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Marine Steam Turbine Design.

By SYDNEY ALBERT SMITH, M.Sc. (Eng.), M.I.Mech.E., M I.N.A. (Member).

(1) General Particulars.

The propelling machinery of the vessel for which the power was determined in the *paper "Powering of Ships" is twin-screw single-reduction geared turbines, each set consisting of an H.P., I.P. and L.P. turbine, the steam conditions being 400lb. per sq. in. gauge, 725° F. at H.P. turbine stop valve, the exhaust from the L.P. turbine being 28½in. vacuum on 30in. barometer.

The high-pressure turbine is of the impulse reaction type, the intermediate-pressure turbine of reaction type with an impulse reversing turbine incorporated, and the low-pressure turbine reaction type with an impulse reaction astern turbine incorporated, the total astern power being about 75 per cent. of the full ahead power.

Steam is to be bled from the turbines to give a feed temperature of 350° F.

(2) Definition of Efficiencies.

Before proceeding to the actual design, the fundamental formulæ will be developed and the symbols used for efficiencies defined.

$$\textit{Stage efficiency, } \eta_s=\frac{\text{actual stage heat drop}}{\text{adiabatic stage heat drop}}=\frac{h_s}{h_r}$$

$$\textit{Internal efficiency, } \eta_1=\frac{\text{actual heat drop turned into work}}{\text{adiabatic heat drop}}=\frac{H_1}{H_r}$$

$$=\frac{\Sigma h_s}{H_r}=\frac{\eta_s\Sigma h_r}{H_r}=\frac{\eta_s H_c}{H_r}=\frac{\eta_s R.H_r}{H_r}=R.\eta_s$$

where,

H_c = cumulative heat drop $=\Sigma h_r$; H_r = adiabatic heat drop;

R = reheat factor $=\dfrac{H_c}{H_r}$

Efficiency Ratio, $\eta=\eta_1-x$

Where x = losses due to friction of bearings, radiation, gland, and dummy, exhaust losses and gearing losses.

Steam Consumption.

$$\text{Theoretical consumption}=\frac{2545}{H_r}\text{lbs. per s.h.p. hour.}$$

$$\text{Actual consumption}=\frac{2545}{\eta.H_r}\quad ,,\quad ,,\quad ,,$$

* Trans. Institute of Marine Engineers, May, 1940, Vol. LII, Part 4, pp. 77-87.

(3) Determination of the number of rows of moving blades.

In determining the number of rows of moving blades, the *curve of efficiency ratios given by the late H. M. Martin (Fig. 1), has been used. The base of the curve is,

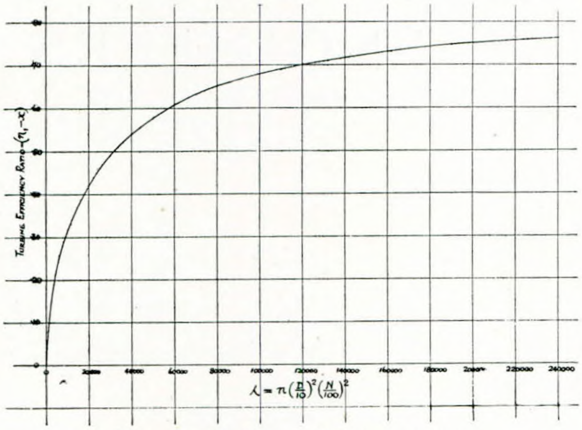


FIG. 1.—*Turbine efficiency curve.*

$$K = n\left(\frac{D}{10}\right)^2\left(\frac{N}{100}\right)^2 \quad \dots \quad \dots \quad \dots \quad \dots \quad (1)$$

where, n = number of rows of moving blades
D = mean blade ring diameter in inches
N = revolutions of rotor per minute.

It will be seen from equation (1) that with a given diameter and revolutions, the number of rows varies directly as K. By raising K the number of rows is increased and for given thermal conditions the steam speed is reduced and hence for a given mean diameter the ratio of blade speed to steam speed (ρ) is increased resulting in an improvement in efficiency.

(4) Impulse Turbine Design Formulæ.

The following values of the stage efficiency of one, two and three ring impulse turbines for various ratios of blade speed to the theoretical steam speed are taken from the tests carried out by Mr. K. Baumann, as published in many textbooks on steam turbines :—

One Ring.

ρ	0	·05	·10	·15	·20	·25	·30	·35	·40	·45	·50	·55	·60
η_s	0	·15	·30	·425	·53	·62	·70	·75	·78	·80	·79	·76	·70

Two Rings.

ρ	0	·05	·10	·15	·20	·21	·22	·23	·24	·25	·26
η_s	0	·23	·42	·56	·66	·67	·675	·68	·675	·67	·66

Three Rings.

ρ	0	·05	·10	·11	·12	·13	·14	·15
η_s	0	·30	·49	·51	·525	·523	·52	·51

$$\rho = \frac{\text{blade speed, ft./sec.}}{\text{theoretical steam speed, ft./sec.}} = \frac{\pi DN}{720 \times 223{\cdot}7\sqrt{h_r}} = \frac{D.N}{51248\sqrt{h_r}}$$

$$\therefore D = \frac{51248\rho\sqrt{h_r}}{N} \quad \dots \quad \dots \quad \dots \quad \dots \quad (2)$$

where, D = mean blade ring diameter in inches
h_r = adiabatic heat drop
N = revolutions per minute.

With regard to the entrance and exit angles of the impulse blading, these appear to conform closely to those obtained by assuming frictionless flow and the effective blade heights to the velocity diagrams allowing for friction.

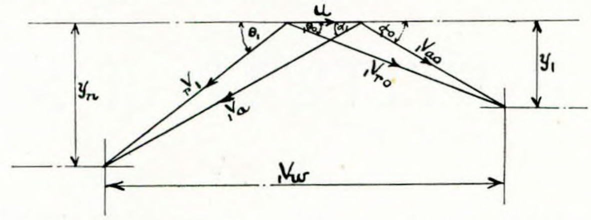


FIG. 2.—*Typical velocity diagram, impulse blading.*

A velocity diagram showing the symbols used is shown in Fig. 2

where, u = peripheral mean blade speed, ft./sec.
$_1V_a$ = absolute velocity of steam jet from nozzle, ft./sec.
$_rV_1$ = relative velocity of steam entering first moving row of blades, ft./sec.
$_1V_{ro}$ = relative velocity of steam leaving first moving row of blades, ft./sec.
$_1V_{a0}$ = absolute velocity of steam entering first fixed row of blades, ft./sec.
$_1V_w$ = velocity of whirl in first stage, ft./sec.
a_1 = jet angle of nozzles.
$_1a_0$ = inlet angle of first fixed row of blades.
θ_1 = inlet angle of first moving row of blades.
$_1\theta_0$ = outlet angle of first moving row of blades.

For the second stage the prefix would be 2 and for the third, 3 and so on.

Work done.

In 1st velocity stage, $E_{b1} = \frac{u}{g}({}_1V_w)$, ft. lb. per lb. steam/sec.

,, 2nd ,, ,, $E_{b2} = \frac{u}{g}({}_2V_w)$, ,, ,, ,,

,, 3rd ,, ,, $E_{b3} = \frac{u}{g}({}_3V_w)$, ,, ,, ,,

Total work done, $E_b = \frac{u}{g}\Sigma V_w$,, ,, ,,

(5) Estimation of velocity coefficients in nozzles and blading.

For a full investigation into the velocity coefficients for nozzles, the reader is referred to the comprehensive series of experiments carried out by the *Steam Nozzles Research Committee. In the nozzle calculations the following values have been taken for the velocity coefficient, k_n.

Theoretical steam speed ($223{\cdot}7\sqrt{h_r}$) ft./sec.	500	750	1000	1250	1500	1750	2000
k_n	·962	·942	·942	·947	·948	·945	·942

For adiabatic heat drops in excess of 100 B.T.U.s,

* "Design and Construction of Steam Turbines", by H. M. Martin.

* Reports of the Steam Nozzles Research Committee. Transactions of Instn. of Mechanical Engineers, 1923, 1924, 1925, 1928 and 1930.

H. M. Martin's formula for nozzle efficiency has been used,

$$\eta_n = 102{\cdot}7 - {\cdot}06H_r$$

Equations for the velocity coefficients of impulse blading have been worked out from a series of curves given by *E. F. Church in which the percentage energy loss in blades is plotted on a base of relative velocity. For $\frac{3}{4}$in. and 1in. width blades the equations obtained were as follows:—

$$\tfrac{3}{4}\text{in. width } k_b = \sqrt{1 - \frac{18 + {\cdot}00545V_r}{100}}$$

$$1\text{in. width } k_b = \sqrt{1 - \frac{14 + {\cdot}00533V_r}{100}}$$

The following table shows the values of k_b for relative velocities from 600 to 3,000ft. per second.

V_r	k_b $\frac{3}{4}$in. width	k_b 1in. width
600	·8871	·9098
800	·881	·904
1000	·8748	·897
1200	·8682	·892
1400	·862	·886
1600	·8558	·880
1800	·849	·8735
2000	·843	·867
2200	·836	·8615
2400	·829	·8545
2600	·823	·848
2800	·816	·8415
3000	·809	·835

(6) Reaction Turbine Design Formulæ. Calculation of mean blade ring diameter.

Let W = weight of steam flow in pounds per second.

v_s = volume of steam at middle of expansion, cubic ft. per lb.

V = exit velocity of steam in feet per second.

D = mean blade ring diameter in inches.

l = effective blade height in inches.

θ = blade exit angle.

ρ = ratio of blade speed to steam speed.

Then,

$$\text{Annular area for flow} = \frac{\pi.D.l.\sin\theta}{144} \text{ sq. ft.}$$

$$\text{Quantity flowing per sec.} = \frac{V.\pi.D.l.\sin\theta}{144} \text{cu. ft.} = W.v_s$$

$$\therefore D = \frac{144\,W.v_s}{V.\pi.l.\sin\theta}$$

Now, $$\rho = \frac{\pi\,D\,N}{12 \times 60\,V} \text{ whence } V = \frac{\pi\,D\,N}{720\,\rho}$$

$$\therefore W.v_s = \frac{\pi^2 D^2 N.l.\sin\theta}{144 \times 720\,\rho}$$

$$D^2 = \frac{10510\,W.v_s.\rho}{N.l.\sin\theta}$$

$$\text{putting } \frac{\text{effective blade height}}{\text{mean ring diameter}} = \frac{l}{D} = z$$

$$D^3 = \frac{10510\,W.v_s.\rho}{N.z.\sin\theta}$$

$$\therefore D = \sqrt[3]{\frac{10510\,W.v_s.\rho}{N.z.\sin\theta}} \quad \ldots \quad \ldots \quad \ldots (3)$$

(7) Labyrinth Gland and Dummy Leakage Formulæ.

Several formulæ have been developed for determining the leakage from labyrinth glands and dummies. The formula used in the calculations is due to the late H. M. Martin, which is as follows:—

$$W = 68A\sqrt{\frac{p_1\left(1 - \frac{1}{x^2}\right)}{v_1(n + \log_e x)}} \quad \ldots \quad \ldots \quad \ldots (4)$$

where, W = weight of steam discharged in pounds per sec.

A = area available for steam flow in square feet.

n = number of throttlings on dummy or gland.

p_1 = initial steam pressure, lbs. per sq. in., absolute.

p_0 = exhaust ,, ,, ,,

v_1 = volume per lb. of steam at pressure p_1

x = ratio of initial pressure to exhaust pressure $= \frac{p_1}{p_0}$

(8) Turbine Design.

It will be appreciated that turbine manufacturers have various empirical and rapid methods of determining the principal dimensions of a set of turbines. These have been evolved from theory in the first place but have been simplified for rapid application and can be used with confidence by the turbine designer as he can make use of actual turbine performances to check his calculations. These methods are not available to the student of the subject and in what follows the writer endeavours to work out a design from first principles adopting certain coefficients from actual practice.

H.P. Turbines.

Each H.P. turbine is fitted with a byepass, the normal power with the byepass closed being 18,500 s.h.p. and the maximum power with the byepass open 19,500 s.h.p. on two screws.

In the design to be described, each H.P. turbine has one two-velocity stage impulse wheel and three reaction ahead expansions. The inlet steam pressure is 415lb. abs., 725° F. and at the first reaction expansion 270lb. abs., 645·9° F., the exhaust to the I.P. turbine being 117lb. abs., 495° F. For a feed temperature of 350° F. at full power, the turbine is bled at 174lb. abs. before the third expansion. With propeller revolutions 125 per minute, the turbine revolutions are 2,350, giving a gear ratio of 18·80 to 1.

I.P. Ahead Turbines.

Each I.P. turbine is all-reaction having five expansions, the revolutions being the same as for the H.P. turbines. The inlet pressure is 117lb. abs., 495° F. and exhaust pressure 20lb. abs.

H.P. Astern Turbines.

In each I.P. turbine an astern turbine is incorporated consisting of a three-velocity stage impulse wheel. The inlet steam pressure is 300lb. abs., 725° F. and exhaust pressure 40lb. abs.

* "Steam Turbines", by E. F. Church. McGraw-Hill Book Coy.

L.P. Ahead Turbines.

All-reaction type having eleven expansions, inlet steam pressure 20lb. abs., exhaust 28½in. vacuum on 30in. bar. Revolutions per minute, 1,700.

L.P. Astern Turbines.

Impulse reaction incorporated in the L.P. turbine casing having one two-velocity stage impulse wheel and five reaction expansions. Inlet steam pressure 40lb. abs., exhaust 28½in. vacuum.

The steam conditions were worked out from the H.ϕ diagram of the late Professor Callendar, the results being as shown in Tables 1 and 2.

TABLE 1.

H.P. TURBINES.

Expansion.	Pressure at entrance lb. sq. in. abs.	Total tempr. °F. and quality.	Volume cubic feet per lb.	Heat content B.T.U.'s per lb.	Volume at middle of expansion cubic ft./lb.
Impulse ...	415	725	1·674	1378	
1st reaction	270	645·9	2·350	1344·5	2·57
2nd „	215	606	2·86	1327·5	3·13
3rd „	174	570	3·415	1312	4·02
Exhaust ...	117	495	4·729	1278	

η_s impulse = 0·67 ; η_1 (reaction) = 0·751.

I.P. TURBINES.

Expansion.	Pressure at entrance lb. sq. in. abs.	Total tempr. °F. and quality.	Volume cubic feet per lb.	Heat content B.T.U.'s per lb.	Volume at middle of expansion cubic ft./lb.
1	117	495	4·729	1278	5·46
2	84	440	6·207	1253·7	7·15
3	60	395	8·280	1233·8	9·50
4	43	345	10·83	1210·8	12·60
5	30	295	14·67	1188·4	18·25
Exhaust ...	20	245	20·59	1166·0	

η_1 = ·752.

L.P. TURBINES.

Expansion.	Pressure at entrance lb. sq. in. abs.	Total tempr. °F. and quality.	Volume cubic feet per lb.	Heat content B.T.U.'s per lb.	Volume at middle of expansion cubic ft./lb.
1	20	245	20·59	1166·0	24·0
2	15·2	213·7	26·20	1151·0	30·0
3	11·7	q = ·991	32·87	1136·0	37·0
4	9·0	q = ·982	41·56	1121·0	48·2
5	6·7	q = ·974	54·61	1106·0	62·0
6	5·0	q = ·961	70·56	1091·0	81·6
7	3·7	q = ·952	94·25	1076·0	108·2
8	2·7	q = ·946	127·71	1061·0	146·2
9	2·0	q = ·939	162·90	1046·0	192·0
10	1·4	q = ·926	249·28	1031·0	276·5
11	1·0	q = ·916	305·0	1016·0	335·0
Exhaust ...	0·735	q = ·906	403·35	1000·0	

η_1 = ·80.

TABLE 2.

H.P. ASTERN TURBINES.

Expansion.	Pressure at entrance lb. sq. in. abs.	Total tempr. °F. and quality.	Volume cubic feet per lb.	Heat content B.T.U.'s per lb.	Volume at middle of expansion cubic ft./lb.
Impulse ...	300	725	2·280	1385	
Exhaust ...	40	484	13·89	1280·7	

η_s = 0·511.

L.P. ASTERN TURBINES.

Expansion.	Pressure at entrance lb. sq. in. abs.	Total tempr. °F. and quality.	Volume cubic feet per lb.	Heat content B.T.U.'s per lb.	Volume at middle of expansion cubic ft./lb.
Impulse ...	40	484	13·89	1280·7	
1	19·5	384	25·42	1235·0	32·5
2	11·1	325	42·16	1207·35	54·0
3	6·0	267	71·70	1179·7	97·0
4	3·1	207	130·44	1152·0	178·0
5	1·52	147	231·0	1124·0	314·0
Exhaust ...	0·735	0·998	444·3	1096·36	

η_s = ·67 impulse ; η_1 = ·615 reaction.

(9) Feed Heating Calculations.

Before the turbine design can be commenced, an estimate of the quantity of steam passing through each turbine must be made. The turbine consumption including interstage feed heating is required to enable this to be done.

Only the saturation temperature of the steam at the bled steam pressures is taken into account in the calculations and no allowance is made for superheat.

The specified feed temperature at maximum power of 19,500 s.h.p. is 350° F. and allowing a saturation temperature of 370° F., the bled steam pressure required is 174lb. per square inch absolute.

With a regenerative condenser the condensate temperature will be not more than 2° F. lower than the vacuum temperature. The specified vacuum is 28½ inches corresponding to 91·6° F., so that the condensate temperature may be taken as 89·6° F.

From each condenser the condensate is pumped by the water extraction pumps through their respective air ejectors and then through the drain cooler to the suction of the main turbo-driven feed pump. The increase of temperature across the air ejectors from previous installations is about 5·5° F. and across the drain cooler a further increase of 18° F. is normal. The temperatures will therefore be as follows :—

Vacuum 28½in. (30in. bar.)	= 91·6° F.
Condensate	= 89·6° F.
Leaving air ejectors	= 95° F. (say)
„ drain cooler	= 113° F.

The feed heaters therefore have to raise the feed water from 113° F. to 350° F. when the turbines are developing 19,500 s.h.p. on two screws. Three-stage feed heating was adopted with the following temperatures.

L.P. heater. Feed enters at 113° F. and leaves at 213° F.
I.P. „ „ „ 213° F. „ „ 300° F.
H.P. „ „ „ 300° F. „ „ 350° F.

To obtain these feed temperatures at outlet from the respective heaters the turbines were bled as follows:—

L.P. heater. 20lb. per sq. in. abs. = 228° F. saturation temp.
I.P. „ 84lb. „ „ = 315·3° F. „ „
H.P. „ 174lb. „ „ = 370° F. „ „

L.P. Heater.

This heater receives the exhaust steam from the main turbo-feed pump amounting to 6,000lb. per hour, also the vapour from the make-up feed evaporator of 6,530lb. per hour. Bled steam from turbines at 20lb. abs. and the combined drains from the H.P. and I.P. heaters.

I.P. Heater.

Receives bled steam from turbines at 84lb. abs. and drain from H.P. heater.

H.P. Heater.

Receives bled steam from turbines at 174lb. abs.

Estimated Steam Consumption.

Steam at H.P. turbine stop valves = 415lb. abs., 725° F.
Exhaust from L.P. turbines = 28½in. vacuum.
Adiabatic heat drop = 471 B.T.U.s per lb.
Overall internal efficiency from $H\phi$ diagram,

$$n_1 = \frac{378}{471} = 0{\cdot}802.$$

Allowing for dummy, gland, radiation, exhaust losses, friction and gearing losses the estimated turbine efficiency ratio has been taken as 0·74.

$$\text{Theoretical steam consumption} = \frac{2545}{471} = 5{\cdot}4\text{lb. per s.h.p./hr.}$$

$$\text{Estimated actual steam} \quad ,, \quad = \frac{5{\cdot}4}{0{\cdot}74} = 7{\cdot}3\text{lb. per s.h.p./hr.}$$

Estimated Total Evaporation without Feed Heating.

Main turbines = 19,500 × 7·3	= 142,350lb./hr.
Two air ejectors	= 1,500 „
Turbo-feed pump	= 6,000 „
Turbo-generator sets	= 10,310 „
Make-up feed	= 6,530 „
Other hotel services	= 5,000 „
	171,690 „

Let x = weight of steam bled to L.P. heater in lb. per hr.
y = „ „ I.P. „ „
z = „ „ H.P. „ „

Then,

Total feed $= 171{,}690 + x + y + z$ lb. per hr. ... (1)

Allowing 5% for radiation losses, we have for the L.P. heater,

Heat given out by steam
$$= (x + 6000 + 6530)L + (y + z)(315{\cdot}3 - 228)$$
$$= 960x + 12{,}028{,}800 + 87{\cdot}3y + 87{\cdot}3z \quad \ldots \quad \ldots (2)$$

Heat required by water
$$= \{(171{,}690 + x + y + z)(213 - 113)\}\,1{\cdot}05$$
$$= 105x + 105y + 105z + 18{,}027{,}450 \quad \ldots \quad \ldots (3)$$

Equating (2) and (3)
$$855x - 17{\cdot}7y - 17{\cdot}7z = 5{,}998{,}650 \quad \ldots \quad \ldots (4)$$

For the I.P. Heater.

Heat given out by steam
$$= y.L + z(370 - 315{\cdot}3) = 901{\cdot}7y + 54{\cdot}7z \quad \ldots (5)$$

Heat required by water
$$= \{(171{,}690 + x + y + z)(300 - 213)\}\,1{\cdot}05$$
$$= 91{\cdot}35x + 91{\cdot}35y + 91{\cdot}35z + 15{,}683{,}881 \quad \ldots (6)$$

From (5) and (6)
$$-91{\cdot}35x + 810{\cdot}35y - 36{\cdot}65z = 15{,}683{,}881 \quad \ldots (7)$$

For the H.P. Heater.

Heat given out by steam
$$= z.L = 858z \quad \ldots \quad \ldots \quad \ldots \quad \ldots \quad \ldots (8)$$

Heat required by water
$$= \{(171{,}690 + x + y + z)(350 - 300)\}\,1{\cdot}05$$
$$= 52{\cdot}5x + 52{\cdot}5y + 52{\cdot}5z + 9{,}013{,}725 \quad \ldots \quad \ldots (9)$$

From (8) and (9)
$$-52{\cdot}5x - 52{\cdot}5y + 805{\cdot}5z = 9{,}013{,}725 \quad \ldots \quad \ldots (10)$$

Solving for x, y and z from equations (4) (7) and (10) we obtain,

x = 7,716lb. per hr. at 20lb. abs. to L.P. heater
y = 20,814lb. „ „ 84lb. „ I.P. „
z = 13,050lb. „ „ 174lb. „ H.P. „

The above amounts are required for feed heating. In addition, 7,510lb. per hour has to be bled at 84lb. absolute for the L.P. evaporator, so that the total bled steam required from the turbines at 19,500 s.h.p. is as follows:—

7,716lb. per hour to L.P. heater
28,324 „ (20,814lb. to I.P. heater + 7,510 to L.P. evaporator)
13,050 „ to H.P. heater.

From the heat drop calculations (Table 1) the live steam equivalents for each bled steam point have been estimated as follows:—

$$\text{At 174lb. abs.} = \frac{312}{378} = {\cdot}825 \times \text{amount of bled steam}$$

$$\text{At 84lb. abs.} = \frac{254}{378} = {\cdot}672 \times \quad ,, \quad ,,$$

$$\text{At 20lb. abs.} = \frac{166}{378} = {\cdot}439 \times \quad ,, \quad ,,$$

Estimated Total Evaporation with Feed Heating.
19,500 s.h.p.

Main turbines = 19,500 × 7·3	= 142,350lb./hr.
H.P. heater = ·825 × 13,050	= 10,766 „
I.P. heater and L.P. evaporator = ·672 × 28,324	= 19,033 „
L.P. heater = ·439 × 7,716	= 3,387 „
Air ejectors	= 1,500 „
Turbo-feed pump	= 6,000 „
Turbo-generator sets	= 10,310 „
Make-up feed	= 6,530 „
Other hotel services	= 5,000 „
	204,876 „

Total steam at two H.P. turbine nozzles boxes	= 175,536lb./hr.
	= 87,768 „ each
	= 24·38lb./sec. „

Normal Full Power=18,500 s.h.p.

Feed heating calculations were also made at 18,500 s.h.p. The total steam at each H.P. turbine worked out at 83,364lb. per hour or 23·16lb./sec.

In estimating the weights of working steam, the dummy leakage in the H.P. turbines has been taken as $4\frac{1}{2}\%$, in the I.P. turbines as $3\frac{1}{2}\%$ and in the L.P. turbines as $2\frac{1}{2}\%$.

Estimated Weights of Working Steam per hour.

At each H.P. turbine nozzle box at 19,500 s.h.p.	=87,768lb./hr.
At each H.P. turbine nozzle box at 18,500 s.h.p.	=83,364 „
At each H.P. turbine 1st reaction stage at 18,500 s.h.p., byepass shut 83364 – $4\frac{1}{2}\%$	=79,613 „
At each H.P. turbine 2nd reaction stage at 19,500 s.h.p., byepass open =87768 – $4\frac{1}{2}\%$	=83,818 „
At each H.P. turbine 3rd reaction stage at 19,500 s.h.p., byepass open =83818 – 6525	=77,293 „
Dummy leakage to H.P. exhaust through equalising pipe ($4\frac{1}{2}\%$)	= 3,950 „
Steam passing to each I.P. turbine	=81,243 „
At each I.P. turbine 1st reaction stage at 19,500 s.h.p. =81,243 – $3\frac{1}{2}\%$	=78,400 „
At each I.P. turbine 2nd reaction stage =78,400 – 14.162 (bled to I.P. heater and L.P. evaporator)	=64,238 „
At each I.P. turbine 3rd, 4th and 5th reaction stage	=64,238 „
Dummy leakage to I.P. exhaust ($3\frac{1}{2}\%$) through equalising pipe	= 2,843 „
Exhaust from each I.P. turbine	=67,081 „
Bled steam to L.P. heater from each I.P. turbine exhaust	= 3,858 „
Steam to each L.P. turbine	=63,223 „
Working steam in each L.P. turbine at 19,500 s.h.p. =63,223 – $2\frac{1}{2}\%$	=61.643 „
Dummy leakage to condenser ($2\frac{1}{2}\%$)	= 1,580 „
Steam passing to each condenser	=63,223 „

(10) H.P. Impulse Turbine Design.

H.P. Impulse Wheel. Two rows.

Initial steam pressure=415lb. abs., 725° F.

Outlet pressure from nozzles=270lb. abs., 645·9° F.

Adiabatic heat drop=50 B.T.U.s per lb.

Theoretical steam speed$=223{\cdot}7\sqrt{50}=1{,}582$ ft./sec.

Assuming $\rho=0{\cdot}25$, blade speed$=\cdot25\times1.582=395{\cdot}5$ft./sec.

Revolutions per minute=2,350.

$$\therefore \text{ Mean ring diameter}=\frac{720\times395{\cdot}5}{\pi\times2{,}350}=38{\cdot}56\text{in., say } 38\tfrac{1}{2}\text{in.}$$

With a nozzle angle of 19° and blade exit angles of the first moving row 22°, first fixed row 27° and second moving row 35°, the inlet angles from the velocity diagram are first moving row 25°, first fixed row 32° and second moving row 45°.

The blade heights are determined by the axial velocity of the steam at the exit edges of the respective blades, the velocity diagram being drawn by adopting the appropriate velocity coefficients, k_n and k_b, given under section (5) above. If y_n be the axial velocity of the steam issuing from the nozzles, y_1, y_2, y_3, etc. at the exit edges of successive blades, then if l_n, l_1, l_2, l_3, etc. be the corresponding radial lengths at exit from the nozzles and blades, it may be proved that,

$$l_n=\frac{\text{constant}}{y_n}$$
$$l_1=\frac{\text{constant}}{y_1}$$
$$l_2=\frac{\text{constant}}{y_2}$$
$$l_3=\frac{\text{constant}}{y_3}$$

whence,

$$\frac{l_1}{l_n}=\frac{y_n}{y_1}$$
$$\frac{l_2}{l_n}=\frac{y_n}{y_2}$$
$$\frac{l_3}{l_n}=\frac{y_n}{y_3}$$

In practice the minimum blade height l_1 is generally not less than 1·5% to 2% of the mean blade ring diameter and l_n not less than $\frac{2}{3}$ of l_1.

From the velocity diagram, not illustrated, the following results were obtained.

In the first row velocity diagram for a theoretical steam speed of 1,582ft. per second the velocity coefficient k_n may be taken as 0·948 from section (5) above, giving ${}_1V_a=1{,}499{\cdot}7$ft./sec. From the diagram ${}_1V_r=1{,}130$ft./sec., so that k_b ($\frac{3}{4}$in. blades)$=0{\cdot}87$, hence ${}_1V_{ro}=983$ft./sec. The diagram gave ${}_1V_{ai}=635$, $y_n=486$, $y_1=369$ and ${}_1V_w=1{,}932$ft./sec.

For the second row velocity diagram, ${}_2V_{ai}=635\times\cdot88=559$ft./sec., ${}_2V_r=277{\cdot}5$, $k_b=\cdot8925$, $\therefore\ {}_2V_{ro}=248$: ${}_2V_{ao}=238{\cdot}6$, $y_2=253{\cdot}5$, $y_3=142$ and ${}_2Vw=306$ft./sec.

Minimum effective height of first row of blades$=\cdot015\times38{\cdot}5=\cdot5775$in., say 0·6in.$=l_1$. Then $l_n=\frac{2}{3}\times0{\cdot}6=0{\cdot}4$in.

$$l_2=\frac{y_n.l_n}{y_2}=\frac{486\times\cdot4}{253{\cdot}5}=0{\cdot}767 \text{ make } \tfrac{7}{8}\text{in.}$$
$$l_3=\frac{y_n.l_n}{y_3}=\frac{486\times\cdot4}{142}=1{\cdot}368 \text{ make } 1\tfrac{1}{2}\text{in.}$$

H.P. Nozzle Plate Design.

The total steam passing through the nozzles of each turbine at 19,500 s.h.p. is 87,768lb. per hour or 24·38lb. per second. The issuing velocity at nozzle exit is 1,499·7ft./sec. as stated above. The volume per pound of steam at 270lb. abs., 645·9° F., is 2·35 cubic feet; hence,

$$\text{Exit area of nozzles } A_o=\frac{144W.v_s}{{}_1V_a}=\frac{144\times24{\cdot}38\times2{\cdot}35}{1{,}499{\cdot}7}=5{\cdot}5 \text{ sq. in.}$$

If x = length of nozzle arc on pitch circle in inches
n = number of nozzles
P = gross pitch of nozzles in inches
c = thickness coefficient, ·064in. vanes, 19° angle = 0·8693 for a gross pitch of 1·5in.
l_n = radial depth of nozzle
α = nozzle angle, in present design 19°

then, $x = n.P = \dfrac{A_o}{c.l_n \sin\alpha} = \dfrac{5{\cdot}5}{{\cdot}8693 \times {\cdot}40 \times {\cdot}3256} = 48{\cdot}55$in.

Taking the gross pitch P as 1·5 inches,

Number of nozzles $n = \dfrac{48{\cdot}55}{1{\cdot}5} = 32{\cdot}36$, make 33 nozzles.

Width of each nozzle $= {\cdot}8693 \times 1{\cdot}5 \sin 19° = 0{\cdot}4246$in.
Area ,, $= {\cdot}4246 \times 0{\cdot}4 = 0{\cdot}16984$ sq. in.
Gross length of arc $= 33 \times 1{\cdot}5 = 49{\cdot}5$in.

The nozzle plate is shown in Fig. 3.

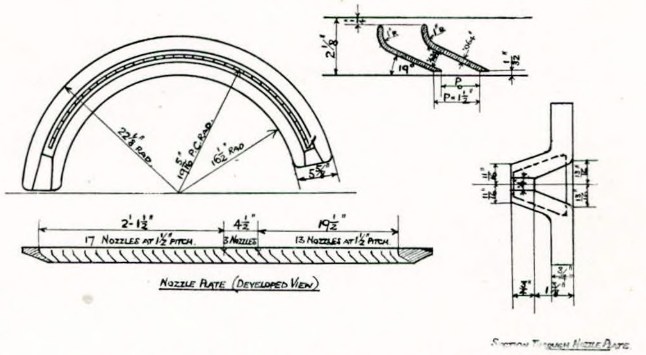


FIG. 3.—*H.P. ahead nozzle plate.*

(11) H.P. Reaction Turbine Design.

In the application of equation 3, section 6 above, for the mean blade ring diameter, it will be realised that assumptions have to be made with regard to the values of ρ and z. In the present design the first reaction stage has to pass the amount of steam required to develop 9,250 s.h.p. in each set of turbines, including feed heating and bled steam to the L.P. evaporator. This quantity from the feed heating calculations above is 83,364lb. per hour, less the dummy leakage, which has been taken as 4½%, leaving 79,613lb. per hour or 22·115lb. per sec.

1st Expansion.

$W = 22{\cdot}115$lb./sec.; Volume at middle of expansion = 2·57 cu. ft./lb.

$\theta = 20°$: assuming $\rho = 0{\cdot}75$ and $z = {\cdot}06$

$$D_1 = \sqrt[3]{\frac{10{,}510 \times 22{\cdot}115 \times {\cdot}75 \times 2{\cdot}57}{2{,}350 \times {\cdot}06 \times {\cdot}342}} = \sqrt[3]{9{,}290} = 21{\cdot}03\text{in.}$$

Effective blade height $= {\cdot}06 \times 21{\cdot}03 = 1{\cdot}2618$, say 1¼in.
Effective drum diameter $= {\cdot}94 \times 21{\cdot}03 = 19{\cdot}768$, say 19¾in.
Mean blade ring diameter $= 19{\cdot}75 + 1{\cdot}25 = 21$in.

Mean blade speed $= \dfrac{65{\cdot}9734 \times 2{,}350}{720} = 215{\cdot}4$ft./sec.

Area through blading $= \dfrac{65{\cdot}9734 \times 1{\cdot}25 \times {\cdot}342}{144} = {\cdot}1959$ sq. ft.

Mean steam speed $= \dfrac{22{\cdot}115 \times 2{\cdot}57}{{\cdot}1959} = 290$ft./sec.

Mean value of $\rho = \dfrac{215{\cdot}4}{290} = 0{\cdot}742$

$z = \dfrac{1{\cdot}25}{21} = 0{\cdot}0595$

2nd Expansion.

At this expansion with the byepass open and each set of turbines developing 9,750 s.h.p., the total steam passing through the 2nd expansion is estimated as 83,818lb. per hour, or 23·283lb. per second. The volume at middle of expansion from Table 1 is 3·13 cubic feet per lb. Blade exit angle $\theta = 20°$. Assuming $\rho = 0{\cdot}75$ and $z = {\cdot}075$,

$$D_2 = \sqrt[3]{\frac{10{,}510 \times 23{\cdot}283 \times {\cdot}75 \times 3{\cdot}13}{2{,}350 \times {\cdot}075 \times {\cdot}342}} = \sqrt[3]{9{,}530} = 21{\cdot}2\text{in.}$$

Effective blade height $= {\cdot}075 \times 21{\cdot}2 = 1{\cdot}59$, say $1\frac{9}{16}$in.
Effective drum diameter $= {\cdot}925 \times 21{\cdot}2 = 19{\cdot}61$in.
Keeping the effective drum diameter 19¾in., mean ring diameter $= 21\frac{5}{16}$in.

The blade speed works out at 218·5ft./sec., area through blading 0·2483 sq. ft., mean steam speed 293·4ft./sec., mean value of $\rho = {\cdot}745$ and $z = {\cdot}0733$.

3rd Expansion.

Before this expansion, 6,525lb. of steam is bled to the H.P. heater, leaving 77,293lb. per hour, or 21·47 lb./sec. to pass through the 3rd expansion when each set of turbines is developing 9,750 s.h.p. From Table 1, the volume at middle of this expansion is 4·02 cubic ft. per lb. Blade angle $\theta = 20°$. Assuming $\rho = {\cdot}75$ and $z = {\cdot}085$,

$$D_3 = \sqrt[3]{\frac{10{,}510 \times 21{\cdot}47 \times {\cdot}75 \times 4{\cdot}02}{2{,}350 \times {\cdot}085 \times {\cdot}342}} = \sqrt[3]{9{,}963} = 21{\cdot}52\text{in.}$$

Effective blade height $= {\cdot}085 \times 21{\cdot}52 = 1{\cdot}8292$, say $1\frac{13}{16}$in.
Effective drum diameter $= {\cdot}915 \times 21{\cdot}52 = 19{\cdot}6908$in.
Keeping the effective drum diamater 19¾in., mean ring diameter $= 21\frac{9}{16}$in.

The blade speed = 221ft./sec., area through blading = 0·2915 sq. ft., mean steam speed = 296·2ft./sec., mean value of $\rho = 0{\cdot}746$ and $z = 0{\cdot}084$.

Number of Rows.

η_1 (reaction) = 0·751. Estimated efficiency ratio = ·751 − ·06 = ·691. K from Fig. 1 = 110,000. Mean ring diameter = 21·2917in.; N = 2,350 revs. per min.

$\therefore$ Number of rows $= \dfrac{110{,}000}{2{\cdot}12917^2 \times 23{\cdot}5^2} = 43{\cdot}9.$

Made 14 rows each expansion.

Schedule of H.P. Turbine Blading.

Two H.P. Turbines.

Impulse Nozzles.

33 nozzles in each H.P. turbine. 19° discharge angle, 1½in. pitch, ·064in. vane thickness, 0·40in. height.
Impulse wheel, 38½in. mean diameter.

Impulse Blading.

1st moving row :
0·60in. effective height. Inlet angle 25°, exit 22°
1st fixed row :
0·875in. ,, ,, 32°, ,, 27°
2nd moving row :
1·50in. ,, ,, 45°, ,, 35°

Reaction Blading. End-tightened $\frac{1}{4}$in. stand out of packing.

Expansion	1	2	3
Effective drum diameter...	$19\frac{3}{4}$in.	$19\frac{3}{4}$in.	$19\frac{3}{4}$in.
Effective blade height ...	$1\frac{1}{4}$in.	$1\frac{9}{16}$in.	$1\frac{13}{16}$in.
Mean ring diameter ...	21in.	$21\frac{5}{16}$in.	$21\frac{9}{16}$in.
Number of rows	14	14	14
θ	20°	20°	20°

Byepass for maximum power before 2nd expansion and bled steam leak off to H.P. heater before 3rd expansion.

(12) H.P. Turbine Dummy Design.

The function of the dummy in conjunction with the adjusting block is to balance the steam thrust on the rotor blading and to put the rotor in equilibrium. The type of dummy usually fitted to H.P. and I.P. turbines is of the contact type, the axial clearance between the face of the fin and the collar on the rotor being the same as the end-tightened blading clearance. In the L.P. turbine a radial dummy is fitted, the radial clearance between rotor fin and casing and casing fin and rotor being usually about $\frac{30}{1000}$in. An equalising pipe is fitted between the exhaust end of the dummy and the exhaust end of the turbine to carry off the steam leakage from the dummy and to balance the pressure at each end of the rotor.

The dummy diameter is calculated by taking the difference between the initial and final pressure in an expansion and multiplying this by the annular area of the rotor blading, the product being the axial thrust in the expansion. This is done for each expansion of the turbine, the sum being the total thrust. It is assumed that half this total thrust is taken by the fixed blades and half by the rotor blades. Dividing the thrust on the rotor blading by the difference between the initial and leak-off pressure on the dummy gives the annular area of the dummy in excess of the rotor drum, from which the dummy diameter is immediately found.

Calculation of H.P. Turbine Dummy.

Annular area of rotor blading at

$$\text{1st expansion} = \frac{\pi}{4}(22{\cdot}25^2 - 19{\cdot}75^2) = 82{\cdot}47 \text{ sq. in.}$$

$$\text{2nd expansion} = \frac{\pi}{4}(22{\cdot}875^2 - 19{\cdot}75^2) = 104{\cdot}62 \text{ sq. in.}$$

$$\text{3rd expansion} = \frac{\pi}{4}(23{\cdot}375^2 - 19{\cdot}75^2) = 122{\cdot}78 \text{ sq. in.}$$

Taking the initial and final pressures at each expansion from Table 1, and assuming half the total thrust to be taken by the rotor blading and half by the fixed blades,

$$\text{thrust on rotor} = \tfrac{1}{2}\{(270-215)82{\cdot}47 + (215-174)104{\cdot}62 + (174-117)122{\cdot}78\} = 7{,}912\text{lb.}$$

Apportioning the thrust, half to the dummy and half to the adjusting block, thrust to be balanced by the dummy = 3,956lb.

Mean pressure on dummy = (270 − 117) = 153lb. per sq. in.

Annular area required in excess

$$\text{of rotor drum} = \frac{3{,}956}{153} = 25{\cdot}85 \text{ sq. in.}$$

Area of $19\frac{3}{4}$in. diameter H.P. rotor drum = 306·35 „

Dummy area required = 332·20 „

Making the mean dummy diameter $20\frac{41}{64}$in. (334·6 sq. in.), load taken by dummy = 4,322lb., leaving 3,590lb. to be taken by the adjusting block. Allowing a loading of 80lb. per sq. in., area of thrust pads = 44·9 sq. in., say 45 sq. in. on 6 pads = 7·5 sq. in. per pad.

Calculation of number of fins required on dummy.

From equation 4, section 7 above,

$$W = 68A\sqrt{\frac{p_1\left(1-\frac{1}{x^2}\right)}{v_1(n+\log_e x)}}$$

Here, $p_1 = 270$lb. sq. in. abs.; $p_o = 117$lb. sq. in. abs; $x = \frac{p_1}{p_o} = 2{\cdot}307$; $\log_e x = 0{\cdot}836$; $v_1 = 2{\cdot}35$ cubic ft. per lb. Axial clearance 0·02in.

$$\text{Area available for steam flow} = \frac{\pi \times 20{\cdot}6406 \times {\cdot}02}{144} = {\cdot}0090 \text{ sq. ft.}$$

Allowing a permissible leakage through dummy of $4\frac{1}{2}$%,

$$W = \frac{{\cdot}045 \times 87{,}768}{3{,}600} = 1{\cdot}0971\text{lb. per second.}$$

From (3) above,

$$n + {\cdot}836 = \frac{68^2\left(1-\frac{1}{2{\cdot}307^2}\right)270 \times {\cdot}0090^2}{1{\cdot}0971^2 \times 2{\cdot}35} = 29{\cdot}02$$

Number of throttlings, $n = 29{\cdot}02 - {\cdot}836 = 28{\cdot}184$.

Making the number of throttlings 30, the leakage from dummy would be 1·064lb. per sec. or $\frac{1{\cdot}064}{24{\cdot}38} = 4{\cdot}366$%.

Hence fit 30 fins in casing and 30 collars on rotor.

A drawing of the dummy is shown in Fig. 4.

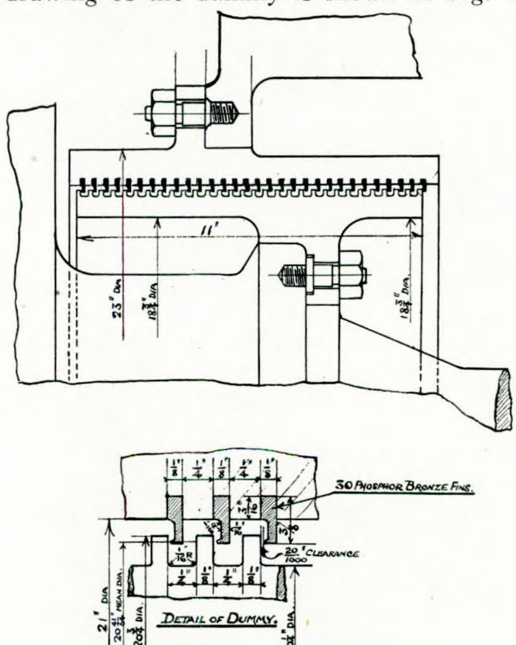


FIG. 4.—*H.P. turbine dummy.*

(13) H.P. Turbine Labyrinth Gland Design.

For high-pressure turbines the gland is usually made with two steam pockets Fig. 5, and sometimes with three, although in the writer's opinion there is little advantage in the three-pocket type. In the present design a two-pocket gland was adopted.

In the latest designs the steam to glands is supplied from a collector, one for each set of turbines, the collector being maintained at a pressure of 3 to 5lb. gauge by steam supplied as required from the auxiliary steam range. When warming up preparatory to starting the turbines, only the outer pockets, where two or more pockets are fitted, are supplied with steam via the

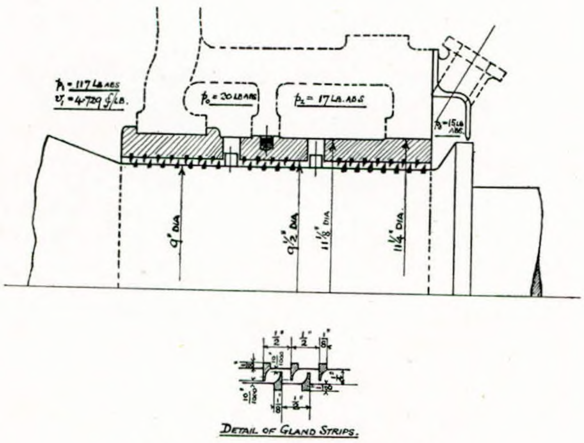


FIG. 5.—*H.P. turbine gland.*

collector by the auxiliary steam range, but under running conditions the steam to the outer pockets is supplied from the inner pockets via the collector.

Calculation of the Number of Throttlings Required.

In the present design the diameter of the rotor shaft in way of the gland was made 9in. The standard fins stand out from the shaft and casing $\frac{1}{4}$in., the radial clearance being $\frac{8}{1000}$in. The mean diameter of the gland is therefore $9\frac{1}{4}$in., giving an area for flow of 0·00202 sq. ft. For the inner pocket the internal pressure $p_1 = 117$lb. per sq. in. absolute, the volume per lb, v_1, being 4·729 cubic ft.

The leak-off pressure from the inner pocket to the collector has been taken as 20lb. absolute$=p_o$, whence $x=\frac{p_1}{p_o}=5{\cdot}85, \frac{1}{x^2}=0{\cdot}02925$ and $\log_e x=1{\cdot}767$. Allowing a permissible leakage of 0·17lb. per sec. between the inner side of gland and the inner pocket,

$$0{\cdot}17 = 68 \times {\cdot}00202\sqrt{\frac{117(1-{\cdot}02925)}{4{\cdot}729(n+1{\cdot}767)}}$$

whence $n=13{\cdot}933$, say 14 throttlings. Hence, fit 7 fins in casing and 7 fins in rotor.

For the number of throttlings required between the inner and outer pockets we have, $p_1=20$lb. abs., volume per lb.$=20{\cdot}06$ cubic feet. The pressure in the outer pocket has been taken as 17lb. abs., whence $x=1{\cdot}177$, $\frac{1}{x^2}=0{\cdot}722$ and $\log_e x=0{\cdot}1630$. Allowing a permissible leakage of 0·025lb. per second between the two pockets, we have

$$n+{\cdot}1630=\frac{68^2 \times {\cdot}00202^2 \times 20(1-{\cdot}722)}{20{\cdot}06 \times {\cdot}025^2}$$

whence $n=8{\cdot}237$, say 8 throttlings. Fit 4 fins in casing and 4 in rotor.

For the outer gland, the outer pocket pressure has been taken as 17lb. absolute, the volume per lb. being 23·35 cubic feet. The external pressure will be 15lb. absolute. $x=\frac{17}{15}=1{\cdot}131$, whence $\frac{1}{x^2}=0{\cdot}78$ and $\log_e x=0{\cdot}123$. The area for flow at each throttling will be 0·00202 sq. ft. and allowing a leakage of 0·015lb. per sec. into the engine room,

$$n+{\cdot}123=\frac{68^2 \times {\cdot}00202^2 \times 17(1-{\cdot}78)}{23{\cdot}35 \times {\cdot}015^2}=13{\cdot}41$$

$n=13{\cdot}287$, say 14 throttlings$=7$ fins in rotor and 7 fins in casing.

A drawing of the gland is shown in Fig. 5 and a sectional arrangement of the H.P. turbine in Fig. 6.

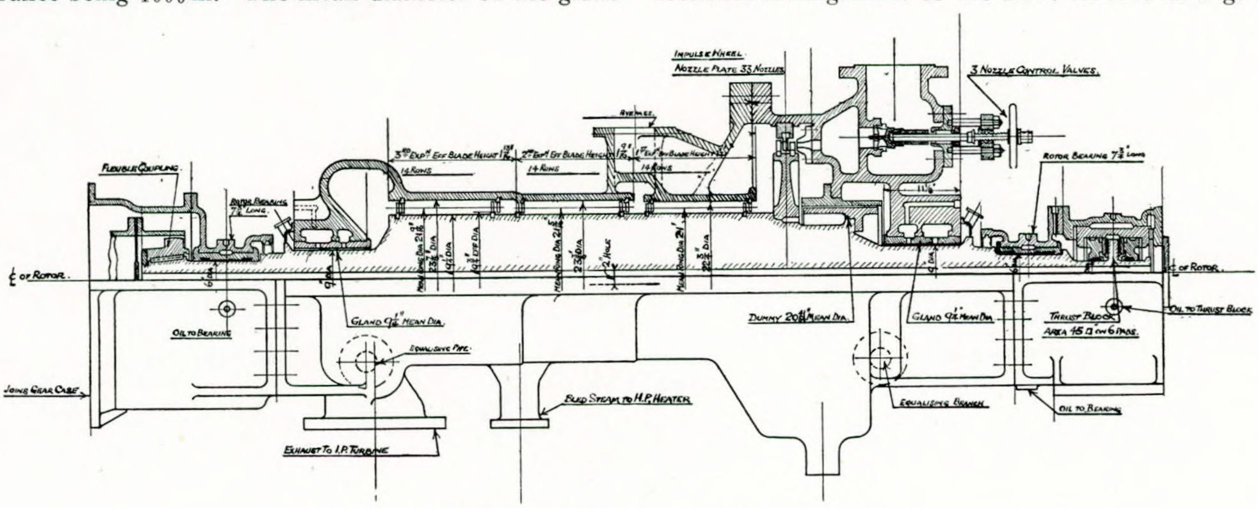


FIG. 6.—*Sectional arrangement of H.P. turbine.*

(14) I.P. Ahead Turbine Design.

The working steam at the first expansion of each I.P. turbine when developing the maximum power of 19,500 s.h.p. is estimated as 78,400lb. per hour, or 21·78lb. per second.

1st Expansion.

$W=21{\cdot}78$lb./sec.: Volume at middle of expansion from Table 1 = 5·46 cubic ft. per lb. Blade exit angle $\theta=20°$. Assuming $\rho=\cdot75$ and $z=0{\cdot}07$

$$D_1=\sqrt[3]{\frac{10{,}510\times21{\cdot}78\times5{\cdot}46\times\cdot75}{2{,}350\times\cdot07\times\cdot342}}=\sqrt[3]{16{,}670}=25{\cdot}54,\text{ say }25\tfrac{1}{2}\text{in.}$$

Effective blade height $=\cdot07\times25{\cdot}54=1{\cdot}788$, say $1\frac{3}{4}$in.

„ drum diameter $=25\frac{1}{2}-1\frac{3}{4}=23\frac{3}{4}$in.

$$\text{Blade speed}=\frac{80{\cdot}1106\times2{,}350}{720}=261{\cdot}5\text{ft./sec.}$$

Area through blading =

$$\frac{80{\cdot}1106\times1{\cdot}75\times\cdot342}{144}=0{\cdot}333\text{ sq. ft.}$$

Mean steam speed =

$$\frac{21{\cdot}78\times5{\cdot}46}{0{\cdot}333}=357{\cdot}3\text{ft./sec.};\ \rho=0{\cdot}732;\ z=0{\cdot}0686.$$

2nd Expansion.

Before this expansion 14,162lb. of steam is bled from each I.P. turbine to the I.P. heater and L.P. evaporator, leaving 78,400 – 14,162 = 64,238lb. per hour or 17·84lb./sec. The volume at middle of expansion is 7·15 cubic ft. per lb., blade exit angle $\theta=20°$. Assuming $\rho=0{\cdot}75$ and $z=\cdot075$.

$$D_2=\sqrt[3]{\frac{10{,}510\times17{\cdot}84\times7{\cdot}15\times\cdot75}{2{,}350\times\cdot075\times\cdot342}}=\sqrt[3]{16{,}680}=25{\cdot}55\text{in.}$$

Effective blade height $=\cdot075\times25{\cdot}55=1{\cdot}917$, say $1\frac{7}{8}$in.

„ drum diameter $=\cdot925\times25{\cdot}55=23{\cdot}633$in.

Making the effective drum diameter $23\frac{3}{4}$in., mean ring diameter $=23\frac{3}{4}+1\frac{7}{8}=25\frac{5}{8}$in.

The blade speed = 263ft./sec., area through blading 0·3585 sq. ft., mean steam speed = 356ft./sec., $\rho=0{\cdot}739$ and $z=0{\cdot}0731$.

3rd Expansion.

$W=17{\cdot}84$lb./sec. Volume at middle of expansion 9·5 cubic ft./lb., $\theta=20°$. Assuming $\rho=0{\cdot}75$ and $z=\cdot093$.

$$D_3=\sqrt[3]{\frac{10{,}510\times17{\cdot}84\times9{\cdot}5\times\cdot75}{2{,}350\times\cdot093\times\cdot342}}=\sqrt[3]{17{,}875}=26{\cdot}15\text{in.}$$

Effective blade height $=\cdot093\times26{\cdot}15=2{\cdot}43$, say $2\frac{3}{8}$in.

„ drum diameter $=\cdot907\times26{\cdot}15=23{\cdot}718$, say $23\frac{3}{4}$in.

Mean ring diameter $=23\frac{3}{4}+2\frac{3}{8}=26\frac{1}{8}$in.

The blade speed = 268ft./sec.; area through blading = 0·4625 sq. ft.; mean steam speed = 366ft./sec. $\rho=0{\cdot}7325$; $z=0{\cdot}091$.

4th Expansion.

$W=17{\cdot}84$lb./sec. Volume at middle of expansion = 12·6 cubic ft. per lb. $\theta=20°$. Assuming $\rho=0{\cdot}75$ and $z=0{\cdot}115$.

$$D_4=\sqrt[3]{\frac{10{,}510\times17{\cdot}84\times12{\cdot}6\times\cdot75}{2{,}350\times\cdot115\times\cdot342}}=\sqrt[3]{19{,}175}=26{\cdot}75\text{in.}$$

Effective blade height $=\cdot115\times26{\cdot}75=3{\cdot}076$, say 3in.

„ drum diameter $=\cdot885\times26{\cdot}75=23{\cdot}674$in.

Making the effective drum diameter $23\frac{3}{4}$in., mean ring diameter $=26\frac{3}{4}$in.

The blade speed = 274·5ft./sec., area through blading 0·5999 sq. ft., mean steam speed = 375·3 ft./sec. $\rho=0{\cdot}731$; $z=0{\cdot}112$.

5th Expansion.

$W=17{\cdot}84$lb./sec. Volume at middle of expansion = 18·25 cubic ft./lb. $\theta=20°$. Assuming $\rho=0{\cdot}75$ and $z=0{\cdot}150$.

$$D_5=\sqrt[3]{\frac{10{,}510\times17{\cdot}84\times18{\cdot}25\times\cdot75}{2{,}350\times\cdot150\times\cdot342}}=\sqrt[3]{21{,}300}=27{\cdot}71\text{in.}$$

Effective blade height $=\cdot150\times27{\cdot}71=4{\cdot}15$, say $4\frac{1}{8}$in.

„ drum diameter $=\cdot850\times27{\cdot}71=23{\cdot}56$in.

Making the effective drum diameter $23\frac{3}{4}$in., mean blade ring diameter $=27\frac{7}{8}$in.

The blade speed = 286ft./sec., area through blading = 0·8580 sq. ft., mean steam speed = 380 ft./sec.; $\rho=0{\cdot}752$; $z=0{\cdot}148$.

Number of Rows of Blades.

From Table 1, $n_1=0{\cdot}752$. Estimated efficiency ratio $=\cdot752-\cdot06=\cdot692$. K from Fig. 1 = 113,000. Mean ring diameter = 26·375in.

$$\text{Number of rows } n=\frac{113{,}000}{2{\cdot}6375^2\times23{\cdot}5^2}=29{\cdot}42,\text{ say 30 rows.}$$

Make each expansion 6 rows.

Schedule of I.P. Ahead Turbine Blading.

Two I.P. turbines. End-tightened blading $\frac{1}{4}$in. stand out of packing.

Expansion	1	2	3	4	5
Effective drum diam.	$23\frac{3}{4}$in.	$23\frac{3}{4}$in.	$23\frac{3}{4}$in.	$23\frac{3}{4}$in.	$23\frac{3}{4}$in.
„ blade height	$1\frac{3}{4}$in.	$1\frac{7}{8}$in.	$2\frac{3}{8}$in.	3in.	$4\frac{1}{8}$in.
Mean blade ring diam.	$25\frac{1}{2}$in.	$25\frac{5}{8}$in.	$26\frac{1}{8}$in.	$26\frac{3}{4}$in.	$27\frac{7}{8}$in.
Number of rows ...	6	6	6	6	6
Blade exit angle, θ ...	20°	20°	20°	20°	20°

(15) I.P. Turbine Dummy Design.

The diameter of the 1st expansion $=25\frac{1}{2}$in. $+1\frac{3}{4}$in. $=27\frac{1}{4}$in.

„ „ 2nd „ $=25\frac{5}{8}$in. $+1\frac{7}{8}$in. $=27\frac{1}{2}$in.

„ „ 3rd „ $=26\frac{1}{8}$in. $+2\frac{3}{8}$in. $=28\frac{1}{2}$in.

„ „ 4th „ $=26\frac{3}{4}$in. $+3$in. $=29\frac{3}{4}$in.

„ „ 5th „ $=27\frac{7}{8}$in. $+4\frac{1}{8}$in. $=32$in.

Annular area 1st expansion × difference in pressure $=(583{\cdot}21-443{\cdot}01)33=$ 4,626·60lb.

„ 2nd „ × difference in pressure $=(593{\cdot}96-443{\cdot}01)24=$ 3,622·80lb.

„ 3rd „ × difference in pressure $=(637{\cdot}94-443{\cdot}01)17=$ 3,313·81lb.

„ 4th „ × difference in pressure $=(695{\cdot}13-443{\cdot}01)13=$ 3,277·56lb.

„ 5th „ × difference in pressure $=(804{\cdot}25-443{\cdot}01)10=$ 3,612·40lb.

Total thrust = 18,453·17lb.

$$\text{Thrust taken by rotor blading}=\frac{18{,}453{\cdot}17}{2}=9{,}226{\cdot}585,$$

say 9,227lb.

Making the I.P. adjusting block identical with the H.P. turbine adjusting block and allowing a pressure per square inch of 80lb., the thrust in the direction of steam flow taken by the block $=80\times 45=3{,}600$lb., leaving 5,627lb. to be taken by the dummy.

Mean pressure acting on dummy $=(117-20)=97$lb.

$$\therefore \text{ Annular area of dummy } =\frac{5{,}627}{97}= 58 \text{ sq. in.}$$

$$\text{Area of } 23\tfrac{3}{4}\text{in. diameter rotor drum} = 443{\cdot}01 \quad ,,$$

$$\text{Dummy area required} = 501{\cdot}01 \quad ,,$$

Make dummy mean diameter $=25\frac{17}{64}$ (501·3 sq. in.).

The dummy is of the axial type, the clearance being $\frac{20}{1000}$ in. The initial steam pressure is 117lb. abs. and leak off pressure 20lb. abs., so that $x=\frac{117}{20}=5{\cdot}85$, $\frac{1}{x^2}={\cdot}02925$ and $\log_e x=1{\cdot}767$. The volume per lb. of steam at the initial pressure is 4·729 cubic feet. With a permissible dummy leakage of $3\frac{1}{2}\%$ the number of

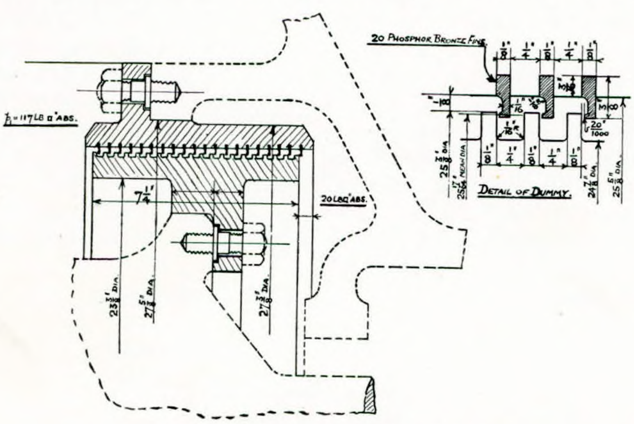


FIG. 7.—*I.P. turbine dummy.*

throttlings works out at 20, so that 20 fins are required in the casing and 20 collars on dummy rotor.

The I.P. dummy is shown in Fig. 7.

(16) I.P. Turbine Ahead and Astern Gland Design.

This gland is situated between the I.P. ahead turbine and the H.P. astern turbine. The pocket is connected to the condenser which ensures that the astern turbines are not subjected to any steam pressure due to leakage through the gland. When going astern similar conditions apply to the ahead turbine.

The diameter of the rotor shaft in way of this gland is 11in., the fins stand out $\frac{1}{4}$in., so that the mean diameter is $11\frac{1}{4}$in. The radial clearance is $\frac{10}{1000}$in., giving an area for flow of 0·00245 sq. ft. Initial pressure $p_1=20$lb. abs. and $p_o=28\frac{1}{2}$in. vacuum, giving $x=27{\cdot}2$, $\frac{1}{x^2}=0{\cdot}00135$, $\log_e x=3{\cdot}30$, $v_1=20{\cdot}06$ cubic ft. per lb. With a permissible leakage of 0·037lb. per second, the number of throttlings works out at 16·9, say 17. Hence fit 9 fins on rotor and 8 fins in casing on inner side of

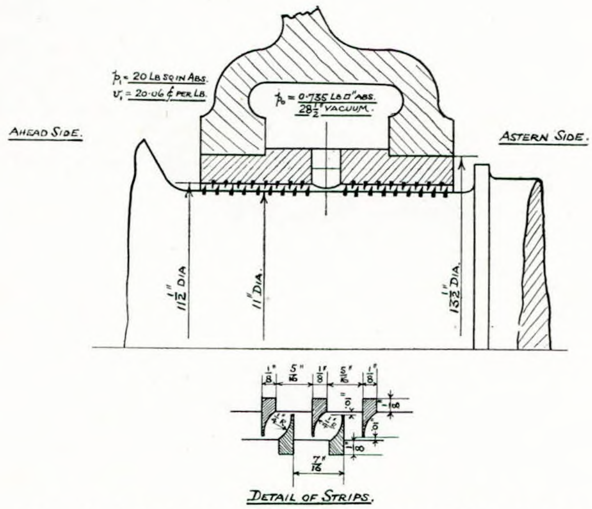


FIG. 8.—*I.P. inner ahead and astern turbine gland.*

gland before pocket connection and a similar number between pocket connection and outer end of gland (see Fig. 8).

(17) I.P. Turbine Outer Glands.

These are of single-pocket labyrinth type, the mean diameter being $9\frac{1}{4}$in., radial clearance $\frac{10}{1000}$in., area for flow $={\cdot}00202$ sq. ft., $p_1=16$lb. abs., $v_1=24{\cdot}71$ cubic ft./lb., $p_o=15$lb. abs., $x=1{\cdot}067$, $\frac{1}{x^2}=0{\cdot}876$, $\log_e x=0{\cdot}0649$. For the outer portion of the gland between pocket connection and vapour outlet to engine room, allowing a permissible leakage of 0·015lb./sec., the number of throttlings works out at 6·695. Fit 3 fins in rotor and 3 in casing; leakage becomes 0·0158lb./sec.

For the inner portion of the gland, $p_1=20$lb. abs., $p_o=16$lb. abs., $x=1{\cdot}25$, $\frac{1}{x^2}=0{\cdot}64$, $\log_e x=0{\cdot}2232$, $v_1=20{\cdot}06$ cubic ft./lb. Assuming a leakage of 0·025lb. per second, the number of throttlings works out at 10·627. Hence fit five fins in rotor and five in casing between turbine side of gland and pocket connection. Fig. 9 shows an illustration of the gland and Fig. 10 a sectional arrangement of the I.P. turbine.

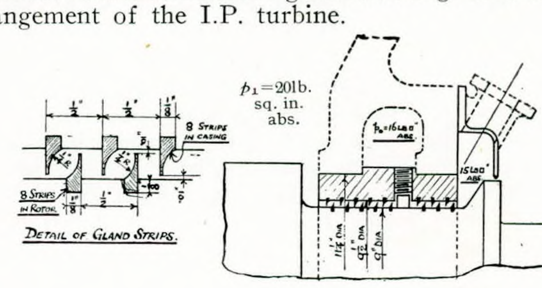


FIG. 9.—*I.P. turbine gland—outer.*

(18) Low Pressure Ahead Turbines.

The calculations for the blade heights and mean ring diameters for these turbines are not given in detail as

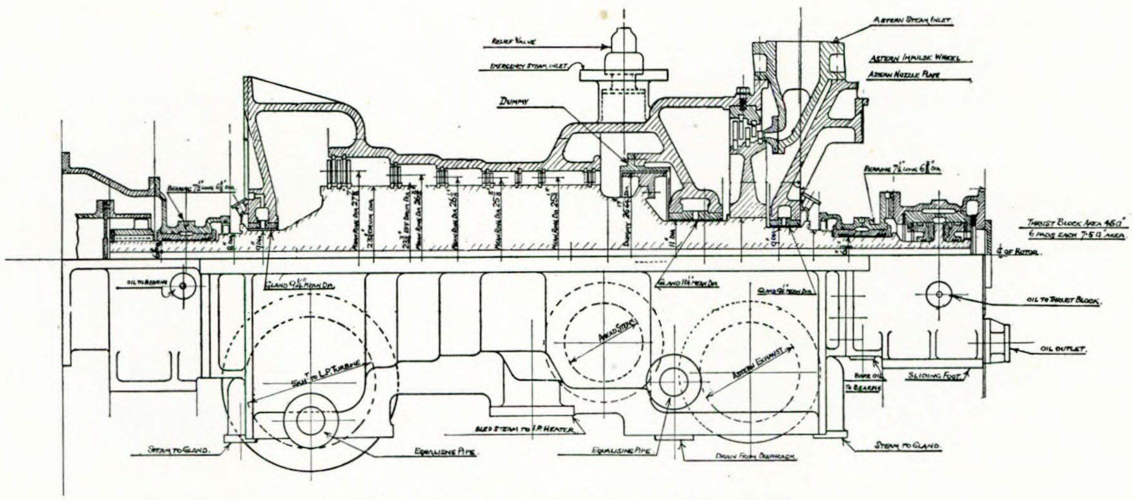


FIG. 10.—*Sectional arrangement of I.P. turbine.*

they follow the lines of those adopted in the case of the H.P. and I.P. turbines. The amount of steam passing through each L.P. turbine is estimated to be 61,643lb. per hour or 17·123lb./sec. and by adopting suitable values for ρ and z the blading schedule worked out as follows:—

Two L.P. Turbines. Single flow. Drum diameter 46in.

Expansion.	Effective blade height.	Mean ring diameter.	Number of rows.	Tip clearance.	Blade exit angle.
1	$2\frac{1}{4}$in.	$48\frac{1}{4}$in.	3	·045in.	20°
2	3in.	49in.	3	·045in.	20°
3	$3\frac{9}{16}$in.	$49\frac{9}{16}$in.	3	·050in.	20°
4	$4\frac{1}{4}$in.	$50\frac{1}{4}$in.	3	·055in.	20°
5	$5\frac{7}{8}$in.	$51\frac{7}{8}$in.	2	·060in.	20°
6	8in.	54in.	2	·070in.	20°
7	$9\frac{1}{4}$in.	$55\frac{1}{4}$in.	1	·080in.	20°
8	$10\frac{3}{4}$in.	$56\frac{3}{4}$in.	1	·085in.	20°
9	$10\frac{3}{4}$in.	$56\frac{3}{4}$in.	1	·085in.	25°
10	$10\frac{3}{4}$in.	$56\frac{3}{4}$in.	1	·085in.	30°
11	$10\frac{3}{4}$in.	$56\frac{3}{4}$in.	1	·085in.	35°

Blading sections are usually stipulated by number and letter. The writer has stated the exit angles of standard Parsons' blading as the former designation would not convey to the reader the exit angles used in the calculations. All the reaction blading in the H.P. turbine and the first four expansions of the I.P. turbine is of the same section, the last stage of the latter turbine having a different section with the same exit angle. In the L.P. turbine the first four expansions have the same section as the last stage of the I.P., the next four have a different section but the same exit angle, whilst the last three expansions are $\frac{1}{4}$ wing, semi-wing and $\frac{3}{4}$ wing blades.

(19) L.P. Turbine Dummy.

The dummies installed in L.P. turbines are of the radial type. The thrust on the rotor blading, calculated in a similar manner to the H.P. and I.P. turbines, works out at 7,566lb. Owing to the mean pressure on the dummy in L.P. turbines being small, the dummy diameter, to take any appreciable proportion of this thrust, would be inordinately large. It is therefore usual practice in L.P. turbines to make the dummy rotor the same diameter as the turbine rotor drum and to allow the steam thrust to be taken by the adjusting block. Accordingly, in the present design the dummy rotor was made 46 inches diameter and with the fins standing out $\frac{1}{4}$in. the mean diameter is $46\frac{1}{4}$in. The amount of thrust taken by the dummy is therefore $(20-\cdot735)(1680-1661\cdot9)=348\cdot7$lb., leaving 7,217·3lb. to be taken by the adjusting block. The latter was made 100 sq. inches in area on 6 pads, the load per square inch in the direction of steam thrust being 72·17lb.

The radial clearance of the dummy fins is $\frac{80}{1000}$in. giving an area for flow of 0·03027 sq. ft. The initial pressure $p_1=20$lb. absolute, $p_o=28\frac{1}{2}$in. vacuum (·735lb. abs.) and with a permissible leakage of 1,580lb. per hour (0·439lb./sec.) the number of throttlings works out at 18·82, say 20 throttlings, *i.e.* 10 fins in the dummy rotor and 10 fins in the casing. Fig. 11 shows the L.P. dummy.

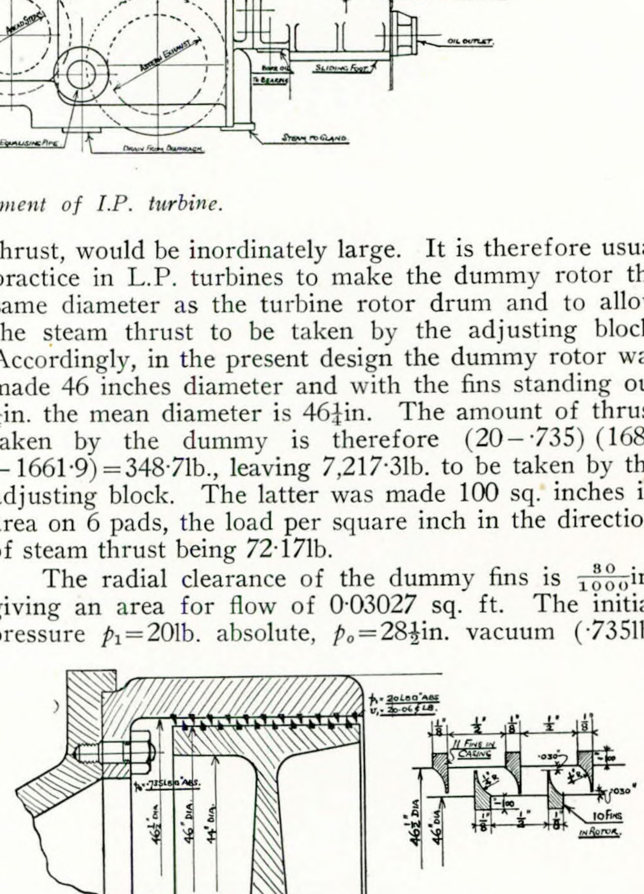


FIG. 11.—*L.P. turbine ahead dummy.*

(20) L.P. Labyrinth Gland Design.

These glands have a mean diameter of $11\frac{1}{4}$in., taking a pocket pressure of 16lb. abs., $v_1=24\cdot71$ cubic ft./lb., clearance $\frac{10}{1000}$in., area of flow$=\cdot00245$ sq. ft., $p_o=15$lb. abs., $x=1\cdot067$, $\frac{1}{x^2}=0\cdot876$, $\log_e x=0\cdot0649$. With a leakage to the atmosphere of 0·02lb. per second, the number of throttlings in the portion of the gland between the pocket connection and vapour outlet to the engine room works out to be 5·5, so that three fins are required in rotor and three in casing.

For the portion of the gland between pocket connection and the turbine, $p_1 = 16$lb. abs., $p_o = 28\frac{1}{2}$in. vac. (0·735lb. abs.), $v_1 = 24{\cdot}71$ cubic ft./lb., $x = 21{\cdot}75$, $\frac{1}{x^2} = {\cdot}0021$, $\log_e x = 3{\cdot}08$. Allowing a steam leakage into turbine of ·037lb. per second, the number of throttlings

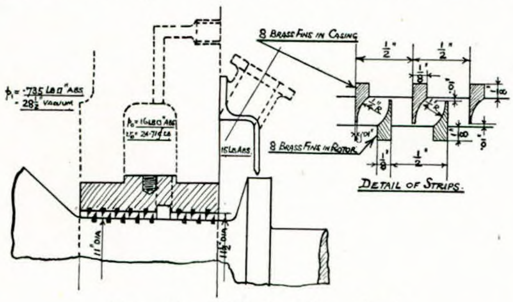


Fig. 12.—*L.P. turbine gland.*

works out at 10·11. Hence 5 fins are required in rotor and 5 in casing. See Fig. 12.

(21) Astern Turbines.

Astern turbines are fitted in the I.P. and L.P. turbine casings to give a total astern power of 75% of the maximum power of 19,500 s.h.p., *i.e.* 14,600 s.h.p. or 7,300 s.h.p. per screw.

Estimation of Steam Passing through Astern Turbines.

The adiabatic heat drop from 300lb. abs. 725° F. to $28\frac{1}{2}''$ vacuum is 456·5 B.T.U.'s per lb. of steam. From Table 2, $H_1 = 288{\cdot}6$ B.T.U.'s so that $\eta_1 = 0{\cdot}6325$. Allowing for dummy, gland, friction, radiation and gearing losses, the turbine efficiency ratio may be taken as 0·57. The theoretical steam consumption would be 5·575lb. per S.H.P. and the estimated actual consumption 9·77lb. per S.H.P. Hence the total steam to two H.P. astern turbines at 14,600 S.H.P. = 142,642lb. per hour or 71,321lb./hour to each = 19·81lb./second.

(22) H.P. Astern Turbine Design.

The adiabatic heat drop from 300lb. abs., 725° F., to 40lb. abs., is 204 B.T.U.s per lb., so that the theoretical steam speed $= 223{\cdot}7\sqrt{204} = 3{,}195$ ft./sec. For this velocity it will be necessary to fit a three-velocity stage wheel in order to keep the diameter and peripheral blade speed within reasonable limits.

From Table 2, $\eta_s = 0{\cdot}511$, hence ρ may be taken as 0·114, giving a blade speed of $\cdot 114 \times 3{,}195 = 364{\cdot}23$ ft./sec. The maximum astern revolutions may be estimated by assuming that the power varies as revolutions cubed, whence turbine revolutions at 14,600 s.h.p. = 2,134.

$$\therefore \text{ Mean wheel diameter} = \frac{720 \times 364{\cdot}23}{\pi \times 2{,}134} = 39{\cdot}1, \text{ say } 39\text{in.}$$

Making the mean diameter of the wheel 39in., actual blade speed = 363ft./sec. and $\rho = 0{\cdot}1136$.

With a nozzle angle of 19° and the following blade exit angles, 1st moving row 18°, 1st fixed row 18°, 2nd moving row 22°, 2nd fixed row 27° and 3rd moving row 35°, the velocity diagrams gave the following inlet angles :—

1st moving row 22°, 1st fixed row 22°, 2nd moving row 25°. 2nd fixed row 33°, and 3rd moving row 40°.

Using H. M. Martin's formula for the nozzle efficiency,

$$\eta_n = 102{\cdot}7 - 0{\cdot}06H_r$$
$$= 102{\cdot}7 - 0{\cdot}06 \times 204 = 90{\cdot}46\%$$

Steam velocity at exit from nozzles $= 223{\cdot}7\sqrt{\cdot 9046 \times 204}$
$= 3{,}030$ft./sec.

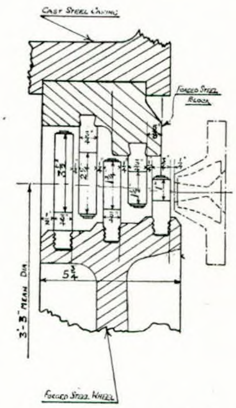


Fig. 13.—*H.P. astern impulse blades.*

The radial height of the nozzle l_n worked out at 0·66in., made 0·7in., and the blade heights as follows :—

1st moving row 1·003in. made 1in., 1st fixed row 1·392in. made $1\frac{1}{2}$in., 2nd moving row 1·681in. made $1\frac{3}{4}$in., 2nd fixed row 2·24in. made $2\frac{1}{4}$in., 3rd moving row 3·45in. made $3\frac{1}{2}$in.

Fig. 13 shows the astern turbine wheel.

(23) H.P. Astern Nozzle Plate Design.

If p_1 be the initial pressure in pounds absolute at a nozzle and p_o the exit pressure in pounds absolute, it may be proved that for maximum discharge $\frac{p_o}{p_1} = 0{\cdot}5457$. If the ratio $\frac{p_o}{p_1}$ is equal to or greater than 0·5457, the nozzle is convergent. If this ratio is less than 0·5457 the nozzle should be of convergent divergent form in order fully to expand the steam and produce orderly flow.

From Table 2, $p_1 = 300$lb. abs. and $p_o = 40$lb. abs., so that $\frac{p_o}{p_1} = 0{\cdot}1333$, hence the nozzles will be convergent divergent.

The weight of steam passing through the nozzles in terms of the throat area may be calculated from the following *equation :—

$$W = 0{\cdot}3155A\sqrt{\frac{p_1}{v_1}} \text{ lb. per second.}$$

Where, A = throat area in square inches; p_1 = initial pressure in lb. per sq. inch abs.

v_1 = volume per lb. of steam at pressure p_1 in cubic feet.

* "Steam Turbines", by Professor Goudie.

The estimated quantity of steam passing per second $=19{\cdot}81$lb., $p_1=300$lb. abs.; $v_1=2{\cdot}28$ cubic ft./lb.; $\therefore$ Throat area A$=5{\cdot}475$ sq. in.

The volume per lb. of steam at exit from the nozzle $=13{\cdot}89$ cubic feet, and steam velocity$=3{,}030$ ft./sec. $\therefore$ exit area $A_o=13{\cdot}10$ sq. in.

With vanes 0·064in. thick and exit angle 19°, the thickness factor $c=0{\cdot}8693$.

$$\therefore \text{ Gross arc } x=nP=\frac{A_o}{c.l_n \sin\alpha}=\frac{13{\cdot}10}{{\cdot}8693\times{\cdot}7\times{\cdot}3256}=66{\cdot}1\text{in.}$$

With a gross pitch of 1·5in., the number of nozzles$=44$. Width of nozzle at right angles to nozzle axis

$$={\cdot}8693\times1{\cdot}5\sin19^\circ=0{\cdot}4246\text{in.}$$

$$\text{Radial height at throat}=\frac{5{\cdot}475}{44\times{\cdot}4246}=0{\cdot}2930\text{in.}$$

If ϕ be the total angle subtended by the exit cone of the nozzle the length between throat and exit$=\dfrac{l_n-l}{2\tan\frac{\phi}{2}}$, where l_n=radial height of nozzle at exit and l=radial height of nozzle at the throat.

Assuming $\phi=15^\circ$ the length between throat and exit$=1{\cdot}543$in., say $1\frac{17}{32}$in.

(24) L.P. Astern Turbine Design.

Astern Impulse Turbine. Two row wheel.

From Table 2 above, $H_r=68{\cdot}2$, hence the theoretical steam velocity$=223{\cdot}7\sqrt{68{\cdot}2}=1{,}848$ft./sec. The efficiency from Table 2=67%, so that ρ may be taken as 0·21. $\therefore$ Blade speed$={\cdot}21\times1{,}848=388{\cdot}08$ft./sec.

The revolutions of the L.P. ahead turbines at 19,500 s.h.p. is 1,700, and from the cube law the revolutions astern at 14.600 s.h.p.$=1{,}544$.

$$\therefore \text{ Mean ring diameter}=\frac{720\times388{\cdot}08}{\pi\times1{,}544}=57{\cdot}6\text{in.}$$

Making the mean ring diameter 58in., the blade speed will be 391·3ft./sec. and ρ becomes 0·2118.

The velocity diagrams were drawn with a nozzle angle of 19° and the following blade exit angles: 1st moving row 22°, 1st fixed row 27° and 2nd moving row 35°, from which the inlet angles were 24°, 30° and 40° respectively. With a theoretical steam speed of 1,848ft. per sec., k_n may be taken as 0·943, giving an estimated steam speed at issue from the nozzles of 1742·7ft. per sec. The nozzle radial height worked out at 1·08in., made $1\frac{1}{16}$in., 1st moving row blades effective height 1·42in., made $1\frac{3}{8}$in., 1st fixed row 1·66in., made $1\frac{3}{4}$in. and 2nd moving row 2·24in., made $2\frac{1}{4}$in.

(25) L.P. Astern Nozzle Plate.

The pressure ratio $\dfrac{p_2}{p_1}=\dfrac{19{\cdot}5}{40}=0{\cdot}4875$, so that the nozzles will be convergent divergent. At a pressure of 40lb. abs., $v_1=13{\cdot}89$ cubic ft./lb. (Table 2) and since $W=19{\cdot}81$lb./sec., the throat area works out at 37·0 sq. in. At exit from the nozzles, the volume per lb. of steam $=25{\cdot}42$ cubic ft., giving an exit area $A_o=41{\cdot}61$ sq. in. Expansion ratio$\dfrac{A_o}{A}=1{\cdot}124$.

Taking a gross pitch of 2 inches, the nett pitch with vanes 0·064in. thick$=2-\dfrac{{\cdot}064}{\sin19^\circ}=1{\cdot}8035$in., so that the thickness coefficient$=0{\cdot}90175$.

$$\therefore \text{ Gross arc } x=nP=\frac{41{\cdot}61}{{\cdot}90175\times1{\cdot}0625\times{\cdot}3256}=133{\cdot}4\text{in.}$$

$$\text{Number of nozzles}=\frac{133{\cdot}4}{2}=66{\cdot}7\text{, say 67.}$$

$$\text{Width of nozzles}={\cdot}90175\times2\sin19^\circ=0{\cdot}5873\text{in.}$$

$$\text{Radial height at throat}=\frac{37}{67\times{\cdot}5873}=0{\cdot}9405\text{in.}$$

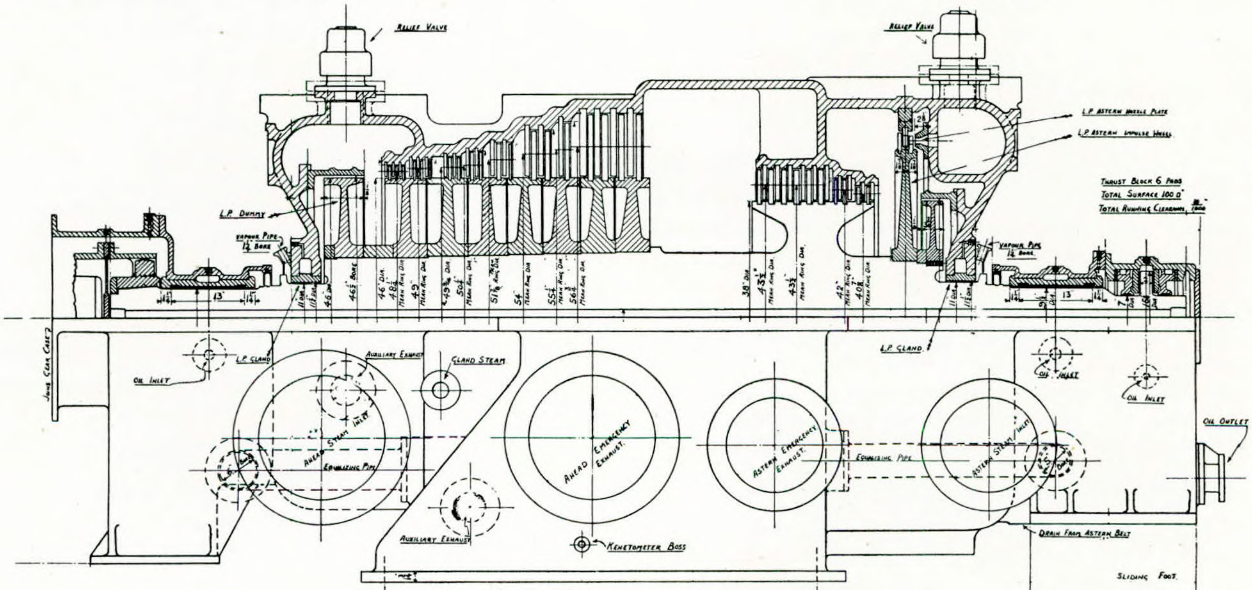


FIG. 14.—*Sectional arrangement of L.P. turbine.*

Taking ϕ, the total angle subtended by the exit cone of nozzle as 10°, the length between throat and exit $=0{\cdot}697$in., say $\frac{11}{16}$in.

(26) L.P. Astern Reaction Turbines.

Since the astern turbines are only used when manœuvring it is not necessary to design them for the highest efficiency. As will be seen from Table 2, the internal efficiency of the L.P. astern reaction turbine is 0·615 as compared with 0·80 for the ahead L.P. turbine. The values of ρ are therefore considerably lower than for the ahead turbine. The estimated turbine efficiency ratio assuming 6% losses for dummy, gland, radiation, friction and gearing would be 0·555, giving a K value of 44,000. The mean ring diameter is 42·65in. and the revolutions 1,544. The rows of moving blades work out at 10·11, made 10 rows, 2 per expansion.

The blading schedule for these turbines calculated from equation 3 by the adoption of suitable values of ρ and z, gave the following results. The complete schedule is given including the impulse wheel.

L.P. Astern Turbines.

Two turbines, impulse reaction.

Impulse nozzles, 67 in each turbine, 19° discharge angle, 2in. pitch, 0·064in. vane thickness, radial height $1\frac{1}{16}$in. at exit, 0·9405in. at throat.

Impulse wheel 58in. mean ring diameter.

Impulse Blading.

1st moving row	$1\frac{3}{8}$in. eff. height.	Inlet angle 24°,	exit 22°
1st fixed row	$1\frac{3}{4}$in. ,,	,, 30°	,, 27°
2nd moving row	$2\frac{1}{4}$in. ,,	,, 40°	,, 35°

Reaction Turbines.

Expansion... ...	1	2	3	4	5
Effective drum dia.	38in.	38in.	38in.	38in.	38in.
Eff. blade height...	$2\frac{7}{8}$in.	$3\frac{7}{8}$in.	$5\frac{1}{2}$in.	$5\frac{1}{2}$in.	$5\frac{1}{2}$in.
Mean ring diameter	$40\frac{7}{8}$in.	$41\frac{7}{8}$in.	$43\frac{1}{2}$in.	$43\frac{1}{2}$in.	$43\frac{1}{2}$in.
Number of rows ...	2	2	2	2	2
Blade exit angle, θ	20°	20°	20°	25°	35°

A sectional arrangement of the L.P. turbine is shown in Fig. 14.

REFERENCES.

"Design and Construction of Steam Turbines", by H. M. Martin. (Longmans, Green & Company).

"Steam Turbines", by W. Goudie. (Longmans, Green & Company).

"Steam Turbines", by E. F. Church. (McGraw Hill Book Company).

ADDITIONS TO THE LIBRARY.

Purchased.

Mercantile Navy List and Maritime Directory, 1940. H.M. Stationery Office, 25s. net.

Presented by the Publishers.

Transactions of The Institution of Engineers and Shipbuilders in Scotland, Vol. 83, containing the following papers:—

"Characteristics of Silent Propellers", by Davis.
"Humps and Hollows in Curves of Resistance", by Robb.
"Effect on the Propulsion of a Single-Screw Cargo Vessel of Changes in the Shape and Dimensions of the Propeller", by Kent and Cutland.
"Welding as a Substitute for Casting", by Dorey.
"A Modern Approach to Building Technique", by Miller.
"Modern Steam Propelling Units and their possibilities for Cargo Steamers", by Sneedon.
"Electrolytic Corrosion in Ship Structures", by Paterson.

Transactions of the Institution of Engineers-in-Charge, Vol. 43, containing the following papers:—

"The Modern Trend of Sanitary Engineering as Affecting Public Buildings", by Shenton.
"Poison Gas", by Whittle.
"Power Transmission by Belt, Rope, Chain and Gearing", by Parry.
"Cold Cathode Tube Developments", by Collins.
"A Study in Steel; and, Engine on the Shed", by Barrie.
"Research in Modern Industry", by Croft.
"The Chlorination of Water", by Burley.
"A Review of Recent Progress in Electric Lighting Practice", by Atkinson.
"Oil from Coal", by Underwood.

Proceedings of the Victorian Institute of Engineers, 1937-38, containing the following papers:—

"The Lubrication of Diesel Engines", by Hall.
"Hot Water Systems", by Borrie.
"Notes on the Efficiency of Water Heaters", by Gamble.
"The Essential Principles of the Lay-out of Industrial Buildings and Equipment", by Bennie.
"The Design of Jigs and Fixtures for Machine Tools", by Trewhella.
"Photogrammetry", by Pyke.
"The Engineer's Part in the Petroleum Industry", by Kneale.
"Purification and Sterilising Equipment for Modern Swimming Pools", by Gamble.

The following British Standard Specifications:—

No. 916-1940. Black Bolts and Nuts (Small Hexagon and Square). (War Emergency Standard).
No. 908-1940. Sisal Rope for General Purposes.
Nos. 382-383-1940. Bronze (Gun Metal) Ingots and Castings for General Engineering Purposes.
No. 431-1940. Manila Ropes for General Purposes.
No. 592-1940. Carbon Steel Castings for Ships and for Marine Engine and General Engineering Purposes.
No. 906-1940. Engineers' Parallels (Steel).
No. 907-1940. Dial Gauges for Linear Measurement (excluding back plunger type).

The specification (906) for Engineers' Parallels deals with the general dimensions and permissible errors in parallel blocks of sizes $\frac{1}{4}$in.$\times\frac{1}{2}$in.$\times$4in. to 2in.$\times$4in.$\times$16in. Provision is made for two grades of accuracy, one for high precision work and the other for general tool room use.

The specification (907) for Dial Gauges applies to the most commonly used type of gauges in which the plunger movement is parallel to the dial. It does not cover gauges with a back plunger movement. The main purpose of the specification for dial gauges is to establish standards of accuracy. Considerable latitude is allowed in the general design of the gauges, although certain requirements are included in respect of matters of detail where these have a direct bearing on the initial accuracy or the maintenance of the accuracy or on the ease with which the dial gauge may be read. The table of permissible errors covers dial gauges from $1\frac{3}{4}$in. to 2in. diameter measuring by steps of 0·001in. 0·0005in., 0·0001in. and 0·01 mm. The appendix to the specification gives notes on various methods of testing dial gauges, with particular reference to the accuracy of calibration.

The Mechanical Testing of Metals and Alloys. By P. Field Foster, M.Sc. Sir Isaac Pitman & Sons, Ltd., 2nd edn., 285 pp., 215 illus., 18s. net.

This book deals in a concise manner with the testing of metals and alloys for engineering purposes.

Commencing with a brief summary of the elementary theory of elasticity, the book includes chapters on the structure of metals, universal testing machines and methods of calibration, tension and bend tests, testing machine accessories, extensometers and recorders, torsion testing, hardness and notched bar impact testing, fatigue and repeated stresses, the testing of wire and sheet metal, and concludes with a discussion of typical test results and phenomena, including strain-hardening effects, hysteresis, compression failures in ductile and brittle materials, temperature effects and creep tests. Some useful tables of the mechanical properties of various metals and alloys are included at the end of the book, whose value is enhanced by a key list giving their general characteristics and engineering applications.

It is thought that the section dealing with universal testing machines might usefully have been enlarged, in view of the importance of this type in commercial testing, whilst the author's reference to bend tests is thought to be unduly brief. Considerable attention is given to the subject of strain-measuring instruments and recorders, and very full descriptions are given of all the best known types. The chapters devoted to hardness and notched bar impact testing will be found of particular value and, together with the following chapter on repeated stresses, deal in a comprehensive manner with the principles and practice of these methods of testing.

The testing of wire and sheet metal, a subject not usually familiar to the average engineer, is treated at some length, and includes an account of various methods of determining the elastic constants of metals by the use of wires and strips. Various commercial tests are described, including the recently introduced Jovignot test for the ductility of sheet metals. In the final chapter, a number of useful diagrams are given, indicating the behaviour of different metals under various types of testing.

The work is generally well arranged and up-to-date, but future editions could be improved by the elimination of a considerable number of errors and misprints, the majority of which were found in the first chapter, but also in chapters X and XI and elsewhere in the book. The value of the book might also be enhanced by the provision of more line diagrams of the various machines and apparatus, and greater clarity in the descriptive text. The published price of 18s. is also thought to be somewhat high.

The book will be found of considerable service to those engaged in mechanical testing and inspection, and to students of engineering wishing to acquire a practical knowledge of testing methods.

Engineering for Nautical Students. By W. A. Fisher, A.M.I.Mech.E., A.R.T.C. Brown, Son & Ferguson, 139 pp., 74 illus., 5s. 6d. net.

This book is written to cover the syllabus in the second year in the scheme for the Instruction of Apprentices. The subject matter in this syllabus is steering gears, pumps and shafting, burning of coal and oil, and the elements of refrigeration, steam turbines and Diesel engines. Electrical units, electric magnets, motors, heating apparatus and wireless telegraphy are also included.

The engineering and electrical knowledge required by this type of student must obviously be of a very elementary character, consisting mainly of the names of the several parts and, as the syllabus states, "a very simple idea of how the machine works".

With this object in view the author has succeeded in keeping down to the standard required and has illustrated his statements by simple sketches. An apprentice who reads the book intelligently should have no difficulty in answering the questions set in the examination at the end of the course.

The Alloys of Iron and Chromium. Vol. II—High Chromium Alloys. By A. B. Kinzel and R. Franks. McGraw Hill Publishing Co., Ltd., 559pp., 133 illus., 40s. net.

The authors in the preface state that this volume, the eleventh in order of publication and the tenth of the regular Monograph series, is the second and final part of a review and summary of important published information and available unpublished data on the alloys of iron and chromium. It deals with alloys containing more than 10 per cent. chromium and includes data on those materials commonly known as "stainless" and "heat resisting" steels.

The authors have drawn well on the world's technical literature and it is unquestionable that this book will have a wide appeal. It gives everyone engaged or interested, such as a practical metallurgist, steelworker, and engineer a detailed and comprehensive survey of iron chromium alloys, that they may be able to make the best of the expert advice on these materials. The book forms perhaps the most complete treatise available on alloys of iron and chromium.

Some idea of its scope will be gathered from the chapter headings :—The Phase Relations—Melting High-Chromium Steels —Fabrication of 12 to 30 per cent. Chromium Steels—Low-Carbon Steels containing 10 to 18 per cent. Chromium—Medium and High-Carbon Steels with the same Chromium Content as above—Plain and Modified Chromium Steels containing 20 to 35 per cent. Chromium—Chromium Cast Iron with 12 to 35 per cent. Chromium; Constitution of Complex Iron-Chromium Alloys—Manufacture and Fabrication of Austenitic Chromium-Nickel Steels—Properties of Austenitic Chromium-Nickel Steels and Resistance to Oxidation and Corrosion—Modified Austenitic Chromium-Nickel Steels—Chromium-Manganese Steels, and Iron-Chromium-Aluminium Alloys.

The remainder of the book, some 70 pages, is given over to three things. First, a most comprehensive bibliography of books and reviews, libraries, engineering societies and the technical press in nine countries. This is followed by a name index and finally the subject index.

Tribute must be paid to the authors for the excellent summary they have made at the end of each chapter, and in having crammed so much useful information within the 500 odd pages comprising the book. The reader's task is lightened by the precise, pleasant style of the authors. Not to be forgotten are the publishers who have provided a worthy setting for another of the Monograph series.

JUNIOR SECTION.

Naval Architecture and Ship Construction (Chapter II).

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

Structural Details.

A ship may be regarded as a vehicle capable of conveying goods and passengers from one point to another across the surface of the seas. From this definition it will be understood that the distribution of materials in a ship will differ considerably from that in a structure of a stationary character, although it is customary to make use of plates, angle bars and various other rolled sections in each case.

Rolled Sections.

The following rolled sections are commonly employed in shipbuilding (Fig. 14).

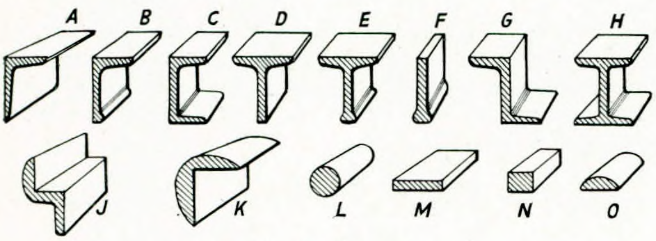


FIG. 14.—*Rolled sections.*

- *A. The angle-bar*—used for general purposes.
- *B. The angle-bulb bar*—used for beams, frames, and stiffeners in medium sized ships.
- *C. The channel-bar*—used for beams, frames and bulkhead stiffeners in large ships. The channel bar is not employed for framing at the ends of the ship, since such frames need to be bevelled, and the process of bevelling a channel section is extremely difficult.
- *D. The "tee" bar* } used for the beams of
- *E. The "tee" bulb bar* } decks which are planked but not plated. It will be noted that in such a case the planks have to be fastened at the beams, and a double flanged beam is more suitable for this purpose.
- *F. The bulb-plate*—used for the construction of "built-up" beams, and for bilge keels.
- *G. The "zed" bar*—used for framing and bulkhead stiffeners in warships.
- *H. The "aitch" bar*—used for strong beams.
- *J. The Tysack* hatch moulding—used as stiffening around the top-edge of the hatch coaming.
- *K. The Tysack* bulwark moulding.
- *L. Round iron*
- *M. Flat iron*
- *N. Square iron*
- *O. Segmental iron*

} used for general purposes such as small shafting, keys, stiffening, etc.

In addition to the foregoing it will be understood that the plates used for constructing the shell, bulkheads and decks are also rolled out from the steel ingots.

Framing.

The frame is frequently defined as a transverse rib, but in modern practice it would be better to regard it as a vertical stiffener to the side plating. The frames are spaced from twenty-one to thirty-six inches apart, with some decrease towards the ends, and forward of the collision bulkhead this spacing is not to exceed twenty-four inches.

Fig. 15 illustrates the various arrangements employed for framing ships. It is not necessary here to say much about the history of structural developments. It will suffice to point out that there have always been

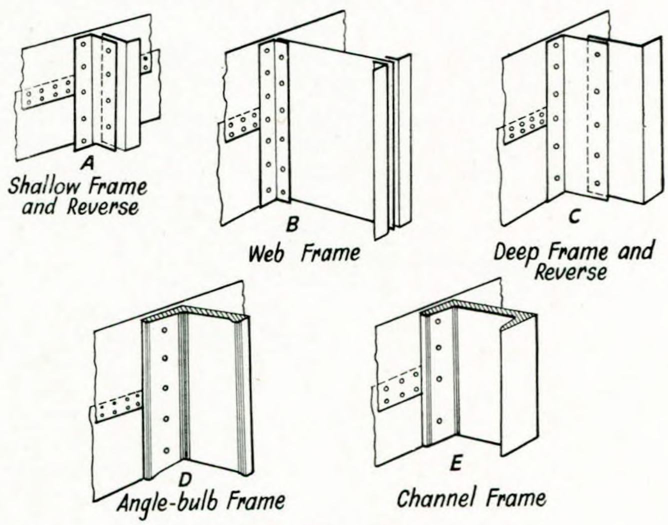


FIG. 15.—*Types of frames.*

two main tendencies, firstly to reduce weight, and secondly to eliminate broken stowage. Thus, heavy web frames in hold spaces have disappeared, and in lieu an arrangement of relatively closed spaced "deep" frames introduced. The term "deep frame" is not entirely a misnomer, since the old-fashioned web frames which were fitted at every fifth or sixth normal frame space, in conjunction with very shallow "frames and reverses", give place to deep "frames and reverses" of a depth varying from about seven inches up to as much as twelve inches. The earlier shallow frames and reverses rarely exceeded four and a half inches in depth. The frame is connected to the plating by a single line of rivets spaced seven diameters apart, and the connection between the frame and reverse bar is also formed by a single line of rivets of the same spacing. The construction of the frame and reverse is comparatively costly, and there is the further disadvantage of rust forming between the flanges. These factors have led to the adoption of the single unit frame, such as the angle bulb or the channel bar.

As pointed out in the previous chapter the attachment between the beam and the frame is of considerable importance. It may be made by forming a crooked end to the beam, known as a beam knee, and riveting the frame to the knee, or a separate bracket may be used. Fig. 16 illustrates a method of forming a beam knee. The beam is first cut along the dotted line in sketch "A"; it is then heated and bent to the form shown in sketch "B"; a piece of plate is next welded in, the surplus material removed, and the finished appearance and dimensions are as shown in sketch "C".

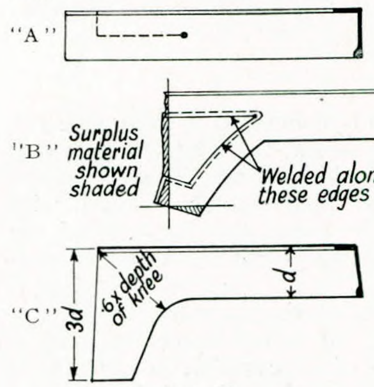


Fig. 16.

A detail of a beam bracket arrangement is shown in Fig. 17, where it will be noticed that the beam does not extend to the shipside, but is cut short where it meets the frame. The number and size of rivets will vary with the size of the ship. However, in a medium-sized cargo vessel about six three-quarter inch rivets are used in each edge of the bracket.

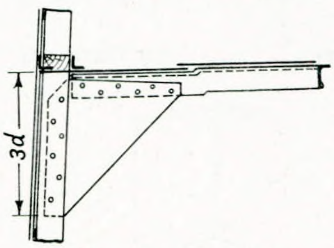


Fig. 17.—*Beam bracket attachment.*

The *beam* is defined as that portion of the transverse member which unites the tops of the frames and supports the deck. Weather deck beams are cambered in order to prevent the deck from falling hollow, and thus lodging small quantities of water. It is doubtful whether camber adds very much to the strength of the transverse member, since the comparative flexibility of the frames would seem to nullify any such effect. The buttresses which take the ends of a cambered bridge girder are quite rigid, a factor of paramount importance.

Camber is usually defined as the "round of beam" and the standard amount equals one-fiftieth of the breadth of the ship. Beams are fitted at every frame position, except for portions of the length under superstructures, where they may be fitted at alternate frames only.

The angle bulb bar or channel bar is the common type of section employed for beams where the spacing is regular. In the machinery spaces, where for obvious reasons a regular spacing of beams is impossible, heavy web or built up beams are employed and attached to web frames. Fig. 18 illustrates the attachment of a "strong" beam and web frame as fitted in a machinery space. Anticipating that at some time parts of the machinery may have to be removed, it is not unusual to make these strong beams in three parts, the middle part being fastened to the outer portions by double butt straps, thus facilitating the removal of the beam when required.

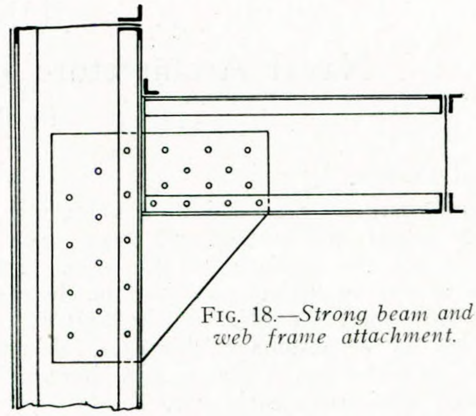


Fig. 18.—*Strong beam and web frame attachment.*

The web frame is attached to the side of the double bottom by lapping the web plate to the bilge bracket and treble riveting the joint. The details will vary in different ships, but one suitable arrangement is indicated in Fig. 19.

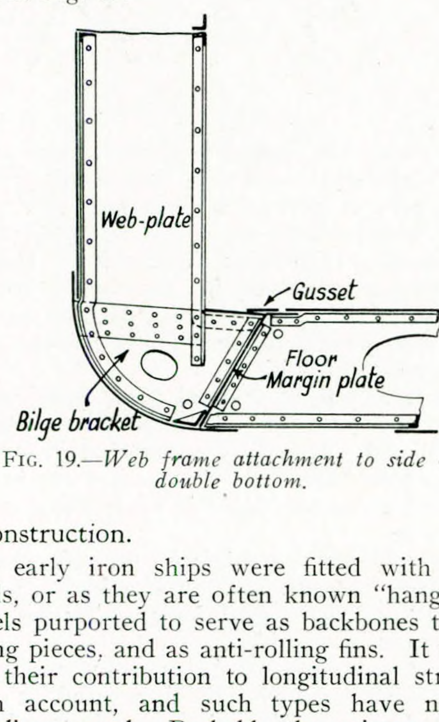


Fig. 19.—*Web frame attachment to side of double bottom.*

Keel Construction.

The early iron ships were fitted with projecting type keels, or as they are often known "hanging" keels. Such keels purported to serve as backbones to the ship, as rubbing pieces, and as anti-rolling fins. It is doubtful whether their contribution to longitudinal strength was of much account, and such types have now almost entirely disappeared. Probably the primary reason for their abolition was the fact that the projecting portion of the keel reduced the effective draught of the ship at a time when depth of water in docks and over sills was none too great, and when every inch of draught meant extra profit to the shipowner. For its historical value only, a sketch of a typical bar keel is included, and at the same time the centre line keelson with which this was associated is also shown (Fig. 20).

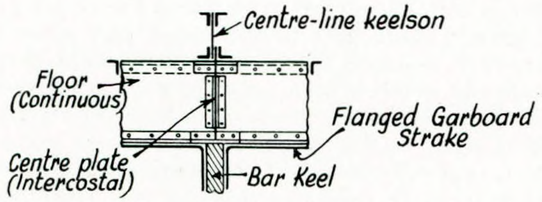


FIG. 20.—*Watertight floor forming boundary to double bottom tank.*

The *flat plate keel* is in reality the middle line strake of outer bottom plating, and when considered in conjunction with the vertical keel and middle line strake of inner bottom plating, it forms a valuable longitudinal stiffener or backbone to the ship. A sketch of this arrangement was given in the previous chapter.

Double Bottoms.

As is well understood, for a vessel to be made seaworthy when sailing in the light condition, it becomes necessary to carry some form of ballast. For many years it was the practice to load pig-iron or rubble for this purpose. This method involved the waste of much time, both for loading and discharging, the operations being often as lengthy as those connected with the handling of cargo. In cases where the sea voyage was of short duration, and where there was no lack of freights, the number of journeys lost per year and consequent loss of profits becomes obvious. The urge therefore for some method of ballasting which would eliminate this loss of time is readily understood.

Instinctively, water suggests itself as offering the best solution to the problem, since the handling thereof could be carried on more or less concurrently with the loading and discharging of cargo. Furthermore the carriage of water in suitably arranged tanks offers great flexibility of control over draft, trim and stability. Finally it may be pointed out that the cost of handling water is relatively negligible.

On the debit side there were one or two constructional difficulties in the early iron ships. It was not proposed originally to fit a continuous sequence of tanks from peak to peak, but rather to leave the existing structure as it was, and to fit tanks between hold bulkheads only. To provide the requisite capacity this demanded that they should be of considerable depth, and thus, the already somewhat limited hold space was still further denuded. One such arrangement was known as the M'Intyre system, in which girders were run longitudinally along the tops of the floor plates, and the tank top plating was secured to the tops of the girders. The whole of the space between tank top and outer bottom was available for water. The tanks were bounded at the sides by means of margin plates, which slotted back over the existing frames. Great difficulty was experienced in making the sides watertight, and in due course it was deemed expedient to sever the frames and connect the margin plate to the shell by a continuous angle on the outside, both edges of which were caulked. As the type is now obsolete it has not been considered desirable to include sketches; mention has been made solely because the modern *cellular double bottom* follows logically from this system.

To the advantages already enumerated for the water ballast system, the cellular double bottom adds yet another. In this arrangement, which is now universal in its application, there is a continuous sequence of tanks stretching from peak to peak. This makes it possible to provide tanks of lesser depth, and to secure a continuous watertight inner skin which it is hoped will remain intact and offer some measure of protection in the event of damage to the outer bottom.

Considerations which fix Ballast Capacity.

It is only natural that the shipowner should not wish his ship to carry around more water than is absolutely necessary, but at the same time it should be borne in mind that the economic performance of the ship is intimately bound up with its good behaviour. Good behaviour requires that the draft should be adequate to prevent the vessel making excessive leeway, the propellers should be well buried, although the trim by the stern must not be exaggerated, and the stability should be ample. To satisfy these requirements would demand a ballast capacity of some twenty-five to thirty per cent. of the deadweight, or what amounts to much the same thing, the ballast draft should be not less than one-half the load draft. In modern ships burning oil fuel, not all the double bottom tanks will be available for ballast, and therefore, use must be made of the peak tanks, and if necessary deep tanks should be fitted. If one deep tank only is required a suitable position is just abaft the engine-room bulkhead. It can extend to the freeboard deck, and if fitted with a large watertight hatch, can be used for cargo when not required for ballast.

Structural Arrangements of Double Bottoms.

As already indicated the cellular double bottom system consists of a series of watertight compartments forming a continuous line of tanks extending from fore peak to after peak. The inner skin is continuous and watertight. Bulkheads fit round the double bottom, they do not pierce the tanks, although in general there will

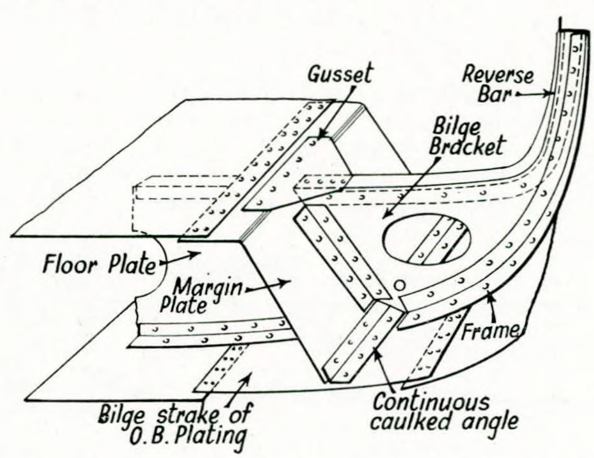


FIG. 21.—*Attachment of frame and reverse to margin plate.*

be a watertight floor immediately under a bulkhead. The side boundaries may be formed by margin plates flanged on the upper edge, and lap jointed to the inner bottom plating, the lower edge being secured to the ship side by a continuous angle bar on the outside of the tank. This is the arrangement when the vessel has side bilges, but where it is decided to dispense with these the tank top extends out to the ship side. In each of these cases the main frame is broken at the tank side, and transverse continuity of strength is preserved by bracketing. The arrangements are illustrated in Figs. 21 and 22.

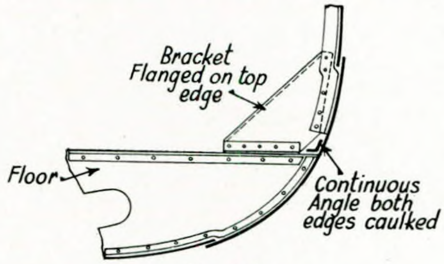
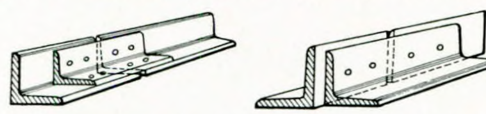


FIG. 22.

The internal structure consists of transverse *floors* fitted at every frame position under engines and boilers and for a portion of the length forward, but at alternate frames elsewhere, and a *continuous longitudinal centre girder,* with one or possibly two *intercostal side girders.* In some of the tanks the centre girder will be made watertight, thus providing independent compartments on either side of the centre line. The primary object of this watertight centre line subdivision is to minimize the risk of instability when running up tanks in a *tender* ship. The general idea would seem to be that divided tanks are provided to facilitate the correction of list, but here a word of warning is necessary. If the list is due to lack of symmetry in the weight distribution, filling a tank on the high side is the lazy man's method; it would be better to redistribute the weight. If the list be due to instability, filling on the high side will make things worse, not better. There is a well-known principle which is inculcated into the minds of all young naval officers and is worth quoting here: "Do not voluntarily introduce water into your ship, leave that to the enemy".

Floors.

A *floor plate* is a transverse vertical plate running across the bottom of the ship and extending from the centre line to the bilge. It has the full depth of the double bottom. *Watertight floors* are employed to divide the double-bottom space into suitable tanks. Such a floor is shown in Fig. 23. It is connected to the centre girder, to the margin plate and to the inner and outer skins by a continuous close fitting angle bar about three to four inches in depth. Both edges of this bar have to be caulked, and to enable this to be done effectively the spacing of rivets in the flanges is four to four and a half diameters, centre to centre. If the tank space under consideration is to be used for fuel oil the rivet spacing would be reduced to three to three and a half diameters centre to centre. It is not possible to employ one length of bar to fit all the way round the edge of the floor, and so shoe pieces are fitted at the centre girder and at the tank margin plate. The butts have to be carefully fitted and caulked. The ends of the bars should be cut in the band-saw, and then filed up. If this be done it will not be necessary to fit *bosom pieces,* as is often the practice. A bosom bar is a short piece of angle-bar fitting into the "bosom" of the parts to be connected and serving as a butt strap (see Fig. 24). Another method



Bosom bar. FIG. 24. *Back bar.*

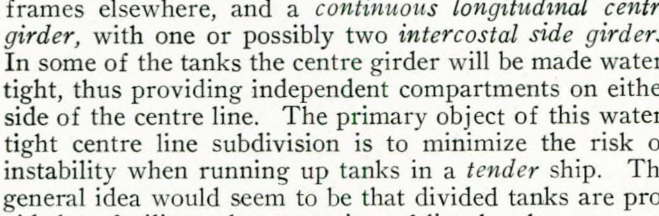


FIG. 23.—*Watertight floor forming boundary to a double bottom tank.*

of connecting two bars together is by means of a back bar, which however is not as efficient and is manifestly unsuitable in this case.

As previously noted, *solid floors* are fitted at every frame position under engines and boilers, and for part of the length forward. They are termed *solid,* even though large holes are cut in them to provide access and to reduce weight. These holes would measure about 18 inches by 21 inches, and since much of the material thus removed is redundant the strength of the plate is not seriously impaired. Nevertheless, immediately under engine seatings, where strength and rigidity are of great importance, no holes will be cut in the floors other than the small ones at the top to release air when running up a tank, and at the bottom to facilitate drainage.

Floor plates and intercostal side girders have their thicknesses slightly increased (about $\frac{1}{10}$ of an inch) under engines due to excessive weight and vibration, and under boilers partly due to excessive weight and partly as a provision against corrosion. It should be noted that the warm moist atmosphere under boilers is highly conducive to corrosion. A typical solid floor with lightening holes cut is shown in Fig. 25.

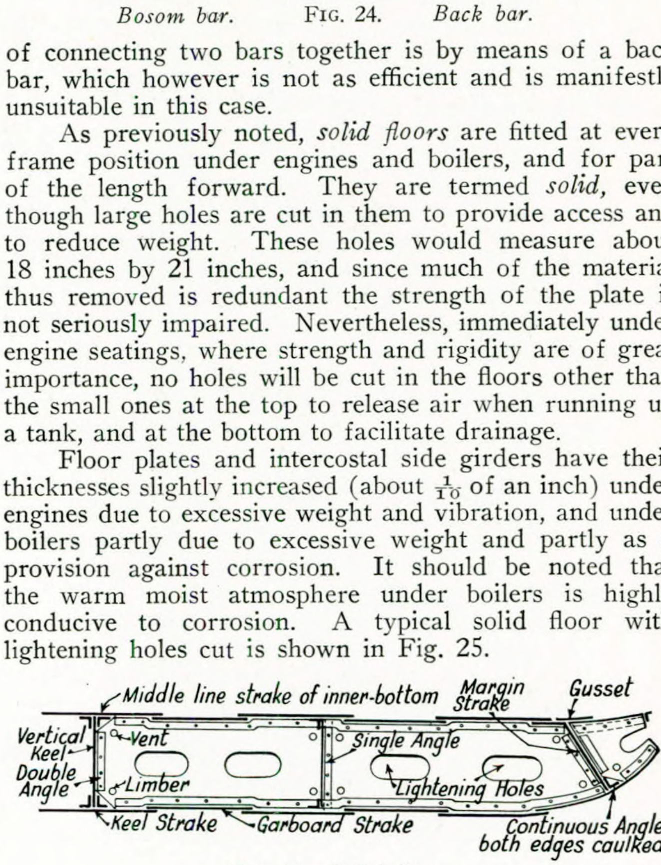


FIG. 25.—*Solid floor.*

Under hold spaces solid floors are fitted alternately, with an arrangement of brackets at the intermediate positions. In some modern ships there may be two ***bracket floor*** positions between the solid floors. At the bracket position angle bars are fitted across the under side of the tank top, and along the inside of the outer bottom. At the centre line and at the margin plate brackets are fitted, the top edges of which have to be long enough to

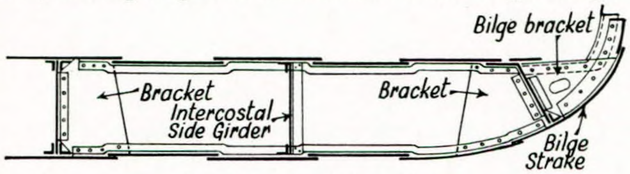


FIG. 26.—*Bracket position.*

take at least three rivets. The arrangement is illustrated in Fig. 26.

Longitudinal Girders in Double Bottoms.

The *centre girder* is an arrangement of relatively heavy plates connected together by double butt straps, united to flat keel by double angle bars, and similarly connected to the inner bottom. In this way a valuable backbone is formed. The importance attached to this part of the structure by Lloyd's is indicated by the fact that it is insisted that it should be continuous from stem to stern, and that no holes shall be cut in the plate other than those required for drainage and for releasing air when filling, for three-quarters of the length amidships. Fig. 27, from which the inner bottom plating has been

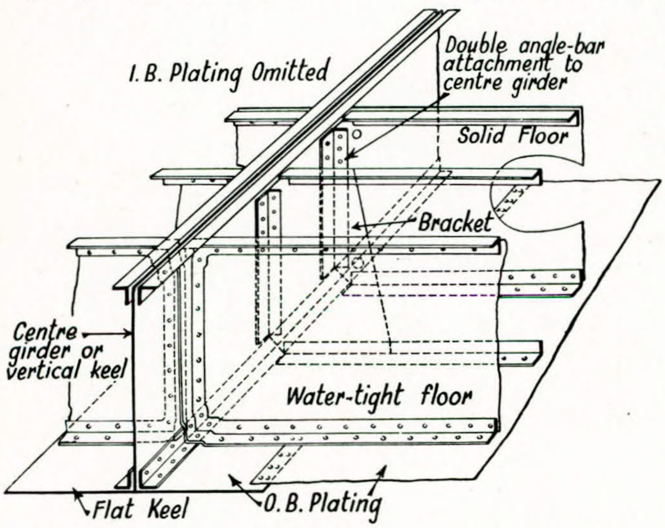


FIG. 27.—*Attachment of floors to continuous centre girder.*

omitted, gives a pictorial idea of the double-bottom structure in way of the centre line. It shows the different types of floors, in their proper sequence, and their attachment to the centre girder.

An *intercostal girder* is one fitted in short lengths between the floor plates. At least one such girder will be fitted on either side of the centre line. An attempt to portray this idea is made in Fig. 28. Reference to the detail of the intercostal plate shows that it is two frame spaces in length, it is slotted in way of the transverse angle bars at the intermediate bracket position, and is cut away where it is likely to foul the angle bars on the solid floor. Single angle bars with rivets spaced

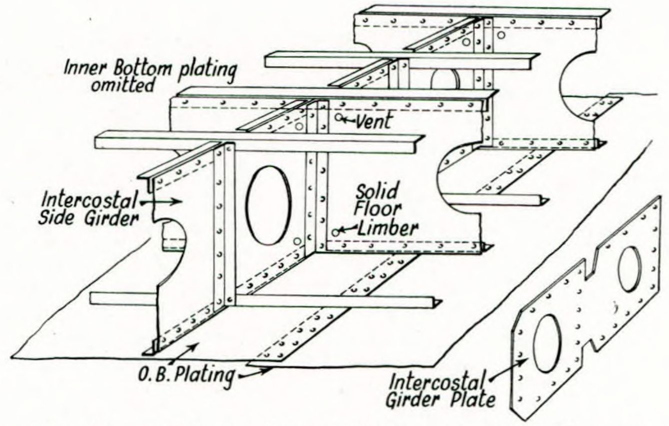


FIG. 28.—*Attachment between intercostal side girder and floor plates.*

seven diameters apart connect the side girder plate to the floors and to the inner and outer bottom.

Inner Bottom Plating.

The inner bottom plating is arranged in longitudinal strakes, having single riveted edges and double riveted butts. The ends of the plates are lapped, not strapped. The centre line strake is somewhat heavier than the remainder, its edges are double riveted and the butts may be treble or even quadruple riveted, according to the size of the ship. The margin strake in vessels with side bilges is flanged on the upper edge and joined to the contiguous plating by a double riveted lap joint. The connection to the shell is through the medium of a continuous angle bar fitted on the outside of the tank with riveting sufficiently close to enable both edges of the bar to be caulked and made watertight. This arrangement has already been illustrated in Fig. 21. Occasionally a transverse strip of plating is worked under a bulkhead, as the task of connecting the latter to the tank and making watertight is much simplified thereby.

As stated in the previous chapter, box girders in engine rooms have not proved very satisfactory. A much better scheme is to rivet the engine bedplate direct to the tank top. When the latter method is adopted additional side girders will be fitted, all angle bar attachments to the inner bottom will be double, and the strakes of plating immediately under the bedplate will be exceptionally thick, reaching an inch in large vessels.

Not much attempt has been made in the foregoing work to introduce precise scantlings, but as an approximate guide to the uninitiated, the following rough values will give some indication of what to expect in a medium-sized tramp steamer :—

Flat keel	about 50in. wide, $\frac{7}{8}$in. thick
Remainder of bottom plating	$\frac{5}{8}$in. ,,
Centre girder	$\frac{5}{8}$in. ,,
Side girders and floor	$\frac{3}{8}$in. ,,
,, ,, ,, (under engines & boilers)	$\frac{1}{2}$in. ,,
Middle line strake of inner bottom	$\frac{1}{2}$in. ,,
Remainder of inner bottom	$\frac{3}{8}$in. ,,
Strakes immediately under engines ...	$\frac{3}{4}$in. to 1in. ,,
Margin strake	$\frac{1}{2}$in. ,,

Preservation of Surfaces inside Tanks.

In older ships a layer of five or six inches of cement was spread over the bottoms of the tanks, the limber holes being arranged so that their lower edges were flush with the top of the cement. The weight involved was considerable, and in due course the classification societies were persuaded to agree to a reduction in this thickness, so that it became common practice to employ a layer of Stockholm tar and cement about one inch thick. Mounds or fillets of cement were built up over rivet heads and plate edges, and to promote drainage slots were cut in the lower angle irons just above the roots. In some quite recent designs cement has been dispensed with entirely.

The remainder of the insides of tanks may be cement washed or iron oxide paint may be employed. In feed tanks and fresh water tanks bitumastic coatings are largely used.

Testing.

All double-bottom tanks both prior to launching and at subsequent special surveys have to undergo pressure tests. In the case of ordinary ballast tanks, the compartments are filled with water and then pressed up to the level of the *light* water line, or to the greatest head likely to be experienced in practice (which is the upper deck when no overflows are fitted). Fuel tanks and fresh-water tanks are tested to the *load* water line or the greatest head likely to be experienced in practice, whichever is the greater.

Access.

Access to double bottom tanks is provided by means of manholes cut in the inner bottom. These are closed by elliptical plates bedded down over studs, or the plates are sometimes dished and secured by strong backs.

Abstracts of the Technical Press

Gas-engined Tug for Siberian River Service.

A twin-screw gas-engined tug for service on the lower reaches of the Yenissei is to be built to the designs of a group of Soviet naval architects and marine engineers. The vessel is to have an overall length of 155ft., a beam of 25½ft., and a moulded depth of 10½ft., with a displacement of 410 metric tons on a maximum draught of 6½ft. The steel hull is to be sub-divided into seven watertight compartments, the bow being saucer-shaped for navigation in ice, the bottom flat and the stern provided with tunnels for the installation of Kort nozzles around the twin propellers. There will be twin balanced rudders. The machinery installation is to consist of two sets of oil engines of Soviet manufacture, adapted for operation on producer gas generated from wood fuel. The designed output of the engines is 680 h.p., corresponding to a speed of 11¾ knots when running free and about 5 knots when exerting a useful towing pull of 20,000lb. The producer-gas equipment will comprise a producer, two scrubbers, a cleaner and a blower, the latter being fitted to facilitate lighting up and the burning of logs up to a metre in length. The total weight of this installation will be about 10¼ tons and the capacity of the vessel's bunkers just over 5,000 cu. ft. It is stated that a gas-engined tug of this type is equal to a steam-driven tug in efficiency, but that it is inferior to one with Diesel engines. Nevertheless, the special conditions prevailing on the great Siberian rivers make the use of wood-burning gas-engined tugs worth while, since they can carry sufficient wood fuel for 24 hours' running and replenish their bunkers at any of their stopping places.—*A. I. Pavlov, "Soudostroienie", Vol. 10, No. 1, January, 1940, pp. 10-11.*

The Sormovo Works at Gorky.

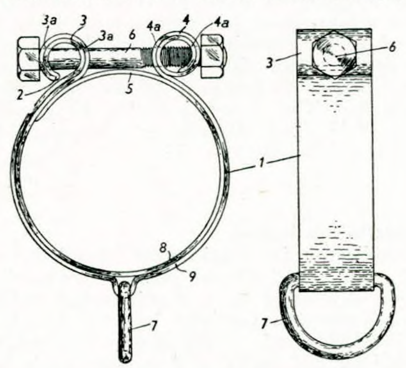
The above establishment recently celebrated its ninetieth jubilee, having been founded at Gorky (formerly Nijny Novgorod) in 1849, since which time its locomotive-building shops, iron and steel foundries and marine engine-building works have attained a very substantial output. Amongst the products of the Sormovo Works in pre-revolutionary days were a number of motor vessels for service on the Volga and in the Caspian. The Soviet Government have, within the past 20 years, devoted special attention to the development of the shipbuilding activities of the works, which include the construction of a large number of barges, lighters and fuel oil barges for river service, put in hand in the year 1923-24. Within the last four years the Sormovo Works have also built some very large fuel-oil barges with a capacity of 12,000 tons, numerous paddle and screw tugs with engines of 150 to 1,200 h.p. and several dredgers and oil-pumping vessels. Some large passenger and cargo motorships for river and sea service and motor tankers have also been built. At the present time the Sormovo Works are stated to be in a position to manufacture marine boilers of the most modern type and machinery of every description. Extensive use is made of electric welding for shipbuilding purposes and the hulls of most of the vessels now in hand are of all-welded construction.—*"Soudostroienie", Vol. 10, No. 1, January, 1940, pp 4-5.*

Marine Engineering Works and Shipyards in North Russia.

The activities of the marine engineering and shipbuilding establishments of the St. Petersburg region of Czarist Russia were almost exclusively confined to naval work and it was only in 1926-27 that the Soviet Government began to devote serious attention to the construction of mercantile tonnage. In order to increase the productivity of the shipbuilding industry and to accelerate the development of the resources of the various works and yards in and around Leningrad, it has since been found desirable to classify these in such a manner as to ensure that the manufacturing facilities available should be used to the best advantage on work of the kind that the establishment in question is able to carry out. Shipbuilding is now proceeding in five of them on the following basis: Baltic Works—hulls and propelling machinery of large merchant vessels and machinery for ships built at the North Yard; North Yard—construction of hulls of sea-going merchant ships of all classes of from 2,000 to 3,000 tons deadweight from components fabricated at other establishments, manufacture of wood and steel cabin furniture and fitting-out of passenger accommodation of all vessels; Marty Works—hulls and propelling machinery of large river craft and sea-going vessels of up to 2,000 tons d.w.; New Admiralty Yard—hulls of barges, lighters and small vessels for special harbour duties and river service; "Red Shipbuilder" Works—hulls and machinery of small river craft. Marine engineering work is carried on in the following six establishments: Baltic Works—gunmetal castings of every kind, ships' propellers, high-pressure steel pipes, electric motors for auxiliary purposes, machining of heavy crankshafts, construction of Diesel engines for ship propulsion and auxiliary purposes; North Yard—helical gearing for marine engines, high-speed turbines for marine propulsion and auxiliary purposes, electric welding work of all kinds, rotary pumps for marine use,; Marty Works (and associated Liebknecht Works)—large-scale production of ships' pipework of every description, auxiliary machinery for all purposes, standard castings and forgings, Scotch boilers for marine use, manufacture of rivets, nuts and bolts; "Red Shipbuilder" Works—ships' fittings and equipment in malleable cast iron, small oil engines for fishing craft and yachts, stampings and pressings for marine engines and ships' fittings, white metal for engine bearings; New Admiralty Yard—construction and equipment of ships' lifeboats, masts and rigging work; Ijorsky Works—large steel castings for marine purposes, heavy forgings for crankshafts, piston rods, etc., and rough machining of same, spindles and shafting of every kind, rolling mills for steel plates for ships and boilers, profiling work for ships' frames, anchors and cables, capstans, windlasses, steering gears and deck fittings, ships' galleys and bakeries, hulls of river service barges and pontoons.—*M. P. Tolkachev, "Soudostroienie", Vol. 10, No. 1, January, 1940, pp. 31-36.*

A Novel Hose Clip.

An improved form of hose clip, which is claimed to be free from the defects common to most of these fittings, has recently



Sketch showing construction of a new type of hose clip.

been patented by two Doncaster inventors, and is illustrated in the accompanying sketch. The clip is made in one piece from flexible strip metal, and the lugs for drawing the ends together are devised in such a manner that they will not bend inwards even when excessive pressure is applied to the drawbolt. One of the lugs is formed by doubling the strip back on itself, and the other is made by turning the end into a spiral of double thickness. The parts numbered in the illustration include the main body of the clip (*1*), the curled end (*3*) bent back at (*2*), and the spiral lug (*4*). The single thickness (*5*) is continued across the gap and overlaps the end in a manner shown by the sketch. Holes (*3a*) and (*4a*) are provided to take the drawbolt (*6*), and the centre part of the bolt practically touches the clip, so that any tendency for the bolt to bend due to excessive tension is avoided. A hook or ring (*7*) is provided for use in certain circumstances and this is inserted between the thickness (*8*) and (*9*) during the course of manufacture.—*"The Motor Boat", Vol. LXXII, No. 1,863, 6th April, 1940, p. 258.*

Launch of Italian Submarine.

The submarine "Francesco-Baracca", launched at Spezia on the 21st April, was laid down in September, 1938, and is due to be completed in the course of the present year. An outstanding feature of this type of "ocean-going" submarine is the large fuel capacity and radius of action. These craft are reported to be able to reach Italian Somaliland ports by the Cape route without refuelling. "The F-Baracca" has a length of about 248ft., a beam of 22ft. 4in., and a surface displacement of 1,036 (metric) tons, the draught at surface trim being about 15ft. 6in. The Diesel engines of 3,600 h.p. are designed to give the submarine a surface speed of 18 knots. The armament will consist of eight 21-in. torpedo tubes, two 3·9-in. guns and several 13-mm. A.A. machine guns. Five further submarines of this class are being constructed.—*"Journal de la Marine Marchande", Vol. 22, No. 1,100, 2nd May, 1940, p. 497.*

Water Circulation in A-type Marine Boilers.

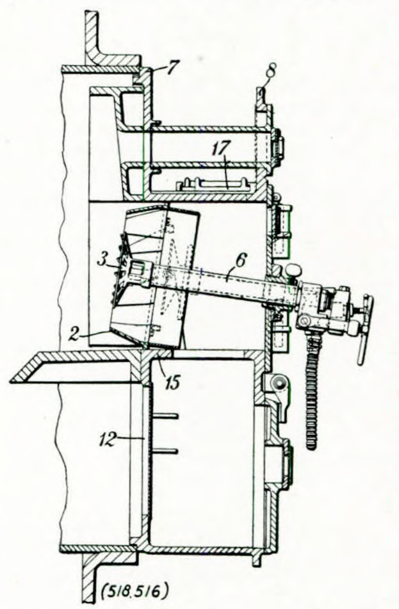
This paper presents a theory which purports to show the water flow conditions existing in an A-type (or 3-drum) boiler in the hope that it will lead to a clearer conception of the fundamental factors which govern the flow through the tubes. The author declares that a progressive increase in the permissible rating of such boilers should be practicable after the determination and elimination of limiting factors in design, which factors usually appear one at a time in succession as the previous limit is lifted. Circulation in A-type boilers has proved adequate for rates of combustion up to about 1lb. of oil fuel per sq. ft. of heating surface and up to about 45lb. of oil fuel per sq. ft. of radiant surface. Above these rates the circulation tends to become inadequate unless special precautions are taken. Hitherto, factors such as furnace design, burner design, and the type of fuel have all tended to make the limit of forcing come within the rate for which circulation was adequate, but at the present time these other limitations have been overcome and a high degree of efficiency is attainable with a small boiler at high rates by using economisers and air heaters, while furnaces and burners are of ample capacity. Nevertheless, a greater steam output can only be obtained by an increased water circulation, as the boiler rating is determined by the heat input which the tubes can absorb. The paper includes 11 charts and four diagrams.—*Paper by E. P. Worthen, "Journal of the American Society of Naval Engineers", Vol. 52, No. 2, May, 1940, pp. 257-294.*

A Water-circulation Indicator.

A visual indicator for liquid flow intended for use in the engine-cooling circuits of Diesel installations and similar purposes, has been developed by a Southampton engineering firm. The device consists of a partially buoyant ball in a protected glass tube. The ball is of a light composition which is claimed to be unaffected by hot water, and its surface is provided with indentations which cause it to revolve in the water stream, thus providing better visual indication and cleaning the interior of the glass tube. The weight of the ball is such that it cannot float and it is kept at an intermediate level whenever there is a water flow. The indentations on its surface make it impossible for a non-return valve action to occur, whilst the presence of a grid at the top of the indicator prevents the ball from closing the circuit at the outlet end. The tubular body of the device is of gunmetal and can be made with various forms of connnections for piping in all sizes from ½-in. bore upwards.—*"The Oil Engine", Vol. VIII, No. 85, May, 1940, p. 29.*

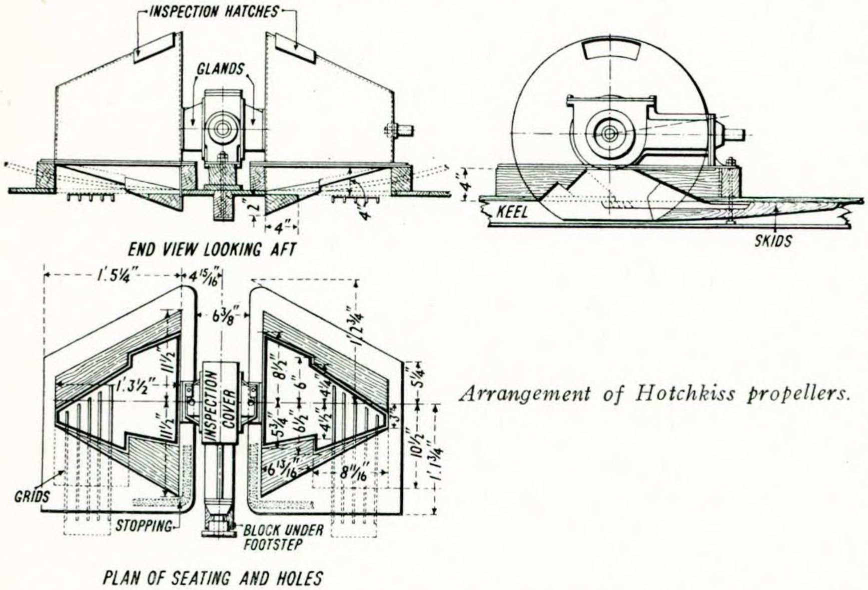
A Novel Form of Furnace Front for Cylindrical Boilers.

A recently published British patent concerns a means for improving the combustion conditions in the relatively restricted space of a cylindrical boiler furnace at different rates of firing. The furnace front of the Scotch marine boiler illustrated in the accompanying drawing consists of a burner nozzle operating by mechanical atomisation, a bladed port ring (*2*) surrounding the nozzle and an impeller plate (*3*), the port ring and impeller plate being vaned to cause rotation in the same direction of the air entering the furnace. The impeller plate is carried by a tube (*6*), which also carries the oil burner barrel. The surface of the impeller plate is conical, with its base towards the furnace, and its vanes discharge air with greater rotation than those of the port ring (*2*). The furnace front is circular and consists of inner and outer plates (*7*) and (*8*). Between these plates, below the horizontal centre line, is a circumferential wall, and forced draught enters between the plates above this level. The furnace front has two doors, the upper door carrying the tube (*6*) and the lower door serving as an ash door in the event of the furnace being coal-fired. An aperture (*12*) in the inner plate (*7*) also serves during coal firing to admit air beneath the grate, but during oil firing it is closed, and a plate which during coal firing occupies an aperature in a horizontal wall (*15*) between the two openings, is removed, so that air flows from the ash opening into the firing opening and thence past the burner into the furnace. Above the firing opening is a ported wall with a slide valve (*17*) which, during coal firing, regulates the flow of upper air to the furnace, but which during oil firing is fully open. Dampers regulate the flow of air to the ash opening, and their operating handle has a projection which moves in front of the firing door and prevents it from opening unless the dampers are closed. The port ring (*2*) is integral with a plate which fits the firing opening through which the furnace, after removal of the port ring, can be coal fired. The nozzle and impeller plate are axially adjustable with respect to the port ring, the tube (*6*) being arranged to slide through a mounting block, which is secured to the firing door. The furnace is fitted with a grate for coal firing, and the burner and port ring direct the fuel and air upwards away from the grate. The port ring (*2*) has a cylindrical unbladed portion, which renders it feasible to retract the nozzle and impeller plate behind the wider end of the converging part. The total length of the port ring with the unbladed portion is of the order of half the greatest diameter of the ring. When the nozzle and impeller plate are retracted, the spray angle is diminished and the range is increased, this feature being of use when altering firing conditions according to the load.—*"Engineering", Vol. 149, No. 3,881, 31st May, 1940, p. 554.*



Cone-propelled Launch for Burma.

The accompanying illustrations show the arrangement of the Hotchkiss propellers of a 25-ft. teak-built launch intended for inspection purposes by the P.W.D., Burma. The propelling machinery comprises a 13/30-h.p. Austin 6-cylinder marine engine which drives the Hotchkiss cone-propeller equipment through a spiral bevel reducing gearbox. The mild-steel impeller blades have a large clearance, so that in the event of a blade being bent through striking a branch of a submerged tree in the Irrawaddy or its tributaries, the blade can be removed



Arrangement of Hotchkiss propellers.

for straightening through the inspection hatches, even with the boat fully loaded. Ordinary rubbish and weeds have no effect on the impeller blades which are almost completely within the hull contour, grids being used to prevent any objects from being drawn into the intakes. Bilge-suction connections are fitted at the smaller ends of the cones, water under pressure for cooling purposes or for fire extinguishing being supplied through a discharge connection at the larger end of one of the cones. The launch is fitted with a large balanced rudder to ensure a high degree of manœuvrability combined with accurate steering astern, for which purpose about 75 per cent. astern speed is available. The full ahead speed of the boat in deep water is 10 m.p.h., with a loaded draught of 10in. The impeller shaft is placed athwartships and the two impellers are used rather in the fashion of the wheels of a car, each rotating in a conical casing of galvanised steel. These cones are mounted about amidships, with their lower sides open to the water, and the centrifugal force of the rotating water causes a spiral flow, expanding towards the larger end of the cone, from which the water is discharged aft or forward according to the direction of rotation, the acceleration of the water providing the thrust which propels the boat.—*"Shipbuilding and Shipping Record", Vol. LV, No. 18, 2nd May, 1940, pp. 444-445.*

Turbo-electric Tugs on the Mississipi.

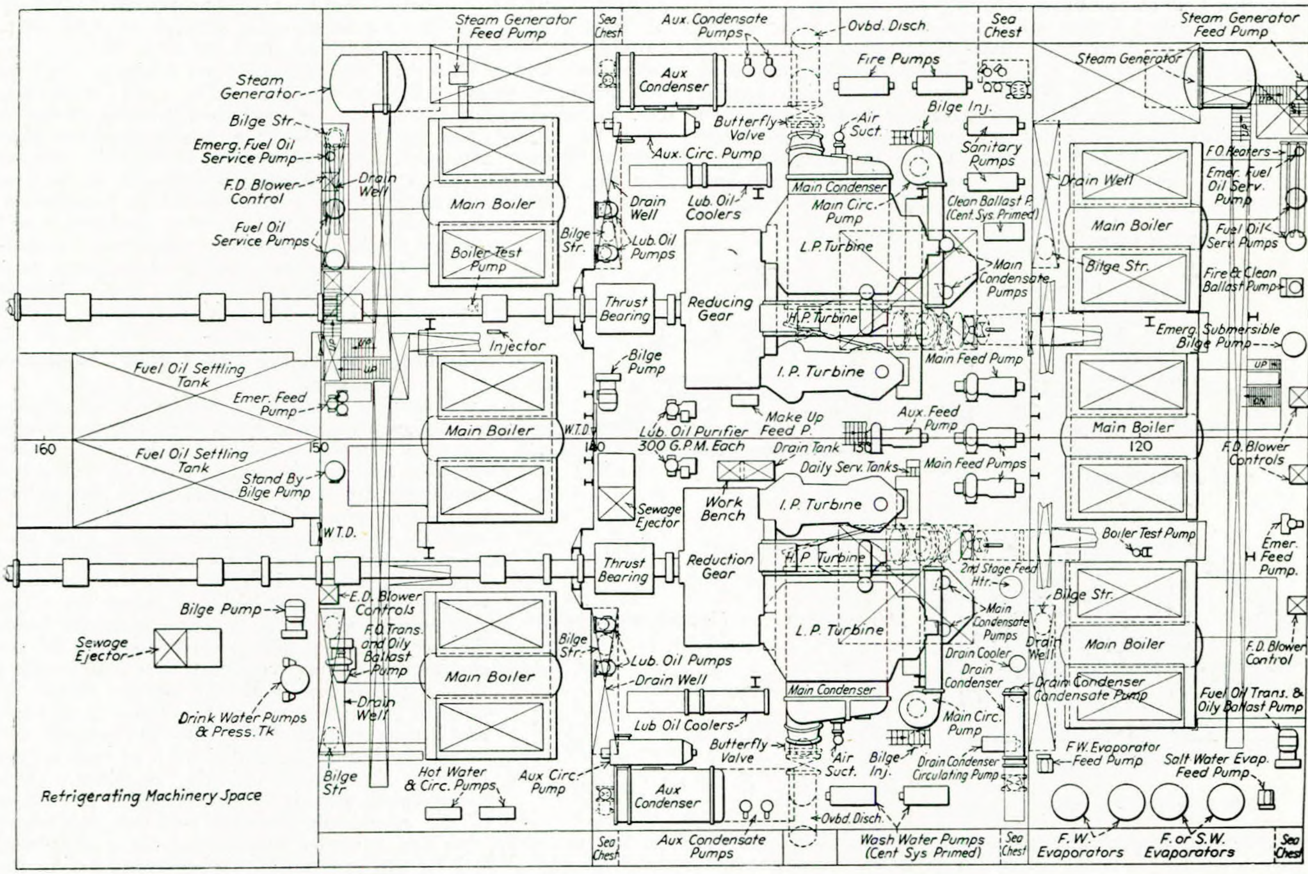
Amongst the numerous steam and Diesel-engined craft used on the Mississipi River, with practically every type of propulsion, it is reported that the Mississipi Barge Line Co.'s twin-screw turbo-electric river tankers "Indiana" and "Louisiana" may be regarded as the most economical. They are sister ships with a length of 200ft. and a beam of 40ft., the maximum draught with a load of 3,600 barrels of oil being 8½ft. These tankers cover the 1,483 miles from Cincinnati to New Orleans in 192 hours, of which about 162 hours are actual running time, so that their average down-stream speed is about 9·2 m.p.h. The up-stream trip only admits of an average speed of something like 4·58 m.p.h. against a current of nearly 2·3 m.h.p. The propelling machinery of these vessels consists of a single main turbine of 2,000 s.h.p. running at 3,600 r.p.m. and driving two 815-kW. 600-volt d.c. generators and a 110-kW. 110-volt exciter on a single shaft at 900 r.p.m. through reduction gearing. The twin propellers, 8ft. 9in. in diameter and 6ft. 1in. in pitch, are directly driven at from 180 to 194 r.p.m. by two 1,000-s.h.p. double-armature propulsion motors, each with a normal output of 815 kW. Steam for the main turbine is supplied by two Babcock & Wilcox boilers. The total weight of the entire machinery installation, comprising boilers, main turbine, generators and exciters, reducting gearing, condenser, propulsion motors, shafting, thrust blocks and propellers is about 245 tons.—*J. L. Bogert, "Marine Journal", Vol. 67, No. 5, May, 1940, pp. 25-28.*

Propelling Machinery for New American Coasters.

The U.S. Maritime Commission are preparing designs of standardised propelling machinery for small vessels primarily intended for coasting service. Several alternative types of propelling machinery are to be available for a single hull and it is anticipated that specifications and drawings will shortly be ready for issue to firms proposing to tender. Two high-speed, supercharged Diesel engines of relatively small size and light weight, each of 1,500 h.p., are called for. The two engines will drive a single propeller shaft through reduction gear located forward of the engines, with the shaft extending aft between them. The engines will be connected to the reduction gear pinions through electric couplings, and a generator of 300 to 400kW. capacity will be mounted on the extended shaft of each engine between the latter and the coupling. This arrangement will, it is stated, enable the whole of the electrically-operated auxiliary machinery to be run while the ship is under way with only a fractional load on one or both of the generators. When manœuvring the ship the normal procedure will be to run one engine ahead and one astern and to drive the propeller ahead or astern by energizing the appropriate coupling. When working cargo in port with the deck winches one engine will be run as a Diesel-generator set, with its coupling disconnected. This arrangement will provide ample power in port and will always leave one engine available for overhaul or maintenance work. A small Diesel-generator set for auxiliary purposes will also be installed, and in the event of circumstances arising which might involve shutting down both main engines at sea, the auxiliary Diesel-generator set will be started up automatically immediately the current from the main generators fails. This set is to be completely independent of the main engines, having its own fuel tank, cooling system, etc. It is probable that radiator cooling will be used.—*"Motorship and Diesel Boating", Vol. XXV, No. 5, May, 1940, p. 271.*

Machinery of U.S. Liner "America".

The machinery arrangement is shown in the accompanying plan. Each set of turbines consists of one H.P., one I.P., and one L.P. turbine in series, the total output of both sets being 34,000 s.h.p. at 128 r.p.m. of the propellers. The astern turbines comprise H.P. and L.P. elements incorporated in the I.P. and L.P. ahead casings, the total astern power being 19,500 s.h.p. at 95 r.p.m. of the propellers. The main condensers are located beneath the L.P. turbines. The reduction gears are of the double helical type, the H.P. turbines driving through D.R. and the I.P. and L.P. turbines through S.R. gearing. The six Babcock and Wilcox 3-drum express-type boilers have a total maximum evaporative capacity of 346,000lb./hr., with feed water at 300° F. The normal working pressure is 425lb./in.² at the superheater outlet, but the boilers are designed for a maximum steam pressure of 500lb./in.². The superheat temperature is 725° F. and a saturated steam connection is provided for each

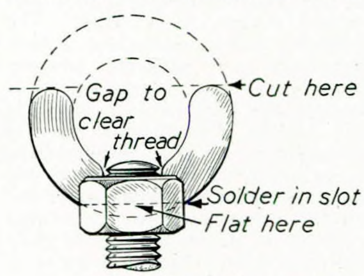


Arrangement of engine and boiler rooms in the "America".

boiler. The normal air temperature at the heater outlets is 307° F. and the funnel gases leave the heaters at 312° F. Each boiler is fitted with six mechanical-atomising fuel-oil burners. There are six motor-driven forced-draught fans of the turbovane type—one for each boiler—drawing air from the boiler rooms and discharging independently to the air heaters on each side of the boilers. Superheated steam is supplied to the main turbines, generator turbines, main and auxiliary turbo feed pumps, main circulating pumps, main and auxiliary air ejectors and H.P. evaporators. De-superheated steam is used for the fresh water and salt water evaporators, ship's heating system, the F.W. heaters, laundry, galley and drain condenser air ejectors. Saturated steam direct from the boiler drums is used for the whistles and reciprocating steam pumps. The contaminated steam system is entirely independent of the boiler feed system and is connected to the fuel-oil heaters, oil tank heating coils and swimming pool salt water heaters. All pumps normally in operation are motor driven except the feed pumps, but steam reciprocating pumps are also provided for emergency use. A central vacuum priming system is installed which serves all centrifugal pumps which have a suction lift and permits the use of standard pumps without separate vacuum-producing equipment. The system includes a vacuum tank of 25 cu. ft. and two automatic start-and-stop motor-driven vacuum pumps.—*"The Journal of Commerce" (Shipbuilding and Engineering Edition), No. 35,045, 30th May, 1940, p. 3.*

An Improvised Wing Nut.

A quick-release fly-nut can readily be made from a bright-steel hexagon nut of the required size by slitting it across the corners to half the depth, by means of a milling cutter or hack-saw blade. The cut must be made wide enough to take the thickness of a bright-steel washer, approximately twice the diameter of the nut. A gap must be cut in this washer of a width equal to the thread diameter, and a flat to suit the width across the corners of the nut cut at right angles to this gap along the outer edge of the washer, which should then be soldered or brazed into the slot with the gap level with the hole in the nut (as shown in the sketch). The washer should then be cut parallel to the top of the nut, so as to leave rather more than half on the nut. Finally, a tap should be run through the nut to clear the solder and the corners of the washer should be rounded off with a file. If the nut is chamfered at both sides, this will obviate the use of a washer underneath it, while if additional bearing surface is required a washer may be soldered or brazed to the under side of the nut during the making.—*D. C. Nicholas, "Practical Engineering", Vol. 1, No. 18, 25th May, 1940, p. 714.*



Improvised wing nut.

New Diesel Generators in Motorship "Silverash".

In January, 1939, while the Silver Line's 5,311-ton cargo motorship "Silverash" was at New York, a disastrous fire in her engine room resulted in severe damage to the Doxford main engine and to most of the electrically-driven dynamos. The ship was subsequently towed from New York to Sunderland, where extensive repairs were carried out. As the fire had completely destroyed the refrigerating machinery and the insulated holds were badly damaged, a new and far more powerful refrigerating installation was fitted on board and the insulated cargo space was trebled. The ship's three 100-kW. Diesel-driven dynamos were replaced by three 125-kW. generators driven at 550 r.p.m. by 6-cylr. four-stroke National Diesel engines fitted with compressed-air starting motors over the flywheels. The reconditioned "Silverash" left the U.K. in January, 1940, for further service between the U.S.A. and Far Eastern ports.—*"The Shipbuilder", Vol. XLVII, No. 370, June, 1940, p. 239.*

Incidence of Breakdowns of Diesel Engines.

According to the statistics compiled by the deputy chief engineer of the Hartford Inspection and Insurance Company relating to breakdowns caused by defects in industrial Diesel engines, the occurrence of such defects or failures of various engine components examined by the Company's inspectors took place as follows:—

Bearings ...	30·0 per cent.	Connecting rods	4·5 per cent.
Cylinders ...	24·3 ,,	Gears	4·5 ,,
Pistons and rings ...	15·3 ,,	Fuel pumps ...	2·8 ,,
		Starting gear ...	2·8 ,,
Valves	7·3 ,,	Exhausts ...	2·3 ,,
Crankshafts ...	4·5 ,,	Miscellaneous ...	1·7 ,,

Thorough examination of crankshafts and other shafts at regular intervals by means of magnetic crack detectors is stated to have resulted in a substantial reduction in the number of cases of failure of such components. As regards the repair of defective shafts by welding, it must unfortunately be admitted that the results have in general been disappointing; in most cases such repairs only endured for a few months.—*"The Locomotive", January, 1940, summarised in "Bulletin Technique du Bureau Veritas", Spl. No., May, 1940, p. 45.*

Building-up Worn Shafts by Spraying.

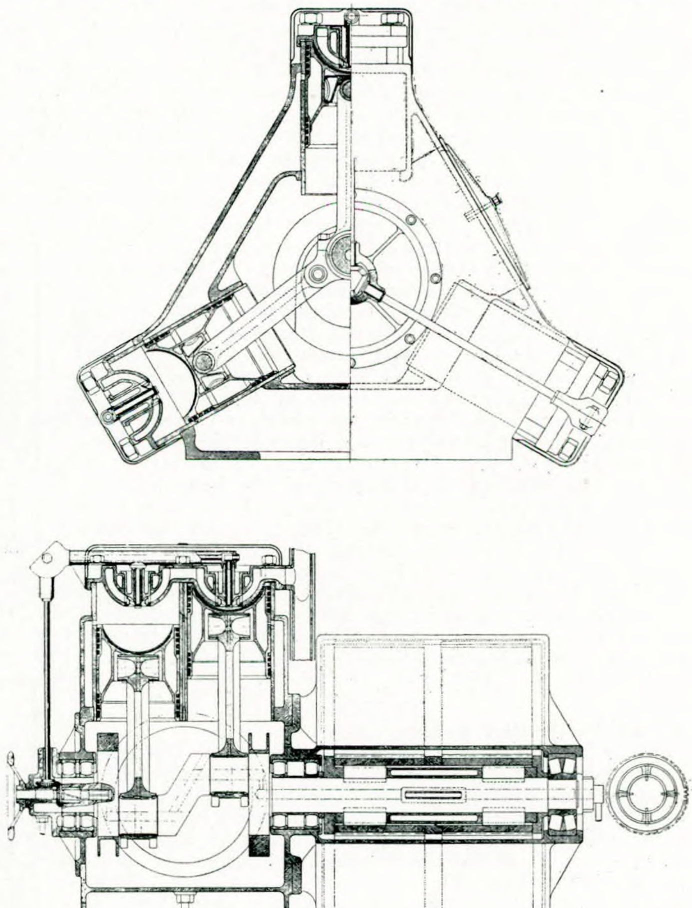
Worn shafts can be successfully built-up by metal-spraying if they are machined down 0·04-0·05in. undersize so that even with maximum wear the actual load between the shaft and the sprayed metal is not reached. The shafts should then be rough-threaded in a lathe to about 25-30 per inch. If the shaft will not have to withstand vibratory or endwise thrusts the tool should be ground with no rake so that it has a tearing action, jagged edges promoting adhesion. For shafts which have to withstand vibration or end thrust a round-nosed tool should be used, followed by sandblasting. With some alloy steels a highly polished surface is developed: this should be finished by sandblasting. Parts to be protected from the metal spray should be covered with electrical insulating tape or coated with a film of blacklead, keyways being filled with the keys smeared with the same material.—*C. Roade, "The Machinist", Vol. 84, No. 16, 8th June, 1940, p. 191*E.

New Lentz Marine Steam Engines.

The well-known marine engine designer Dr.-Ing. E. L. Lentz has recently developed an improved form of reciprocating steam engine for marine propulsion, which is claimed to possess all the advantages of a multi-cylinder crossheadless internal-combustion engine as regards construction, lightness and ease of operation. The general design of the new "steam motor" approximates closely to that of a Diesel engine and it is intended to be run at a high speed, driving the propeller shaft through gearing. The unit operates on the parallel-flow principle, taking steam at a very high pressure on one side of the pistons only. The steam inlet valves of the individual cylinders are of the poppet type, operated by a camshaft, and exhaust takes place through ports in the cylinder walls. The employment of trunk-type pistons enables the overall height of the engine to be kept down to a minimum. Reversal of the direction of rotation is effected by displacement of the camshaft and the designer claims that exceptional manœuvrability is an outstanding feature of his steam motor. The output can be increased, if required, by varying the point of cut-off and by the admission of superheated steam to special H.P. cylinders located before the main cylinders. The condenser is built into the bedplate of the engine. Two standard types of the new steam motor are to be manufactured, the first being a 6-cylinder unit with an output of 250-1,250 i.h.p. at 900-2,000 r.p.m., and the second a 9-cylinder unit developing 375-1,875 i.h.p. at the above r.p.m. By grouping two or more steam motors together to drive a propeller through gearing, it is claimed that machinery installations with an output of 7,500 i.h.p. per shaft can be made available. An installation of this type comprising two 600-h.p. steam motors driving a single propeller through S.R. gearing would only weigh about 3 tons (or 5·6lb./i.h.p.), the weight of each engine being about 1,322lb. and that of the gearing 1·75 tons, while the corresponding weight of the necessary boiler installation would not exceed 66lb./i.h.p. Dr. Lentz has also applied the above principles of construction to a radial steam engine of extremely compact design. He has built one for demonstration purposes, with six cylinders arranged in two groups of three around two cranks and developing 600 i.h.p. at 900 r.p.m. The cylinders have a diameter of 600mm. and the piston stroke is 300mm. The engine can operate on superheated steam at any pressure up to ·735lb./in.^{2} and temperature up to 932° F. The steam consumption at full power is reported to be 6·6lb./i.h.p.-hr. It is proposed to instal a number of these Lentz steam motors in vessels navigating the inland waterways of Germany in order to test them under service conditions.—*"Schiffbau", Vol. 41, No. 4, 15th February, 1940, p. 54.*

A Revolutionary Marine Steam Engine.

With reference to the foregoing abstract, the accompanying drawings are taken from an advertisement of the new Lentz radial engine which appeared in a recent issue of *Werft * Reederei * Hafen.* As may be noted the steam inlet valve is



General arrangement of the new Lentz uniflow marine steam engine.

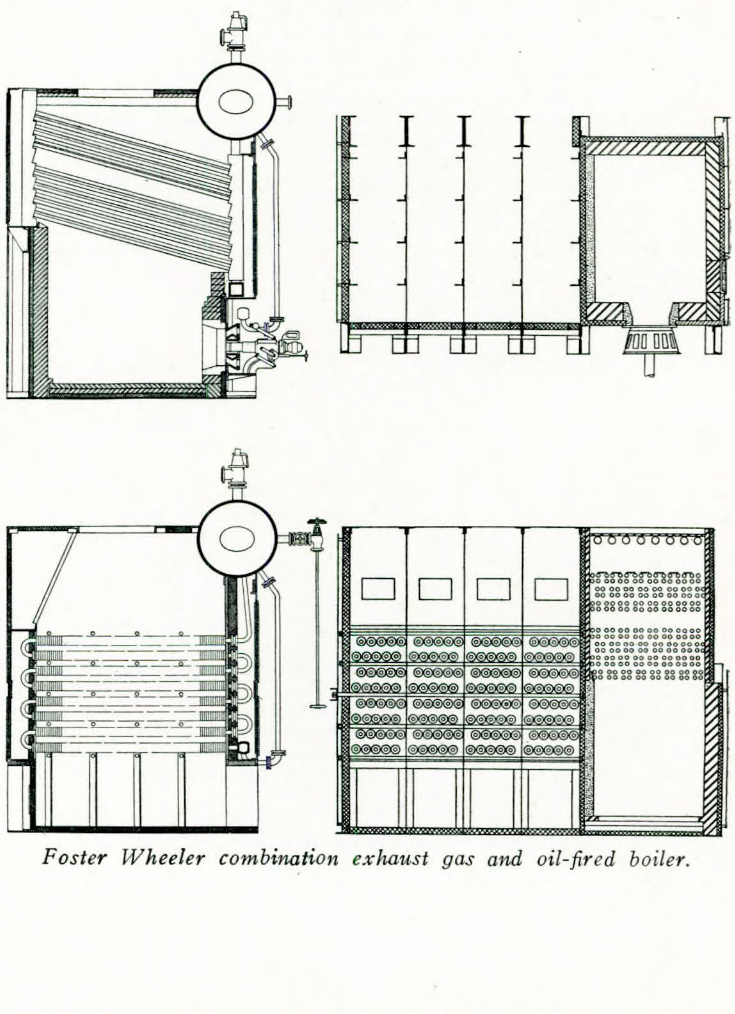
placed at the centre of the cylinder cover, each pair of cylinders in the double-bank engine illustrated having a common cylinder-cover casting. These inlet valves are operated by an overhead camshaft as in certain types of petrol engine, there being three overhead camshafts for the engine. These, it will be observed, are driven by bevel gearing. The actual valves appear to be similar to the double-beat poppet valves which are a usual feature of Lentz designs. Exhaust takes place through ports in the cylinder liner and it will be noted that the stroke is appreciably less than the cylinder bore. Direct reversing by hand setting of the camshaft is provided and increased starting torque is obtainable by simply increasing the cut-off during starting. The drawings show that a compact arrangement of gearing is envisaged, the design of the pinion shaft being interesting. No information is available as to the materials used for the construction of the engine, but presumably light alloys figure prominently in the proposals. The exceptionally low specific steam consumption claimed as well as the weights given must be based upon experimental data, otherwise the designer and pioneer users are likely to be disappointed when the first commercial engine is tried out. The two-bank type illustrated is apparently intended for cargo ship work. The double-star lay-out is favoured because it relieves the load on the roller-type main bearings and allows complete balancing to be achieved; the balance weights on the webs of the drilled two-bearing crankshaft will be noted. The steam jacketing of the cylinder covers and the insulating of the light hollow trunk-type pistons by evacuation should be effective in minimising steam consumption. Lubrication of the totally-enclosed running gear, including the gudgeon pins, is by a copious supply of oil which also serves to carry away the heat generated by friction. A split-type big end for the master connecting rod is used, with articulated rods for the other two cylinders. The piston speed adopted is said to result in minimum wear and maximum reliability, but the actual value is not given. The low steam consumption and guaranteed mechanical efficiency of 92 per cent. at full load should show remarkably low daily fuel consumptions, while it is claimed that only one-fifth of the cooling water necessary for an equivalent steam turbine is required. The condenser is therefore much smaller and the auxiliaries considerably simpler, lighter and smaller than would otherwise be the case. The high speed of the engine reduces its dimensions in a striking manner, the saving in weight being 85 to 92 per cent., while the size is only some 8 to 15 per cent. of machinery of "normal" design. Steam piping is small due to the low steam consumption and the permissible higher steam velocities, and is secured against radiation losses by the fitting of an exterior pipe around the main steam pipe, a vacuum being maintained in the annular space between the two pipes. The effectiveness of this novel method of insulation ensures a high standard of economy and also helps to maintain a low temperature in the engine room. Although the average temperature inside the cylinders is of the order of 500° F., low-quality lubricating oil is not burnt because, unlike the I.C. engine, there is no air available for such burning. The entire design is simple and certainly appears to lend itself to easy and cheap mass production. Most of the information forming the subject of this article is taken from the German paper *Hansa.—"The Marine Engineer", Vol. 63, No. 754, May, 1940, pp. 107-108.*

A New Waste Heat Boiler.

The U.S. Maritime Commission's series of C-3 motor cargo vessels—of which the "Mormacpenn" is the first to enter service—are propelled by four single-acting 2-stroke Busch-Sulzer engines, each rated at 2,250 b.h.p. at 240 r.p.m. and geared to a single propeller shaft through electric couplings, the normal shaft output, allowing for losses, being 8,500 s.h.p. It may happen, not only during manœuvring, but also on other occasions, that all of the engines will not be running together, and it was therefore decided to instal a Foster Wheeler exhaust-gas boiler with four separate waste-heat sections. This allows any individual engine to be shut down without affecting the operation of the boiler, apart from reducing the amount of heat available for raising steam. If desired, one section of the waste-heat boiler may be cleaned after shutting down the corresponding engine without interfering with the working of the other three units. At the end of the four exhaust-gas sections is the direct oil-fired combustion chamber, the steam drum extending across the entire unit with separate connections to each section. This drum has a diameter of 36in. and is about 16ft. long. In the waste heat-fired sections are 2-in. diameter horizontal tubes upon which are shrunk gilled annular castings arranged in the manner shown in the sectional drawing. In the oil-fired section there are straight tubes which are expanded into forged-steel headers connected to the steam drum by 4-in. tubing. The bottom element receives water from a distributing header, and steam and water are discharged from the top element into the steam and water drum. With the engines developing full power it is estimated that the amount of steam raised should be something over 0·5lb./b.h.p. The exhaust-gas temperature of the 2-stroke main engines is lower than with 4-stroke units, so that the steam production is correspondingly less. The boiler is installed in the engine room well above the level of the engines. It is claimed that the exhaust-gas sections may be operated dry for a considerable period without damage.—*"The Motor Ship", Vol. XXI, No. 244, May, 1940, p. 43.*

Docking Float Built in Record Time.

A new speed record for construction of steel-welded river craft is claimed by the Dravo Corporation's Neville Island shipyards (U.S.A.) by their completion, early in April, of a 175×26×6ft. landing float within a period of only 14 days from the contract date, without any previous design or drawings. When the contract was awarded on 29th March, ten draughtsmen were instructed to prepare the necessary drawings. These were completed on the following day and fabrication of steel was begun on 31st March. The production plan divided the

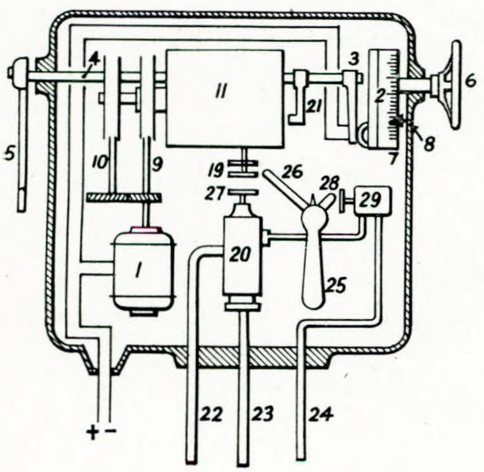


Foster Wheeler combination exhaust gas and oil-fired boiler.

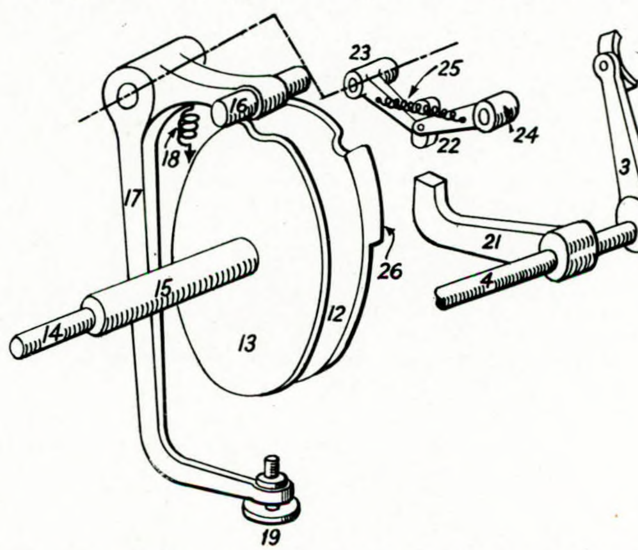
structure into 56 shop units—22 deck sections, 22 bottom sections and 12 wide sections. These units were fabricated, fitted and welded into pre-assemblies prior to moving to the barge shop for final assembly. Six days after the order these shop units were ready for final assembly and this work was then proceeded with, the 56 sub-assemblies being joined and completely welded by 12th April.—*"The Shipping World", Vol. CII, No. 2,451, 5th June, 1940, pp. 665-666.*

Automatic Blow-down.

An interesting automatic control system for boiler blow-down cocks, intended to permit the evacuation of the sludge formed in the boiler by the water softeners used, has been developed in France. The control is based on the idea that the formation of sludge will depend upon the amount of steam produced. By providing a system which will open the blow-down valves for a sufficient length of time to permit drainage of the sludge only, at intervals the frequency of which is dependent upon the consumption of steam, complete freedom from sludge can be maintained, it is claimed. The control system is enclosed in a case, all the parts except the actual control cylinder mounted on the drain valve being contained in this case. The control is operated by a small electric motor (1) fed from any convenient source of current, such as a lighting circuit, at a constant voltage, and the speed of this motor is regulated by a differentially-mounted on a bell-crank lever (17) and is held in its position against the cams by a spring (18). Each of the two cams has an indentation at a point along its circumference. The indentation on the cam (13) has an angular opening sufficiently large to permit the drain cocks to open and close, while the cam (12) has one large enough to permit cam (13) to make one complete revolution while cam (12) travels through the distance represented by the angular opening of its indentation. Operation of the system is as follows:—The roller (16) in contact with the edge of the two cams is maintained in a raised position by the cams. When the indentations of the two cams happen to coincide at the point of their circumference at which the roller rests, this roller can drop down under the pull of the spring (18). When this occurs, the pressure knob (19) at the end of the lever (17) is pressed against the control (27) of the air valve (20), admitting air to a control piston on the drain cock. With this system it might happen that the engine might be stopped at the moment when the drain cock was open. Since the motor (1) stops with the regulator of the engine, this would result in holding the cock open so long as the engine was stopped, and end in excessive blowing-down of the boiler. To avoid this, a safety device has been added which closes the cock when the engine stops, and holds it shut for a certain time after restarting. This is accomplished by a contact lever (21) mounted on the same shaft (4) as the arm (3) controlling the contacts of the rheostat.



Diagrammatic arrangement of automatic boiler blow-down device.



mounted rheostat. (2) This is arranged by having the sliding contacts of the rheostat mounted on a arm (3) carried on a shaft (4) which is connected by a lever (5) and a suitable tie rod to the regulator of the machine being fed. When the regulator opens, the contact arm (3) progressively cuts out resistances on the rheostat and speeds up the motor. When the regulator is closed, the contacts are brought back to a maximum position, in which the resistance on the rheostat is sufficient to stop the motor. In order to permit regulation of the system to take into consideration the degree of hardness of the water, the rheostat itself is mounted on a shaft which ends in a wheel (6) outside the enclosing case. A graduated scale (7) on the rheostat, moving in front of an indicator (8) permits this regulation. Speed of the motor (1) is thus regulated to the double variables of degree of hardness of the water and amount of steam produced. A pair of gears (9, 10) is driven by the motor, the two geared shafts operating at different speeds. The gear ratio is so determined that one will revolve more rapidly than the other in a ratio determined by the system of softening used and the amount of sludge formed by it. These two gears operate a pair of cams enclosed in a separate small case (11). One of these cams (12) is shrunk on to the shaft (14) driven by gearwheel (10), while the second cam (13) is on the shaft (15) driven by the other wheel (10). These two cams (12 and 13) both act upon a single roller (16), which presses against both their edges. It is

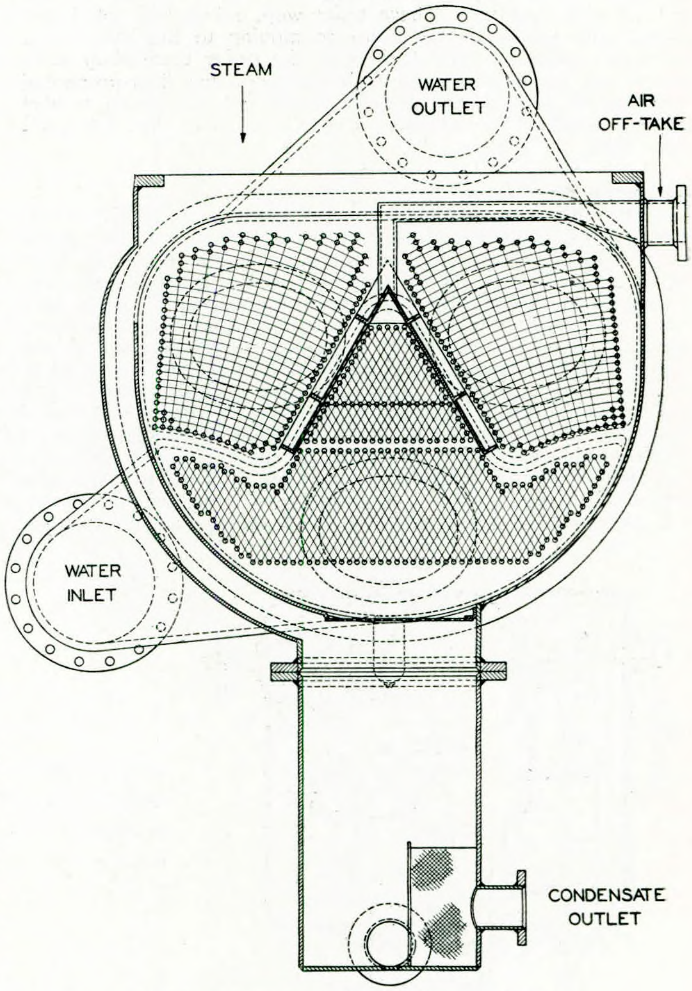
When the regulator turns the rheostat to the stop position, this arm (21) comes into contact with the roller (22) mounted at the pivot of the two half levers (23, 24) which are held by a spring (25) in either an upper or a lower position to form a cranked lever, as shown. One of the half levers (23) is mounted on the same shaft as the bell-crank lever (17) which carries the roller (16) acted upon by the cams of the system, so that whenever this half lever (23) is raised into its upper position the roller (16) is raised above the upper surface of the cams regardless of their position, and maintained there. A projection (26) on the side of one of the cams (12) strikes lever (23-24) as soon as the system again begins to revolve, returning it to its lower position and allowing the roller (16) again to rest on the cams and be acted upon by them. The air valve (20) of the system has a barrel containing a double valve, the two valves being on a single control rod (27), operated from outside by the contact of the "pressure knob" (19) on the end of the control rod. This rod is normally maintained in a raised position by a spring. In this position, the pipe (23) which runs from the air valve to the control cylinder on the drain cock is connected with the atmosphere by the opening of one of the two valves inside the valve barrel. On the other hand, when the pressure knob (19) presses the control rod (27) of the air valve down, this connection is closed, and the pipe (22) to the control cylinder is connected by the opening of the second valve in the air valve barrel to the source of compressed air (24). Manual control of the drain cock is also possible by means of a lever (25) on the front of the case containing the apparatus. On the outside of the case there is merely a handle and an indicator moving in front of three markings on the case representing stoppage of the system, automatic operation, and manual opening of the drain cocks. Internally, the shaft on which the handle is mounted carries two levers. One of these (26), when the handle is moved into the position "manual opening", presses against the control rod (27) of the air valve, operating the valve in exactly the same fashion as if the automatic control had worked. The second lever (28), when the handle is moved into the stop position, presses against a control valve (29) of the compressed air admission pipe (24), closing it and thus preventing automatic operation of the drain valves. The control cylinder, mounted on the drain cocks, is quite simple. It consists of a cylinder, compressed air being admitted under its piston by an opening in the end of the cylinder, connected by a tube to the air valve (20) of the control system. When this air valve sends air to the control cylinder, the piston is pushed upwards. When the air valve cuts off the admission of compressed air and opens the connection to the atmosphere, a spring on the control cylinder piston pushes the piston back into its original position. Mounted on the piston rod of this control cylinder is a pin which slides in a forked lever, fixed to the barrel of the drain cock. The position of the lever is such that when the piston is pushed by the incoming compressed air the cock opens, allowing drainage of sludge from the boiler, whereas when the compressed air is cut off, the return of the piston causes the cock to close. The cock can be replaced, if desired, by any type of valve.—*"The Marine Engineer", Vol. 63, No. 754, May, 1940, pp. 117-118.*

American Yard for Welded Ship Construction.

A recent development in the U.S. shipbuilding industry is the completion of the new Ingalls yard on the Gulf coast designed specially for the construction of all-welded ships. Eight 8,500-ton cargo vessels for the Maritime Commission are already being built there. The yard occupies a site of 50 acres, with nearly a mile of water front. There are long open shops and large erection platforms, with special tools and equipment, cranes, etc. The two-storey mould loft has plate and shape storage, and laying-out tables served by 15-ton bridge cranes. At right angles to it is the plate and angle shop, with overhead cranes and a petrol locomotive. It has its heavy machine-tool equipment arranged for straight-line movement, and houses an assembly platen of 150ft.×35ft. There are no punches, but burning tables for bevel burning. Manual welding with portable machines on the platens permits welding to proceed as soon as the plating is assembled. Automatic power welding machines are also provided. The furnace and forging shop has angle and plate furnaces served by overhead travelling cranes. All heavy machines are served by 5-ton electric jib cranes. The power house contains a 500-cu. ft./min. compressor to serve the pneumatic tools. Five building ways of reinforced concrete are available, with a slope of ⅝-in. to the foot, the launching ways being extended to a depth of 30ft. at low water. The ways are served by 35-ton revolving cranes on travelling gantry towers 66ft. high, with electric drive on a track of 20ft. gauge. The yard has a welding school and a special personnel department which selects employees for training. Owing to the absence of reaming and riveting machines, the work is practically noiseless, while the erection work is cleaner and there is less débris in the form of lost bolts, nuts, washers and rivets, and also ashes from rivet heaters. It is estimated that ships can be built far more quickly by the use of welding.—*"The Engineer", Vol. CLXIX, No. 4,404, 7th June, 1940, p. 514.*

An Interesting Marine Condenser Design.

With the wider use of high-economy water-tube-boilered geared turbine installations, the importance of the design of the condenser is of prime consideration. The condenser which is illustrated is a Foster Wheeler product and has been used in several new American steamships. It is of conventional shape and of the underhung type; in the case of some ships, the water-

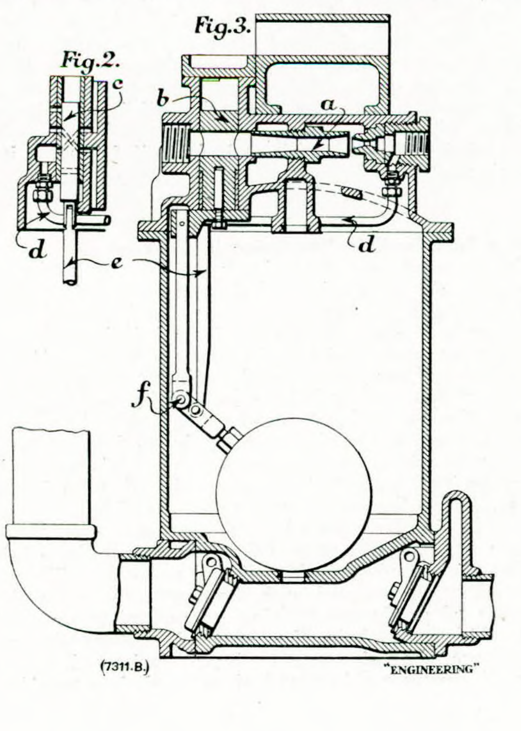


Cross-section of F-W. marine condenser.

box is arranged to give a two-pass water flow. The major part of the steam enters at the top of the shell, and is divided into two symmetrical paths by the inverted Vee baffle shown. The main tube nest is arranged on radiating line spacing, with sufficient cross section between tube rows to afford low velocity and low pressure drop. By this tube arrangement the condensing of the exhaust steam is gradual and the work of cooling is distributed uniformly throughout the entire tube bundle so that the steam is condensed by the time it reaches the lower part of the shell. Around one side of the all-welded steel shell is a steam by-pass to permit a small part of the exhaust steam to come into direct contact with the condensate as it drops into the hotwell. A perforated plate in the shell, just above the hotwell, breaks up the condensate into tiny drops and keeps a thin layer of water above the plate at all times. Thus the by-pass steam is restricted to the hotwell and reheats the falling condensate practically to the temperature corresponding to the prevailing vacuum. Since the pressure in the hotwell is approximately the same as that at the top of the shell, the amount of steam that enters the hotwell through the by-pass belt is automatically regulated by the quantity and temperature of the condensate. When the condensate is heated to the corresponding vacuum temperature, it will not condense any more steam and the flow of steam through the by-pass ceases until the pressure and temperature again falls. Air is driven off from the condensate by this heating and rises through a small vent to the air-cooling section immediately above the hotwell. This air together with the air and non-condensable vapours contained in the main exhaust steam is accumulated under the "V" baffle and drawn upwards through the air-cooling and devaporizing section, which is cooled by the incoming circulating water. The baffle arrangement makes it possible, it is claimed, to cool the air to within 5 or 10° F. of the inlet water temperature. This minimises the amount of exhaust steam carried off with the air, thereby reducing the steam consumption of the air ejector. This arrangement of main tube surface and air cooling surface is said to give a most efficient design, in moderate overall dimensions, especially height. Great attention has been given, the designers point out, to the proportion of air cooling space and to the close spacing of the tubes in this section of the condenser. Careful cross-baffling has enabled the velocity of the vapours to be kept high, with consequent improvement in heat transfer rates. This is most important, for if films of air display a tendency to gather on the tube surfaces heat-transfer efficiency falls.—*"The Marine Engineer", Vol. 63, No. 754, May, 1940, p. 104.*

Portable Pneumatic Sludge Pump.

A portable sludge pump, operated by compressed air and capable of handling water containing up to 15 per cent. of solids, has been developed by a well-known firm of pneumatic tool manufacturers in this country. It is claimed that the design is such that there are no expensive parts to be worn out by abrasion, any parts that are exposed to such action being readily and cheaply replaced. The construction of the pump is shown by the sectional views (Figs. 2 and 3) and it operates by a combination of the ejector and displacement principles, *i.e.*, the suction stroke is effected by creating a vacuum in the pump chamber by means of an ejector, the water thus drawn into the chamber being forced out by its displacement by compressed air, so that pumping is not continuous but cyclic, the discharge being a pulsating flow similar to that from a reciprocating hand pump or diaphragm pump. The total height of the pump is 24½in., the width of the base is 10¾-in. and the weight 95lb. The body of the pump consists of a cylindrical chamber with a rectangular base, a detachable cover (fitted with a lifting handle) giving access to the discharge valve, and a detachable box carrying the suction valve. This box, when removed, also gives access to the discharge valve for the clearance of obstructions, in case splinters of wood or pieces of wire drawn in by the suction should hold the valve open and stop the pump. The pump chamber is sealed



at the top by a cover carrying an ejector nozzle *a*, an ejector control valve *b*, and a pilot valve *c*. The compressed-air inlet, which is of ⅜-in. bore, is at the right of the ejector nozzle *a*, the exhaust branch, of 1-in. bore, being connected at the left, in line with the inlet, to a drum-shaped silencer. The pump suction, 2in. in bore, is seen at the right-hand at the bottom, with the 2½-in. discharge at the left. The control valve *b* travels vertically on a central guide rod which prevents it from turning, and is moved up or down by compressed air admitted and released by the pilot valve *c*, the supply to this latter valve being obtained from the air inlet through the pipe *d*. The pilot valve is actuated by the rod *e* attached at *f* to the arm of a float in the pump chamber. The water enters and leaves the chamber through the orifice seen below the float. As shown in Fig. 3, the several parts of the pump are in the position occupied at the completion of a discharge stroke, the float being then in its lowest position and sealing the opening. The control valve is also in its lowest position and the transverse port in it is in line with the ejector nozzle and exhaust outlet. The air supply has, in consequence, a free passage, and the action of the ejector creates a vacuum in the chamber. The control valve is held down by the air pressure admitted through the pilot valve, which, as shown in Fig. 2, is provided with diagonal ports so that when it is in its lowest position (as it is in Fig. 2) the pressure supply communicates with the top of the control valve and the bottom of that valve is open to the atmosphere. A vacuum having been formed in the chamber by the operation of the ejector, water is drawn into the chamber and the float ultimately rises to its top position. This movement causes the pilot valve to rise so that a horizontal port in it opens communication between the pressure supply and the underside of the control valve. The latter is thus driven upwards so that its solid part, covered with an abrasion-resisting bush, shuts off the ejector from the exhaust branch. The air supply is not, however, cut off at the ejector nozzle, but discharges into the chamber, driving the contained water out through the discharge valve. The float falls and the cycle then repeats itself. The pump has few working parts and no packed surfaces such as glands and pistons. Any want of tightness in the control valve will be indicated by air blowing through directly to exhaust, so that the defect can be corrected. When all water concerned has been discharged the air also discharges to exhaust and the supply cock is shut. The capacity of the pump is, of course, dependent on the pressure of the air supply. With air at 100lb./in.^{2} the discharge will vary from about 14·5 tons/hr. at a total head of 50ft. to one of about 4·5 tons/hr. at a total head of 200ft.—*"Engineering", Vol. 149, No. 3,884, 21st June, 1940, p. 610.*

New Diesel Engines for American Submarines.

An article in a recent issue of the technical periodical *Le Génie Civil* describes several new designs of Diesel engines for submarines, developed in the U.S.A. In 1932 the Bureau of Engineering invited a number of prominent American designers and builders of Diesel engines to submit designs for submarine engines weighing not more than 28lb. per b.h.p., no restrictions being imposed in regard to the cycle of operations, number or arrangement of cylinders, speed of rotation or mean effective pressure. Seven U.S. firms subsequently contracted to supply typical engines or units complying with these conditions and the whole of this machinery has now been subjected to comparative tests. F. B. Stearns, the pioneer of the valveless automobile engine in the U.S.A., received an order for a diamond-section 2-stroke engine with 8 cylinders arranged in pairs, the two lower pairs being arranged in V-formation and the two upper ones as an inverted Vee. This engine has an output of 320 b.h.p. at 1,300 r.p.m., the cylinders having a bore of 133mm. and piston stroke of 216mm. The total swept volume of the cylinders is 24,200 c.c. and the m.e.p. is 67lb./in.^{2}. The Winton Company (now known as General Motors) constructed a 12-cylinder 2-stroke V-type engine having an output of 950 b.h.p. at 720 r.p.m. The Sun Shipbuilding and Dry Dock Company produced a 6-cylinder opposed-piston 2-stroke engine with two crankshafts, the cylinders having a bore of 165mm. and piston stroke of 248mm., with a total swept volume of 63,100 c.c. This engine has a rated output of 510 b.h.p. at 650 r.p.m. with an m.e.p. of

80lb./in.^{2}. The engine frame is of aluminium. The fuel consumption trials produced remarkable results, the consumption of fuel oil proving to be only 0·358lb./b.h.p.-hr. with the engine driving an electric generator at 2,900 r.p.m. through D.R. gearing. A single-crankshaft version of this engine has also been constructed. The Continental Motors Corporation built a radial 2-stroke engine (similar in design to the 250-b.h.p. engines fitted in some of the U.S. army tanks), with 10 cylinders of 165mm. bore and a piston stroke of 190mm., the total swept volume being 40,800 c.c. The engine has an output of 480 b.h.p. at 1,100 r.p.m. with an m.e.p. of 70lb./in.^{2}. The Electric Boat Company, with their wide experience of submarine-building, preferred a supercharged 4-stroke V-type engine with 16 cylinders of 178mm. bore and 210mm. piston stroke, developing 635 b.h.p. at 1,150 r.p.m. and driving a generator running at 1,800 r.p.m. located under the engine, through gearing. The engine has an aluminium crankcase. The Fairbanks-Morse Company constructed an opposed-piston 2-stroke engine with two crankshafts. The Hooven-Owen-Rentschler Company produced a double-acting M.A.N.-type engine developing 1,300 b.h.p. at 700 r.p.m., having eight cylinders 230mm. in diameter with a piston stroke of 330mm., the volumetric capacity being 222,000 c.c. and the m.e.p. 54lb./in.^{2}. The fact that this engine weighs only just over 11lb. per b.h.p appears to indicate that the minimum specific weight prescribed by the Bureau of Engineering was far too high. As a sequel to the design competition of 1932, it was decided to instal opposed-piston 2-stroke Fairbanks-Morse engines with twin crankshafts in several submarines of the improved "Porpoise" class, while others were to have Winton, Nelseco and H.O.R. (M.A.N.) Diesel engines. These submarines have a length of 300ft., a beam of 25ft. and a surface displacement of 1,330 tons on a draught of about 13ft. 6in. Their 3,200-h.p. Diesel engines are intended to give them a surface speed of 17 to 20 knots. The armament is to consist of six 21-in. torpedo tubes, a 3-in. A.A. gun and a machine gun. The general design of these craft does not appear to be of a very advanced type, despite their ultra lightweight Diesel propelling machinery. No information is available concerning their radius of action, an all-important matter in the case of submarines. It is probable that these new underwater craft are considered as being of an experimental nature and that if their machinery proves satisfactory in service similar submarines with an appreciably higher surface speed are likely to be constructed.—*"Journal de la Marine Marchande", Vol. 22, No. 1,105, 6th June, 1940, p. 651.*

Speed of German Submarine Building.

During the war of 1914-18 the shortest time taken by the Germans to build their smallest "U-B" type of submarines (127-141 tons) was 100 days. These craft had a 160-h.p. Daimler Diesel engine and one 120-h.p. motor generator, their surface speed being 6½ knots and submerged speed 5½ knots. Their armament comprised two torpedo tubes. The "Deutchland" class submarines for blockade-running purposes were built in 4½-5½ months, but it took 11½ months to construct submarines of the U-51-56 type of 742 tons surface displacement, driven by two 1,200-h.p. engines. The submarines U 62-65, with a surface displacement of 860 tons and two 1,600-h.p. engines, were built in the same time. Similar submarines could of course be constructed more rapidly at the present time, but the question of obtaining trained crews to man them still remains. Many of the U-B submarines were built in sections for despatch by rail to different ports for erection and use, some being sent to the Adriatic and others to the coast of Flanders. The overall length of these small craft was 92ft., and the beam 10ft. 3in. The radius of action was stated to be 1,600 miles. The first boat of this class was completed 101 days after the order was placed. She was then dismantled and sent to Flanders for re-erection, and the following boats were sent off without erection and trials. —*"The Motor Ship", Vol. XXI, No. 245, June, 1940, p. 84.*

German Diesel-electric Ships Captured by Allies.

Among the German vessels captured by the Allies is the 7,400-ton Diesel-electric cargo liner "Wuppertal", completed in 1936 and equipped with an a.c. propulsion motor of 6,800 h.p. supplied with current at 2,000 volts by three 1,900-kW. generators driven at 250 r.p.m. by three 2,600-b.h.p. 7-cylinder single-acting 2-stroke M.A.N. engines. The propelling motor is a synchronous unit running at 125 r.p.m. supplemented by an induction motor of 900 h.p., running at 62·5 r.p.m. and fitted in the same housing as the synchronous motor. The purpose of the induction motor is to facilitate manœuvring the vessel at slow speeds, but as it has been found that the main synchronous motor is very simple and flexible to operate, the use of this "fog motor" as it is called, in later ships, is to be dispensed with. The "Wuppertal's" machinery installation also includes three converters to excite the machines, each comprising an induction motor and a d.c. generator. Only two of the converters are required for ordinary service, the third being spare or used for the auxiliary d.c. system for operating the ship's winches and windlass. The remaining auxiliary machinery runs on three-phase a.c. supplied by two 170-kVA Diesel-generator sets. The engine-room staff of the "Wuppertal" includes five engineers and two electricians. Her service speed, when fully loaded, is some 16 knots, the corresponding fuel consumption being about 32 tons per 24 hours. The second Diesel-electric ship seized by the Allies is the Hamburg-Amerika passenger and cargo liner "Vogtland", a twin-screw vessel of 6,608 tons gross, completed in 1924, as an ordinary geared motorship and re-engined in 1938. Her present machinery consists of five 8-cylinder 4-stroke Diesel engines direct-coupled to 600-kW. generators supplying d.c. at 350 volts to two propulsion motors which drive the twin propellers through gearing at 85 r.p.m. The two outboard engines also drive 125-kW. 220 volt d.c. generators for supplying power to the auxiliaries, but the 600-kW. main generator driven by No. 3 engine can likewise be utilised for this purpose, if necessary.—*A. C. Hardy, B.Sc., "The Journal of Commerce" (Shipbuilding and Engineering Edition), No. 34,051, 6th June, 1940, p. 3.*

Thinner Oils for Diesel Engines.

In a paper recently read at a meeting of the Institution of Engineers-in-Charge by Mr. D'Arcy Evans, the author gave particulars of the qualities which he considered desirable in lubricating oils for various types of machinery, including Diesel engines. He stated that he did not subscribe to the very prevalent practice of using a heavy oil of 400-500 Redwood viscosity at 140° F. in moderate-speed high-powered marine engines and referred to his own experience of using a straight mineral oil of good quality of 160-180 Redwood viscosity at 140° F. for lubricating and piston cooling in three different types of large marine installations. The result, he declared, showed a low lubricating-oil consumption, cleaner pistons and reduced liner wear. The author is now carrying out a 600-hour running test with an engine in which a specially inhibited lubricating oil is used to overcome the piston ring sticking so commonly experienced through the dilution of the lubricating oil with fuel oil following incomplete combustion.—*"The Motor Ship", Vol. XXI, No. 245, June, 1940, p. 72.*

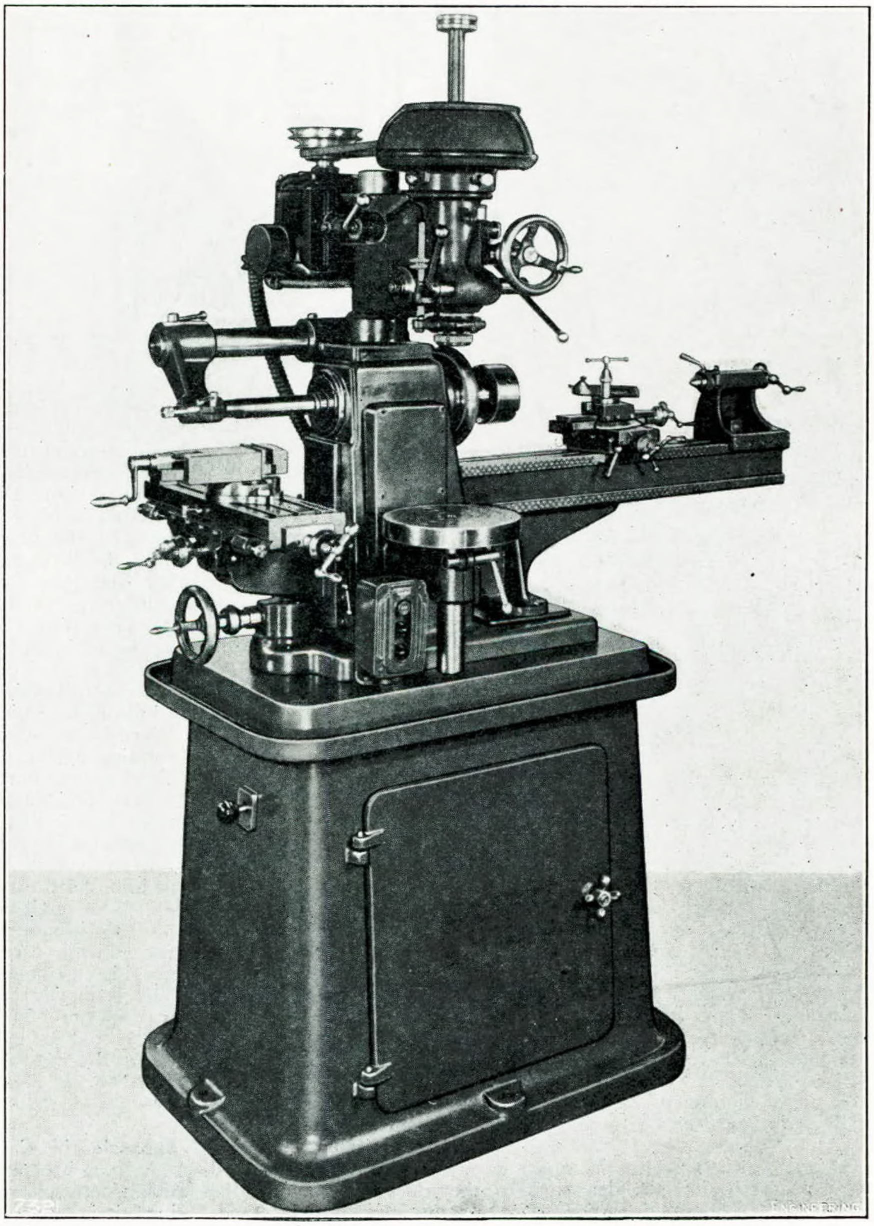
Peat as a Fuel for Ship Propulsion in Germany.

The growing use of producer gas in conjunction with I.C. engines for the propulsion of certain types of vessels on the inland waterways of Germany is causing increased attention to be devoted to the various kinds of solid fuels, mostly based on coal, wood or peat, available for gas producers. Minimum ash content and sulphur content, and minimum formation of tar are essential attributes of such fuels, in addition to ability to respond rapidly to the fluctuating demand of the engine by prompt alteration of combustion to suit variations in suction. Wood charcoal and peat coke possess these properties to a high degree, but wood is now required urgently in Germany in other vital branches of industry, hence increased importance attaches to peat and its products. Abundant supplies of peat are available in N.W. Germany and its utilisation offers the possibility of important reductions in the consumption of coal and coke. The unpleasant odour so often associated with the engine-exhaust and producer flue gases of producers running on anthracite or semi-coke is considerably reduced in the case of peat-burning producers. Furthermore, the ash content of peat is low (2 to 3 per cent. of non-clinkering residue) and gives much less trouble than the

15 per cent. or more of ash often associated with fuels derived from coal. It is also claimed that the natural moisture of peat avoids the necessity for the steam or water injection which adds to the difficulty of constructing and operating anthracite and coke-burning producers. On the other hand, the low density of peat, about 22lb./cu. ft., involves more frequent recharging of the producer, besides the provision of larger bunker space. These disadvantages are, however, not insuperable, and in the case of the river-tug "Bober", built specially for the use of peat, a bunker of adequate size is located immediately over the producer, thus avoiding the need for any conveyor or other equipment for the transfer of fuel. According to the German Peat Utilisation Department, further progress may be expected in the densification of peat by preparatory treatment. Tests indicate consumptions ranging from 1·41 to 1·68lb./h.p.-hr., while as regards the requirements imposed on the engine-room personnel, the difference between one producer fuel and another should not occasion any difficulty to men who have adapted themselves to the general operating routine of producer gas plant.—*"Shipbuilding and Shipping Record", Vol. LV, No. 23, 6th June, 1940, p. 586.*

Combined Lathe, Milling Machine and Drilling Machine.

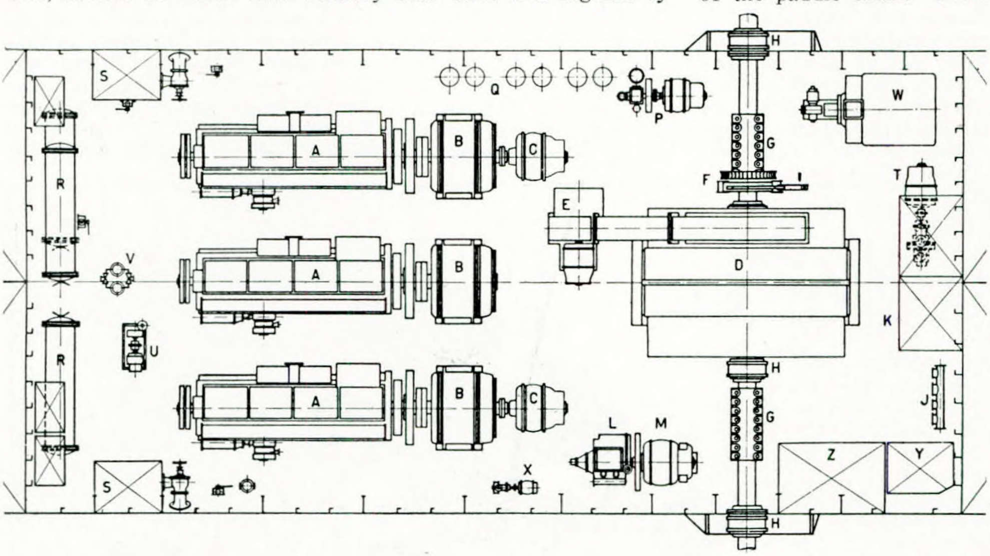
The accompanying illustration shows a combined machine, primarily intended for small workshops with limited space, which comprises a lathe, a horizontal milling machine, a vertical milling machine and a drilling machine. The horizontal milling machine at the left, the drilling machine in the centre, and the lathe on the right can all be readily identified, together with the driving motor at the top of the machine. The vertical milling machine is arranged for by removing the arbor of the horizontal milling machine and retracting and swinging the overarm out of the way. The drilling head is then rotated on the vertical column on which it is carried so that the spindle lies above the table of the horizontal milling machine and the drill spindle carries the milling tool. The vertical milling machine and the drilling machine only are driven by the upper motor, the control push-buttons of which are not visible. The upper motor is reversible and of ½-h.p. at 1,725 r.p.m. It drives the vertical spindle by a single V-belt on a four-step pulley, four speed changes, from 380 to 2,150 r.p.m. being obtainable by belt-shifting. The push-button panel seen near the drilling-machine table controls a 1-h.p. motor situated in the machine base, which also runs at 1,725 r.p.m. and drives the spindles of the lathe and the horizontal milling machine, 12 spindle speeds, from 85 to 1,250 r.p.m. being available. The horizontal milling machine has a knee-type support with a vertical traverse of 5¼in., actuated by a hand-wheel fitted with a micrometer dial. The support carries a saddle 7⅝-in. long by 6in. wide, and has a hand cross-traverse of 3in. The table has a T-slotted working surface of 15in. by 5in., and is provided with an adjustable gib on the saddle. The longitudinal hand feed traverse is 10in. Quick-acting locking screws and adjustable dog-stops are provided, while the table and saddle-traverse screws have precision Acme threads and are fitted with micrometer scales. The spindle is of heat-treated, hardened and ground chrome-nickel steel and runs in ball bearings. The maximum height between its centre line and the surface of the table is 5½-in. A swivelling vice with a graduated base, for attachment to the table, is provided for use when milling small parts. The lathe has a box-section bed of close-grained semi-steel and is long enough to take work of 18in. between centres. The maximum swing over the bed is 9in. The tailstock has ground gibs, for adjustment, on all bearing surfaces, and is fitted with a hardened and ground quill. A three-jaw chuck is supplied for use instead of the fixed centre. The spindle is fitted with a three-jaw universal chuck and the nose is formed to take adaptors having tapered shanks. The compound slide is suitable for precision work and although the motions are normally hand-operated, the machine can also be supplied arranged for screw-cutting, in which case a power feed and change gears are provided. The drilling machine has a table 9in. in diameter, adjustable for height, the maximum distance between the chuck and the table being 8in. The drill has a vertical traverse of 4in. and alternative methods of drill feed are provided, *viz.*, a hand lever for sensitive feed and a worm-gear feed actuated by a hand-wheel and having a dial graduated in 0·001in. The spindle is 1⅜-in. in diameter and is traversed in a 2½-in. diameter quill, both

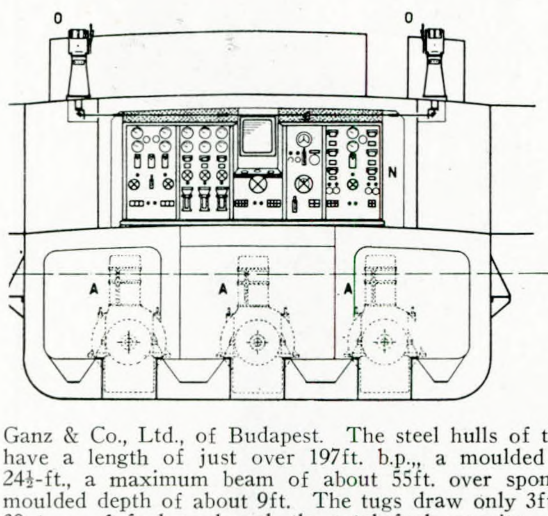


spindle and quill being of hardened and ground chrome-nickel steel. The quill is mounted in heavy ball bearings. The spindle nose is formed to take a No. 10 Brown and Sharpe collet. The machine is manufactured in Chicago and is distributed in the U.K. through a London firm.—*"Engineering", Vol. 149, No. 3,884, 21st June, 1940, pp. 600-601.*

Diesel-electric Paddle Tugs for Service on Danube.

The accompanying drawings show the arrangement of the engine-room machinery of the Hungarian tugs of the "Széchenyi" class, several of which have recently been built and engined by





Ganz & Co., Ltd., of Budapest. The steel hulls of these vessels have a length of just over 197ft. b.p.,, a moulded breadth of 24½-ft., a maximum beam of about 55ft. over sponsons and a moulded depth of about 9ft. The tugs draw only 3ft. 10in. with 30 tons of fuel on board, the total fuel capacity of the tanks forward and aft of the machinery space being 90 metric tons, which is sufficient for over 450 hours' normal running. The total weight of the 1,000-s.h.p. Diesel-electric machinery installation is very little greater than that of an oil-engined tug of equivalent power with direct screw propulsion or that of the engines and boiler of a steam paddle tug of 550 s.h.p. Moreover, the fuel carried in these two types of tugs is only enough for 150 hours' and 50 hours' running respectively. The deck machinery equipment of the "Széchenyi" includes an anchor windlass forward and a towing winch abaft the wheelhouse, both driven by 10-h.p. electric motors. The triple Hitzler rudder is operated by a 3-h.p. electric motor controlled by Ward-Leonard gear. Referring to the drawings of the machinery layout, the total length of the engine room is only some 50ft., notwithstanding the large amount of space available. The three main Diesel engines (*A.A.A.*) are 8-cylinder single-acting 4-stroke Ganz-Jendrassik units of the non-reversible type, with cylinders 216mm. in diameter and a piston stroke of 310mm. The normal power output of each engine is 400 b.h.p. at 750 r.p.m., and the engines are always run at that speed, irrespective of the speed of the ship. The normal m.e.p. is about 74lb./in.², which may be increased to about 85·3lb./in.² (or to any intermediate pressure) if required, a special feature of the installation being the Ganz-Jendrassik patent governor gear which controls the output of the fuel pumps on the engines according to the speed of rotation of the paddle shaft. Fresh-water cooling on the closed-circuit system is provided for each engine, for which purpose there are two centrifugal circulating pumps—for fresh water and river water, respectively—driven off each engine, with horizontal coolers (*R.R.*) and inlet strainers (*S.S.*) to deal with the notoriously muddy water of the Danube. A lubricating-oil filter of a well-known British make serves the three engines in turn, and although it is somewhat unusual, this arrangement has proved quite satisfactory in practice. The three Diesel engines are coupled by Voith-Mauser flexible couplings to three 280-kW. dynamos (*B,B,B*) generating direct current at 180 volts, the shafts of the two wing machines being extended to drive 30-kW. exciter dynamos (*C,C*) generating direct current at 220 volts for the field coils of the propelling motor (*D*). Under ordinary running conditions one exciter dynamo suffices for this purpose, the second machine being available for power and lighting requirements. There is, in addition, a 25-kW. 220-volt d.c. generator (*M*) for auxiliary purposes, driven by a 40-h.p. two-cylinder Ganz Diesel engine of the four-stroke type (*L*). This dynamo is likewise available for use as an exciter for the field coils of the propelling motor (*M*). When the tug is running light the three main generators (*B,B,B*) are operated in parallel, but when towing two—or all three—are run in series, the full voltage of 540 volts corresponding to 1,000 b.h.p. developed by the propelling motor (*D*) at 35 r.p.m. The maximum overload output of 1,200 b.h.p. of which the latter is capable, is developed at 45 r.p.m. The speed and direction of rotation of the motor are controlled from a switchboard (*N*) over the engine room, in addition to which bridge control gear (*O,O*) operated on the Ward-Leonard system, is fitted. The speed and direction of rotation of the propelling motor (*D*) are varied as necessary by altering the current in the field coils and this arrangement gives a very fine degree of manœuvrability, the speed of the paddle shaft being capable of regulation to within ½-r.p.m. and the time required to reverse the motor from "Full Ahead" to "Full Astern" (or *vice versa*) being between 15 and 20 seconds. The paddle shaft turns in six roller bearings (*H*) in which the friction is so small that the flow of the current past the paddles when the ship is moored turns them at a speed of from 2 to 7 r.p.m., for which reason an effective braking device (*F*) has had to be provided to keep the shaft from turning under these conditions. The reduction in draught due to 1 hour's fuel consumption under way is only 0·02in. in the case of the "Széchenyi" as against nearly 0·07in. in that of the steam tugs referred to above.—*V. Gras, "Werft * Reederei * Hafen", Vol. 20, No. 16, August, 1939, pp. 247-256, and "The Motor Ship", Vol. XXI, No. 245, June, 1940, pp. 100-102.*

Expansion in Air Compressor Crankshafts.

In a brochure bearing the above title issued by a firm of manufacturers specialising in a well-known make of roller bearings, it is suggested that an attempt should be made to calculate the difference in expansion between the length of the crankshaft taken over the bearing centres and that of the crankcase over the same centres. If this difference should exceed, say, 0·003 to 0·004in., then an arrangement with a pair of bearings located axially at one end of the shaft with a similar pair floating at the other should be adopted. Alternatively, a parallel roller bearing of suitable capacity could be used instead of the floating pair of bearings. Where the conditions are not fully known and the amount of difference in relative expansion cannot be estimated, the better course is to adopt the latter type of mounting, *i.e.*, a pair of bearings located at the one end with a floating pair at the other end. This arrangement is suitable for the crankshafts of Diesel and other engines and pumps, as well as for compressors.—*"The Motor Ship", Vol. XXI, No. 245, June, 1940, p. 90.*

Fire-fighting Equipment for Electrical Installations.

An article bearing the above title which appeared in a recent issue of the *Journal of the Institution of Electrical Engineers* includes a section dealing with fires liable to occur in cable runs or ducts exposed to the atmosphere, such as are generally found on board ship. A number of experiments were carried out in which water, foam, methyl bromide and CO_2 were used for extinguishing fires produced under conditions simulating those met with in practice, and various methods of applying these materials were tried. It was found that if there is no danger of risk through ventilation, inert gas protection is effective under conditions of rapid operation, although delay may result in a sufficient evolution of heat to cause smouldering with the attendant risk of re-ignition. Where through ventilation cannot be automatically checked, water or foam should be used.—*"Shipbuilding and Shipping Record", Vol. LV, No. 23, 6th June, 1940, p. 571.*

Special Valves for Tanker.

An article in a recent issue of the *Glenfield Gazette* contains details of the A-Y-R patent tanker valve designed to overcome certain disadvantages peculiar to oil service. The principal feature of the valve is the method of guiding the wedge to ensure that the faces shall always come to the original fitting. The valve spindle is extended in a plain portion through the wedge to a footstep bearing in the bottom cleaning door. Thus, the wedge is constrained to travel accurately on to its seat so that the valve always closes tightly, even to motor spirit. Other features of the A-Y-R valve include a large sump and cleaning door at the bottom of the valve, an extra deep stuffing-box for the spindle, and a patent gastight gland which can be fitted when required.—*"Shipbuilding and Shipping Record". Vol. LV, No. 24, 13th June, 1940, p. 619.*

Self-aligning Valve Seats.

An improved design of valve with a self-aligning seat has been introduced by a well-known engineering firm in this country. The essential feature of this valve is a stainless-steel ball in place of the usual and conventional disc or mushroom type clack, used in conjunction with a stainless-steel seat ring which is renewable and reversible. The seat is held in position by a bronze retaining ring screwed down by means of a special key. The seat is free to expand and contract independently of the valve body and is therefore free to centralise itself and form a perfect closure with the ball. Results of tests carried out with the valve appear to indicate that its efficiency is of a particularly high order.—*"Shipbuilding and Shipping Record", Vol. LV, No. 24, 13th June, 1940, p. 599.*

Rotary Blowers of Stork Engines.

The employment of rotary blowers for Stork 2-stroke and supercharged 4-stroke engines enables the overall length of such engines to be shortened, as the rotary blower is built level with the crankshaft and driven from the middle of the latter. These blowers give a more regular supply of air than single- and two-cylinder piston-scavenging air pumps. The accompanying diagram (Fig. 16) shows the relative output of a single-cylinder air pump, of a double-cylinder air pump with cranks at 90°, of a rotary blower of the Roots type, and of a Stork rotary blower with three sections arranged at an angle of 120°. This last

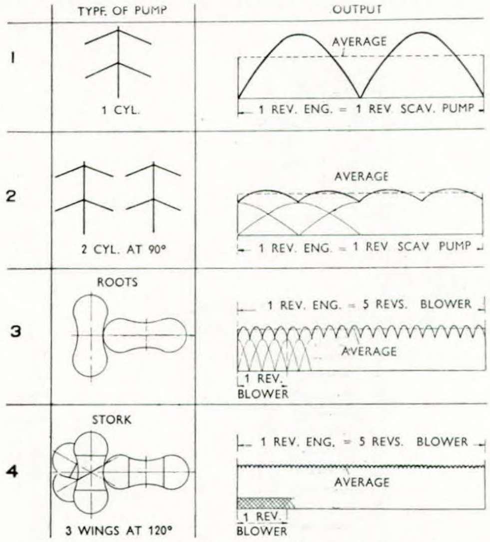


FIG. 16.—*Relative outputs of various scavenging pumps.*

arrangement ensures a much more regular torque on each of the shafts than the normal Roots blower in which the torque varies during each revolution between zero and a maximum, causing a good deal of noise where gearwheel drive is used (Fig. 17). The design of the Stork impeller with its three lobes

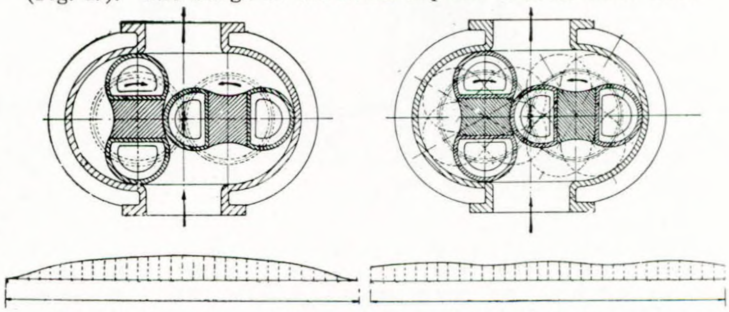


FIG. 17.—*Torques of Roots (left) and Stork (right) rotary blowers.*

is shown in Fig. 18. The rotor casing is of light metal, the circumference and ends being grooved to enable running-in to take place with little clearance and no risk of fouling, and the rotors being rigidly secured to the square shafting by means of

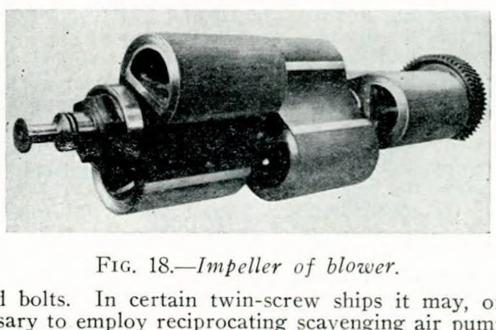


FIG. 18.—*Impeller of blower.*

reamered bolts. In certain twin-screw ships it may, of course, be necessary to employ reciprocating scavenging air pumps at the front of each engine owing to lack of space for rotary blowers, but although such an arrangement enables entirely satisfactory results to be achieved, they are not comparable with those attained where rotary blowers are used.—*G. Wieberdink, "The Motor Ship", Vol. XXI, No. 245, June, 1940, pp. 76-80.*

Improved Fuel-injection Valves in Stork Engines.

Until quite recently the larger sizes of Stork engines were equipped with Hesselman compression-disc or membrane valves, which possessed the advantage of closing very rapidly. The construction of these valves is shown in Fig. 10. It was found, however, that when used in the top cylinder covers of double - acting engines they were liable to give trouble owing to slight distortion of the membrane column caused by

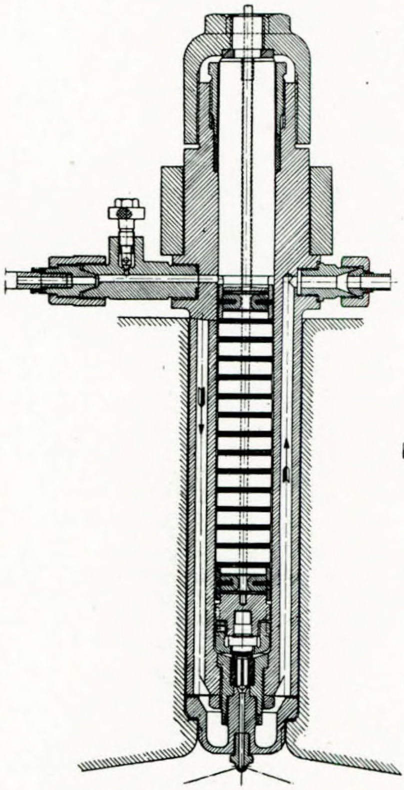


FIG. 10.—*Original Hesselman membrane fuel valve and—*

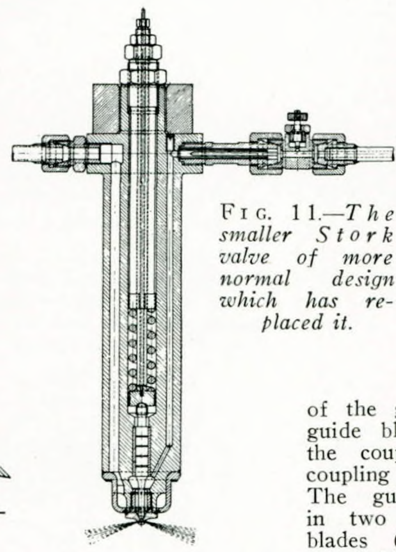


FIG. 11.—*The smaller Stork valve of more normal design which has replaced it.*

the high temperature prevailing there. Under such circumstances tightening down was ineffective and dribbling set in. No difficulties were ever experienced with the membrane valves which were fitted to the bottom covers of double-acting two-stroke engine cylinders, this being explained by the fact that the valves were fitted to the sides of the combustion chamber (because of the piston rod), where they were not subjected to such high temperatures.. Apart from the distortion of the membrane column of the valves used in the top covers, there was another drawback in the shape of possible fuel leakage into the cooling-water space of the valve housing. Fig. 11 shows the new Stork needle-type fuel valve in which the valve seating is situated very near the atomizer nozzle. The membrane valve had the disadvantage that the distance between the seat of the valve and the atomizer was relatively great and that also tended to promote dribbling. The new needle valve has the further advantage that there is no possibility of fuel leakage into the cooling-water space of the valve housing. This construction has been patented. It is stated that the needle valve possesses the additional advantage of intensive cooling of the atomizer, which, in turn, makes the engine less sensitive to the quality of the fuel.—*G. Wieberdink, "Gas and Oil Power", Vol. XXXV, No. 417, June, 1940, pp. 130-131.*

Series-built Auxiliary Sailing Ships.

Series-built "motor-sailers" with steel hulls have been constructed in Sweden during the last five years, a typical vessel of this kind being launched recently at the Oskarshamm shipyard on the east coast of Sweden. This ship has a length of 111ft. and a d.w. capacity of 350 tons. There are two cargo holds served by two electric winches. The propelling machinery consists of a semi-Diesel engine developing 225 b.h.p. and the vessel also carries about 3,760ft.^{2} of sail on her three masts. The ship has an elegant yacht-like shape with a clipper bow.—*"The Journal of Commerce" (Shipbuilding and Engineering Edition), No. 35,069, 27th June, 1940, p. 7.*

Maintaining Full Engine Output at Reduced Speed.

A German invention which is claimed to effect the above purpose is covered by a recently published British patent. With the arrangement illustrated in Fig. 3, a triple-screw vessel has a proportion of the machinery, say the centre shaft, running at its full output, although the ship's speed is reduced and the revolutions have to be correspondingly lower. The specification mentions the case of three 8,000-b.h.p. engines running at 250 r.p.m. At reduced speed it would not be sufficient to keep only one unit in operation, as with a reduction of power from 24,000 to 8,000 b.h.p. the r.p.m. fall from 250 to something like 180. All the shafts have positive dog-clutch couplings, in addition to which the centre shaft is also equipped with a speed-reducing coupling of the fluid transformer type running empty when the propeller revolutions are at the maximum rate. Referring to the diagrams, the three propellers (*1, 2, 3*) are driven by the engines (*4, 5, 6*) or sets of geared engines, if necessary. The centre shaft (*7*) has a coupling (*8*) for reducing the propeller revolutions as required, in addition to a dog-clutch coupling (*9*) which normally connects the engine shaft (*10*) to the propeller shaft (*7*), when the coupling (*8*) is inoperative. The construction of these components is shown on an enlarged scale in the left-hand bottom diagram. The fluid transformer comprises primary and secondary members (*11, 12*) respectively, the former being connected to the engine shaft by hollow shafting (*13*). The full output of the engine is transmitted to the propeller shaft (*7*) by a suitable choice of the guide and running members of the coupling. The guide blades (*14*) have cast-in through-passages for filling the coupling. Two of the diagrams show a form of coupling for use with both directions of rotation. The guide blades (*b*) of the blade rows (*a.b.c*) are in two parts, one (*b*^{1}) contiguous with the secondary blades (*c*), being set so that the guide blades direct the fluid against either side of the secondary blades. This setting is effected from outside the coupling by linking all the parts (*b*^{1}) of the guide blade row to a ring enclosing the circuit. The specification describes further embodiment of the invention, which is not confined to a centre-shaft drive and may be applied to separate units of multi-engined geared sets.—*"The Motor Ship", Vol. XXI, No. 245, June, 1940, p. 103.*

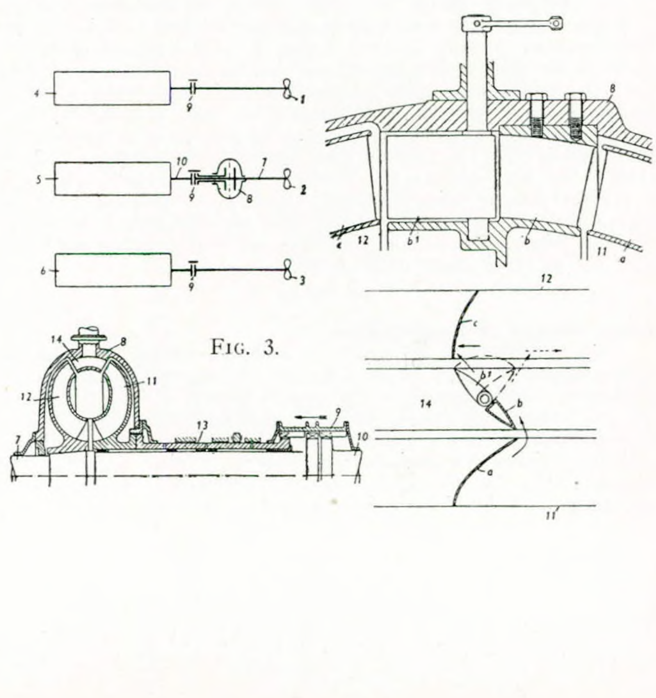
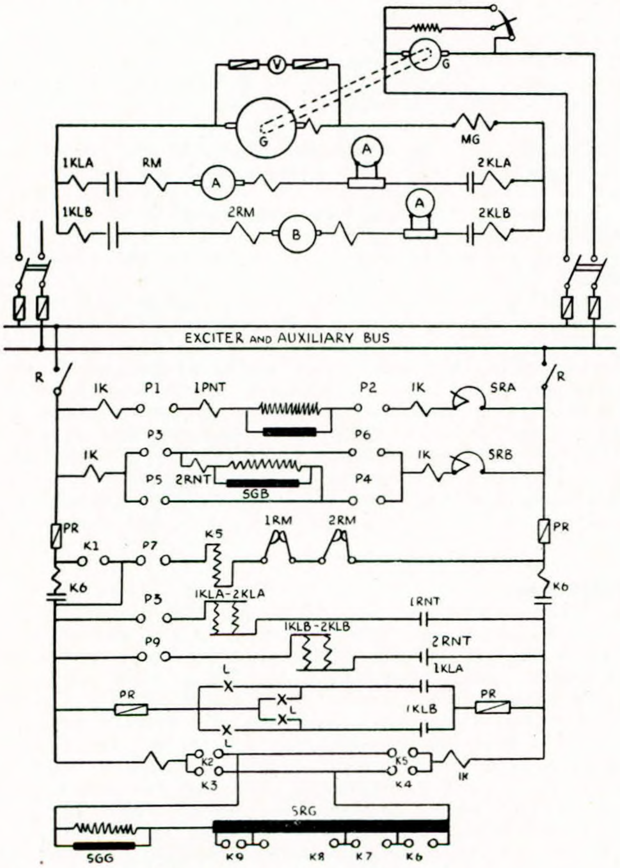


FIG. 3.

First Russian Stern Wheeler with Diesel-electric Drive.

A special type of passenger and cargo vessel with Diesel-electric drive was recently developed for service on the inland waterways of the U.S.S.R., and the first ship of this kind has now been completed at the Gorky Shipyard. The length of this vessel is 133ft. 6in., the beam 26ft., and the displacement 165·3 tons on a draught of from 12 to 20in., the designed speed being 14km./hr. (=7·6 knots). In order to achieve maximum manœuvrability on the above shallow draught, double stern-wheel propul-



Ward Leonard control circuit.

G—Main generator.
G^1—Auxiliary generator.
A—Port propulsion motor.
B—Starboard propulsion motor.
MG—Compound winding of generator.
K—Control contact.
P—Control contact.
KLA—Port line switch.
KLB—Starboard line switch.
KB—Control circuit switch.
RNT—Minimum relay.
RM—Maximum relay.
SRA—Shunt regulator, port motor.
SRB—Shunt regulator, starboard motor.
SRG—Shunt regulator of generator.
SGA—Protective resistance for port motor.
SGB—Protective resistance for starboard motor.
SGG—Protective resistance for generator.
L—Signal lamp.
PR—Protective switch.
R—Hand switch of control circuit.
IK—Arc-suppressing coil.

sion with Diesel-electric drive and separately controlled wheel drive has been adopted, the normal speed of the paddle wheels being 50 r.p.m. The power unit consists of a 6-cylinder single-acting 2-stroke Burmeister and Wain engine rated at 250 b.h.p. at 1,000 r.p.m. and weighing only 1,000kg., *i.e.*, 8·8lb./b.h.p. This engine is directly coupled to an over-compounded d.c. generator of a nominal capacity of 150 kW. and to an auxiliary generator of 12 kW. at 230 volts, which supplies all the ship's auxiliaries besides furnishing the excitation current for the generator and wheel motors. There is also an auxiliary set consisting of a single-cylinder Diesel engine driving a 7·5-kW. 230-volt d.c. generator at 650 r.p.m. The whole of this machinery is located in an engine room in the fore part of the vessel, the two propulsion motors being arranged aft alongside the stern wheels. Each of these motors has a capacity of 66·5 kW. at 1,070 r.p.m. and is coupled to its respective wheel shaft through double-reduction gearing having a ratio of 21·4 to 1. The arrangement of the electrical circuits of the propelling machinery is shown in the accompanying diagram, from which it will be seen that the two stern-wheel motor drives are operated in parallel through two Ward-Leonard circuits. A five-step drum-type control is provided in the exciter circuit of the main generator and gives a range of speed control of the wheel motors from 430 r.p.m. (corresponding to a power consumption of 5·5 kW. per motor) to the normal speed of 1,070 r.p.m., corresponding to the 66·5 kW. rating of the wheel motors. The total weight of the propulsion plant is about 9 tons—although the Diesel engine itself weighs less than one ton. A geared Diesel-drive installation with a 1,000-r.p.m. engine driving the wheels through reduction gearing at 50 r.p.m. would have weighed only 5·3 tons, but it is claimed that the Diesel-electric drive gives incomparably greater flexibility and enables the ship to be reversed from full speed ahead to full speed astern in 30 seconds. When turning at "third speed" by working one wheel ahead and the other astern, and without using the two rudders, 2 min. 5 sec. to 2 min. 30 sec. are found to be required for a 180° turn, while a 360° turn can be made in just over 5 min. The turning circle of the ship under these conditions is stated to be between 1 and 1·5 ship's lengths. A 180° turn at fifth speed with 45° of helm, working one wheel ahead and the other astern, can be effected in 2 min. with a turning circle of 1·5 ship's lengths.—*Prof. V. L. Lichkovsky," Vodny Transport", M.8, 1939, reproduced in "The Marine Engineer", Vol. 64, No. 755, June, 1940, pp. 140-142.*

U.S. Maritime Commission's Training School for Ships' Firemen.

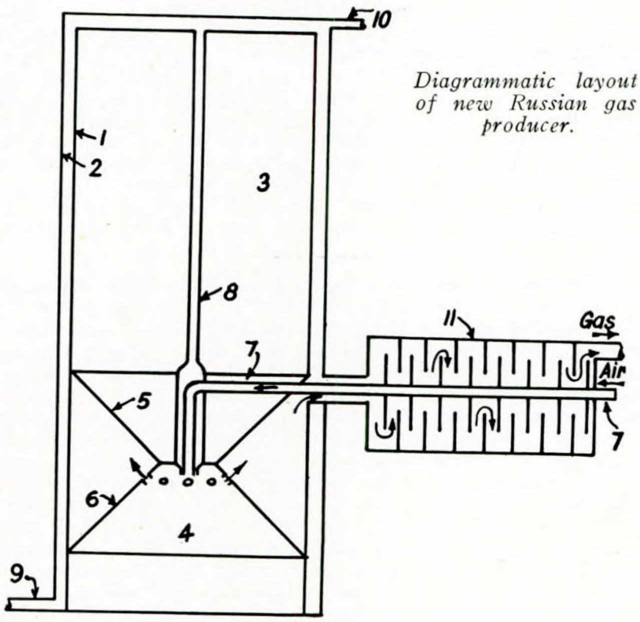
In order to improve the efficiency of the firemen of the U.S. Merchant Service, the Maritime Commission have established a training school at Hoffman Island, N.Y., in which men who have already had from two to twenty years' sea service undergo a technical course of three months' duration which is intended to make them into first-class firemen. Landsmen are not eligible for entry. The first five weeks of the course are mainly devoted to instruction in arithmetic, the sixth week being spent in undergoing examinations. If the student fails to pass he can go on with the course and is given a weekly opportunity of passing in the arithmetic examination, but if he does not succeed in the long run, he cannot qualify for a first-class fireman's rating when he leaves the school. In the engineering class students receive instruction in the combustion of fuels, special attention being paid to the safe and economical handling of oil. After dealing with the various systems of burning oil fuel, the students undergo instruction in marine boilers, more especially in the types installed in the ships of the U.S.M.C. Lantern slides are extensively used in this section. The students are then taught something about boiler-room and engine-room instruments in common use and undergo instruction in various methods of fire-fighting and carrying out minor repairs. The last part of the course is devoted to elementary instruction in turbines, feed-water analysis, thermodynamics, Diesel engines and various other subjects.—*"Shipbuilding and Shipping Record", Vol. LV, No. 24, 13th June, 1940, p. 598.*

New Grace Liner.

The Grace Line, N.Y., recently took delivery of three fast C.2-type cargo steamers for service between Pacific Coast ports and the West Coast of S. America. The first of the three to arrive at San Francisco was the 7,169-ton "Stag Hound", an oil-burning steamship with a speed of 15½ knots propelled by turbine machinery comprising H.P. and L.P. turbines of the impulse-reaction type rated at 6,000 s.h.p. and driving a single propeller shaft through D.R. gearing. The two Babcock and Wilcox single-pass sectional-header boilers are located aft of and slightly above the turbines, in the same compartment. The control position for the engineer is on the second grating level, and standing there he has a full view aft over the turbines and through the fore-and-aft firing space between the two boilers directly at the gauge board of the control system.—*"The Syren", Vol. CLXXV, No. 2,285, 12th June, 1940, pp. 389-390.*

A Russian Heat-recuperating Gas Producer.

A new type of gas producer, designed to permit the recuperation of a large part of the waste heat involved in the producing process, is described by I. Menzine in U.S.S.R. patent 52291. The producer is simple in design, as the sketch shows. It comprises a single large sheet-metal cylinder (1), surrounded by a water jacket (2). This cylinder provides space for both the fuel hopper (3) which occupies the upper part of the cylinder and the producer proper (4) which is contained in the lower part. Fuel is admitted through a hermetically-closing door in the top of the hopper, and ash is removed through a hermetically-sealed door in the bottom of the producer; these present no features calling for comment. Division between hopper and producer is formed by two truncated sheet-metal cones (5, 6), one of which (5) forms the floor of the fuel hopper while the other (6) provides the top to the gas producer. In this latter, a series of small holes is provided around the whole circumference, near the top, to serve as a gas outlet. Air is admitted to the producer through a pipe (7) which opens into the producer in the centre of the opening through which the fuel passes, providing a radial cross-draught running from the air pipe to the centre to the holes all round the circumference of the producer roof (6). Water is used as an auxiliary fuel and is provided from the water jacket through a pipe (8), which runs down from the water jacket at the top of the hopper. This ends in a very thin annular slit around the end of the air inlet pipe (7) in such a way that the incoming stream of air will provide a sort of annular atomiser to spray the water very finely into the combustion zone. Although this spraying of water on to fire may seem curious, it is really less so than it seems, on account of the heat recuperation which makes it possible. This works as follows: In the first place, water for the jacket is supplied by an inlet (9) which is supplied with hot water from the engine. This water is further heated by the waste heat escaping from the producer itself, so that by the time it reaches the top and is allowed to descend into the producer, it is near boiling point. Surplus water is carried off by an overflow pipe (10) at the top of the producer, and flows to the radiator for cooling and return to the engine. Secondary recuperation of waste heat is provided on the air inlet pipe. This runs through a heat exchanger (11) just outside the producer. In this exchanger the air pipe is provided with radial ribbing, while the exchanger itself is composed of a simple box with baffle plates projecting from the sides inward. Gas from the producer is admitted directly to this exchanger box through an outlet pipe from the producer (as shown by the arrows), concentric with the air inlet pipe (7), and is obliged by the baffle plates to pass over the ribbing on the air inlet, thus giving up the greater part of its heat to the incoming air. By the time the gas leaves the heat exchanger, it has been sufficiently cooled to permit immediate filtering without the need for further cooling, while the air on reaching the producer is sufficiently hot to be well above the boiling point of water, so that the water vapour sprayed on to the burning fuel in the producer is instantly vaporized and turned into superheated steam even before it reaches the combustion zone. Temperature of combustion is sufficient to cause this steam to be broken down into oxygen, which combines with some of the carbon of the fuel to provide carbon monoxide, while the remaining free hydrogen passes on with the carbon monoxide through the gas outlet and so to the engine. So far from being an impediment, this addition of hydrogen to the producer gas actually improves it, since the heat of combustion of hydrogen in the engine is greater than that of the carbon monoxide which forms the basis of producer gas.—*"Gas and Oil Power", Vol. XXXV, No. 417, June, 1940, p. 129.*



Diagrammatic layout of new Russian gas producer.

Sandblasting.

There are two kinds of sandblasting. One uses high-pressure air (from 5lb./in.^{2} to 120lb./in.^{2} or more) and accomplishes its work by the sharp impact of the abrasive thus propelled, while low-pressure sandblasting on the other hand, works not so much by the great cutting force of the abrasive as by the repeated impact of the sand under low-pressure air. Each process has its own field. High-pressure sandblasting is chiefly used for removing scale from drop forgings, cleaning casehardened parts such as transmission gears, producing a clean, uniform metallic surface on carburettor bodies, and removing core sand from the interior of cylinder-block castings, gearcases and similar components. Low-pressure sandblasting is mainly employed on sheet metal products and die castings. The process accentuates the natural colour of these and if they are to be electroplated or otherwise treated the blast produces a matt finish or surface to which the plate or enamel will readily adhere. This process is also largely used for frosting radio and electric light bulbs and for stencilling letters on glass, metal and bakelite. The sand is not closely graded, the grades varying with different producers. Sharp grains cut faster, but rounded grains produce smoother surfaces. Flint shot or a metallic abrasive, such as crushed steel or steel shot, is often used, but should be kept dry as it is apt to rust and form pellets or balls if exposed to moisture. If a hose is used, care must be taken to avoid bending or kinking to prevent wear where bent and wastage of both the air and power necessary to remove the accumulation of sand that collects in the hose. Sandblasting is capable of reducing grinding costs by 25 to 50 per cent. and produces a more attractive finish. The sand blast room should be simple in construction, and illuminated by electric lights at the sides, protected from the flying sand by wire screens. Special dust-proof hoods with inserted lenses must be worn by the operators.—*A. E. Peters, "The Machinist", Vol. 84, No. 17, 15th June, 1940, pp. 201-202 E.*