thereafter the rest of the fuel is injected normally. The dual spurt is intended solely for use at from half to full load, the initial injection only being used at lesser outputs, *i.e.,* the governor controls only the second injection period.—*"The Oil Engine", Vol. VII, No. 82, February, 1940, p. 319.*



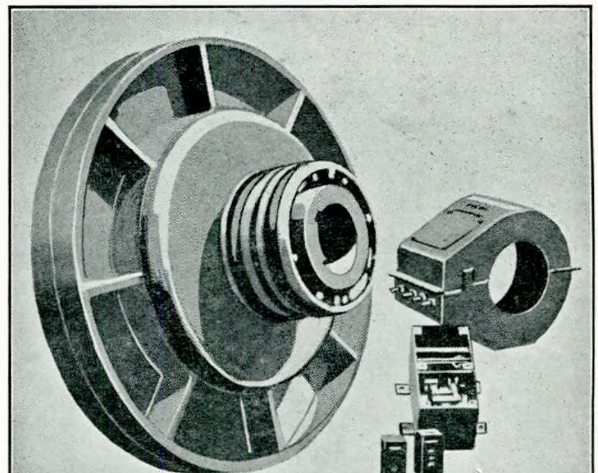
L'Orange delivery valve for two-stage fuel injection.



Gardner injection pump plunger giving two-stage delivery.

Torque Limiter Type of Electro-magnetic Coupling.

An interesting type of coupling is the torque limiter, and the author has designed some which are by far the largest magnetic units made in this country. Four such clutches are fitted in a Diesel-engined dredger in which two engines, each developing 550 b.h.p. at 250 r.p.m., are installed and fitted with clutches at each end, one coupled to the propeller and the other to centrifugal sand pumps. Interlocks prevent the two clutches being engaged at the same time. The complete equipment is remotely controlled from the bridge, and a man is available in the engine room only at odd times. The capacity of the sand pumps is very large, stones 12in. in diameter, planks and chains being readily

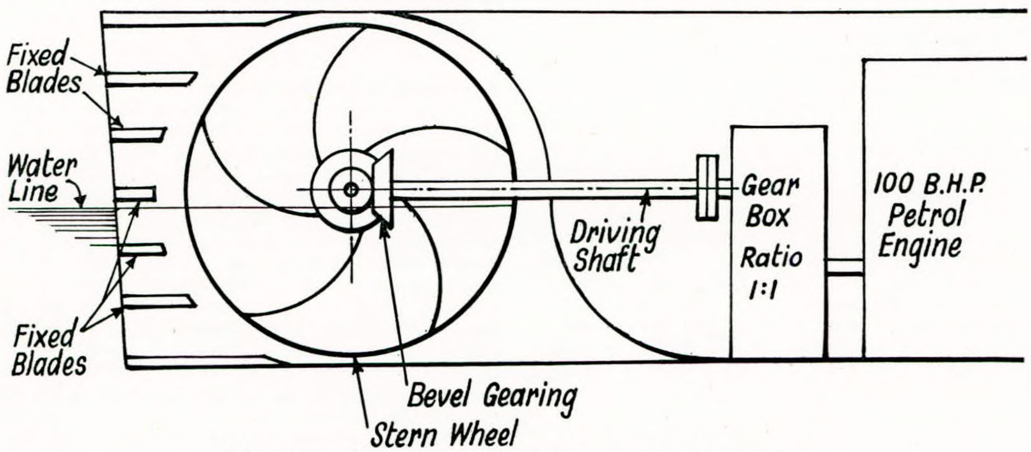


One of a pair of 44-in. diameter couplings for the 500-b.h.p. oil engines of a dredger.

sucked up; it is thus easy for some article such as a wire rope to be drawn into the pump and to jam it. To prevent stalling the engine the coupling is set to slip at slightly above full-load torque. After slipping for a short period the temperature of the linings rises to a specified figure, whereupon thermostats built into the couplings automatically act to disconnect the unit. The engines are kept running continuously and the torque limiter serves to pick up the load, the employment of thermostats and contactor gear providing exceptional means of control. Furthermore, the thermostat feature is a safety device which can be relied upon regardless of attendance. The accompanying illustration shows one of the pair of 44-in. dia. couplings fitted in this dredger.—*H. E. Hunter, "The Oil Engine", Vol. VII, No. 82, February, 1940, pp. 310-311.*

Novel French Sternwheel Propulsion System.

A river barge with a new form of sternwheel drive, which permits a 30-ton capacity on a draught of under 20in., was recently completed by the Ateliers and Chantiers de Bordeaux for the Benin-Niger Railway Company. The barge is of the ordinary flat-bottomed type with a square flat stern at which the propulsion equipment is installed. This comprises a set of eight relately small (33·3-in. diameter) wheels on a common shaft running the full width of the stern. Each of these paddle wheels has a hub on which a series of blades is mounted, but these blades—unlike those of an ordinary paddle wheel—are not flat but helicoidal and mounted in such a manner as to strike the water along one edge, instead of hitting it flat. The eight wheels are mounted on their common shaft so that the ends of the blades of successive wheels form a helix. This arrangement is claimed to reduce vibration and provides a constant thrust. The paddle-shaft bearings are located about 2in. above the waterline of the barge and as the diameter of the paddle wheels is exactly twice the draught and the bearings are above the waterline, the paddles can work in any water in which the barge can float, irrespective of the draught. A curved casing covers the wheels, as shown in the diagrammatic sketch, in order to keep the resistance of the entering water as low as possible. Behind the wheels is a series of fixed horizontal blades which run the full width of the stern, these being arranged so as to leave just enough clearance for the wheels inside the casing. When the wheels revolve they throw up a considerable volume of water which is projected sternwards between the fixed blades in the form of a stream which, it is claimed, adds its thrust component to the normal propulsive effort of the paddle wheels. In order to secure the desired propulsive effect the stern wheels are rotated at a far higher rate than that usual for ordinary paddle wheels, their turning speed being approximately equal to that of an equivalent screw propeller. Power is supplied by a 100-b.h.p. petrol engine installed in a compartment in the stern, with a short driving shaft, flanged on to the output shaft of the gearbox, terminating in a bevel gear which engages a level on the paddle shaft. These gears are protected against the water although there is actually no imperative necessity to render the shaft opening watertight, since the entire shaft is located above the waterline. A simple system of baffles serves to keep out any flying spray churned up by the movement of the wheels. The bevel gear between the driving shaft and the paddle shaft has a ratio of 1 to 1. This system of propulsion can, in certain cases, be made even more simple by placing the engine in the stern with its crankshaft at right angles to the axis of the barge and flanging two half-paddle shafts directly on to the ends of the crankshaft, presumably with friction clutches interposed. The efficiency of this system of propulsion is stated to be quite satisfactory, and considerably greater than that of a screw propeller working in a tunnel. Moreover, the small paddle wheels running at relatively high speeds permit the use of compact internal-combustion engines, instead of large slow-speed steam engines such as are employed with normal directly-coupled paddle wheels. The fact that all the bearings and moving parts, except the ends of the wheel blades, are above the waterline, facilitates their inspection and refit without having to drydock the barge.—*"The Marine Engineer", Vol. 63, No. 751, February, 1940, p. 41.*



Diagrammatic arrangement of the new sternwheel drive

Motor Yacht of Aluminium Alloy.

The motor yacht "Edi" recently built by the Götaverken for the well-known Swedish engineer Torsten Kreuger, has an overall length of 103ft., a beam of 15ft. 7in. and a draught of 3ft. at a displacement of 35 tons, her speed being 28 knots. The hull and propellers are constructed entirely of light metal, the frames and other components are of Bondur (4 per cent. copper, 0·5 per cent. magnesium, 0·5 per cent. manganese and 95 per cent. aluminium), whilst the hull plating is of an alloy containing at least 5 per cent. magnesium, ¼ per cent. manganese and the remainder aluminium, this being proof against corrosion. The deck beams are of silumin (12·5 per cent. silicon and 87·5 per cent. aluminium). The Z-sectioned frames of 50mm. × 35mm. × 35mm. are spaced 300mm. apart, the thickness of the plating varying between 4mm. and 5mm. The propelling machinery is installed right aft, with a fireproof and watertight bulkhead separating the engine room from all the accommodation, which includes six beds and six sofa berths. The crew's quarters are right forward, separated from the other spaces by a watertight bulkhead. The machinery is of a very experimental nature and consists of three Wright Typhoon 12-cylinder V-type engines with a bore of 146mm. and a piston stroke of 159mm., normally rated at 540 h.p. at 1,900 r.p.m. In order to attain maximum economy in fuel consumption, however, the two wing engines have been converted to the Hesselman system, this being the first time such a modification has been carried out on engines of so high a power and speed. With the Hesselman engine, starting is effected on petrol and electric plugs are fitted, but when the engine is warmed up it is changed over to heavy fuel, the fuel oil being supplied by pumps. In the case of the "Edi", the pumps are fitted in the V of the two banks of cylinders, with a drive from the crankshaft, the resulting arrangement being extremely compact and satisfactory in operation. In modifying the engines to the Hesselman system it was necessary to change the inlet valves, and a smaller amount of air is drawn in than when operating as petrol engines. As a result, the engine output is reduced to 425 h.p. at 1,900 r.p.m., but whereas the petrol engines consume 33·35 gall./hr., the consumption as modified is only 25¼-27½ gall./hr. of much cheaper Diesel fuel. The centre engine has also undergone a slight alteration to enable it to operate with a higher compression and on a petrol of increased octane value. By this means the output has been raised to 650 h.p. at 1,900 r.p.m. The yacht has tanks for 770 gallons of petrol in addition to 1,650 gallons of Diesel oil. Electric current is supplied by a 1½-kW. 32-volt dynamo driven by a 2-cylinder petrol engine and there is also a storage battery of 26 Nife accumulators. The propelling machinery can be controlled either from the engine room or from the wheel-house, Hyland remote controls being provided for the reverse gears of the three engines.—*"The Motor Boat", Vol. LXXII, No. 1,855, 10th February, 1940, pp. 110-111.*

Specification of New American Liners.

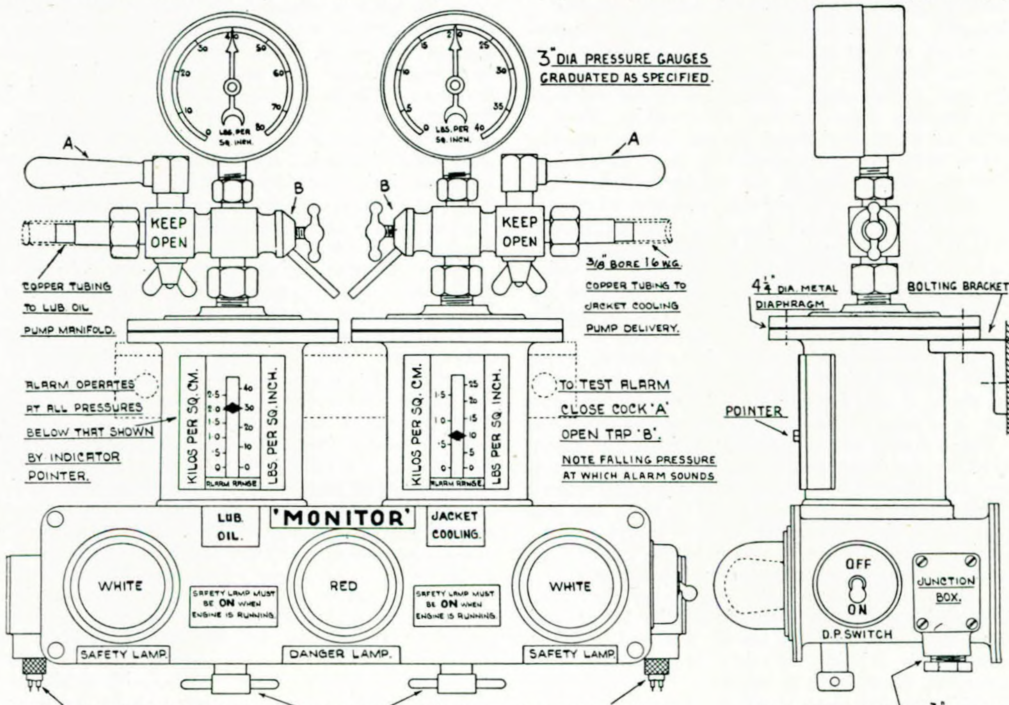
The specification of the two Transpacific liners to be constructed under the auspices of the U.S. Maritime Commission provides for ships of 35,000 tons gross, with a speed of 24 knots. They are to be 759 ft./in. overall length, with a beam of 99ft. and a maximum draught of 32ft. The cargo capacity is to be 535,000 cu. ft. of which 60,000 cu. ft. will be refrigerated cargo space, and there will be accommodation for over 1,000 passengers. The ships are to be driven by steam turbines and the cargo-handling equipment is to be of the most modern type. All cabins will be air-conditioned and the design of the hull provides for extensive sub-division and fireproofing. Light armour-plate will be used in the hull structure and the ships are referred to as "P4 convertible liners". The estimated cost is between fourteen million and twenty-two million dollars per ship. —*"The Shipping World", Vol. CII, No. 2,434, 7th February, 1940, p. 247.*

Submarine Cable Laying.

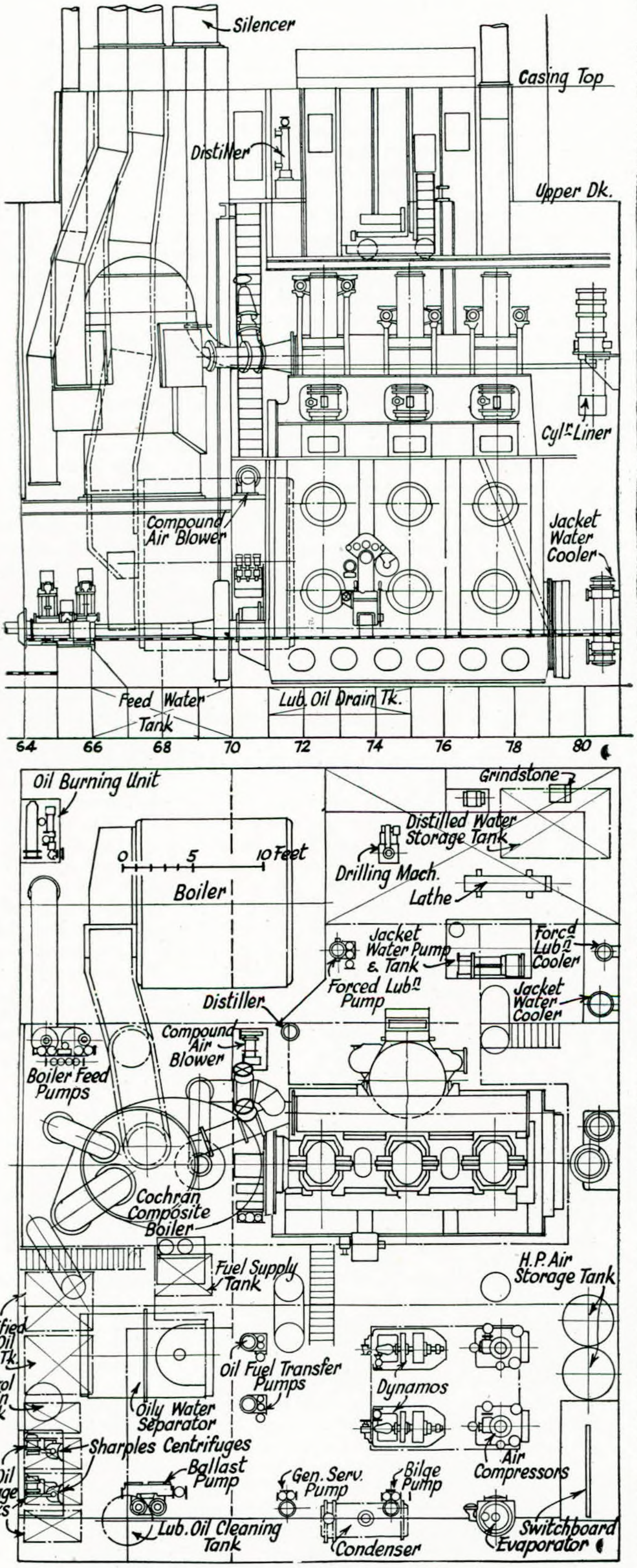
The Western Union Telegraph Company are reported to have developed a device which digs a trough in the ocean bed and lays a submarine cable in it. The plough weighs 10½ tons. Although ploughing can only be carried out at sub-surface depths of up to 2,000ft., a line 4,200ft. long is required to enable the plough to be operated from a cable-laying ship at an angle of 22° from the horizontal, and this line has to be capable of withstanding a load of at least 65,000lb. It must also be as light and flexible as possible to permit handling with the ordinary equipment of a cable ship. The usual type of chain cable would be far too heavy and cumbersome for this purpose, since a crane chain of carbon steel would have to be of at least 1½-in. link size to withstand the above load. Such a cable of 4,200ft. length would weigh 95,000lb. or more, a deadweight exceeding the proof strength. The chain actually used is "Di-Lok" 3½ per cent. nickel steel chain, made in one continuous length instead of in the 90-ft. lengths common in marine chain work. The total weight is only 43,000lb., each link weighing just under 3½lb. Heat treatment to a hardness of 300 Brinell enabled the chain to withstand a proof load of 84,000lb., and it showed an ultimate tensile strength of 117,000lb. on test.—*"The Shipping World", Vol. CII, No. 2,434, 7th February, 1940, pp. 241-242.*

An Improved Monitor Alarm.

The accompanying illustration shows a group of two Monitor pressure alarms with an integral indicator panel for use with oil-engine-driven generators in ships as well as in the main engine rooms of small motor craft. The "Keep Open" cocks are connected to the pump discharge pipes leading to the engine and a 2-core cable is led from the engine-room lighting circuit to the junction box next to the double-pole switch. A similar 2-core cable connects the Klaxon horn of the system to the other junction box. There is no electrical connection to the pressure gauges as the alarm works on the diaphragm system, the electrical gear being housed in an aluminium box below the alarm-range indicator. The usual alarm range is from 40lb./in. to zero for the lubricating oil and 25lb/in.² to zero for the cooling water, the adjustment for any pressure being effected by turning the handles at the bottom. When the engine is running the safety lamps (white) will be "on" provided the pressures of the lubricating oil and cooling water systems are above danger point. Should the pressure in either system fall below these values, the safety lamp will go out and the danger lamp (red) will light up; at the same instant the Klaxon horn will sound the alarm. The alarm can be stopped by opening the tumbler switch, whereupon all the lamps in the panel will be extinguished. The "Keep Open" cocks are only for testing the alarms while the engine is running. When the alarm operates, the pressure on the gauge should correspond with the pointer of the alarm range indicator. In the drawing, the oil side alarm is shown as being set to operate if the pressure falls below 30lb./in.² and the cooling water alarm set for 10lb./in.².—*"The Marine Engineer", Vol. 63, No. 751, February, 1940, p. 45.*



The new Monitor pressure alarm combined with an indicator panel.



Engine-room of the "African Prince".

Doxford-engined Motorship "African Prince".

The single-screw cargo motorship "African Prince" recently completed by the Furness Shipbuilding Co., Ltd., for the Prince Ltd., is a finely modelled open shelter-deck vessel of 4,625 gross tons, $437 \times 56\frac{1}{2} \times 28$ft. with a d.w. capacity of 9,366 tons on a draught of 25ft. 1in. There are seven transverse watertight bulkheads and a continuous cellular double bottom arranged for the carriage of oil fuel or water ballast. There are two large cargo holds and a deep tank forward of the machinery space and two cargo holds aft of the latter. There is a steel centre-line division in all four holds clear of the hatches, while shifting boards and feeders are provided to facilitate the carriage of grain cargoes. The deck and engineer officers' accommodation is arranged in a deckhouse around the machinery casing, while the crew are berthed aft. The propelling machinery consists of a set of Doxford 2-stroke opposed-piston Diesel engines having three cylinders of $20\frac{1}{2}$-in. diameter and a combined stroke of 82in. The normal rating of the engine is 2,040 i.h.p., or 1,780 b.h.p. at 114 r.p.m. A scavenging-air pump is driven from the crosshead of No. 2 cylinder by means of rocking levers which also serve to drive a 20-ton forced-lubrication pump, a 120-ton salt-water circulating pump and a 75-ton jacket-water pump. A four-feed cylinder lubricator, driven off the rear camshaft, is provided for each cylinder. The Michell-type thrust bearing bolted directly to the after main bearing girder, is lubricated from the engine f.l. system through a special fine filter. A Zeiss solid bronze 4-bladed propeller is used, this being 14ft. in diameter with a pitch of $10\frac{1}{2}$ft. at the tips and $8\frac{1}{2}$ft. at the roots of the blades. In accordance with the latest practice in cargo motorships, the whole of the engine-room and deck auxiliary machinery is steam-driven, the sole exceptions being two 3-ton centrifugal pumps for supplying distilled cooling water to the fuel valves, starting valves and relief valves of the main engines. The exhaust gases from the latter are used for generating steam for auxiliary purposes when at sea, for which purpose there is a Cochran boiler of the composite type, designed to work at a steam pressure of 120lb./in.^{2} and having an exhaust-gas heating surface of 434ft.^{2} and an oil-fired heating surface of 538ft.^{2}. There is also a multi-tubular Scotch boiler with a heating surface of 1,390ft.^{2} and generating steam at a pressure of 120lb./in.^{2}, for harbour use. The oil-burning equipment of both boilers is of the Wallsend low-pressure type. The steam-driven auxiliary machinery in the engine room includes two 15-kW. generating sets, two 3-stage air compressors having a free air capacity of 75 cu. ft./min. at 350 r.p.m., and discharging at 600lb./in.^{2}, a 200-ton ballast pump, a 40-ton bilge pump, two 10-ton oil fuel transfer pumps, a 75-ton stand-by jacket cooling water pump, a 20-ton stand-by forced-lubrication pump, a 40-ton general service pump, two 1-ton centrifugal oil purifiers, two 6-ton boiler feed pumps and an evaporator with a capacity of 10 tons per 24 hours. There is a well-equipped engineers' workshop, and a 4-ton motor-driven crane spans the engine room. The deck machinery comprises 13 cargo and warping winches, a windlass and steam steering gear. The general arrangement of the engine room is shown in the accompanying drawings.—*"The Marine Engineer", Vol. 63, No. 751, February, 1940, pp. 34-35.*

New 1,500-h.p. French Tug.

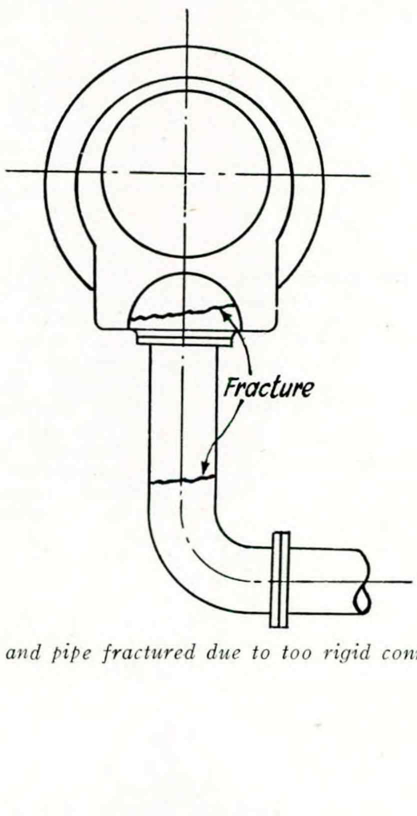
The Ateliers et Chantiers de Provence, of Marseilles, recently completed the tug "Marseillais 9" for the Compagnie Chambon of that port. The new tug is a single-screw vessel of $75\frac{1}{2} \times 22 \times 15\frac{1}{2}$ft., and a mean draught of about 14ft. She is claimed to possess a towing pull of 20 metric tons and should, therefore, be capable of towing the largest liners using the port, in any weather. The tug's rudder is of the Oertz type and she is equipped with "Souplex" spring towing hooks. The propelling machinery consists of a set of triple-expansion engines with cylinders $16\frac{1}{2}$in., $26\frac{1}{2}$in. and $43\frac{1}{4}$in. in diameter and a stroke of $26\frac{3}{4}$in., developing 1,500 i.h.p. Brown starting gear is fitted and the main circulating pump is independently driven. Steam at a pressure of 235lb./in.^{2} is supplied by a 4-furnace Prudhon-Capus boiler fitted with Todd oil-burning equipment. The smoke tubes are provided with double-helical retarders and forced draught is maintained by a turbo-blower located in the boiler uptake.—*"Journal de la Marine Marchande", Vol. 22, No. 1,087, 1st February, 1940, p. 121.*

A Portable Explosive-gas Detector.

An instrument known as the M.S.A. Explosimeter, recently placed on the British market, enables direct readings of gas concentrations in ships' holds, fuel tanks and bunkers, to be taken on a dial meter graduated in percentages of the lower explosive limit. By operating a small plunger pump, a sample of the atmosphere to be tested is drawn through a length of hose into the explosimeter. The hose may be of almost any convenient length, as there is no lag in the indicator reading except that equivalent to the actual time required to draw the sample through the suction hose. In the explosimeter the gas sample flows over a hot platinum wire which forms part of a balanced electrical circuit energised by current from a pair a small dry-cell batteries. This detector unit is balanced against the filament of a small electric-light bulb burning in an inert atmosphere. The combustion of the gases in contact with the detector filament increases the temperature and resistance of the wire, thereby causing the electrical circuit to be unbalanced. This unbalancing of the circuit produces a deflection of the pointer of the electrical meter, proportional to the concentration of gas in the atmosphere being tested. The explosimeter compares in size and weight with a small folding camera and can be carried about either in a pocket case or by means of a shoulder-strap. In addition to use aboard ship, the instrument is claimed to possess a wide field of application in oil refineries, chemical and paint works, and similar establishments.—*"Shipbuilding and Shipping Record", Vol. LV, No. 5, 1st February, 1940, p. 108.*

Exhaust Pipe and Silencer Troubles.

Excessive cylinder lubrication in an internal-combustion engine may result in the passage of considerable quantities of oil into the exhaust pipe, where it becomes congealed by the heat. In time, the sectional area of the pipe may thus become so much reduced that the escaping gases are throttled and a back pressure is created which reduces the power of the engine. Thus, a gradual fall in output—such as might also be caused by faulty ignition timing and valve setting, leakage past pistons and valves, unsatisfactory cooling and lubrication and restricted areas at the exhaust valves—may be brought about by a restriction of the exhaust pipe area. In many small engines there is no means of ascertaining whether the back pressure is unduly high, in which case one may search for the cause of labouring without a reliable clue. Furthermore, if the gases cannot escape freely the temperatures tend to increase and there is greater risk of overheating with its attendant troubles. Since the exhaust pipes are exposed to great heat they will expand considerably, and unless adequate provision for free expansion has been made, excessive stresses may be induced not only in the exhaust pipes themselves but in the cylinder castings to which they are attached, with a consequent risk of fracture. Such a case occurred in the breech end of an engine in which the breech end was bolted to the cylinder and had a nose cast on it to which the exhaust pipe was secured. The arrangement of the exhaust piping was unsatisfactory, any movement due to changes of temperature or vibration being communicated to the breech end, and imposing an undue stress. The breech end eventually fractured across the nose, and the pipe fractured completely across, near to the nose, as shown in the accompanying illustration. Expansive movements transmitted from the exhaust pipe to the breech end may account for persistent trouble from leakage at the joint where the two are connected. If the silencer be placed on rollers, this may permit the exhaust piping to expand freely away from the engine and thus relieve the stresses. The periodic discharge of the gases through an exhaust pipe tends to shake or vibrate the pipe, and where the latter is long there is considerable expansion, so that the question of providing for this and at the same time anchoring the pipe at certain places to prevent it from shaking, may need careful consideration, as serious vibration may occur if the period of the exhaust discharges should happen to synchronise with the natural period of vibration of the pipe. Any alteration in the arrangement of the pipe would probably overcome the trouble as it would alter the natural period of vibration. Lengthening or shortening the pipe a little might suffice. A change of speed of the engine means a change in the periodicity of the exhausts, and the engine is therefore an important factor in regard to vibration caused by synchronism. Thus, if there is vibration at a certain speed, the trouble may disappear completely if the engine be run slightly above or below that speed. A silencer, by breaking up the waves of gas which travel along the exhaust pipe, tends to prevent vibration, so that when trouble arises an alteration of the position of the silencer might have a beneficial effect. In some instances an improvement has been effected by changing the silencer or adding to the arrangement already in use. One objection to the latter course is that it may increase the back pressure. An occasional trouble is "back-firing", *i.e.*, explosion of the gas in the exhaust pipe. This may happen if a charge is not properly fired, so that unburnt gas passes into the exhaust pipe and silencer and is ignited, before it passes away, by the products of the succeeding charge. It may also happen if the exhaust valve does not close properly, or if it closes very late, so that some of the charge passes into the exhaust pipe. An exhaust-pipe explosion usually sounds far worse than it is, the noise being terrifiic, but it rarely causes any damage to the engine, although the pipe itself may be injured.—*E. Ingham, "Gas and Oil Power", Vol. XXXV, No. 413, February, 1940, pp. 33-34.*



Breech and pipe fractured due to too rigid connection.

Hydro-icecutters.

According to *Pravda* the Soviet icebreaker "Lenin" is to be fitted with the new hydro-icecutter invented by V. P. Chizhikov, for the purpose of testing it under practical conditions during next March and April. This hydro-icecutter employs two or three powerful jets of water under high pressure (750lb./in.^{2}) which are played on the surface of the ice from specially constructed hydro-monitors. The jets cut into the ice to a considerable depth, reducing its resistance, it is said, by almost 90 per cent., thus making it unnecessary for the icebreaker to cut its way through the ice by impacts and permitting the ship to move forward at the rate of one to two miles per hour. In the course of a series of experiments in the cutting of ice by means of a high-pressure water jet carried out in Leningrad in 1937, it was found that an apparatus of only 250 h.p. enabled ice 31½in. thick to be cut at the rate of 13ft./min. As the installation fitted in the "Lenin" is to be of 1,500 h.p., it should allow the ice-breaker to travel at a speed of two miles per hour through ice 6½ft. thick. The apparatus, which comprises a very powerful engine and pump, has already been completed and despatched to the icebreaker "Lenin" for installation.

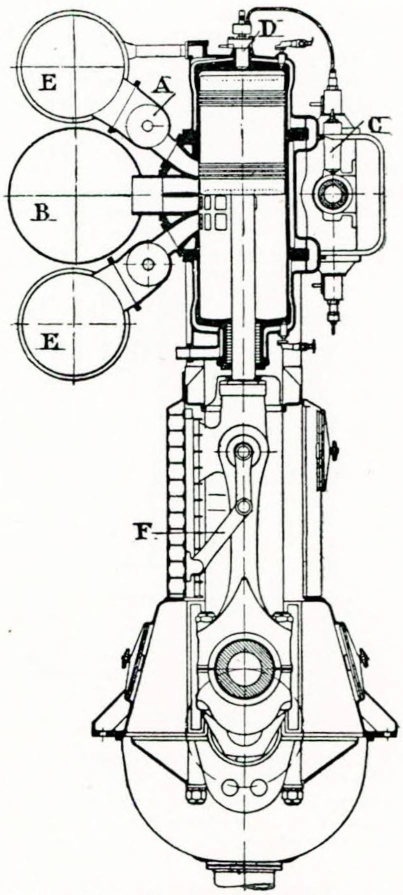
If the experiments with the new device prove successful, all Soviet icebreakers will be fitted with hydro-icecutters.—*"The Journal of Commerce" (Shipbuilding and Engineering Edition), No. 34,944, 1st February, 1940, p. 3.*

The Removal of Oily Deposits.

The removal of oily deposits from the inside of the tubes of oil heaters and coolers, etc., from the discharge pipes of air compressors and the exhaust pipes of reciprocating steam engines, etc., frequently presents difficulties. A novel way of cleaning bundles of oil-cooler tubes, recently described in *"Power"*, consists in placing the tube bundle in a horizontal position, supported a few inches off the ground by blocks. A jet of compressed air is then directed on to the tubes, and a little petrol allowed to fall on the air stream. Thus, the external surfaces of the tubes (which are the surfaces to be cleared of oil, as under working conditions, the oil surrounds them and the water flows through them) are subjected to the action of a powerful jet impregnated with petrol, which is known to have a strong solvent effect on oil. Cleaning in this manner should, of course, be carried out with due care, and well away from buildings or sheds, owing to the highly inflammable nature of petrol. In air-compressing plant, oily deposits tend to accumulate in intercoolers, discharge pipes and receivers, which are highly dangerous, owing to the risk of fire and explosions. To avoid the formidable task of breaking the numerous joints necessary for dismantling and cleaning in the usual manner, it is often advantageous to wash the pipes, etc., through with a solution of soap and water. As the use of a soap solution, however, tends to cause rusting of the cylinders and pistons when the compressor is shut down, the compressor should be run for about 20 minutes before shutting down, with oil lubrication. It may be necessary to repeat the washing-through process a number of times before all the oily deposits are removed. Paraffin should never be introduced into an air compressor, as it may cause an explosion. Internal cleaning of exhaust-pipe systems, etc., may also be carried out by means of the I.C.I. Degreasing Process, which employs trichlorethylene in a special plant for the removal of oil, grease, wax and tar from metal surfaces. There are two main types of such degreasing plant, one a vapour plant, and the other a liquor plant. From these two types, a wide variety of designs has been developed to meet various requirements. The vapour type—which is confined to the process of degreasing—consists essentially of a tank in which a small quantity of trichlorethylene is boiled, and a bank of condensing coils arranged near the top of the plant. The tank becomes filled with trichlorethylene vapour, and when an oil-coated article is suspended in this vapour some of the latter condenses on the oil-coated surfaces, dissolving the oil which falls into a sump at the bottom of the plant. The cleaning action continues until the temperature of the article being degreased reaches that of the solvent vapour, by which time all traces of grease have usually been removed. In many instances, the whole process occupies only five or ten minutes.—*E. Ingham, "The Power and Works Engineer", Vol. XXXV, No. 404, February, 1940, pp. 37-38.*

Double-acting Engines Weighing 2·65lb. per b.h.p.?

According to an article in *Le Genie Civil,* the 32 *S* class motor torpedo boats now building for the German Navy are to be driven by high-speed double-acting 2-stroke M.A.N. engines weighing only 2·65lb./b.h.p. at maximum power. Each boat is to be fitted with two such engines developing 1,200 b.h.p. at 1,000 r.p.m., with a normal rating of 900/1,000 b.h.p. They have seven cylinders of 190mm. diameter and a piston stroke of 300mm. About 20 per cent. of the power is absorbed by the engine-driven rotary scavenge pump, piston-cooling oil pump, lubricating-oil and circulating-water pumps. Nevertheless, it is claimed that the engine has a fuel consumption of only 0·38lb./b.h.p. The weight of the engine without water or lubricating oil, is stated to be 3,610lb., and this, at the service rating of 900 b.h.p., corresponds to about 4lb./b.h.p., which seems to be remarkably low. Unless some very exceptional steps have been taken for lighten-



Section through cylinder of high-speed M.A.N. double-acting engine.

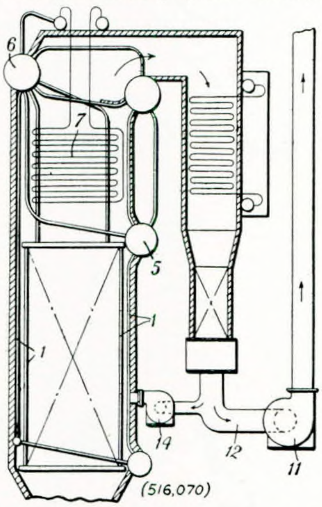
ing the engines—such as the utilisation of light metals for components not hitherto so constructed—it appears improbable that the weight could really be reduced to this extent. The *S* boats have a length of 92½ft. and a displacement of 62 tons, with a main armament of two 19·7-in. torpedo tubes. They carry a crew of 17. The speed of these craft is reported to be 30-36 knots, the lower figure probably being the speed on continuous service. It has been stated that the Germans intend to use boats of this type or of higher speed for surface raids on shipping in the summer, when the sea conditions might be favourable.—*"The Motor Ship", Vol. XX, No. 241, February, 1940, p. 419.*

Safety of Ships.

Among inventions that aim at preventing ships from sinking is one by the Russian engineer Yourkevitch, who designed the hull of the liner "Normandie". As torpedoes or mines explode on impact with the hull and plates are torn outwards, the hull is prevented from being flooded by keeping air under pressure in the holds and maintaining it against the water pressure by air pumps. The holds are made airtight by a specially-prepared rubber sheeting. No particulars of the invention have yet been published, neither have any details of the installation and air-compressing equipment been disclosed, but as Mr. Yourkevitch is said to have gone to the United States to arrange for the exploitation of his invention in that country, it may be anticipated that something more will be heard of it.—*"The Engineer", Vol. CLXIX, No. 4,386, 2nd February, 1940, p. 124.*

Novel Method of Steam Superheating.

A recently-published British patent concerns an invention which is a method of raising the final steam temperature in a boiler without increasing the rate of firing. Referring to the accompanying drawing, the generating surface of the boiler is mainly constituted by the water tubes (*1*) lining the walls of the furnace, and leading across the top from the water drum (*5*) to the steam and water drum (*6*). Above the furnace chamber is a convectively-heated superheater (*7*) from which the flue gases pass to a reheater, economiser and air heater. After leaving the latter, the gases are drawn by a fan (*11*) through a duct (*12*), whence they pass into the uptake. Another fan (*14*) draws flue gases from the duct (*12*) and delivers them into the



furnace chamber, the quantity of these gases being regulated by varying the speed of the fan. In operation, the transfer of heat from the newly-generated gases to the water tubes (*1*) is reduced owing to the reduction of the furnace temperature caused by the admission of the returned products of combustion, but the weight of the gases available to pass over the superheater elements (*7*) is materially increased, resulting in the required steam temperature being obtained without having an excessive flue-gas temperature at the superheater elements. Furthermore, a higher reheat temperature is obtained. The steam temperature may be controlled by varying the quantity of the cooled gases returned to the combustion chamber, the speed of the fan (*1* being varied automatically according to the final steam temperature.—*"Engineering", Vol. 149, No. 3,869, 8th March, 1940, p. 268.*

Fatigue Strength of Crankshafts.

The Research and Standardisation Committee of the Institution of Automobile Engineers are conducting research work on the fatigue strength of crankshafts. Although the investigation is primarily concerned with the study of fatigue strength in bending of crankshafts for compression-ignition engines, the information obtained is also proving of value in the design of all types of crankshaft. Three machines are employed to apply reversed bending moments to the specimens, which are usually multi-throw crankshafts, one throw being broken at a time. The deflection is applied through a variable-throw eccentric, and calibrated in terms of bending moments. Failure occurs across the web, the crack starting from a fillet, and the nominal stress figure being calculated in the ordinary way from the modulus of the area of fracture. The fatigue limit is determined on a basis of ten million repetitions. A good deal of work has been carried out with these machines to study the fatigue strength of various designs of shaft in relation to the material employed. Recently, particular attention has been devoted to various methods of surface hardening, including nitriding, chromium plating, etc. In addition, owing to the importance under war-time conditions of salvaging worn parts for further use, a study is being made of the effects of certain processes, such as metal-spraying, for building up crankshafts.—*"Metallurgia", Vol. 21, No. 124, February, 1940, p. 104.*

New Design of Bunker.

A novel design of bunker has been adopted for the Norwegian steamer "Favor", 2,475 tons d.w., which has just run trials. She is a vessel with a length of 249ft. 8in., a breadth of 41ft. 6in. and depth (to shelter deck) of 25ft. 10in., with a hull strengthened for navigation in ice. The ship was built at Oslo for the Great Lakes trade and has eight winches and derricks, steam steering gear with telemotor control from the bridge and a streamlined rudder. Three two-berth passenger cabins are provided. The propelling machinery, which is located amidships, consists of a compound steam engine working in conjunction with a Bauer-Wach exhaust turbine, and develops a total of 1,200 i.h.p. at 110 r.p.m. Steam at a pressure of 230lb./in.^{2} and superheat temperature of 590° F. is supplied by two coal-fired cylindrical boilers. On her trials the vessel attained a speed of 13 knots on a coal consumption of 0·98lb./i.h.p. The coal bunker extends vertically from the navigating bridge to the tank top between the main boilers, this arrangement being claimed to give increased stability and to eliminate trimming, the bunker being self-emptying.—*"Lloyd's List and Shipping Gazette", No. 39,092, 21st February, 1940, p. 11.*

New Norwegian Fruit Carrier.

The recently-completed Norwegian motorship "Panama Express" is specially designed for the carriage of bananas, citrus fruits, pears and apples through tropical waters. She is a vessel of 350ft. by 50½ft. by 34ft., propelled by an 8-cylinder single-acting 2-stroke Sulzer engine of 5,600-b.h.p., designed to give the ship a speed of 17 knots fully loaded. There are four hatches and 10 derricks worked by electric winches. The ship has 12 passenger cabins, each with an adjoining bath or shower-bath. The total refrigerated cargo capacity is 230,000 cu. ft. in 12 compartments which can be maintained at any temperature between 54° and 32° F. by means of a CO_2 cooling installation with brine circulation. The refrigerating equipment is of Danish make and comprises two vertical double-acting CO_2 compressors, each directly driven by a 175-h.p. motor at any speed between 280 and 365 r.p.m.; two CO_2 condensers with copper coils in cylindrical cast-iron casings; two CO_2 sub-coolers for enhancing the performance of the plant in tropical waters; two evaporators with coils of seamless steel tubes; six air coolers with galvanised-steel pipes through which brine is circulated by two centrifugal pumps driven by 15-h.p. motors, one of the pumps being intended as a stand-by; six streamline propeller fans driven by 6/15 h.p. reversible motors, which draw air from the cargo chambers through ducts along the ship's sides and blow it through the air coolers, whence it returns to the chambers at the low tem-pumps with 7½-h.p. motors for the CO_2 condensers. The brine system is designed on the 3-temperature system so that the various compartments may be maintained at different temperatures by means of either freezing brine, chilling brine or thawing brine, as may be required. The temperatures of the chambers can be controlled from the main engine room by means of an electric distance-thermometer installation with 32 points. There is also a two-chamber refrigerating installation with a small 7-h.p. electrically-driven plant for the use of the passengers, officers and crew.—*"The Journal of Commerce" (Shipbuilding and Engineering Edition), No. 34,962, 22nd February, 1940, p. 8.*

From Destroyer to Fruit Carrier.

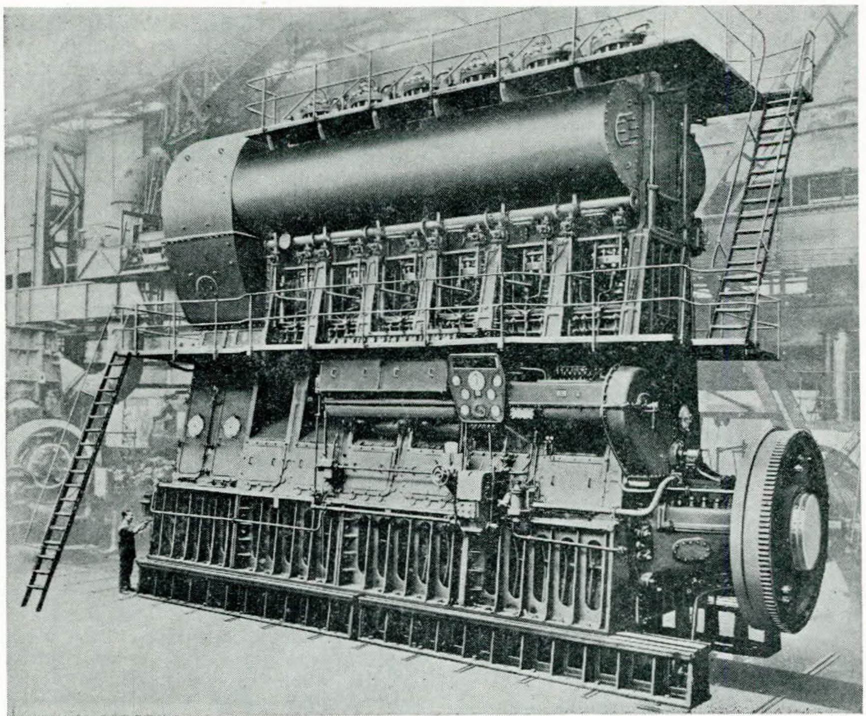
Soon after the last World War, three U.S. destroyers, surplus to naval requirements, were bought by an American fruit company and converted into banana carriers. These destroyers were built in 1901, and had a length of 259½ft., with a displacement of 433 tons and a speed of just over 29 knots. As originally constructed, the ships had four boilers in separate boiler rooms, with reciprocating engines of 8,300 i.h.p. driving three screws and situated in a compartment between each pair of boiler rooms. The destroyers carried 184 tons of coal in their bunkers and had a complement of 76 officers and men. On conversion into fruit carriers, the whole of the space occupied by the machinery and boilers was gutted and converted into an insulated fruit hold with the exception of the original No. 4 (after) boiler room, which was made into a motor room. Two 6-cylinder 200-h.p. Wolverine kerosene engines were fitted to the existing wing propeller shafts, giving the ships a maximum fine-weather speed of 12 knots. Hatches were fitted on the clear deck space, accommodation and a navigating bridge was built up aft, and what had once been fast torpedo boat destroyers then became fruit carriers of the coastal type. In this capacity, all three ships were employed in regular trading between Nicaraguan ports and Houston or Galveston, Texas, sailing under the Nicaraguan flag with an average load of 15 to 16 thousand bunches of bananas per ship, the return cargo being a general one.—*A. C. Hardy, B.Sc., "The Shipping World", Vol. CII, No. 2,435, 14th February, 1940, pp. 265-266.*

Fiat Marine Engines.

The accompany illustration shows the main propelling engines of the 10,000-ton Italian tanker "Giulio Giordani", which was recently launched from the Genoa yard of the Soc. Anon. Ansaldo with the whole of her machinery and propeller shafting in place. The engines are of the Fiat double-acting 2-stroke type, with six cylinders of 640 mm. diameter, and a piston stroke of 1,160 mm. The full-power rating is 7,400 b.h.p. at 147 r.p.m., the overload capacity being 8,000 b.h.p., or just over 1,300 b.h.p. per cylinder. The bedplate is in four sections and each main column is bolted to it by two steel through bolts. The cylinder liners are in three portions, the middle section having ports for the exhaust gas and scavenging air. Automatic valves in the scavenging-air receiver control the admission oi air and the two scavenging pumps at the forward end of the engine are driven by an extension of the crankshaft. The piston are likewise made in three parts, the upper and lower sections being of forged steel, while the centre portion consists of a cast-iron skirt. The cylinder covers are of cast steel. Lubricating oil is used as the cooling medium for the pistons and piston rods. The highest powered Fiat marine engines of this type so far built, are the 10-cylinder units ($29\frac{1}{2}$-in. diameter and $49\frac{3}{16}$-in. stroke) of the 24,469-ton twin-screw passenger liner "Vulcania" which have a total output of 36,000 b.h.p.—*"The Shipping World", Vol. CII, No. 2,437, 28th February, 1940, pp. 313-314.*

The Application of the Two-stroke Heavy-oil Engine to Aircraft Propulsion.

A paper on the above subject was read by Mr. W. S. Burn at a general meeting of the N.E. Coast Institution of Engineers and Shipbuilders held on the 9th February, 1940. The author stressed the desirability of taking the fullest advantage of experience gained in developing marine oil engines, and suggested a horizontal opposed-piston aero engine of his own design which complied with the greatest number of aeronautical requirements and was yet basically efficient. He also suggested that the growing intricacies of the engine installation would make it desirable, in the larger aircraft, that the navigating staff shou.d be relieved of all detail engine control and that the practice which obtains on board ship of having an engineer in charge, should be followed more closely. The accessible engine within the wing construction made that practicable. It was also becoming apparent that the mass of engine-driven auxiliaries would have to be removed from the main engine and separately driven either directly or indirectly electrically from a completely accessible auxiliary oil engine. It would be found that there was a convenient space for this between the engine and the wing root. Such an auxiliary engine could be made as efficient as the main engine. In the case of the 2,100-b.h.p. size of engine which the author had under consideration for a proposed flying boat, no fewer than 18 cylinders would be required for an air-cooled radial type of engine, or 24 cylinders in the case of a water-cooled double Vee or X engine. A major advantage of the oil engine was that it imposed no limitation in cylinder size. The engine Mr. Burn proposed to use for 2,100 b.h.p. was one with only six opposed-piston cylinders, three on either side of the crankshaft, with 12 pistons in all, six controlling the air inlet and six the exhaust. The diameter of the cylinders would be 7in. and the combined piston stroke 12in., the mean effective pressure 150lb./in.^{2} and the speed 2,000 r.p.m. It has been demonstrated that, for aero-dynamic reasons, the most suitable geometric form is a flat engine, and as there is ample wing length the engine could be of almost unlimited length in a horizontal plane. That made possible the use of relatively long cylinders and pointed the way to a horizontally-opposed-piston type of engine. The ideal construction seemed to be the use of



Fiat main engine of the "Giulio Giordani".

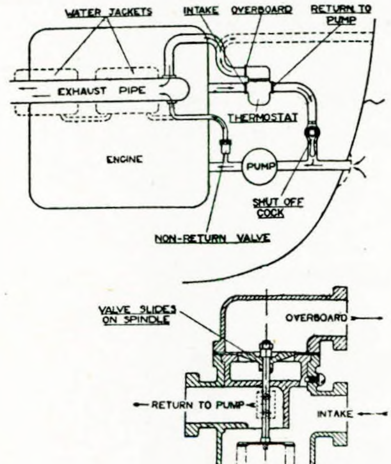
such an engine, but to connect the outer exhaust pistons by side rods to a central three-throw crank, thereby utilising the weight-economising principle of the radial engine to the greatest extent possible in a flat engine.—*"Lloyd's List and Shipping Gazette", No. 39,086, 14th February, 1940, p. 9.*

"The Relative Importance of Piston Ring Dressing in Relation to Cylinder Wear".

The author suggests that the problem of preventing excessive cylinder wear has in the past been left to the metallurgist, but that, in his opinion, much can be done by the engineer to further decrease excessive wear at the top of the liner. After setting out what he regards as the main factors responsible for such wear, the author proceeds to discuss the prevention of abrasion, the problem of cylinder lubrication, the action of the piston rings, and the various materials used for the construction of pistons. Particulars of a series of tests with different pistons and liners carried out by the author in a four-cylinder "R.N." engine developing 32 b.h.p. at 900 r.p.m. are briefly set out, and some details of a sleeve piston of novel design—the "R.N." patent sleeve piston—are given. The paper concludes with a summary of requirements for the best liner and piston results. The text is illustrated by five diagrams and drawings.—*Paper by J. H. Bradbury", "Bulletin of the Liverpool Engineering Society", Vol. XIII, No. 7, February, 1940, pp. 12-22.*

The Use of Kerosene in Marine Petrol Engines.

Vaporizing equipment enabling petrol engines to be operated on kerosene usually consists of a special form of exhaust manifold with passages through which the incoming air and paraffin spray mixture from the carburettor pass into the cylinders. These passages may be subjected to exhaust heat either externally or internally (in the latter case they form a jacket around the exhaust pipe); or in some cases a combination of both external and internal heating is employed. Unless the petrol engine is of the low-compression type, the compression ratio will have to be reduced to about 4½ :1 for running on kerosene, as the use of too high a compression causes detonation and excessive formation of carbon deposit. A number of different proprietary vaporizers are available for attachment to petrol engines for conversion purposes, but in order to run these satisfactorily on



Arrangement of Smith thermostat.

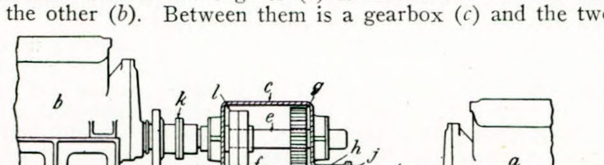
kerosene it is most important that a correct temperature should be maintained, as failure to do so results in incomplete vaporization and poor combustion, in addition to which there may be dilution of the lubricating oil through kerosene finding its way past the pistons into the sump, which may lead to serious damage. Dilution up to 0·5 per cent. per hour by volume of the lubricating oil is permissible, but this figure should not be exceeded. A temperature of about 180° F. should be attained by the circulating water before attempting to run the engine on kerosene, and in order to maintain the correct temperature it may be desirable to provide some form of automatic temperature regulator. One such device, marketed by a well-known firm of motor accessory manufacturers, consists of a thermostat installed in a pipe running between the cylinder block and the inlet side of the pump, as shown in the sketch. When starting from cold the whole of the water passes from the thermostat to the pump inlet, thereby quickly raising the temperature to 158° F., whereupon the expanding bellows begin to operate, lifting the valve from its seating and allowing part of the water to pass through another pipe overboard. The valve opening is progressive with increasing temperature until, at 185° F., the whole of the circulating water passes directly overboard, the by-pass back to the pump being closed. A special feature of the thermostat is that the valve head is loose on its spindle in order to prevent air from being drawn into the system, as might happen with a fixed-head valve. An alternative system of temperature control comprises a T-piece of special form attached to the cylinder head and supplied with water from the circulating pump, the bulk of the water passing straight across the T to the discharge pipe. A branch pipe on the inlet side of the T passes water to the bottom of the cylinder jackets and as the engine warms up the water in the jackets rises and passes out through the T-piece into the discharge pipe, its place being taken by cold water descending from the branch pipe. It is claimed that this system enables the engine to be maintained at the correct temperature when idling or running slowly and that excellent results have been obtained with this regulator on many different makes of engine. Petrol engines converted to run on kerosene generally give a lower output on the latter fuel, the reduction in power varying between 10 and 20 per cent. according to the type and the speed of rotation.—*"The Motor Boat", Vol. LXXII, No. 1,858, 2nd March, 1940, pp. 154-157.*

Motorships for Philippine Islands' Service.

The motorship "Doña Aniceta" recently completed by the Cantieri Riuniti dell'Adriatico, Trieste, is the third of three passenger and cargo motorships built for Philippine owners for service between the Phillippine Islands and New York *via* the Panama Canal. They are sister ships of the shelter-deck type with a second 'tween deck in No. 1 hold. The main dimensions are 410ft. 10in. by 55ft 6in. by 29ft. with a gross tonnage of 5,011 tons. The vessels have a rounded stem, cruiser stern, two masts and a single funnel, and the hull is divided into eight watertight compartments, including three cargo holds forward of the machinery space and two aft. A continuous cellular double bottom is arranged for the carriage of oil fuel, lubricating oil, feed water, fresh water and water ballast. Wing tanks for the carriage of vegetable oils are provided at the sides of the shaft tunnel. The total tank capacities amount to 770 tons of oil fuel and 625 tons of vegetable oils in addition to 860 tons of tank space that may be used either for water ballast or for oil fuel. Apart from 108 tons of fresh water carried in the D.B. tanks, two fresh-water tanks on the main deck forward of the engine casing hold a further 25 tons. The five cargo holds have a total capacity of 483,000 cu. ft. and the cargo handling equipment comprises twelve 5-ton derricks and one 30-ton derrick, worked by eight 3-ton and two 5-ton electric winches. The passenger accommodation includes single and double-berth cabins for 12 passengers. The propelling machinery consists of an 8-cylinder single-acting 2-stroke C.R.A.-Sulzer Diesel engine rated at 6,350 h.p. at 120 r.p.m., and having cylinders 720mm. in diameter and a piston stroke of 1,250mm. A piston-type scavenging pump is driven off the main engine, but the whole of the auxiliary machinery—which includes a 150-ton ballast pump, a 100-ton bilge pump, a 200-ton cargo oil pump and the necessary lubricating oil, cooling water, oil transfer, fire, sanitary and fresh water pumps—is electrically driven, as are the steering gear and windlass. Electric current is furnished by three 75-kW. generators, each driven by a 3-cylinder 4-stroke C.R.A.-Sulzer Diesel engine. There is also a 15-kW. emergency generator in a compartment on the promenade deck. An exhaust-heat boiler with alternative oil firing provides steam for hotel services and for the steam-heating coils in the vegetable oil tanks. All three ships exceeded 17 knots on their sea trials, the machinery developing a maximum power of 7,600 h.p. at 130-r.p.m.—*"Shipbuilding and Shipping Record", Vol. LV, No. 7, 15th February, 1940, pp. 152-154.*

An Improved Twin-engine Drive.

A recently-published British patent relates to a new method of driving a single propeller shaft by two engines, each of which normally runs in the same direction. The arrangement provides for two engines having the same hand of rotation with a gearbox between them, clutches being provided so that either one or both of the engines can drive the propeller shaft, which runs below and can be reversed by a train of gears shown in the sketch reproduced. One of the engines (*a*) is turned end for end and faces the other (*b*). Between them is a gearbox (*c*) and the two



Two Thornycroft engines driving a single shaft through gearing.

shafts (*e, f*) are geared to run at the same speed through spur wheels (*g, h*). Each engine is fitted with flexible coupling (*i, k*) and the drive is taken through oil-operated clutches (*j, l*). In the gearbox is a shaft (*m*) which is coupled to the propeller shaft (*n*). The gearwheels (*o, p*) on the driving shaft engage those (*q, r*) on the gearbox shaft (*f*), and these also embody oil-operated clutches. The installation in the vessel is arranged in such a manner that if the wheels (*o, q*) drive the propeller shaft, the motion will be in the ahead direction, the clutch of the lower left-hand wheel being in the engaged position; whereas, if the wheels (*q, r*) take up the drive through a third pinion (*s*) in the train, the shaft will run astern when the internal clutch of the lower wheel (*p*) is engaged. The size of the extra gear-wheel (*c*) determines the ratio of the reducing-reversing gear and a suitable device is provided to ensure that there should be no tendency for one of the oil-operated clutches to become engaged while the other is taking up the drive.—*"The Motor Boat", Vol. LXXII, No. 1,858, 2nd March, 1940, p. 165.*

Diesel Emergency Sets.

In a paper presented to the London Local Technical Group of the Electrical Power Engineers' Association on the 6th February, 1940, Mr. R. J. Welsh advocated the use of internal combustion engines for driving emergency generators. For up to about 5kW. he said, it was usual to employ petrol engines, but for the larger sizes Diesel engines were preferable. The simplest of all automatic starting arrangements was, he remarked, similar to that used for d.c. telephone installations, in which a relatively heavy flywheel is fitted to the motor generator set normally used, coupled directly to a petrol engine through a friction clutch which is normally held in the disengaged position by a solenoid or latch of the no-volt release type. The stored energy of the flywheel gives the necessary starting impulse to the petrol engine without more than a momentary drop in voltage. With small sets a secondary battery is the obvious solution, and although the switchgear involved is somewhat complicated, British-made sets of this kind have proved very reliable. For a.c. sets of over 12-kW. or so, low-voltage batteries are employed to operate 12- or 24-volt starters of the automobile type, engaging the engine flywheel through a device similar to a Bendix drive. To avoid sticking if left idle for long periods, the whole starter armature is arranged to slide axially under the influence of a magnetic pull, and the starter is constructed in such a way that it does not develop its full torque until the pinion is properly engaged in the toothed ring. For engines running at under 750 r.p.m., or greater than 100 h.p., compressed air is the only satisfactory method of starting, but in all compressed-air systems precautions against losses of starting-air pressure through leaks at pipe joints, etc., during long periods of stand-by should be taken by fitting an electrically-driven air compressor with automatic pressure-operated "start-stop" arrangements. The closing down of emergency sets should be done by hand, and if the plant has to restart automatically the engine must have six or more cylinders, so that it can be started from any position in which it might have come to rest. The author stressed the relative simplicity with which Diesel engines could be adapted to automatic working.—*"Electrical Review", Vol. CXXVI, No. 3,247, 16th February, 1940, p. 187.*

Welding as a Substitute for Casting.

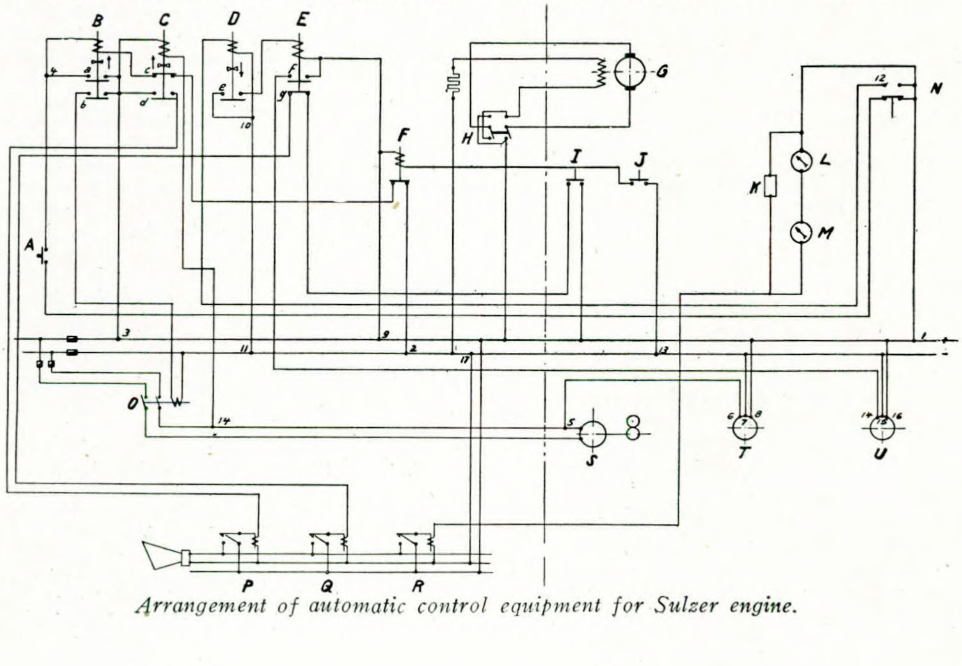
The object of the paper is to indicate the progress that has been made in the substitution of welded mild steel parts for ferrous castings. The author expresses the view that the substitution of weldings for castings is desirable where it is important to restrict weight to a minimum or when it is considered that a better or more suitable article can be made by the use of welding. Both these conditions can be met not only by the use of lighter scantlings, but also by the more efficient distribution of material obtained by fabricating tne part in mild steel, the main advantages of the latter over cast iron being its higher elastic modulus and greater resistance to the effects of temperature. For the same strain the ratio of the permissible stress in mild steel as compared with cast iron is about 2:1, and it is this factor which is responsible for the saving in weight effected. In marine work, welded mild steel is now commonly used for main and auxiliary engine bedplates, pipes and pipe flanges, oil and water heater bodies, condensers, evaporators, steam traps, superheater headers, eduction pipes, turbine cylinders and gear cases, armature hubs and stator frames and bedplates for windlasses, capstans and winches. Sundry valves, brackets, and pedestals formerly made of cast iron, are now also frequently manufactured in the form of weldings. One of the main advantages of welding over casting is that the designer is not tied to a range of standard castings that might be available, which is of special importance for such items as pedestals, yokes, switchgear, and hubs and bedplates for generating plant, where each job can receive individual consideration and special features can readily be incorporated in the welding which would otherwise be prohibited for a casting for which standard patterns would be used. Reference is made to various examples of marine engine weldings of which illustrations are given. The author points out that welding has very definite limitations where steel castings are concerned owing to the fact that the weldability of certain alloy steels is still in the experimental stage, and that for certain parts of special design intended for high internal pressure, accessibility for welding may be too difficult to ensure sound welding, since highly stressed welds should always be made from both sides of the joint. For certain parts of machinery a useful combination of welding and steel castings can be employed, and the author cites a number of examples. He then makes a brief reference to the principal defects liable to occur in fusion welding and to the precautions to be taken for their avoidance. He explains the nature of the inspections and tests to which welded work is habitually subjected, including the use of X-ray examinations. He also lays special stress on the human element in welding. In conclusion the author states that there appears to be no ground for suggesting any serious competition between welding and iron casting, since each process has its legitimate sphere of usefulness and scope. The paper is illustrated by 13 plates depicting various examples of welded work.—*Paper read by S. F. Dorey, D.Sc., at a meeting of the Institution of Engineers and Shipbuilders in Scotland, on the 20th February, 1940.*

The Rating of Electrical Machinery.

The author predicts the possibility of effecting a 50 per cent. reduction in the weight of electrical machinery in ships, with a substantial saving in cost. The article is the first of a series of four which delineate a plan for the development of light-weight machinery, the steps recommended being similar to those taken by the author in his successful developments of heat— and flame-resisting cable. He declares that obsolescent methods of electrical machinery design must be enlivened and that better use must be made of recent improvements in insulating materials. He bases his diagnosis on the exposure of present design standards and suggests better yardsticks together with co-operative endeavour.—*J. B. Lunsford, "Journal of the American Society of Naval Engineers", Vol. 52, No. 1, February, 1940, pp. 1-25.*

Automatic Control Equipment for Diesel Engines.

In recent years there has been an increasing demand for automatic control of the starting and stopping of Diesel engines, both by distant press-button control and by fully automatic gear, where the engine is to run whenever there is a failure of a main supply. Such automatic control equipment has, in some cases, been fitted to large Diesel engines driving electric generators and other types of plant, in which the automatic system has to fulfil the following functions:—(1) Start the auxiliary lubricating oil pump in order that the forced lubrication system of the engine may come under pressure before the latter is started. (2) Open the starting-air valve and bring the regulating shaft into the starting position. (3) Shut off the starting air and put the regulating shaft into the running position as soon as firing speed is reached. (4) Open the cooling water inlet valve when starting and close it again when stopping. (5) The person in charge of the engine must be warned: (*a*) if the engine should fail to start at the proper time; (*b*) if the cooling water pressure is too low or the cooling water temperature is too high; and (*c*) if the engine is not in the starting position after it has been stopped (this only applies to engines which cannot start in any position of the crankshaft). The diagram shows the arrangement of the automatic control gear of a Sulzer 4-stroke engine developing 550 b.h.p. at 600 r.p.m., fitted with a Büchi supercharger and arranged for distant automatic control. The starting valve *T* is a compressed-air valve operated by a small electric motor. The regulating shaft is actuated by a compressed-air servomotor and operates the contactor *N*. The cooling-water pressure gauge, fitted with a minimum contact, and the lubricating-oil pressure gauge, fitted with a maximum contact, are shown at *L* and *M* respectively. The engine is alo equipped with a camshaft signal contactor *I* and centrifugal swich *J*, the latter operating when the engine reaches firing speed and being arranged on the end of the exciter shaft of the generator driven by the engine. The electric motor *G* is provided for the remote control of the engine speed and is located just behind the governor. The auxiliary lubricating-oil pump *S* and the automatic cooling-water inlet valve *U* can be arranged in any convenient and readily accessible position near the engine, while the starting push-button *A*, the various relays *B* to *F*, the auxiliary oil-pump motor contactor *O*, the signal apparatus *P*, *Q* and *R*, and the speed-control motor switch, are all arranged on the generator-control panel at the central station from which the control of the engine is to be exercised. The system works in the following manner:—I. To start the engine the control button *A* is pressed to close a circuit taken from *1* through the starting relay B. This circuit also includes the regulating-shaft contactor *N*, the upper contactor of the battery switch relay *C*, and the contactor *F*. The relay *B* has a slight time lag in order to prevent the engine from starting up if the button *A* should be accidentally pushed down for a moment. After the time lag has expired the contactors *a* and *b* close and the winding of the relay *B* remains connected through *3*, *a* and *4* to the positive pole of the battery circuit after the push button *A* has been released. The contactor *O* of the motor for the auxiliary lubricating-oil pump set *S* is closed by the contactor *b*, so that the pump starts up and supplies oil to the lubricating system of the engine. At the point *5* on the connecting cable between *0* and *3*, is a branch circuit to the automatic starting-air valve *T*. The motor for this valve is fitted with special control gear (not shown in the diagram), so that according to the position of the air valve, only one of the two negative terminals *6* and *7* is in circuit at the same time. Compressed air now enters the servomotor, turning the regulating shaft into the starting position. The air also passes through the control valve into the starting-air valves on the cylinders and the engine begins to turn. As the regulating shaft is turned by the servomotor, the contactor *N* moves into its upper position, closing a circuit from *1* to *12* through the retarding relay *D*. This relay operates at once but has a retarding device which prevents the switch it controls from re-opening for a period of about five minutes after the circuit to the relay has been broken. The purpose of this device is described under IV below. After the contactor *e* closes, the



Arrangement of automatic control equipment for Sulzer engine.

winding of the relay *E* is connected in circuit from *9* through *e, 10* and *11,* thereby closing the contactor *f* and starting the motor actuating the automatic cooling-water inlet valve *U*. This motor is fitted with special control gear which allows it to rotate in one direction or the other until the corresponding end position is reached. II. When the engine begins to fire and attains a predetermined speed, the centrifugal switch *J* closes and allows current to pass to the auxiliary relay winding *F* through *9, J* and *13*. Operation of the auxiliary relay *F* interrupts the current to the starting relay *B,* thereby causing the auxiliary lubricating-oil pump contactor *O* to open, while the motor of the automatic starting valve *T* is rotated further over the terminals *7* and *8* until the initial position is reached. This cuts off the starting air from the engine. Forced lubrication is now provided by the main lubricating-oil pump driven directly by the engine, and the oil pressure turns the regulating shaft into the running position. III. The engine is normally stopped by hand. As soon as the regulating shaft is moved into the "stop" position, the contactor *N* opens and cuts the current off from the relay *D*. As already stated, the contactor *f* remains closed for some time due to the action of its retarding mechanism. Afterwards the current is also cut off from the winding of the relay *E* and the contactors *f* and *g* fall back into the starting position, the contactor *f* interrupting the connection to the cooling-water inlet valve motor *U* and causing the latter to rotate in the opposite direction through the connections *15* and *16* and close the inlet valve. IV. The system includes a number of safety devices. As soon as the contactor *O* of the auxiliary lubricating-oil pump is closed, the winding of the protecting relay *C* is brought into circuit through *3* and *14*. This relay is fitted with a retarding device. Only if the starting operation should last for an abnormally long period—*e.g.,* more than 40 seconds—will the contactor *c* open, cutting off current from the starting relay and consequently interrupting the starting process. At the same time the contactor *d* closes, connecting the signal apparatus *P* through *3, d, P* and *17,* and the warning syren is sounded. Simultaneously, a signal lamp on the switchboard lights up the panel "Diesel engine failed to start". A thermostat *K* is connected in parallel with the contacts for the cooling water pressure gauge *L* and the lubricating-oil pressure gauge *M*. When contact is made at either of these instruments the circuit to the signal device *R* is closed and the warning syren is sounded, while the switchboard panel marked "Cooling-water temperature too high or pressure too low" lights up. To prevent the signal apparatus from being in circuit while the engine is stopped, the contacts for gauges L and *M* are placed in series, that on *M* being arranged for a maximum contact, so that this is always closed when the engine is running. When the engine is stopped the circuit is broken by this gauge. In the event of the lubricating-oil supply failing, the engine is immediately stopped by a servomotor fitted for the purpose. If the engine does not come to rest in a starting position of the crankshaft when stopped, the contactor *I* on the camshaft closes the circuit through *g* to the signal *Q*. This sounds the syren and lights up the panel marked "Diesel engine not in starting position" on the switchboard, whereupon an attendant can bar the engine round to the starting position. As it is undesirable that this signal should be sounded immediately the engine comes to rest, the relay *D* is fitted with the retarding device previously mentioned, which keeps the contactor *e* closed for about five minutes after the current through the winding of the relay *D* has been interrupted. In addition to the gear shown in the diagram, the starting-air system is provided with automatic control of the auxiliary compressor which starts up and recharges the starting-air bottles when the pressure in these falls below a pre-determined minimum. The starting-air compressor stops as soon as the maximum air bottle pressure is reached.—*"Gas and Oil Power", Vol. XXXV, No. 414, March, 1940, pp. 59, 62 and 72.*

Marine Engine Production.

One of the greatest difficulties facing British marine engine-builders at the present time is to secure deliveries of large forgings such as crankshafts, but although the production of solid-forged or semi-built-up multi-throw crankshafts for high-powered marine oil engines represents one of the most difficult marine forging propositions, the ordinary three-throw built-up crankshaft of the average triple-expansion engine such as is being fitted in the majority of cargo ships now under construction, must still be the slowest-flowing item in this all-important production stream. In an advanced marine oil engine design developed some years ago, a cast-steel crankshaft was used and although the power per cylinder in this engine was fairly high, the crankshaft proved quite satisfactory. Engines of this particular type have not been built for several years, but it is suggested that the idea of employing a cast-steel crankshaft, or one of the semi-built-up type with cast-steel webs, might be worth considering. One of the most popular cars has a crankshaft of special cast iron running in lead-bronze bearings, which has proved remarkably successful, well over four million of these shafts being in service all over the world. Recently, a well-known British maker of industrial Diesels introduced a small air-cooled single-cylinder heavy oil engine which has the additional novel feature of a cast-iron crankshaft. These two different engine types demonstrate the practicability of the special cast-iron crankshaft.—*"The Shipping World", Vol. CII, No. 2,436, 21st February, 1940, pp. 289-290.*

A Field Tube Boiler.

The so-called Field tube employed in the Lewis boiler was invented about 100 years ago by Perkins, a pioneer of high-pressure steam, but its construction was greatly improved by Edward Field in 1862 for fire-engine boilers A Field tube is

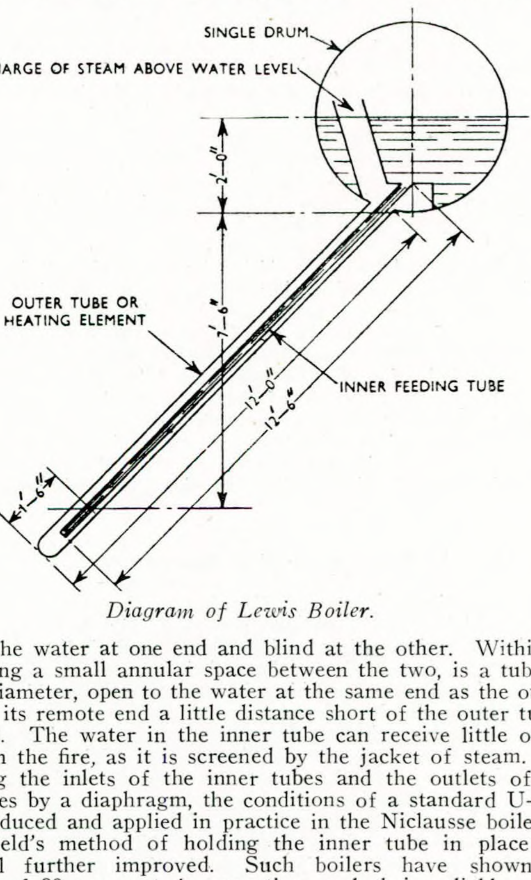


Diagram of Lewis Boiler.

open to the water at one end and blind at the other. Within it, and leaving a small annular space between the two, is a tube of smaller diameter, open to the water at the same end as the other, and with its remote end a little distance short of the outer tube's blind end. The water in the inner tube can receive little or no heat from the fire, as it is screened by the jacket of steam. By separating the inlets of the inner tubes and the outlets of the outer tubes by a diaphragm, the conditions of a standard U-tube are reproduced and applied in practice in the Niclausse boiler, in which Field's method of holding the inner tube in place has been still further improved. Such boilers have shown an efficiency of 90 per cent. in operation, and their reliable water circulation makes it possible to run them on sea water for considerable periods without incurring trouble. In the Lewis boiler there is only one drum and from the bottom of this the Field tubes project downwards radially, or rather, fanwise. The steam formed in the annular spaces is conveyed through a special compartment to a level above the water in the drum, so that each tube constitutes an individual heating unit with its own independent circulation and separate supply of water, and the performance of each tube may be calculated in the same way as for a standard U-tube. This makes it possible to ensure that all the sensible heating of the water should take place before it enters the steam tubes, by the provision of a separate set of water tubes arranged as a rear bank, where the gas temperatures are lowest. The water from the feed pump passes through the drum to spigots which direct it into the mouths of the downcomers of these sensible heating tubes, so that it only mixes with the drum water after it has been raised to steam temperature. It then has the properties required in the downcomers of the steaming elements, retaining the weight of water, but being ready to be turned into steam when it emerges into the annulus. A recently completed Lewis boiler for AR.P. duties has a total weight of under 10 tons, which includes the feed pumps, etc.; it evaporates 6,000lb. of water per hour at 185lb./in.^{2} pressure, under very low natural draught, with hand-firing; and it is compact enough to be mounted on a four-wheeled trailer. It has been suggested that these characteristics should make this type of boiler suitable for marine purposes and, more especially, for naval work. The arrangement of the tubes in relation to the boiler drum is shown in the accompanying diagram.—*"The Syren", Vol. CLXXIV, No. 2,272, 13th March, 1940, p. 469.*

Creep at High Temperatures.

The paper deals with the creep characteristics of metals with special reference to the behaviour of carbon and molybdenum steels. The manner in which the tensile properties, such as ultimate stress and limit of proportionality, are influenced by creep is first described, and an account is given of the general behaviour of metals during creep and of the influence of temperature, stress, and time. The second half of the paper is entirely devoted to the creep properties of carbon and molybdenum steels. Considerable variations in the creep properties of carbon steels of similar carbon content have been encountered, and this important feature is simply illustrated by the data quoted, from which it would appear that steels containing from 0·13 to 0·4 per cent. carbon have much the same properties around 840° F. Molybdenum steels of similar composition also vary considerably in creep properties, although they are greatly superior to carbon steels. The addition of a small percentage of vanadium to molybdenum steels produces further improvement. The author describes the effect of heat treatment, more especially the effect of prolonged heating during service, and concludes by giving brief details of methods for estimating working stresses, as well as numerical data and curves to show how experimental results may be utilised.—*Paper by H. J. Tapsell, read at a General Meeting of the N.E. Coast Institution of Engineers and Shipbuilders, on the 22nd February, 1940.*

New Phase in Steam Generating Approaching.

In the course of his address to the managers and members of the American Bureau of Shipping at the annual meeting recently held in New York, the president, Mr. J. L. Luckenbach, stated that reciprocating engines and Scotch boilers with steam pressures around 200lb./in.^{2} have faded from the picture in the U.S.A., virtually no new ship with this type of machinery having been built in the last decade, except for service on the Great Lakes. However, the last four vessels constructed on the Lakes some two years ago were equipped with high-pressure water-tube boilers and geared turbines using superheated steam. Some of the newest American merchant ships use steam at a pressure of 625lb./in.^{2} with a temperature of 910° F., in conjunction with electric propulsion and non-reversing turbines. Most of the U.S. steamships now being built are designed to operate with steam at 450lb./in.^{2} pressure and a superheat temperature of 800° F., but contracts have been placed for two vessels designed for 1,500lb./in.^{2} boiler pressure and 960° F. total steam temperature. The new passenger ships shortly to be ordered will embody the principle of reheating the steam between stages for improved economy. All these developments, observed Mr. Luckenbach, represent a reduction in machinery weight and a consequent economy in fuel consumption as well as a saving in space. Oil-burning equipment has also been improved by the adoption of wide-range burners giving more effective combustion control, and further development has taken place in the design of economisers for heat conservation. A type of steam generator is rapidly approaching which will be operated directly in conjunction with the main engines with little or no steam storage, the steam being generated directly in accordance with requirements, for which purpose forced circulation will play an important part. Experiments along these lines have been carried out both in the U.S. and abroad, and a practical application may be expected to be made available for commercial use in the near future.—*"The Journal of Commerce" (Shipbuilding and Engineering Edition), No. 34,968, 29th February, 1940, p. 7.*

An Unusual Bearing Failure.

The unaccountable failure of a bearing of a turbo-feed pump recently caused some consternation, as no reason was forthcoming as to why the bearing had failed. The pump in question was one of two turbine-driven centrifugal pumps used in a boiler room, one pump being in use and the second employed as a stand-by. The rotors of these small turbines were mounted on ball bearings of reputable make and the plant was new, the first pump having only been run for a few weeks before it was decided to put the second pump into service for a time. This pump had undergone a satisfactory running test when first installed, but on starting it up when required for service bearing trouble quickly developed and the pump had to be stopped. An examination of the bearing revealed the fact that the ball-race had a groove worn round it and that the balls were considerably pitted, so that a new bearing was necessary. The pump had not run for any length of time, so that there could not be any question of ascribing the damage to running; furthermore, the bearings of the first pump, which had been running for a considerable period, were as good as new. The one theory that seems tenable is that the bearing had been damaged by the static vibration of other machinery in the vicinity of the pump, the effect of which caused the balls to be repeatedly flung against the race.—*"Mechanical World", Vol. CVIII, No. 2,772, 16th February, 1940, p. 143.*

Multiple Engines for Geared Diesel and Electric Propulsion.

It has been suggested that a wider employment of the geared Diesel system might be useful under present-day conditions, as it would enable single-screw ships of, say, 2,000 to 4,500 s.h.p., to be powered with either two or four engines—according to their size—of a type which could not otherwise be utilised for marine propulsion. Thus, stationary-type engines complying with Lloyd's Register requirements could be adapted for this purpose even when they cannot be made directly reversible (an unlikely eventuality) by the use of Vulcan gearing with an astern hydraulic coupling. Some years ago, Mr. H. R. Ricardo—admittedly not a marine man—made a proposal before the Institution of Mechanical Engineers concerning the possibility of employing Diesel-electric drive for a single-screwship of considerable power, by means of multiple Diesel generators in sound-deadening "packing cases" stowed in tiers in the engine room. He suggested that high-speed road transport engines derated down to about 100 b.h.p. each for continuous operation could be used for this purpose and that overhauling, from the sea-going engineer's standpoint, would merely take the form of disconnecting one or more of the "packing cases", sending it ashore for refit and replacing it with another unit. Such a scheme might involve the use of perhaps 60 or more main engines (with 720 valves to grind in and 720 rocker clearances to check!) which is excessive, but a modified form of the general idea might be worth considering during the present emergency. The use of a number of relatively small engines in association with a.c. electric drive should be a practical proposition for a variety of craft, since quite a large d.w. capacity ship can be built with, say, 1,500 s.h.p., provided nothing higher than a low convoy speed is required. Such powering needs could easily be met by multiple high-speed engines of moderate power, such as many British makers turn out. These engines, in their class, lead the world and would be fully up to marine service of the type suggested. The British oil engine industry is well equipped for meeting such demands, and if either vertical or horizontal engines were acceptable a wide field of manufacture, capable of making good deliveries, would be available. The electrical side would present no difficulties, practical or otherwise.—*"The Shipping World", Vol. CII, No. 2,437, 28th February, 1940, p. 313-314.*

American Trawler with High-speed Diesel Engines and Reduction Gears.

The 70-ton trawler "Vagabond' of New Bedford, Mass., was originally equipped with a 180-h.p. Diesel engine. The main dimensions of the vessel are 86ft. 19.1ft. with a carrying capacity of some 38 tons of fish on a draught of 8.6ft. Quite recently the trawler has been re-engined with two Gray-General Motors Diesels, each rated at 135 h.p. at 1,600 r.p.m. The engines are placed in tandem on the fore-and-aft centre line, one facing aft and the other facing forward. Between the two is a Farrel-Birmingham reduction gear with both engines connected to the pinion through Twin Disc clutches and reduction gears and Morse flexible couplings. The combined ratio of the engine and propeller-shaft gears is 4:1, so that the propeller runs at 400 r.p.m. when the engines run at normal speed. The propeller shaft extends aft under the after engine in the same vertical plane as the pinion. This use of two compact high-speed Diesels on a single screw saves over seven tons of machinery weight. Although the engine power and speed have been increased, fuel consumption has not increased proportionately due to improved efficiency. Furthermore, all running repairs can now be carried out at sea by merely uncoupling one engine from the gear and continuing to fish with the other, which will drive the vessel at about three-quarter speed. The reduction of machinery weight permits better trim at less than full load, which is unusual. Major repairs that would ordinarily lay the trawler up can now be effected without interrupting fishing operations, by disconnecting one engine and lifting it out of the boat during a cargo-discharging period, a complete spare unit being put in its place if considered desirable. The throttles of both engines are normally interlocked and the two are handled as a single unit with wheel-house control, but these interlocks can be instantly disengaged for local operation of each engine separately. On the end of each engine opposite from the gears is a shaft extension and clutch connected to a salt-water pump and air compressor. Auxiliary power is furnished by a 7½-h.p. single-cylinder Diesel engine which operates a winch on deck as well as a dynamo, a general-service pump, and a compressor for the air whistle. A 32-volt battery is used for starting the engines. The extra seven tons of carrying capacity made available by the installation of the new machinery has been used for additional fuel storage, the original fish storage capacity being considered adequate. The trawler's fishing (cruising) range has thus been increased by 25 per cent.—*"Motorship and Diesel Boating", Vol. XXV, No. 2, February, 1940, pp. 106-107.*

British-built Motor Torpedo Boats for Swedish Navy.

Two 60-ft. motor torpedo boats recently completed for the Royal Swedish Navy by Vosper, Ltd., of Portsmouth, are of a new type and possess some special features to fit them for operation in northern waters. The hull is of wooden construction, divided into six watertight compartments. The accommodation for the crew consists of a large forecastle with four cots and padded lockers giving additional sleeping facilities, and there is a cabin for two officers aft, in addition to a small wireless room and a galley, etc. The propelling machinery is in a compartment in the after part of the boat and consists of two sets of 12-cylinder Isotta-Fraschini engines of Italian manufacture, driving twin screws through gears. The engine cylinders are arranged in three banks, *i.e.*, in "broad arrow' formation, the central bank being located centrally between the two outer ones which form a wide-angle vee. The engines are essentially marine units and not modified aero engines. Fresh-water cooling is employed, and the magnetos, carburettors, water pumps, etc., are all readily accessible. Air-starting equipment is provided, a Reavell compressor being used to charge air bottles connected to automatic non-return valves on the engine cylinders. The engine throttles can be controlled from the wheelhouse. Modified Ford V-8 engines, with sea-water cooling, are fitted for cruising purposes, these being arranged just aft of the main engines. The drive is first taken forward, through a special coupling, to a gearbox, and thence aft to the propeller. These cruising engines are started by electric motors in the usual way, and can be used to start the main engines if required. The designed output of the latter is 2,500 b.h.p., and they are capable of giving the boats a continuous cruising speed of 41 knots. The V-8 engines are each rated at 75 b.h.p. and enable the boats to cruise at 8/9 knots. The armament of each vessel includes two 18-in. torpedo tubes. Special heating arrangements are provided for these and for the living accommodation, cold air from outside the boat being passed over heating coils by an electric fan and then distributed to various points through ducts. In port, a paraffin-heated Primus-pattern boiler in the galley serves to heat the water used in the system, while at sea two exhaust-gas boilers are "fired" by the gases from the V-8 engines and supply the necessary heat. When running on the main engines, hot water is by-passed from the main fresh-water cooling system, but this does not give as much heat as the boilers.—*"The Marine Engineer", Vol. 63, No. 752, March, 1940, pp. 64 and 74.*

Italian Oil-engined Trawlers.

The motor trawlers "Genepesca I" and "Genepesca II", built by the Cantieri di Riva Trigoso, of Genoa, for the Compagnia Grande Pesca, of Leghorn, sailed on the 10th March for the Newfoundland fishing grounds. These trawlers are vessels of 253ft.×40ft.×21ft. 3½in. with a maximum speed of 13½ knots and a service speed of 11 knots. They have a salted cod-fish hold capacity of 35,400 cu. ft., a frozen codfish hold capacity of 10,600 cu. ft. and a cod liver oil tank capacity of 4,420 cu. ft. The oil-fuel tank capacity of 550 tons and lubricating-oil tank capacity of 30 tons enable the trawlers to make a voyage of 150 days, if necessary, without putting into port. The engine room is aft, while the holds are forward, and there is accommodation for two passengers and a crew of 65. The propelling machinery consists of a Fiat single-acting 2-stroke engine and there are three Diesel generator sets for supplying lighting and power. It is claimed that these two new motor trawlers are the only fishing craft afloat provided with a modern plant for drying the clothing of the crew after a day's fishing.—*"The Shipping World", Vol. CII, No. 2,439, 13th March, 1940, p. 396.*

Simplification of Water-tube Boiler Design.

In a paper read at an extra meeting held on the 1st March by the Steam Group of the Institution of Mechanical Engineers, the authors, Messrs. W. Y. Lewis and S. A. Robertson, suggested that a simplification in the design of water-tube boilers should reduce weight, space and complications, improve efficiency and reliability, and lessen the frequency and duration of shut-down and soot-blowing periods, if designers could be provided with a rational theory of circulation. The authors proposed the establishment of a simple type of boiler, consisting of a single drum and a single U-tube, as a standard of excellence with which other boilers might be compared qualitatively. They claimed that flow circulation proved such a simple device to be satisfactory, provided that the downcomer leg of the U-tube was not heated. The authors also pointed out that departures from the suggested simple standard U-tube (such as those met with in practice, including the use of several tubes, downward discharge of the steam, use of bottom drums, so-called "high-head" boilers and long tubes needing forced circulation) all produced effects that were disadvantageous. An improved type of tube that approached very nearly the suggested standard U-tube had been evolved by the authors, who claimed to have proved its circulation to be adequate over the full range of pressures and higher possible rates of heating. The new kind of tube was said to enable a greatly simplified type of boiler to be designed. The assured excellence of circulation, especially in the back tubes, enabled full advantage to be taken of the higher gas velocities and greater heat transmission rates, with consequent important reductions in the area of heating surface required. Other rational eliminations and improvements were also rendered possible, enabling substantial reductions to be effected in the cost of manufacturing, installing and maintaining boilers. *—"Electrical Review", Vol. CXXVI, No. 3,250, 3th March, 1940, p. 276.*

The Electric Propulsion of Ships Originated in Russia.

The credit for the first practical attempt to drive a vessel by electricity is due to the Russian scientist Yakobi, who, in 1839, built a boat capable of carrying 12 passengers and propelled it by an electric motor, fed by a battery of 69 Grove cells. A speed of 2½ m.p.h. was obtained, but owing to imperfections in the motor the idea did not receive any wide practical application and the experiment was soon forgotten. In 1903, however, the Russian shallow-draft tanker "Vandal", an 1,100-ton vessel built and engined in Sweden for service on the Volga and in the Caspian Sea, was fitted with three 3-cylinder single-acting 4-stroke oil engines, placed amidships, and each driving a dynamo and exciter at 240 r.p.m. The three screws were driven by motors placed aft. Normally each generator was coupled to its corresponding motor on the Ward-Leonard system, giving propeller speeds of from 30 to 300 r.p.m. with voltages up to 500. Cross-connection was possible, and reversal took from 8 to 12 seconds. The weight of the engines, each of which was rated at 12 b.h.p., was 48 tons, that of the electrical equipment amounting to 31·2 tons. The cost of the installation was $3,000, while it was estimated that that of a steam installation of equivalent power would have been $21,500. An increase of 10 per cent. in carrying capacity justified the higher first cost, although it is probable that the electrical equipment was really fitted as a convenient, if experimental, method of reversing the oil engines. Nevertheless, the ship continued in service until 1913, and similar installations followed.*—A. C. Hardy, B.Sc., "The Journal of Commerce" (Shipbuilding and Engineering Edition), No. 34,974, 7th March, 1940, p. 3.*

The Mean Pitch Determination of Variable Pitch Propellers.

The paper deals with the problem of the marine propeller whose face pitch is locally adjusted at each radius to produce the best efficiency. No attempt is made to discuss the optimum pitch distribution, the paper being chiefly concerned with the comparison of any pitch distribution with the equivalent constant face-pitch screw, and from the standpoint of assigning a mean pitch to the former to admit of the usual slip calculations being correctly made in practice. The problem involves the appreciation of the local blade-section properties and the determination of sectional zero-lift angles. The relation of a typical section to the effective pitch of the whole blade must be considered and Gawn's recent work on Admiralty propellers is used to illustrate the main facts of effective pitch variation. The effective pitch is also modified by blade section departures from the purely segmental, and particularly so by low-drag root sections. These sections act as segmentals of lower face pitch, and in deriving a mean pitch for a blade the equivalent segmental local pitch must first be determined. This is discussed in detail and an accurate tabular method is given for the calculation of the zero-lift angle of any section likely to be met with in marine practice. Such preliminary work then allows mean pitch determination to be finally discussed. Alternative methods are considered and various examples are given to illustrate the points at issue. The author concludes by remarking that high-duty propellers must literally be designed *and* manufactured correct to minutes of arc in face pitch, effective pitch and blade section details, as errors of design or workmanship may easily convert a cavitation-free design into a propeller which cavitates badly, vibrates excessively and erodes abnormally. The paper is illustrated by means of four diagrams.*—Paper by E. V. Telfer, D.Sc., Ph.D., at a General Meeting of the N.E. Coast Institution of Engineers and Shipbuilders, on the 8th March, 1940.*

Weight Saving by Welding in Ship Construction.

Experts writing in American periodicals are claiming great advantages for welding as compared with riveting in ship construction, basing their conclusions on results which, they say, have been obtained with ships built in the U.S.A. One writer estimates that a saving of 16 per cent. in weight is obtainable by the use of welding and the employment of improved propelling machinery, while other evidence quoted is to the effect that a saving in weight of 1,000 tons in 6,000 tons has been realised, about 800 tons of the decrease being attributed to the use of welding and 200 tons to the reduction of machinery weights. The Engineering Foundation, which has inquired into the matter, considers that savings in weight through welding and the use of improved propelling machinery amount in some cases to 25 per cent.*—"Fairplay", Vol. CLIV, No. 2,966, 14th March, 1940, p. 419.*

Cargo Steamers with Boilers on Deck.

There are about 26 cargo vessels in service, including four of over 7,000 tons d.w., which have their boilers placed on deck and they were built in Norway for foreign owners. Wm. Gray & Co., Ltd., of West Hartlepool, recently completed an oil-burning steamship of this type for the Dene Ship Management Co., Ltd., and are now building a similar vessel for another firm. The ship already in service is the "Elmdene", a vessel of the open shelter-deck type with top gallant forecastle, of 406ft. b.p., 56ft. moulded breadth ,and 36ft. 4½in. moulded depth to shelter deck. The deadweight is 9,000 tons on a summer load draught of 25ft. and the gross tonnage of the ship is 4,853 tons, with a block coefficient of 0·75. The normal sheer of the hull is carried on the shelter deck, but the upper deck throughout is parallel to the keel. Three holds forward and two holds aft of the engine room, with 'tween decks superimposed, are available for cargo and are fitted with permanent steel grain bulkheads clear of hatches. No. 3 hold contains a deep water ballast tank and permits the adaptation of three double-bottom tanks for oil fuel only. The total oil fuel capacity of the ship is 864 tons, carried in three double-bottom tanks under the engine room and deep tank, and in wing tanks on either side of the former. The arrangement of boilers on deck is claimed to provide increased cubic capacity for cargo, the unusually high stowage rate of 70 cu. ft. per ton being attained in this vessel. The total capacity of the five holds and their corresponding 'tween decks is 513,319 cu. ft. grain (478,841 cu. ft. bale), in addition to which the deep tank has a capacity of over 35,000 cu. ft. of general cargo (or 1,060 tons of water ballast). The cargo holds are served by five cargo hatches and a smaller hatch aft to the stern cargo compartment, with ten 5-ton and one 3-ton steel tubular derricks worked by steam winches. The whole of the crew are accommodated above the shelter deck amidships—seamen and firemen forward of the boiler room with petty officers over, and officers, engineers and captain in a separate deckhouse forward of No. 2 cargo hatch. The propelling machinery consists of a set of triple-expansion engines of the most modern design, with the usual auxiliary machinery, including a Diesel-driven emergency generator. Two cylindrical boilers fitted with Howden's forced draught and superheaters, are carried on special supports in the boiler room, which is on the upper deck, aft of the engine casing. The working pressure is 200lb./in.². As the "Elmdene" was only delivered in December, 1939, and has been operating under the convoy system, it is not yet possible to give an analysis of speed and consumption figures, but reports received by the owners are said to indicate entirely satisfactory performance and seaworthiness.*—W. B. Strang, "The Shipping World", Vol. CII, No. 2,439, 13th March, 1940, pp. 367-372.*

Reversible Fans.

For certain purposes, notably for the transport of refrigerated cargoes, it is the practice to reverse the direction of circulation of the air through the hold from time to time, and whereas the reversal of the electric fan motors does not present any special difficulty, it is often found that the efficiency of an axial-flow type of fan is liable to undergo an appreciable reduction when its direction of rotation is reversed. To obtain the full benefit of reversal it is essential that the rate of air circulation should be the same in both directions in order that the transfer of heat from the cargo to the cooling medium should remain unchanged, and this can only be achieved when the efficiency of operation is the same in both directions. The *Report of the Food Investigation Board for 1938,* which has just been issued, contains an account of an investigation into the theory underlying the design of a reversible fan, together with particulars of tests carried out with a fan of normal design and with one designed to operate equally well in both directions. The report states that figures obtained from the lower hold of a ship show that with the normal type of fan the efficiency on reversing dropped from 76 to 22 per cent., with a reduction of 63 per cent. in the rate of circulation of air, while for a pre-cooling chamber the efficiency was reduced from 61 to 12 per cent., with a diminution of circulation to 46 per cent. The efficiency of the axial flow fan is given as $\frac{\text{thrust x rate of air flow}}{\text{torque x r.p.m.}}$ The thrust and torque are functions of the aerodynamic properties of the blade section and the angle of pitch of the blade, so that a formula can be evolved for the efficiency in terms of the shape of the blade section and its position in relation to the flow of the air. Blade sections for both the normal fan and for the reversible fan are given, the latter being symmetrical both about a line joining the leading and the trailing edges and about an axis perpendicular thereto. The efficiency of the two types is then determined mathematically for different values of the variables in order to obtain the optimum conditions before proceeding to the actual design. These conclusions were tested by means of a reversible fan with a diameter of 33in. over the blade tips and a boss diameter of 15in., circulating 7,200 cu. ft. of air per minute against a head of 0·86in. of water when running at 2,100 r.p.m. This fan was run in an air tunnel which had a flared inlet permitting an undisturbed flow of air at uniform velocity to pass through the fan. The resistance against which the latter operated was increased by placing gauze screens across the outlet so as to obtain data for a range of working pressures both above and below that for which the fan was designed. The mean pressure over the fan disc was noted and the quantity of air was also determined, reversal being effected by turning the fan round on its spindle. The results were plotted and the graphs show that a fairly close agreement is obtained between the actual results and the theoretical calculations, while except at the vicinity of the boss, there is a very close agreement in the air delivery pressure in both directions over a radius ranging from 11·5 to 16·5in. The actual efficiency of the blades alone is 60 per cent., which is only 4 per cent. less than the predicted value. It is, however, suggested that this particular design does not represent the most efficient obtainable, since by increasing the diameter of the boss from 15in. to 23in. there would be a gain of 10 per cent. in the theoretical efficiency, in addition to which careful fairing of the fan would enable better results to be secured, so that an overall efficiency of 60 per cent. should be obtainable in a practicable commercial design of fan.—*"Shipbuilding and Shipping Record", Vol. LV, No. 10, 7th March, 1940, pp. 220-221.*

The 16-knot Cargo Liner "Mormacpenn".

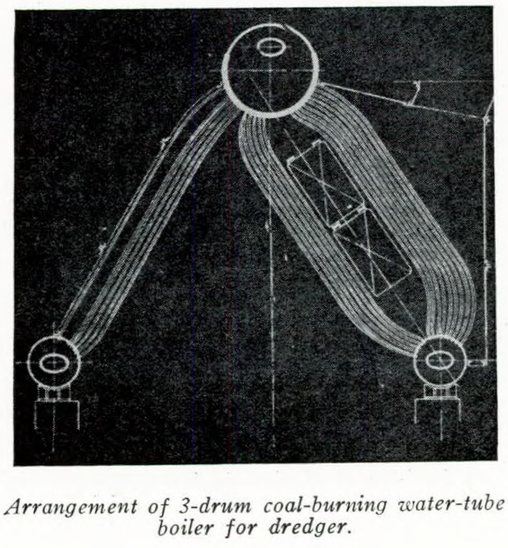
The 7,680-ton motorship "Mormacpenn" is the first of the 25 cargo vessels of the C3 class to be completed for the U.S. Maritime Commission, who have also ordered 11 more ships of similar design with accommodation for between 60 and 90 passengers. The service speed of all these vessels is to be about 16-16½ knots when loaded. In the "Mormacpenn" four 2,250-b.h.p. Busch-Sulzer engines drive a single propeller through reduction gearing, with a Westinghouse electric coupling between each engine and the reduction gear. The engines run at 240 r.p.m. for full speed, the propeller speed being 85 r.p.m. The electric couplings comprise inner and outer members which are not in direct contact, there being 24 laminated poles in each outer member with strap field windings, while the inner members or armatures have short-circuited squirrel-cage windings. The field current can be reduced to 70 per cent. of the normal figure when the ship is running at half speed and in addition to the "full speed", "half speed" and "stop" positions, the switch gear includes an "astern" position. Usually, when manœuvring, two engines are kept running ahead and two astern, with the clutches out of operation. Corresponding ahead or astern rotation of the propeller is simply effected by energizing the field windings of the appropriate couplings, as desired. Three similar ships are being built with couplings of the same type and dimensions. In four Sun-Doxford oil-engined cargo and passenger vessels under construction the propelling machinery installation consists of two 4,375-b.h.p. Doxford-type engines running at 180 r.p.m. and driving a single propeller shaft through electric couplings and reducing gear at about 90 r.p.m. These ships will be the first in which Doxford-type engines have been employed with indirect drive. The engines, moreover, are of higher speed than any Doxford engines of equal power. In seven other vessels, also being built for the U.S. Maritime Commission, there are to be two 2,100-b.h.p. engines driving the propellers through electric couplings and gearing. The auxiliary machinery of the "Mormacpenn" includes three 275-kW. generators driven at 450 r.p.m. by 400-h.p. Cooper-Bessemer oil engines.—*"The Motor Ship", Vol. XX, No. 242, March, 1940, p. 434.*

A New Insulating Material.

A new material for the insulation of bulkheads, cabins and other spaces on board ship has recently been developed by an American firm. The product is composed of alternate layers of flat and corrugated asbestos felt, jointed with a newly-developed waterproof adhesive. It has the characteristics of a structural material and is approved for Class A-1 and Class B bulkheads by the U.S. Bureau of Marine Inspection and Navigation. The material is supplied in the form of a board or panel which is claimed to be hard, light, rigid and strong, proof against rot and vermin, fire-resisting and capable of withstanding a pressure of 75lb./in.² even after soaking in water for 72 hours. The new product is made in thicknesses from ½-in. upwards and in sizes up to 48in. × 120in. Although developed primarily for the purpose of insulating bulkheads, the strength and fireproof nature of the material make it suitable for use in boiler and furnace casings, oven walls, casings for air-conditioning units, and as an insulating lining for funnel uptakes.—*"Shipbuilding and Shipping Record", Vol. LV, No. 10, 7th March, 1940, p. 219*

A New Russian Marine Boiler.

Prior to 1936 the majority of Soviet river steamers were equipped with cylindrical boilers, but since that year marine water-tube boilers for such vessels have been constructed at Kiev and Gorky (formerly Nijny Novgorod). The accompanying illustration shows the arrangement of a 3-drum coal-burning water-tube boiler for a sea-going dredger of 2,630 cu. ft. per hour capacity constructed at the Sormovo Works in Gorky. According to the Russian periodical *Sovetskoe Kotloturbostroenie,* this boiler generates steam at a pressure of 200lb./in.² with a final temperature of 617° F. The heating surface of



Arrangement of 3-drum coal-burning water-tube boiler for dredger.

the boiler is 2,690ft.^{2} and the temperature of the preheated air is 302° F. The boiler drums are of welded construction, the inside diameter of the upper one being 43·3in., and that of the two lower ones 23·6in. The upper drum is made up of two plates and has, therefore, two welded seams; the lower plate is just over 1in. thick and the upper one 0·55in. in thickness. The wrapper plates of the lower drums are 0·79in. thick and these drums have only one seam. All three drums have spherical dished ends. The boiler tubes are 1·75in. outside and 1·48in. inside diameter. The uptake is arranged at one side and the superheater is divided into two sections, as shown in the illustration, consisting of tube coils of seamless tubing of 1·25-in. outside and 1·0-in. inside diameter. The heat transfer rate in the brick-lined furnace is stated to be only 4,250 B.Th.U.'s per cu. ft./hr. There are two ashpits under the furnace, separated by an airtight wall, with dampers for air regulation. Each section may be cleaned separately without affecting the working of the boiler. The air preheater is of the tubular type with tubing 20in. in diameter outside and 1·8in. in inside diameter, arranged at the back of the boiler, one side of the preheater forming part of the boiler rear wall, thus reducing radiation losses. The forced draught equipment is at the front of the boiler. There is a 2-stage tubular feed-water heater, heated by exhaust steam from the auxiliary machinery. The feed water leaves the first-stage heater at 194° F. and the second-stage heater at 246°-252° F. The feed pump is of the vertical Worthington type.—*"The Marine Engineer", Vol. 63, No. 751, February, 1940, p. 46.*

Feed-water Heating.

A novel form of boiler economiser has been developed by a British firm specialising in this type of equipment, which is suitable for use both in multitubular cylindrical and water-tube boilers. Two designs are available, one of which employs cast-iron tubes and is intended for low-pressure boilers, while the other has solid-drawn steel tubes and is suitable for higher working pressures, the economiser in both cases being placed on the delivery side of the feed pump, so that feed temperatures considerably in excess of 212° F. can be obtained. The cast-iron tubes are of diamond cross-section with rectangular gills, whereas the solid-drawn steel tubes have cast-iron gilled sleeves shrunk on to them. The gas passages in both designs are, therefore, of the same shape, yielding adequate turbulence and a high rate of heat transfer. The headers are so arranged that the joints are all outside the gas chamber.—*"Shipbuilding and Shipping Record", Vol. LV, No. 10, 7th March, 1940, pp. 219-220.*

Italian Motorships for Red Sea Coasting Trade.

The twin-screw coasting vessels "Lago Tana" and "Lago Zuai" were recently launched from the Legnano yard of the Franco Tosi Company. The ships are being built for the Eritrean Shipping Company, of Rome, who are reported to be about to place orders for three further vessels with the same builders. The main dimensions of these ships are, it is stated, to be similar to those of the first two, *viz.*, 220ft. × 32ft. 4in. × 14ft. 8in. with a mean draught of 9ft. 1in. on a displacement of 1,100 tons and a cargo capacity of about 32,800 cu. ft. The propelling machinery consists of two sets of 720-b.h.p. 6-cylinder Tosi engines fitted with built-in Büchi superchargers. The output of these engines on the test bed was 800 b.h.p. at 225 r.p.m. during a two-hours' trial, but the contract speed of the machinery at sea is only 210 r.p.m. The main engine cooling-water and lubricating-oil pumps are engine driven and there is a separate fuel-injection pump for each cylinder. The whole of the auxiliary and deck machinery is electrically driven, current being supplied by two 220-volt 30-kW. generators, each driven at 500 r.p.m. by a 45-b.h.p. twin-cylinder Tosi engine. The equipment of the vessels will include air-conditioning plant. —*"The Motor Ship", Vol. XX, No. 242, March, 1940, pp. 450-451.*

Training Engineers for the U.S. Mercantile Marine.

According to a recent announcement of the U.S. Maritime Commission future officers for the American merchant marine are to be selected on a new basis. During the past few weeks applications for enrolment as cadets in the engineering and deck departments have been considered, and the names of a number of young men who satisfy all the mental and physical tests prescribed for candidates have been placed on a roster from which they will be appointed to American merchant vessels engaged in foreign trade as vacancies occur. Under the regulations laid down by the Commission all cadets in ships owned or subsidised by the U.S. Government are selected by the Commission, and it is anticipated that satisfactory arrangements with private shipowners receiving no subsidy will shortly be made to open that field also to cadets. Out of 166 young men who passed the first cadet examination more than 100 had been appointed to merchant ships before the end of last year. The programme of training prescribed for these cadets is a purely civilian one, directed by Mr. Richard McNulty and supervised by a retired officer of the U.S. Navy, Rear-Admiral Henry A. Wiley, of the U.S. Maritime Commission. During this training, engineer cadets will learn the duties of the engine-room department and they will also be required to complete courses of study, while serving in their ships, in engineering, physics, mathematics, first aid and ship hygiene. The minimum pay for cadets is $50 a month in addition to food and quarters. They will mess with the junior officers and will be quartered with them. After three to four years' service as such, cadets may take the U.S. Bureau of Marine Inspection and Navigation's examination for a third assistant engineer's licence and after obtaining this they will receive the status of cadet officers until such time as they secure employment as third assistant engineers. Cadets and cadet officers will not be used to displace members of the crew. During their first three months' service they may learn how to clean and polish, but thereafter these duties will be reduced to a minimum and after six months they will not be required to perform this type of work. Applicants for engineer cadetships must be between the ages of 18 and 25 years on the 1st July next, and unmarried American citizens who can produce evidence of good moral character and able to pass the strict physical and scholastic examinations prescribed by the Commission. Only young men in possession of the required scholastic certificates are accepted as candidates and special importance is attached to seagoing experience and to completion of special academic courses in mechanical engineering. Candidates are advised that before applying they should carefully consider whether or not they are "naturally adapted for a seafaring career which involves confinement aboard a vessel for long periods of time. Youths who are not rugged in physique and resolute and ambitious in spirit or who are unstable of purposes or at all afraid of work, wet, cold, or the general privations that may accompany a seafaring life, are unfit to become cadets". The official statement goes on to say that applicants for engineer cadetships must have mechanical aptitude. This is essential because marine engineering aboard ship demands a practical mechanical skill in machine-shop work, handling and use of tools, electricity, and assembling ability. The above essentials are coupled with a sound theoretical knowledge of the working principles involved in the propelling machinery, refrigerating plant, electrical equipment, and all of the auxiliaries.—*"The Marine Engineer", Vol. 63, No. 752, March, 1940, p. 60.*