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Marine Steam Condenser Design and Practice.

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The condensers adopted in all modern marine steam turbine installations are of the regenerative type. With the older type of condenser the temperature drop across the condenser was of the order of 10° F. to 15° F., equivalent to an increase in fuel consumption of approximately 1 to 1·5 per cent.

In the regenerative condenser, as is well known, a portion of the exhaust steam passes directly to the bottom of the condenser, thereby raising the temperature of the condensate to approximately the steam temperature corresponding to the vacuum. In practice the difference between the exhaust steam temperature entering the condenser and that of the condensate is usually between 1° F. and 2° F.

With this type of condenser working at the high vacua demanded by modern marine steam turbines, it is essential to embody an air cooler as an integral part of the condenser, since a small difference in pressure at high vacua means an appreciable difference in temperature owing to the temperature gradient being very steep at low pressures. A reference to the steam tables will show that for a condenser working at 25·6in. vacuum with 0·2in. pressure drop across the condenser, the temperature difference is $(128{\cdot}8-127{\cdot}1)=1{\cdot}7°$ F., whereas with a condenser working at 29in. vacuum with 0·2in. pressure drop, the temperature difference is $(78{\cdot}9-72{\cdot}2)=6{\cdot}7°$ F., or nearly four times the temperature difference for the same pressure drop, the barometer being 30in. in each case.

The pressure drop across the condenser is an important factor in the maintenance of high vacuum and must be reduced to a minimum. Sim, in his book "Steam Condensing Plant", has developed an expression for the estimation of the pressure drop on the rational basis that the pressure drop is dependent upon the steam velocity over the top row of tubes, the steam head corresponding to the pressure drop being inversely proportional to the density of the steam entering the condenser. From his investigations on the pressure drop in condensers, he found that a large proportion of the pressure drop takes place over the first few rows of tubes at the top of the condenser. From fundamental principles it can be shown that, if v be the steam velocity, d the pressure drop in inches of mercury, and w the density in pounds per cubic foot, the velocity corresponding to the pressure drop is $v=67{\cdot}2\sqrt{\frac{d}{w}}$. Sim's experiments showed that the average value of v, under practical conditions could be taken as,

$$v=21{\cdot}1\sqrt{\frac{d}{w}} \qquad \ldots \qquad (1)$$

The pressure drop is a determining factor affecting the mean temperature difference between the steam and the circulating water. The greater the pressure drop the lower is the rate of heat transmission per square foot of cooling surface per hour, and consequently the higher is the condensing surface required for given vacuum, velocity of flow and circulating water temperature conditions.

The determination of the mean temperature difference is due to the conventional equation of Grashof which may be stated as follows:—

$$\theta_m=\frac{\theta_1-\theta_2}{\log_e\left(\frac{\theta_1}{\theta_2}\right)} \qquad \ldots \qquad (2)$$

where,

θ_m = mean temperature difference between the steam and circulating water, °F.

θ_1 = difference in temperature between the condensate and inlet circulating water, °F.

θ_2 = difference in temperature between the exhaust steam entering the condenser and the outlet circulating water, °F.

A detailed investigation of the validity of this equation was made by Dr. H. L. Guy and Mr. E. V. Winstanley in their paper on the "Design of Surface Condensing Plant" read before the Institution of Mechanical Engineers in 1934. The conclusion formed from this paper was that Grashof's equation could be used without sensible error for two- and three-pass condensers.

The effect of the pressure drop on the mean temperature difference and the percentage increase in condensing surface required in excess of zero pressure drop for the maintenance of 28, 28½ and 29 inches of vacuum (30in. bar.) with inlet sea-water temperatures of 82° F., 73° F. and 60° F. respectively and outlet temperatures 10° F. above the inlet, are stated below:—

28 inches.

Pressure drop, ins. Hg.	0	0·1	0·2	0·3	0·4	0·5	0·6
Mean temp. difference, °F.	13·39	12·70	11·95	11·16	10·30	9·40	8·15
Increase in condensing surface, % ...	0	5·43	12·05	19·97	30·0	42·44	64·30
Temp. drop across condenser, °F. ...	0	1·7°	3·5°	5·4°	7·3°	9·4°	11·6°

28½ inches.

Pressure drop, ins. Hg.	0	0·1	0·2	0·3	0·4	0·5	0·6
Mean temp. difference, °F.	12·95	12·07	11·07	10·0	9·2	7·16	5·09
Increase in condensing surface, % ...	0	7·29	16·97	29·50	40·76	80·86	154·4
Temp. drop across condenser, °F. ...	0	2·2°	4·6°	7·1°	9·8°	12·7°	15·9°

29 inches.

Pressure drop, ins. Hg.	0	0·1	0·2	0·3	0·4
Mean temp. difference, °F.	13·30	11·99	10·5	8·5	5·98
Increase in condensing surface, % ...	0	10·91	26·65	56·46	122·4
Temp. drop across condenser, °F. ...	0	3·2°	6·7°	10·7°	15·1°

Resistance to Heat Transmission.

The resistance to heat transmission of commercially clean tubes in a surface condenser is made up of the two following factors:—

(1) The resistance to heat transmission from tube to water which includes the resistance at the surface of separation of the tube wall and the water ... $=\frac{I}{K_W}$

(2) The resistance to heat transmission from steam to the water side of the tube metal which includes the resistance due to the water film on the steam side of the tube, the resistance due to the air blanket surrounding the tube and the resistance of the metal wall of the tube ... $=\frac{I}{K_S}$

The total resistance to heat transmission will therefore be

$$\frac{I}{K_T}=\frac{I}{K_W}+\frac{I}{K_S}$$

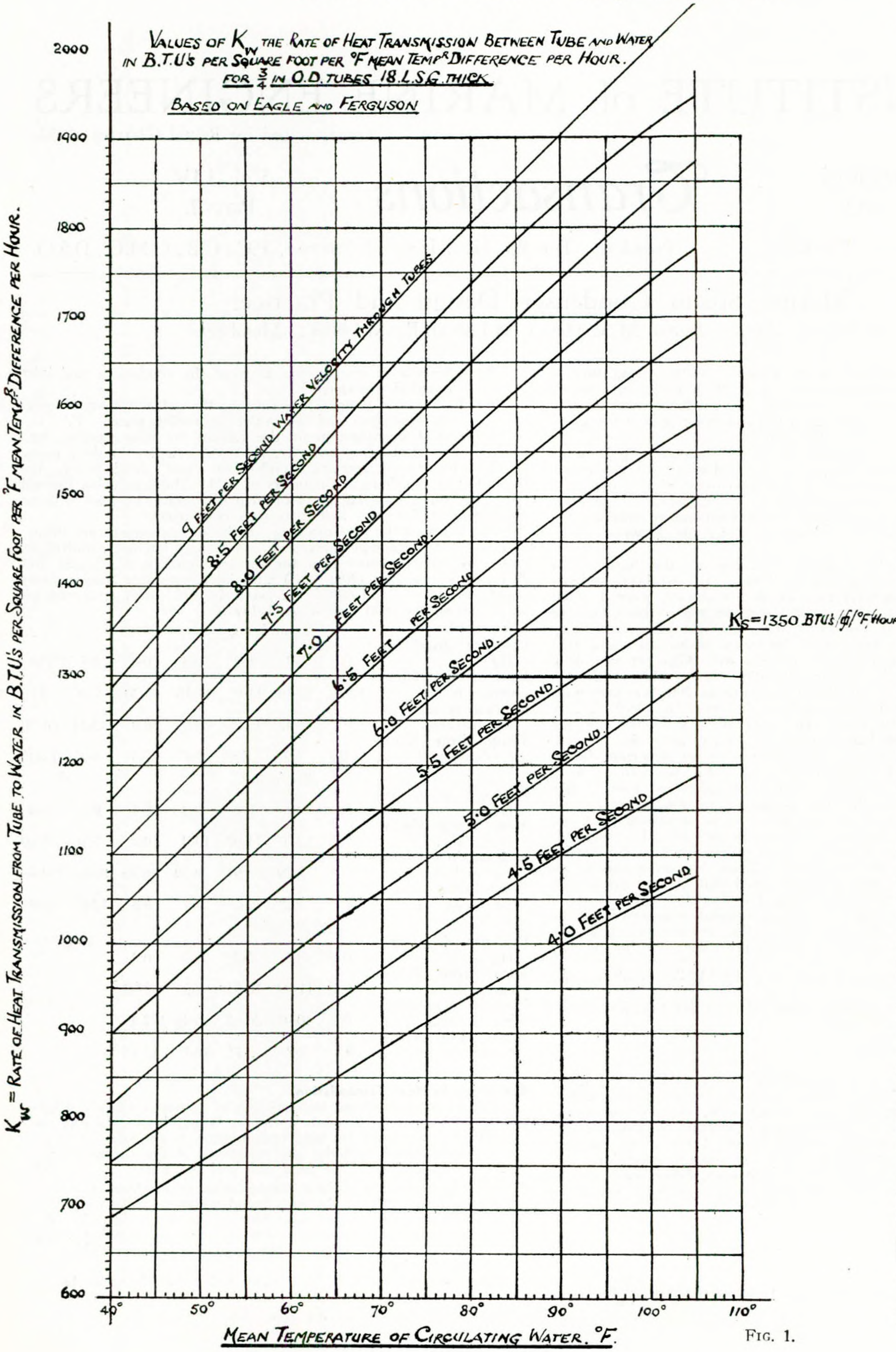


FIG. 1.

Where,

K_W = rate of heat transmission from tube to water in B.Th.U.'s per square foot per °F. mean temperature difference per hour.

K_S = rate of heat transmission from steam to water side of tube metal in B.Th.U.'s per square foot per °F. mean temperature difference per hour.

K_T = overall rate of heat transmission from steam to water in B.Th.U.'s per square foot per °F. mean temperature difference per hour.

The values of K_W are due to the extensive experiments carried out by Mr. A. Eagle and Mr. R. M. Ferguson which were published in their paper before the Institution of Mechanical Engineers in 1930 and later given by Dr. H. L. Guy and Mr. E. V. Winstanley in their paper before the same Institution in 1934 in terms of the outside diameter of the tube.

The author has redrawn the curves of K_W on a base of mean circulating water temperature at water velocities through the tubes from 3ft. to 9ft. per second, as the mean circulating water temperature will seldom be an even figure as shown in the curves mentioned above, whereas the designer will generally adopt some definite velocity of flow for determining the surface and particulars of the condenser. The curves of K_W are shown in Fig. 1.

With regard to K_S, the rate of heat transmission from steam to tube in B.Th.U.'s per square foot per °F. mean temperature difference per hour, it has been shown by Guy and Winstanley from the experiments of

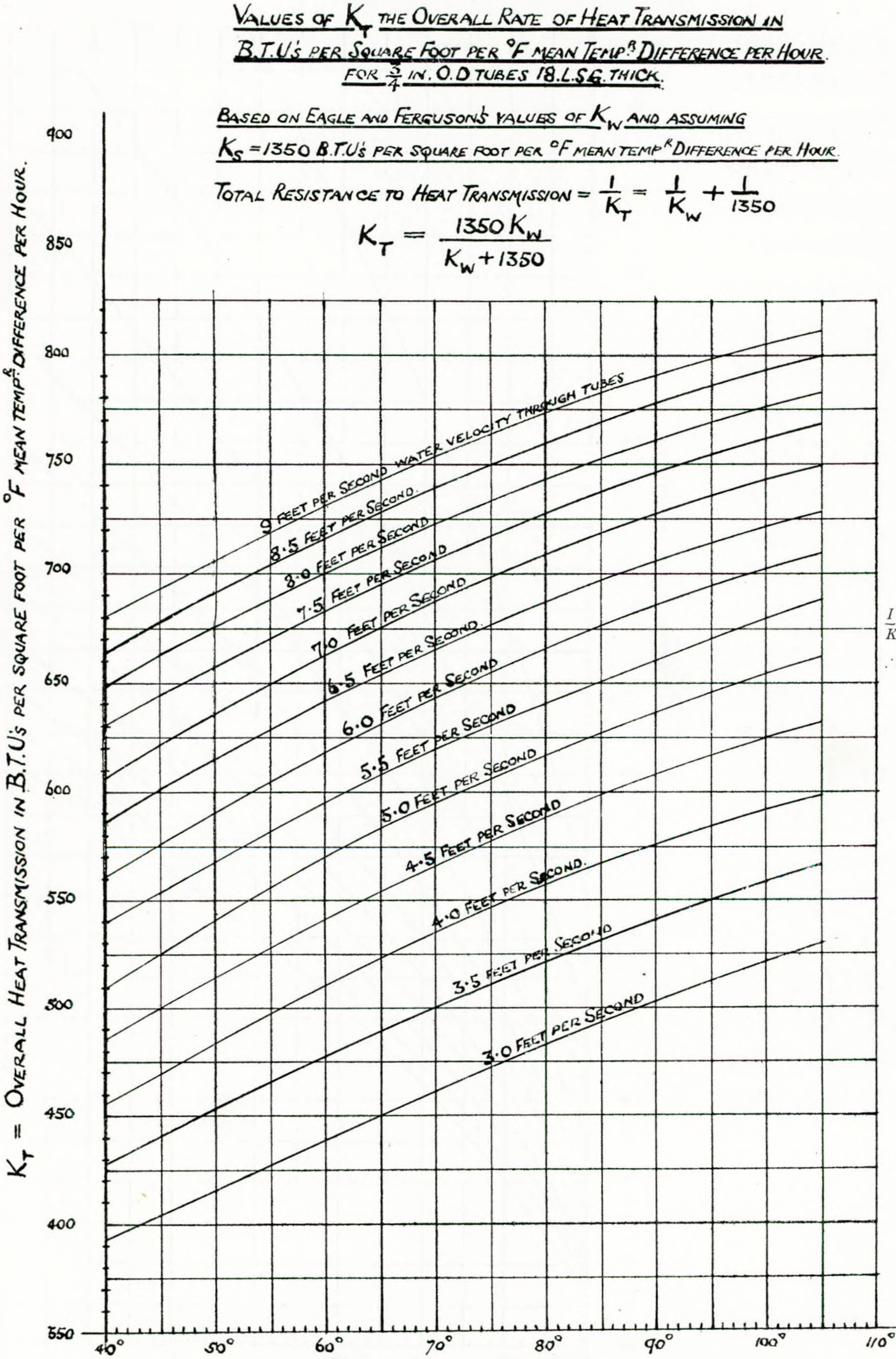


FIG. 2.

Lee and Jakob that the variation over the practical range of circulating water temperatures pertaining in condensers is small, being about 1,200 B.Th.U.'s at 45° F. and 1,400 B.Th.U.'s at 100° F. mean circulating water temperature. Their own experiments showed wide variations in K_S, depending principally upon the cleanliness and loading of the condenser. It would appear, however, from the experimental data at present available that the value adopted by these authors, *viz.* 1,350 B.Th.U.'s may be used with confidence as a basis for design, as will be shown in the marine condensers, given later in the present paper, of which the writer has experience.

Using this value for K_S we have,

$$\frac{1}{K_T} = \frac{1}{K_W} + \frac{1}{K_S} = \frac{1}{K_W} + \frac{1}{1,350}$$

$$\therefore\ K_T = \frac{1,350 K_W}{K_W + 1,350} \quad \dots \quad (4)$$

The values of K_T are given in Fig. 2, and Table I has been prepared from the curves for ready reference. The margin to be allowed on these overall rates of heat transmission, based on a number of actual marine condensers, is given later in this paper.

Rate of Heat Transmission.

The rate of heat transmission per square foot of cooling surface per hour will depend directly upon the mean temperature difference between the steam and circulating water and the value of K_T, the overall rate of heat transmission in B.Th.U.'s per square foot per °F. mean temperature difference per hour. Hence, the rate of heat transmission per square foot per hour $= \theta_m . K_T \quad \dots \quad (5)$

The rate of heat transmission per square foot will also equal,

$$\frac{W.q.L.}{S} \quad \dots \quad \dots \quad (6)$$

TABLE I.

Values of K_T=Overall Rate of Heat Transmission in B.Th.U.'s per Square Foot per °F. Mean Temperature Difference per Hour.

$$= \frac{1350\ K_W}{K_W + 1{,}350}$$

$\frac{3}{4}$in. Outside Diameter Tubes, 18 L.S.G. Thick.

Mean temperature of circulating water, °F.	45°	50°	52°	54°	56°	58°	60°	62°	64°	66°	68°	70°	72°	74°	76°	78°	80°	82°	84°	86°	88°	90°	92°	94°	96°	98°	100°
3·0ft. per second water velocity	404	415	420	425	430	435	439	443	447	452	456	461	465	470	475	479	483	487	491	495	499	503	507	510	513	517	521
3·5 " " "	440	452	458	463	468	473	477	482	486	491	496	500	505	509	513	517	521	525	529	533	537	541	545	548	551	554	558
4·0 " " "	469	483	490	495	500	505	510	516	521	525	530	535	540	543	547	552	556	560	564	567	571	575	578	582	585	588	592
4·5 " " "	500	513	520	526	531	537	542	547	552	557	562	567	571	575	580	584	587	592	596	600	604	607	611	615	618	622	625
5·0 " " "	525	540	547	553	559	565	570	575	580	585	590	595	598	602	607	611	615	620	624	628	632	635	639	643	647	651	655
5·5 " " "	553	567	572	578	583	590	595	600	605	610	615	620	624	628	632	636	640	644	648	652	656	660	664	668	672	676	680
6·0 " " "	576	590	596	602	607	612	618	624	629	634	639	644	648	652	657	661	665	670	674	677	681	685	689	692	695	698	701
6·5 " " "	600	614	620	626	631	636	642	647	651	656	661	665	670	674	678	682	686	690	694	698	702	705	708	712	715	719	722
7·0 " " "	621	635	642	647	652	658	663	668	673	678	683	687	692	696	700	704	708	712	716	720	723	726	730	733	737	740	743
7·5 " " "	644	656	662	667	672	677	682	687	692	696	700	705	709	713	717	722	727	730	733	737	741	745	748	752	755	758	761
8·0 " " "	662	675	681	686	691	696	700	705	710	714	719	723	727	731	736	740	744	747	751	754	757	760	764	767	771	774	777
8·5 " " "	677	690	696	701	706	711	715	721	726	730	735	739	743	747	751	755	759	762	766	770	773	777	780	783	787	790	793
9·0 " " "	692	705	710	715	720	725	730	735	740	745	750	754	757	761	765	768	772	775	779	783	786	790	793	796	799	802	805

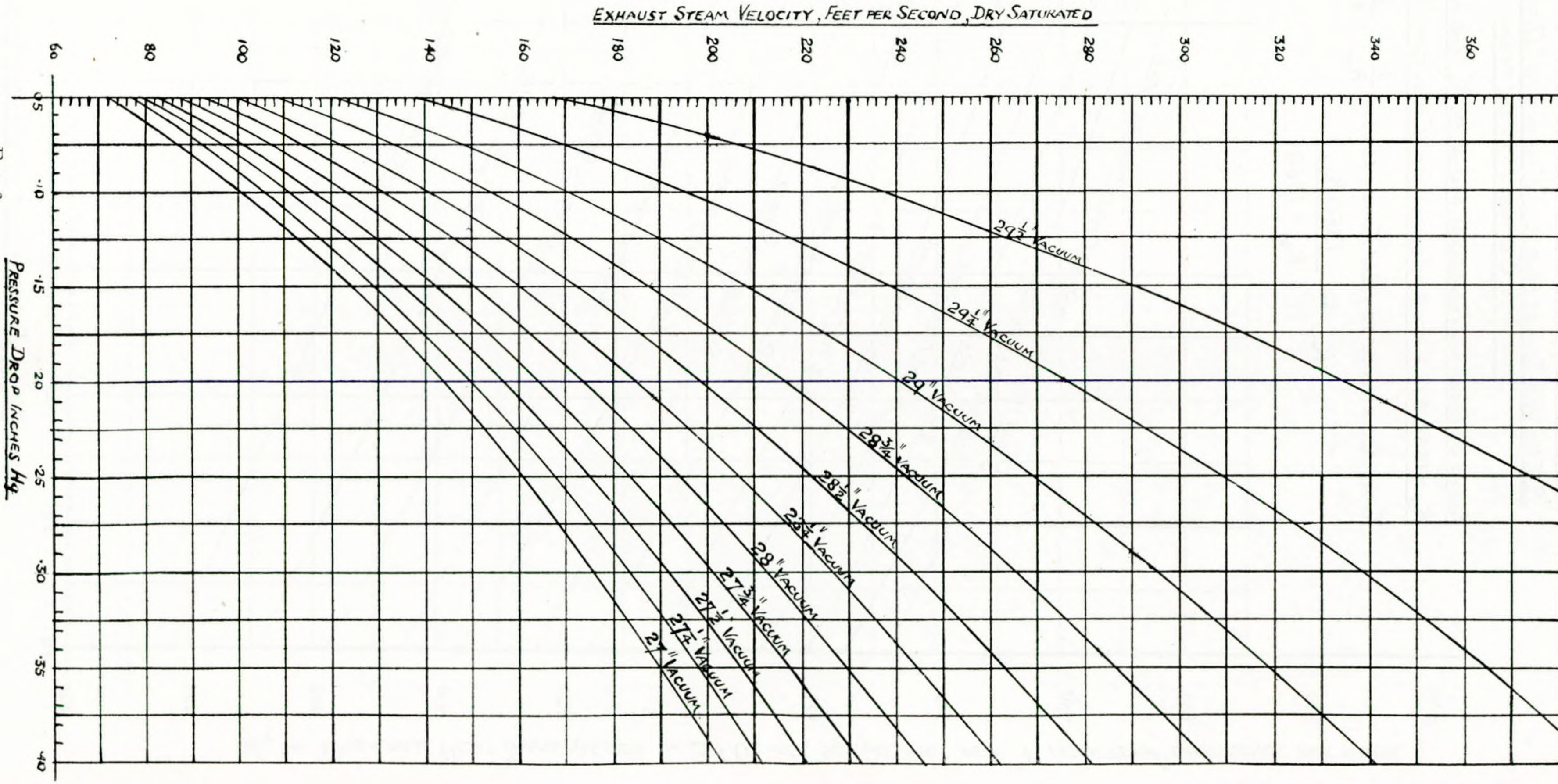


Fig. 3.

Where, W = weight of steam condensed in pounds per hour.
q = dryness fraction of exhaust steam entering the condenser.
L = latent heat of dry saturated steam at the given vacuum in B.Th.U's per pound.
S_c = condensing surface in square feet.

Combining equations (5) and (6) we have,

$$\text{Condensing surface, } S_c = \frac{W.q.L.}{\theta_m.K_T} \quad \ldots \quad (7)$$

$$= \frac{n_c.\pi.d.l}{12} \quad \ldots \quad (8)$$

Where, n_c = number of tubes in condensing portion of the condenser
d = outside diameter of the tubes in inches.
l = length of the tubes in feet.

Air Leakage.

The air leakage allowance by the British Electrical and Allied Manufacturers Association rule is, $m = \frac{W}{2,000} + 3$ ... pounds per hour ... ... (9)

With regard to the surface baffled off as air cooling surface through which the water passes once only, the author has selected the following marine regenerative two-pass condensers, from which it will be seen that the portion baffled off varies from about 10 to 14½ per cent. of the condensing surface. In each of the condensers given below the vacuum recorded by Kenotometer under actual service conditions was 28½ in. (30in. bar.) with sea-water inlet temperature 73° F. Some actual results of tests obtained from condensers "A" and "E" under service conditions are given later in the paper.

TABLE II.

Condenser.	*Total tubes in condenser.*	*Tubes in air cooling portion.*	*Tubes in condensing portion.*	*Surface in air cooling portion, sq. ft.*	*Surface in condensing portion, sq. ft.*	*Total surface of condenser, sq. ft.*	*Surface baffled off. Total surface. %*	*Surface baffled off. Condensing surface. %*
A ...	4,120	454	3,666	1,069	8,631	9,700	11·02	12·39
B ...	5,440	590	4,850	1,356	11,144	12,500	10·85	12·17
C ...	3,016	276	2,740	714	7,086	7,800	9·15	10·08
D ...	5,124	502	4,622	1,127	10,373	11,500	9·80	10·86
E ...	5,752	730	5,022	2,030	13,970	16,000	12·68	14·53
F ...	5,172	480	4,692	1,299	12,701	14,000	9·28	10·23
G ...	4,220	428	3,792	1,150	10,190	11,340	10·14	11·29

The author therefore suggests that the condensing portion of the condenser should be determined from equation (7) to which should be added an allowance of not less than 10 per cent. for the tubes baffled off as air cooling surface in order to obtain the total surface required with clean tubes, that is without any margin for fouling. The amount of the allowance for fouling from the investigation of the above condensers is given in Table VII, from which it will be seen that the margin adopted was between 15½ and 18 per cent.

With the allowances for the surface baffled off and a margin of 16 per cent. for fouling given in the previous paragraph, based on the above successful marine two-pass regenerative condensers, the total surface required would be as follows:—

$$S_R = \frac{S_c}{0{\cdot}84} + \frac{0{\cdot}10S_c}{0{\cdot}84} = 1{\cdot}309S_c \quad \ldots \quad (10)$$

Where,
S_R = recommended total surface of condenser in square feet.
S_c = condensing surface of condenser in square feet without margin $= \frac{WqL}{\theta_m K_T}$

Number of Tubes.

The number of tubes per pass will depend upon the surface per pass, the length between tube plates and the velocity of flow through the

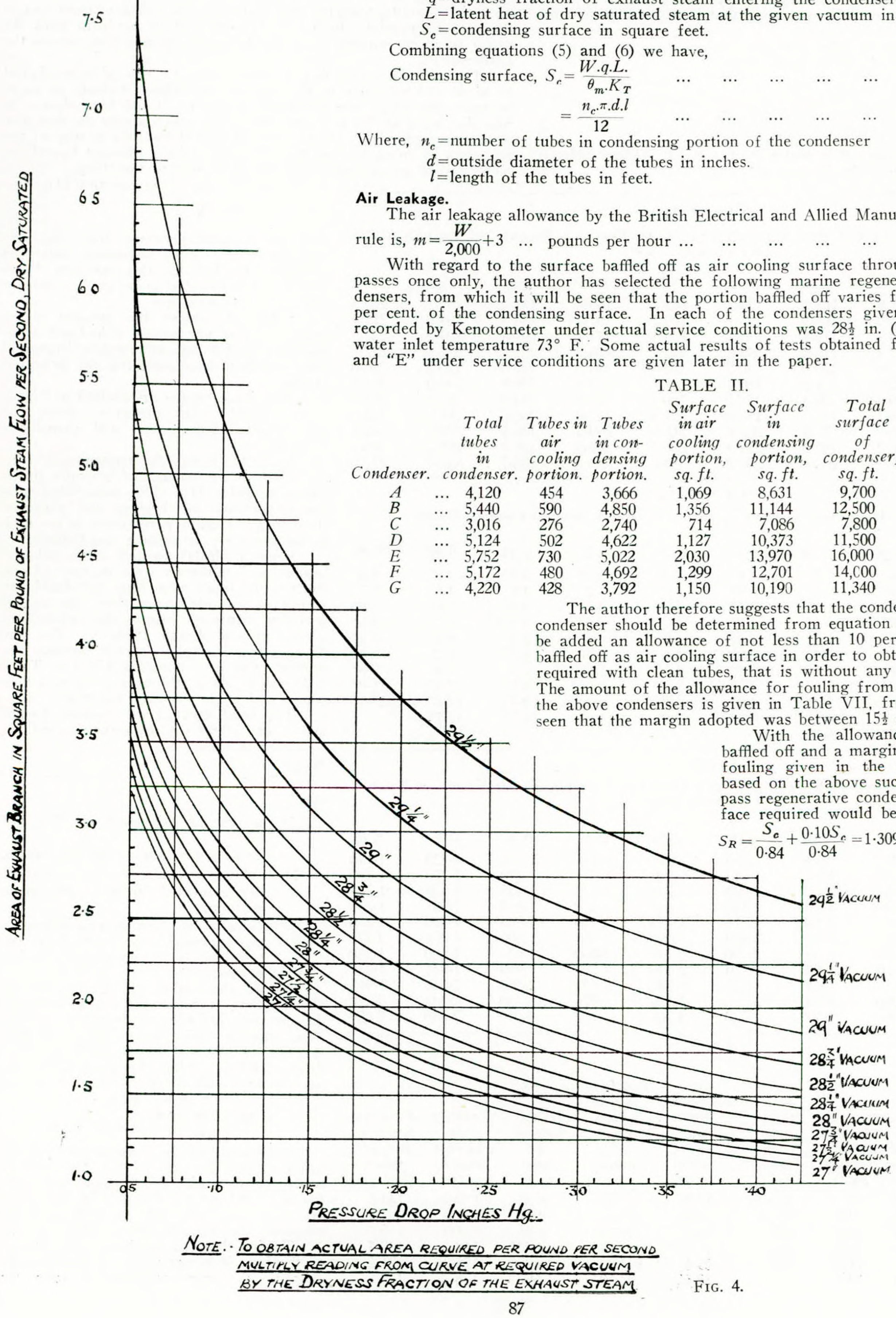


FIG. 4.

tubes. The total number of tubes in the condenser will be twice the number per pass. With ¾in. outside diameter tubes, 18 L.S.G. thick, the minimum number of tubes per pass, that is without margin for fouling, would be :—

Minimum number of tubes per pass=

$$\frac{6{\cdot}6\ S_c}{\pi\times 0{\cdot}75l}=\frac{2{\cdot}8\ S_c}{l}=\frac{1{\cdot}143\times \text{galls./min.}}{v_w} \quad \ldots \quad \ldots \quad \ldots \quad (11)$$

Recommended number of tubes per pass (based on equation 10)=

$$\frac{7{\cdot}854\ S_c}{\pi\times 0{\cdot}75l}=\frac{3{\cdot}33\ S_c}{l}=\frac{1{\cdot}143\times \text{galls./min.}}{v_w} \quad \ldots \quad \ldots \quad \ldots \quad (12)$$

Where, l=length between tube plates in feet and v_w=water velocity through tubes in feet/second.

Area of Exhaust Inlet to Condenser and Area at Top Row of Tubes.

The area of the exhaust branch on the condenser will depend upon the quantity, vacuum and quality of the exhaust steam and in the writer's opinion should be proportioned to conform with the steam speed corresponding to the designed pressure drop across the condenser.

The area for steam flow in way of the top row of tubes should be made at least equal to the area of the exhaust branch, so as to maintain the steam speed sensibly constant. It has been shown by Sim that most of the pressure drop takes place across the first five or six rows of tubes. On no account should the area in way of the top rows of tubes be less than the area of the exhaust branch or there will be an excessive pressure drop due to throttling.

It will be seen from equation (1), *viz.*:

$$v=21{\cdot}1\sqrt{\frac{d}{w}}$$

that for a given pressure drop, the permissible steam speed increases with the vacuum. Incidentally, this equation shows that it is not possible to design a condenser with zero pressure drop.

Table III shows the exhaust steam speeds in feet per second calculated from equation (1) for various pressure drops and vacua on 30in. barometer for dry saturated steam.

The above results are plotted in Fig. 3, from which the pressure drops at any exhaust steam speed and vacua can be read.

Table IV shows the temperature drop in ° Fah. at the vacua and pressure drops given in Table III. This table illustrates the importance of keeping the pressure drop to a minimum, particularly at the high vacua necessary in turbine installations.

From Table III the area of the exhaust branch and minimum area in way of the top row of tubes have been calculated per pound of exhaust steam flow per second for dry saturated steam; the results are shown in Fig. 4 and Table V. For any dryness fraction (q) it is only necessary to multiply the areas given in Fig 4 or Table V by (q) to obtain the area per pound flow per second at the required vacuum.

The total area of the exhaust branch and in way of the top rows of tubes will be,

Area in square feet=

$$\frac{W.q.V_s}{3{,}600.v} \quad \ldots \quad \ldots \quad \ldots \quad \ldots \quad \ldots \quad (11)$$

Where,

W=weight of exhaust steam flow per hour.

q=dryness fraction of exhaust steam.

V_s=volume in cubic feet per pound of dry saturated steam at the given vacuum.

v=velocity of flow corresponding to the permissible pressure drop, determined from equation (1) in feet per second.

As an illustration of the use of the above tables, suppose we have a condenser dealing with 72,000lb. of steam per hour at 29in. vacuum (30in. bar.) with a dryness fraction of 0·90 and a permissible pressure drop of 0·10in. Then, the area of the exhaust branch and in way of the top row of tubes should not be less than,

$$\frac{72{,}000\times 0{\cdot}90\times 3{\cdot}83}{3{,}600}=68{\cdot}94 \text{ sq. ft.}$$

Particulars of the areas of the exhaust branches and in way of the top row of tubes, together with the designed velocities, for the condensers mentioned in this paper are given in Table VI. In each case the recorded vacuum was 28½in. (30in. bar.).

To find the area required at the bottom of the lane it is necessary to calculate the quantity of steam required to raise the

TABLE III.

Exhaust Steam Speeds for Dry Saturated Steam in Feet per Second, at Various Vacua and Pressure Drops.

Pressure drop ins. hg.	0·05	0·10	0·15	0·20	0·25	0·30	0·35	0·40
27in. vac. ...	71·9	101·6	124·5	143·7	160·8	176·0	190·1	203·3
27¼in. „ ...	74·9	105·9	129·7	149·7	167·4	183·4	198·0	211·7
27½in. „ ...	78·3	110·8	135·7	156·6	175·1	191·8	207·2	221·5
27¾in. „ ...	82·3	116·4	142·5	164·5	184·0	201·5	217·6	232·7
28in. „ ...	87·0	123·0	150·7	174·0	194·5	213·0	230·2	246·0
28¼in. „ ...	92·6	131·0	160·4	185·2	207·0	226·9	245·0	262·0
28½in. „ ...	99·6	140·8	172·5	199·1	222·6	244·0	263·4	281·5
28¾in. „ ...	108·5	153·5	188·0	217·0	242·6	266·0	287·0	307·0
29in. „ ...	120·6	170·6	208·8	241·0	269·6	295·4	319·0	341·0
29¼in. „ ...	138·2	195·5	239·3	276·3	309·0	338·5	365·7	386·4
29½in. „ ...	167·3	236·6	290·0	334·6	374·0	409·7	442·7	473·0

TABLE IV.

Temperature Drop Across Condenser ° Fah. for Various Pressure Drops.

Pressure drop ins. hg.	0·05	0·10	0·15	0·20	0·25	0·30	0·35	0·40
27in. vac. ...	0·6°	1·2°	1·8°	2·4°	3·1°	3·7°	4·3°	5·0°
27¼in. „ ...	0·6°	1·2°	1·9°	2·5°	3·2°	3·9°	4·6°	5·3°
27½in. „ ...	0·7°	1·4°	2·1°	2·9°	3·6°	4·4°	5·2°	6·0°
27¾in. „ ...	0·8°	1·6°	2·4°	3·2°	4·0°	4·9°	5·7°	6·6°
28in. „ ...	0·9°	1·7°	2·6°	3·5°	4·4°	5·4°	6·3°	7·3°
28¼in. „ ...	1·0°	1·9°	2·9°	4·0°	5·0°	6·1°	7·2°	8·4°
28½in. „ ...	1·1°	2·2°	3·4°	4·6°	5·8°	7·1°	8·4°	9·8°
28¾in. „ ...	1·3°	2·6°	4·0°	5·4°	6·9°	8·5°	10·1°	11·8°
29in. „ ...	1·6°	3·2°	4·9°	6·7°	8·7°	10·7°	12·2°	15·1°
29¼in. „ ...	2·0°	4·1°	6·4°	8·9°	11·6°	14·5°	17·7°	
29½in. „ ...	2·9°	6·1°						

TABLE V.

Area of Exhaust Branch in Square Feet per Pound of Exhaust Steam Flow per Second, Dry Saturated.

Pressure drop ins. hg.	0·05	0·10	0·15	0·20	0·25	0·30	0·35	0·40
27in. vac. ...	3·222	2·28	1·860	1·614	1·441	1·316	1·220	1·140
27¼in. „ ...	3·360	2·375	1·940	1·680	1·501	1·370	1·270	1·189
27½in. „ ...	3·518	2·483	2·030	1·760	1·571	1·435	1·330	1·243
27¾in. „ ...	3·690	2·610	2·130	1·849	1·651	1·509	1·395	1·305
28in. „ ...	3·906	2·760	2·255	1·954	1·749	1·594	1·476	1·380
28¼in. „ ...	4·160	2·940	2·400	2·080	1·860	1·699	1·571	1·470
28½in. „ ...	4·470	3·160	2·580	2·235	2·000	1·825	1·690	1·580
28¾in. „ ...	4·870	3·442	2·810	2·439	2·180	1·989	1·841	1·721
29in. „ ...	5·410	3·830	3·125	2·708	2·421	2·208	2·046	1·915
29¼in. „ ...	6·200	4·390	3·580	3·100	2·776	2·530	2·347	2·220
29½in. „ ...	7·500	5·305	4·330	3·760	3·360	3·070	2·840	2·659

TABLE VI.

Condenser.	*Steam condensed per hour, pounds.*	*Area of exhaust branch, sq. ft.*	*Area of central lane at top row of tubes, sq. ft.*	*Area between top row of tubes, sq. ft.*	*Total area in way of top row of tubes, sq. ft.*	*Area at bottom of lane, sq. ft.*	*Velocity in exhaust branch, feet/sec.*	*Velocity past top row of tubes, feet/sec.*
A ...	63,500	51	38·125	12·75	50·875	8·25	139·6	140·0
B ...	82,500	54·5	58·50	28·80	87·30	9·09	169·0	105·4
C ...	58,000	44·0	38·80	19·95	58·75	8·04	148·0	110·7
D ...	77,500	53·0	48·0	15·10	63·10	9·225	164·0	137·5
E ...	111,200	70·0	73·0	29·80	102·80	16·96	178·0	121·2
F ...	103,000	84·6	53·5	38·60	92·10	12·92	136·7	125·5

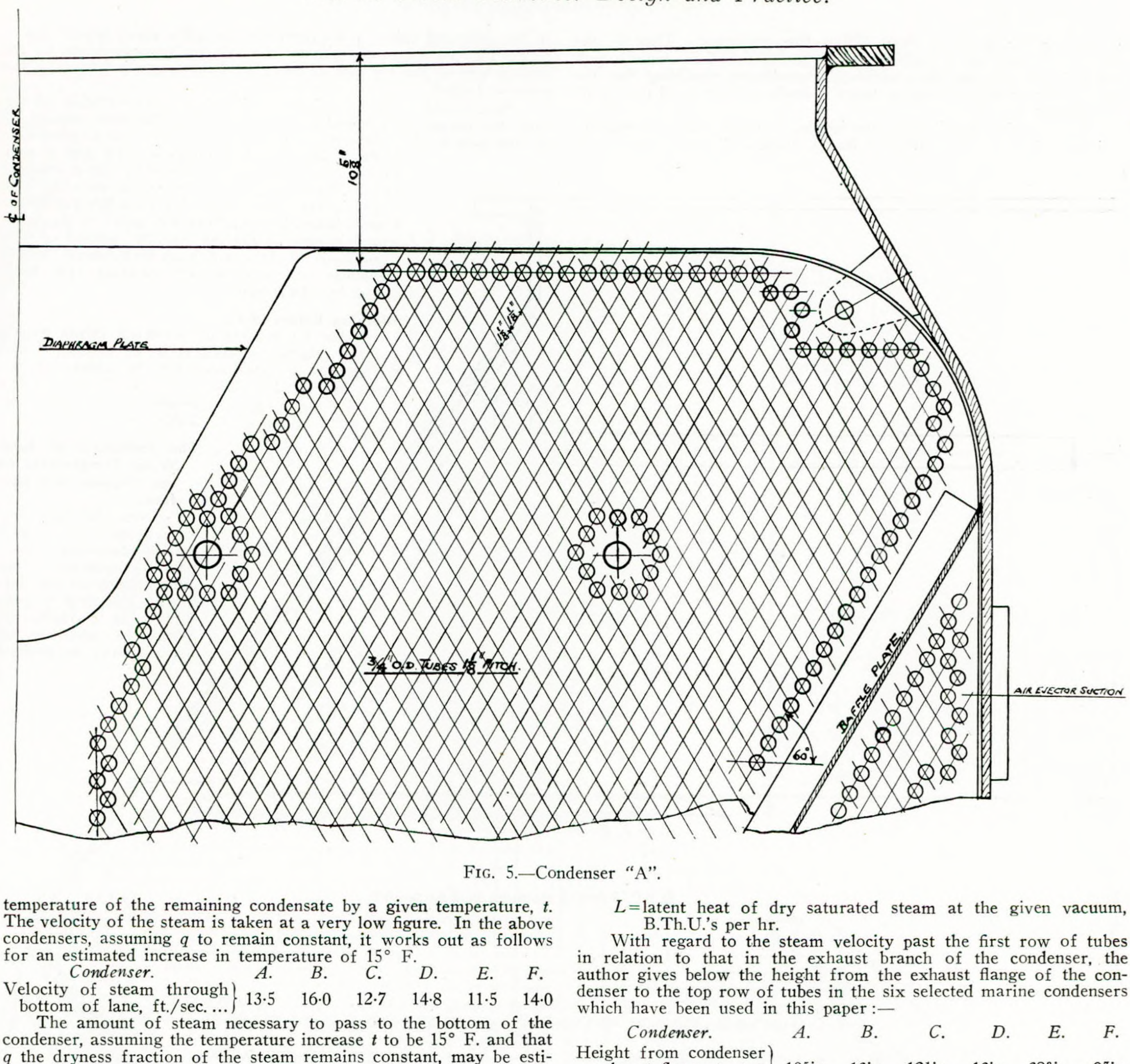


FIG. 5.—Condenser "A".

temperature of the remaining condensate by a given temperature, t. The velocity of the steam is taken at a very low figure. In the above condensers, assuming q to remain constant, it works out as follows for an estimated increase in temperature of 15° F.

Condenser.	*A.*	*B.*	*C.*	*D.*	*E.*	*F.*
Velocity of steam through bottom of lane, ft./sec. ...	13·5	16·0	12·7	14·8	11·5	14·0

The amount of steam necessary to pass to the bottom of the condenser, assuming the temperature increase t to be 15° F. and that q the dryness fraction of the steam remains constant, may be estimated from equation (12).

$$w=\frac{15.W}{(q.L+15)}\text{ lb. per hour} \qquad \ldots \quad (12)$$

Where,

W=total weight of steam entering condenser, pounds per hour.
q=dryness fraction.
L=latent heat of dry saturated steam at the given vacuum, B.Th.U.'s per hr.

With regard to the steam velocity past the first row of tubes in relation to that in the exhaust branch of the condenser, the author gives below the height from the exhaust flange of the condenser to the top row of tubes in the six selected marine condensers which have been used in this paper:—

Condenser.	*A.*	*B.*	*C.*	*D.*	*E.*	*F.*
Height from condenser exhaust flange to top row of tubes ...	10⅝in.	13in.	12½in.	13in.	38¾in.	9⅞in.

In the writer's opinion, unless there is a distance of the order of that shown for condenser E, which is for turbo-electric drive, between the exhaust flange and the top row of tubes, with easy slopes to the ends and sides of the condenser, the making of the area in way of the top row of tubes considerably in excess of the area through the exhaust branch will have practically no effect on the steam speed past the top row of tubes. In geared turbine installations the height available for the condenser is dependent upon the height of the shafting above the tank top to suit the gearing, and in consequence the distance between the exhaust flange and the top row of tubes is limited and the slopes to the ends and sides of the condenser are fairly flat.

It is in view of this fact that the author suggests that the velocity in the exhaust branch should be used in equation (1) for

TABLE VII.

WORKING VALUES OF K_T.
THE OVERALL RATE OF HEAT TRANSMISSION IN B.TH.U'S PER SQUARE FOOT PER DEGREE F. MEAN TEMPERATURE DIFFERENCE PER HOUR.
MEAN TEMPERATURE OF CIRCULATING WATER=78° F.
VACUUM 28½IN. (30IN. BAR.). DRYNESS FRACTION 0·906.

Condenser.	*A*	*B*	*C*	*D*	*E*	*F*
Velocity of circulating water through tubes, ft./sec. ...	6·0	5·9	7·5	5·9	7·5	7·75
Working value of K_T ...	545	549	606	554	592	601
K_T from Fig. 2 ...	661	656	722	656	722	730
% allowance ...	17·4	16·3	16·05	15·52	18·0	17·7
Condensation rate, lb. per sq. ft./hour	6·66	6·70	7·42	6·77	7·20	7·35

calculating the pressure drop across the condenser. That is, the area of the exhaust branch should be proportioned to comply with the designed pressure drop across the condenser, providing that the area in way of the top row of tubes is made at least equal to that of the exhaust branch.

In the turbo-electric drive the height available for the condenser is independent of the height of the shafting and hence easy slopes to the ends and sides of the condenser afford a much better opportunity for the steam to take up the velocity corresponding to the area in way of the top row of tubes, if this is made in excess of the exhaust branch.

Figs. 5 and 6 show condensers A and E in explanation of the previous paragraphs, also the arrangement of the tube spacing of the top pass of the condensers. It will be observed that in condenser E the spacing is of rectangular form and in way of the top pass the horizontal distance between centres has been made 1⅜in. to increase the area available for steam flow. This condition is sometimes obtained by making the angle to the horizontal in way of the top pass 30° instead of 60°. Condensers B, D and F have this feature, whilst condenser C has rectangular spacing 1¼in. horizontal by 1⅛in. vertical.

Working Values of K_T

Table VI enables the working values of K_T for the selected condensers to be calculated and the percentage allowance on the values of K_T given in Fig. 2 or Table I to be obtained. The results are shown in Table VII.

The Influence of Inlet Water Temperature on the Vacuum and Surface.

From the performance results of several marine regenerative two-pass condensers, the author suggests the following temperature conditions as a basis for design, to determine the maximum maintained

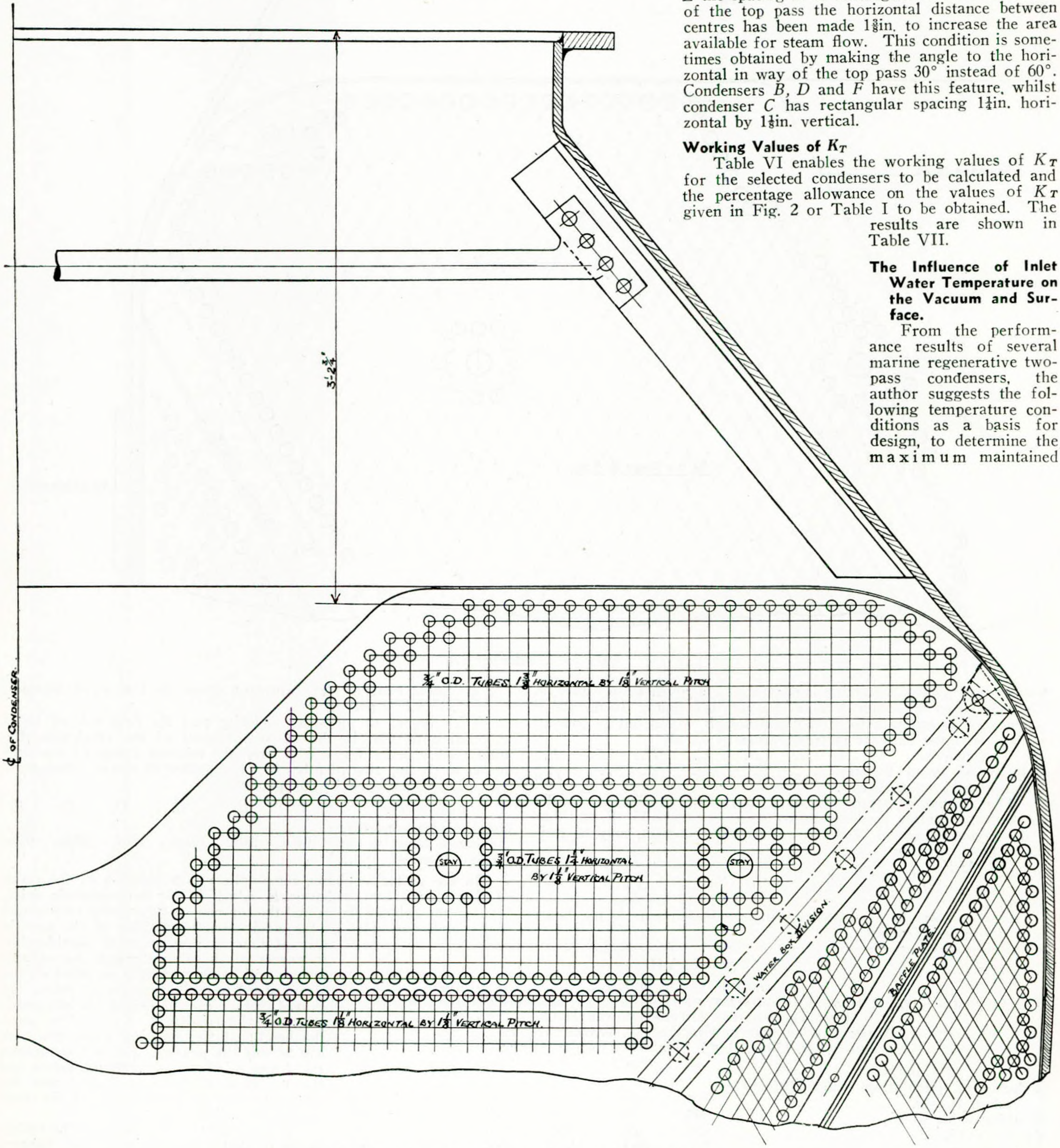


FIG. 6.—View of diaphragm plate, condenser "E".

TABLE VIII.

PROPERTIES OF LOW PRESSURE DRY SATURATED STEAM FOR EACH 0·01-IN. VACUUM
(INTERPOLATED FROM CALLENDAR'S STEAM TABLES).

Vacuum, in. hg.	Temperature, °F.	Latent heat B.Th.U.'s/ lb.	Volume, cu. ft / lb.
27·5	108·6	1029·2	275·4
27·51	108·46	1029·28	276·4
27·52	108·32	1029·56	277·4
27·53	108·18	1029·44	278·5
27·54	108·04	1029·52	279·5
27·55	107·90	1029·60	280·6
27·56	107·76	1029·68	281·7
27·57	107·62	1029·76	282·84
27·58	107·48	1029·84	284·0
27·59	107·34	1029·92	285·1
27·60	107·20	1030·00	286·2
27·61	107·06	1030·08	287·3
27·62	106·92	1030·16	288·4
27·63	106·78	1030·24	289·6
27·64	106·64	1030·32	290·7
27·65	106·50	1030·40	291·8
27·66	106·34	1030·48	293·0
27·67	106·18	1030·56	294·2
27·68	106·02	1030·64	295·5
27·69	105·86	1030·72	296·7
27·70	105·70	1030·80	297·9
27·71	105·56	1030·88	299·1
27·72	105·42	1030·96	300·3
27·73	105·28	1031·04	301·6
27·74	105·14	1031·12	302·8
27·75	105·00	1031·20	304·0
27·76	104·84	1031·28	305·3
27·77	104·68	1031·36	306·6
27·78	104·52	1031·44	308·0
27·79	104·36	1031·52	309·3
27·80	104·20	1031·60	310·6
27·81	104·04	1031·68	312·0
27·82	103·88	1031·76	313·4
27·83	103·72	1031·84	314·7
27·84	103·56	1031·92	316·2
27·85	103·40	1032·00	317·3
27·86	103·24	1032·08	319·0
27·87	103·08	1032·16	320·2
27·88	102·92	1032·24	321·8
27·89	102·76	1032·32	323·2
27·90	102·60	1032·40	324·5
27·91	102·44	1032·50	325·8
27·92	102·28	1032·60	327·2
27·93	102·12	1032·70	328·8
27·94	101·96	1032·80	330·2
27·95	101·80	1032·90	331·8
27·96	101·64	1032·97	333·1
27·97	101·48	1033·05	334·8
27·98	101·32	1033·14	336·2
27·99	101·16	1033·22	338·0
28·00	101·00	1033·30	339·8
28·01	100·82	1033·40	341·1
28·02	100·64	1033·50	342·8
28·03	100·46	1033·60	344·4
28·04	100·28	1033·70	346·0
28·05	100·10	1033·80	347·8
28·06	99·94	1033·89	349·5
28·07	99·78	1033·98	351·2
28·08	99·62	1034·06	353·0
28·09	99·46	1034·14	354·8
28·10	99·30	1034·20	356·6
28·11	99·12	1034·30	358·4
28·12	98·94	1034·40	360·1
28·13	98·76	1034·50	362·0
28·14	98·58	1034·60	364·0
28·15	98·40	1034·70	365·6
28·16	98·22	1034·80	367·5
28·17	98·04	1034·90	369·5
28·18	97·86	1035·00	371·5
28·19	97·68	1035·10	373·3
28·20	97·50	1035·20	375·4
28·21	97·32	1035·30	377·2
28·22	97·14	1035·40	379·2
28·23	96·96	1035·50	381·2
28·24	96·78	1035·60	383·2
28·25	96·60	1035·70	385·3
28·26	96·40	1035·80	387·2
28·27	96·20	1035·90	389·5
28·28	96·00	1036·00	391·5
28·29	95·80	1036·10	393·7
28·30	95·60	1036·20	396·1
28·31	95·42	1036·30	398·1
28·32	95·24	1036·40	400·4
28·33	95·06	1036·50	402·8
28·34	94·88	1036·60	405·0
28·35	94·70	1036·70	407·2
28·36	94·50	1036·80	409·6
28·37	94·30	1036·90	412·0
28·38	94·10	1037·00	414·4
28·39	93·90	1037·10	416·8
28·40	93·70	1037·20	418·9
28·41	93·48	1037·32	421·2
28·42	93·26	1037·44	424·0
28·43	93·04	1037·56	426·6
28·44	92·82	1037·68	429·2
28·45	92·60	1037·80	431·9
28·46	92·40	1037·90	434·6
28·47	92·20	1038·00	437·2
28·48	92·00	1038·10	440·0
28·49	91·80	1038·20	442·9
28·50	91·60	1038·30	445·2
28·51	91·38	1038·42	448·3
28·52	91·16	1038·54	451·0
28·53	90·94	1038·66	454·0
28·54	90·72	1038·78	457·0
28·55	90·50	1038·90	460·0
28·56	90·28	1039·01	463·0
28·57	90·06	1039·12	466·0
28·58	89·84	1039·24	469·0
28·59	89·62	1039·38	472·0
28·60	89·40	1039·50	475·2
28·61	89·16	1039·60	478·2
28·62	88·92	1039·72	481·5
28·63	88·68	1039·84	484·8
28·64	88·44	1039·96	488·0
28·65	88·20	1040·10	492·0
28·66	87·96	1040·21	495·0
28·67	87·72	1040·34	498·5
28·68	87·48	1040·47	502·0
28·69	87·24	1040·60	505·8
28·70	87·00	1040·70	509·4
28·71	86·76	1040·84	513·3
28·72	86·52	1041·00	517·3
28·73	86·28	1041·12	521·2
28·74	86·04	1041·26	525·2
28·75	85·80	1041·40	529·1
28·76	85·54	1041·52	533·2
28·77	85·28	1041·68	537·3
28·78	85·02	1041·80	541·3
28·79	84·76	1041·95	545·4
28·80	84·50	1042·10	549·5
28·81	84·24	1042·22	553·1
28·82	83·98	1042·35	557·7
28·83	83·72	1042·50	562·3
28·84	83·46	1042·64	566·9
28·85	83·20	1042·80	572·5
28·86	82·92	1042·94	577·3
28·87	82·64	1043·08	582·1
28·88	82·36	1043·22	586·9
28·89	82·08	1043·36	591·7
28·90	81·80	1043·50	596·6
28·91	81·52	1043·66	602·1
28·92	81·24	1043·82	607·6
28·93	80·96	1043·98	613·1
28·94	80·68	1044·14	618·6
28·95	80·40	1044·30	624·1
28·96	80·10	1044·46	629·9
28·97	79·80	1044·62	635·6
28·98	79·50	1044·78	641·4
28·99	79·20	1044·94	647·1
29·00	78·90	1045·10	653·0
29·01	78·58	1045·28	659·6
29·02	78·26	1045·46	666·3
29·03	77·94	1045·64	672·9
29·04	77·62	1045·82	679·6
29·05	77·30	1046·00	686·2
29·06	76·98	1046·16	693·2
29·07	76·66	1046·32	700·3
29·08	76·34	1046·48	707·3
29·09	76·02	1046·64	714·4
29·10	75·70	1046·80	721·4
29·11	75·36	1046·96	729·5
29·12	75·02	1047·12	737·6
29·13	74·68	1047·28	745·8
29·14	74·34	1047·44	753·9
29·15	74·00	1047·60	762·0
29·16	73·64	1047·78	770·7
29·17	73·28	1047·96	779·5
29·18	72·92	1048·14	788·2
29·19	72·56	1048·32	797·0
29·20	72·20	1048·50	805·7
29·21	71·80	1048·72	816·1
29·22	71·40	1048·94	826·5
29·23	71·00	1049·16	836·9
29·24	70·60	1049·38	847·3
29·25	70·20	1049·60	857·6
29·26	69·80	1049·82	868·9
29·27	69·40	1050·04	880·3
29·28	69·00	1050·26	891·6
29·29	68·60	1050·48	903·0
29·30	68·20	1050·70	914·3
29·31	67·78	1050·92	927·8
29·32	67·36	1051·14	941·4
29·33	66·94	1051·36	954·9
29·34	66·52	1051·58	968·5
29·35	66·10	1051·80	982·0
29·36	65·64	1052·04	997·2
29·37	65·18	1052·28	1012·4
29·38	64·72	1052·52	1027·6
29·39	64·26	1052·76	1042·8
29·40	63·80	1053·00	1058·0
29·41	63·30	1053·28	1076·4
29·42	62·80	1053·56	1094·8
29·43	62·30	1053·84	1113·2
29·44	61·80	1054·12	1131·6
29·45	61·30	1054·40	1150·0
29·46	60·76	1054·68	1171·4
29·47	60·22	1054·96	1192·8
29·48	59·68	1055·24	1214·2
29·49	59·14	1055·52	1235·6
29·50	58·60	1055·80	1257·0

vacuum at any given inlet sea-water temperature and the capacity of the main circulating pump.

Inlet sea-water temp. $= t°$ F.
Outlet " " $= t° + 10°$ F.
Vacuum steam " $= t° + 19°$ F.
Condensate " $= t° + 17°$ F.

Table VIII gives the properties of dry saturated steam for each 0·01in. vacuum interpolated from Callendar's steam tables covering the range of vacua associated with turbine installations, whilst Table IX shows the practical maintained vacuum from condensers designed on the above temperature conditions with surfaces determined in accordance with equation (10). The results are plotted in Fig. 7, from which the surface, vacuum and quantity of circulating water per minute for inlet sea-water temperatures from 50° F. to 94° F. per 1,000lb. of dry saturated exhaust steam per hour can be obtained. To obtain the actual surface and capacity of circulating water per min. per 1,000lb. of exhaust steam as received per hour, it is only necessary to multiply the surfaces and capacities given in Fig. 7 or Table IX by the dryness fraction as determined from the $H\phi$ diagram.

The writer considers that the condenser should be designed to maintain a specified maximum vacuum at the average sea-water temperature over the normal round voyage upon which the vessel will be engaged. The later stages of the L.P. turbine must also be designed to take full advantage of the vacuum so determined. In this respect it is suggested that for Eastern-going vessels, passing *en voyage* through the tropics, the designed maintained vacuum should be 28½in. (30in. bar.) with a sea-water temperature of 73° F., and for North Atlantic vessels 29·1in. at 57° F.

If the same vacuum is desired at a higher sea-water temperature than that given in Table IX, it may be found that the capacity of the main circulating pump and the surface of the condenser become uneconomical. For instance, it may be shown that to maintain a vacuum of 28½in. at an inlet sea-water temperature of 76° F. outlet 83° F. with a velocity of flow through the tubes of 6·5ft. per second, the condensing surface without margin for fouling would be 11 per cent. greater and the main circulating pump capacity 43 per cent. greater than for a condenser designed to maintain 28½in. vacuum at 73° F. inlet water temperature.

The surface in conjunction with the quantity of circulating water given in Table IX may require modification to suit the length available in the engine room for the accommodation of the condenser, but it is recommended that the surface and quantity of circulating water stated in the table should be

TABLE IX.

WORKING VACUUM, SURFACE PER 1,000 LB. OF DRY SATURATED EXHAUST STEAM PER HOUR AND CIRCULATING WATER PER MINUTE PER 1,000 LB. OF DRY SATURATED EXHAUST STEAM PER HOUR.

CONDITIONS: RISE IN TEMPERATURE OF CIRCULATING WATER ACROSS CONDENSER = 10° F.
STEAM TEMPERATURE 19° F. ABOVE INLET SEA WATER.
CONDENSATE TEMPERATURE 17° F. ABOVE INLET SEA WATER.
WATER VELOCITY THROUGH TUBES, 6·5 FT. PER SECOND.
MEAN TEMPERATURE DIFFERENCE BETWEEN STEAM AND CIRCULATING WATER, $\theta_m = 12{\cdot}58°$ F.

Inlet sea water temp., °F.	*Mean circulating water temp., °F.*	*Working vacuum, ins. hg.*	K_T *from Fig. 2 or Table 1.*	$Sc. = \frac{W.L.}{\theta_m K_T}$	*S. Total surface without margin.*	S_R *Total surface allowing 16% margin for fouling.*	*Circulating water, gallons per minute.*
50	55	29·28	628	133·0	146·30	174·1	175·00
52	57	29·23	634	131·5	144·65	172·2	174·85
54	59	29·18	639	130·25	143·27	170·5	174·69
56	61	29·12	644	129·0	141·90	169·0	174·52
58	63	29·06	649	128·0	140·80	167·5	174·35
60	65	29·00	653	127·2	139·92	166·5	174·20
62	67	28·93	658	126·0	138·60	165·0	174·0
64	69	28·86	662·5	125·0	137·50	163·6	173·82
66	71	28·78	667	124·0	136·40	162·3	173·65
68	73	28·70	671·5	123·2	135·52	161·3	173·47
70	75	28·62	676	122·0	134·20	159·7	173·3
72	77	28·53	680	121·3	133·43	158·9	173·1
74	79	28·43	684·5	120·5	132·55	157·8	172·92
76	81	28·33	689	119·7	131·67	156·5	172·75
78	83	28·23	693	118·8	130·68	155·5	172·57
80	85	28·12	697	118·0	129·80	154·5	172·4
82	87	28·00	700	117·3	129·03	153·5	172·2
84	89	27·87	704	116·6	128·26	152·5	172·0
86	91	27·75	707	115·9	127·49	151·7	171·83
88	93	27·60	711	115·1	126·61	150·9	171·65
90	95	27·47	714	114·3	125·73	149·8	171·5
92	97	27·30	717	113·7	125·07	148·7	171·32
94	99	27·15	720	113·1	124·41	148·0	171·13

TABLE X.

VALUES OF C FOR COMMERCIALLY CLEAN TUBES ⅝-INCH OUTSIDE DIAMETER BY 18 L.S.G. THICK. (BASED ON GUY AND WINSTANLEY).

Mean circulating water temperature °F.	50°	60°	70°	80°	90°	100°	110°	120°
3ft./sec. ...	0·0375	0·036	0·0345	0·033	0·03225	0·0315	0·0306	0·0297
4ft./sec. ...	0·0345	0·033	0·03165	0·03075	0·0297	0·0291	0·0285	0·02775
5ft./sec. ...	0·033	0·0315	0·030	0·0291	0·0282	0·0276	0·02685	0·02625
6ft./sec. ...	0·0315	0·030	0·0288	0·02775	0·0270	0·0264	0·0258	0·0252
7ft./sec. ...	0·030	0·0285	0·0276	0·0267	0·0261	0·0252	0·02475	0·0243
8ft./sec. ...	0·0288	0·0276	0·0267	0·0258	0·0252	0·02475	0·0240	0·0234

adhered to as closely as practical limitations of space will permit.

Frictional Loss of Head Across a Condenser.

The frictional loss of head across a condenser is made up as follows :—

(1) Loss of head due to friction through tubes.

(2) Loss of head at entrance and exit from tubes.

(3) Loss of head in the water boxes.

The fundamental formula for the loss of head due to the flow of water through tubes may be written :—

$$h_t = \frac{C.l.v^2}{2g.d.} \text{ft.} \quad \ldots \quad (13)$$

Where, C is a coefficient; l = length of tube in ft.;
v = velocity of flow in ft. per second; d = internal diameter of tube in ft.
g = 32·2ft. per second per second.

The coefficient C is dependent upon the velocity of flow, the mean temperature of the circulating water and the internal diameter of the tube. The values of C, based on the experimental work of Guy and Winstanley, are given in Table X, for various mean circulating water temperatures and velocities of flow.

It was suggested by these authors that C could be taken as 0·030 for commercially clean tubes when the water speed was 5ft. per second and the mean circulating water temperature 70° F. for ⅝in. outside diameter tubes, 18 L.S.G. thick.

A check on the value of 0·030 for C has been made by the author from the experiments of Box, who showed that the loss of head due to friction may be obtained from the following expression :—

$$h_t = \frac{0{\cdot}005724 l.v^2}{d} \quad \ldots \quad (14)$$

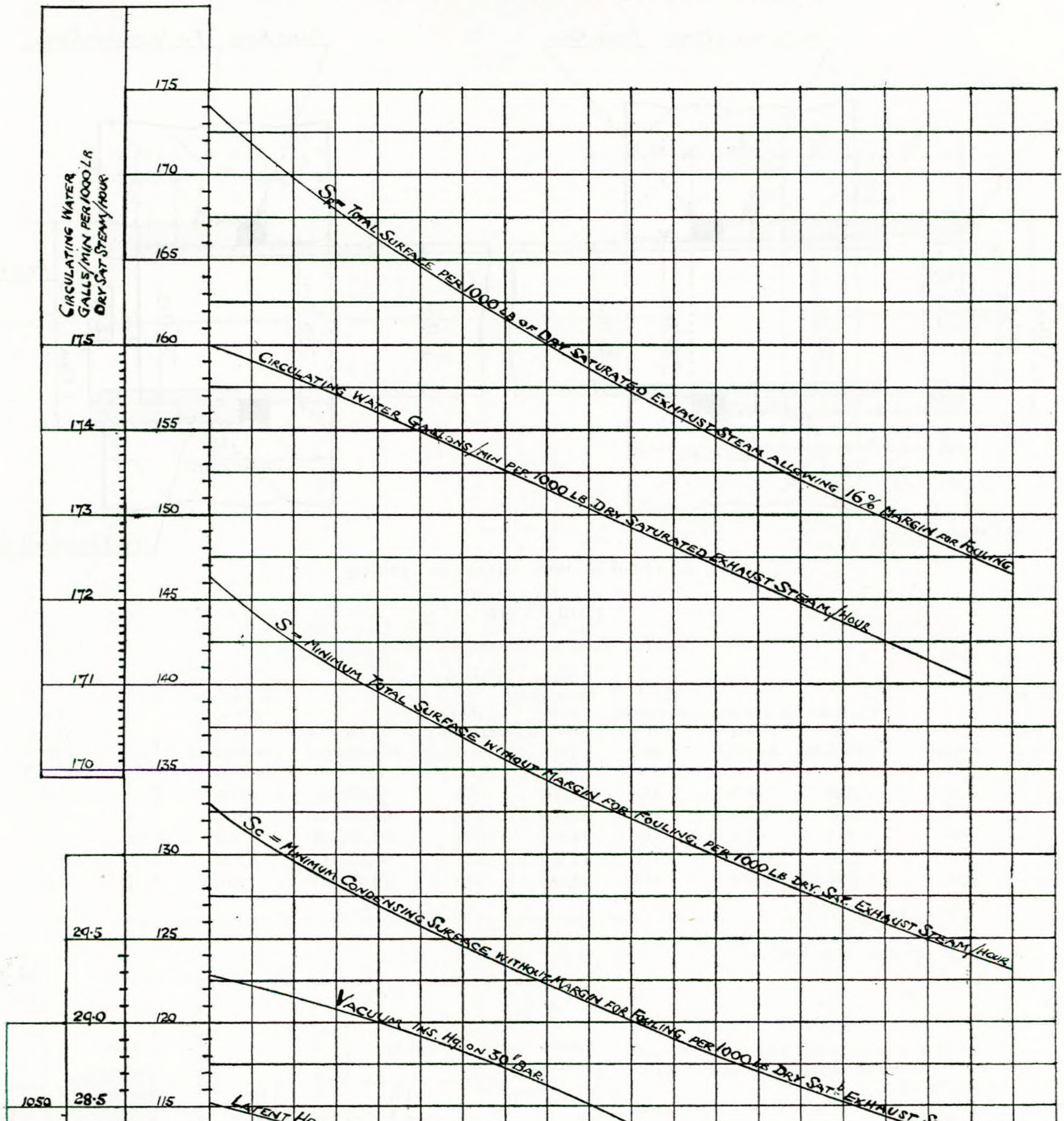
Where, l and v are in feet as before and d is in inches.

TABLE XI.

Condenser.		*A.*	*B.*	*C.*	*D.*	*E.*	*F.*
Velocity of circulating water through tubes, ft./sec.		6·0	5·9	7·5	5·9	7·5	7·75
Length of tubes		12ft. 2½in.	12ft. 0in.	13ft. 6in.	11ft. 8½in.	14ft. 6in.	14ft. 0in.
Velocity through inlet and outlet branches, ft./sec.		9·17	7·62	8·4	8·975	7·10	8·85
Frictional head loss through tubes in ft.	Guy and Winstanley...	8·79	8·37	14·48	8·21	15·35	16·0
	Box	9·47	9·026	15·09	8·84	17·06	17·76
Frictional head loss in water boxes in ft.	Guy and Winstanley...	1·306	0·902	1·095	1·25	0·783	1·217
	Sim	1·50	1·50	1·50	1·50	1·50	1·50
Total loss of head across condenser in ft.	Guy and Winstanley...	10·096	9·272	15·575	9·46	16·133	17·217
	Box and Sim	10·968	10·526	16·59	10·34	18·56	19·26

TABLE XII. SERVICE RESULTS, CONDENSER "A".

Sea temp., °F.	*Over-board dis-charge temp., °F.*	*Con-densate, °F.*	*Vacuum on 30in. bar, in.*	*Vacuum steam temp., °F.*	*Main circulating pump, turbine, r.p.m.*	*Main circulating pump, gallons per min.*	*Velocity of flow through tubes, ft./sec.*	*B.Th.U.'s transferred per hour.*	*B.Th.U.'s per sq. ft. of condensing surface.*	θ_m, *°F.*	K_T *from service results.*	K_T *from Fig. 2.*
58	71·0	76·75	29·0	78·9	2,425	6,847	3·80	53,406,600	6,188	12·57	492·3	510
58	71·75	79·75	28·9	81·8	2,394	6,760	3·756	55,770,000	6,461	15·10	427·8	505
58	72·0	80·5	28·9	81·8	2,425	6,847	3·80	57,456,000	6,657	15·27	435·9	512
61	73·0	80·0	28·9	81·8	2,000	5,650	3·14	40,680,000	4,713	13·24	356·0	465
71	82·0	89·0	28·6	89·4	2,150	6,070	3·37	40,062,000	4,641	11·93	388·0	502
75	85·0	90·5	28·5	91·6	2,500	7,058	3·92	42,348,000	4,906	10·40	471·7	553
77	87·0	93·0	28·4	93·7	2,200	6,210	3·45	37,260,000	4,317	10·70	403·5	520
80	90·0	94·5	28·25	96·6	2,500	7,058	3·92	42,348,000	4,906	10·0	490·6	562
82	92·0	97·0	28·19	98·0	2,500	7,058	3·92	42,348,000	4,906	9·82	499·7	565
84	94·0	100·0	28·08	100·0	2,483	7,005	3·89	42,030,000	4,870	10·20	477·4	567
86	96·0	102·0	27·90	102·6	2,425	6,847	3·80	41,082,000	4,760	10·61	448·6	570
88·5	98·5	104·0	27·71	105·7	2,500	7,058	3·92	42,348,000	4,906	10·82	453·4	575



NOTE. TO OBTAIN ACTUAL RECOMMENDED TOTAL SURFACE MULTIPLY S_R BY THE DRYNESS FRACTION OF THE EXHAUST STEAM. ALSO CIRCULATING WATER/MIN FROM CURVE MUST BE MULTIPLIED BY THE DRYNESS FRACTION TO OBTAIN ACTUAL QUANTITY REQUIRED.

FIG. 7.

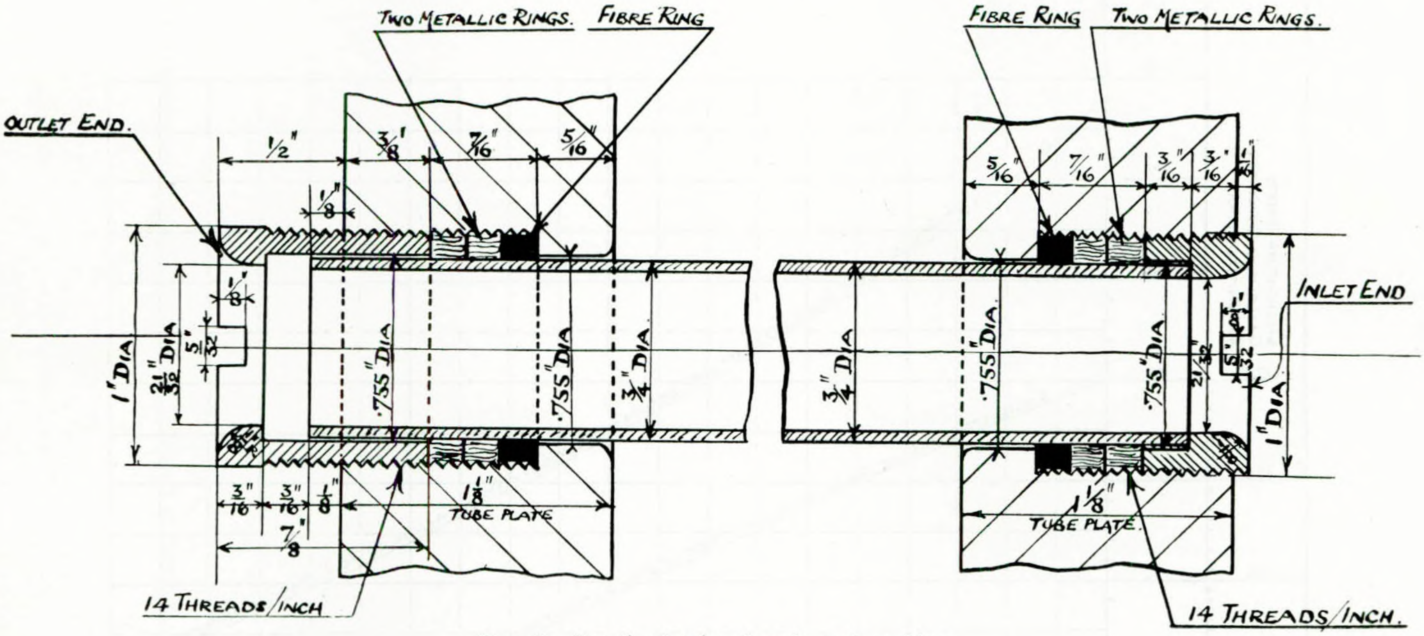


FIG. 8.—Detail of tube, ferrule and packing.

TABLE XIII.

SERVICE RESULTS, CONDENSER "E".

Sea temp., °F.	*Over-board dis-charge temp., °F.*	*Con-densate, °F.*	*Vacuum on 30in. bar, in.*	*Vacuum steam temp., °F*	*Main circulating pump. Pump r.p.m.*	*Main circulating pump, gallons per min.*	*Velocity of flow through tubes, ft./sec.*	*B.Th.U.'s transferred per hour.*	*B.Th.U.'s per sq. ft. of condensing surface.*	θ_m *°F.*	K_T *from service results.*	K_T *from Fig. 2.*
60	71	76	29·0	78·9	445	12,700	5·07	83,820,000	6,000	11·5	521	590
70	81	88	28·6	89·4	490	14,000	5·58	92,400,000	6,610	12·6	525	630
80	92	100	28·1	99·3	510	14,580	5·81	104,976,000	7,510	12·6	596	670
88	99	107	27·6	107·2	510	14,580	5·81	96,228,000	6,890	12·5	550	680
90	102	109	27·4	109·9	510	14,580	5·81	104,976,000	7,510	12·63	595	690

Putting d in inches, equation (12) becomes,

$$h_t = \frac{12C.l.v^2}{2gd}$$

so that $\frac{12C}{2g} = 0{\cdot}005724$, whence $C = 0{\cdot}030719$.

In the formulæ of Box it is assumed that C is constant for all velocities of flow, mean circulating water temperatures and bore of tubes.

With regard to the entrance and exit losses, these may be stated in the form $\frac{C_2.v^2}{2g}$ where C_2 for tubes ferruled at both ends, from the same source, may be taken as 1·6. This value is confirmed by Sim from the experiments due to Box which gives $C_2 = 1{\cdot}59$.

The water box losses are difficult to measure and assess and there is a considerable difference put forward by the various sources investigated by the author.

Sim suggests the following:—

For single-pass condensers	... ...	1·0ft.
„ two- „ „	... ...	1·5ft.
„ three- „ „	... ...	2·0ft.

Guy and Winstanley suggest that the water box loss may be obtained from the value of $\frac{v_b^2}{2g}$ where v_b is the velocity of flow through the branches, providing this value is used in computing C from the frictional head loss in actual condenser tests. This formula makes no allowance for the number of passes in the condenser. The following formula for the total loss of head across the condenser was found by these authors to give results remark-

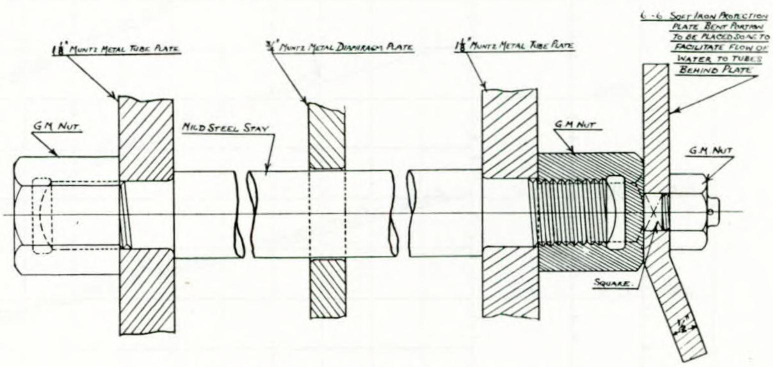


FIG. 9.—Detail of tube plate stays.

ably near the actual test figures obtained for two-pass condensers in which the tubes had been wire-brushed and were ferruled at both ends.

$$\text{Total loss of head in ft. } H = n\left(\frac{12.C\ l.v^2}{2gd} + \frac{1{\cdot}6v^2}{2g}\right) + \frac{v_b^2}{2g} \quad \ldots \quad (15)$$

Where n = number of passes.

C = coefficient taken from Table 10.

l = overall length of tube in ft.

v = velocity of flow through tubes in ft. per second.

d = bore of tube in inches.

v_b = velocity of water through branches of condenser.

Adopting the values stated above due to Box and the waterbox

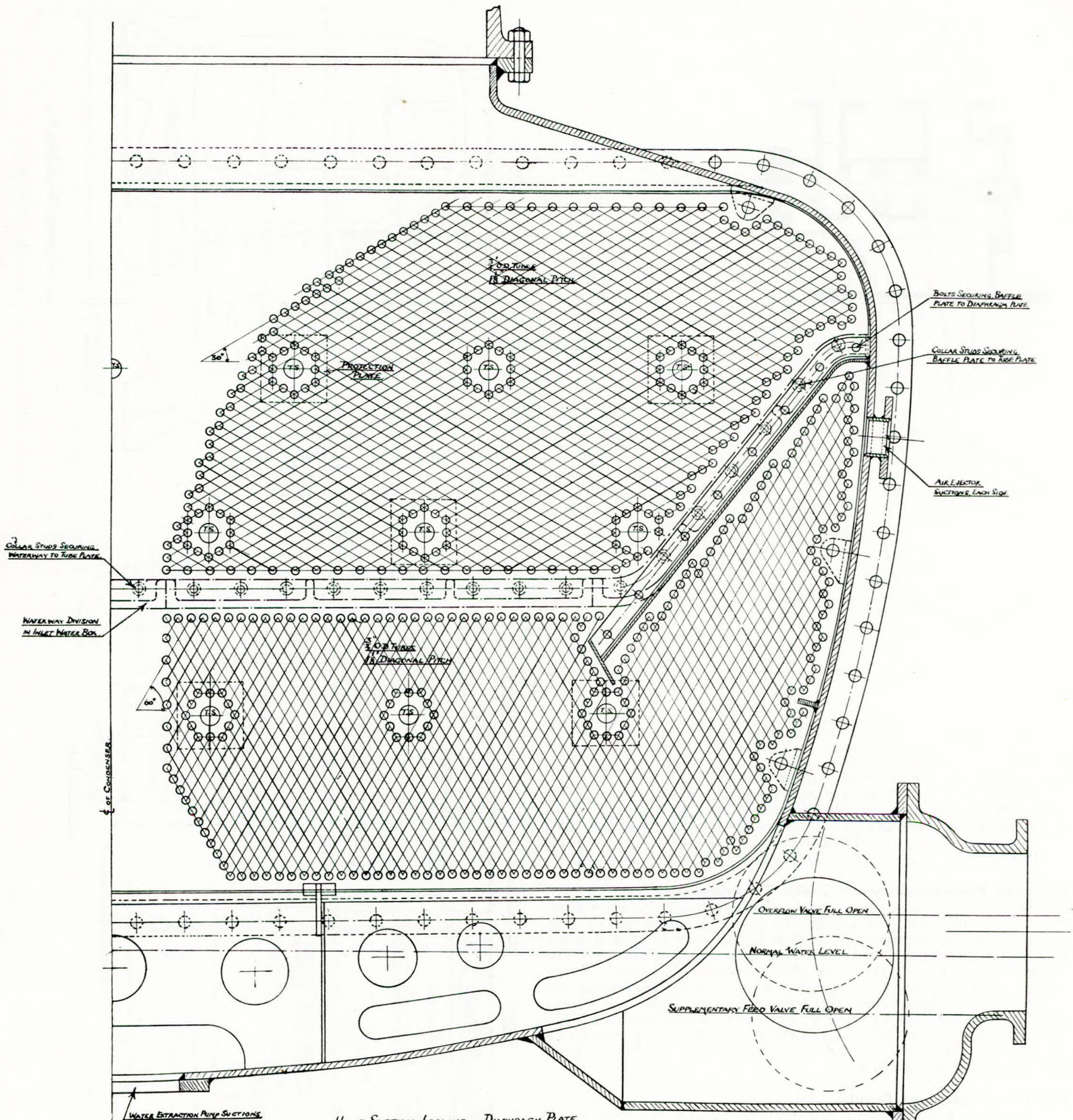


FIG. 10.—Typical tube layout of regenerative marine two-pass condenser.

losses given by Sim, the total loss of head across the condenser in feet would be as follows:—

For single-pass condensers, $H=\left(\frac{12\times 0{\cdot}0307l.v^2}{2g.d.}+\frac{1{\cdot}59v^2}{2g}\right)+1{\cdot}0$ ... (16)

For two-pass condensers, $H=2\left(\frac{12\times {\cdot}0307l.v^2}{2g.d.}+\frac{1{\cdot}59v^2}{2g}\right)+1{\cdot}5$ ... (17)

For three-pass condensers, $H=3\left(\frac{12\times {\cdot}0307l.v^2}{2g.d.}+\frac{1{\cdot}59v^2}{2g}\right)+2{\cdot}0$ ... (18)

Table XI shows the frictional head loss across the two-pass condensers of this paper, adopting the corresponding value of *C* from Table X as compared with using equation (17) above in which *C* has been taken as 0·0307, the mean circulating-water temperature being 78° F. in each case and the vacuum 28½in. (30in. bar.).

At lower mean circulating water temperatures the difference between the formulæ will be less. At higher temperatures and velocities it will be greater and the coefficient *C* from Table X will probably give the more accurate result.

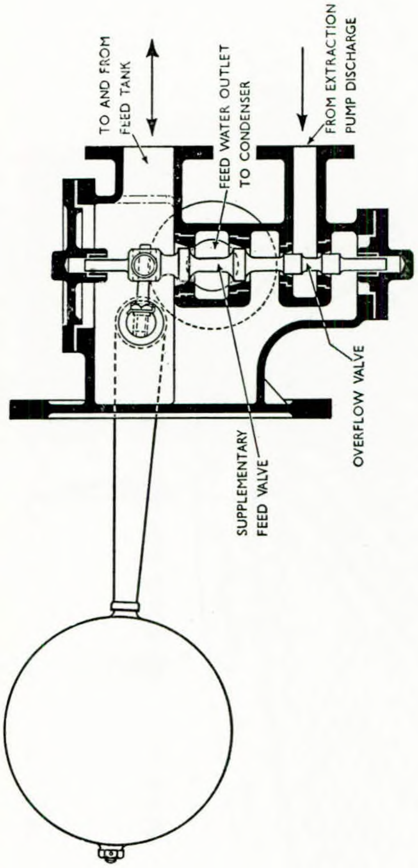


FIG. 11.

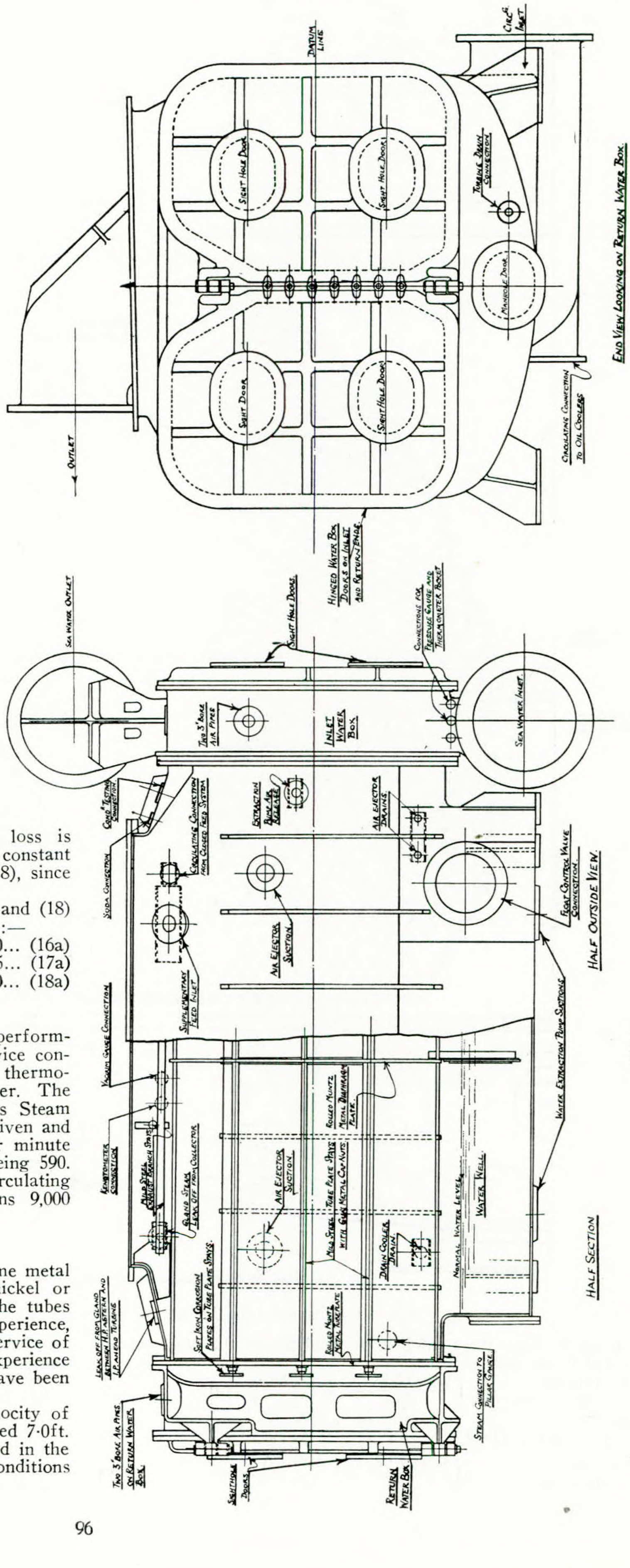


FIG. 12.—Main condenser assembly arrangement.

For practical purposes, seeing that the water box loss is empirical, the author considers it preferable to adopt a constant coefficient $C=0{\cdot}0307$ as given in equations (16), (17) and (18), since the head thereby obtained errs on the safe side.

By taking $C=0{\cdot}0307$ as a constant, equations (16), (17) and (18) reduce to the following for $\frac{3}{4}$in. o.d. tubes, 18 L.S.G. thick:—

For single-pass condensers, $H=(0{\cdot}00874\, l.v.^2+0{\cdot}02469\, v^2)+1{\cdot}0$... (16a)

„ two- „ „ $H=2(0{\cdot}00874\, l.v^2+0{\cdot}02469\, v^2)+1{\cdot}5$... (17a)

„ three- „ „ $H=3(0{\cdot}00874\, l.v^2+0{\cdot}02469\, v^2)+2{\cdot}0$... (18a)

Condenser Performances under Service Conditions.

To ascertain the working values of K_T, the following performance figures are given for condensers A and E under service conditions. The temperatures were taken by mercury in steel thermometers and the vacuum by kenotometer on a 30-in. barometer. The vacuum steam temperature was obtained from Callendar's Steam Tables. In condenser A the circulating pump is turbine driven and under test conditions delivered 12,000 gallons of water per minute at a turbine speed of 4,250 r.p.m. the pump revolutions being 590. For condenser E two vertical spindle electrically-driven circulating pumps were fitted. Each delivered under test conditions 9,000 gallons per minute at 630 r.p.m.

Material of Tubes and Ferrules.

The tubes and ferrules should both be made of the same metal and it is recommended they should be of 70/30 cupro-nickel or aluminium brass. In a number of condensers in which the tubes are of the former material and of which the author has experience, very few tubes have required renewal during a period of service of over 12 years. The writer has not the same length of experience with aluminium brass, but in vessels so fitted the results have been very satisfactory.

The author considers that in condenser design the velocity of flow through the tubes at maximum power should not exceed 7·0ft. per second and would state that in the condensers mentioned in the previous paragraph the velocity of flow under service conditions did not exceed 6·0ft. per second.

The tube packing in all cases should be of the metallic type, in which the condenser should be tight without the ferrules. The inlet ferrules for preference should be of the flush-checked type to reduce the frictional head and to improve the water flow at entrance to the tubes (see Fig. 8).

Tube-plate Stays.—The tube-plate stays are usually of mild steel with rolled naval brass or gunmetal nuts in the water space. A typical drawing of the stays is shown in Fig. 9, together with the soft iron protection plates. The latter should be maintained in an efficient condition as experience has proved that they are a decided advantage in the protection of the water boxes.

Some Notes on Regenerative Condensers and Details of Design.

At the bottom of all regenerative condensers a well is formed having a water capacity according to the size of the installation to ensure steadiness and stability to the closed-feed system under normal running conditions. In this well a float is fitted operating the supplementary feed and overflow valve which maintains the level of water in the well.

An arrangement of the float is shown in the typical tube-plate layout, Fig. 10, and the supplementary feed and overflow valve in Fig. 11. In the tube-plate layout the top pass is shown with the tubes at an angle of 30° to give additional space between the tubes.

The height of the water is shown by a gauge glass adjacent to the condenser, the connection for which is indicated in the general assembly drawing, Fig. 12.

It is essential for the supplementary feed to enter the condenser through a sprayer as high up on the condenser as possible and above the air baffle plate to ensure that the water is properly de-aerated before passing to the feed system. In this connection, if the supplementary feed water is below the temperature corresponding to the vacuum it may be questioned whether this water is evaporated and the air liberated. However, in modern closed-feed systems the de-aeration of the supplementary feed water is assured as the temperature of the water in the feed tank is normally well above the temperature corresponding to the vacuum. De-aeration of the feed water is of paramount importance in high pressure water-tube boiler installations and is a feature of the regenerative condenser.

All inlet connections to the steam space of the condenser should be fitted with a baffle to prevent impingement on the tubes.

The inlet and return water end doors should preferably be hinged and large manholes should be fitted in both the inlet and return doors to enable tubes to be stopped or repacked without having to open or remove the complete door.

Water boxes should be made as deep as possible and air pipes should be fitted to each water box to prevent air locks and to remove air from the circulating system. A typical drawing of a condenser showing the mountings, doors and general assembly is shown in Fig. 12.

In high pressure installations, in which clean steam is a *sine qua non*, the steam side of the tubes remains exceptionally clean but the water sides of the tubes should be wire brushed at least every three to four months.

REFERENCES.

"Some Factors in the Design of Surface Condensing Plant", by H. L. Guy, D.Sc. and E. V. Winstanley. Trans. of Institution of Mechanical Engineers, February, 1934.

"Steam Condensing Plant", by J. Sim, B.Sc.(Eng.). Published by Blackie & Son, Ltd.

"The Coefficients of Heat Transfer from Tube to Water", by A. Eagle, B.Sc. and R. M. Ferguson, M.Sc., A.M.I.Mech.E. Trans. of Institution of Mechanical Engineers, November, 1930.

*"Steam Tugs, Past, Present and Future"—Discussion.

Mr. A. M. Riddell (Member), referring to the reply made by Mr. G. T. Shoosmith to the discussion on the above paper, has written as follows:—"Virtual disc area" demonstrates the effect if a nozzle were to be fitted round the *existing* propeller of the "Redeem"—44·5 sq. ft. The theoretical jet efficiency would thereby be increased to that of an open propeller of 67 sq. ft. disc area without alteration to hull, stern frame or propeller.

The major factor which governed propeller diameter in the "Sir Francis Spring" was the protection afforded by the deck line, provision having to be made for the vessel heeling. Mr. Shoosmith quotes 10ft. diameter for the nozzle, but such diameter lies forward of the propeller tips where greater beam is available for protection.

Tunnel sterns are usually limited to shallow draught and/or Diesel vessels. In Thames steam tugs permissible pitch/diameter ratio would generally make the use of tunnel sterns unprofitable; the tunnel stern, however, is embodied in the nozzle design where conditions warrant it.

The tests in Grimsby were not "stern to stern" as stated by Mr. Shoosmith. The method of trials was explained in an article in the "Shipbuilding and Shipping Record" dated 16th June, 1939. A member of Mr. Shoosmith's staff was actually present at these trials.

Mr. Shoosmith "has not been able to find any figures which fully substantiate" Mr. Tritton's statement that "the Kort nozzle tug has the potentiality on a bollard pull of a normal tug of practically double the horse power". Trials carried out with the "John Hamilton" proved that the horse powers required with nozzle for the same static pulls as without nozzle, were less than 50 per cent.; this reduction in power was maintained over a range of from half to full power. These results were published in the technical press. In the L.N.E.R. tug trials, the maximum pull obtained without nozzle was 2·65 tons on 159 s.h.p. and a pull of 2·6 tons was obtained with nozzle for only 85 s.h.p.; these latter trials were also witnessed by a member of Mr. Shoosmith's staff.

Mr. Shoosmith mentions tugs with a "pull of 54 s.h.p. per ton and better". It is clear that he appreciates the effect propeller loading has on available static pull per s.h.p. The Diesel-electric tug "Acklam Cross" obtained pulls of from 28 to 58 s.h.p. per ton, the former being a "freak" result, as the propeller was only transmitting 1 s.h.p. per sq. ft. of disc area. The writer dealt with this matter in his contribution to the discussion on Mr. Shoosmith's paper "Diesel-Electric Propulsion" read before The Institute on November 9th, 1937. Taking "freak" results, the N.P.L. figures showed that the "John Hamilton" without nozzle obtained 48 s.h.p. per ton at 2·6 s.h.p. per sq. ft., and with nozzle 25 s.h.p. per ton at 2 s.h.p. per sq. ft. of disc area.

Mr Shoosmith's statement regarding power required for 9·27 knots in the L.N.E.R. non-nozzle tug is incorrect, this being 205, not 107 i.h.p. He also mentions reduction in free speed of the "Scud". The writer furnished him with an article, from which presumably he quoted, in which it was stated that the nozzle was designed specifically for towing duties, free speed being of no consequence.

Regarding Mr. Shoosmith's comment on the unsatisfactory astern steering of nozzle tugs, it is generally acknowledged that astern steering in single-screw vessels is feeble and not sufficient to overcome contrary external forces such as wind, current or the inertia from an opposite "swing" due to a previous ahead manœuvre. Steering trials with and without nozzle were carried out at the N.P.L. tank and the writer would be pleased to furnish Mr. Shoosmith with a copy of graph and report for these trials. As distinct from model trials, the following occurs in the aforementioned article in "Shipbuilding and Shipping Record":—"but the master of the tug states that when coming astern he can place more reliance on the astern manœuvring ability of the nozzle tug", this statement being approved by the L.N.E.R. Co. Where astern steering is important, rudders are fitted to the fore sides of the nozzle, in which case the astern steering moment would be about 60 per cent. of the ahead steering moment.

In connection with vulnerability, the writer has been promised but has not yet received a comparison of the propeller damage prior to and after the fitting of the nozzle to the tug mentioned as having been in trouble with driftwood. Last week (early June) the writer received information from the owners of the "Grangeburn" that during the blackout the vessel "contacted" a large twin-screw freighter. The port shaft (11in. diameter) and one manganese bronze propeller blade of the freighter had to be renewed. The damage to the "Grangeburn's" nozzle was limited to two dents in the outer plating of the nozzle entry and a small fracture in one plate; nothing else was distorted, the only repairs carried out being the arc-welding of the fracture "on site".

Mr. Shoosmith has made no comment on the main point raised in the discussion, that is, the cost of the nozzle in relation to its efficiency.

Mr. G. T. Shoosmith, in reply, has written: Mr. Riddell states that pitch/diameter ratio limitations would usually make the use of

* Paper published in the January, 1942 issue of the TRANSACTIONS (Vol. LIII, No. 12, p. 185). The discussion and author's reply, to which Mr. Riddell refers, appeared in the April, 1942 issue (Vol. LIV, No. 3, p. 27).

a tunnel stern unprofitable on London River tugs. It is difficult to see on what grounds this statement is made, since this ratio is commonly as high as 1·5 on such tugs, whereas the optimum is usually considered to be one.

The trials carried out at Grimsby and mentioned by Mr. Riddell bore no resemblance whatsoever to London River towing conditions. The use of a comparatively long tow line prevented any possibility of reaction between tug and tow, which is one of the largest factors under London River conditions.

The author would again emphasize that the propellers fitted to the majority of tugs are, in his opinion, of too small a diameter and too coarse a pitch. This is borne out by the figures Mr. Riddell quotes and was alluded to by the author in his paper. The point is that the increase in bollard pull claimed for the nozzle would not be so great if it were compared with an open propeller designed to utilize the absolute maximum disc area (*i.e.* an area similar to that of the nozzle itself).

Regarding the free-running speed of the non-nozzle L.N.E.R. tug, the author based his figures on information supplied by the builders and has since been advised that these figures were incorrect.

The question of astern steering of single-screw vessels is vastly complicated. The author has made observations on many tugs in this respect and there are many factors affecting the results obtained. He would be interested to receive the figures to which Mr. Riddell refers.

The author has no experience of a reaction rudder on a tug, though he intends to experiment on these lines as opportunity arises, and he believes from observations already made that an improvement in astern steering will be one of the secondary advantages obtainable with this type of rudder. Regarding vulnerability, the author can only state that the owners of one nozzle tug (whose name has been transmitted to Mr. Riddell) confirm that the nozzle appears to attract driftwood and that this can cause serious trouble.

On the score of cost, it must be borne in mind that the average Thames tug, as opposed to a ship-handling tug, spends her life towing at between 4 and 8 knots, with a certain amount of free running. Not only is the added efficiency of the nozzle when towing close-lashed barges in question, but also, as Mr. Riddell will admit, any increase which may be obtained must necessarily fall off as the speed through the water increases. The author believes the L.N.E.R. tug nozzles for 200 i.h.p. cost about £750, or probably more than 10 per cent. of the cost of the vessel. Whilst this may be justifiable for ship handling, there is reason for considerable doubt in the case of barge-handling vessels.

ADDITIONS TO THE LIBRARY.

Presented by the Publishers.

Practical Marine Diesel Engineering. By Louis R. Ford, M.A. Simmons-Boardman Publishing Co., New York, 3rd edition. 590 pages, 307 illus., $5.00 net.

Intended as a handbook for American motorship engineers and particularly for those interested in the standard vessels created by the U.S. Maritime Commission, this handy volume (9½in. by 6in.) is a very welcome addition to the literature of Diesel engineering.

The opening chapters on fundamental principles are clearly and simply written and, far from frightening the reader off the subject, arouse interest and pave the way for a deeper appreciation of his job.

The subdivision of the subject matter is intelligent and follows the lines of principle, general description with diagrammatic sketches and particular description with working drawings.

The chapters on indirect drive systems, Diesel tugs, supercharging, and shipyard repairs are particularly informative.

The print is large and clear and misprints are almost entirely absent. The schematic diagrams are a feature of this work, being clear and free from confusing extraneous matter. Photographs, so often a meaningless blur and waste of valuable space, are well produced and carefully chosen.

The chapter on obtaining a license is of particular interest to British marine engineers, and it may be mentioned that a candidate for an original license has to pass an oral examination on ship sanitation and first-aid and a medical examination.

The technical Engish in this volume is a model of clearness and lucidity. It is concise without being terse and although some of the words employed are American, such as "schematic" for "diagrammatic" and "spray" for "blast", the meaning is quite clear.

At the present time, when our relations with the United States are closer than ever and when British engineers are called upon to man American-built vessels, this book is of special interest.

Engineering Questions and Answers, Vol. 3. Emmott & Co., Ltd., 176 pp., illus., 6s. net.

This third volume of engineering questions and answers continues the record in permanent form of selected answers from the Questions and Answers section of the "Mechanical World". This section has been an interesting feature of the "Mechanical World" for many years.

The value of the collection lies in the fact that the questions which are answered all arose originally out of the difficulties and problems encountered by engineers in the course of their work.

Prospective readers, no matter in what branch of engineering their special interest lies, can feel assured that the book will provide them with a great deal of interesting and instructive reading in a form that is intriguing and relatively light.

Merchant Ship Construction. By H. J. Pursey. Brown, Son & Ferguson, Ltd., 209 pp., copiously illus., 21s. net.

The ability to make descriptive sketches is of great value when discussing structural details and principles of construction, and Mr. H. J. Pursey has made very effective use of perspective in the planning and presentation of his book "Merchant Ship Construction". Instead of the ordinary plan and sectional views with their puzzling combinations of drawn and dotted lines, Mr. Pursey shows plates, stiffeners, stringers and all the dozen and one details of a steel ship's structure by means of well-drawn perspective diagrams. Facing the diagrams are short and direct explanations of the various elements illustrated, together with references which are helpful in enabling the drawings and scantlings to be linked up with Lloyd's Rules.

Short sections of the book are devoted to discussing classification, the "Numerals" used by Lloyd's Register in fixing scantlings, periodical surveys and so on; other sections to welding and riveting, general and particular types of steel ship—regarded from the structural point of view; also to launching and shipyard practice. A good deal of attention is devoted to the construction of oil tankers.

The book is splendidly produced, the line drawings standing out very clearly, and the text being arranged so that notes may be added if desired. The volume should be of great assistance to marine engineers as well as deck officers, while students should find it of especial value. It will also be welcomed as a book of reference by superintendents and those having to do with ships in a shore capacity.

The book is priced 21s. and can rightly be regarded as a sound purchase as a "course" from the pen of an expert lecturer. Mr. H. J. Pursey holds an Extra Master's Certificate, and is Lecturer in Ship Construction to the Department of Navigation, University College, Southampton.

ELECTION OF MEMBERS.

List of those elected by the Council during the period 2nd June to 27th July, 1942.

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Student.

Harry Kay.

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Albert Ridings.

Transfer from Associate to Member.

John William Helps.
Frederick Stephen Parker.
Cyril Oliver Tabbitt.

Transfer from Graduate to Associate Member.

Donald Charles Chapman.
Thomas Lea.

Abstracts of the Technical Press

American Scheme for Aircraft Propulsion by Steam Turbine.

Four different arrangements of power units are dealt with in recently published U.S. patent acquired by the United Aircraft Corporation, one of the patentees being the world-famous aircraft designer Igor Sikorsky. The declared objects are to conserve engine power by the utilisation of waste heat, to improve engine cooling, to enhance the aerodynamic characteristics of the wing sections and also include a scheme for a V-type, liquid-cooled engine, a method of returning the power generated by the waste heat to the main engine, and the use of auxiliary power to drive an electric generator for cabin heating. The engine is enclosed in a cowling and the cooling air is drawn in through the central forward aperture by the suction of a blower. The air passes between the cylinders to remove excess heat and is led into a chamber formed by the leading edge of the wing, where it may serve to impart heat and prevent ice formation. The blower is driven by an exhaust-gas turbine and a steam turbine operating on a common shaft, the gas turbine being connected directly to the exhaust conduit of the engine, whilst the steam turbine is driven by steam supplied from a boiler heated by the efflux of the gas turbine. Steam leaving the turbine is passed through a condenser and the condensate is returned by a pump to a feed tank or directly to the boiler. From the leading edge chamber the blower delivers the air to a mid-wing chamber, where it is forced through the condenser into an internal duct, which communicates with a spanwise slot located at the rear of the upper surface of the wing. It is claimed that such an arrangement converts practically all the heat of the exhaust gas into mechanical energy which is utilised for cooling the engine and increasing the aerodynamic efficiency of the wing. A jet of air forced at high velocity from the rearwardly directed spanwise slot adds its energy to the air flowing along the rear portion of the upper wing surface. By raising the velocity of this airflow, the area of diminished pressure may be increased, and any tendency of the airflow to break away from the surface near the trailing edge may be materially delayed. The proportion of the potential energy of the fuel supplied to the engine which is converted into useful work may, it is suggested, be greatly increased, thus permitting a substantial improvement in the ratio of the size and weight of the power plant to the speed and load-carrying capacity of the aircraft. As specific fuel consumption is generally high, any improvement in efficiency is to be welcomed.—*"Flight", Vol. XLI, No. 1,740, 30th April, 1942, pp. 422-423.*

New Ideas on Aircraft Propulsion.

It is reported that the Breguet Aircraft Works at Toulouse are constructing an experimental jet-propelled machine to the designs of the well-known French engineer R. Leduc. It is said to employ an air compressor unit driven by a steam turbine of the VUIA type running at 3,000 r.p.m. with a steam pressure of 1,910lb./in.^{2} and an estimated output of 1,200 h.p. Experiments, presumably on the test bed, are claimed to have given satisfactory results, but no details are disclosed of the boiler or the necessary condenser plant. The steam system would obviously have to operate on a closed cycle, and this raises problems which are not easy of solution in an aircraft installation. The condenser would, in all probability, be placed in the main air stream so that the heat transferred from the steam would be usefully absorbed for the propulsive jet. According to present-day standards it would not seem possible for the rate of fuel consumption to be as low as that of an I.C. engine doing the same work. The projected speed of the new machine is estimated to be over 310 m.p.h. Leduc's earlier designs were all of the compressorless type in which a divergent duct was used to raise the pressure of the "relative wind" admitted when the aircraft was in motion. Fuel was introduced through a comparatively large number of nozzles disposed across the main stream, so that the combustion of the fuel would therefore be relatively inefficient owing to the low degree of compression attainable, and consumption might be expected to be inordinately high. Such a compressorless scheme of propulsion, although ingenious, appeared to be based on the somewhat impracticable idea of "getting something for nothing". Some form of mechanically-driven compressor is clearly necessary, and M. Leduc's latest project appears to confirm this view.—*"Flight", Vol. XLI, No. 1,740, 30th April, 1942, p. 422.*

Maintenance of Norwegian Motorships.

About half of the four million tons of Norwegian shipping under Allied control is equipped with Diesel machinery. Although most of the ships concerned are comparatively new, the problem of upkeep, maintenance and the supply of spare parts is of vital importance for the successful and continuous operation of the Norwegian mercantile fleet, and as a considerable proportion of the vessels trade to or frequently call at American ports, it was decided to arrange, not only for the repairs, where possible, to be carried out there, but also for the supply of spare parts to be maintained from America. Plans were therefore made to manufacture spare parts for B. & W., M.A.N. and other types of Diesel engines not normally built in America. New York, Boston and Baltimore are the chief repair ports for Norwegian ships at the present time. Spare parts are manufactured there, and the vessels are also equipped with degaussing gear and other defensive appliances by the yards located at these ports. It is stated that between four to five million dollars have been paid to shipyards and engine works on the U.S. Atlantic coast alone in the past three months for work done for the Norwegian mercantile fleet.—*"The Motor Ship", Vol. XXIII, No. 267, April, 1942, p. 5.*

Standard Ship Fittings.

It is reported that some British shipyards are still specifying numerous different designs of ship fittings, which require special patterns, although it is of the utmost importance at the present time to obtain quick delivery of all such fittings. Standard type fittings, manufactured in bulk, and delivered from stock, accelerate the completion of new ships, and there is usually little difference between these products and those of the preferred types. Deck castings, such as bollards, fairleads, mooring pipes, etc., when cast from standard patterns, effect a saving in material and labour. Tables have been compiled showing the required size of fittings for ships of different dimensions. These tables are based on the requirements of the various classifification societies for the size of hawsers, warps and cables, and they show the required thicknesses of metal and the number of bolts or rivets, so that it is merely necessary for the shipbuilders to quote a number or letter from the table to place an order for the particular fitting required. It is suggested that this system of standardisation and tabulation be extended to various plumbers' fittings, such as valves, bulkheads, W.T. connections, air- and sounding-pipe covers, ballast and deep tank fittings, etc.—*"Shipping", Vol. XXX, No. 357, April, 1942, p. 34.*

Alternating Current for Ships' Auxiliaries.

Over 50 motorships building in America are to have their auxiliary machinery driven by a.c. motors, and 105 tankers out of 128 on order for the U.S. Maritime Commission will likewise utilise the a.c. system. Many vessels now under construction in European yards are being similarly equipped. It is true that in a large number of cases the ships concerned are also designed for electric propulsion, but it is not on this account that a.c. is to be used for the motors driving the auxiliary machinery, as the current for this purpose is derived from independent generators which might equally well have been of the d.c. type, were this considered more economic. In the 105 American tankers the electrical auxiliary equipment is large, and power is derived from two 500-kVA. 400-volt 60-cycle alternators, although the cargo oil pump motors can also be supplied by the main propulsion generator through 2,300/440-volt transformers with frequencies variable from 48 to 60 cycles. Apart from the gyro compass and sounding machine, all the motors are supplied with alternating current, and with the exception of that for the shaft-turning gear, are of the squirrel-cage type. The shaft-turning-gear motor is of the wound-rotor design. The large main cargo pump motors, as well as the main circulating-water pump motors, are equipped with compensator-type starters, whilst the remainder of the motors are arranged for "across the line" starting and draw starting currents approximately 700 per cent. greater than the current required under full-load running conditions. All the electric motors, excluding those for the turning gear, oil-fuel transfer pump, oil-fuel service pump and forced-draught fan, are of the constant-speed type designed to run at 440 volts, three-phase 60 cycles. The speed of the above pump and fan motors is varied by means of resistances in series with insulated squirrel-cage windings. In all, each ship is equipped with 50 auxiliary motors ranging in power from ½ h.p. to 200 h.p. In order to obtain the full advantage of a.c. motors, it is, of course, necessary to make use of the squirrel-cage design for direct starting. Such motors are now being produced in this country, and their simplicity and reliability, as well as their safety from the aspect of electric shock compared with d.c. machines, to say nothing of the absence of vulnerability to moisture and condensation, are obvious advantages in favour of a.c. drive. The starting current taken by a modern squirrel-cage motor is only about four times the full-load value, instead of the 10 or 20 times which which was common some years ago and made this design unsuitable for shipboard work. Alternating-current generators, control gear and motors are also considerably less expensive to manufacture than d.c. machines and equipment, and they can be produced more rapidly. The employment of a.c. motors for driving ships' auxiliaries could be accompanied by a rise in the temperatures in the electric cables, as compared with those allowed under the present regulations. If higher temperature rises were permitted in the cables, the weight and size of the electrical plant could be reduced, and the number of man hours required for its manufacture correspondingly diminished. Such higher temperature rises should now be permissible because so-called Class B (inorganic) materials have become available in forms suitable for the insulation of the various parts of all the electrical machinery on board ship. Until recently the only material of this class obtainable for insulating small conductors was asbestos, which is hygroscopic and not altogether suitable for marine work. New possibilities are, however, opened up by the development of glass fibre insulation, which is capable of application to round conductors in the form of yarn, and can be used for insulating purposes in numerous situations where it would not be satisfactory to utilise asbestos. Many marine electrical engineers consider that the temperature rises now permitted by Lloyd's Rules when using Class B insulating material are unnecessarily low, particularly for a.c. motors. There is little likelihood of any sustained overload on the motors driving auxiliary machinery on board ship, so that there should be no objection to rating all the motors, generators and electrical equipment on a continuous maximum rated basis. Hence the temperature rises for Class B material as laid down in *British Standards Specification No. 206* should be permitted even for ships trading in tropical waters so long as adequate ventilation of the machinery spaces is provided. The acceptance of this principle would enable the weight, size and cost of electrical machinery in ships to be reduced.—*"The Motor Ship", No. XXIII, No. 267, April, 1942, p. 10.*

5-kW. Generator for M.T.B.s.

The electrical equipment of most American motor torpedo-boats includes a Lawrence auxiliary generator of 5 kW. capacity driven by a twin-cylinder horizontal air-cooled petrol engine of about 488 c.c. The total weight of the entire unit is only 200lb. and the rating of 5kW. is its constant output at 4,000 r.p.m. The dynamo generates current at 28·5 volts and there is a regulator for maintaining the pressure within 2 per cent. of the rated voltage. The engine cylinders are of chrome molybdenum steel with aluminium alloy heads and cooling fins, each complete cylinder weighing 6¼lb. The temperature of the oil in the crankcase is maintained automatically at 120° F. The engine is controlled by a spring-loaded centrifugal governor and an automatic mixture control is provided on the carburettor. There are two magnetos. The engine can be started up by motoring the dynamo, and there is a commutator brush device to minimise arcing and W/T interference. The cylinders are cooled by air drawn over the fins of the E.R. ventilating fans. The space taken up by the entire unit is just under 33in. by 24in. with a height of 15in.—*"The Motor Boat", Vol. LXXV, No. 1,894, April, 1942, p. 92.*

Arrangement of Auxiliary Machinery.

The relative importance of engine-room auxiliaries not only in turbine vessels and motorships, but also in oil- or coal-burning tramp steamers, is far greater to-day than was the case twenty years ago. Despite differences of opinion regarding the types of these auxiliaries, which are mainly pumps for one purpose or another, a certain degree of uniformity has now been attained, so that rotary pumps, whether of the centrifugal, turbine or vane type, are employed for the circulating-water, feed-water and certain oil services, whilst the bilge, ballast and general service pumps are generally of the plunger type, rotary pumps not being suitable for draining a compartment. The great advantage of the rotary type of pump is, of course, the small number of its working parts and the simplicity of its construction. The arrangement of the E.R. auxiliaries should provide adequate accessibility for overhaul and operation, and their bases should be at a convenient height above the E.R. floor plates, the fewer the number of ladders that have to be climbed and descended the better. Fittings such as filters and coolers should not be placed below the platform, for "out of sight, out of mind". The lead of the various pipes in the engine room should be made to conform to a system, and crossings and changes of direction should be avoided. Many piping arrangements could be simplified and the engineers would be saved much trouble if, in addition, pipes were painted in different colours as a means of identification. With the ever-growing complexity of modern engine rooms, a little care in the design stage may increase the efficiency of the machinery by reducing the work of the ship's engineers.—*"Fairplay", Vol. CLVIII, No. 3,074, 9th April, 1942, pp. 438-439.*

Ship Propulsion by Combustion Turbine.

The ostensible advantages to be derived from the adoption of internal-combustion turbines for ship propulsion have recently been explained by R. Schmid, of Brown, Boveri & Co. The relative thermal efficiencies of the usual types of marine propelling machinery of the present day are approximately as follows:—

Reciprocating steam engines ...	16-17 per cent.
Turbines	18-23 „ „
Diesel engines	33-38 „ „

Recent improvements in the design of axial turbine-driven compressors, operating with turbines supplied with gas at an inlet temperature of 1,100° F., should render it possible for machinery of this type to compete with all the above classes of plant. The simplest form of marine installation is shown diagrammatically in Fig. 1. Atmospheric air is supplied to the axial compressor (*1*) which is driven by the turbine (*2*). Air from the compressor passes through the air preheater (*6*) and the combustion chamber (*5*), into which oil is injected. The gaseous products of combustion are delivered to the turbine (*2*) driving the compressor, as well as to the ahead turbine (*3*), which drives the propeller through gearing (*7*). For astern operation there is an astern turbine (*4*) with a power output of about 50 per cent. of the ahead turbine. At normal load, the thermal efficiency of such an installation is about 18-20 per cent., but by employing two-stage compression and combustion (Fig. 2), in conjunction with an air preheater, the thermal efficiency can be increased to 23 per cent., *i.e.*, to a figure equal to that of the best steam-turbine installations. Any bunker oil capable of being used for oil-fired boilers can be utilised in the combustion turbine, and the ability of the latter to compete with Diesel propelling machinery is therefore dependent on the difference in the cost of bunker oil and Diesel oil, having regard to the difference in efficiency between Diesel and combustion-turbine propulsion. The latter class of plant, however, is considerably lighter and takes up far less space.

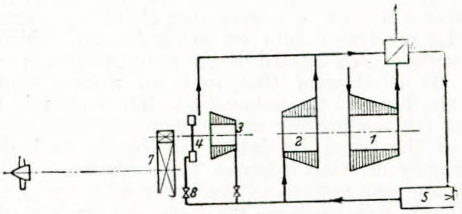


FIG. 1.—*Marine combustion turbine installation. Single-stage compression and combustion with ahead and astern turbines.*
1.—Compressor. 2.—Turbine driving compressor. 3.—Ahead turbine. 4.—Astern turbine. 5.—Combustion chamber. 6.—Air preheater. 7.—Gear. 8.—Reversing valve.

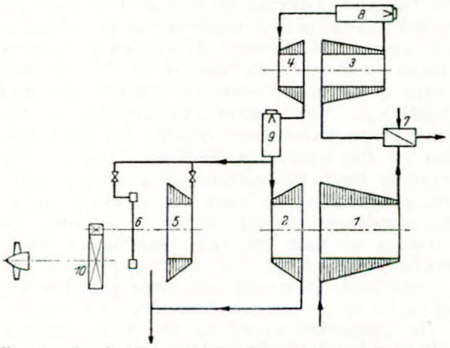


FIG. 2.—*Combustion turbine with two-stage compression and combustion.*
1.—L.P. compressor. 2.—Turbine driving L.P. compressor. 3.—H.P. compressor. 4.—Turbine driving H.P. compressor. 5.—Ahead turbine. 6.—Astern turbine. 7.—Air cooler. 8.—H.P. combustion chamber. 9.—L.P. combustion chamber. 10.—Gear.

The total weight of a two-stage compression and combustion turbine installation of 6,200 b.h.p., designed to drive a 10,000-ton ship at 16 knots, would amount to only 90 tons as against 530 tons for double-acting two-stroke Diesel engines, and although the specific fuel consumption of the latter (0·39lb./s.h.p.-hr.) would be lower than that of the combustion-turbine installation (0·64lb./s.h.p.-hr.) the total machinery and fuel weight would be about the same for a 10,000-mile voyage of some 26 days. The lay-out for a twin-screw combustion-turbine installation of 14,200 b.h.p. is shown in Fig. 3. There are four complete units and each pair of propulsion turbines drives one shaft through reduction gearing. Only one unit of each pair incorporates an astern turbine. The approximate thermal efficiency at a fuel consumption of 0·64lb./s.h.p.-hr. would be 21·8 per cent. The lay-out includes an exhaust-gas boiler installation with an output of about 4,000lb. of saturated steam at 70 to 100lb./in.^{2} pressure per 1,000 b.h.p. of the combustion turbines. This steam reduced to 0·64lb./b.h.p.-hr., which is equivalent to the best results obtained with high-pressure steam installations. A similar fuel consumption figure could be attained by employing air preheaters instead of exhaust-gas boilers, the exhaust gases from the L.P. turbine being used to heat the air delivered by the H.P. compressors. In such an installation the auxiliary power would have to be generated by Diesel-driven dynamos and no steam plant whatever would be employed. The necessary auxiliary machinery for the propelling installation and the total power involved is much smaller with combustion turbines than with modern steam turbine installations, so that the generating plant can be designed for a lower output. Moreover, there is the possibility of arranging for the H.P. compressor, with its practically constant speed of rotation, to drive a generator, in which case the sole function of the auxiliary generators would be the provision of the necessary current in port and for starting the combustion turbines. The simplicity and accessibility

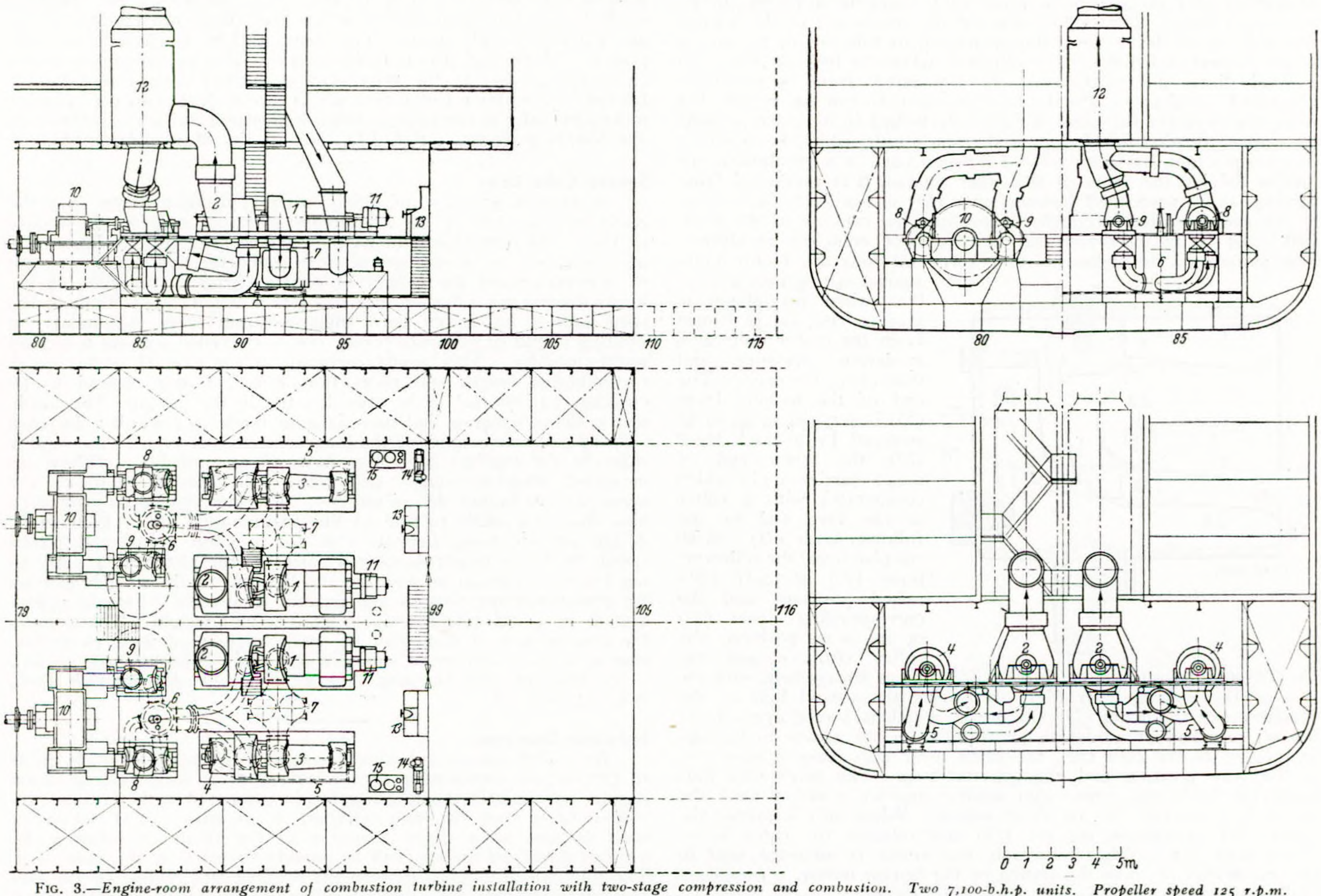
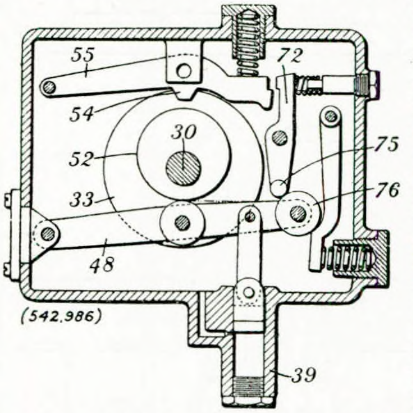


FIG. 3.—*Engine-room arrangement of combustion turbine installation with two-stage compression and combustion. Two 7,100-b.h.p. units. Propeller speed 125 r.p.m. 1.—L.P. compressor. 2.—Turbine driving L.P. compressor. 3.—H.P. compressor. 4.—Turbine driving H.P. compressor. 5.—H.P. combustion chamber. 6.—L.P. combustion chamber. 7.—Air cooler. 8.—Propelling turbine with astern turbine incorporated. 9.—Propelling turbine. 10.—Reduction gear. 11.—Starting motor. 12.—Exhaust-gas boiler. 13.—Control platforms and switchboard. 14.—Fuel pump. 15.—Fuel heater. Distance between frames 109 and 116 represents saving in space compared with a steam-turbine installation. Space between frames 99 and 109 reserved for auxiliary machinery.*

may be supplied to turbo dynamos, the corresponding output being 95-100 h.p. per 1,000 b.h.p. propulsive power, or the steam may be utilised in an additional turbine driving the propeller shaft through gearing, thus adding to the propelling power. The two arrangements may be employed simultaneously, in which case the steam not utilised in the turbo dynamo may be used in a propelling turbine. The regulation of this portion of the machinery installation can be effected automatically in a simple manner, the oil-regulated nozzle valves of the propelling turbine being opened or closed as necessary, according to the steam pressure. The exhaust-gas boilers could also be equipped with oil burners for use when the propelling machinery is not running, or when the demand for steam for auxiliary purposes is high. The employment of steam raised by the exhaust gases enables the thermal efficiency of the entire installation to be increased to 22-23 per cent. When using fuel with a heat value of 18,000 B.Th.U. per lb. the fuel-oil consumption would be of the combustion-turbine plant should ensure reliable operation under all conditions with partly unskilled personnel, and it is claimed that, in comparison with steam turbines or Diesel-engined plant, the E.R. staff is reduced, the replacements needed are fewer and the lubricating-oil consumption is smaller. The set is started up by an electric motor or steam turbine in about 10 minutes, whilst reversing is carried out in a similar manner to that employed with steam turbines.—*"The Motor Ship", Vol. XXIII, No. 269, June, 1942, pp. 90-91.*

Hydraulic Remote-control System.

A hydraulic remote-control device which is especially applicable to ships' telegraphs and similar purposes, is the subject of a recently published British patent. The apparatus comprises a transmitter unit connected by a pair of pipe lines with a pair of motor units arranged in parallel across the latter. The spindle of each motor rotates one step for each revolution of the transmitter-unit shafts. Referring to the accompanying sectional drawing, the casing of the transmitter unit contains an operating shaft (*30*) carrying a pair of ratchet drums (*33*), each free on the shaft, and carrying a pawl which is accommodated in a recess in the drum and engages a ratchet wheel keyed on the operating shaft (*30*). The ratchet drums transmit a drive in opposite directions. The base of the casing houses a pair of cylinders (*39*) connected to the respective pipe lines. The piston in each cylinder is linked to a double follower-lever (*48*) carrying a roller which engages an eccentric cam (*52*) fixed on the corresponding drum (*33*), so that as the latter turns clockwise, one piston is operated, whereas anti-clockwise rotation of the shaft (*30*) reciprocates the other piston. Each complete revolution of the shaft (*30*) corresponds with one step of the motor units, and a lock prevents the direction of rotation of the shaft (*30*) from being reversed except when it is in the zero position determined by a projection (*54*) on a locking lever (*55*) engaging a recess in the associated drum. The sloping sides of the recess and of the projection (*54*) on the lever cause the projection to ride out of the recess as the respective drum rotates, although when the locking lever (*55*) is held down the rotation of its associated drum is positively prevented. Either of the locking levers (*55*) can be raised, but when one is raised the other is positively locked in its down position by a spring-loaded interlock. If the shaft (*30*), which is shown in its zero position, has been rotated through part of a revolution, one ratchet driving the drum of the other ratchet, it is prevented from turning, as its associated locking lever (*55*) is positively held down by the spring-loaded interlock. Therefore, the rotation of the shaft (*30*) must be in uni-directional units of one complete revolution. The pistons in the cylinders (*39*) force oil into the motor units against spring pressure, so that when the piston is free to rise, oil is forced from the motor units at a moderate pressure and thus raises the piston. The end of the locking lever (*55*) is stepped so as to be engaged by a catch lever (*72*), the lower end of which carries a pin which co-operates with a roller at the free end of the follower-lever (*48*). With the piston and the follower-lever (*48*) in their fully raised positions and the corresponding drum (*33*) in its zero position, the roller (*76*) engages the pin (*75*) and deflects the catch lever (*72*) out of engagement with the step of the locking lever (*55*). If during the second half of the revolution of the drum (*33*), the piston, which is forced upwards by the oil returned at moderate pressure from the motor units, lags in relation to the cam (*52*), the catch lever (*72*) remains operative in the zero position and engages the step. The lever (*55*) thus positively holds the drum (*33*) against further rotation until the piston has reached its top dead centre. When this happens, the roller (*76*) re-engages the pin (*75*) and releases the catch lever. When used for a ship's telegraph, one motor is mounted next to the transmitter to show the setting of the distant motor.—*"Engineering", Vol. 153, No. 3,985, 29th May, 1942, p. 440.*



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The Internal Combustion Turbine.

It may be anticipated that in the immediate post-war period considerable progress will be made in developments affecting the so-called gas or internal-combustion turbine. The restriction on the working temperatures by the materials available for turbine blades and other parts exposed to the hot gases, has hitherto imposed limitations on the thermal efficiency of the gas turbine, but there is reason to believe that recent research work on heat-resisting metals will enable this handicap to be overcome, or at any rate mitigated, and that, at the same time, additional improvements such as increased recovery of heat from the exhaust and better efficiency of the turbine itself and of the air compressor which it drives, will raise the thermal efficiency to a point which will make the I.C. turbine fully competitive with the reciprocating Diesel engine. Even with the best materials likely to be available, the combustion temperatures, so far as they affect the actual turbine, are not expected to be abnormally high, since they are kept down by the excess air. The burning of the fuel may be regarded as a means of heating the air which constitutes the working fluid, rather than as a source of hot burnt gases. These moderate temperatures, together with the fact that the I.C. turbine is not essentially a high-pressure machine, should make it suitable for marine purposes. Other points in its favour are simplicity and lightness, favouring low cost and facilitating maintenance, while its general compactness and the relatively small headroom required to accommodate it should give it a definite advantage over an oil engine of conventional design. Cooling requirements, too, are greatly reduced and simplified, since the air which forms the working fluid performs this function to a very large extent, and there are no moving parts to be cooled or complicated components requiring jacketing. Gearing between the turbine and propeller will, of course, have to be provided, as in the case of a steam turbine or high-speed reciprocating oil engine. No separate supercharger is needed, since this function is performed by the rotary blower or compressor, which is mounted on the turbine shaft in the usual manner. A point of considerable interest is the possibility of obtaining, with not more than one turbine per propeller shaft, larger individual power outputs than those now attainable with an ordinary Diesel engine. The steam turbine is basically a high-power machine, and it is probable that this also applies to the rotary I.C. engine, whereas the reciprocating Diesel's output is definitely limited by the amount of power which can be developed per cylinder, more especially in the case of small high-speed units.—*"Shipbuilding and Shipping Record", Vol. LIX, No. 19, 7th May, 1942, p. 488.*

Square Cube Law.

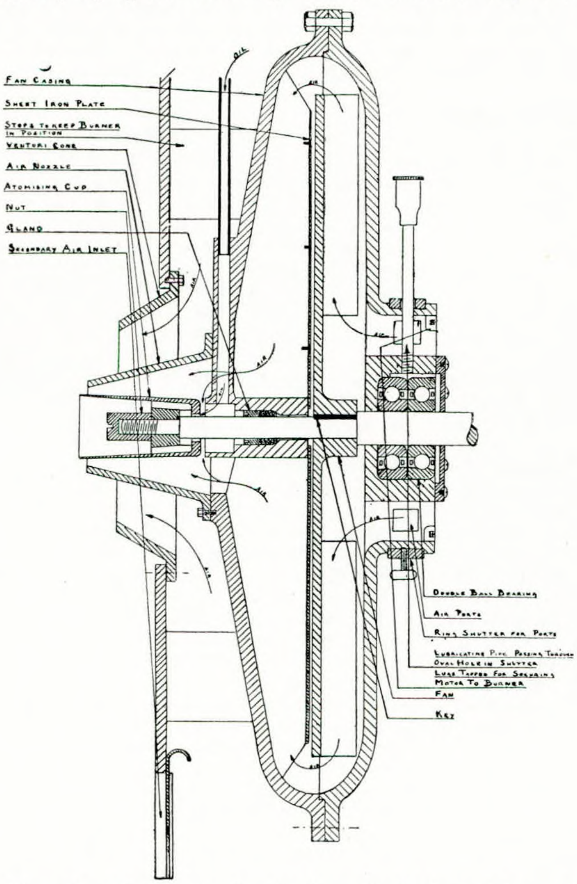
A general principle of fairly wide application states that the load-carrying capacity of a structure increases only as the square of the linear dimensions, or as the surface area, as the dimensions are increased, assuming geometrical similarity and equal strength of material, while the weight of course varies as the cube of the linear dimensions. For bridges this sets a definite upper limit to the span, and it has often been suggested that there is similarly a limiting length of ship which may not be exceeded without a serious weight penalty. This would certainly be the case if ocean waves of unlimited length and proportionate height were bound to be encountered and had to be provided for in the design. The available evidence suggests that there is some tendency towards a decrease of proportionate height for the longer waves, but even so the largest ships in the various categories now afloat probably do show an increased structure-weight percentage, and this in spite of the acceptance of higher design stresses. The law applied to machinery also, more especially to that of the reciprocating type. Elementary design considerations indicate that for a given m.e.p., and piston speed, the horse-power developed varies as the piston area. Allowing for some margin on scantlings in the case of the smaller engines for manufacturing reasons and general robustness, it would appear that there is still definite gain in h.p. per unit weight which favours the employment of a number of small, high-speed units in preference to a single low-speed engine where other considerations permit. —*"Shipbuilding and Shipping Record", Vol. LIX, No. 18, 30th April, 1942, pp. 462-463.*

Indicator Diagrams.

As is well known, indicator diagrams constitute a valuable guide to the correct functioning of the values of a reciprocating steam engine, particularly if each diagram is compared with the one previously taken from the same cylinder. In the case of a Diesel engine such changes as a slight variation in the clearance between the various cams and tappets will be quickly revealed in the same way. The time which elapses between the opening and closing of the fuel valve is much less than that of either the air or the exhaust valve, but by superimposing the draw or pull card on the normal card the apparent period of combustion is greatly magnified. Instead of the drum being operated by the engine in the usual way, at the moment of combustion, the cord is pulled sharply by hand, and thus the combustion line can be spread over the entire length of the card. In this way any irregularities in the combustion process are clearly shown.—*"Shipbuilding and Shipping Record", Vol. LIX, No. 17, 23rd April, 1942, p. 439.*

Rotary Oil Burners for Low-pressure Steam Boilers.

Many factories and industrial establishments in the United States do not use high-pressure steam boilers, and obtain their power supply from the large electric companies. However, they use a good deal of steam for heating and processing, as well as for pumping, etc., and this created a demand for oil-burning equipment of a type intermediate between the standard high-pressure equipment requiring steam pressures of 50lb./in.^{2} and upwards, and the small-capacity domestic type which could not deal with heavy loads. This demand led to the development of the rotary burner which required only gravity feed for atomisation and was capable of an output equal to that of a high-pressure burner. The earliest rotary burners could only utilise light grades of oil of low flash point and viscosity which did not require preheating, but it was subsequently found that the high latent heat of low-pressure steam made it possible to maintain an oil temperature of about 100° F. for the average burner output by means of ordinary heaters. This temperature, applied to oil at a pressure of up to 50lb./in.² supplied by an electric pump, sufficed to produce good atomisation and combustion with the heaviest grades of oil and enabled rotary O.F. burners to be operated with the same grades of fuel as standard high-pressure burners, although they could not compete with the latter when high pressures were obtainable. A typical rotary oil burner may be described as a motor with an extended shaft on which were mounted a spinning cup and fan, the former protruding into the furnace and the latter enclosed in a casing outside the boiler. The burner was mounted on a hinged plate at the furnace front and could be swung outwards when not in use. The general principle of its construction is shown in the accompanying sectional diagram. Oil enters through a vertical



pipe, falling directly on to the horizontal shaft which carries it along into the atomising cup. A screwed gland, threaded to the opposite hand to the direction of rotation, prevents the oil from running back into the fan casing, and it is found in practice that oil does not penetrate the space between the boss and the inner end of the cup, as air enters the cup through this space and facilitates combustion. The base of the cup forms a boss having large slots in its circumference, and these serve to guide the oil along the inside walls of the cup to the rim, whence it is thrown off by centrifugal force in the form of a fine mist or spray which mixes directly with the air entering through the surrounding nozzle. The air supply is regulated by a shutter which opens or closes the air ports to the fan, whilst the oil delivery is controlled by a needle valve in the supply pipe (not shown). A small hole in the bottom of the air nozzle, together with a plug in the base of the fan casing, provides for any oil drip. Air at natural draught is also admitted through the Venturi cone enclosing the annular air nozzle by means of a secondary-air inlet controlled by a shutter below the burner. When starting up, oil is first circulated past an electric heater, through the hollow hinge pin of the burner, until a suitable temperature, depending upon the grade of oil, is attained. The burner motor is then started up, and when this has reached its full speed of rotation, the oil is ignited by a lighted torch held before the cup with the air partially shut off, the oil control valve being gradually opened as ignition takes place, whereupon the full supply of air is given to the flame. The quantities of air and oil admitted to the burner are then regulated to suit requirements and, when steam is raised, the electric heater is shut off and the steam heater put into operation. The relationship between the length and diameter of the flame cannot be altered, as the position of the burner is fixed and it cannot be moved in or out of the furnace. The shape of the flame is governed by the design of the burner and depends upon the clearance between the inner circumference of the annular air nozzle and the atomising cup, as well as on the distance the latter protrudes beyond the air nozzle. The form of the flame is also affected by the design of the cup, which varies with different makers, some cups being cylindrical before flaring off near the end, whilst others are conical. A cup with too thin an edge will often curl right out as if it had been flattened against some object. Some makers run the oil supply right into the cup by means of a small-diameter pipe led through the small opening between the cup and the shaft, but this arrangement is liable to be a source of choking up when heavy oil is used. A coupling is sometimes fitted between the motor and the burner, in order that a defective motor may be exchanged more easily, but the arrangement in which the armature is wound directly on the burner spindle gives a more rigid construction. The entire burner unit can be exchanged at any time by merely removing the hinge pin after disconnecting the oil feed and circulating pipes. Among the disadvantages of the rotary burners was the fact that each burner required a separate motor, and that failure of the latter put the burner out of action. Furthermore, burners had to be made in a variety of sizes for varying capacities, and with motors to suit the particular current and voltage available. This made the system expensive, and where the heaters and electric pumps were run from the same supply, an attempt was made to reduce the cost by driving the pump from the burner motor. Another type was produced in which the burner was rotated by air pressure supplied by a single external fan capable of feeding a battery of burners, the design providing for the maintenance of a constant maximum of air for rotation while permitting that for combustion to be varied. However, such an arrangement was only partially successful, as the best pump speeds were 1,200 r.p.m. or less, whereas the best spinning cup speed was 3,500 r.p.m., so that where the pump was incorporated in the burner, gearing had to be employed and the motor made powerful enough for the extra load. The use of a single large fan to feed all the burners showed little saving as it was essential to provide an additional fan as a stand-by. On the other hand, the rotary type of burner had the advantages of providing a ready means of regulating the rate of firing without changing tips, having no tip to clean, giving better air distribution and being simpler to operate than the high-pressure type of burner. Choking of the rotary burner at the control valve could be relieved by temporarily opening it out a little, and it was frequently possible to dispense with O.F. strainers entirely. Improvements in the process of refining oil fuel enabled the rotary type of burner to handle the lighter grades better than the other types of burners, since there were no contracted atomising grooves or tip orifices to consider. The difficulties of burning the heavy type of bunker fuel oil remained very great, however, and the rotary burner really found its highest expression with the lighter grades of oil which did not require preheating. Many rotary burner installations were arranged for fully automatic operation, the burners being shut off due to excessive temperature, too high a steam pressure, too low a water level or failure of oil supply and, when these conditions rectified themselves, the burners were relighted automatically, after clearing the furnace of fumes, by means of a self-igniting gas jet. The attention of the operator was only required in the event of a second failure of a burner to relight itself and resume operation. The CO_2 readings and boiler efficiencies obtainable with the rotary type of burner rivalled those attained with standard high-pressure equipment, and its performance improved with the better grading of oil, but attempts to produce a rotary burner capable of burning the heaviest grade of fuel with fully automatic control have not proved successful, and the rotary burner remains to-day a simple medium for burning any kind of oil without the elaborate installation and skilled attention necessary with the high-pressure type.—*W. E. Bruce, "Industrial Power", Vol. XVIII, No. 199, May, 1942, pp. 40-41 and 44.*

Reclaiming Worn Pump Parts.

The presence of suspended abrasive matter in the liquids handled by oil-fuel, brine or bilge pumps, often causes serious wear and corrosion of the impellers, seal rings and shafts, with the result that these parts may have to be reconditioned long before their usual period of overhaul. Experience however indicates that much of this replacement can be obviated by the application of a hard protective coating to the surfaces exposed to maximum wear, whilst the hard-facing of new pumps before they are put into service gives them an increased life at a low cost. It has been found that the best material for such hard-facing is a stainless, non-ferrous alloy of cobalt, chromium and tungsten, which possesses high resistance to wear and corrosion and a low coefficient of friction. This alloy is usually applied to a machined steel sleeve by the oxy-acetylene process, and it flows on readily in a relatively thin layer, *i.e.*, $\frac{1}{16}$in. to $\frac{1}{8}$in. To illustrate the economies achieved by hard-facing pumps in this manner, a few examples may be cited. The shaft of a main bilge pump, fitted with hard-faced sleeves to obviate the necessity for frequent renewal and constant repacking of glands, ran for three years. During this time, occasional tightening of the glands sufficed to keep the pump in good condition. Another bilge pump shaft, hard-faced in the same manner, showed a reduction in diameter at the gland of less than $\frac{1}{16}$in. after four years' running, although it had formerly proved difficult to keep the gland properly packed due to the excessive wear experienced. After installing the hard-faced shaft it was only found necessary to renew the packing once in the four years. Being of the open-impeller type, this pump also lost its efficiency rapidly due to rounding of the corners of the blades. A small bead of similar hard-facing alloy deposited along the edges and ground to shape solved this problem, and the impeller blades showed no noticeable wear after two years. It is of great importance that the hard-facing operation should be carried out with the utmost care, and that the finish of the hard-faced work should be such as to ensure a smooth and accurate surface. Furthermore, proper preparation of the work facilitates the application of the hard-facing material and affects the quantity used. The welding operator should keep in mind that any metal deposited in excess of that needed will have to be ground off, thereby increasing the cost of both welding and finishing. The finish of the prepared steel part must be smooth, and all sharp corners must be rounded off to offset any tendency of the parent metal to alloy with the hard-facing material and result in a softer, less wear-resistant deposit. The surface of the work surface must also be quite free from oil, dirt or scale. Formerly, hard-facing material was applied directly to a shaft, but this practice has been discontinued to avoid the possibility of the shaft warping. The removable sleeve method is preferable because the sleeve can be finished before being keyed, screwed or shrunk on the shaft, thus avoiding any risk of warping the latter. The comparatively small amount of metal in the sleeve is easily heated and can be readily hard-faced, thereby reducing the time required for making the deposit. At the same time this method eliminates the risk of cracks because the tubing absorbs any strain set up by the contraction of the hard-facing alloy as it cools. In addition, grinding is made easier because the work is less bulky and more accessible. The ready removal of the shaft sleeves when replacement is needed is an added advantage. After the steel tubing has been hard-faced, ground and machined, it is keyed, shrunk or screwed on to the shaft where wear occurs. Although this procedure is not always applicable, it can generally be utilised either by re-designing the shaft or by boring out the packing gland and pump housing. Under ordinary conditions, a finished layer of 0·05 to 0·06in. thickness of hard-facing material is ample, and any part worn in excess of this amount should first be rebuilt with steel and then machined or ground in preparation for the hard-facing. The inside of the removable shaft sleeve should, in the first instance, be rough-bored to within $\frac{1}{8}$-in. of the finished diameter, after which the outside of the sleeve should be turned for the desired depth of deposit and length of wearing surface, allowing about 3in. on one end for securing in the lathe chuck. The sleeve should also be turned to a diameter slightly smaller than finished size at this end for a length equal to the width of the grinding wheel plus $\frac{1}{4}$in., to obviate any grinding of the steel tubing. Dry grinding of the steel may have a tendency to load the grinding wheel. It is desirable to leave an $1\frac{1}{8}$-in. band of metal at the end opposite the chuck and to turn it to the required finished outside diameter, in order to provide a small shoulder which will facilitate the starting of welding and serve as a guide for the welder by indicating the depth of deposit required. The hard-facing material is applied in the usual manner, special care being taken to keep the flame adjustment constant while welding by maintaining the welding gases at a steady pressure. Reheating is necessary and should be done in a furnace where possible. Welding is started at the end opposite that to be chucked and proceeds in a spiral manner round the sleeve, the entire operation being carried to completion without interruption. After the surface has been coated, the work is placed in a vertical position and protected by asbestos paper, or placed vertically in powdered lime, mica or other loose insulating material to induce slow cooling and to eliminate strain as much as possible. It is of the utmost importance that the hard-facing deposit should be uniform and of just the right amount for finishing, since any attempt to respot the deposit later causes distortion unless extreme care is taken. The use of a simple rotating welding jig for holding the work is beneficial, as it enables the operator to rotate the work at will and leaves both his hands free for welding. In order to grind the sleeve well and in as short a time as possible, a cylindrical grinding machine with a spindle of the ball-bearing type should be used, whilst the best wheels for the purpose are vitrified aluminium oxide with 46 grit or a 24 combination grit. The peripheral wheel speed should not exceed 4,200ft./min., using a 10-in. wheel 1-in. wide. A fast feed with slow turning of the work in the opposite direction to the wheel gives good results. Wet grinding causes less distortion of the work and less dust, but good results can be achieved with dry grinding. In order to prevent excessive vibration of the work while grinding, a wooden plug is inserted into the bushing, in which the tailstock centre can run. When grinding is completed, the sleeve should be finish-bored. If it is to be shrunk on, not more than 0·0005in. per inch of shaft diameter should be allowed for shrinkage. When putting the sleeve on to the shaft, only just enough heat should be applied to ensure it going into place. Heating is preferable to pressing.—*"The Journal of Commerce" (Shipbuilding and Engineering Edition), No. 35,637, 16th April, 1942, p. 1.*

Floating Workshops.

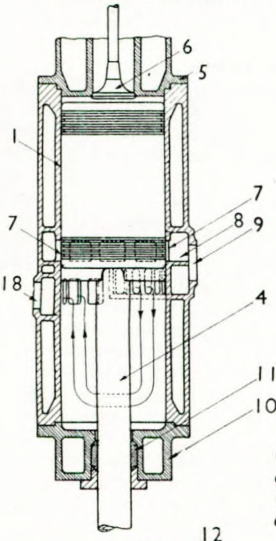
The U.S. Navy is building a number of floating workshops in shipyards which do not deal with naval contracts but normally construct coasters, tugs and yachts. These workshops are vessels of some 1,200 tons displacement and are being given a fair ship form forward to facilitate towing, but will have no propelling machinery. The space that this would occupy is taken up by a large generating installation and an elaborate equipment of lathes, drilling machinery, welding plant, smithies and everything necessary for floating repairs of considerable importance. The machinery has cost about $200,000 for each ship, which means that they are well equipped. The functions of these useful little mobile workshops will, of course, be distinct from those of the large ocean-going fleet repair ships already possessed by the American Navy.—*"Shipbuilding and Shipping Record", Vol. LIX, No. 17, 23rd April, 1942, p. 439.*

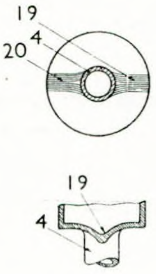
New Pressure-charged Marine Engine.

The National Gas and Oil Engine Co., Ltd., are developing a new series of medium-speed marine Diesel engines with pressure charging which is claimed to represent the last word in engines of this class. The first unit of the new series to be completed is a 6-cylr. engine rated at 540/600 b.h.p. at 600 r.p.m. and driving the propeller at 300 r.p.m. through an SLM-type oil-operated reverse-reduction gearbox. The cylinders are 10in. in diameter and the stroke is 13in., the piston speed thus being 1,300ft./min., whilst the brake m.e.p. at maximum output is 128lb./in.^{2}. The general design of the engine follows the makers' current practice in being of the open combustion chamber type. The main bearings are water cooled, so that no oil cooler is required. Sea-water cooling is normally utilised for the liners and cylinder heads, although F.W. cooling may be arranged for if desired. A reciprocating type circulating-water pump is driven by a cross shaft at the forward end of the engine, this pump being balanced by a bilge pump which is interchangeable through suitable cross connections. Uncooled C.I. pistons are used, and H-section connecting rods are employed instead of the round type hitherto favoured for National engines of approximately this size. A new feature of the design is the provision of a camshaft at almost the level of the liner flange instead of at a level near the bottom of the liner. This modification enables the length of the push rods to be reduced and decreases the inertia effects of the valve gear, in addition to which it renders the injection pumps more readily accessible for inspection and adjustment. A Büchi turbo-charging set is installed at the forward end of the engine just above the engine-driven pump group. The blower speed is about 17,000 r.p.m. and an air silencer is provided. A starting air compressor is driven through a clutch at the after end of the primary shaft of the reverse-reduction gear. The main engine controls are arranged at the after end, the left-hand wheel being for starting and stopping the unit, whilst the right-hand one controls the speed and reversing of the propeller. If desired, bridge control of engine speed and reversing can be provided through suitable bevel gearing and universally-joined shafts. The full-load fuel consumption of the new engine on the test bed was under 0·40lb./b.h.p.-hr., the curve being remarkably flat down to almost half load. The exhaust-gas temperature at the cylr. outlets was between 750° and 800° F. at full load, while just in front of the turbine it was 1,000° F. The pressure of the air delivered to the inlet valves was 8 to 9in. of mercury at full load.—*"Gas and Oil Power", Vol. XXXVII, No. 439, April, 1942, p. 75.*

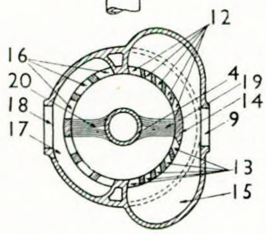
A Götaverken Double-acting Engine.

A British patent was recently granted to the A.B. Götaverken, Gothenburg, on an improved design of double-acting two-stroke Diesel engine with uniflow scavenging in the top portion of the cylinder and return-flow port scavenging at the bottom. Referring to the accompanying illustrations, the two larger diagrams show the cylinder in sectional elevation, while the smaller diagrams are of the bottom end of the piston and include part of the piston rod. In the upper part of the cylinder (*1*) the cover (*5*) is provided with an exhaust valve (*6*). The scavenging-air ports (*7*) are arranged all round the periphery of the cylinder and are in communication with an annular passage or belt (*8*). Air for both ends of the cylinder is admitted through an inlet (*9*). The bottom cylinder cover (*10*) has a stuffing box (*11*) for the piston rod (*4*). There are two groups of air inlet ports for this end of the cylinder. These groups (*12, 13*) are shown in plan, from which it will be seen that the ports are directed convergently towards each other. Between the two groups of ports there is a solid portion (*14*). The admission of air for the bottom end takes place through a receiver (*15*) which branches from the air intake (*9*). On the opposite side of the cylinder is a series of exhaust ports (*16*) which are in communication with a belt (*17*) and an exhaust outlet (*18*). The bottom of the piston is shaped to form an air guide (*19, 20*), so that when the scavenging ports are uncovered the air is directed axially downwards. The lower cylinder cover deflects the air upwards, and consequently through the exhaust ports and the outlet (*18*), the path followed by the air being indicated by lines and arrows in the sectional elevation of the cylinder.—*"The Motor Ship", Vol. XXIII, No. 269, June, 1942, p. 73.*





Cylinder and piston diagrams of the Götaverken double-acting engine.



Improved Design of Götaverken Diesel Engine.

The Swedish motor tanker "Zelos", which recently completed her sea trials, was built and engined by the A.B. Götaverken, Gothenburg, and is a single-screw vessel of 483ft. 9in. o.a. length, with a beam of 59ft., a depth of 35ft. 6in., and a carrying capacity of 12,730 tons d.w. on a draught of 27ft. 6in. The propelling machinery, which is of Götaverken design, and embodies a number of improvements, consists of a 6-cylr. single-acting two-stroke engine developing 5,200 i.h.p. at 112 r.p.m. and is designed to give the ship a service speed of 13 knots in a loaded condition. The engine differs from earlier ones of the Götaverken type in not being provided with the large scavenge pump which used to be located at the forward end of the engine and was driven directly by the crankshaft, whilst the scavenging-air chamber along the engine and the large air intakes mounted on the scavenge pump have also been dispensed with. These changes have been made possible by utilising the lower part of the cylinders as scavenge pumps. The cylinders have also been fitted with exhaust valves for the scavenge ports operated from the valve gear. According to the *Svensk Sjöfartstidning*, these improvements have enabled the weight of the engine to be reduced by about 28 tons, in addition to which its length has been shortened by approximately 5ft., whilst the mechanical efficiency has been increased by about 1 per cent.—*"Lloyd's List and Shipping Gazette", No. 39,789, 27th May, 1942, p. 5.*

Manning the Merchant Navy.

Credit is due to the Ministry of War Transport and its predecessor, as well as to the men themselves, for the way in which the manning problems of the Merchant Navy have been satisfactorily met. After 2½ years of total warfare and the sinking of about a third of the tonnage under the British flag, there has never been a serious shortage of officers and men, although it is common knowledge that early in the war there was a shortage of junior certificated engineers, which sometimes delayed the sailing of ships. The regulations were wisely relaxed so as to enable men with sufficient practical experience and good records to discharge duties of greater responsibility than would formerly have been possible owing to insufficient service or failure to pass examinations. In such cases temporary permits were issued for service under prescribed conditions. It is understood that about 500 of these permits have been issued to date and the problem has virtually been solved. Other measures adopted to achieve elasticity included an arrangement under which officers and ratings in reserve might serve, if required, one rank below their previous rank—with a corresponding reduction in pay. In May, 1941, men formerly employed at sea had to register, and nearly 60,000 names were thereby added to the Reserve Pool. About one-third of these men have been approached and some 7,000 of them have already returned to service at sea. Shore employment is now being drawn on at a rate of over 100 a week, chiefly seamen and firemen. These men have responded very well indeed, and there is, of course, a considerable reserve still available. At the same time, the rate of entry at the bottom has been accelerated as far as possible, and what is known in industry as up-grading, has been encouraged. The first school in the country for training firemen for service in merchant ships was opened in Cardiff last week under the auspices of the Shipping Federation.—*"The Shipping World", Vol. CVI, No. 2,549, 22nd April, 1942, pp. 294 and 296.*

"Refrigerating" Old Ships.

According to a recent survey of the Carrier Corporation, practically every merchant ship building in the U.S.A. is having part of its cargo space adapted for the carriage of refrigerated cargo, in addition to which many older vessels are now being equipped with refrigerating plant. Amongst such older ships dealt with by the Carrier Corporation is the Mexican motor vessel "Ensenada", which is now engaged in the transport of frozen foodstuffs and may be regarded as a typical example of such a conversion. In order to transport frozen foodstuffs successfully, the temperature in the cargo hold must be held at 0° F. In the case of the "Ensenada", this involved lining the hold with 6in. of insulation and dividing it into three compartments by insulated bulkheads or partitions. Two "cold diffuser" units are employed for low-temperature air conditioning. These units are installed just forward of the bulkhead separating the refrigerated hold from the engine room and are connected by air ducts to all three cold-storage spaces, any or all of which can be supplied by either or both diffusers. Refrigeration is supplied by two "Freon-12" compressors driven by Diesel engines which also drive the all-bronze S.W. circulating pumps for the two shell-and-tube, vertical, multi-pass condensers. To provide for emergencies, each system consisting of one cold diffuser and one refrigerating plant, is arranged as a single unit with only one cross-connection, the latter being fitted to permit a transfer of the refrigerant charge from one system to the other. The "Ensenada" normally carries a refrigerant charge of 300lb. of "Freon-12" divided equally between the two compressors. Either system has sufficient capacity to maintain the insulated-hold temperature of 0° F. when the ship is loaded with frozen food products.—*"Modern Refrigeration", Vol. XLV, No. 529, April, 1942, p. 70.*

Roughened Hull Surface.

The paper describes resistance and screw experiments on models with both smooth and rough surfaces, and the author suggests a method of extrapolating the results to ships with allowance for time out of dock. Warship trials have shown that skin-friction resistance increases at an average rate of about 0·5 per cent. per day in warm waters and 0·25 per cent. per day in temperate waters, although much depends on the conditions of service. The author shows that Prandtl's grain-size coefficients are inconsistent with deductions from speed trials, the character of the surface roughness being important. Experiments on models coated with various compositions indicate that the difference in resistance between smoothly painted surfaces is extremely small. The author also investigates and identifies the various marine growths on steel rafts and on ships. Such growth proceeds slowly at first, but may become luxuriant and varied after three months' exposure, depending on the service of the ship. Biological examination shows adaptations to resist the toxic effect of paints. Tests of the model rafts at various periods show that fouling may increase resistance by as much as 0·5 per cent. per day.—*Paper by R. W. L. Gawn, R.C.N.C., read at a general meeting of the N-E. Coast Institution of Engineers and Shipbuilders on the 24th April, 1942.*

Forward Butts.

Some years ago it was suggested abroad that there would be some definite gain in overcoming hull resistance, with the usual type of overlapped butt joint of the strakes of shell plating, by having the butt ends facing forwards, instead of aft, in accordance with the common practice. Experimental evidence in support of this contention was produced, and although the proposed change met with little response at the time, the matter is deserving of re-examination. In the first place, the standard arrangement, dating from the beginning of iron and steel shipbuilding, was almost certainly adopted without any special investigation, the advantage being regarded as sufficiently obvious without any trial of the alternative. More recently faired rearward butts have been tried in an endeavour to reduce their resistance, which is recognised as being appreciable, though it seems doubtful whether a really adequate length of fairing is practicable. On the other hand, the fairing principle might be applied to the forward-facing joint with what seems to be much better prospects of success, since a quarter-round fairing—little more than a full weld-metal fillet—should be amply sufficient in the light of modern knowledge of streamline flow.—*"Shipbuilding and Shipping Record", Vol. LIX, No. 18, 30th April, 1942, p. 463.*

Plastic Materials for Ships.

The possibilities of plastic materials in connection with the present merchant shipbuilding programme are, it is stated, being examined by the authorities. There is an immediate scope for the employment of plastics in place of plywood for cabin bulkheads, doors, and so on, provided the plastic materials are more readily obtainable than the wood, and it is understood that a merchant ship is planned which will have her entire deckhouse made in plastic material. This may never be an economical proposition when wood is readily obtainable, but the properties of the plastics used will have been put to a practical test in a way that might not have been possible if expediency had not assumed a greater importance than economics. Plastic materials can cater for very many needs, but much depends on whether the correct plastic is selected to suit the particular purpose. In an article by A. C. Hardy in the March issue of *"Plastics"*, concerning possible uses for plastics in shipbuilding, the following questions regarding their special characteristics are put forward: *(1)* Are plastics lighter than steel? *(2)* Are they lighter than aluminium alloy? *(3)* Can they be married structurally to light metals or steel? *(4)* What is their strength compared with other materials of construction? *(5)* Are they waveproof, wind-proof and corrosion-proof? *(6)* Are they likely to deteriorate in structures at the point of their "fastenings" to the essential strength hull of the ship? *(7)* Are they easily repairable in the sense that a patch can be welded on a steel plate? *(8)* Can they be made fire-resistant? *(9)* Will they take paint readily? *(10)* From the aspect of practical engineering economics, is it possible to mould, say, one or two large "castings" and then adapt the machinery for other purposes? The most promising field for plastics is where lightness is essential, since the sp. gr. of plastic materials is about half that of aluminium and one-sixth that of steel. Weight for weight, the strongest plastics are practically as strong in tension as dural or steel, while with the increased bulk there should be increased rigidity. It should be remembered, however, that the high cost of plastics makes mere volumetric replacement by such materials impracticable. A recent development is the employment of a plastic material known as Isoflex for the insulation of refrigerated spaces on board ship. The weight of Isoflex is said to be about one-tenth that of slab cork, and the material is built up in layers from corrugated cellulose acetate film. It is claimed to be rot-proof and non-inflammable, and does not disintegrate when subjected to vibration.—*"The Shipping World", Vol. CVI, No. 2,550, 29th April, 1942, p. 315.*

New Westinghouse Recording Appliances.

The research laboratories of the Westinghouse Electric Manufacturing Company have developed a new instrument for the accurate measurement of the performance of turbine governors of all types. The instrument includes a number of recording devices which are placed at strategic points of the control system and relay their information electrically to an oscillograph for simultaneous record. One of these devices is the valve travel recorder which serves to indicate the movement of the valve in terms of a proportional electrical voltage by a magnetic strain gauge. This device gives an accurate record of any valve travel from 0·05in. to 8·0in. Other appliances simultaneously record the several oil pressures in the entire control system, by means of a bellows on another strain gauge which amplifies very small changes in pressure and transmits a proportional change in voltage to the oscillograph. This device is extremely rapid in its response and can record variations in pressure over as little as $\frac{1}{60}$-sec. It has already served to detect the presence of an unknown high-frequency variation in governor oil pressure. The same firm have also developed a controller for timing spot welding. The controls are mounted on hinged insulated panels which, for maintenance, may be swung open by removing a single bolt. The welding operator places plugs in the proper holes in the controller panel to establish an automatic timing sequence.—*"The Nautical Gazette", Vol. 132, No. 4, April, 1942, pp. 43 and 47.*

A Multiple-tappet Arrangement for Fuel Pumps.

The Daimler-Benz Company, Stuttgart were the original owners of a recently published British patent for a multiple-tappet oil-engine fuel pump, the arrangement of which is shown, sectionally, in Fig. 3. The tappets are located side by side in such a manner as to guide each other. The pump casing *(1)* has a transverse bottom *(4)* above the space *(3)* containing the camshaft *(2)*, the pump casing *(1)* and the camshaft housing enclosing the space *(3)* being in one piece. In the bottom *(4)*, holes *(5)* are provided for the reception of the tappets of the pump plungers *(7)*. Each tappet bears upon a cam *(8)* by means of a shaped bottom surface comprising two sloping faces *(6)*. At the sides, the tappets are formed, at diametrically opposite points, with two plane surfaces *(10)*. The holes *(5)* in the transverse bottom are made with a milling cutter of the same diameter *(r)* as the tappets, and are pitched so as to intersect one another in the axial direction of the shaft *(2)*. The amount *(x)* by which the holes intersect is designed to provide broad engaging surfaces *(10)* for the guidance of the tappets, so that the facing surfaces of any two adjoining tappets engage one another and afford free guidance without twisting. The surfaces on the tappets which have to be accurately machined are relatively small, and it is stated that the production of the intersecting cylindrical bores *(5)* is a very simple operation. Should any excessive wear of a single tappet take place, it is only necessary to renew that particular one.—*"The Oil Engine", Vol. X, No. 110, June, 1942, p. 53.*

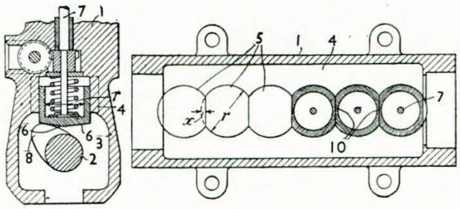


FIG. 3.

Blade Thickness of Wide-bladed Propellers.

Wide-bladed propellers are employed to reduce or eliminate cavitation. Screws of this type are commonly made with ogival sections, elliptical outline and uniform thickness variation with radius. Such propellers are more highly stressed at the root than towards the tip. Thinner sections near the tip would probably result in reduced cavitation and improved efficiency. The purpose of this paper is *(1)* to give a suitable method of evaluating the stresses experienced by a wide-bladed propeller, and *(2)* to investigate how thin the blades of a typical propeller may be made, consistent with the maximum stresses being within safe limits everywhere, bearing in mind that propeller blades are subject to fatigue and erosion. The author points out that the latter need not be troublesome as it is found that the highest stresses in a typical propeller are experienced at the leading edge; whereas erosion always occurs near the trailing edge, where the stresses are lower.—*Paper (No. 4) by N. Hancock, R.C.N.C., prepared for publication in the "Transactions of the Institution of Naval Architects", for 1942.*

Checking Marine Propellers.

For many years past marine propellers were made in such a way that the driving faces of the blades were parts of the helicoidal surfaces of the required pitch. In some cases the faces were machined after casting, this being a relatively simple matter, since the cutting tool was only required to move in a straight line through the axis of the boss along successive generatrices of the helicoidal surface. The tool marks were then removed, and the finished face could be used as a datum from which to check the shape of the back of the blade. With the introduction of varying pitch over the blade, and the adoption of aerofoil sections, with curved driving faces, no machine was available to perform the necessary operations. Two things must be possible in the manufacture of such propellers—the determination of the amount of work to be done on the original casting, and the checking of the finished product. The accompanying drawing shows sections of a propeller blade formed by the intersection of cylinders of different radii all co-axial with the shaft axis—which may be termed, for convenience, cylindrical sections. Templates of $\frac{1}{32}$-in. sheet steel can be made for both the face and back contours of these sections, and by suitable reference data these templates are correctly located on the blade. A typical template of this kind is shown in Fig. 1. The points J and K are on the face pitch line outside the blade limits, whilst the points M and L are on the centre line of the propeller blade. The templates are set to the

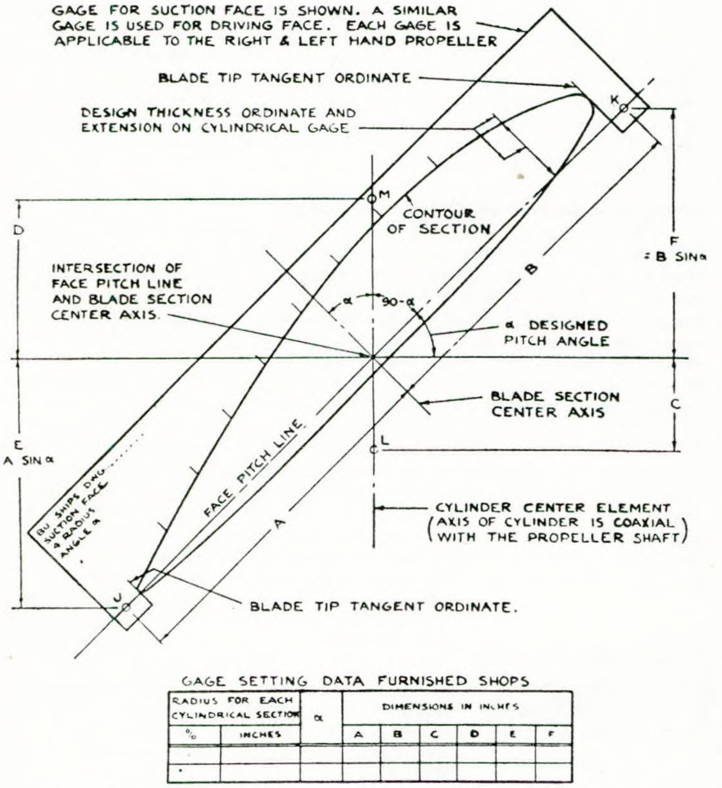


FIG. 1.—*Cylindrical section gauge for checking propellers.*

correct cylindrical curvature, and if each of the face ones in turn is set so that the points J, K and L are correctly located, the surface of the blade will be defined. The back can then be established in the same way. The templates are applied after the boss has been machined and the propeller set up on a face plate. Each template is centred by the line on which M or L is located, the pitch angle being determined by the points K and J, the heights of which have been pre-determined in the manner shown in Fig. 2. If the blade section differs from the designed shape, the template will be above the computed values at the points K and J, and the blade surface will therefore have to be cut into along the section until the correct contour is established. This is done at a number of radii, for both the face and the back, the surplus metal being then chipped away and the blade finished by grinding and polishing. Other templates may be made for the shape of the fillets by taking sections through the blade roots by planes normal to the propeller axis. With care in the foundry, little excess metal has to be removed, the blades are closely similar when finished, and dynamic balancing is greatly simplified. This system of applying templates has been employed for over a year with highly satisfactory results as regards the checking of propeller castings, determining the amount of work to be done on them, and ensuring that the finished propeller conforms to the design.—*Paper by L. H. Kenney, "Journal of the American Society of Naval Engineers", February, 1942, summarised in "The Marine Engineer", Vol. 65, No. 779, June, 1942, p. 114.*

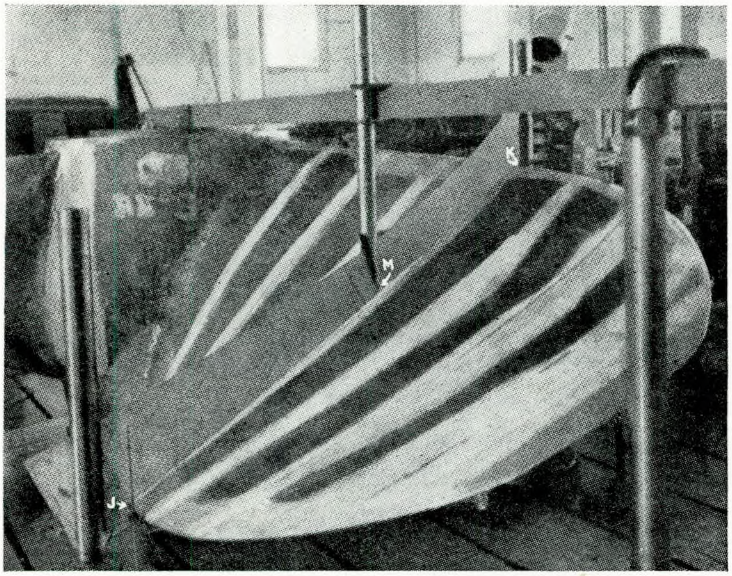


FIG. 2.—*The gauge is set to the designed cylindrical curvature after the boss has been machined. The pitch angle is determined by the points K and J.*

An Investigation into the Cause of the Failure of the Main Bearings of Internal-combustion Engines.

Bearing failures in general may be ascribed to two causes, *viz.*: (*a*) fatigue of the whitemetal; and (*b*) excessive bearing loads. Failures due to (*a*) generally occur in crankhead, top-end and crosshead bearings, and are associated with very rapid changes of load. In such bearings fissures develop at the bearing face and progress inwards towards the bearing shell, where they link together to form a mosaic of cracks. Finally, portions of the bearing metal, bounded by cracks, break away from the shell, and renewal of the bearing becomes necessary. Failures due to (*b*) are quite different, in these cases the temperature of the bearing rises to melting point at the surface of the bearing which then assumes, instead of a dull matt surface, a bright metallic lustre resembling solder which has been cut with a chisel. If the load is increased, the bearing becomes what is commonly termed "wiped", the oil wells and grooves being fitted up with metal which has reached such a temperature that flow at the surface of the bearing occurs. In some instances the load may not be great enough to cause "wiping" but excessive wear of the bearing results. Such failures are confined in general to main crankshaft bearings, and it is to this type of failure that the authors' investigation refers. Various papers dealing with the causes of the failure of engine crankshaft bearings have been published during the last few years, and in these papers failure was ascribed to a "whirl" condition of the crankshaft at the critical speed. No method, however, was suggested for estimating the loads on the bearings, the inference being that such critical speeds involved excessive loads on certain bearings which resulted in the failure of the latter. The authors' experimental work on this problem leads to an entirely different conclusion, indicating that with a 6-throw crankshaft of a single-acting four-stroke engine without balance weights, the shaft bends in approximately one plane; the magnitude of the bend and the position of the plane of bending, relative to the cranks, are in agreement with calculated results based on the well-known theory of the flexure of beams. Using the same theory, the bearing loads may be estimated. It is also shown how the influence of the reciprocating masses may be taken into account in estimating bearing loads, and the effect of bearing clearance on bearing loads is also indicated. The general conclusions drawn by the authors clearly demonstrate the advantages to be gained by the fitting of balance weights to the crankshafts of I.C. engines in order to reduce the loads on the end and centre bearings, and that in some cases the r.p.m. and centrifugal loading are such as to make the provision of balance weights essential. It is pointed out that many crankshafts have failed through bending fatigue, such fatigue loading being due to excessive weardown of the main bearings. There appears to be little doubt that the fitting of balance weights reduces wear-down and the risk of such crankshaft failures. The authors express the view that this whole subject has hitherto been clouded by the earlier "whirl" theories put forward as the cause of failure. The investigations described in this paper are claimed to show that such "whirl" theories are unsound, and that bearing loads may be calculated from quite simple theory.—*Paper (No. 8) by S. F. Dorey, D.Sc., and G. H. Forsyth, M.Sc., prepared for publication in the "Transactions of the Institution of Naval Architects" for 1942.*

Marine Watertube Boilers.

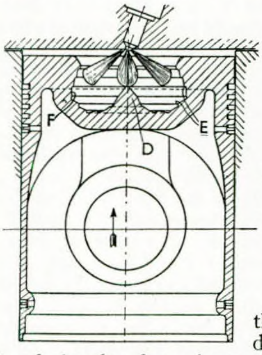
An improvement on the "D" type design of watertube boiler installed in many American merchant vessels is now being introduced in the first of a fleet of tankers building in the U.S.A. The new design is termed the "P" type boiler, and has an exceptionally high evaporative capacity, which makes it possible to meet the steam requirements of a 12,000-s.h.p. machinery installation with only two boilers arranged in a single casing on a flat above the main engine room, giving the vessels, which are of 9,800 tons displacement, a service speed of 16½ knots. Each boiler consists of a large steam drum with a slightly smaller water drum offset below it in order to give sufficient inclination to the main bank of tubes, between the rows of which is arranged the superheater. The furnace is water-walled by tubes all round. The working pressure is 375lb./in.^{2} and the superheat temperature 700° F.—*"Shipbuilding and Shipping Record", Vol. LIX, No. 19, 7th May, 1942, p. 487.*

An Investigation of the Failure of a Single Helical Geared Turbine System due to Combined Axial and Torsional Vibration.

The paper deals with the failure of single helical gearing due to a form of combined axial and torsional vibration, and the authors show that if the natural frequency of this combined vibration divided by the r.p.m. of the pinion shaft gives an integer, then patterned pinion wear quickly results, this being followed by pinion failure. Methods of avoiding this form of failure are discussed; these include the division of the system axially by the provision of claw or elastic couplings and the fitting of an additional thrust adjacent to the pinion. The advantage of flexible couplings as a means for preventing resonance in turbine blades is indicated. The paper also illustrates the value of high-frequency vibration instruments in the solution of problems relating to gearing.—*Paper (No. 10) by S. F. Dorey, D.Sc., and G. H. Forsyth, M.Sc., prepared for publication in the "Transactions of the Institution of Naval Architects" for 1942.*

Crossley Engine Combustion Chamber Design.

A new design of oil-engine combustion chamber, for which Crossley Motors, Ltd. (and other parties), recently secured a British patent, is illustrated in Fig. 2. It is claimed that with this design



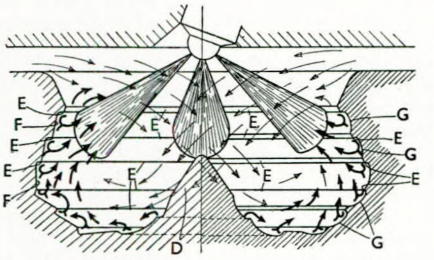


FIG. 2.

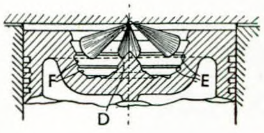
there is no layer of air which does not get directed into the fuel jets. The chamber is of circular form in sectional plan and has a raised projection (D) in its base which makes the chamber resemble an annulus. The whirling air from the cylinder, when transferred to the combustion chamber, is caused to swirl in helices around this annulus. The downward flow of air into the combustion chamber which causes the swirl is indicated by faint arrows. The wall of the combustion chamber is provided with inwardly directed ridges (E) approached by straight surfaces (F) or by curved surfaces (G). These approach surfaces are either straight or curved all the way round the combustion chamber, the centre diagram being intended to illustrate the two forms in one view. A dividing line, therefore, passes through the middle of the chamber. As the left-hand diagram indicates, all the approach surfaces are straight in this particular sectional view, but whether straight or curved, they are circles in any sectional plan view.—*"The Oil Engine", Vol. X, No. 110, June, 1942, p. 53.*

Comparative Tests on the Lubrication of Crankshaft Bearings.

The authors have shown in another paper *(No. 8)* how certain bearings of a crankshaft are liable to carry excessive loads with consequent rapid failures of the bearings. The present paper gives the results of some preliminary tests carried out on three types of bearings to ascertain whether the loading of bearings of the orthodox type can be increased by means of an alteration in the lubricating arrangements. Although the tests described are essentially comparative, the results obtained indicate the desirability of further tests with equal specific loading per unit area of bearing and equal quantity of oil circulating through each bearing. In view of the relatively high percentage of wear of the journals in the total wear taking place, the authors are also proposing to carry out tests with nickel-chrome steel journals having a Brinell number somewhat higher than 300. As regards the value of the breakdown load in relation to wear with normal loading in practice, the authors believe that the tests already carried out and those at present being carried out (which latter include temperature measurements at the bearings) will indicate whether or not the breakdown load depends on the ability of the bearing to maintain an oil wedge. If this is so, the resulting wear in practice will depend on the thickness of the oil wedge in relation the size of the abrasives present in the lubricating oil. In this case, then, the wear on the journal and bearing in practice will be a function of the temperature and state of cleanliness of the oil. Reduced wear would then involve a design of bearing in which an oil wedge may be maintained, an increase in quantity of oil circulated through the bearing, and more attention to the purification of the oil.—*Paper (No. 9) by S. F. Dorey, D.Sc., and G. H. Forsyth, M.Sc., prepared for publication in the "Transactions of the Institution of Naval Architects" for 1942.*

Mechanical Stokers in German Warships.

The experience obtained by the North German Lloyd Company with mechanical stoking in ships was dealt with in a paper recently read by Herr Schultz, of Gummersbach, at a meeting of an engineering society in Hamburg. Steinmüller mechanical stokers were first installed on board a steamer having three cylindrical boilers in 1937, when the company found it necessary to modernise several of their vessels to make them more economical in service. In 1938 and 1939 five more sea-going steamships were fitted with new boilers and mechanical stokers, and underwent certain other alterations, with the result that their service speed was increased from 10-11 knots to 13-13½ knots. On installing the new stokers in the first vessel it was found that the amount of steam generated was increased by 28 per cent., while three fewer firemen were needed. Improvements introduced as a result of experience with the first installation led to the ten stokers installed in the five ships giving every satisfaction from the point of view of reliability, economy and flexibility. By making suitable arrangements for bunkering, conveying the coal to the stokers and removing ashes, it was found possible to reduce the E.R. personnel of a coal-burning steamer to the number required for a Diesel-engined vessel of similar size. Projects for equipping steamships of the Hamburg-American Line and other vessels with mechanical stokers of this type could not be realised owing to the war, and had further development not been prevented, it might be assumed that the new method of firing would play an important part in the reconstruction of the German mercantile marine after the war.—*"Lloyd's List and Shipping Gazette", No. 39,778, 13th May, 1942, p. 11.*



Re-boilering of a Great Lakes Freighter.

The recent re-boilering of the self-unloading Great Lakes freighter "T. W. Robinson", is an excellent example of the technical and economic effectiveness of modern marine boilers. The ship is a coal-burning turbo-electric cargo vessel of 7,726 gross tons with a single propulsion motor of 3,000 s.h.p., and was originally equipped with three boilers occupying a floor space of 43ft.×14½ft., with a height of 19ft. The new boiler installation comprises only two "D" type Foster Wheeler boilers arranged in a single steel casing and occupying a space of 36ft.×11ft., with a height of 18ft. The new boilers therefore only take up 60 per cent. of the space required for the old ones, whilst their weight is 20 per cent. less. Each boiler normally generates 22,500lb./hr. of steam at a working pressure of 315lb./in.^{2} and total temperature of 650° F. The water-cooled furnaces are stoker-fired by rotary type stokers feeding on to sectional grates, which are stated to be very simple in operation, whilst the ashes are removed by means of a hydraulic jet installation.—*Article in "Heat Engineering", October, 1941, summarised in "The Marine Engineer", Vol. 65, No. 778, May, 1942, p. 99.*

New Swedish Cargo Steamer with Götaverken Turbo-compressor Machinery.

The cargo steamship "Rosa Smith", recently launched at the Lindholmens yard, Gothenburg, where she has been built for Swedish owners, is a vessel of the closed shelter-deck type, some 300ft. b.p., about 44½ft. in breadth and 26½ft. in depth moulded to shelter deck, with a carrying capacity of 3,050 tons d.w. The four hatches are fitted with Nielsen patent rolling hatch beams and the cargo-handling equipment comprises eight 5-ton derricks and one 10-ton derrick operated by eight steamdriven winches. The propelling machinery consists of a triple-expansion engine working in conjunction with a Götaverken turbo-compressor with a normal output of 1,550 i.h.p. at 97 r.p.m., but capable of developing about 1,750 i.h.p. The diameters of the cylinders are 20in., 29½in. and 52⅝-in. respectively, with a piston stroke of 35½in. The auxiliary machinery includes a 12-kW. 115-volt dynamo driven by a vertical steam engine. The ship is designed for a speed of 11¾ knots.—*"Lloyd's List and Shipping Gazette", No. 39,778, 13th May, 1942, p. 9.*

Use of Welding to Expedite Ship Construction.

It is reported that Mr. R. B. Shepheard, one of the senior ship surveyors on the staff of Lloyd's Register of Shipping, who has had much experience of welding before the war in Hamburg, and who has been supervising the construction of the 30 all-welded cargo steamers building to the order of the British Government in Richmond, California, has been lent to the Department of Merchant Shipbuilding and Repairs to advise on welding questions in order that the use of welding can be extended in this country without interfering with output. The British Oxygen Co. are acting as agents for the Department in the matter of the application of the Unionmelt welding process to British yards. A number of yards have been selected by the Admiralty as being suitable in general lay-out and existing electrical capacity for the application of welding on the wide scale on which it is proposed to utilise it for all forms of ship construction. The Unionmelt process is actually more of a resistance fusion process than an electric arc process, comparatively small and readily portable welding machines being employed to carry out the functions of distributing the Unionmelt powder over the seams to be welded, carrying and feeding the welding rods, applying the powder to the welds, and travelling along the seam at the correct speed to enable the welding to be performed. The process derives its name from the granulated slag-forming material known as "unionmelt". Within the layer of this powder intense heat is developed, which causes the welding rod and the edges of the steel plates being welded to be melted and fused together. Molten metal from the rod is intimately mixed with the molten base metal to form the weld proper (see Fig. 1) the heavy flux blanket making it

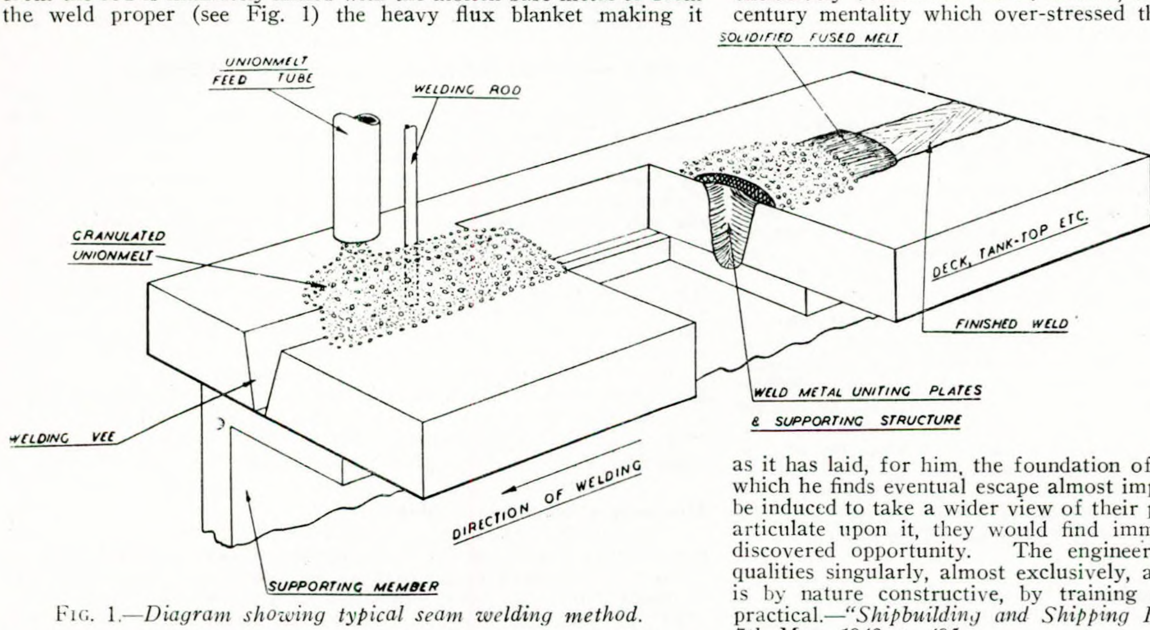


FIG. 1.—*Diagram showing typical seam welding method.*

possible to use much higher current densities than with ordinary hand arc welding. This automatic process is used to a very great extent in every big American yard, from 15 to 40 per cent. of the total amount of welding involved in the construction of an all-welded ship being carried out by the Unionmelt process, and the remainder by the normal hand processes already well known in this country. The Unionmelt automatic welding machines will shortly be distributed to the various British yards concerned, and while it is unlikely that their use will have an immediate effect on ship production, it is certain that when the planned application of automatic welding attains its peak, a marked acceleration of the merchant shipbuilding programme should result.—*"The Journal of Commerce" (Shipbuilding and Engineering Edition), No. 35,675, 11th June, 1942, p. 3.*

Welding Applications in Naval Machinery.

The first welded high-pressure steam boilers were accepted by the U.S. Navy authorities in 1930 and have now given over eight years of satisfactory service. Prior to 1930 the application of arc welding was limited, because of the relatively low ductility obtainable with bare or lightly coated electrodes. When the development of covered electrodes enabled sound welds to be made in heavy plate with both the strength and ductility of the plate itself, responsible engineers approved the acceptance. Electrodes used for welding naval machinery must be of an approved type. All naval welders are required to pass root and face bend tests, using the well-known guided bend tests. Pipe welders must pass these tests using specimens cut from pipe similar in size, thickness and composition to the pipe they will be required to weld. The special safeguards necessary in the welding of boiler drums are summarised; the welding of all longitudinal and circumferential seams must be carried out by the automatic arc process. Details are given of welded boiler casings and funnels; high and low pressure piping; H.P. and L.P. turbine casings; gears and gear cases; Diesel engines; heat-transfer equipment; electrical machinery, and maintenance of machinery. In conclusion, the authors discuss stress relieving. The article includes 14 illustrations and diagrams.—*Abstract of article by H. W. Hiemke and J. D. Bert in "Welding Journal" (U.S.A.) of March, 1942, reproduced in "Welding Literature Review", Vol. 4, No. 2, May, 1942, pp. 698-699.*

The Engineer and His Opportunities.

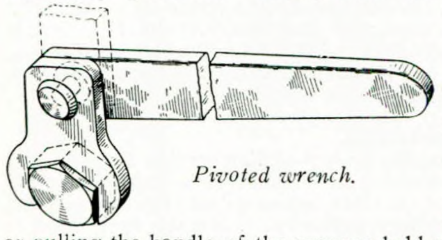
Addressing members of the Liverpool Engineering Society, Mr. Thos. Graham, B.Sc., commented on the fact that in a war which is wholly an engineers' war, not one distinguished engineer is to be found in a position of plenary authority. Months ago, when it became necessary to send a man out East, with Cabinet authority, where his only possible usefulness would have been co-ordination of technical resources, a professional politician was sent out, who, in the nature of things, could do nothing but listen and report. The speaker expressed the view that this was wrong, and that the genesis of correction would seem to lie in an enlargement of the training of engineers, which is hopelessly departmental because it is too exclusively vocational. It is, in fact, a survival of that nineteenth century mentality which over-stressed the strictly utilitarian values of knowledge, and is responsible for the promotion of the impression that "once an engineer, always and only an engineer"; the myth that outside the realm of the technical, the engineer is an anomaly. The speaker urged that all so-called vocational education should be eliminated from our elementary educational system, because it is fundamentally wrong for no other reason than that it is not education. The engineer has suffered more from it than any other person, as it has laid, for him, the foundation of a technical enclosure from which he finds eventual escape almost impossible. If engineers could be induced to take a wider view of their profession and become more articulate upon it, they would find immense fields of, as yet, undiscovered opportunity. The engineer is possessed of the very qualities singularly, almost exclusively, adapted to exploit it, for he is by nature constructive, by training accurate, and by necessity practical.—*"Shipbuilding and Shipping Record", Vol. LIX, No. 19, 7th May, 1942, p. 495.*

A British All-welded Swim Barge.

One of the most successful applications of welding in a British shipyard has been in connection with the building of an all-welded swim barge of the type used on the Thames, in a Thames shipyard. This vessel, which carries about 130 tons of cargo, is of the usual design and scantlings, the single hold being arranged with a watertight bulkhead at either end. The outside plating is in general ⅜in. thick, and the frames are 6-in. bulb plates at the sides and 6-in. angles at the bottom, the toe being welded to the plating in each case. The total weight of the steel, including that of the deposited metal, is 44 tons, as against 49 tons for a similar craft of riveted construction. This reduction of 10 per cent. in the weight of the vessel's structure represents an increase of 5 per cent. in the weight of the cargo which can be carried. Had the weight of cargo remained the same as in the riveted vessel, the dimensions of the all-welded barge could have been reduced slightly, thereby effecting a further saving in steel. A craft of this type lends itself to prefabrication, and this method of construction was extensively adopted. In estimating the advantages of electric welding, it is necessary to take into account the saving due to the decrease in the weight of the steel, which usually amounts to about 15 per cent. In the case of a medium-sized cargo vessel, this would represent a saving of something like £4,000 at present-day prices. This must be offset, of course, by the increased cost due to welding, which in the initial stages is bound to be high; but it may be anticipated that when further experience has been gained with welding by British shipyards, it will prove to be more economical than riveting.—*"Fairplay", Vol. CLVIII, No. 3,081, 28th May, 1942, pp. 631-632.*

Wrench for Awkward Nuts.

The accompanying illustration from *"Power"* shows a pivoted wrench or spanner for dealing with packing-gland or other nuts in inaccessible positions. The spanner can be made readily in any workshop, the component parts consisting of two pieces of flat steel formed to the dimensions required and connected by a loose rivet. There are several ways of using the tool, according to circumstances. While a nut is loose, it can be turned by pushing or pulling the handle of the spanner held at right angles to the jaw-piece, but where more power is required, it may be more convenient to use the handle as a lever with some fixed object, or temporarily inserted bar, etc., as a fulcrum. In other cases, the handle can be utilised simply to hold the jaw-piece on the nut while it is knocked round by means of a hammer and striker bar.—*"The Power and Works Engineer", Vol. XXXVII, No. 432, June, 1942, pp. 177-178.*



Pivoted wrench.

Machine Flame Cutting in Shipbuilding.

The authors describe a number of flame-cutting applications in shipbuilding, dividing these into shop operations, and field and ship operations. The first heading comprises straight and line cutting and plate-edge preparations; the preparation of special structural shapes, *e.g.*, cutting I-beams and H-beams in half to produce two T-bars, and removing flanges or reducing the flange width of beams and channels; the scalloping of butt straps, seam straps, angle flanges and beam flanges by a single blowpipe machine: shape cutting from templates or independent lay-out; the cutting of elongated manholes in floors, longitudinals and similar members; cutting square edges on heavy sections preparatory to thermit welding; stack cutting; and the cutting of J- and U-groove joints. Under field and ship operations, the following are described: A special vertical cutting machine for cutting the vertical straight-line portions of bevelled openings in bulkheads; portable cutting machines for trimming the tops and edges of bulkheads; the cutting of circular openings on board ship; the preparation of bevelled openings for manholes and hatches; and the development of a special precision cutting machine designed to cut the bottoms of heavy riveted structures to a metal-to-metal fit, which can be used to cut either from the outside or the inside of a cylinder. Full details are given of all the above operations, and there are 23 illustrations.—*Abstract of article by E. R. McClung and H. L. Wagener in "Welding Journal" (U.S.A.) of February, 1942, reproduced in "Welding Literature Review", Vol. 4, No. 2, May, 1942, p. 659.*

Life-saving Craft of New Design.

An outstanding feature of a reversible sailing lifeboat, designed, constructed and patented by Mr. J. L. Spooner, of Hull, is a semi-circular wooden keel, which is fitted and fixed as a centre board. The keel is reversible and locked in place with two locking pins on chains. When the boat is stowed in position on skids or on deck the keel is uppermost to facilitate the launching of the craft, the reversing apparatus to be brought into operation if the boat takes the water in the same position as it is stowed, being worked by withdrawing the two brass drop pins and replacing them after the keel has been turned round on its centre pin. The stem and stern of the boat are shaped for steering, and it is equipped with a mast, yards, sails and oars, some of this gear being stowed outboard along each side so as to be available whichever side up the craft floats. The timbers used in the construction of the hull are stiffened with galvanised-steel corner and nosing plates. Watertight provision lockers, Kapok-filled buoyancy tanks and water tanks are other features of the design, the buoyancy allowance being well in excess of the regulations. The craft has been named the "Spoonfost" and is intended to be built in sizes to accommodate 10-12 and 20-22 persons, the surface per person being 4·1ft.^{2}. It is claimed that the steering and navigational capabilities of the craft will give those using it a reasonable chance of making a landfall, and that they fulfil all the requirements for an improved life-saving boat.—*"The Journal of Commerce" (Shipbuilding and Engineering Edition), No. 35,639, 30th April, 1942, p. 2.*

Swedish Life-saving Raft.

A short time ago reports were published of the amazing rescue in the Atlantic of two seamen from the torpedoed Swedish motorship "Yngaren". They had been drifting on a raft in cold and bad weather for a whole month, but when found they were still alive, although completely exhausted. Of the rest of the crew no trace was found. The rescue of the two sailors is ascribed to the construction and seaworthiness of the raft and its equipment. The raft, designed by Capt. Y. Cassel of the lost vessel, was of unusually large dimensions—about 13ft. by 16ft.—and was fitted with 32 buoyancy tanks arranged in two tiers. Above and below these tanks were wooden floors surrounded on all four sides by a wooden bulwark about 4ft. high, as a protection against wind and sea. Benches were fitted along the bulwark and over the seating space was fixed a pole, from the one short side to the other, across which a tarpaulin covering could be drawn. The top and bottom of the raft were similar, so that it could be used no matter which side came up when taking the water. Stores were stowed in a locker built into the centre of the raft and covered with sailcloth on the outside. On the top and bottom of the locker were manholes with covers of sheet iron secured with screw clamps. The contents of the locker consisted of blankets, jersey, oilskins, tinned provisions, matches, primus stoves, paraffin, lanterns, rockets, etc. A wooden receptacle for fresh water was fixed at the side of the locker and was, like all other supplies, accessible from both sides of the raft.—*"The Engineer", Vol. CLXXIII, No. 4,507, 29th May, 1942, p. 449.*

Rising Temperature in Cylinder Jackets of Diesel Engine.

An article in a recent issue of *"Power Plant Engineering"* concerns a 225-h.p. three-cylr. Diesel dynamo engine in a flour mill, in which for 3½ years the difference between the inlet and outlet temperatures of the cylr. jacket water was from 12° to 15° F., but which has lately risen to 35° F. The jacket walls are clean, fuel economy "better than ever", and all conditions apparently normal except that the exhaust temperature has increased by 50° F. Opinions are published as to the cause of the rise in jacket temperature. Reference is made to the suggestion that a heavy deposit of carbon has built up on the under side of the piston heads, due to various possible causes which are suggested, including change of lubricating oil, excessive amount of oil, and emulsification of oil by leakage water or condensed moisture. Other correspondents attribute the trouble to the formation of deposits in the circulating pump and/or pipes, leakage of lubricating oil into the combustion chamber (which would account for the alleged reduction in fuel consumption) or some trouble in the fuel-injection system.—*"The Power and Works Engineer", Vol. XXXVII, No. 431, May, 1942, p. 154.*

Efficiency of Single-screw Ships.

Despite the large number of slow-speed cargo vessels now under construction in the U.S.A., the favourable results obtained with the relatively high-speed cargo fleet of the Maritime Commission will probably cause American ship-owners to revert to higher speeds after the war, and for that reason alone it is likely that British owners will have to adopt a similar course, irrespective of the economic advantages, or disadvantages, entailed. Twin-screw machinery may become more common for cargo vessels, as it is not always convenient to develop the 9,000 h.p. required to drive a ship carrying, say, 10,000 tons d.w., at 16 knots, on a single shaft. It should be noted that, on the whole, the single-screw ship is more efficient from a propulsion point of view than a twin-screw vessel, for not only is it possible to have a larger and, consequently, slower running propeller, which is an advantage, but the shaft brackets, or shaft bossing, of the twin-screw installation are eliminated. The resistance of the shaft appendages has quite an appreciable effect on the power required to propel the ship, although, if stream flow tests are made, this source of waste may be reduced, but never entirely eliminated. The steering of a twin-screw ship is also less efficient, for the propellers rarely run at exactly the same r.p.m., and the helm angles necessary in service are usually greater than in the case of a single-screw vessel. Furthermore, the twin-screw ship is more complicated structurally.—*"Fairplay", Vol. CLVIII, No. 3,079, 14th May, 1942, pp. 574 and 576.*

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