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The Computation of the Stresses in a Propeller Blade Section.

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In estimating the stresses and determining the required thickness of a propeller blade section, it is necessary to separate the fore and aft moment produced by the thrust and the transverse moment produced by the torque.

The method of separating these moments is due to the late Admiral D. W. Taylor, and is given in his book "Speed and Power of Ships".

Taylor shows that the fore and aft moment M_1 at any diameter d_1 due to the thrust is,

$$M_1 = \frac{T_0}{6(d+d_1)}\left\{2d^2 - dd_1 - d^2_1\right\} \quad \dots\dots\dots\dots\dots\dots \quad (1)$$

where,

M_1 = fore and aft moment in pound feet
T_0 = thrust per blade in pounds
d = tip diameter of the propeller in feet
d_1 = diameter at which moment is required, in feet.

In determining the thrust per blade the assumption is made that the propeller efficiency equals (1 – real slip) which is usually on the safe side so that,

$$T_0 = \frac{33{,}000 \times \text{power absorbed per blade}}{\text{face pitch of propeller} \times \text{revolutions per minute.}}$$

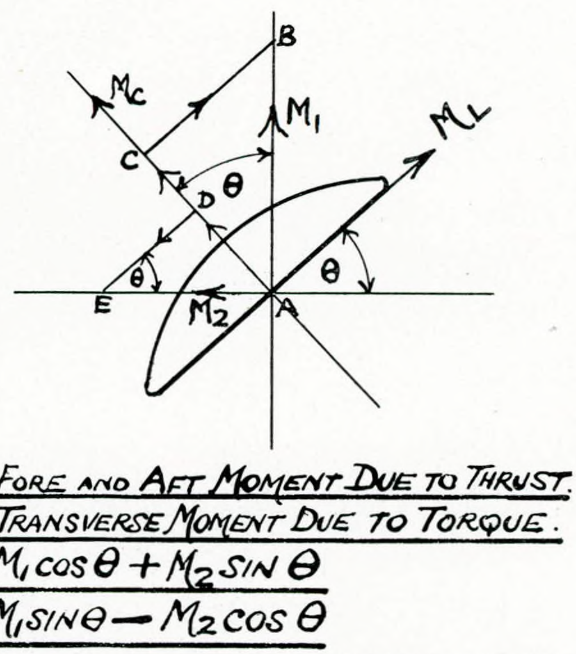


FIG. 1.—*Diagram of moments for stress calculations on propeller blade sections.*

Taylor also shows that the transverse moment M_2, due to the torque, is

$$M_2 = \frac{5252P_1(d-d_1)^2}{N(d^2-d_1^2)} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (2)$$

where,

M_2 = transverse moment in pound feet.
P_1 = power absorbed per blade.
N = revolutions per minute.

Before these moments can be used for the determination of the stresses on the blade section they must be resolved at right angles and parallel to the face of the blade section at which the stress is required.

Referring to Fig. 1, the moment at right angles to the blade face, $M_c = M_1\cos\theta + M_2\sin\theta$ .... (3)

The moment parallel to the blade face,

$$M_L = M_1\sin\theta - M_2\cos\theta \quad \ldots\ldots\ldots\ldots \quad (4)$$

where θ = pitch angle at the given radius of the section.

From a number of root sections of actual propellers with aerofoil contour, the author has found from graphical integration that to facilitate the calculations necessary for the computation of the stresses, the following formulæ may be adopted.

Referring to Fig. 2, if l = total width of the section in inches and t = maximum thickness in inches,

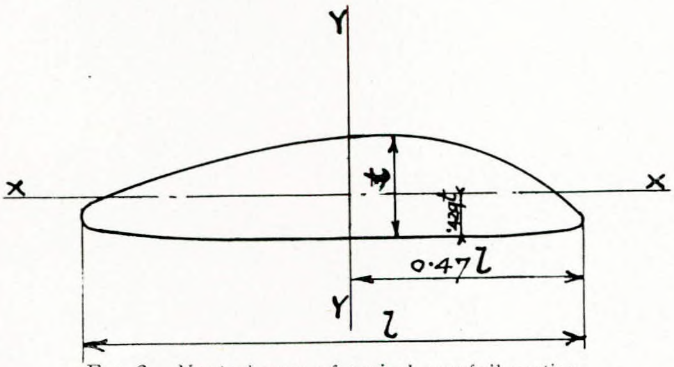


FIG. 2.—*Neutral axes of typical aerofoil section.*

Area of section $A = 0{\cdot}725\ l.t$ square inches ...... (5)

Neutral axis XX from face of blade section $= 0{\cdot}429\ t$ inches.............. (6)

Neutral axis YY from leading edge of blade section $= 0{\cdot}47\ l$ inches................. (7)

Moment of inertia about XX $= 0{\cdot}05\ lt^3$, inches^4........... (8)

Moment of inertia about YY $= 0{\cdot}04\ l^3t$, inches^4........... (9)

From these formulæ we may write,

Maximum tensile stress due to M_c on face of blade

$$= \frac{M_c \times 0{\cdot}429t}{0{\cdot}05l.t^3} = \frac{8{\cdot}58M_c}{lt^2} \text{ lb. per sq. inch} \ldots \quad (10)$$

Maximum tensile stress due to M_L on face of blade

$$= \frac{M_L \times 0{\cdot}53l}{0.04l^3t} = \frac{13{\cdot}25M_L}{l^2t} \text{ lb. per sq. inch} \ldots \quad (11)$$

Compressive stress due to M_c on back of blade at line of maximum thickness

$$= \frac{M_c \times 0{\cdot}571t}{0.05lt^3} = \frac{11{\cdot}42M_c}{lt^2} \text{ lb. per sq. inch} \ldots \quad (12)$$

Stress due to M on back of blade at line of maximum

$$\text{thickness} = \frac{M_L y}{0{\cdot}04l^3t} = \frac{25M_L}{l^3t} \text{ lb. per sq. inch} \ldots \quad (13)$$

where M_c and M_L are expressed in pound-inches, and y is the distance of the line of maximum thickness from the neutral axis YY.

In addition to the stresses due to M_c and M_L, there are those due to centrifugal force and centrifugal moment, the latter being due to the usual rake of the blade aft. The centrifugal stresses cannot be determined until the weight of the blade beyond the root section is known.

The thickness of the blade section can be calculated however from equations (10) and (11) above, the width of the section, l, having been previously determined from the contour of the blade to obtain the required developed surface. To allow a margin for the additional centrifugal stresses, the writer suggests that in equations (10) and (11) a tensile stress for manganese bronze of 5,000 to 6,000 pounds per square inch should be used.

In connection with the centrifugal stresses the centre of gravity of the blade beyond the root section also has to be found.

The author has taken a number of propellers and constructed curves of cross sectional areas, the area of this curve being the volume of the blade, together with a first moment figure for the determination of the centre of gravity of the blade beyond the root section.

The volume of the blade beyond the root section and the position of the centre of gravity from the shaft axis for the calculation of the centrifugal stresses from the results of these investigations may be approximated as follows :—

For Built Propellers.

Volume of blade beyond root section in cubic inches

$$= \left\{\begin{array}{l}\text{Developed} \\ \text{area of} \\ \text{blade in} \\ \text{sq. ins.}\end{array}\right\} \left\{\frac{\sum\left(\begin{array}{l}\text{Maximum} \\ \text{thickness of} \\ \text{each section}\end{array} \times \begin{array}{r}\text{Corresponding} \\ \text{radius of} \\ \text{each section}\end{array}\right)}{\sum(\text{radii of the sections})}\right\} \ldots\ldots\ldots \quad (14)$$

Centre of gravity from axis in feet =
$0{\cdot}535 \times$ radius of propeller in feet......... (15)

For Solid Propellers.

Volume of blade beyond root section in cubic inches $= 1{\cdot}15 \left\{\begin{array}{l}\text{developed area of} \\ \text{blade in sq. ins.}\end{array}\right\}$

$$\left\{\frac{\sum\left(\begin{array}{l}\text{Maximum} \\ \text{thickness of} \\ \text{each section}\end{array} \times \begin{array}{r}\text{Corresponding} \\ \text{radius of} \\ \text{each section}\end{array}\right)}{\sum(\text{radii of the sections})}\right\} \ldots\ldots \quad (16)$$

Centre of gravity from axis in feet =
$0{\cdot}48 \times$ radius of propeller in feet......... (17)

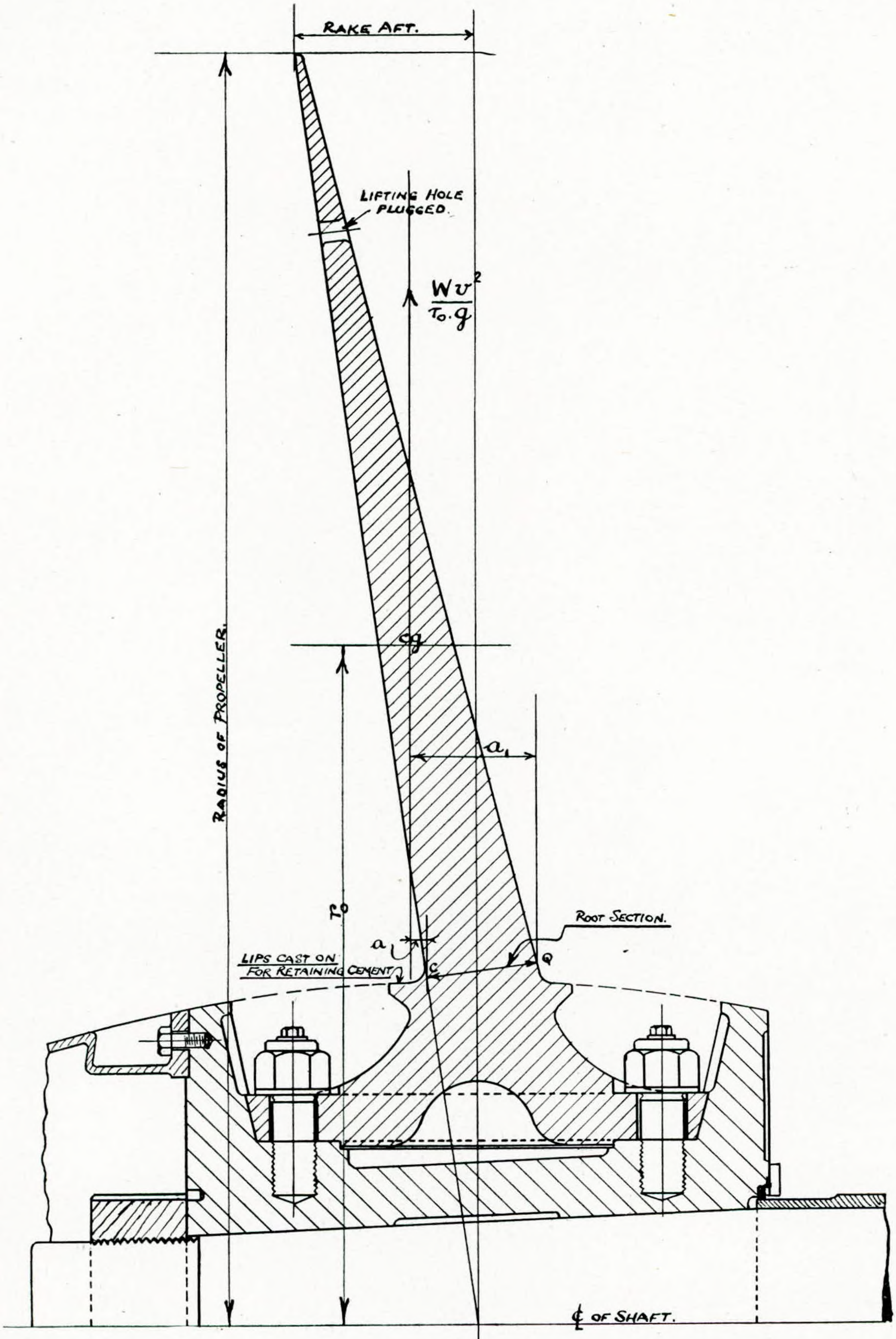


FIG. 3.—*Section of blade for centrifugal stress calculations.*

Centrifugal Stresses.

Stresses due to centrifugal force.

Let W = weight in pounds of the blade beyond section for which the stress is required

r_0 = radius of the centre of gravity of this portion from axis in feet

N = revolutions per minute

v = velocity of centre of gravity in feet per second.

Then,

$$\text{Centrifugal force} = \frac{W.v^2}{r_0.g.} \text{ ....pounds.........} \quad (18)$$

$$\text{Centrifugal stress} = \frac{W.v^2}{r_0.g.A} = \frac{W.v^2}{0{\cdot}725 r_0.g.l.t.} \text{ ......} \quad (19)$$

Stresses due to Centrifugal Moment.

Referring to Fig. 3.

$$\text{Centrifugal moment for tension at } C = \frac{W.v^2.a.}{r_0.g} \text{ ....} \quad (20)$$

Centrifugal moment for compression at Q

$$= \frac{W.v^2 a_1}{r_0.g.} \text{ ......} \quad (21)$$

Moment of inertia about XX (equation 8) $= 0{\cdot}05 lt^3 ins.^4$

Distance of furthermost fibre from neutral axis XX for tension $= 0{\cdot}429t$.

Distance of furthermost fibre from neutral axis XX for compression $= 0{\cdot}571t$.

Hence,

Tensile stress at C due to centrifugal moment

$$= \frac{0{\cdot}429 W.v^2.a.t}{0{\cdot}05 r_0.g.l.t^3} = \frac{8{\cdot}58 W.v^2.a.}{r_0.g.l.t^2} \text{ ....} \quad (22)$$

Compressive stress at Q due to centrifugal moment

$$= \frac{0{\cdot}571.W.v^2.a_1.t}{0{\cdot}05 r_0.g.l.t^3} = \frac{11{\cdot}42 W.v^2.a_1}{r_0.g.l.t^2} \text{ ......} \quad (23)$$

From the equations given above, we have,

Maximum tensile stress in pounds per sq. inch

$$= \frac{8{\cdot}58 Mc}{l.t^2} + \frac{13{\cdot}25 M_L}{l^2.t} + \frac{W.v^2}{0{\cdot}725 r_0.g.l.t} + \frac{8{\cdot}58 W.v^2.a}{r_0.g.l.t^2} \text{ ......} \quad (24)$$

Maximum compressive stress in pounds per sq. inch

$$= \frac{11{\cdot}42 Mc}{l.t^2} + \frac{25.M_L.y}{l^3 t} - \frac{W.v^2}{0{\cdot}725\, r_0.g.l.t} + \frac{11{\cdot}42\, W.v^2.a_1}{r_0.g.l.t^2} \quad (25)$$

The above formulæ will now be applied to obtain the thickness and stresses in the root section of the propeller for the vessel whose power and propeller dimensions were determined in the author's paper "The Powering of Ships", published in the TRANSACTIONS of The Institute for May, 1940.

In this paper the propeller dimensions for a twin-screw ship of 19,500 s.h.p. at 125 revolutions per minute, were as follows :—

Diameter	=	17ft. 9in.
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	=	66·9 per cent.
Power per propeller	=	9,750 s.h.p.
Power per blade	=	3,250 s.h.p.

The propellers were of the built type with sunk blades and a streamlined boss, and it will be seen from the propeller design, Fig. 4, that the diameter of the root section is 4ft. 10in. and the pitch at this section has been reduced to 14ft. 3in.

$$\text{The thrust per blade, } T_0 = \frac{33{,}000 \times 9{,}750}{3 \times 17{\cdot}4 \times 125} = 49{,}310\text{lb}$$

$$d = 17{\cdot}75\text{ft. and } d_1 = 4{\cdot}833\text{ft.}$$

From equation 1 above,

$$M_1 = \frac{49{,}310(2 \times 17{\cdot}75^2 - 17{\cdot}75 \times 4{\cdot}833 - 4{\cdot}833^2)}{6(17{\cdot}75 + 4{\cdot}833)}$$

$$= 189{,}274\text{lb.-ft.}$$

From equation 2,

$$M_2 = \frac{5{,}252 \times 3{,}250(17{\cdot}75 - 4{\cdot}833)^2}{125(17{\cdot}75^2 - 4{\cdot}833^2)} = 78{,}106\text{lb.-ft.}$$

For the pitch angle θ at the root section we have,

$$\tan\theta = \frac{14{\cdot}25}{6{\cdot}28 \times 2{\cdot}417} = 0{\cdot}93881 = \tan 43° - 12'$$

$$\sin\theta = 0{\cdot}6845 \text{ and } \cos\theta = 0{\cdot}7290.$$

From equation 3,

moment at right angles to blade face, M_c

$= 189{,}274 \times {\cdot}7290 + 78{,}106 \times {\cdot}6845$

$= 191{,}444\text{lb.-ft.} = 2{,}297{,}328\text{lb.-in.}$

Moment parallel to blade face M

$= 189{,}274 \times {\cdot}6845 - 78{,}1C6 \times 0{\cdot}7290$

$= 72{,}619\text{lb.-ft.} = 871{,}428\text{lb.-in.}$

The weight of the blade cannot be found until the thickness is determined. The width, l, of the sections is known from the contour of the blade surface and in the present design the width of the root section was 46⅛in. Assuming a tensile stress of 5,500 lb. per sq. in. to cover for the stress due to M_c and M_L, we have from equations, 10 and 11,

$$5{,}500 = \frac{8{\cdot}58 M_c}{lt^2} + \frac{13{\cdot}25 M_L}{l^2 t}$$

$$= \frac{8{\cdot}58 \times 2{,}297{,}328}{46{\cdot}125 t^2} + \frac{13{\cdot}25 \times 871{,}428}{46{\cdot}125^2 t}$$

whence, $t = 9{\cdot}3321$in., say 9⅜in. maximum root thickness.

The maximum thickness at the tip of the blade was made $\frac{11}{16}$in. and the thickness and the cross sections at the other radii are as shown in Fig. 4. The rake aft of the blades was made 15in. in order to obtain a tip clearance between the hull and the propeller of about 33in.

Fig. 5 shows the curve of cross sectional areas for this propeller and also the first moment figure. The scale of the diagram as drawn was 1in. vertical equals 12in. and 1in. horizontal equals 50 sq. in., so that 1 sq. in. equals 600 cubic in. The area of the cross-sectional area curve was 27·72 sq. in. giving the volume beyond the root section as 16,632 cubic ins. Taking the weight of manganese bronze per cubic in. as 0·301lb., the weight beyond the root section is 5,006lb. The area of the first moment figure was 13·17 sq. in. and the axes were chosen at 10ft. apart, so that the c.g. from the axis of the shaft

$$= \frac{13{\cdot}17 \times 10}{27{\cdot}72} = 4{\cdot}75\text{ft.} = {\cdot}535 \times \text{radius of propeller.}$$

At this distance from the axis the arm (a) for tension at C, Fig. 3, is 1·25in. and the arm (a_1) for compression at Q, Fig. 3, is 10·5ins.

From the combined equations 10 and 11, with $t = 9{\cdot}375$in. as determined above,

Maximum tensile stress due to M_c and M_L =

$$\frac{8{\cdot}58 M_c}{lt^2} + \frac{13{\cdot}25 M_L}{l^2 t} = \frac{8{\cdot}58 \times 2{,}297{,}328}{46{\cdot}125 \times 9{\cdot}375^2} + \frac{13{\cdot}25 \times 871{,}428}{46{\cdot}125^2 \times 9{\cdot}375}$$

$= 5{,}453$lb. per sq. in.

Centrifugal Stresses.

$$\text{Velocity of c.g.} = \frac{2\pi r_0 . N}{60} = \frac{2\pi \times 4{\cdot}75 \times 125}{60}$$

$= 62{\cdot}147$ft. per sec.

Weight of blade beyond root section = 5,006lb.

$$\text{Centrifugal force} = \frac{5{,}006 \times 62{\cdot}147^2}{4{\cdot}75 \times 32{\cdot}2} = 126{,}400\text{lb.}$$

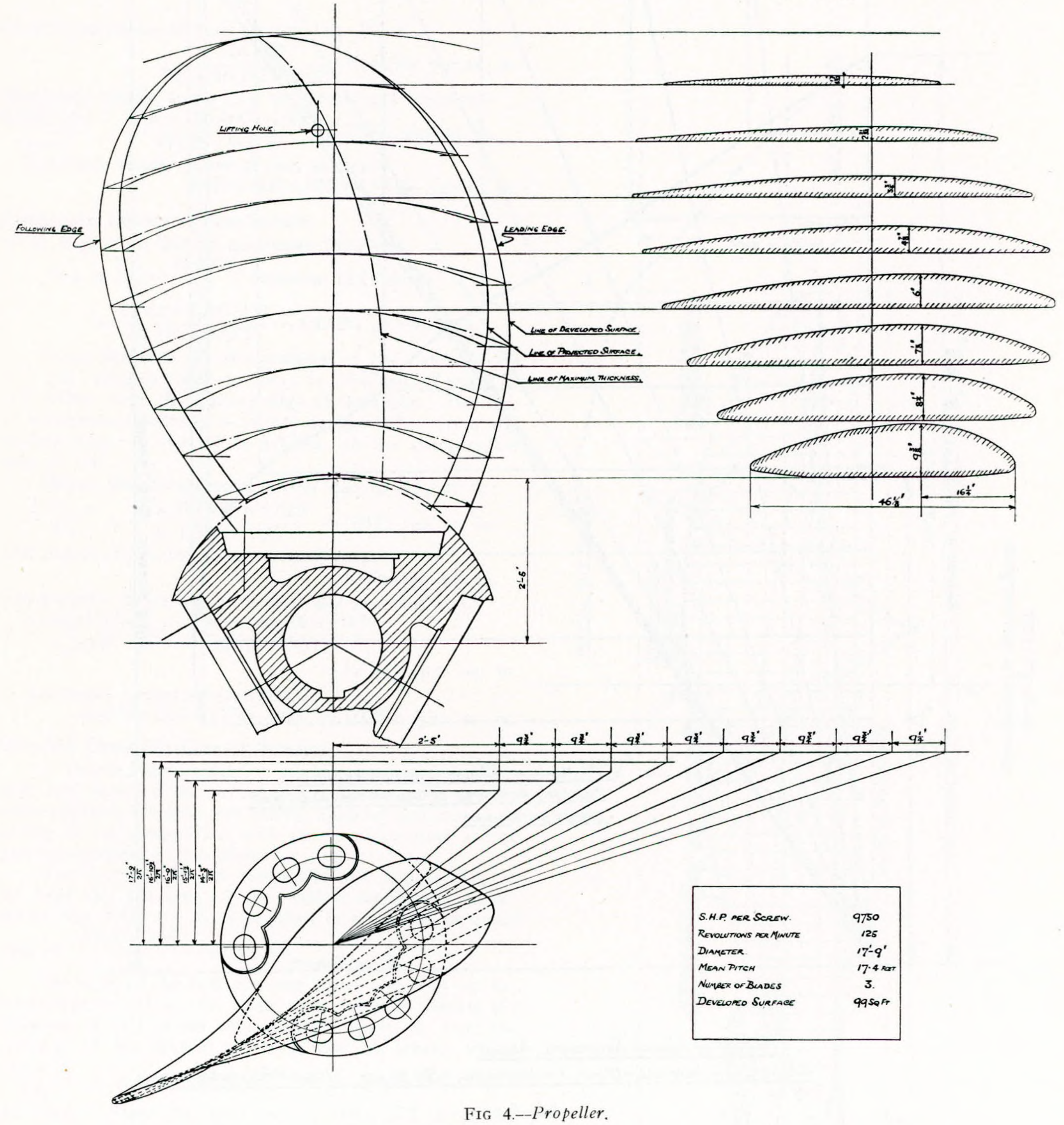


FIG 4.—*Propeller.*

the moment of inertia about $XX = I_{LL} - Ah^2 = 17{,}775 -$

The Computation of the Stresses in a Propeller Blade Section.

$324 \times 7^2 = 1{,}900\text{in.}^4$ as compared with $\cdot 05lt^3 = \cdot 05 \times 46\cdot 125 \times 9\cdot 375^3 = 1{,}900\cdot 3\text{in.}^4$, given in equation (8).

Position of Neutral Axis and Moment of Inertia for M_L (Fig. 6).

The reading of the first moment figure was 9·19 sq. in., equivalent to 147·04 sq. in. The axes were chosen 48in. apart, and since the area of the section is 324 sq. in. the position of YY from the axis $KK = \dfrac{147\cdot 04 \times 48}{324} = 21\cdot 78$in., or $0\cdot 472l$, as compared with $0\cdot 47l$ given in equation (7).

The reading of the second moment figure was 5·17 sq. in., equivalent to $\dfrac{5\cdot 17 \times 144}{9} = 82\cdot 72$ sq. in. Since the axes are 48in. apart, the moment of inertia about $KK = 82\cdot 72 \times 48^2 = 190{,}550\text{in.}^4$. The moment of inertia about $YY = I_{KK} - Ah_1{}^2 = 190{,}550 - 324 \times 21\cdot 78^2 = 36{,}855\text{in.}^4$ as compared with $0\cdot 04l^3t = \cdot 04 \times 46\cdot 125^3 \times 9\cdot 375 = 36{,}800\text{in.}^4$ (eq. 9).

Tensile Stresses on Face of Root Section. (Referring to Fig. 6).

Tensile stress at A.

A is 2in. from XX and 24·345in. from YY.

Tensile stress at A due to M_c

$$= \frac{M_c y_{cA}}{1{,}900} = \frac{2{,}297{,}328 \times 2}{1{,}900} = 2{,}418\text{lb. per sq. in.}$$

Tensile stress at A due to M_L

$$= \frac{M_L \cdot y_{LA}}{36{,}855} = \frac{871{,}428 \times 24\cdot 345}{36{,}855} = 576\text{lb. per sq. in.}$$

Centrifugal stress $= 403$lb. per sq. in.

Centrifugal Moment.

A is 3in. from the vertical through c.g. and 2in. from XX (Figs. 3 and 6).

The centrifugal force $= 126{,}400$lb.

Tensile stress at A due to centrifugal moment

$$= \frac{126{,}400 \times 3 \times 2}{1{,}900} = 399\text{lb. per sq. in.}$$

Total tensile stress at $A = 2{,}418 + 576 + 403 + 399$
$= 3{,}796$lb. per sq. in.

Stress at B. B is $3\frac{3}{8}$in. from XX and 18·345in. from YY.

Tensile stress at B due to M_c

$$= \frac{M_c y_{cB}}{1{,}900} = \frac{2{,}297{,}328 \times 3\cdot 375}{1{,}900} = 4{,}080\text{lb. per sq. in.}$$

Tensile stress at B due to M_L

$$= \frac{M_L y_{LB}}{36{,}855} = \frac{871{,}428 \times 18\cdot 345}{36{,}855} = 433\text{lb. per sq. in.}$$

Centrifugal stress at $B = 403$lb. per sq. in.

Centrifugal Moment. B is 1·75in. from the vertical through c.g., so that tensile stress at B due to centrifugal moment

$$= \frac{126{,}400 \times 1\cdot 75 \times 3\cdot 375}{1{,}900} = 393\text{lb. per sq. in.}$$

Total tensile stress at $B = 4{,}080 + 433 + 403 + 393$
$= 5{,}309$lb. per sq. in.

Stress at C. C is 4in. from XX and 12·345in. from YY.

Tensile stress at C due to M_c

$$= \frac{M_c y_{cc}}{1{,}900} \quad \frac{2{,}297{,}328 \times 4}{1{,}900} = 4{,}836\text{lb. per sq. in.}$$

Tensile stress at C due to M_L

$$= \frac{M_L y_{LC}}{36{,}855} = \frac{871{,}428 \times 12\cdot 345}{36{,}855} = 292\text{lb. per sq. in.}$$

Centrifugal stress at $C = 403$lb. per sq. in.

Centrifugal Moment. C is 1·25in. from the vertical through c.g.

Tensile stress at C due to centrifugal moment

$$= \frac{126{,}400 \times 1\cdot 25 \times 4}{1{,}900} = 332\cdot 6, \text{ say } 333\text{lb. per sq. in.}$$

Total tensile stress at $C = 4{,}836 + 292 + 403 + 333$
$= 5{,}864$lb. per sq. in.

Stress at D. D is 4in. from XX and 6·345in. from YY.

Tensile stress at D due to $M_c = 4{,}836$lb. per sq. in.

Tensile stress at D due to M_L

$$= \frac{M_L y_{LD}}{36{,}855} = \frac{871{,}428 \times 6\cdot 345}{36{,}855} = 150\text{lb. per sq. in.}$$

Centrifugal stress $= 403$lb. per sq. in.

Centrifugal Moment. D is 1·25in. from the vertical through c.g., hence centrifugal moment stress
$= 333$lb. per sq. in.

Total tensile stress at $D = 4{,}836 + 150 + 403 + 333$
$= 5{,}722$lb. per sq. in.

Stress at E. E is 4in. from XX and 0·345in. from YY.

Tensile stress at E due to $M_c = 4{,}836$lb. per sq. in.

Tensile stress at E due to M_L

$$= \frac{M_L y_{LE}}{36{,}855} = \frac{871{,}428 \times 0\cdot 345}{36{,}855} = 8\cdot 0\text{lb. per sq. in.}$$

Centrifugal stress $= 403$lb. per sq. in.

Centrifugal moment stress $= 333$lb. per sq. in.

Total tensile stress at $E = 4{,}836 + 8\cdot 0 + 403 + 333$
$= 5{,}580$lb. per sq. in.

Stress at F. F is 4in. from XX and 5·625in. on compressive side of YY.

Tensile stress at F due to $M_c = 4{,}836$lb. per sq. in.

Compressive stress at F due to M_L

$$= \frac{M_L y_{LF}}{36{,}855} = \frac{871{,}428 \times 5\cdot 625}{36{,}855}$$

$= 133$lb. per sq. in.

Centrifugal stress $= 403$lb. per sq. in.

Centrifugal moment stress $= 333$lb. per sq. in.

Total tensile stress at $F = 4{,}836 - 133 + 403 + 333$
$= 5{,}439$lb. per sq. in.

Stress at G. G is 4in. from XX and 11·625in. on compressive side of YY.

Tensile stress at G due to $M_c = 4{,}836$lb. per sq. in.

Compressive stress at G due to M_L

$$= \frac{M_L y_{LG}}{36{,}855} = \frac{871{,}428 \times 11\cdot 625}{36{,}855} = 274\text{lb. per sq. in.}$$

Centrifugal stress $= 403$lb. per sq. in.

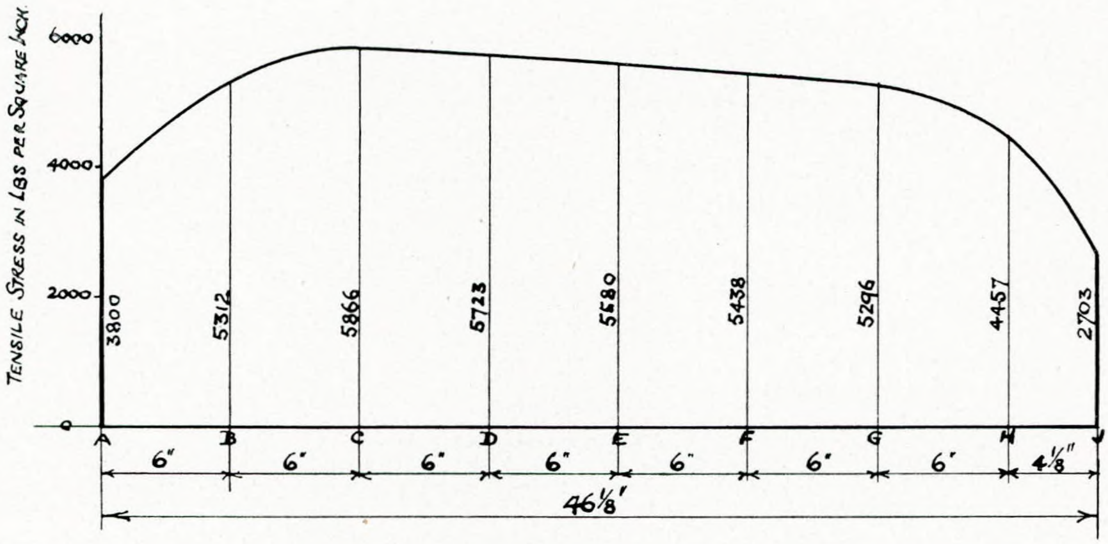
Centrifugal moment stress $= 333$lb. per sq. in.

Total tensile stress at $G = 4{,}836 - 274 + 403 + 333$
$= 5{,}298$lb. per sq. in.

Stress at H. H is $3\frac{3}{8}$in. from XX and 17·625in. on compressive side of YY.

Tensile stress at H due to M_c

$$= \frac{M_c y_{cH}}{1{,}900} = \frac{2{,}297{,}328 \times 3\cdot 375}{1{,}900} = 4{,}080\text{lb. per sq. in.}$$



TENSILE STRESS ON FACE OF BLADE.
AT ROOT SECTION.

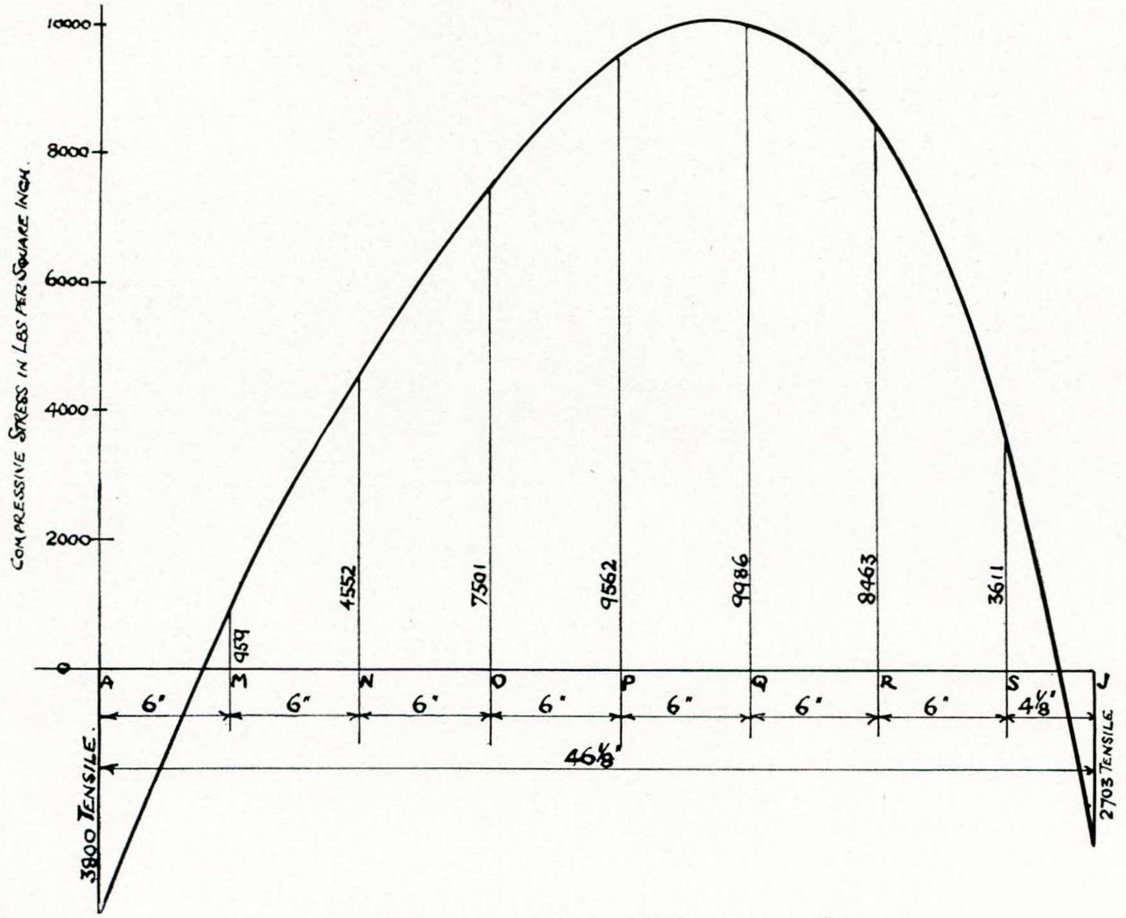


FIG. 7.—*Compressive stress on back of blade at root section.*

Compressive stress at H due to M_L

$$= \frac{M_L y_{LH}}{36{,}855} = \frac{871{,}428 \times 17{\cdot}625}{36{,}855} = 416\text{lb. per sq. in.}$$

Centrifugal stress = 403lb. per sq. in.

Centrifugal Moment. H is 1·75in. from vertical through c.g. so that the centrifugal moment stress = 393lb. per sq. in.

Total tensile stress at $H = 4{,}080 - 416 + 403 + 393$ = 4,460lb. per sq. in.

Stress at J. J is 2in. from XX and 21·78in. from YY.

Tensile stress at J due to M_c = 2,418lb. per sq. in.

Compressive stress at J due to M_L

$$= \frac{M_L y_{LJ}}{36{,}855} = \frac{871{,}428 \times 21{\cdot}78}{36{,}855} = 514\text{lb. per sq. in.}$$

Centrifugal stress = 403lb. per sq. in.

Centrifugal moment stress = 399lb. per sq. in.

Total tensile stress at $J = 2{,}418 - 514 + 403 + 399$ = 2,706lb. per sq. in.

Compressive Stresses on Back of Root Section.

Stress at M. M is $1\frac{1}{8}$in. from XX and 18·345in. from YY.

Compressive stress at M due to M_c

$$= \frac{M_c y_{cM}}{1{,}900} = \frac{2{,}297{,}328 \times 1{\cdot}125}{1{,}900} = 1{,}360\text{lb. per sq. in.}$$

Tensile stress at M due to M_L

$$= \frac{M_L y_{LM}}{36{,}855} = \frac{871{,}428 \times 18{\cdot}345}{36{,}855} = 433\text{lb. per sq. in.}$$

Centrifugal stress = 403lb. per sq. in.

Centrifugal Moment. M is $6\frac{1}{4}$in. on compressive side of vertical through c.g.

Compressive stress at M due to centrifugal moment

$$= \frac{126{,}400 \times 6{\cdot}25 \times 1{\cdot}125}{1{,}900} = 468\text{lb. per sq. in.}$$

Total compressive stress at $M = 1{,}360 - 433 - 403 + 468$ = 992lb. per sq. in.

Stress at N. N is 3in. from XX and 12·345in. from YY.

Compressive stress at N due to M_c

$$= \frac{M_c y_{cN}}{1{,}900} = \frac{2{,}297{,}328 \times 3}{1{,}900} = 3{,}627\text{lb. per sq. in.}$$

Tensile stress at N due to M_L

$$= \frac{M_L y_{LN}}{36{,}855} = \frac{871{,}428 \times 12{\cdot}345}{36{,}855} = 292\text{lb. per sq. in.}$$

Centrifugal stress = 403lb. per sq. in. tensile.

Centrifugal Moment. N is $8\frac{1}{8}$in. on compressive side of vertical through c.g.

Compressive stress at N due to centrifugal moment

$$= \frac{126{,}400 \times 8{\cdot}125 \times 3}{1{,}900} = 1{,}622\text{lb. per sq. in.}$$

Total compressive stress at $N = 3{,}627 - 292 - 403 + 1{,}622$ = 4,554lb. per sq. in.

Stress at O. O is $4\frac{3}{8}$in. from XX and 6·345in. from YY.

Compressive stress at O due to M_c

$$= \frac{M_c y_{cO}}{1{,}900} = \frac{2{,}297{,}328 \times 4{\cdot}375}{1{,}900} = 5{,}290\text{lb. per sq. in.}$$

Tensile stress at O due to M^L

$$= \frac{M_L y_{LO}}{36{,}855} = \frac{871{,}428 \times 6{\cdot}345}{36{,}855} = 150\text{lb. per sq. in.}$$

Centrifugal stress = 403lb. per sq. in.

Centrifugal Moment. O is $9\frac{1}{2}$in. on compressive side of vertical through c.g.

Compressive stress at O due to centrifugal moment

$$= \frac{126{,}400 \times 9{\cdot}5 \times 4{\cdot}375}{1{,}900} = 2{,}765\text{lb. per sq. in.}$$

Total compressive stress at $O = 5{,}290 - 150 - 403 + 2{,}765$ = 7,502lb. per sq. in.

Stress at P. P is $5\frac{1}{4}$in. from XX and 0·345in. from YY.

Compressive stress at P due to M_c

$$= \frac{M_c y_{cP}}{1{,}900} = \frac{2{,}297{,}328 \times 5{\cdot}25}{1{,}900} = 6{,}350\text{lb. per sq. in.}$$

Tensile stress at P due to M_L

$$= \frac{M_L y_{LP}}{36{,}855} = \frac{871{,}428 \times 0{\cdot}345}{36{,}855}$$

= 8lb. per sq. in.

Centrifugal stress = 403lb. per sq. in. tensile.

Centrifugal Moment. P is $10\frac{3}{8}$in. on compressive side of the vertical through c.g.

Compressive stress at P due to centrifugal moment

$$= \frac{126{,}400 \times 10{\cdot}375 \times 5{\cdot}25}{1{,}900} = 3{,}623\text{lb. per sq. in.}$$

Total compressive stress at $P = 6{,}350 - 8 - 403 + 3{,}623$ = 9,562lb. per sq. in.

Stress at Q. Q is $5\frac{3}{8}$in. from XX and 5·625in. on compressive side of YY.

Compressive stress at Q due to M_c

$$= \frac{M_c y_{cQ}}{1{,}900} = \frac{2{,}297{,}328 \times 5{\cdot}375}{1{,}900} = 6{,}500\text{lb. per sq. in.}$$

Compressive stress at Q due to M_L

$$= \frac{M_L y_{LQ}}{36{,}855} = \frac{871{,}428 \times 5{\cdot}625}{36{,}855}$$

= 133lb. per sq. in.

Centrifugal stress = 403lb. per sq. in. tensile.

Centrifugal Moment. Q is $10\frac{1}{2}$in. on compressive side of vertical through c.g.

Compressive stress at Q due to centrifugal moment

$$= \frac{126{,}400 \times 10{\cdot}5 \times 5{\cdot}375}{1{,}900} = 3{,}755\text{lb. per sq. in.}$$

Total compressive stress at $Q = 6{,}500 + 134 - 403 + 3{,}755$ = 9,985lb. per sq. in.

Stress at R. R is $4\frac{5}{8}$in. from XX and 11·625in. on compressive side of YY.

Compressive stress at R due to M_c

$$= \frac{M_c y_{cR}}{1{,}900} = \frac{2{,}297{,}328 \times 4{\cdot}625}{1{,}900} = 5{,}590\text{lb. per sq. in.}$$

Compressive stress at R due to M_L

$$= \frac{M_L y_{LR}}{36{,}855} = \frac{871{,}428 \times 11{\cdot}625}{36{,}855} = 274\text{lb. per sq. in.}$$

Centrifugal stress = 403lb. per sq. in. tensile.

Centrifugal Moment. R is $9\frac{3}{4}$in. on compressive side of vertical through c.g.

Compressive stress at R due to centrifugal moment

$$= \frac{126{,}400 \times 9{\cdot}75 \times 4{\cdot}625}{1{,}900} = 3{,}000\text{lb. per sq. in.}$$

Total compressive stress at $R = 5{,}590 + 274 - 403 + 3{,}000$ = 8,461lb. per sq. in.

Stress at S. S is $2\frac{1}{8}$in. on compressive side of XX and 17·625in. on compressive side of YY.

Compressive stress at S due to M_c

$$= \frac{M_c\, y_{cS}}{1{,}900} = \frac{2{,}297{,}328 \times 2{\cdot}125}{1{,}900} = 2{,}570\text{lb. per sq. in.}$$

Compressive stress at S due to M^L

$$= \frac{M_L\, y_{LS}}{36{,}855} = \frac{871{,}428 \times 17{\cdot}625}{36{,}855} = 416\text{lb. per sq. in.}$$

Centrifugal stress = 403lb. per sq. in. tensile.

Centrifugal Moment. S is $7\frac{1}{4}$in. on compressive side of vertical through c.g.

Compressive stress at S due to centrifugal moment

$$= \frac{126{,}400 \times 7{\cdot}25 \times 2{\cdot}125}{1{,}900} = 1{,}025\text{lb. per sq. in.}$$

Total compressive stress at $S = 2{,}570 + 416 - 403 + 1{,}025$
$= 3{,}608$lb. per sq. in.

Curves of the tensile and compressive stresses are shown in Fig. 7.

The maximum tensile stress from the detailed calculations is at C and equals 5,864lb. per sq. in. as compared with 6,191lb. per sq. in. using the formulæ given in the first part of the paper. This difference is due to assuming in the formulæ that C is at the maximum distance from YY as well as from XX. The formulæ calculations are therefore on the safe side and the writer submits may be used with confidence.

From a number of test sheets of manganese bronze an ultimate strength of 35 tons per sq. in. may be taken as an average, so that the factor of safety works out at 13·365 from the detailed calculations and 12·663 from the formulæ calculations.

Size of Studs Required for Built Propellers.

In propellers of the built type each blade is secured to the boss usually by five studs on the tension side and four on the compression side when going in the ahead direction. When going astern the stresses will be reversed.

The size of the studs may be computed by resolving the fore and aft moment M_1 and the transverse moment M_2, Fig. 8, the stresses due to the centrifugal force and moment being allowed for by adopting a low tensile stress to cover for M_1 and M_2.

The resultant moment due to M_1 and M_2, Fig. 8, will be,

$$M_3 = \sqrt{M_1^2 + M_2^2} \text{ pound-inches} \quad (26)$$

This moment must equal,

$$n \times a \times f \times d \text{ pound-inches} \quad (27)$$

where, n = number of studs in tension,
a = area of stud at bottom of thread in sq. inches.
f = tensile stress in lb. per sq. in.
d = pitch circle diameter of studs in inches.

Hence,

$$a = \frac{M_3}{n.f.d.} \quad (28)$$

The angle at which M_3 acts relative to the fore and aft axis of the shaft may be found from,

$$\tan \alpha = \frac{M_2}{M_1} \quad (29)$$

Applying the above formulæ to the propeller blade which is the subject of this paper we have,

$M_1 = 189{,}274$lb.-ft.; $M_2 = 78{,}106$lb.-ft.
$M_3 = \sqrt{189{,}274^2 + 78{,}106^2} = 204{,}760$lb.-ft.
$= 2{,}457{,}120$lb.-in.

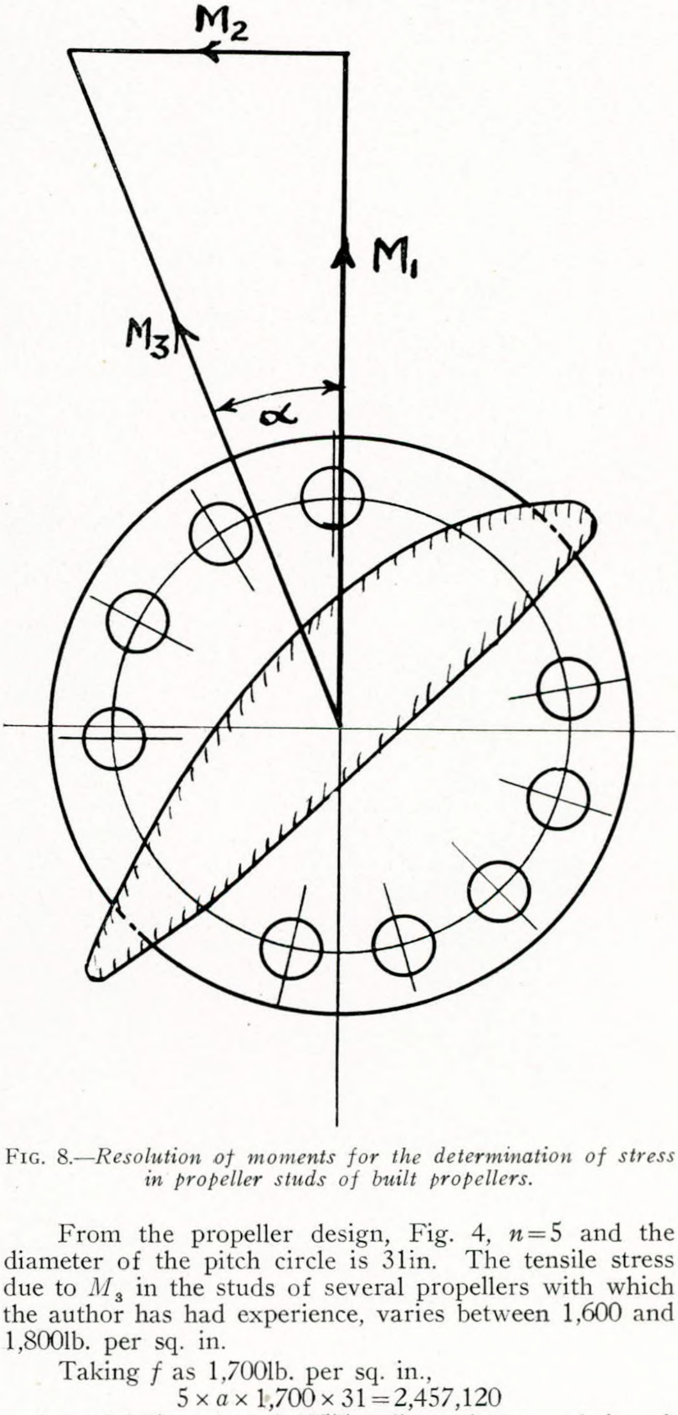


FIG. 8.—*Resolution of moments for the determination of stress in propeller studs of built propellers.*

From the propeller design, Fig. 4, $n = 5$ and the diameter of the pitch circle is 31in. The tensile stress due to M_3 in the studs of several propellers with which the author has had experience, varies between 1,600 and 1,800lb. per sq. in.

Taking f as 1,700lb. per sq. in.,

$$5 \times a \times 1{,}700 \times 31 = 2{,}457{,}120$$

$\therefore a = 9{\cdot}325$ sq. in. $= 3{\cdot}4457$in. dia. at bottom of thread.

The studs are usually screwed 4 threads per inch giving a reduction in diameter of 0·32in., so that the studs will be $3{\cdot}4457+0{\cdot}32=3{\cdot}7657$in. diameter, say $3\frac{3}{4}$in. diameter.

The angle at which M_3 acts relative to the fore and aft axis of the shaft from eq. 29 will be,

$$\tan\alpha=\frac{M_2}{M_1}=\frac{78{,}106}{189{,}274}=0{\cdot}41266$$
$$\therefore\ \alpha=22°\text{-}25'.$$

DISCUSSION.

Contributions by correspondence, not exceeding 1,000 words in length, are invited to the discussion on this paper. Such contributions will be published, with the author's reply, in a subsequent issue of the TRANSACTIONS.

Discussion—"Research at the William Froude Laboratory."

Dr. G. S. Baker, replying to Mr. J. Hamilton Gibson's remarks published in the March, 1941, issue, pp. 37-8, stated that Mr. Gibson's suggestion was very interesting. In several cases they had in fact proceeded on the lines suggested, but these had always been picked ships which had been thoroughly tested in the tank, so that they had basic data with which to compare the ship data. All Mr. Kent's full scale rough water work was based on a similar procedure. They would be willing to take up such work for any firm and would welcome any co-operation of this kind. It should be added that as full a report as possible was made to the firm concerned, but data was only published when the results were regarded as a good illustration of some scientific principle, or as showing some new feature in design, and then only after the formal permission of the firm concerned had been obtained.

ELECTION OF MEMBERS.

List of those elected by the Council during the period 25th January to 4th April, 1941.

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Henry Augustus Cowe.
Duncan Cameron.
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Justus Schwersenski.
Robert James Wilson.

Students.

George Stewart Ramsay Gordon.
Vinayagamoorthy Veluppillai.
Arthur Wood.

Transfer from Associate to Member.

Stanley Walter Edwards.
Clement Philip Harrison.
William Wright.

ADDITIONS TO THE LIBRARY.

Presented by the Publishers.

The British Corporation Register of Shipping and Aircraft, 1941.

British Standard Specification No. 938-1941 for Metal Arc Welding as applied to Tubular Steel Structural Members.

Manual of Electric Arc Welding. By E. H. Hubert.

In the February, 1941, issue of the TRANSACTIONS, page 13, a review appeared in error of the above book, which has been published for some years. Also, the publishers and price of the book are other than that stated, and members are therefore advised to ignore the reference to the book in the February issue.

Drawing and Development for Practical Welding. By F. W. Sykes. Sir Isaac Pitman & Sons, 72 pp., 71 illus., 4s. net.

This book should prove of practical assistance to welders and young engineers, particularly those in the plate and welding shops. Chapter I introduces the beginner to the first principles of sketching, isometric and perspective, with many simple examples which when mastered should enable the student to acquire a valuable ability to make clear and bold sketches of correct proportions and sequence of views. The next chapter deals with standard welds and symbols, and also refers to the appropriate B.S. specifications. An example in figures of the stress formula on page 26 might have been given to assist the junior, also the angle for plate edge preparation of vee butts.

Much useful information is given concerning various types of fillet welds, whilst the "break joint", or staggered longitudinal seam for large pipe welding, is stated to avoid concentration of weld and stresses at one point. It should also be stated that where it is necessary to fit and weld reinforcing plates or cover straps, abrupt and equal changes of section should be avoided where possible.

Chapter III deals with the principles of plate development for welding fabrication, and forms a very useful reference for lobster back, cone, triangular and branch to pipe developments, the object being to indicate the basic principles and so put the student on the right track; having mastered and learnt to apply these principles, this should enable him to improve his position and prospects.

The author in chapter IV touches only briefly on the problem of distortion by welding and how it should be avoided or counteracted, stating that this is not within the scope of this book. However, several simple examples are given, but much more could be said on this subject.

It is altogether a very useful little book.

Philips Practical Welding Course. Philips Lamps Ltd., 102 pp., 108 illus., 3s. 6d. net.

"Philips Practical Welding Course" is a useful handbook, well printed and illustrated, and contains much information in a simple form. Commencing with a historical description of welding, a useful little chapter on the fundamental principles of electricity follows. Then a description is given of the welder's plant and tools with information of various types of electrodes, functions of the coating, and electrodes for special metals. As regards special metals the reader should bear in mind that the temperature of the source of heat has to be considered, and that in certain cases the oxy-acetylene or other processes are for this reason more suitable. The manipulation of the arc in various positions and detailed examples of welding practice are considered and the stresses set up. Examination of weld metal including X-rays is discussed as well as considerations governing the cost of work; finally some useful general tables are given.

Scientific Facts and Data. Negretti & Zambra, London. No published price. 64 pp., illustrated.

The previous publication entitled "Meteorological and other Facts and Data" issued by this well-known firm proved so popular that they decided to produce a revised edition containing many more interesting and useful scientific, meteorological, and other facts, and this decision has resulted in the present publication. For easy reference, items have been placed alphabetically, under "ocean", "sun", etc.; but an exception has been made with temperatures, under which heading will be found temperatures of metals, ignition temperatures, etc.

Members of The Institute will be interested to learn that they can obtain copies of the book gratis on application through the Secretary.

JUNIOR SECTION.

Naval Architecture and Ship Construction (Chapter VIII).

By R. S. HOGG, M.I.N.A.

The Fore End of the Ship.

Before proceeding to a discussion of the structural arrangements at the fore end of the ship, it may be desirable to refer to two well-known attempts in recent years to improve propulsive performance by certain modifications in the design of the underwater form forward.

The designer finds it convenient to divide the underwater body into three portions:—

(1) *The entrance* is that portion of the *tapering* underwater body situated *forward* of the mid length.

(2) *The run* is that portion of the *tapering* underwater body situated *abaft* the mid length.

(3) *The parallel body* is that portion of the underwater body which lies amidships and for which the cross section is constant (Fig. 82).

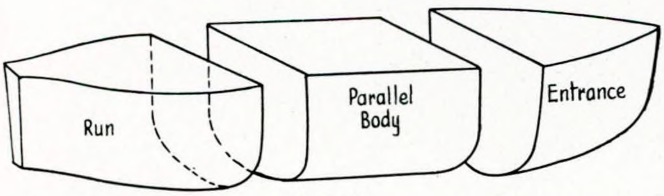


FIG. 82.

Now one of the first things which has to be done in producing a new design is the drawing out of a *curve of cross sectional areas.* This curve is drawn on a base of length, and the ordinates measure to scale the cross sectional area of the underwater form at the specified position in the ship's length. When it is understood that the *block coefficient of fineness* of a modern cargo steamer is in the region of ·8, it is not difficult to see that the midship section must be very full (mid-section area coefficient will almost always be well above ·95) and that this section must be maintained for a considerable length amidships. This is tantamount to saying that the vessel possesses considerable *parallel body* and of course that part of the cross sectional area curve amidships will be a straight line parallel to the base (Fig. 83). Accepting

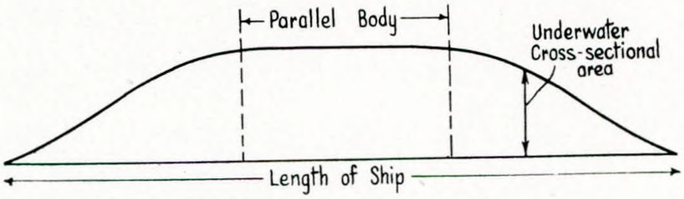


FIG. 83.—*Curve of cross-sectional areas.*

these conditions, the problem of the designer is to find how to complete the cross sectional area curve so as to produce the least resistful form. Discussions on this absorbing subject go right back to the monumental work of Scott Russell, who on the suggestions of Brunel designed the famous "Great Eastern". Scott Russell used no parallel body, and employed a *snubbed* versine curve for his cross sectional areas. A versine curve is one whose ordinates measure to scale the values of $(1 - \text{cosine } \theta)$, and *snubbing* the ends amounts in effect to straightening them out. There is much more that might be said on the subject, but it will be sufficient to hint that modern designers are naturally coloured in their views by the published experience of such men as Scott Russell, William and R. E. Froude, and G. S. Baker, to mention only the more important of these investigators. It is not surprising therefore to find in the *Maier* system that much attention was given to the idea of *straightening out* the forward portion of the cross sectional area curve.

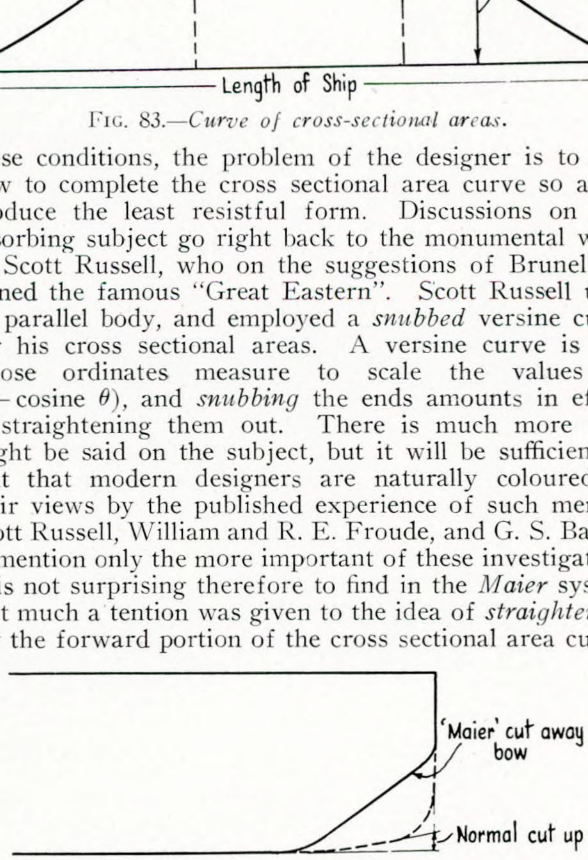


FIG. 84.—*Cut away bow.*

What does appear as surprising is that the same amount of care should not have been given to the rest of the curve. Apparently the early Maier investigations were carried out in respect of an ice-breaker, which necessarily has a cut away bow (Fig. 84). The ship seems to have been a success and the investigators were impressed with the idea that the cut away bow contributed appreciably to the efficiency of the form. They therefore incorporated this arrangement in their patents. Summarized, the *Maier patent* as originally set out would appear to be : *An attempt to improve the propulsive performance of a ship by straightening out the forward portion of the curve of cross sectional areas, and by fitting a cut away bow.* So far as is understood, the normal procedure is to submit a standard type *body plan* to the patentees, who then modify it in the light of their patents, without disturbing any other characteristics of the ship, such as stability or trim, or may be flow of water round the stern. The altered form is guaranteed to provide an improved efficiency of propulsion of some 5 per cent. or more. The reader may object, that in spite of the foregoing explanation, he is still unable to discern wherein the Maier form differs from an ordinary form (apart from the shape of the bow). To which the answer is, that the difference becomes apparent only after a critical analysis of the shape and area of the forward sections. It is not, in the popular sense of the term, a visual difference.

Another idea worthy of some mention, is the employment of U-shaped sections at the forward end. Comparing U sections with triangular sections (Fig. 85), it

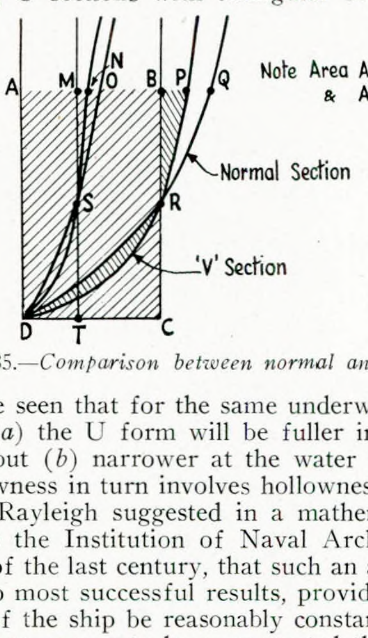


Fig. 85.—*Comparison between normal and V-shaped sections.*

will be seen that for the same underwater cross sectional area (*a*) the U form will be fuller in the region of the keel, but (*b*) narrower at the water line. This relative narrowness in turn involves hollowness of the water line. Lord Rayleigh suggested in a mathematical dissertation before the Institution of Naval Architects towards the close of the last century, that such an arrangement should lead to most successful results, provided the draught and trim of the ship be reasonably constant. Admiralty constructors appear to have supported the idea at the time. It would seem that the condition of relatively constant draught and trim, to which should be added speed, provides the crux of the idea. A hollow water line which might prove most helpful in one set of conditions, might become a dangerous "shoulder" if those conditions be changed, and it is suggested therefore that the general principle lacks flexibility, and is not suitable to be applied to the problem of the slow-speed cargo steamer, subject to wide variations of draught, trim and weather conditions. It is claimed, with much justification, that U-shaped sections provide a ship which will lift more easily to a head sea, and one which, may be, is less likely to become a victim of pounding.

Although not strictly relevant to the work of this chapter, it is proposed to refer to the *Isherwood Arcform.* In this case the characteristic is the employment of a *circular midship section.* Isherwood argued that instead of devoting so much time to the improvement of the ends of the vessel, it might be well to give a little consideration to the middle, or main body as it were. The employment of a circular midship section involves the acceptance of a lower mid section area coefficient. If for a given draught the displacement is to remain the same, then the ends of the ship must be filled out. Provided the length on the water line is generous, fuller underwater form forward and aft is not regarded as detrimental but rather as advantageous, and most remarkable claims have been made in respect of the propulsive efficiency of the "Arcform". Suggestions have been made in some quarters that a vessel with a circular section would show a predilection towards heavy rolling, but those who have served in this class of vessel will know to what extent the criticism is justified. There are some reasons why the reverse might be the case, *e.g.* the circular section implies tumble home, which is conducive to easy rolling.

Structural Arrangements.

In reviewing the structural arrangements at the fore end of the ship, it has been thought advisable to deal only with the simple straight vertical stem type, and to discuss only such features as would appear in a general cargo carrier built to the requirements of one of the classification societies.

(*a*) All vessels require to be fitted with a *collision bulkhead* situated not less than 5 per cent. of the vessel's length from the stem. (This distance is measured on the load water line, from the fore edge of stem). The bulkhead is to extend to the uppermost continuous deck, even though, as in the case of the shelter deck type of vessel, other watertight bulkheads stop at the deck next below.

(*b*) The double-bottom construction will terminate at the collision bulkhead.

(*c*) Special strengthening arrangements are required in respect of (1) panting, (2) pounding.

(*d*) The stem bar must be strongly connected to the keel and vertical centre girder.

(*e*) A chain locker will be fitted either just before or just abaft the collision bulkhead.

(*f*) The capstan and "Hawse" pipes should be arranged in such a way as to make the passage of the cable from hawse pipe via capstan or windlass to chain locker as easy and direct as possible.

The stem bar is a rectangular bar of wrought iron or mild steel about 10in. × $2\frac{1}{2}$in., made usually in two pieces and scarphed and riveted together. The scarph has a length equal to 9 times the thickness of the bar. In the case quoted for example the length of scarph would be 1ft. $10\frac{1}{2}$in. The lower end of the stem bar is hammered out and rabbeted to receive the forward plate of

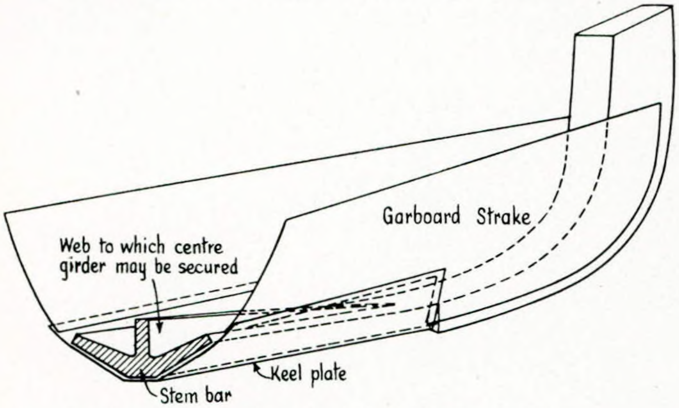


FIG. 86.—*Attachment of keel plate to stem bar.*

the keel (Fig. 86). In most cases a web is worked to which the intercostal vertical centre plates fitted on the fore-side of the collision bulkhead may be riveted (Fig. 87).

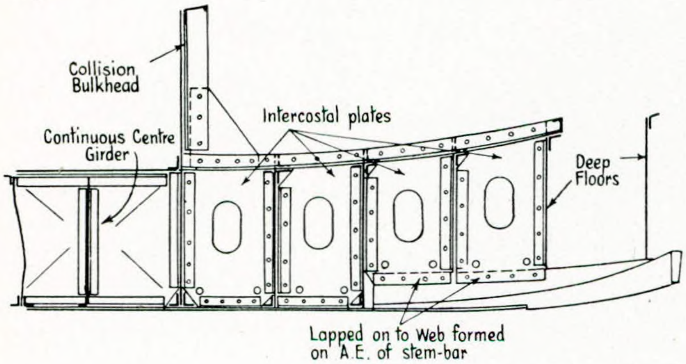


FIG. 87.

Fig. 88 typifies a general arrangement profile for the fore end of the vessel, indicating the positions of the bulkheads and decks, extent of double bottom, etc. Quite obviously there will be many differences as between ship

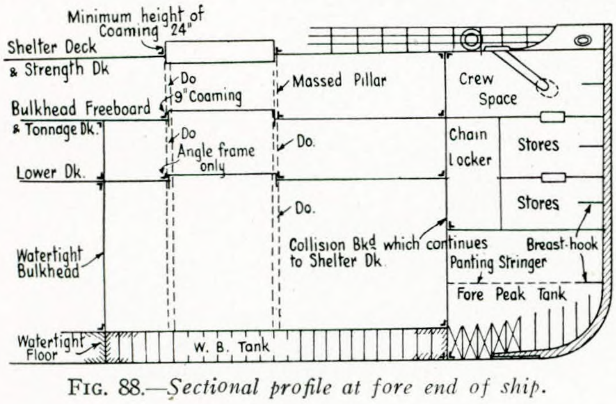


FIG. 88.—*Sectional profile at fore end of ship.*

and ship, for example in a vessel of the *"three island type"* there would be a high forecastle, which in this case has been omitted. Again in another ship the chain locker might extend to the bottom of the vessel, or as already noted it might be situated abaft the collision bulkhead. These differences, however, make little alteration in the general principles of stiffening and protection needed, and which will now be discussed.

The *collision bulkhead* is required to be of a higher standard of strength than in the case of a hold bulkhead. In the event of damage forward it might be called upon to withstand considerable water pressure for a prolonged period, and in a serious case (unless it is decided to tow the ship stern first) there would be the additional pressure caused by the vessel forcing its way through the water. For these reasons the classification societies require the vertical stiffeners to be spaced not more than 24in. apart. They are to be bracketed top and bottom, and of the same size as required for ordinary bulkheads. The plating also is to be retained of the same thickness as for ordinary bulkheads. The riveting of seams and butts will for the most part be single, although the bottom strake or two may be double. The rivet spacing is of course watertight, *viz.* $4\frac{1}{2}$ diameters centre to centre. The boundary angles are to be *double,* or if a single bar be used the bulkhead *flange* is to be deep enough to take a double row of rivets.

Since the bulkhead has to undergo a complete water test (the compartments on the fore side being flooded and the head of water carried up to the highest deck),

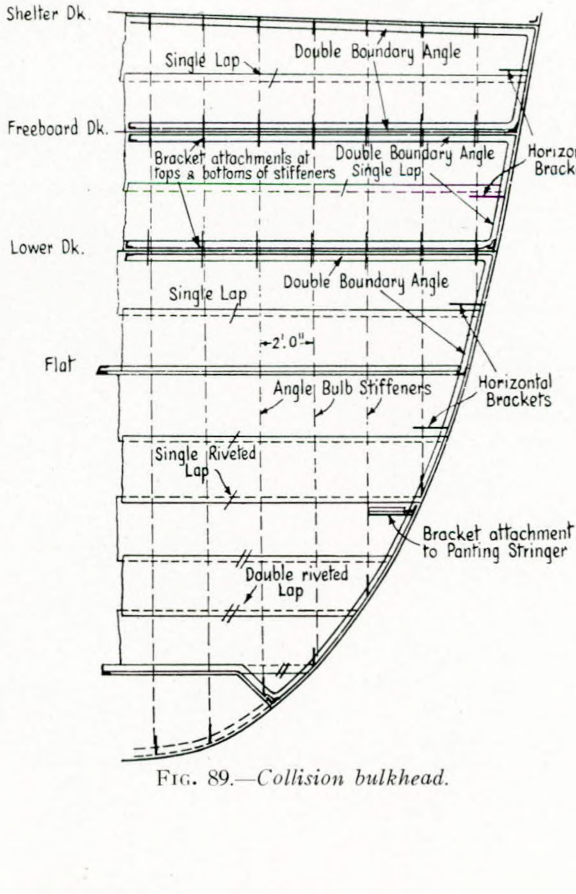


FIG. 89.—*Collision bulkhead.*

it is imperative that all plate edges and butts be absolutely unimpeded, so that caulking operations can be carried out as and when necessary during the test. Hence it follows that the vertical stiffeners should be fitted on the fore side of the bulkhead. No horizontal stiffening bars are required by law, but should they be fitted, they would be on the after side of the bulkhead (Fig. 89). Some horizontal bracketing will always be employed, and reference to the sketch dealing with panting will illustrate how this is done.

Panting Arrangements. The meaning of panting and the nature of the strains produced in the fore part of the ship have been discussed in an earlier chapter. It now remains to give some account of the structural precautions introduced to deal with this type of strain. Briefly they are as follows:—

(*a*) Tiers of panting beams are to be fitted below the lowest continuous deck. The tiers are to be 6ft. apart, and the beams constituting each tier are to be fitted to alternate frames.

(*b*) Stringer plates are to be fitted in line with the beams, to which they are securely connected by means of heavy horizontal brackets. At positions intermediate between the beams large vertical brackets connect the stringer to the frames (Fig. 90).

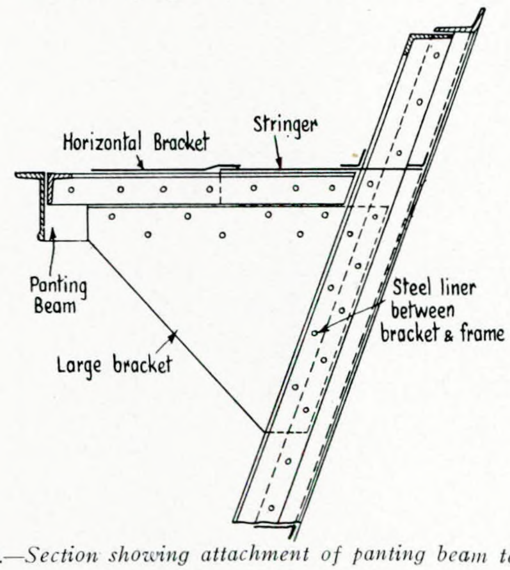


FIG. 90.—*Section showing attachment of panting beam to frame.*

(*c*) At the extreme fore end breast hooks unite the port and starboard stringers.

(*d*) At the collision bulkhead horizontal brackets tie the stringers to the bulkhead, and to avoid discontinuity brackets are also fitted abaft the bulkhead. The general scheme will be appreciated by studying Fig. 91.

(*e*) It was the practice at one time to fit a number of heavy web frames at every 4th normal frame space, and a system of stringers *abaft* the collision bulkhead for a distance of $7\frac{1}{2}$ per cent. of the vessel's length. This arrangement however leads to considerable broken stowage, and as a result of the representations made by shipping companies (particularly those engaged in carrying frozen cargoes, where the boxing off and insulation of webs, beams and stringers makes very serious inroads into the cargo space), the classification societies now agree to a somewhat

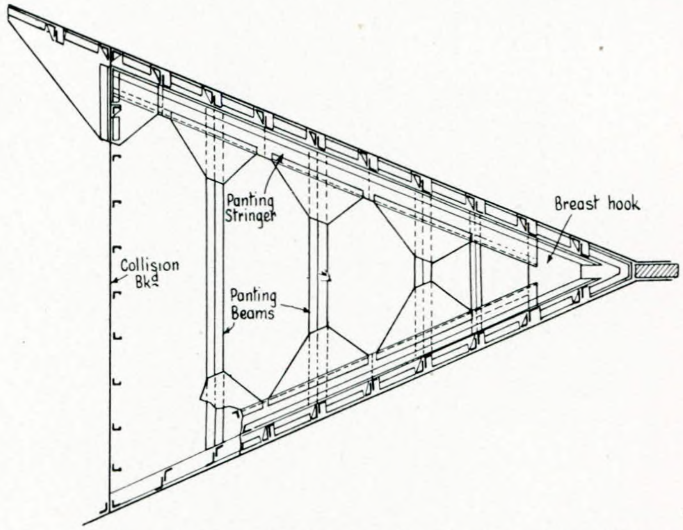


FIG. 91.—*Plan of fore end showing details of panting stringer and beams.*

simpler and less obtrusive scheme. So far as stiffening abaft the bulkhead is concerned they are prepared to entertain proposals for a modified arrangement of frame spacing and scantlings, provided it can be shown that the stiffness thus provided is comparable with that which would be obtained under the old regulations. In this way No. 1 hold becomes a relatively clear uninterrupted space, which is easily insulated, and eminently suitable for the carriage of frozen cargoes.

Pounding also has been defined and discussed in a previous chapter. The necessary additional stiffening required to combat these effects is as follows:—

(*a*) The two strakes of bottom plating on either side of the keel have their midship thicknesses maintained to the extreme fore end of the ship.

(*b*) Solid floors are fitted at every frame position from a point one-fifth of the vessel's length abaft the stern up to the extreme fore end.

(*c*) The floor plates forward of the collision bulkhead are made some 50 per cent. deeper.

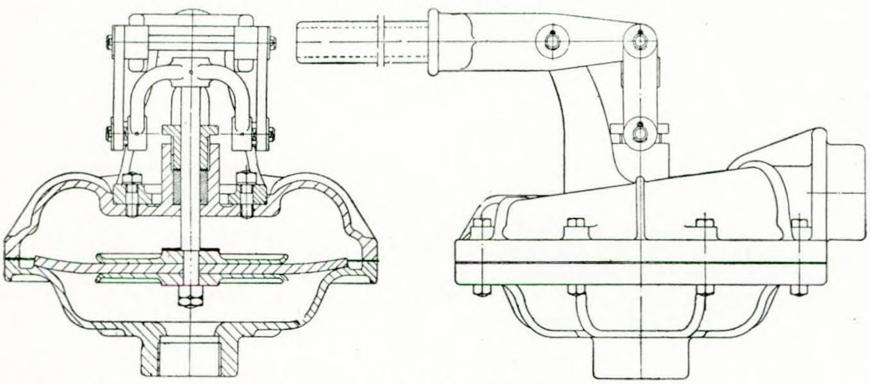
(*d*) If thought necessary additional side girders could be introduced.

(*e*) Special care should be given to the riveting and attachments of the frames to the side of the double bottom. Gusset plates should be fitted at every frame position (see chapter dealing with double-bottom construction).

Abstracts of the Technical Press

A Valveless Diaphragm Pump.

A pump of a new type, suitable for handling oil fuel, as well as for pumping bilges and similar work, is being produced by Tecalemit, Ltd. The unusual feature of this pump, which is of the diaphragm type, is that the diaphragm acts also as a valve, and there are no other valves in the mechanism. As will be seen from the drawings, the pump is of simple construction. The main casing is divided by a horizontal diaphragm into a suction chamber beneath and a delivery chamber above. The centre of the diaphragm is clamped between two discs carried on a vertical spindle, which is moved up and down by the operating handle. Assuming the centre of the diaphragm to be depressed to its lowest position and commencing its upward motion, the depression set up in the suction chamber will cause the outer edge of the diaphragm to be drawn tightly on to the annular abutment surface upon which it rests. Continuation of the upward movement causes the liquid to flow up the suction pipe into the lower



General arrangement of the new Tecalemit diaphragm pump.

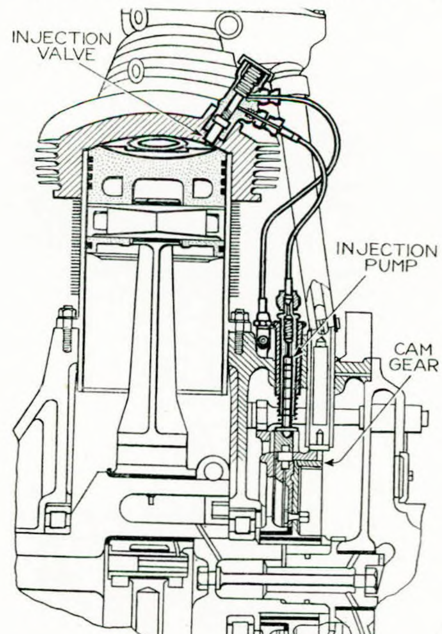
chamber, whilst the liquid above the diaphragm in the delivery chamber is forced outwards through the delivery pipe. On the downward stroke the edge of the diaphragm is raised from its seating and the liquid already in motion continues to flow past the edge of the diaphragm through the pump. Provided the pump is not operated too slowly, its action maintains continuous motion in the column of liquid with an intermittent acceleration at each stroke. It is claimed that the pump will suck from a depth of 5ft. without a foot valve, whilst if a foot valve is provided a suction of 15ft. is possible. The pump will deliver to a head o 15ft. At present, three sizes are available: S.J.5, which has a capacity of 5 gallons per minute; S.J.8, with a capacity of 10 gallons per minute; and S.J.10, with a capacity of 15 gallons per minute. The two smaller sizes are provided with oil-resisting diaphragms; the largest is supplied with a diaphragm suitable for water only.—*"The Motor Boat", Vol. LXXIV, No. 1,879, January, 1941, p. 10.*

The Diesel Oil Fuel Question.

It seems likely that, in the future, owing to the variation in the demand for different qualities of fuel—from aviation petrol to boiler oil—the oil industry will produce fuels for Diesel-engine operation of lower specification than those which have hitherto been available, and sooner or later users will have to reckon on the possibilities of utilising heavier residual oils than in the past. A great deal of work is being carried out in this connection in the United States, and the greatest influence on the problem has been exerted by the United States Maritime Commission insisting that all the motorships built to its order—and they represent a tonnage of nearly half a million—shall have machinery capable of running on relatively low-grade residual oil such as has not been utilised with European motorships. This demand has extended to the running of auxiliary Diesel engines in the ships in question, and as these auxiliary units are, to all intents and purposes, similar to the stationary plants built by the manufacturers supplying them, these manufacturers have had to modify their designs to comply with the specifications. Hence we find that many American oil engines of from 200 b.h.p. to 450 b.h.p., running at speeds of 350 r.p.m. to 600 r.p.m., are being designed to run on these low-grade fuels. Since the engines are similar to the stationary plants made by the same manufacturers, it is clear that sooner or later large numbers of American stationary oil engines will be designed to comply with the specification of the United States Maritime Commission. The manufacturers will thus be able to offer their users engines guaranteed to run on cheaper and lower-grade fuels than those required for the British oil engines with which they are in competition. We, therefore, recommend British oil engine manufacturers to obtain the necessary specifications of the United States Maritime Commission for the oils upon which the engines in question must run. It will not be difficult to make the alterations needed to enable many British engines to run on heavier residual oils, but a certain amount of experimental work is required, and this should be carried out whilst the war is on. —*"The Oil Engine", Vol. VIII, No. 93, January, 1940, p. 222.*

The Guiberson Aero Diesel Engine.

According to *Automotive Industries,* the Guiberson aero Diesel engine has been awarded a certificate of approval by the



A sectional view of one cylinder of the Guiberson Diesel.

U.S. Civil Aeronautics Board and is now in course of production for lightweight Army tanks and other purposes. The engine is a 9-cylr. radial air-cooled unit with a cylr. bore of 5⅜in. and a piston stroke of 5½in. rated at 250 h.p. at 2,200 r.p.m., but claimed to be capable of developing a maximum of 265 h.p. at 2,250 r.p.m. Solid injection is employed and the compression ratio is 15:1. The specific fuel consumption is stated to be 0·395lb./h.p.-hr. at 85 per cent. rated speed and full power, and 0·415lb./h.p.-hr. at rated speed and rated power and also at 60 per cent. rated speed and full power. The fuel pumps, arranged radially at the front of the engine within the cam-gear housing, are operated through the same gearing as the cam ring. Fuel is delivered to these pumps by an aircraft fuel supply pump at a pressure which can be regulated from 4 to 8lb./in.^{2} in the fuel duct. The engine is fitted with pressure lubrication, the oil pressure being maintained at 85 lb./in.^{2} by an engine-driven pump. The dry weight of the complete unit is 616lb.—*"Flight", Vol. XXXIX, No. 1,671, 2nd January, 1941, p. 9.*

New Combination Waste-heat Boiler and Silencer.

Among the exhibits shown by the makers of Maxim silencers at the Thirty-sixth Annual National Motor Boat Show recently held in New York, was a combination waste-heat boiler and silencer of novel design. No tubes whatever are used in its construction, the heat being recovered by a simple arrangement of angle irons which form an extended heating surface. The device is a development from the well-known Universal silencer originated by the makers some years ago, and therefore no additional silencing unit need be employed. The boiler may also be combined with a spark arrestor, if required, and takes up little more space than the ordinary form of exhaust muffler. It may be operated dry over long periods without injury.—*"Motorship and Diesel Boating", Vol. XXVI, No. 1, January, 1941, p. 56.*

Marine Boilers—Their Troubles and Maintenance.

This paper is included in the second edition of the Institute's handbook *The Running and Maintenance of Marine Machinery,* and deals with the operation of all types of boilers in general use afloat, including Scotch boilers, water-tube, thimble-tube and high-pressure boilers. Some of the common troubles experienced with Scotch boilers are enumerated and the various methods of making good the defects are briefly described. Reference is also made to evaporators and their working, and the paper concludes with some brief notes on welding repairs to boilers.—*Paper by Eng. Lt.-Cdr. H. S. Humphreys, R.N.(ret.), "Transactions of the Institute of Marine Engineers", Vol. LII, No. 12, January, 1941, pp. 223-234.*

Recent Developments in Marine Boilers.

Marine boilers have developed enormously in the last few years, not only in thermal efficiency, but in general design and, above all, in size. Whereas the "Queen Mary" has 24 Yarrow boilers, the "Queen Elizabeth" of about the same power (200,000 s.h.p.), which immediately followed, has only 12 boilers of similar type; but, of course, correspondingly larger, though occupying much less total space. The same evolution is noticeable in naval practice, as the latest destroyers have only two large boilers instead of the three or four units installed in earlier ships of comparatively recent construction. The outstanding advance in steam generation is that working pressures have doubled, as has the temperature at which steam is delivered. Pressures now average 450 to 500lb./in.^{2} and final superheat 750° to 800° F. Practically every marine boiler now incorporates a superheater, and in addition air preheaters are usually installed to feed the furnaces with hot air in order to increase the combustion temperature, although the latter has to be kept within certain limits to prevent damage to the furnace fittings and brickwork through overheating. About 250° F. of preheat is usually considered adequate, but 300° to 350° F. may be used in oil-fired boilers where there are no firebars or mechanical grates to complicate matters. The higher pressures quoted refer only to water-tube boilers, while the so-called Scotch boiler—which still retains a high degree of popularity—is limited by its form to working pressures not exceeding 250lb./in.^{2}. The Scotch boiler has, however, undergone a radical change and improvement in the design of the back-ends, as well as a simplification in construction, in the type known as the Howden-Johnson boiler. The latter is virtually a single-ended boiler with its combustion chambers and water-backs removed, and having its flat back shell coincident with the back tube-plate, thus forming a plain shortened "cheese" in which the furnaces and smoke tubes go right through from end to end. This results in a great simplification in construction, as a common combustion chamber is formed by a close cage of bent water tubes, the top ends entering the back-plate just above the stack of smoke tubes, and the lower ends entering below the furances. The cage is enclosed by a casing with suitable brickwork, and the combustion gases are thus deflected back through the return smoke tubes to the uptake. Considerable weight (about 25 per cent.) and space are saved by the adoption of this type of boiler and, as the rating may be at least double that of the older type, correspondingly fewer units are required. Water-tube boilers as now made with every pressure part truly cylindrical, are eminently suitable for the higher pressures employed and can indeed be stressed considerably higher than as designed without fear of distortion or failure. For the same reason, the standard combinations of steam and water drums with connecting stacks of small-bore straight tubes can be made with largely increased overall dimensions, the working stresses in the materials being well within the prescribed limits, and, in any case, considerably lower than those in the machinery, shafting, etc., which, of course, are not subject to the same fluctuations of temperature. These developments, while not necessarily improving the thermal efficiency of boilers, have resulted in immense savings in total weight of material and water, which, taken in conjunction with the saving of space occupied, are of paramount importance in general ship design. The water-tube boiler is likewise undergoing modification in fundamental design. The Johnson water-tube boiler, for instance, has only one water drum, connected by bent tubes to the steam drum above and thus forming a totally enclosed furnace. The Lewis single-drum boiler, which has already done well on land, goes a step further in that there are no water drums or lower headers whatever.—*J. Hamilton Gibson, O.B.E., M.Eng., "The Shipping World", Vol. CIV, No. 2483, 15th January, 1941, pp. 78-80.*

American Marine Diesel Engines.

Brief particulars of 482 oil engines for marine propulsion and auxiliary purposes are set out in a table arranged in alphabetical order according to the names of the various makers. The great majority of the models enumerated are low-powered high-speed units of under 1,000 h.p. operating on the four-stroke cycle. Of the 27 manufacturers named, only nine appear to produce Diesel engines of over 1,000 h.p., and of these only two—the Hooven, Owen, Rentschler Division of the General Machinery Corporation and the Nordberg Manufacturing Company—seem to specialise in the construction of high-powered units. The first-named makers' double-acting 2-stroke engines of from 3,600 to 9,000 h.p. appear to be the only ones listed suitable for direct drives, their speeds varying from 100 to 171 r.p.m. The table does not include any particulars of Sun-Doxford engines and the Sun Shipbuilding & Dry Dock Company is omitted from the list of marine Diesel-engine manufacturers.—*"Motorship and Diesel Boating, Vol. XXVI, No. 1, January, 1941, pp. 71-80.*

Mexican Motor Tankers Built in Italy.

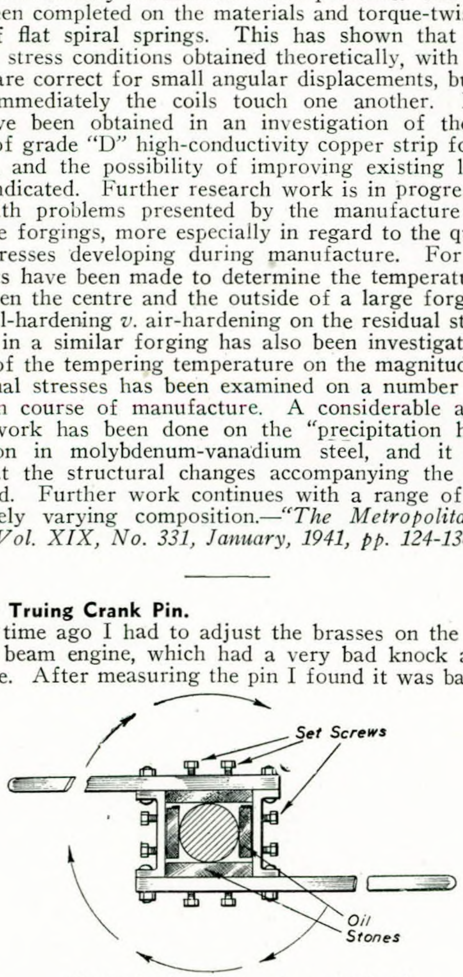
The Ansaldo shipyard has completed the two motor tankers "Poza Rica" and "Panuco" for the Mexican Oil Distributing Company, of Mexico City, the first being delivered in October and the second just having finished her trials. A third and similar vessel, the "Minatitlan", was launched in December. The main dimensions of these tankers are about 465ft. in overall length, with a beam of 62½ft. and a depth of 32ft., the gross tonnage being 7,800 and the d.w. capacity 10,900 tons on a mean draught of 22½ft. The propelling machinery of each vessel consists of a six-cylinder double-acting two-stroke Fiat Diesel engine with a cylr. diameter of 600 mm. and piston stroke of 1,100 mm., having a normal output of 2,400 b.h.p. at 125 r.p.m. and intended to give the ship a service speed of 12 knots. It is reported that all three sets of engines developed 3,500 b.h.p. during their trials.—*"The Nautical Gazette", No. 1, January, 1941, p. 26.*

Metropolitan-Vickers Research Work in 1940.

Apart from a considerable amount of research work carried out in the laboratories of the firm's research department during the past year, further investigation of the creep properties of metals has been continued, and, in addition, theoretical analyses have been made in regard to certain aspects of the creep phenomena. Included in the general work of creep testing, checks have been made on various types of moly-vanadium steel forgings of different sizes. An investigation of the effect of additions of chromium on the creep properties of moly-vanadium steel is still proceeding, but it may be said that whilst the additions appear to have advantages over short periods, this gain disappears when long periods, comparable with service, are considered. Work is also in progress to determine the creep properties of certain austenitic steels. Following on the theoretical investigation of the creep behaviour of steam pipes subjected to initial bending or torsional deformations, tests have now been completed on small steel tubes under constant internal pressure, and with superimposed constant bending or torsional deformations. These tests have qualitatively confirmed the results of the theoretical work, and indicate that decline of the initial bending or torsional stress is accelerated by the internal steam pressure. An investigation has been completed on the materials and torque-twist characteristics of flat spiral springs. This has shown that the most favourable stress conditions obtained theoretically, with the outer end fixed, are correct for small angular displacements, but become modified immediately the coils touch one another. Important results have been obtained in an investigation of the thermal softening of grade "D" high-conductivity copper strip for generator rotors, and the possibility of improving existing limitations has been indicated. Further research work is in progress in connection with problems presented by the manufacture of large high-tensile forgings, more especially in regard to the question of thermal stresses developing during manufacture. For example, experiments have been made to determine the temperature difference between the centre and the outside of a large forging. The effect of oil-hardening *v.* air-hardening on the residual stress after tempering in a similar forging has also been investigated, whilst the effect of the tempering temperature on the magnitude of residual internal stresses has been examined on a number of actual forgings in course of manufacture. A considerable amount of research work has been done on the "precipitation hardening" phenomenon in molybdenum-vanadium steel, and it has been shown that the structural changes accompanying the effect are complicated. Further work continues with a range of alloys of progressively varying composition.—*"The Metropolitan-Vickers Gazette", Vol. XIX, No. 331, January, 1941, pp. 124-130.*

Method of Truing Crank Pin.

Some time ago I had to adjust the brasses on the crank pin of an old beam engine, which had a very bad knock at the end each stroke. After measuring the pin I found it was badly worn;



Method of truing up a crank pin.

in fact, you could say it required a new pin fitting. This being a big job, I decided to devise some means to make the present pin workable. It had been neglected as regards adjusting the brasses and the pin was slightly oval, therefore I made the appliance as shown in the sketch. It consists of four fine oil stones enclosed in a frame of set screws for setting up the stones. When mounted on the pin I oiled the stones turning the frame round with a slight side movement from one end of the pin to the other, gradually adjusting the screws. This gave a perfect finish to the pin and, when tested, was found to be perfectly round.—*G. M. (Bolton); "Practical Engineering", Vol. 3, No. 54, 1st February, 1941, p. 46.*

New American Merchant Shipbuilding Programme.

The 200 additional merchant ships to replace tonnage lost in the war to be constructed under the $350,000,000 shipbuilding programme announced by President Roosevelt on the 3rd January, are, it is reported, to be cargo vessels of about 7,500 gross tons, with a length of 425ft. They will have oil-burning water-tube boilers and a sea speed of 10 to 11 knots, simplicity and ease of construction being the main feature of their design. The vessels will, therefore, not be fitted with the elaborate technical equipment of the ships built for the U.S. Maritime Commission.—*"The Nautical Gazette", Vol. 131, No. 1, January, 1941, p. 21.*

Dynamic Transmission and Reverse Gear.

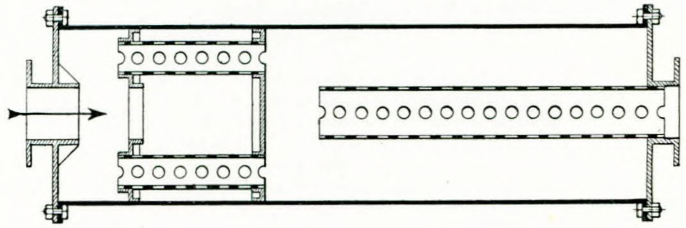
The dynamic marine transmission and reverse gear fitted to the Waukesha-Hesselman oil engines exhibited at the recent National Motor Boat Show in New York, is a product of a well-known American firm manufacturing power-absorption dynamometers, infinitely-variable transmission drives and electric clutches. As applied to a marine engine, it is a development of the principle employed for the constant-speed drive of the Pullman car air-conditioning equipment, in which it is used for driving the air compressor from the car axle, maintaining a constant speed of the compressor regardless of the axle speed. In this drive there is a controlled slip between the engine-driven rotating member consisting of a steel ring with wound pole-pieces bolted to its inner periphery, and an inner rotatable member similar to a squirrel-cage induction motor with short-circuited copper conductors embedded in it. By sending a direct current through the windings of the outer member, induced currents are set up in the driven ring, which magnetise it and increase or decrease the magnetic drag between the two members, depending upon the intensity of the current flow in the windings of the outer one. In applying this principle to the operation of a marine reverse gear, two of these drives are employed. The flywheel of the engine carries a hollow drum, the inner circumference of which has the two sets of pole pieces with their windings attached to it. These revolve at engine speed whenever the engine is in motion and are the driving members. Two driven members to correspond with each of these driving members are placed within these revolving pole sleeves, one of them being mounted on a shaft piloted in the flywheel with the other end supported at the outer end of the gear box. The other unit is mounted upon a sleeve which rotates independently of the shaft in ball bearings carried within the housing On the end of this sleeve is a pinion which, through an idler gear, drives the propeller shaft in the reverse direction. With no current flowing through the field coils of the driving members there is no drag set up and no movement of the propeller shaft. To control this mechanism there is a control lever connected to a two-way switch by means of which the current can be sent through one or the other of the two wound drive members, depending upon which side—to the right or left—of the vertical "off" position the control lever is placed. When the control lever is moved to right or left a small amount of current is admitted to the windings of the drive member, setting up induced currents in the driven member and causing the shaft to rotate slowly. As the control lever is moved further towards the "full speed" position, more current—and finally a maximum current—passes through the field coils and the driven shaft picks up speed rapidly with less and less slip until it is transmitting the full load of the engine at approximately 98 per cent. of the engine speed. In this way it is possible to obtain an infinitely variable range of speeds with the electric drive alone, while further speed changes may be effected by operating the engine throttle as well. The reversal of the drive sets up no mechanical stresses except in the gear case itself, for from the moment that the current is cut off from the energising members there is no drag whatever at the engine. When the current is passed through the other set of windings the load is picked up gradually because there is no direct mechanical connection, and until the propeller shaft comes to a full stop and reverses, the output of the engine is used as a brake upon it, slowing it down to a stop and reversing it with imperceptible smoothness. The current required for operating the gear is taken from a 32-volt battery charged by an engine-driven generator, the maximum current required at full output being 30-35 amperes. Forward and reverse speed may be selected from any desired station in the ship by operating a small switch, and the speed can be controlled independently of the engine speed by means of a hand rheostat on the control. The range of infinitely variable speeds available by this means enables the vessel to be manœuvred at a speed so slow as to be almost imperceptible. The performance is claimed to equal the smoothness of any of the conventional forms of electric drive in regard to operation, but the efficiency is higher and the weight and cost much lower. The normal efficiency of the electric portion of the drive, including the excitation current losses, is about 96 per cent. at two-third of full engine speed and 95 per cent. at full engine speed. The elimination of all mechanical connections between the engine and the propeller is claimed to give the drive a high degree of flexibility and smoothness coupled with freedom from mechanical wear and tear. The largest engine in the Waukesha Motor Company's range to be equipped with dynamic transmission and reverse gear is the 285-h.p. Reliance model, a six-cylr. four-stroke unit developing its rated output at 900 r.p.m. — *"Motorship and Diesel Boating", Vol. XXVI, No. 1, January, 1941, pp. 82-83.*

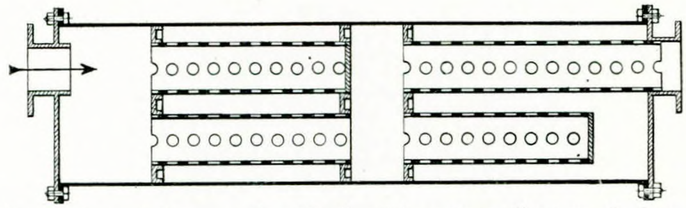
Automatic Steering Equipment for Small Craft.

A device which has been in use for several years on the Pacific Coast but was shown for the first time on the Atlantic Coast at the National Motor Boat Show recently held in New York, is the Photo Electric Pilot. Its outstanding feature is that it makes automatic steering available for even the smallest commercial or pleasure boats. The Photo Electric Pilot is essentially a device for use in connection with the ordinary hand- or power-operated steering equipment of a boat and operates through the medium of the regular steering wheel. It consists of a compass unit and a power unit connected by a flexible cable which permits a wide latitude in the relative positions of these two components. The compass unit comprises a standard magnetic compass, with a small light and photo electric cell so arranged that the slightest movement of the boat's head from the set course causes the motor of the power unit to be energized and the motor to turn the steering wheel in the appropriate direction to bring the boat back on its course. The power unit consists of a heavy-duty motor, operated from the boat's storage battery, which is connected to the steering wheel by a worm reduction gear, sprocket chain and clutch. The clutch allows instant engagement or disengagement of the pilot, so that automatic or hand steering may be used at will. The electric control elements, through which the compash controls the motor, are contained inside a weatherproof cabinet mounted on the power unit. The main switch for disconnecting the pilot from the boat's battery when not in use, is fitted on the front of the cabinet. To change over from hand to automatic steering it is only necessary to steady the boat on the desired course and throw in the clutch. To change to a new course the clutch is disengaged, the boat's head steadied on the new course, and the clutch thrown in again.—*"Motorship and Diesel Boating", Vol. XXVI, No. 1, January, 1941, p. 55.*

A New Type of Silencer.

The makers of Burgess silencers for I.C. engines have developed a new type of exhaust "snubber" which differs from the conventional or resonator form of silencer by making use of the principle of dissipating the exhaust gas through a series of perforated tubes in the silencer chamber. In the Burgess exhaust snubber, two types of which are illustrated in the accompanying diagrams, the object is to trap and dissipate the trouble-making slug of high-velocity, high-pressure gas before it reaches the atmosphere. It is claimed that this dissipating action takes





Two types of Burgess "snubber". The upper diagram depicts the pattern with concentric entry and exit, while the lower shows that with eccentric openings for awkward positions.

effect before the slug can do any harm, because the arrangement of parallel paths in the snubber offers no resistance to the flow of low-pressure, low-velocity gases. The combination of the chamber and tubes are said to change the rapidly fluctuating exhaust impulses into a steady, noiseless flow of gases into the atmosphere. The makers state that this type of silencer is equally suitable for use with two- or four-stroke engines and with any arrangement or length of exhaust piping.—*"Shipbuilding and Shipping Record", Vol. LVII, No. 6, 6th February, 1941, p. 132.*

Fire Extinguishing in War-time.

A new order, which came into force on the 1st January, 1941, supplements the 1932 rules relating to fire-extinguishing appliances in cargo vesels. The new rules stipulate that every foreign-going cargo vessel of 2,000 tons gross and upwards must be equipped with apparatus whereby smothering gas can be conveyed through a permanent pipe system into any compartment used for the carriage of cargo. In steamships, and in motorships with auxiliary boilers of a certain capacity, steam may be used as the extinguishing agent. In addition, fire pumps and hoses must be provided so that two powerful jets of water may be directed simultaneously into any part of the ship. Fire-extinguishing arrangements must be installed in the machinery spaces of all oil-burning steamships and oil-engined vessels, and it is also compulsory to provide equipment for dealing with incendiary bombs.—*"Fairplay", Vol. CLII, No. 3,011, 23rd January, 1941, p. 125.*

Electrically-driven Ships.

One direction in which it may be possible to increase the output of ships is by adopting the principle involved in the construction of fabricated ships and utilising the vast potentiality of the electrical manufacturing industry for the production of the propelling machinery, leaving the shipyards to deal with the construction of the hulls. The majority of the large electrical manufacturing firms in this country have their works situated inland and are equipped for the construction not only of electrical generators, motors and switchgear, but also of the prime movers, *i.e.,* steam turbines and oil engines, used for driving the generators. Indeed, the design and construction of steam turbines and heavy-oil engines forms, in many instances, an important part of the work undertaken in peacetime. In present circumstances the manufacture of electrical generating equipment, including the prime mover, is largely in abeyance, particularly as regards units of large and even moderate power, while the boiler manufacturers find themselves in a somewhat similar position. It is suggested, therefore, that if the shipyards could increase the output of hulls, the propelling equipment for the additional vessels thus produced could be constructed in its entirety by the electrical manufacturers and their associated boiler manufacturers when steam turbines are installed. Since such a policy would involve a consideration of the problem of which types of vessel are best suited to the application of the electrical drive, it will be of advantage to give briefly a general survey of the main features of the electrical system for the propulsion of ships. As indicated by the broad classification of tonnage under the headings of steamers and motorships, two distinct types of prime mover are being used for propulsion. The steamer can be further sub-divided into the turbine-driven steamship and that in which the triple-expansion reciprocating engine is used. The latter, although still very extensively employed in cargo ships of small and moderate tonnage—mainly because of its rugged simplicity—does not lend itself to the electric drive and need not be further considered here. The motorship almost invariably employs the Diesel engine as its source of power. This type of prime mover has two points in common with the steam turbine, the first, that its economic speed is greater than that of the propeller and the second, that it cannot be readily reversed. Both of these disadvantages are overcome by the electric drive. The economic speed of the steam turbine is very high, and in all modern turbine-driven steamships some form of speed-reduction gearing must be employed. The Diesel engine is generally designed to run at low speed so that it can be directly coupled to the propeller shaft, but since the most efficient propeller speed for ships of the type under review which usually have a speed of 12 to 15 knots, is about 100 to 120 r.p.m. this means that an engine of large size and excessive weight in relation to the power developed must be employed, both these factors having a very unfavourable influence on the economic operation of the ship. The ability to use engines having higher speeds of rotation, although it would necessarily involve the use of some form of speed-reduction gearing, would produce a considerable saving both in the weight and the space occupied by the machinery, resulting in an increase in the capacity of the cargo holds for a given size of ship. Thus, the electric drive using a high-speed prime mover direct-coupled to a generator, the current from which is fed to a motor direct-coupled to the propeller shaft, can be regarded, first and foremost, as a form of speed-reduction gearing, silent in operation, yielding an infinitely flexible coupling and having an overall efficiency only slightly less than that of mechanical gearing. It has already been mentioned that neither the turbine nor the heavy-oil engine can be readily reversed. In the case of the turbine having mechanical speed-reduction gearing, this involves the use of specially-constructed turbines running in the reverse direction to the main turbines, since the powers involved are too great to permit the use of reversing mechanisms in conjunction with the gearing itself. These astern turbines, often of sufficient size to develop more than 50 per cent. of the ahead power, may only be required for half an hour or so at the beginning and end of a voyage when manœuvring out of and into port, this representing a "load factor" of a very low order indeed, to say nothing of the uneconomical effect of the weight of the ship's astern turbines and the space which they occupy. With the marine heavy-oil engine, except those of small power in which gearing is employed not dissimilar to that used in a motor car, reversing is effected by the provision of special astern cams on the cam shaft, these being brought under the valve rockers when running astern. With the electric drive, reversing is effected by the use of simple switchgear which alters the connections on the propulsion motors so that in the one case the astern turbines are eliminated and in the other a far simpler engine is obtained. In addition, the reversing process is greatly facilitated by the use of the electric drive, so much so that, if desired, it can be effected directly from the navigating bridge without the intervention of the usual engine-room telegraph, a feature which makes for far more efficient manœuvrability. Another distinctive feature of the electrically-propelled ship is its high efficiency at all speeds between dead slow and full speed. With the fixed mechanical gearing of the normal turbine-driven steamship and with the direct-coupled Diesel engined vessel, speed regulation is necessarily effected at the engine itself. With both of these prime movers, however, efficiency falls off rapidly on either side of the maximum, which is designed to occur at the normal rated output or the engine. With the electric drive, particularly if two or more generators are employed to supply the maximum power, not only is a very high degree of flexibility obtained but the installation can be so designed that high efficiency is obtained at all the different speeds at which the vessel may be required to run when in service. This feature would prove to be of great value in wartime when vessels may be called upon to proceed in convoy, while the capability of being rapidly manœuvred if bridge control is fitted would obviously prove of supreme value in any emergency which might arise. Finally, attention may be directed to the superior prospects of electrically-propelled vessels after the war is over. Admittedly, the initial capital cost of such ships will be somewhat greater than that of similar vessels fitted with the more usual forms of drive, although it should be recognised that the cost of the machinery is but a portion of the cost of the ship as a whole. But when these vessels are called upon to run on a competitive basis with ships of this and other countries, it will be found that not only will the cost of operation as represented by the charges for fuel, lubricating oil and personnel be considerably less, but the earnings will be greater due to the relatively larger volume and weight of cargo which can be carried. The maintenance charges comprising the cost of overhauls, spares and so on will also prove to be in favour of the electrically-propelled ship. It does therefore appear highly desirable that the possibilities of the electric drive under the particular conditions of the present emergency should be investigated without delay. There can be no doubt but that its adoption would add substantially to the potential output of machinery for the propulsion of ships by utilising the productive capacity of inland manufacturing firms and it would also increase the output of the shipyards themselves since they could concentrate more particularly on the construction and fitting out of hulls. Moreover, apart from the special value of such ships at present when the provision of the maximum amount of tonnage is a matter of vital importance to our war effort, the electrically-propelled ships would prove an asset of particular value when the merchant fleets revert to their peacetime services.—*A. Regnauld, B.Sc., "Electrical Review", Vol. CXXVIII, No. 3,296, 24th January, 1941, pp. 282-283.*

Improved Wood.

One of the technical advances which was just beginning to become known in the years immediately preceding the outbreak of war, and which has doubtless made considerable progress since, is the manufacture and use of compressed resin-impregnated wood possessing greatly increased strength as compared with ordinary timber. The process of production resembles that employed in the making of ordinary plywood, except that the individual veneers are usually thinner, special adhesives, frequently of the bakelite type, are applied, and considerable pressure and heat are necessary for the formation of the board, which may be of practically any desired thickness. The density of the material increases, and the tensile and compressive strength along the grain is increased in similar or slightly greater ratio. The great advantage lies in the large percentage increase in the cross-grain and shear strength, resulting in a finished product of vastly improved homogeneity, almost approaching that of metal, while the ease of working is retained and the density remains low even compared to that of the light alloys. A wide range of timbers can be used, including varieties not normally regarded as of great commercial value, and the finished board is not merely a substitute for ordinary hard woods, but is suitable for applications where the employment of wood could not usually be considered. It should find numerous uses in ship construction and boat-building, enabling scantlings to be reduced, and it may be that when sufficient supplies are available complete superstructures will be built of the new material, with a large saving in weight.—*"Shipbuilding and Shipping Record", Vol. LVII, No. 4, 23rd January, 1941, p. 75.*

Gas for Small River Craft.

An interim report of the Gas Traction Development Committee embodies a suggestion for the utilisation of methane (CH_4) as fuel for small craft. It could be carried either in compressed form, or, more attractive still, in the liquified state. The gas is obtainable from collieries, coke ovens, or sewage farms, and is also available in the natural state in large quantities in certain parts of this country. Methane can be used for the operation of internal-combustion engines as an alternative to petrol, coal gas, or producer gas, and the report includes an estimate of the cost of compressing, which, taking 140 cu. ft. as equivalent to a gallon of petrol, works out at 3·8d. per gallon equivalent. It might even be possible to carry cylinders of compressed gas or bottles of liquid gas. — *"The Engineer", Vol. CLXXI, No. 4,437, 24th January, 1941, p. 72.*

Storage of Coal Under Water.

It is known that the storage of coal under water is resorted to in a few isolated places, primarily for safety's sake, and occasionally to mitigate the dust nuisance; however, there is no systematised information available on the practice and consequences of under-water storage. By the courtesy of a Durham firm of colliery proprietors, the Fuel Research Board have been able to record an observation bearing on the deterioration of coal under water. At the Lambton coke works there is a spoil bank of pit and washery refuse abutting on a pond, on which it forms a bank some 200 yards long. The bank also contains good coal and some of this coal, which had lain for at least 10 years below the level of the pond, which does not vary, was taken up and examined. There is, not unnaturally, no analysis of fresh coal actually corresponding to the original coal, but it has been possible to make a comparison of the characteristics of the old coal and the fresh as marketed at the present day. In order to avoid any effect due to the contamination of the material, especially the old coal, by dirt or washing tailings, the analyses were conducted on fractions floating in a liquid with a sp. gr. of 1·4. The results of these observations appear to indicate that underwater storage of coal has the desirable effect of nearly or completely suppressing its deterioration. Since the old coal is perfectly preserved in caking powder, it is likely that no other deterioration would be detectable. It may be mentioned that this particular type of Durham coal is usually susceptible to the deleterious effects of exposure.—*"The Iron and Coal Trades Review", Vol. CXLII, No. 3,804, 24th January, 1941, p. 82.*

Roller Bearings for Sheaves.

Among many other important applications of roller bearings is their use in connection with sheave blocks, particularly those of the multiple type, and a well-known firm of roller-bearing manufacturers are now stated to have developed a type made up of a double-row outer race and two single-row inner races or cones. The races and rollers are ground to precision limits, so that when the cones are assembled into the cup the front faces of the cones contact and the proper running clearance is provided. The front cone faces are slotted and chamfered to provide an entrance for the lubricant to the bearings, this feature being particularly advantageous in multiple-sheave blocks where it is required that the bearings be lubricated through the stationary pin. The bearings are designed to a minimum width and with an inside diameter large in relation to the outside diameter. As the unit is an anti-friction thrust bearing as well as a radial bearing, the sheaves may rotate freely without axial play and it is claimed that wear between the sheaves and side plates is wholly eliminated. — *"Shipbuilding and Shipping Record", Vol. LVII, No. 5, 30th January, 1941, p. 99.*

Impact Testing.

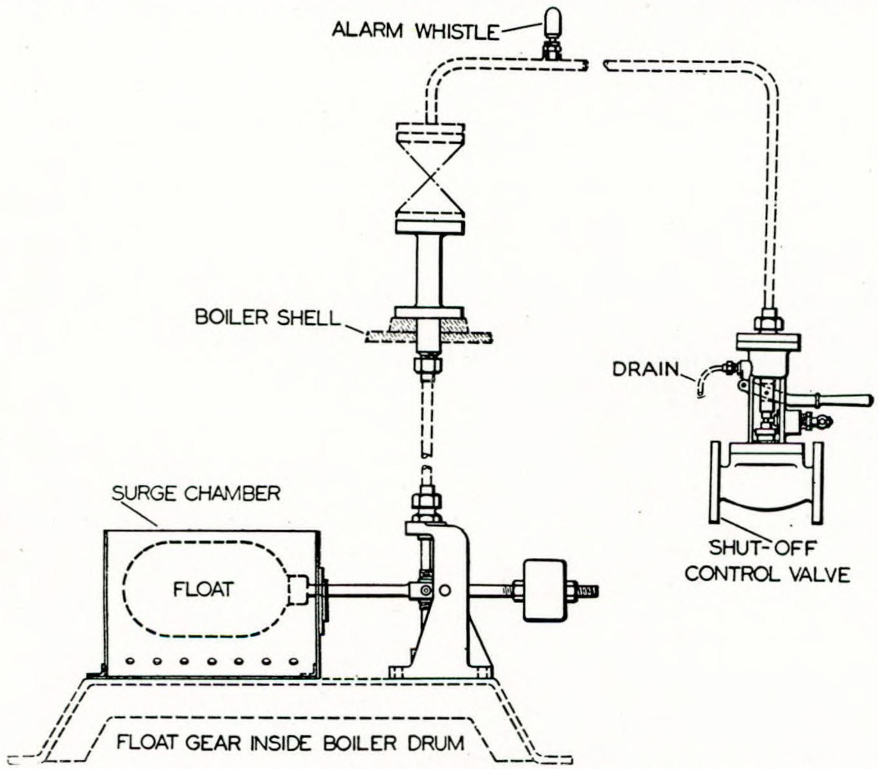
In comparison with the well-known and old-established tensile and bend tests which have so long been standardised for shipbuilding material, the method of testing according to which the specimen is broken by means of a hammer blow is a relatively modern procedure, but it has been extensively used for investigating the properties of deposited weld metal and of welded joints, particularly butt joints. In some degree it combines the functions of the straightforward tensile pull test, giving the ultimate tensile strength of the material, and of the bend test, regarded as a means for measuring the ductility, since a high-tensile which is not also ductile cannot be expected to give a good impact figure. The impact test is no less precise than the others mentioned, for the movement of the swinging hammer after the blow, which is noted, gives an exact measure of the energy required to fracture the specimen, depending on both the stress at rupture and the distortion. The value of the impact energy is sometimes regarded as the most important property of the material, and the best guide to its serviceability. A simple type of specimen can be used, requiring a minimum amount of material and machine, whilst the apparatus involved is likewise very simple and compact. Consideration might well be given to making the impact test the standard and unique routine acceptance test for shipbuilding steel, in substitution for the more lengthy and elaborate tests now required.—*"Shipbuilding and Shipping Record", Vol. LVII, No. 5, 30th January, 1941, p. 99.*

"Progress in Marine Engineering as Influenced by the Classification of Ships".

The paper bearing this title begins by a reference to ship insurance in Greek and Roman times, and then deals with the early days of Lloyd's, the publication of *Lloyd's List,* and the amalgamation of the *Underwriters' Green Book* and *Red Book* in 1834, to form the familiar *Lloyd's Register Book* in its present form. Early in the history of the Society in its reconstituted form, the regulations relating to steamers were very brief and did not call for the survey of the machinery by the Society's surveyors. The rules were based on government regulations then in force and merely stipulated that the machinery should be surveyed twice yearly by a competent master engineer, who was to sign a certificate describing the condition of the engines and boilers. A reproduction of one of these certificates is given in the paper. Under the headings of main propelling machinery, boilers, oil fuel, electricity, refrigeration and so on, the author outlines the developments which have occurred since these refinements were first fitted in ships, and how Lloyd's Register surveyors closely watched each new idea and only approved its use after satisfactory tests. The experience gained through surveys carried out in shipyards, engineering works, steelworks, ports and harbours all over the world led to the accumulation of a vast amount of information and data which proved of inestimable value for the formulation of new rules, or the amendment of existing regulations. Such information has frequently been freely placed at the disposal of all interested parties in the form of scientific papers read before the technical societies both at home and abroad. The paper includes a considerable amount of historical information not generally known to marine engineers, *e.g.,* that electric lighting was first installed in a steamship in 1879, and that experiments in the burning of oil fuel were carried out 70 years ago.—*Thirteenth Thomas Lowe Gray Lecture by Dr. S. F. Dorey, read at a general meeting of the Institution of Mechanical Engineers on the 24th January by W. D. Heck, summarised in "Shipbuilding and Shipping Record", Vol. LVII, No. 5, 30th January, 1941, p. 100.*

Low-water Safeguards and Alarms.

The Factories Act, 1937, has created a number of new regulations concerning steam boiler operation, among which is that requiring the provision of a suitable fusible plug or efficient low-water alarm for every water-tube boiler. Fusible plugs depend for their action upon the rise in temperature resulting from shortness of water, oil, or excessive scale deposits, and certain types of low-water alarm devices also incorporate fusible metal discs on which they depend for their operation. Other types employ a float which is actuated by the rise and fall of the water in the boiler (*see abstract on p. 69 of "Transactions", April, 1939*) and these devices may be fitted either inside the boiler drum or outside it. There are a number of such low-water alarms on the market, and Fig. 3 shows the arrangement of the Weir Mumford low-water alarm and oil-fuel control gear as designed for use in marine oil-fired water-tube boiler installations. The device depends for its action upon the movement of an internal float operating a valve through which steam is admitted to an alarm whistle and at the same time shuts off the oil-fuel supply to the boiler burners. When the oil-fuel supply is shut off it is impossible to re-start it until the water in the boiler has been restored to safe level, during which time the whistle continues to function. The float is of Monel metal and is located in a surge chamber to damp down disturbances caused by the movement of the ship. The needle valve admits steam to the small pipe shown leading out of the boiler, and in addition to sounding the alarm whistle, the steam acts on a piston integral with the oil-fuel shut-off valve, and causes the latter to close. Fig. 6 shows the arrangement of the Dewrance high- and low-water alarm, in which the operating weights B and C are suspended from the opposite ends of two independent balance beams H,



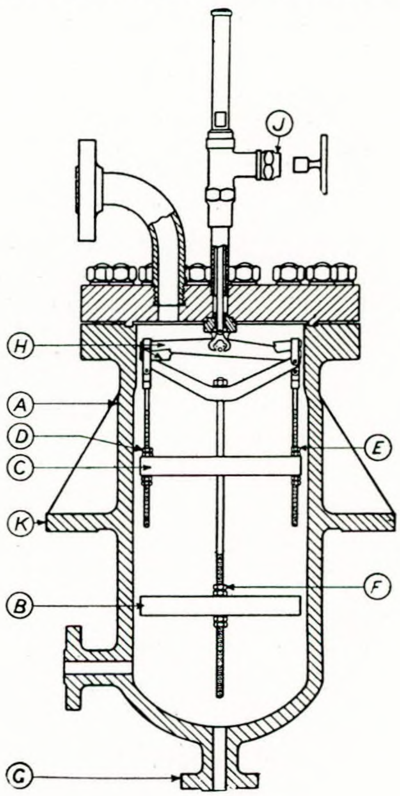


FIG. 3 (above).—*Diagrammatic arrangement of Weir-Mumford low-water alarm and oil fuel control gear* (Courtesy G. & J. Weir Ltd.)
FIG. 6 (above right).—*Dewrance high- and low-water alarm.*

arranged so that each balance beam has half the weight of B acting on it at one end and half the weight of C at the other end. The adjustment is such that when B is in water and C is in steam, the whistle valve is held shut. If, however, the water level rises to the weight C or falls to the weight B, the beams swing over and open the whistle valve. This type of alarm is particularly suitable for high-duty water-tube boilers, and the operation of the mechanism is unaffected by any surging of the water inside the drum.—*S. D. Scorer, "Boiler House Review", Vol. 54, No. 8, February, 1941, pp. 252-256.*

Safeguarding of Air Receivers.

Particulars of an explosion of a large receiver appear in the January issue of *Vulcan*, the house journal of the Vulcan Boiler and General Insurance Co., Ltd. The receiver in question was 67in. long and 25in. in diameter, with a shell thickness of $\frac{3}{16}$in. There were no longitudinal seams, and the dished ends were fire-welded to the shell and $\frac{1}{4}$in. thick. The certified working pressure was 100lb./in.^{2}, but the explosion occurred while the receiver was under pressure of only just over 75lb./in.^{2}, one end-plate being blown off without warning. The compressor was not capable of raising the pressure above 150lb./in.^{2}, so that the resultant stresses could not have been sufficiently high to cause fracture of the receiver. Inquiry into the accident showed that the end-plate had been forced from the shell at the weld for about 7$\frac{1}{4}$in. of the circumference, and that the shell itself had fractured for the remainder of the circumference, the break being from $\frac{1}{4}$in. to 1in. from the weld; this strip still adhered to the end-plate. The weld was of the ordinary standard accepted in fire welds. A metallurgical examination disclosed no evidence of a progressive fracture in any particular region, the indications being that the whole fracture occurred suddenly, commencing at the inner surface of the shell and proceeding in a brittle manner for about half the thickness of the plate, after which some indications of ductility were apparent. All the evidence went to show that defective welding was not a contributory factor to the failure, which was due to the application of pressure, much in excess of the compressor capacity, in the receiver. The conclusion arrived at was that the air from the compressor was at a temperature exceeding the flash point of the lubricating oil used, and that this hot compressed oil-laden air provided an explosive charge. Ignition might have occurred from a spark developed by friction in the compressor cylinder; from heated dust particles; or from fragments of carbon heated to a glow by the high temperature of the compressor discharge. While it is not suggested that the danger of such explosions is continuously present, since the proportions of air and oil vapour in the mixture may vary and the mixture may not always be capable of being fired, the risk is nevertheless by no means negligible. To combat it, it might be advantageous to fit a fusible plug in the compressor discharge pipe, the alloy of the plug being designed to fuse at a temperature below the flash point of the compressor lubricating oil. A suitable plug developed by the Vulcan Insurance Company is available in two sizes, for screwing into pipes below and above 2in. bore, respectively. The screwed portion is prolonged internally into a plain tube, sealed at the top with the alloy and grooved to increase the heat conductivity. Externally, there is a hexagonal cap, bored on the faces with holes communicating with the hollow interior of the plug. The latter should be screwed into the discharge pipe as close as possible to the receiver. Should the delivery temperature of the air approach the danger point, the alloy fuses and allows the air to escape, with an audible warning sound, through the holes in the cap of the plug.—*"Engineering", Vol. 151, No. 3,916, 31st January, 1941, p. 100.*

New Type of Large Fast Cargo Liners for U.S. Maritime Commission.

It is reported that the technical experts of the Maritime Commission are preparing designs for a fourth type of cargo vessel, to be known provisionally as the C-4 type, which will be the fastest cargo carrier ever seen in the foreign trade of the United States. The new ships are likely to be of some 10,000 tons gross with a guaranteed sea speed of about 20 knots and space for 12,000-13,000 tons of cargo, together with accommodation for at least 200 passengers. The dimensions of the new ships will, it is stated, be about 550ft. by 73ft., and the output of the propelling machinery at least 12,000 s.h.p.—*"The Journal of Commerce" (Shipbuilding and Engineering Edition), No. 35,253, 30th January, 1941, p. 2.*

"Effect of Some External Factors on the Performance of Single-screw Ships".

The paper bearing the above title is divided into three sections. The author points out that in predicting the performance of a ship, emphasis is usually laid on the hull resistance and on the quasi-propulsive efficiency as determined from self-propulsion tests carried out under still-water conditions. There are, however, other influences, some of which are reflected in the results of still-water tank experiments carried out with and without overload, while others can only be surmised or imperfectly gauged from extensive analysis of service performance. These two methods of dealing with the problem are interdependent and can sometimes be used as a check on each other. The author attempts to do this in Sections I and II of the paper, and devotes Section III to "Weather Effect", an aspect which does not readily lend itself to an experimental study.—*Paper by Alex. Kari, M.Sc., read at a meeting of the N.E. Coast Institution of Engineers and Shipbuilders, on the 31st January, 1941.*

The Recovery of Electric Lamp Caps.

It is an important point in the national salvage campaign that the valuable brass which can be recovered from electric lamps should not be wasted, and especially that the caps should, whenever possible, be used again for their original purpose instead of being melted as scrap. The glass bulb or other parts of the lamp except the cap have no value for re-use. Therefore, to save bulk and weight in transport, it is preferable to remove most of the bulb and other parts before packing caps for return to the factory, and this can be done easily and safely by anyone provided with a pair of gloves, an old file, a barrel or other receptacle for the rubbish, and suitable protection for the eyes. For the latter, a cheap pair of goggles will suffice, the only danger being the accidental entry of a small splinter of glass. The cap of each lamp to be disposed of should be grasped firmly in one hand, while the bulb is first sharply tapped with the sharp edge of a file to crack it and afterwards knocked off piecemeal reasonably close to the edge of the cap. Do not overdo this or one may damage the cap itself, which, if badly bruised or dented, cannot be re-used. It is not recommended that any attempt should be made to remove any part of the glass "foot" which protrudes from the inside of the cap, as the process is tedious, the amount of space saved small and the danger of damage to the cap considerable. On no account should any attempt be made to remove the neck of the bulb or the cement which attaches it to the cap. This requires special means and also removes the support which will otherwise sustain the cap in its transit back to the factory and prevent it from being damaged by its neighbours. The solder on the cap contacts should also be left as it is. Throughout the operation, all possible care should be taken to avoid denting or scratching the brass surfaces or cracking the black glass insulator which fills the butt end of the cap and carries the contacts. Caps treated in this way can safely be packed into barrels, wood or cardboard boxes (not sacks), of which the gross weight should preferably not exceed 2 cwts. per package, and returned to any branch of any member of the Electric Lamp Manufacturers' Association.—*"Foundry Trades Journal", Vol. 64, No. 1,276, 30th January, 1941, p. 68.*

Using a Small Lathe on Turbine-generator Repair.

An electric utility company had trouble with the bearings of a 12,500-kVA steam turbine-generator. Bearing surfaces on both ends of the shaft were badly scored and had to be remachined. Operators discarded the idea of removing the 22-ton shaft assembly and taking it to a machine shop because of the time and money involved. Work was further complicated by the fact that there was no lathe in any nearby shop large enough to swing the shaft and 8-ft. dia. turbine wheels. The engineer in charge of the job showed his skill and resourcefulness by using a small bench lathe to remachine the journals without removing the shaft assembly from the turbine. He took off the turbine bearing caps and bolted the bed and carriage of a 9-in. swing lathe to the turbine frame. The lathe-bed vee-ways were carefully aligned with the shaft. A special goose-neck cutting tool, forged to suit the job, was mounted in the toolpost of the lathe. An old hoist driven by a 5-h.p. motor was set up on the floor and geared to the turbine in such a way that it would drive the shaft at a suitable speed. After starting the turbine shaft, the forged cutting tool was fed along the scored journal surface by turning the lathe carriage handwheel. A smooth finish was obtained and a first-class job turned out quickly at low cost.—*N. D. Jackson (in "Power"), "Mechanical World", Vol. CIX, No. 2,822, 31st January, 1941, p. 87.*

The Future of Diesel-electric Propulsion in America.

According to Mr. H. C. Coleman, marine manager of the Westinghouse Electric Manufacturing Co. (U.S.), 1940 developments in connection with Diesel-electric propulsion indicate that the day of the low-speed engine is over, and that all future installations will make use of small light-weight medium speed engines. There are also signs that alternating current is likely to be adopted, wherever practicable, both for Diesel-electric propulsion and for ships' auxiliary machinery. Probably the most outstanding installation of the year was the Diesel-electric propelling machinery of the tuna clipper "Challenger" (*see abstract on p. 193 of "Transactions", December, 1940, for description*), which was the first American Diesel-electric ship to have a.c. transmission. Reports of operation since the "Challenger" was completed in June prove the soundness of the design and indicate that it has many advantages for this particular service. There had been considerable progress abroad in the last four years, Mr. Coleman said, in the application of a.c. machinery to Diesel-electric propulsion, but all these installations had been in large passenger or cargo liners and employed large medium-speed engines and direct-connected synchronous propulsion motors, so that they were of an entirely different type. Studies had been made and designs carried through for similar installations to be made in America, and it was believed that a.c. drive presented a practical means of utilising medium high-speed engines to drive a single screw in large ships in which d.c. transmission would be out of the question because of the cost and weight, and where frequent manœuvring was not required. He thought there would be an increase in the use of this type of propulsion in the next few years.—*"The Shippng World", Vol. CIV, No. 2,486, 5th February, 1941, pp. 155-156.*

Atlas Diesel Rotary Blower for Reversible Engines.

One of the latest Atlas Diesel developments is to introduce a rotary blower in place of their reciprocating pump hitherto employed as standard. The blower is gear-driven from the main shaft, and if only one is employed some form of reversing gear is necessary for the machine to act both ahead and astern. In order to avoid this complication, which has the effect of increasing the length of the engine and adds to the noise of the gearing, the arrangement shown in Fig. 6 is adopted. Two vane wheels are employed, one running for astern and the other in normal use for ahead operation. It is stated that the loss of power due to idling is less than the decrease in efficiency when a reversing blower is used. The blowers may have separate intakes, as shown in the left-hand diagram, or a common intake, as the right-hand view indicates. Referring first to the left-hand diagram, there are two pump housings (5, 6) having tapered passages (7, 8). The vane wheels (9, 10) are secured to a common shaft (11) and have air intakes (12, 13), respectively. Air is discharged under pressure through the conduits (14, 15), which communicate with the scavenging air receiver (17). From the small diagram on the left it will be seen that there is a flap valve (18), which closes one or the other of the conduits (14, 15), according to the passage of air. Assuming the left-hand vane wheel to be designed for ahead running, the flap valve would normally be horizontal and cover the orifice (15) from the astern blower. When reversal takes place, the valve is raised and the ahead blower conduit is isolated. In the case of the alternative arrangement shown on the right, the common air intake (23) is located between the blowers. The toothed gearing (19, 20, 21, 22) is mounted alongside the engine, whereas in the former design the gears are carried between the blowers, as the diagram indicates.—*"The Motor Ship", Vol. XXI, No. 253, February 1941, p. 387.*

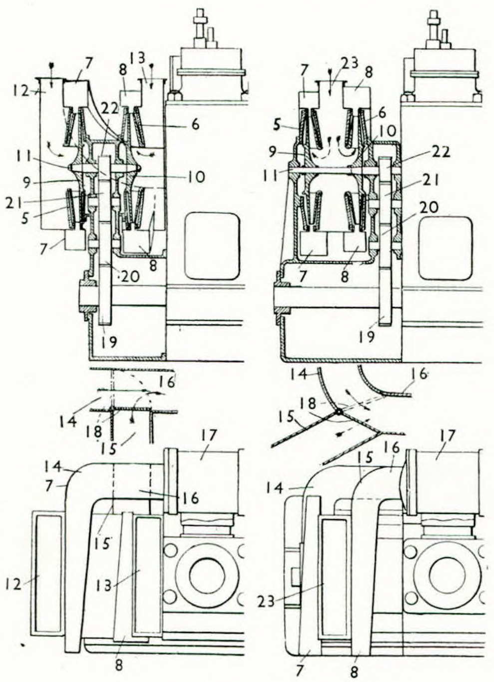


FIG. 6.

Success of Electric Couplings in U.S. Maritime Commission's Motorships.

The first of the Maritime Commission's vessels to be equipped with electric couplings, the "Mormacpenn" and her three sister ships, have now been in service for some time, and it is reported that the couplings are operating in a very satisfactory manner. Each ship has four electric couplings, each rated at 2,230 h.p. at 240 r.p.m., each coupling connecting a Diesel engine to a reduction gear which, in turn, is connected to the single propeller shaft. The great flexibility in manœuvring achieved by the use of the couplings has been frequently demonstrated. Two of the engines are run in the ahead direction and the other two in the astern direction, while the propeller is operated ahead or astern by energising the respective couplings. Thus, any order except "full ahead" or "full astern" can be complied with by the operation of a single lever controlling the coupling excitation, together with the appropriate changes of engine speed, without any consumption of starting air. By this method manœuvring while docking or passing through intricate channels can easily be carried out in the engine room as fast as the orders are rung down on the telegraph. — *"The Shipping World", Vol. CIV, No. 2,486, 5th February, 1941, p. 156.*

The Colour of Used Diesel Lubricating Oil.

Used oils from Diesel engines differ from new oils in that they contain certain products of decomposition and impurities. In a circulation system, agitation in the pump and passage through pipes and ducts promotes an intimate mixture of impurities and lubricating oil. When the operator shuts down the engine, the free water settles and collects at the bottom of the crankcase or sump. As the temperature drops, soluble impurities precipitate. Suspended foreign matter and sludge may settle extremely slowly and may be light enough to remain in suspension. As soon as the engine starts again, the settled impurities will be drawn into circulation once more and some of them will dissolve in the warm lubricating oil. Purification of Diesel-lubricating oil should aim at removing sludge and suspended impurities and at restoring the oil to its original condition. The purified oil will be darker than the original oil, the darker colour merely indicating the presence of harmless colloidal carbon and not poor condition or inferior oil.—*S. S. Hansen (in "Power") "Mechanical World", Vol. CIX, No. 2,823, 7th February, 1941, p. 104.*

Corrosion of Superheaters.

Apart from such corrosion as may occur in any part of a boiler installation due to oxygen or other impurities in the feed water, the superheater elements are also liable to peculiar forms of corrosion due to the decomposition of the steam consequent upon the very high temperature to which it is subjected. It is well known that one of the simplest methods of preparing hydrogen is to pass steam over red-hot iron, and circumstances may arise in the superheater tubes which likewise result in the decomposition of the steam. The oxygen thus set free actually tends to form a compact protecting film of iron oxide (Fe_3O_4), but any movement of the surface due to changes in temperature may set up molecular friction which breaks this film and exposes the iron to the direct corrosive effects of the hydrogen. Any magnesium chloride present may decompose, liberating hydrogen, but the use of tri-sodium phosphate leads to the precipitation of the magnesium chloride in the form of magnesium phosphate, which is harmless.—*"Shipbuilding and Shipping Record", Vol. LVII, No. 6, 6th February, 1941, p. 123.*

Butt-welding of Wires.

The process of butt-welding can be employed for the joining of tubes, wires and other similar sections by the use of special machines instead of electrodes to produce the necessary connecting fillet of metal, the resistance of the two sections in contact to the passage of electricity causing them to become sufficiently hot to weld together when the necessary pressure is applied. A well-known British firm of electrical manufacturers have recently produced a small machine of this type primarily intended for the butt-welding of wires, although a modification of the jaws which constitute the two electrodes makes it possible to weld other sections. The machine has a consumption of only 2·5 kVA and is capable of welding copper and aluminium wires from No.20 S.W.G. to No. 8 S.W.G., with an increase up to No. 4 S.W.G. in the case of steel. It is suggested that in the wiring of switchboards and those parts of the electrical distributing system on board ship where bare wires are used, butt-welding would prove superior to the ordinary method of jointing.—*"Shipbuilding and Shipping Record", Vol. LVII, No. 6, 6th February, 1941, p. 123.*

Synthetic Lubricating Oils.

A recent report of the Fuel Research Board contains a reference to current research work in connection with the production of lubricating oil from coal and coal products, and in some of the processes mentioned synthesis is being employed. The problem is being dealt with in four different ways, *viz.:* (*a*) by the chlorination and condensation of saturated hydrocarbons obtained by the synthesis of hydrocarbons from carbon monoxide and hydrogen, (*b*) the polymerisation of the liquid product obtained as above, (*c*) the polymerisation of unsaturated hydrocarbon produced by the carbonisation of coal, and (*d*) the hydrogenation of tars. Previous work has shown that oils can be obtained which are similar in properties to natural oils except that they fail in the oxidation test, but recent experiments, in which further hydrogenation treatment was given, indicate that with carefully selected starting materials a lubricating oil can be obtained equal in all respects to the best natural products.—*"Shipbuilding and Shipping Record", Vol. LVII, No. 6, 6th February, 1941, p. 123.*

Kort Nozzle Tugs for U.S. River Service.

Particulars of three similar twin-screw Diesel-engined river tugs, fitted wih Kort nozzles, appeared in a recent issue of the New York *Marine News.* The three vessels concerned, the "Ductilite", "Semet-Solvay", and "Victory", were all built in the Dravo Corporation's Pittsburgh yard at about the same time, which made it possible to rotate the use of welding jigs and positioning devices in a way closely approaching mass-production methods. The tugs are 135ft. in overall length, 27ft. in beam moulded, and 11ft. 9in. in depth at side, moulded, to raised deck; they have a draught of 6ft. 3in. forward and 5ft. 9in. aft, and a freeboard of 4ft. 4in. forward and 3ft. 6in. aft. The superstructure and hull of each vessel are of all-welded steel construction, the hull being built on the transverse-frame principle and divided into compartments by five main W.T. bulkheads extending to the raised deck. The sections for these were preassembled in an inverted position to permit a maximum of downhand welding. Longitudinal bulkheads on each side of the hull are made oiltight to form fuel-oil tanks in the wings. The propelling machinery of each tug consists of two sets of Cooper-Bessemer six-cylr., four-stroke Diesel engines developing a total of 760 h.p. at 310 r.p.m. Each engine drives a 5½-ft. propeller partially encircled by the tunnel design of the stern and fully enclosed by a modified Kort nozzle. It was anticipated that the Kort nozzles, in conjunction with the tank-tested hull lines, would increase the effective push developed by these tugs by 18 to 25 per cent. at normal towing speeds above that which would otherwise have been developed with the rated h.p. of the engines. Actually, the trials of the "Ductilite" more than fulfilled this expectation. The flanking tests were likewise highly satisfactory, and the manœuvring and handling qualities under the control of the ship's six rudders were found to be unusually good, while the free-running speed at normal power was 12 m.p.h. A comparison of the "Ductilite's" trial performance with that of the stern-wheel steam tug "La Belle", built in 1921, is significant. "La Belle", which has compound engines of 700 i.h.p., is 174ft. in length (26ft. being taken up by the wheel) and 28ft. 4in. in beam. Under bollard test conditions, the forward push of the "Ductilite" exceeded that of "La Belle" by 45 per cent., the backing power by 125 per cent., the forward steering power by 400 per cent., and the flanking power by 50 per cent. The fact that a screw-propelled tug is able to surpass a stern-wheeler in flanking and backing performance is noteworthy.—*"Lloyd's List and Shipping Gazette", No. 39,394, 12th February, 1941, p. 7.*

Portable Moisture Detector.

The degree of moisture content in wood, plaster, textiles, and other materials is often required by manufacturers and shippers, but until recently it has only been possible to determine this by tedious methods of weighing, drying and calculating. A new apparatus for indicating the degree of moisture content has just been developed by an American firm, in the form of a battery-operated detector, measuring 3½in.×5in.×7in., and weighing only 7lb. The operation of the device depends on the fact that the electrical resistance of many materials is a function of the moisture content, and the resistance varies as the moisture content. With the development of vacuum-tube circuits it has become possible to make use of this new technique in measuring electrical resistance and, therefore, the amount of moisture present. A range of 8 to 24 per cent. may be measured with the instrument, which is capable of recording 20 readings per minute. No skill is required on the part of the operator, as all that is necessary is the plunging of the electrode needles into the material and the turning of a dial. — *"Shipbuilding and Shipping Record", Vol. LVII, No. 7, 13th February, 1941, p. 146.*

What Is Cast Iron?

In the *Bulletin of the British Cast Iron Research Association,* Mr. J. G. Pearce gives the following definition, for the use of students as distinct from legal authorities, of cast iron:—"Cast irons are alloys of iron and carbon, with or without other elements, usually containing from 1·7 to 4·5 per cent. of carbon, which are not usefully wrought at any temperature. White cast irons contain free carbide; grey cast irons contain graphite; mottled cast irons contain both. Pig-irons are usually the product of the blast furnace, and are cast irons which may be either grey, white or mottled. Refined pig-irons may also be white, grey or mottled, and have had their composition, structure and properties modified either before solidification from the blast furnace or by a subsequent melting process. Malleable cast iron is the product obtained by the thermal treatment of a white cast iron in an oxidising or neutral atmosphere, so that part or the whole of the carbide carbon is transformed into temper carbon. In the whiteheart process the carbide of the iron is partially removed by an oxidising (decarburising) ore packing, and partially converted into temper carbon nodules. In the blackheart process the carbide of the iron is wholly or partly converted into temper carbon nodules in a neutral packing without decarburisation".—*"Foundry Trades Journal", Vol. 64, No. 1,278, 13th February, 1941, p. 102.*

Supplying Charging Air in Stages to Harland and Wolff Four-stroke Engines.

A diagrammatic view of a six-cylinder four-stroke pressure-charged engine is illustrated in Fig. 4. It is proposed to supply air under pressure in two or more stages, any of which can be isolated if necessary. The first stage may embody the well-known under-piston supercharging arrangement, or separate pumps reciprocated by the piston rods may be employed, while the second stage comprises an exhaust gas-driven blower or an independently driven compressor. The diagram shows a simple form in two stages with under-piston supercharge and an exhaust gas turbo-blower. By using the first-stage air from the under-piston supercharging system at a low pressure, leakage from the

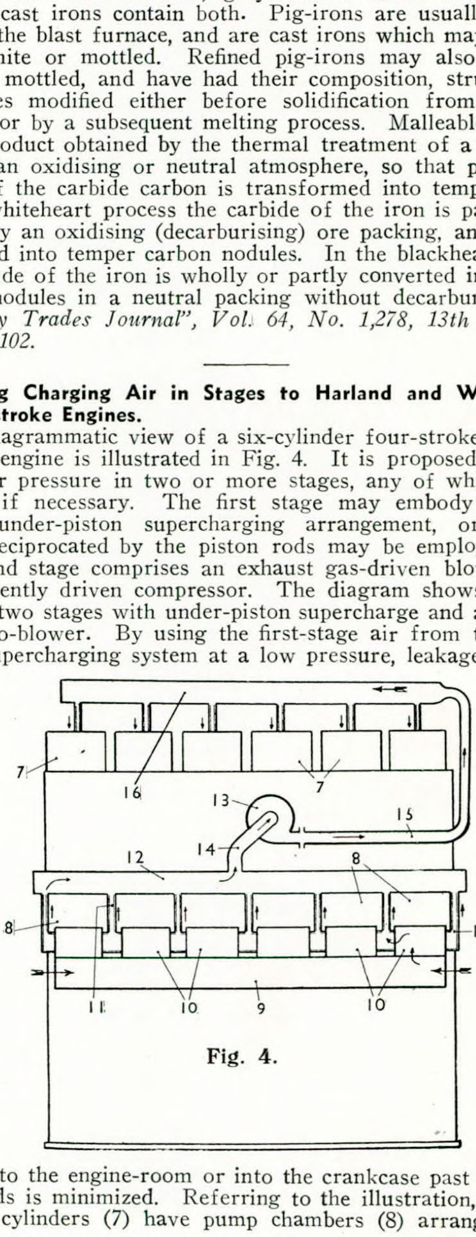


Fig. 4.

pumps into the engine-room or into the crankcase past the piston rod glands is minimized. Referring to the illustration, the main working cylinders (7) have pump chambers (8) arranged below the pistons. Air is drawn in through an inlet manifold (9) communicating with the suction and delivery valve chests (10), discharge being effected through the outlet pipes (11) to a manifold (12). The rotary blower (13) receives the air under pressure from a pipe (14) and after further compression has been effected the final supply is discharged through a pipe (15) to the distributing manifold (16). Valves may be provided in order to allow the air to pass to the manifold (16) without being dealt with by the turbo-blower and on the other hand the air may be taken direct from the inlet manifold (9) to the blower (13) without passing through the pump chambers (8). In other words, single-stage compression, either from the pumps or the blower, may be employed if the necessity arises. Various groupings of pumps are described in the specification and it is also mentioned that each cylinder may have its own turbo-blower. —*"The Motor Ship", Vol. XXI, No. 253, February 1941, p. 38.*

Emergency Air Supply Devices for Submarines.

Two forms of emergency air fitting for an external supply to submarines are shown in Fig. 5. Multiple-valve distribution boxes are provided at different positions and on the inside of the hull, pipes are carried to different compartments and tanks. Each pipe is fitted with a valve, preferably of the non-return type, so that water from a damaged compartment or tank can be prevented from entering the air distribution system. Non-return valves are also fitted at the tank connections and ordinary stop valves in the crew's accommodation. In the upper diagram a

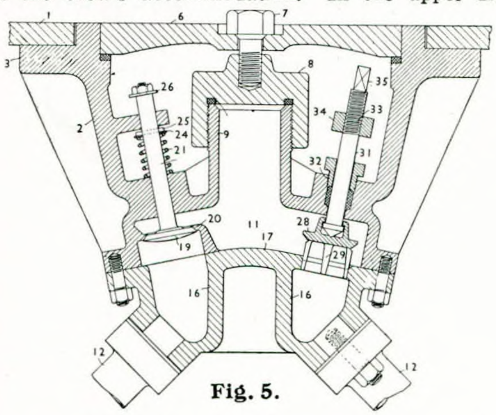
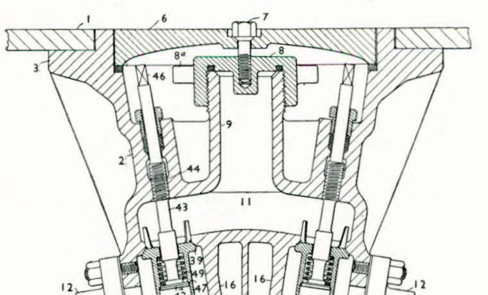


Fig. 5.



distribution box (2) is shown attached to the hull (1) by a flange (3). The cover (6) is secured by a bolt (7) which screws in a cap (8) tightened over the air supply connection (9). From the annular space (11) air can pass to the pipes (12), which lead to positions inside the vessel. Below the valves, the box has a cover (17) with openings (16) leading to the pipes. The left-hand valve illustrated is of the non-return type with a head (19) normally bearing on its seating (20), a spring (21) being provided. Above the spring is a washer (24) secured by a pin (25). The washer (26) at the top acts as a stop to limit the opening of the valve. An alternative fitting is a stop valve (28) with guide webs (29). The spindle (31) passes through a gland (32) and the screwed part (33) works in a boss (34) forming a portion of the body (2). The valve is operated by a key fitting on the squared end (35) of the spindle. In the event of an accident to the submarine, a diver removes the bolt (7) and the cover (6); the cap (8) is then unscrewed and an air pipe coupled on to the connection (9). The air pressure opens the non-return valves, but if stop valves are employed, these would have to be opened by the diver. The general construction of the box is the same in the lower diagram, except that it is in one ipece; also, the cap (8) has lugs (8a) for ease of removal. The valve bodies (16) are integral with the main body (2) and the openings below the valves have screwed caps (36) provided with squared sockets (38). The valves (39) are of the combined non-return and hand-operated type, each having a tubular extension (41) surrounding the head (42) of the spindle (43). The screwed part (44) works in the body (2) and there is a squared end (46) to take a suitable key. In the diagram the valves are shown shut and the spindles (43) are in their uppermost position. The head (42) of each valve spindle bears on a shoulder (47) and holds the valve (39) on its seat. When the spindle is screwed downwards, the head is moved away from the shoulder and the valve is free to open under the pressure of air (or oxygen), but remains subject to the influence of the return spring (49). The ends of the extensions (41) are closed by plugs (50) after the parts have fitted in place.—*"The Motor Ship", Vol. XXI, No. 253, February, 1941, p. 38.*

"Experiments in Rough Water with a Single-screw Ship Model".

The paper bearing this title describes a series of resistance and propulsion experiments carried out in rough water with a model hull of a single-screw 12-knot cargo ship of 0·75 block coefficient at deep and ballast draughts, the model being driven or towed "head on" against waves of short, medium and long wavelengths of varying height. The results of the tests indicated the effect of wave height upon resistance and propulsion over a range of ship speed at both deep and ballast draughts; and the percentage margin of power necessary to prevent serious loss of ship speed in rough weather was deduced for slow and fast cargo vessels of this type. The motion of the model during its passage through the different sets of rough water is recorded and described in the paper, together with some observations on the effect of the form of the entrance of ships upon their seaworthiness.—*Paper by J. L. Kent and R. S. Cutland, read at a meeting of the Institution of Engineers and Shipbuilders in Scotland on the 11th February, 1941.*

Organised Training Needed for Oil Engine Attendants.

The wide varieties of engine attendant employed in the innumerable Diesel-engine installations throughout this country range from the unskilled labourer, who merely fills lubricators and can start and stop the engine, to men like ex-naval E.R. ratings, with an extensive mechanical knowledge and a striking maintenance ingenuity. Although this country possesses a number of educational establishments which provide instruction in the theory and, to some extent, practice of heat engines, there are insufficient facilities for the training of works enginemen. Large Diesel installations are frequently attended by personnel recruited from the fitting and erecting staffs of Diesel-engine makers, but the smaller plants are not likely to attract such men. This is where there is a scope for inexpensive tuition on practical lines at not too widely spaced centres. In many cases it may be necessary to provide instruction for men of more advanced ages than are usual with engineering classes.—*"The Oil Engine", Vol. VIII, No. 94, February, 1941, p. 257.*

Employment of Heavy Residual Oil in Auxiliary Diesel Engines.

The accompanying diagrams show some of the modifications made by the Atlas Imperial Diesel Engine Company in the design of the 450-b.h.p., 6-cylr. four-stroke generator engines constructed at their Oakland works for the C-1 ships building on the Pacific Coast, in order to adapt them for running on the heavy fuel used in the main and auxiliary Diesel engines of the U.S. Maritime Commission's vessels (*see abstract on p. 188 of "Transactions",* *December, 1940, for report of shop tests.*) Owing to the low ignition index of this fuel, it was found desirable to increase the compression pressure from 400 to 500lb./in.^{2}. This resulted in exceptionally smooth running, with no more combustion noise than when using a lower compression pressure. Although the engines are equipped with the Bosch fuel-injection system and jerk pumps in order to comply with the Maritime Commission's specifications, the manufacturers believe that the common-rail

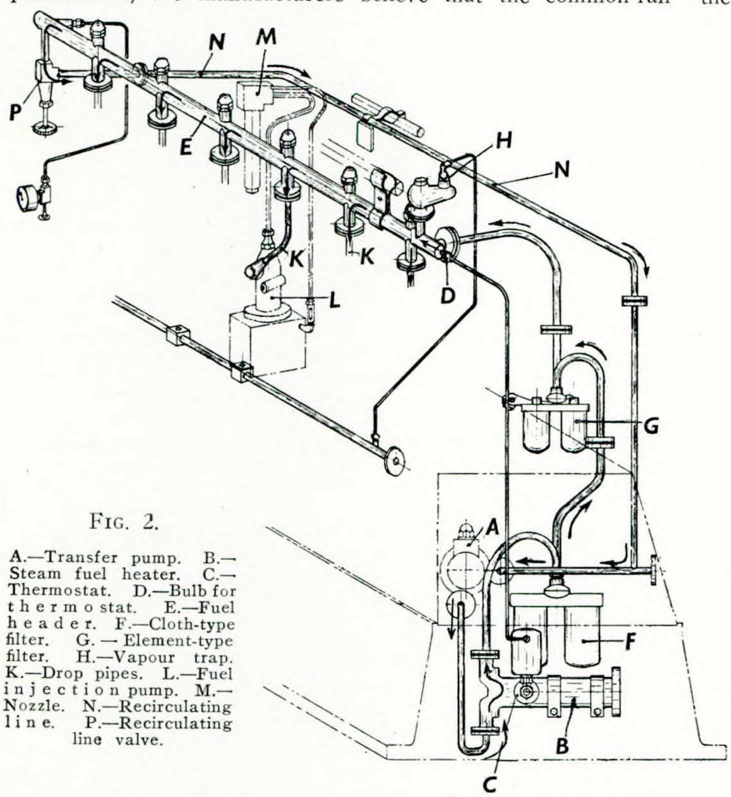


FIG. 2.

A.—Transfer pump. B.—Steam fuel heater. C.—Thermostat. D.—Bulb for thermostat. E.—Fuel header. F.—Cloth-type filter. G.—Element-type filter. H.—Vapour trap. K.—Drop pipes. L.—Fuel injection pump. M.—Nozzle. N.—Recirculating line. P.—Recirculating line valve.

system normally employed for similar engines is even less sensitive to different fuels than a jerk-pump system. The principal modifications adopted in the Bosch fuel-injection system to make it suitable for the burning of heavy fuel are:—(1) The fuel is

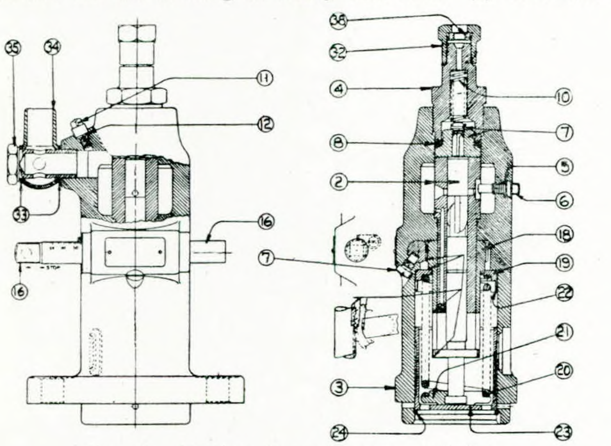


FIG. 3.—*Section through pump cylinder.*
2.—Plunger and barrel assembly. 3.—Pump housing. 4.—Holder. 5.—Gasket. 6.—Setscrew. 7.—Valve assembly. 8.—Gasket. 10.—Spring. 11.—Screw. 12.—Gasket. 16.—Rod control. 17.—Securing screw. 18.—Control sleeve. 19.—Plunger spring seat. 20.—Plunger spring. 21.—Plunger spring seat. 22.—Spring ring. 23.—Plunger guide. 24.—Split ring. 31.—Lead plug. 32.—Delivery nipple nut. 33.—Fuel inlet union gasket. 34.—Fuel inlet union. 35.—Screw. 38.—Washer.

heated to a viscosity of about 100 sec. to enable the injection system to handle it without excessive friction loss and to make the passage through the filters easier. With the fuel used this means heating to between 190° and 200° F. (2) The heavy fuel being more corrosive than standard Diesel oil, all parts in the pumps and injector valves are made of corrosion-resisting materials. (3) As the heavy fuel is more apt to contain abrasives, the question of filtering has received special attention and the plungers in the Bosch injection pumps are hydraulically balanced, *i.e.*, the usual hydraulic unbalance due to the relief around the control valve helix has been eliminated by a corresponding relief on the other side of the plunger. (4) Care has been taken to lead off any vapours which might be formed before the fuel is delivered to the pumps, as gases and vapours given off by heavy fuels are often extremely corrosive. (5) To minimise carbon formation on the nozzles, direct water cooling of the nozzle and holder was adopted and has proved to be most essential for successful operation. The arrangement of the fuel system is shown in Fig. 2, the path of the fuel being indicated by arrows. After passing through the transfer pump (A) the fuel enters the heater (B) in which saturated steam at 50lb./in.^{2} serves to raise the fuel temperature to 190°-200° F. The steam supply is controlled by a self-contained temperature controller (C) with its bulb (D) in the fuel header (E), the temperature maintained at the inlet of the fuel header being about 175° F. After being heated, the fuel passes successively through a cloth-type filter (F) and metal-element-type filter (G) to the header, any vapours or gases entrained in the fuel being trapped and led off by the vapour trap (H) at the top of the header. The latter is provided with drop pipes (K) leading down to the individual fuel-injection pumps (L) which discharge the fuel to the nozzles (M). In order to maintain the fuel delivered to all the pumps at a fairly even temperature,

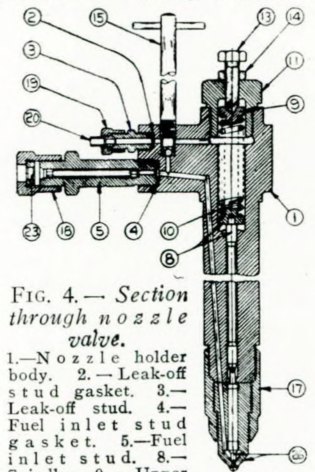


FIG. 4.—*Section through nozzle valve.*
1.—Nozzle holder body. 2.—Leak-off stud gasket. 3.—Leak-off stud. 4.—Fuel inlet stud gasket. 5.—Fuel inlet stud. 8.—Spindle. 9.—Upper spring seat. 10.—Pressure adjusting spring. 11.—Spring retainer cap nut. 13.—Spring adjustinng screw. 14.—Adjusting screw locknut. 15.—Bleeder screw. 17.—Nozzle cap nut. 18.—Fuel inlet nipple nut. 19.—Leak-off nipple nut. 20.—Leak-off connection. 23.—Washer. 26.—Nozzle assembly.

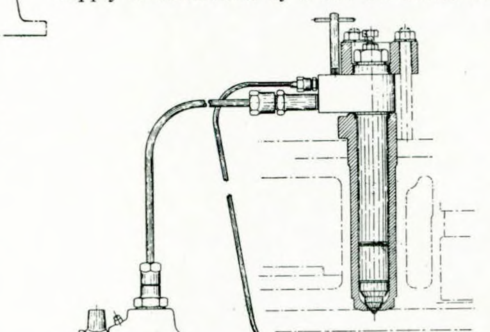


FIG. 5.—*Arrangement of pump and nozzle valve.*

a recirculating pipe (N) is led from the far end of the header, the amount of fuel to be recirculated being determined by the valve (P), which is opened sufficiently to cause a pressure drop of about 15lb./in.^{2} in the fuel header. Under these conditions the temperature difference between the first and last pumps is only about 10° F. In other words, if the transfer pump by-pass valve is set for, say, 45lb./in.^{2} pressure with the recirculating line valve closed, the latter is opened enough to cause the pressure to drop to approximately 30lb./in.^{2}. The nozzle and nozzle holder are illustrated in Figs. 4 and 5. They are mounted in a cage with separate water connections, the water flowing down along the nozzle holder body and up again through two cored channels in the cage. At the lower end the water is circulated across the nozzle from one of the cored passages to the other, thus cooling it effectively. At the normal rating of the engine the brake m.e.p. is 79·9lb./in.^{2} and the piston speed 933ft./min. The full-load exhaust temperature is 770° F. and the maximum combustion pressure approximately 695lb./in.^{2}. Although the engines are rated at 450 b.h.p. at 250 r.p.m. and are capable of developing 550 b.h.p. at 350 r.p.m., it is anticipated that the normal load on each engine when installed on board will be 325 b.h.p.—*"The Motor Ship", Vol. XXI, No. 253, February, 1941, pp. 376-377.*

Causes of Diesel Engine Breakdowns.

A paper which was recently awarded a prize by the Edison Electrical Institute, in America, deals with the causes of breakdowns in Diesel engines, the author having analysed data concerning failures or minor mishaps of well over 200 Diesel engines in a period of five years. Including accidents, overhaul, inspection, adjustment, wear and tear, it was found that there were 31 different causes for engines being out of service, and the accompanying table gives the most important of these. The majority of the engines concerned are of the stationary type, and it is possible that in marine work there would be some difference in the causes. Apparently the examination, repair and replacement of pistons and piston rings represent the main cause of engine stoppage, apart from inspection and adjustment, and air compressors appear to be subject to failure to a considerable extent. This would not apply in the case of modern engines of the airless-injection type. Crankshaft troubles seem to be astonishingly frequent in respect of total hours of operation, but the number of engines affected is relatively small, and it is obvious that in each case it is a question of the plant affected being out of operation for a prolonged period. Crosshead pins and bearings are singularly free from serious trouble, and this appears to be all the more remarkable in view of the fact that most of the engines to which the statistics refer are of the trunk-piston type and no doubt run at a relatively high speed.

Engine part or other cause of failure or stoppage.	Total hrs. of stoppage in 5 years.	Per cent. of hrs. of stoppage in 5 years.	No. of engines affected.
(1) Inspection and adjustment	21,043	24·0	127
(2) Main pistons and piston rings	11,562	13·2	139
(3) General overhaul	10,911	12·4	15
(4) Crankshaft	7,368	8·4	10
(5) Air compressors (air-inj. engines)	4,976	5·7	67
(6) Cylinder liners	3,710	4·2	28
(7) Crank and piston-pin bearings	3,704	4·2	91
(8) Exhaust valves (4-stroke engines)	2,049	2·3	70
(9) Main bearings (incl. thrust and outboard)	1,967	2·2	54
(10) Fuel-spray valves (priming valves, etc.)	1,807	2·0	87
(11) Cylinder heads	1,653	1·9	60
(12) Governors and governor drives	1,540	1·8	41
(13) Water cooling (incl. scale trouble)	1,447	1·7	66
(14) Air inlet valves	1,401	1·6	74
(15) Fuel pumps, distributors, drives, etc.	1,273	1·4	90
(16) Lubricating-oil system ...	1,230	1·4	71
(17) Fuel-oil system	1,044	1·2	38
(18) Cracked frames and cylinders	991	1·1	8
(19) Crosshead pins and bearings	964	1·1	15

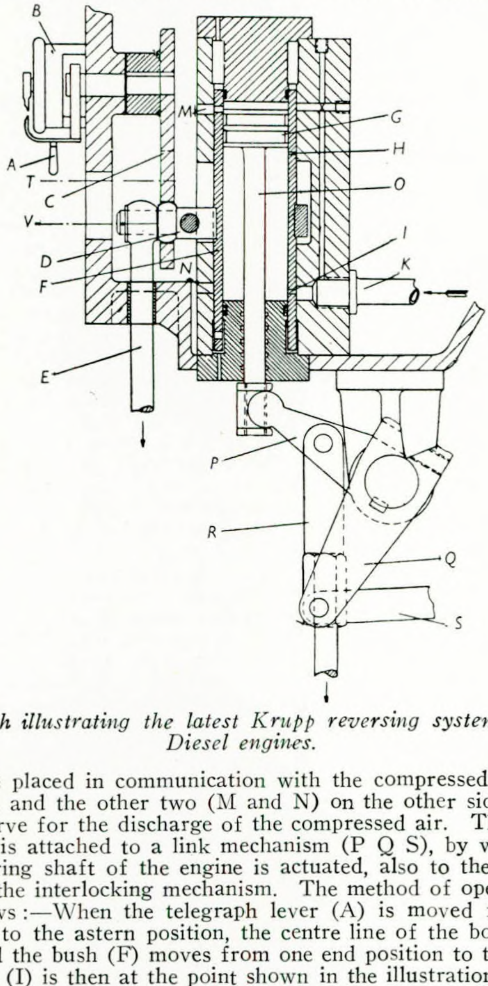
—*"The Motor Ship", Vol. XXI, No. 253, February, 1941, p. 385.*

Spare Parts for Motorship Machinery.

The question of providing spare parts for the machinery of some of the enemy ships taken over by the British Government, also certain of those operated by our Allies, and particularly Norway and Holland, may become difficult if the war is prolonged and no special provision is made for maintaining an adequate number of spares. Most of the vessels concerned rely for their spare parts in normal times upon manufacturers in Continental countries, but such sources of supply are not now available. It is fortunate that a large proportion of these ships have Burmeister and Wain-type engines which are manufactured on a large scale in this country and in respect of which the supply of spare parts presents no difficulty. There are, however, many acquired ships with Fiat engines, of which no modern types have ever been built here. Some Dutch and Norwegian vessels have Stork engines, which have not at any time been constructed in the U.K., whilst several are equipped with M.A.N. machinery of the single-acting or double-acting types, which, again, have not been produced in this country, as the British firms holding licences for their consrtruction have only had very limited opportunities to build the designs in question. As regards ships with Sulzer or Werkspoor engines, the position is less difficult, as these types of engines have been constructed on a relatively large scale in this country. It is unlikely that the problem will grow acute for some time, but it will undoubtedly become so in the course of the next 18 months, and it would therefore seem desirable to make plans to ensure that no ship shall, in any circumstances, be laid up for lack of spare parts. The matter is by no means as simple as it might appear, because not only is a detailed knowledge of the design of the respective parts required, but the composition of material used and other factors must be thoroughly understood, while the persons responsible for the production of the components concerned must have considerable experience of the engines in question. The best plan would undoubtedly be to equip a new factory with specialised machinery for the manufacture of these spare parts. Such a factory would have great after-war possibilities as a commercial concern, and, in the end, instead of being a drain upon the country's resources, would probably pay for itself over and over again, while during the war period—if the war continues into 1943—the value of such a centralised production plant for Diesel engine spare parts, which might prevent ships being laid up for a considerable period, would be impossible to assess. It is obvious that under present conditions, when Diesel marine machinery is so urgently needed, manufacturers fully engaged in the production of plant for new ships, cannot be expected to undertake the supply of spare parts, other than those for engines they have built and are building. —*"The Motor Ship", Vol. XXI, No. 253, February, 1941, p. 357.*

A German Diesel Engine Reversing System.

Krupp have developed a new system of reversing oil engines, the control being carried out from the engine telegraph lever. In general, when such a method is employed, the supply of the compressed air to the reversing piston is regulated when the telegraph lever is moved so that air is delivered to one side of the piston or the other, according to the direction of rotation required. In the new arrangement a simplification has been effected, and distribution of the air is carried out in such a way that the bush in which the reversing piston moves becomes the means of distribution of the air, suitable inlet and discharge openings being provided. The arrangement is shown in the illustration. A is the telegraph lever and B the engine telegraph, A being fixed to a disc (C) in which is a curved slot. In this slot is a bolt (D) which on one side is coupled to the interlocking lever (E) and on the other side to the bush (F), which serves not only as a guide for the piston (G), but also as a distributing valve. For this purpose it is carried in a bearing so as to be capable of moving longitudinally, and is provided with four holes, of which the two (H and I) may alternately be placed in communication with the compressed-air inlet pipe (K), and the other two (M and N) on the other side of the piston serve for the discharge of the compressed air. The piston rod (O) is attached to a link mechanism (P Q S), by which the manœuvring shaft of the engine is actuated, also to the rod (R) fixed to the interlocking mechanism. The method of operation is as follows:—When the telegraph lever (A) is moved from the forward to the astern position, the centre line of the bolt (D) is at V, and the bush (F) moves from one end position to the other. The hole (I) is then at the point shown in the illustration, so that compressed air enters through the pipe (K) under the piston (G), which moves upwards, operating the reversing mechanism and discharging the air above the piston to the ports (M). When the telegraph lever (A) is brought back to the ahead position (the centre line of the bolt (D) is now at T), compressed air enters through the hole (H) and the piston is forced down. — *"The Motorship", Vol. XXI, No. 253, February, 1941, p. 369.*



Sketch illustrating the latest Krupp reversing system for Diesel engines.

The American Motorship "Rio Hudson".

The 9,000-ton passenger and cargo liner "Rio Hudson", recently launched from the yard of the Sun Shipbuilding and Dry Dock Co., is one of four similar ships under construction for the Moore-McCormack Lines, Inc., and is the first vessel ever built to be equipped with geared engines of the Doxford type. The main dimensions of the ship are $492 \times 69\frac{1}{2} \times 42\frac{1}{2}$ft., with a draught of 27ft. 3in. and a d.w. capacity of 10,280 tons. The total cargo capacity is 560,000 cu.ft., of which 62,000 cu.ft. are taken up by the refrigerated cargo spaces. The vessel is also provided with deep tanks for the carriage of oil cargo, the capacity of these being 1,970 tons. The ship's fuel-oil tanks in the double bottom forward and aft of the machinery space hold about 1,500 tons and will give the vessel a cruising radius of some 14,000 sea miles. The total daily fuel consumption at the normal sea speed of 16½ knots with the ship fully laden, should be in the neighbourhood of 34-35 tons. The propelling machinery will consist of two 4,500-b.h.p. Sun-Doxford engines running at 180 r.p.m. and driving a single propeller at about 79 r.p.m. through reduction gearing and Westinghouse electric slip couplings. The latter are excited from the main generators at 230 volts and absorb about 2 per cent. of the power transmitted. They have short-circuited squirrel-cage secondary windings, with salient-pole primary winchings. Steam for heating the ship is provided by a combined waste-heat and oil-fired boiler, to which the exhausts from both main engines are led and which also serves as a silencer. The total length of the machinery space is about 54ft. The whole of the auxiliary and deck machinery is electrically driven; it includes 20 three-ton cargo winches, an electric windlass and the steering gear. The emergency generating plant is installed in a compartment in the funnel, which is elliptical, 24ft. in length and about 9ft. wide. Part of the funnel is divided off to form an air duct direct to the engine room for the supply of scavenging air to the propelling machinery. The passenger accommodation is exceptionally roomy and comprises a number of public rooms, 22 single and 34 double cabins, each of which is air conditioned and has an adjoining bathroom, and 20 de luxe suites each consisting of a bedroom, sitting-room, bathroom, large luggage room and private veranda. The arrangement of these cabins is quite new and similar to that adopted in the Swedish-American Line motorship "Stockholm" launched in Italy just before that country entered the war. The main passenger corridor is in the centre of the ship and the de luxe suites occupy the space between it and the ship's sides. All the public rooms are air conditioned and a new system of air conditioning is being fitted for the cargo holds. The vessel will carry a total crew of 110, and provision is being made for mounting a 6-in. gun aft and a 3-in. gun on each side of the promenade deck. It is anticipated that the "Rio Hudson" will be completed in May and that she is likely to be employed on the regular service between New York and the East Coast ports of South America. It is understood that the cost of the ship is about £650,000, but since the order was placed prices have risen and the latest vessels of this type will cost about 25 per cent. more.—*"The Motor Ship", Vol. XXI, No. 253, February, 1941, pp. 360-361.*

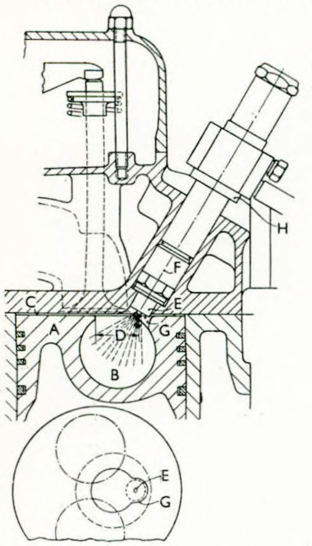
Fallacies of Lubrication.

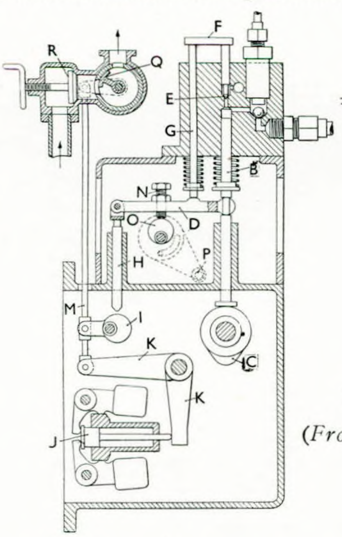
A number of erroneous views on lubricants were dealt with by Mr. Wilfred E. Gooday in a paper recently read before a meeting of the Chemical, Metallurgical and Mining Society of South Africa. The author, basing his remarks on experience in Great Britain, called attention to the gaps still existing in our knowledge of crude oils and the heavier petroleum products, as well as to the fact that the engineer cannot define accurately what he wants in a lubricant, any more than a lubrication technologist can describe precisely what he recommends. Recognising these basic limitations as to what can be done, it is easier to admit what is impossible or indeterminate and thus the origin of a fallacy. First and foremost comes the fact that it is not possible to judge an oil solely by the source of its crude, as owing to improvements in refinement and blending, the place-origin of lubricating stocks is no longer a criterion of the finished product. Furthermore, attempts to express the lubricating value of oils in terms of their distinctive physical characteristics, have led to many fallacies. The specific gravity or relative density of an oil, for example, bears no relation to its performance as a lubricant, and in view of the fact that oils, as used, are often blends of stocks from different sources, it is evident that specific gravity is no reliable indication of origin. Again, however useful the flash point may be in determining the fire risk of an oil in transport and storage, as an aid to comparison or differentiation between oils, and as a means of detecting contamination, it is certainly no basis for estimating the comparative rates of evaporation of lubricating oils. Some years ago, an oil of given viscosity was much darker than it is to-day and there is still some tendency to regard light colour as an indication of light body or low viscosity. Actually, no such correspondence can be established and for all but special applications, the colour of lubricants may be ignored. Viscosity is the best known and most useful property for which a test is applied in the technology of lubricating oils, and the old fallacy "the heavier the oil the better" has now nearly expired. Too much "body" results in excessive fluid friction with additional generation and retarded removal of heat, and no evidence of the superior "guts" formerly supposed to be associated with high viscosity. Citation of viscosity at a temperature remote from that at which the oil is to work is probably the most prevalent mistake to-day, but this is rather an oversight than a fallacy, since the effect of temperature on viscosity is well known. Most of the fallacies to which the author of the paper referred are due either to misunderstanding or to sheer lack of information on the part of the user, the supplier or both. This is particularly evident in the field of acidity, where three kinds of acid have to be considered, the petroleum or naphthenic acids, free fatty acids introduced by compounding, and mineral acids arising from defective refining or subsequent contamination. Confusion engendered by different methods of expressing acidity is avoided by referring all forms of acid to neutralisation values, and provided all due account is taken not only of the actual acidity of an oil but also of the rate at which it has been developed, and the conditions of service of the oil, useful conclusions can be reached about the resistance of the lubricant to deterioration in service. On the other hand, judgment on an incomplete basis may be very mistaken, and in particular lubricants have often been blamed for deposits and corrosion caused by the electrolytic action of eddy or stray currents. The author also made some cogent remarks on misunderstandings arising from the widely different meaning of "sludge" in different contexts, and uttered a warning concerning the inadvisability of mixing oils from different suppliers or adding undue proportions of new to used oil. He likewise dealt with the fallacy that oil "loses its nature" by filtration, and concluded his remarks by stressing the vital importance of care in the framing of specifications for lubricating oil. Precise definition of the best lubricant is inherently impossible in our present state of knowledge, and merely to describe an oil known to be satisfactory may automatically exclude better products. All that can be done with safety is to draft a protective specification, prohibiting unsuitable or contaminated oils but imposing no unnecessary restriction on genuine products. — *"The Power and Works Engineer", Vol. XXXVI, No. 416, February, 1941, pp. 25-26.*

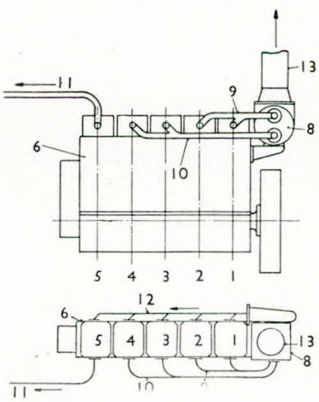
Recent Inventions of Oil Engine Interest.

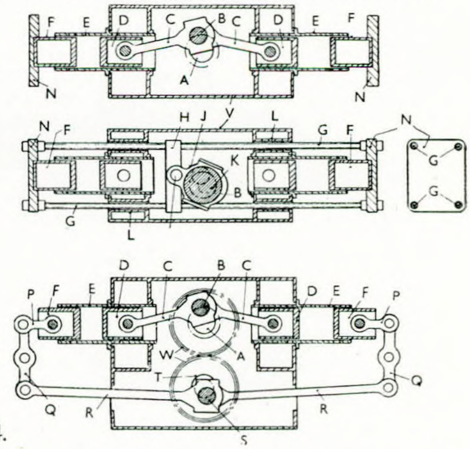
M.A.N. Combustion Space.

A modification to the M.A.N. high-speed type of engine having the combustion space in the piston is illustrated in Fig. 1. The principal feature is the inclusion of a cut-out extension below the fuel injection nozzle (H), so that the inclination of the injector can be increased with a view to convenience in fitting and avoiding any interference with the valve gear. It will be seen from the diagram that the combustion space (B) in the piston (A) is spherical. The piston clearance (C) is reduced to a minimum. Fuel is injected through the nozzle (E) and the inclination of the axis (F) appears to be determined by the spread of the jet (D), which should not impinge on the edge of the opening and should reach the centre of the combustion space. The desired effect is obtained by the provision of a clearance (G) around the jet, designed so that there shall be no unfavourable influence on combustion.









(*From left to right*) Figs. 1, 2, 3, 4.

Crossley Dual-fuel Control.

In Fig. 2 is illustrated a Crossley device for changing over an engine from liquid to gaseous fuels, at the same time maintaining efficient governor control. The injection pump has a constant stroke and a spill valve is regulated by the governor when the unit works as an oil engine. When gas is used the governor controls the supply and the pump gives small injections for pilot oil ignition. The fuel-pump plunger (B) is operated by a cam (C) and a tappet lever (D) actuates the spill valve (E) through a spindle (G) and a crossbar (F). A rod (H) on one end of the lever engages an eccentric (I) turned by the governor (J) through levers (K) and a spindle (M). On the tappet lever is a stop (N) engaging an eccentric (O) which can be adjusted by a handle (P). The governor controls the gas inlet area by a valve (Q). When the engine is working on gas with pilot oil ignition the valve (R) is open; the eccentric (O) engages the stop (N) and keeps the rod (H) clear of the governor-operated eccentric (I). The spill valve (E) ensures that the quantity of oil is sufficiently small. In order to run the unit as a compression-ignition engine the valve (R) is closed and the eccentric (O) is turned to clear the stop (N), so that the rod (H) makes contact with the eccentric (I) and places the spill valve under the control of the governor. It is stated that the engine may work with part oil and part gas by appropriate adjustment of the valve (R) and the eccentric (O).

Turbo-charging with Five or Seven Cylinders.

It is known that when exhaust gas turbo-charging is applied to engines with five or seven cylinders certain difficulties have to be overcome, due to the requirements of timing. A solution has been found in the employment of three exhaust manifolds and the same number of gas turbine chambers which, however, add to the cost of the engine. An effective method of dealing with the problem is to discharge the exhaust gas from one cylinder either to the atmosphere or into the waste gas pipe from the turbine. The diagram (Fig. 3) shows a five-cylinder engine (6) having a firing order of 1, 3, 5, 4, 2. Cylinder No. 5 exhausts to the atmosphere through a pipe (11), whilst cylinders 1 and 2 discharge into a pipe (9) and cylinders 3 and 4 are connected to a pipe (10). The turbo-blower (8) is fitted with gas inlet connections for the pipes (9, 10) and an exhaust uptake (13) into which the pipe from cylinder No. 5 may be led. Charging air from the blower passes to all the cylinders through a pipe (12). The minimum intervals of ignition for the pairs of cylinders which exhaust into the turbine is 288°. In a seven-cylinder engine the sequence of ignition may be 1, 3, 5, 7, 6, 4, 2 and the exhaust gas from cylinder No. 2 would be taken to the atmosphere or to the turbine outlet. Cylinders 1, 5, and 6 discharge into one manifold, whilst cylinders 3, 4 and 7 are connected to the other, and the crank angles in this instance are 206°.

Harland and Wolff Horizontal Engine.

In Fig. 4 the two upper diagrams refer to a horizontal opposed-piston engine in which the crankpin (B) is common to a pair of connecting rods (C) attached to the inner pistons (D). The cylinders (E) are coaxial and are arranged on opposite sides of the crankshaft (A). The outer pistons (F) are secured to yokes (N) and there are four tie-rods (G) attached to a frame (H). The frame has an aperture to allow for the free movement of the left-hand connecting rod. An alternative form of construction, however, would be to employ separate cross-pieces, one for each pair of tie-rods. The frames or cross-pieces are provided with journals to take eccentric straps (J), one on each side of the crankpin. Guides (L) in the engine framing (V) are provided to take the side-thrust on the tie-rods due to the action of the eccentrics (K). The bottom diagram shows a different form of construction in which there are two shafts. The outer pistons (F) are connected by links (P) to rocking levers (Q). These levers are attached to coupling rods (R), which are in turn connected to the crankpin (or eccentric) (S) on the shaft (T). The engine crankshaft (A) and the auxiliary shaft (T) are geared together by toothed wheels (W). Further arrangements are described in the specification.

Exhaust-turbo Generator.

A compact design of exhaust-gas turbo-driven generator is shown in Fig. 5. The set comprises the generator (10) with a rotor shaft (12), and the gas turbine (25). Due to the close

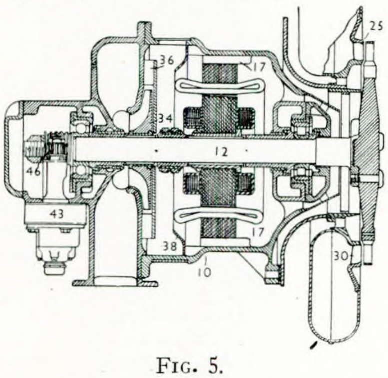


Fig. 5.

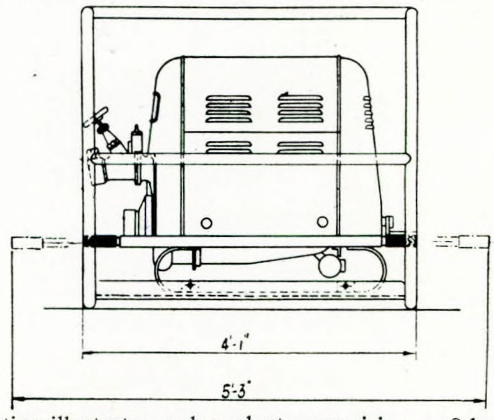
spacing of the turbine and generator, special cooling means are necessary to protect the latter. There is an air impeller (34); its blades compress the air and a diffuser (36) is provided. In its passage through the generator, the air travels along all the ducts, gaps and other available spaces where cooling effects are obtainable. A flared ring (38) is fitted to reduce the flow through the large ducts (17) and thus ensure that the smaller spaces are properly supplied with air. The oil pump (43) is driven by a worm gear (46).—*"The Oil Engine", Vol. VIII, No. 94, February, 1941, p. 279.*

An Australian Kort Nozzle Conversion.

After the owners of the 1,100-i.h.p. single-screw tug "Woonda", of Port Adelaide, had decided to fit the vessel with a Kort nozzle, the outbreak of war made it impossible for the London manufacturers to supply the nozzle from this country. It was therefore arranged to make and fit the nozzle in Australia in accordance with drawings furnished by the makers, the work being carried out under the direction of the superintendent engineer of the Adelaide Steamship Company, at their works. In the manufacture of these nozzles it is of paramount importance that the various complicated contours should be rigidly adhered to. Practically all the plates forming the "skin" of the nozzle have to be shaped to within very fine limits, so that the workmanship and arc welding must necessarily be of the highest standard. Nozzles are stiffened internally by vertical webs and radial diaphragm plates, the vertical webs being attached by means of welded lugs to the underside of the hull plating. All the vulnerable butt welds are so arranged as to be reinforced by the webs or diaphragm plates. This internal stiffening, together with the external plating of the nozzle, not only provides for a very rigid connection of the hull and stern frame, but also makes the nozzle itself a particularly robust fitting which affords protection to the propeller. No alterations of any kind were made in the hull, the nozzle being superimposed on to the existing hull plating and stern frame by means of arc welding, as is the usual practice in single-screw conversions. On completion of fitting the nozzle was water-tested to a head of about 8ft. Prior to fitting the nozzle, careful tests were carried out which demonstrated that the "Woonda", when developing 792 i.h.p., obtained a static pull of $10\frac{5}{8}$ tons, the propeller being 11ft. 2in. in diameter. Further tests made after fitting the nozzle were carried out with the original propeller "dressed down" to 11ft. in diameter, which enabled a static pull of $16\frac{1}{4}$ tons to be obtained with 830 i.h.p. This pull is claimed to make the "Woonda" one of the most powerful single-screw tugs afloat. The increase in pull of 52 per cent. was obtained with only 5 per cent. more power than prior to the fitting of the nozzle. It is calculated that with a new propeller specially designed for the nozzle, an increase in standing pull to about 19 tons will be obtainable without any detriment to the free speed. A new propeller design is being prepared. This is the largest propeller which the Kort Propulsion Company, of London, have yet had to deal with. The "Woonda's" nozzle being the first to be fitted at Port Adelaide, the Adelaide Steamship Company had had no previous experience in making or fitting nozzles, but the results obtained have caused the tug's owners to express their unqualified satisfaction with the fitment.—*"Lloyd's List and Shipping Gazette", No. 39,400, 19th February, 1941, pp. 6-7.*

Portable Emergency Pumping Plant for Ships' Use.

Among the new regulations prescribed by the Merchant Shipping (Fire Appliance) Rules, 1940, is one which requires all ocean-going motorships to carry a portable emergency power pump, while in ships of more than 4,000 tons gross at least two pumps must be provided. The fuel tanks of such pumps must be of sufficient capacity to enable them to be run for not less than one hour. Although it is understood that the Ministry of Shipping recommend a small air-cooled engine of about 3 h.p. for driving the pump, it is probable that most shipowners would prefer a higher-powered unit for larger vessels. The accompany-



ing elevation illustrates such a plant, comprising an 8-h.p. Morris engine coupled to a Sigmund pump with a delivery of 220 g.p.m. at 60lb./in.^{2} or 120 g.p.m. at 120lb./in.^{2} pressure. This unit is, of course, larger and heavier than the air-cooled engine-driven pumps, but its height is only 3ft. $4\frac{1}{2}$in. and it can be carried by four men through any normal ship's alleyway. Every ship must also carry a stirrup with 100ft. of tubing for dealing with fires caused by incendiary bombs.—*"The Motor Ship", Vol. XXI, No. 253, February, 1941, p. 373.*

The late WILLIAM LIVINGSTON ROXBURGH.

OBITUARY.

WILLIAM LIVINGSTON ROXBURGH.

We record with regret the death of Mr. William Livingston Roxburgh, which occurred at Bearsden, Glasgow, on Thursday, 13th March, 1941.

Born in 1868 at Kilmarnock, where he was educated, Mr. Roxburgh served his apprenticeship partly at Messrs. Grant, Ritchie & Co.'s and subsequently at the works of Messrs. Glenfield & Co., hydraulic engineers of Kilmarnock. For some years afterwards he was in shore employment with Messrs. Barclay's Locomotive Works, Messrs. John Penn & Co., Ltd., and Messrs. Humphreys & Tennant, Ltd.

Mr. Roxburgh then commenced his sea career with Messrs. Elder Dempster & Co., later serving with the Red Star Line and Houlder Bros. About 1916 he became superintendent engineer of Messrs. Coast Lines, Ltd., a position in which he served with distinction until his retirement about two years ago. Mr. Roxburgh was also on the board of The Ardrossan Dockyard, Ltd.

For many years a very keen Member of The Institute, Mr. Roxburgh was elected to the Council in 1930 and proved an able and valuable Member throughout his three years in that office, his services being recognised by his colleagues by nomination for a further term of office. By this time, however, Mr. Roxburgh's health was failing somewhat, and he was compelled to restrict his active participation in The Institute's work.

Mr. Roxburgh, who was also a Member of the Institution of Naval Architects, was well-known in marine engineering and shipping circles in London, where his death has occasioned widespread regret.