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The Importance of Governing Arrangements for Marine Installations, with Special Reference to Torsional Vibration.

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Synopsis.

The investigation described in this paper illustrates the importance of additional torsional loading between the engine and the propeller under conditions of engine racing in rough weather. It is shown that the additional loading may be due to:—

(1) Severe torsional vibration due to the absence of damping when the propeller is not fully immersed.

(2) The accelerations resulting from the general speed surges during racing. Of these additional torsional loads, it is shown that (1) above is by far the most important with engines fitted amidships. The effect of fitting governors and the positioning of the major one-node criticals are discussed with reference to rough weather conditions.

The author's attention has been drawn on many occasions to cases of shaft failures resulting from torsional vibration both in the engine crankshaft and in the screwshaft. In the case of failure in the latter, fractures have generally exhibited the well-known characteristics of corrosion-fatigue. In other cases, however, failures, whilst indicating the presence of fatigue, have not shown corrosion-fatigue markings. On making investigations into the torsional vibration characteristics, it has in many instances been difficult to account for failures of the latter type. There have been sufficient numbers of failures in vessels, where the 1-node major critical was, say, 10-15 per cent. above the normal running speed, to indicate that a margin of some 30 per cent. was necessary to ensure freedom from shaft failure. The importance of this, however, was not fully realized until recent investigations were carried out under conditions of engine racing in rough weather.

Before proceeding further with the investigations to be described, the author considers it well to draw attention to a paper entitled *"The Relation of Fatigue to Modern Engine Design", of which the following is an extract. Figs. 15 and 16 of that paper are included as Figs. 1 and 2 of the present paper.

* Trans. N.E.C. Institution of Engineers and Shipbuilders, Vol. LI, 1935.

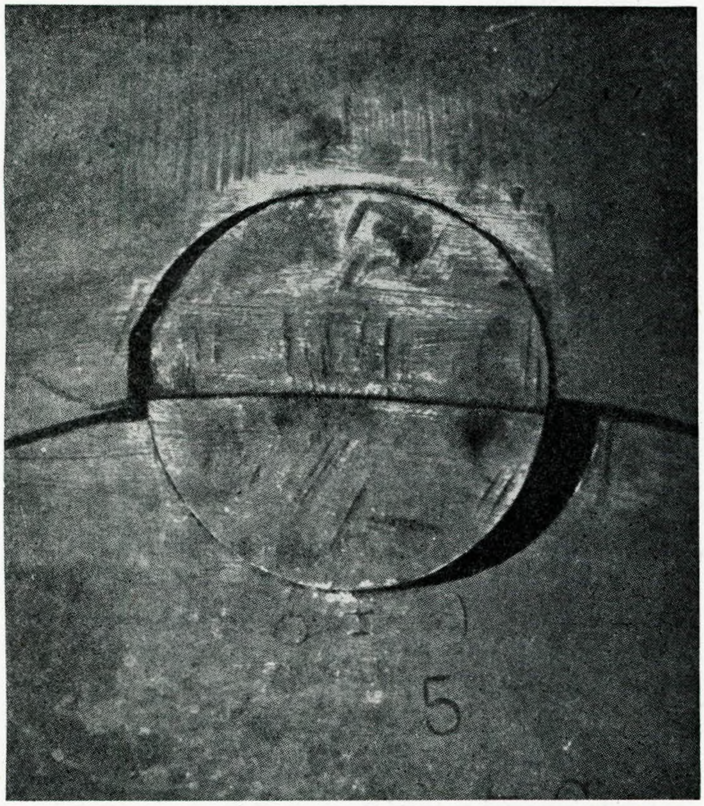


Fig. 1.—*Ship's crank-web moved on journal by impact.*

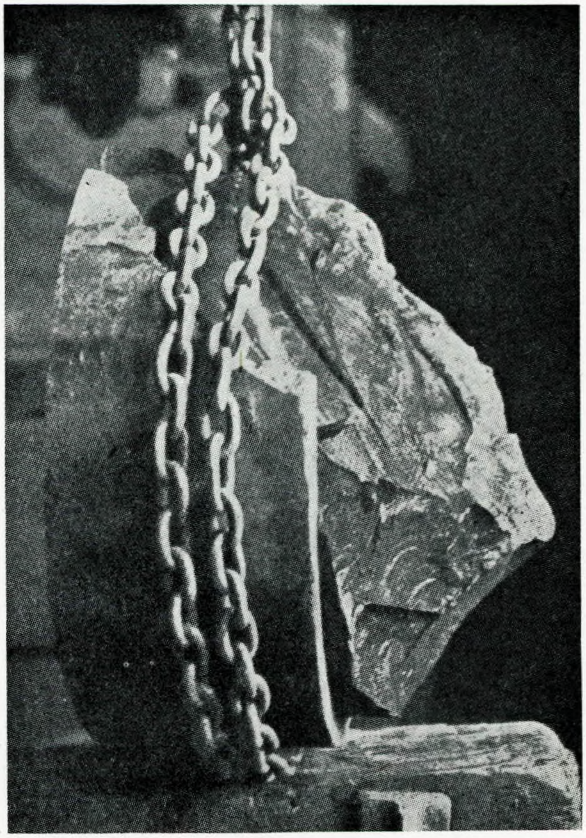


Fig. 2.—*Tail shaft connected to above crank broken in service by few severe blows.*

"Figs. 15 and 16 are photographs of damage caused at the same time to a ship's crankshaft and its propeller-shaft. No fouling of the propeller is reported, but the combined quadruple-expansion and turbine engines raced violently under full head of steam. Fig. 15 shows the crank-web having slid on its journal to an extent indicated by the dowel and dowel hole. The shrinkage was good as revealed by wholesale seizing between the journal and bore of the web. Enormous forces must have been applied, for a few beats of the propeller, as its blades entered the water in a heavy seaway. The tail-shaft exhibits a very unusual type of fracture in that it looks very like a fatigue fracture in texture, but it lacks *(a)* point of initiation, *(b)* splinters, *(c)* age-rings or *(d)* final granular ruptured area. The whole fracture is made up of many smaller ones located, and travelling, in a haphazard direction. It would seem that a few very heavy blows have brought about step-by-step impact cracking. On examination the material was found to be excellent:—

Carbon	0·29 per cent.
Tensile strength	29·6 tons.
Elongation in 3in.	35·0 per cent.
Bends 1in. × $\frac{3}{4}$in.	180° flat uncracked.

This example is not a true case of fatigue cracking, but it is quite interesting in itself. The object of its inclusion is to illustrate by an extreme case the kind of conditions a marine engine has to face in service on occasion".

The type of brittle fracture discussed, which is not true fatigue cracking, is particularly interesting in view of the cause of failure to be described.

The torsional load necessary to create the failure shown in Fig. 1 is very high indeed. In this connection it may be stated that the factor of safety of a shrink fit is in the neighbourhood of four. Whether or not torsional strength is added or subtracted by addition of a dowel is a debatable point, on which much has been written. It is clear, from Fig. 1, however, that dowel shear area in itself does not produce the necessary strength, since in practice it is found that the shrink fails due to the web lifting clear of the journal, exposing corners at the journal and web where flow of metal takes place quite independent of shear of the dowel itself. Taking the probable factor of safety as four, the torsional load to produce such a failure would represent a stress in the intermediate shafting in the neighbourhood of 20,000lb. per sq. inch.

It is difficult to account for such high torques being developed, even taking into consideration the impact torque from the propeller upon being re-immersed after racing. *Investigations previously carried out using a stroboscopic torsion meter indicated that such impact torques would be in the neighbourhood of twice the mean transmission torque.

The author's attention was recently drawn to the case of a vessel fitted with steam reciprocating triple-expansion engines, in which six crankshaft failures had occurred during the period April, 1938 to November, 1942. In each case the failure occurred at the L.P. crank, the after web of this crank moving on the journal in a similar fashion to that shown in Fig. 1, which has been found upon enquiry to be that of a failure at the L.P. crank. These cranks were constructed by different well-known engine builders whose experience in crankshaft construction was beyond doubt. Various sizes of dowels were fitted from $1\frac{3}{4}$in. dia. to $2\frac{1}{4}$in. dia. by about 4in. long. It was at first considered that engine seatings might be the cause of the trouble, and since there was a complaint that movement did occur at the L.P. crank, very special care was taken, after the sixth crankshaft failure, to make the thrust seating integral with the engine seating, and to add longitudinal stiffness in way of the L.P. crank and extending forward of that crank-pit. Although such stiffening had been considered necessary in view of the movements which had been complained of, it was not thought that the real cause of failure, *i.e.* very high torques, had been ascertained. Furthermore, it was considered that lack of alignment would have set up fatigue bending of the crankshaft which would have resulted, if severe enough, in fatigue cracks rather than a torsional movement of the journal and the web.

In view of the foregoing, it was decided to investigate fully the torsional vibration characteristics of the engine and shafting system, and to take torsiograph records to confirm the calculations made.

Appendix I gives the torsional vibration calculations made before the trial and it will be noted that the natural 1-node frequency between the engine and propeller has been calculated at 324 v.p.m.; with service r.p.m. 65-70 it was necessary to calculate the stresses arising due to the 5th and 6th orders. The vibration stresses due to these orders amounted to 142lb./in.^{2} and 1,350lb./in.^{2} respectively, these values being derived as shown in Appendix II. In spite of this low estimate of stress, it was still considered necessary to take Geiger records because of the number of L.P. crank failures which had occurred.

Trials at Sea.

It should be stated at this point that the vessel had the following particulars as given in Lloyd's Register, *viz:*—

Length	280·0ft.
Breadth	40·1ft.
Depth mld.	21·33ft.
Load draught ...	18·5ft.

The draughts forward and aft during the trials were 5ft. 11in. and 12ft. 1in. respectively, the vessel being without cargo or bunkers, and the propeller blade being about 1ft. 6in. out of water when at the top of the aperture. The weather during the trial could be described as "seas moderate to heavy".

Arrangements were made for taking Geiger torsiograph readings just aft of the thrust block. It had been hoped to take records over the speed range of 30-80 r.p.m. at intervals of 5 r.p.m., but owing to the state of the weather it was quite impossible to obtain or maintain any desired speed. In the circumstances there was no alternative but to take continuous records under conditions of engine racing. In spite of these apparently unsatisfactory conditions for taking records, very valuable results were in fact obtained. The records have yielded consistent results upon analysis and show that even with very rapid speed surges during engine racing, the various orders of torsional vibration all appear on the record as the corresponding critical speeds are passed through.

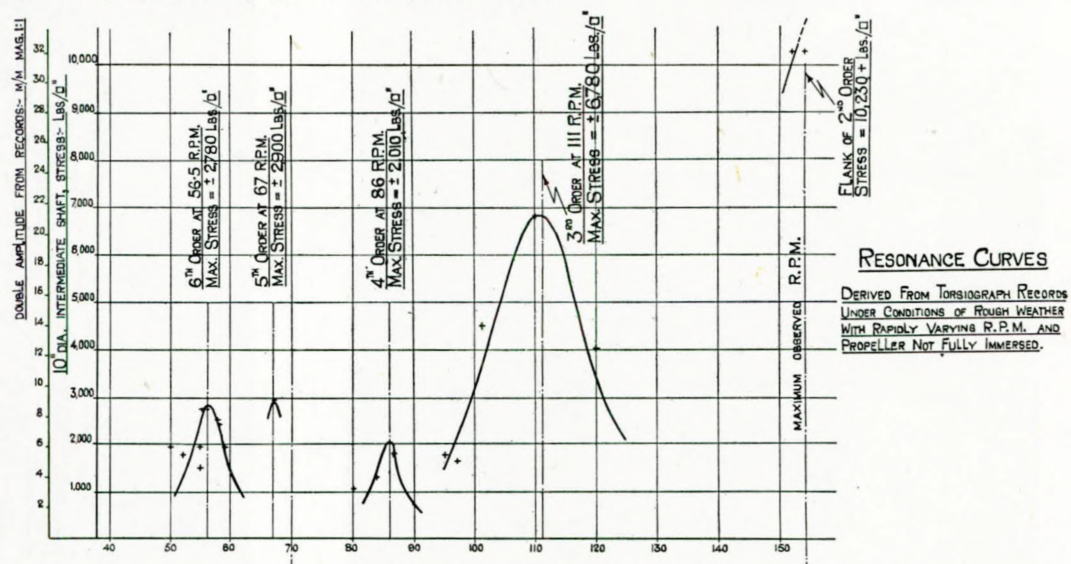


FIG. 3.

Discussion of Trials and Torsiograph Records.

Fig. 3 shows the resonance curves obtained from the records, and it will be noted that the natural frequency is about 339 v.p.m. This figure is somewhat higher than that calculated, and this can be accounted for to some extent by the fact that the calculations allowed 25 per cent. for entrained water, whereas in fact the propeller was not fully immersed and under conditions at sea the amount of the immersion would vary very considerably. The stresses in Fig. 3

* Ref. "Strength of Marine Engine Crank Shafts", by S. F. Dorey, D.Sc., Trans. N.E.C. Institution of Engineers and Shipbuilders, 1939.

are also higher than the calculated figures, as might be expected from considerations of propeller damping. This is an important point since propeller damping is very large under normal conditions, and prevents the development of enormous vibration stresses due to 1-node criticals. Lack of complete immersion of the propeller has the effect of reducing propeller damping, and this will be shown more fully in dealing with vibration stresses which occurred at 154 r.p.m.

Figs. 4 and 5 show resonant portions of the torsiograph readings corresponding to the 3rd, 4th, 5th and 6th orders. The revolutions per minute of the engines have been derived by the use of the fixed vibrator and by a make-and-break trace operating once per revolution. The values of revolutions so obtained have been indicated above the fixed vibrator trace, and it will be noted that violent changes in speed occur due to the propeller being alternately partially or completely out of the water and immersed. Under these conditions no steady resonant vibration was recorded, *but at the correct speeds for the various orders transient resonant vibrations appear, and the rapidity with which such resonance builds up is a remarkable feature of the records.*

In Fig. 5 the speed surges were even greater than shown in Fig. 4, and it will be seen that speeds of 154 r.p.m. were attained. Moreover, the rise and fall of speed occurred in a very short space of time, namely, 2-2½ secs. respectively. In order to form some estimate of the torque loading in the straight shafting due to the speed surges, Fig. 6 has been obtained from the records shown in Fig. 5. The curves indicate the variation of velocity at the engine during two speed surges whilst racing. These curves have been differentiated to derive the maximum values of the angular acceleration during increase and decrease of engine speed. These calculations are shown in Appendix III from which it will be noted that the maximum value of stress in the intermediate shaft resulting from these speed surges amounted to 3,810lb./sq. inch. Such a value in itself could not have accounted for the type of failure shown in Fig. 1. It should be mentioned, however, that apart from the records of racing shown in Fig. 6, twice during the trial more serious speed surges occurred whilst adjustments were being made to the Geiger. The impression, however, from ordinary observation was of very severe conditions for the crankshaft.

Apart from the racing, another and more important fact emerges, namely, the very violent torsional vibration at 154 r.p.m., where a distinct 2nd order occurs. It should be explained with reference to Fig. 5 that the fixed vibrator trace is not in line with the vibration trace, the latter being 19 mm. to the left of the former. At 85 r.p.m. and 113 r.p.m. during racing, the 4th and 3rd orders respectively, also clearly appear in the records, resonance building up with amazing rapidity. When the speed reached 154 r.p.m., although the instrument was hitting the stops hard, the 2nd order vibration frequency was maintained. With a double amplitude equal to that permitted by the stops, the stress would be equal to ±10,000lb. per sq. inch. By examination of the record and by the sound made by the instrument hitting the stops, all the indications were that the true amplitude was considerably greater than that which could be recorded. The conclusion to be drawn from this is that with the propeller out of water and a consequent surge of speed of 154 r.p.m., the shafting is thrown into a state of violent torsional oscillation. The energy input to the vibration comes from the 2nd order harmonic of the engine turning moment and the large amplitude is due to the absence of propeller damping, which would otherwise be present with an immersed propeller. Further, the higher orders excite vibration during the speed surges at revolutions corresponding to resonance.

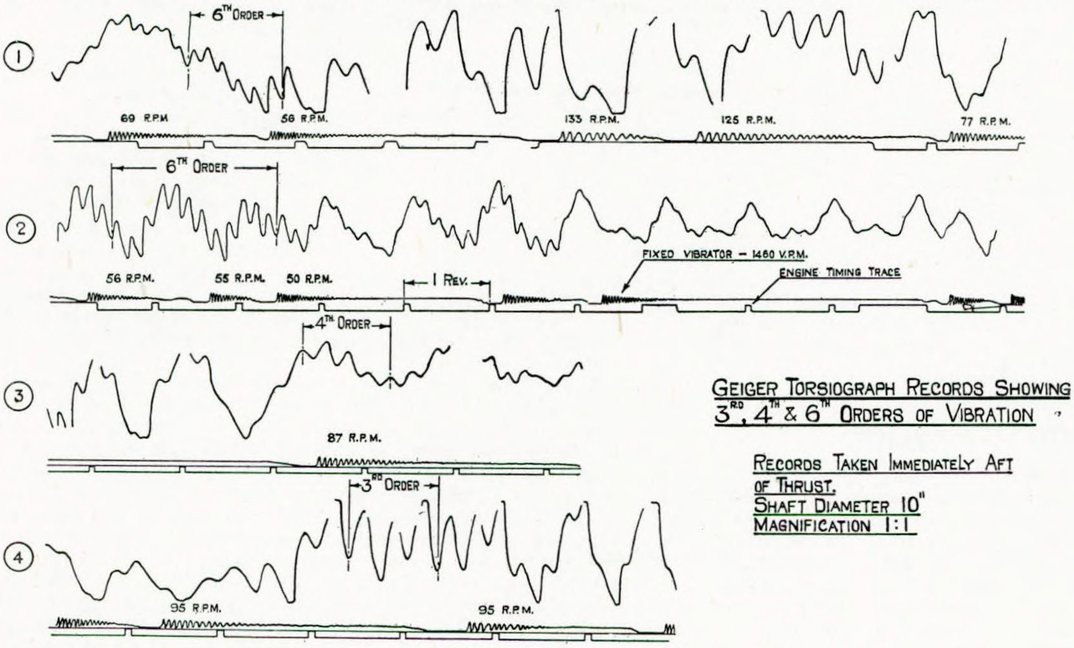


FIG. 4.

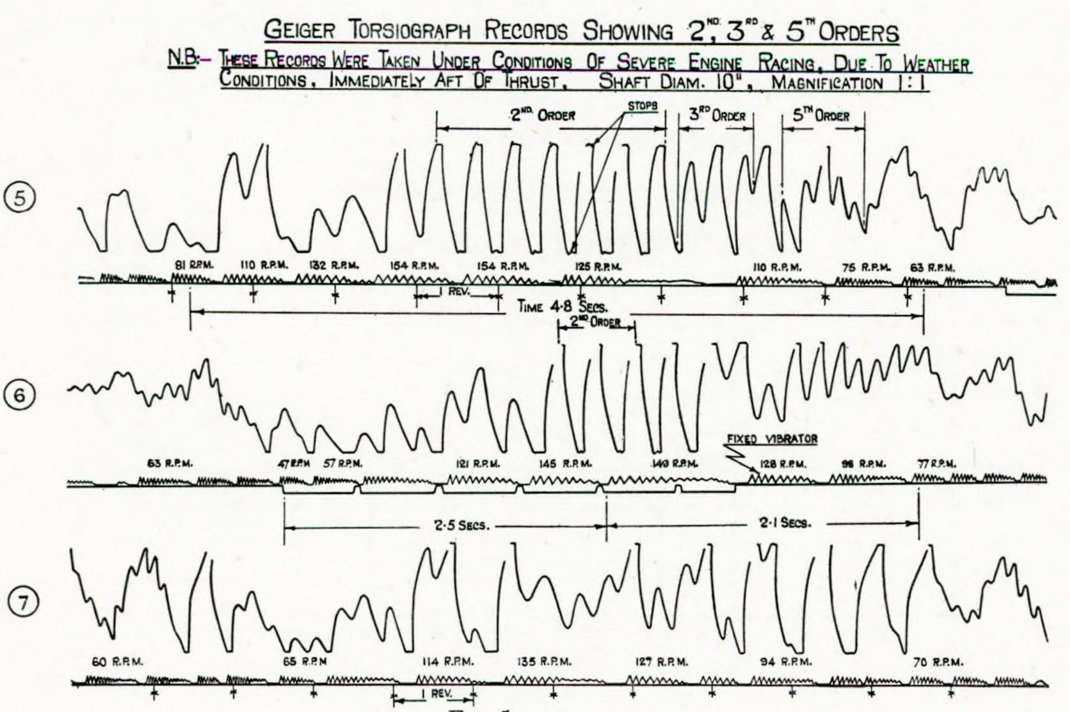


FIG. 5.

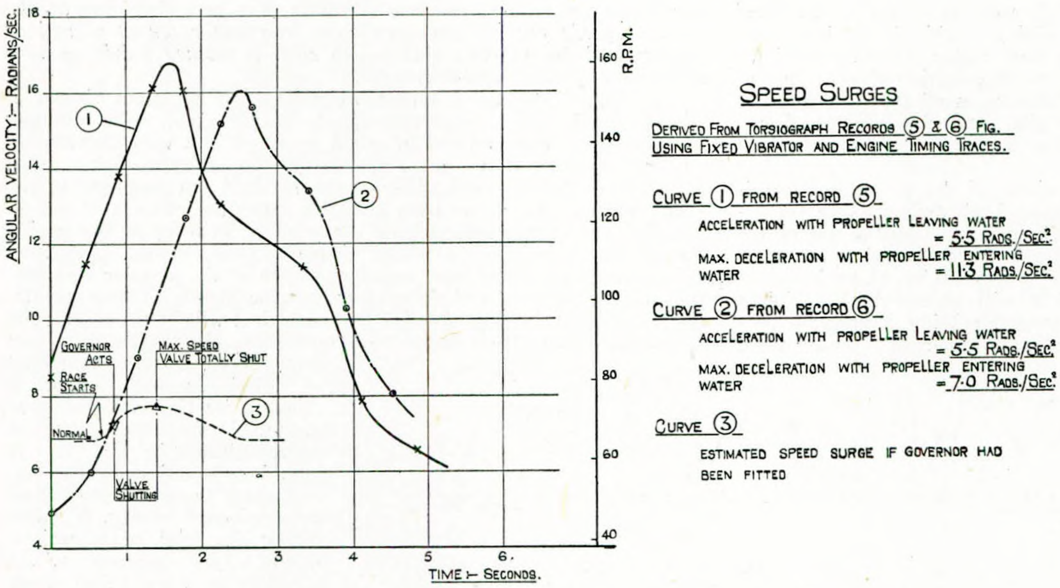


Fig. 6.

From the foregoing it is seen by addition of the torque due to the general speed surge to that due to torsional oscillation, that the resulting stress in the straight shafting at 154 r.p.m. will be :—

$$= \left\{ \begin{matrix}\text{speed surge} \\ \text{stress}\end{matrix} + \begin{matrix}\text{vibration} \\ \text{stress}\end{matrix} \right\}$$

$$= \{3{,}810 + (>10{,}000)\} \text{ lb. per sq. inch}$$

$$= >13{,}810\text{lb. per sq. inch.}$$

As already explained, the torsional load required to cause slippage at the crankshaft shrink would correspond to a stress of about 20,000lb. per sq. inch in the intermediate shafting. In comparing this approximate stress with that recorded, it should be remembered that records were not taken during the most severe periods of racing. In spite of this, it will be noted that the margin of safety is very small. From general observations during racing, the conditions for the crankshaft seemed very severe. In consequence, before the conclusion of the trial, enquiry was made of the captain as to whether such conditions were typical or extreme. The inference drawn from this enquiry was that the conditions of loading of the vessel, namely, no bunkers and no cargo, were extreme, but that the racing was similar to that encountered during bad weather.

The engine was examined during the trial, particularly in way of the L.P. crank. All crankshaft bearings showed nothing more than a normal running heat with the exception of the after H.P. bearing (*i.e.* No. 2 bearing) which was slightly above normal running heat. No transverse movement was noted at any of the crankshaft journals, and a vibrograph set up adjacent to the pocket of the after L.P. bearing showed no movement or pulsation whatever, excepting those due to the general movement of the hull in space.

At the conclusion of the trial the crankshaft was examined, and it was found that failure had again occurred at the after web of the L.P. crank. The type of failure was precisely as shown in Fig. 1, the opening at the sides of the dowel being about 10/1,000in. and the lift of the web on the journal about 2/1,000in. Torsiograph records had thus been obtained under severe conditions of racing, leading to the same type of failure as had formed the *raison d'être* for the investigation.

Conclusions from Torsiograph Records.

(1) Fig. 3 indicates that under normal service conditions envisaging maximum revolutions of 70 per minute, the torsional vibration characteristics, even with propeller partially immersed, are entirely satisfactory.

(2) With weather and loading conditions during this trial, the torsional vibration characteristics are shown to be severe, due primarily to :—

(*a*) Increases of speed to 154 r.p.m. bringing in a 2nd order vibration.

(*b*) The absence of propeller damping, which is of great importance for reducing 1-node vibration torques. The records clearly show that at 154 r.p.m. the vibratory torque on the L.P. crank was considerably in excess of that corresponding to a stress of ±10,000lb. per sq. inch in the intermediate shafting, the actual measurement being outside the range of the instrument.

(3) Apart from the vibratory torque mentioned in (2) above, there is an additional torque due to the general speed surge, which corresponds to a stress in the intermediate shafting of 3,810lb. per sq. inch. In this connection, it has already been remarked that twice during the trial, as far as could be judged from ordinary observation, more severe racing occurred than that recorded. If this was in fact the case, both the vibratory torque mentioned in (2) above and that due to the speed surge would be greater, the former because the 2nd order critical speed would in all probability be reached.

(4) It has been stated that a stress of 20,000lb. per sq. inch in the intermediate shafting would produce a dangerous torque for the crankshaft shrink. Whilst such a load was not recorded, a torque corresponding to a stress of at least 13,810lb. per sq. in. in the intermediate shaft was actually measured, and all the indications are that the true maximum stress was actually considerably in excess of this figure. It is of interest to note that the failure under discussion in

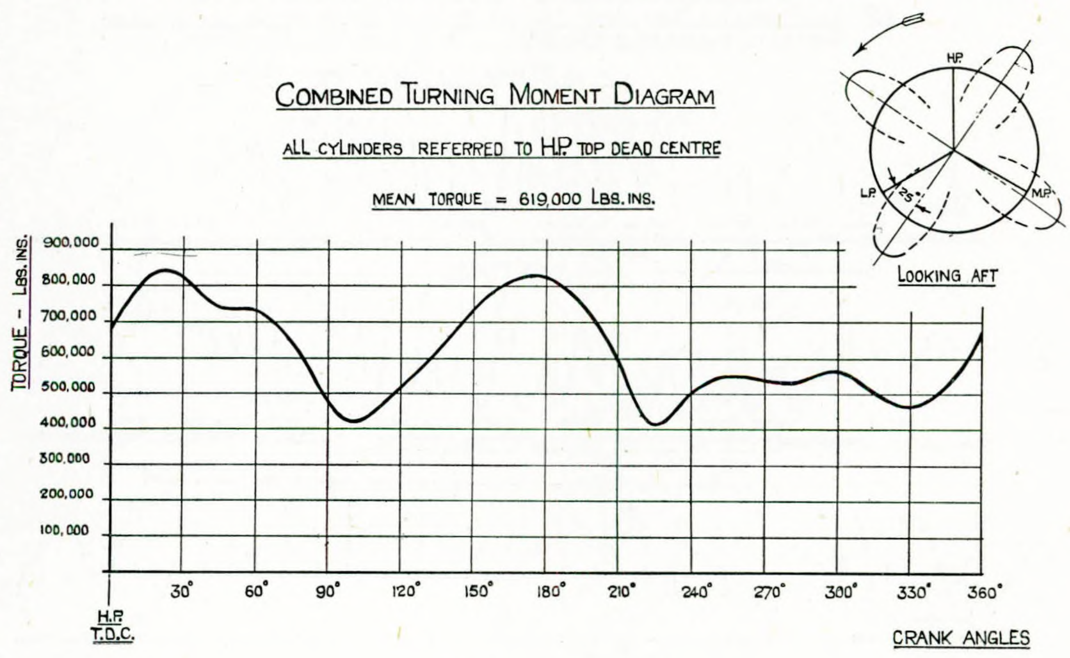


Fig. 7.

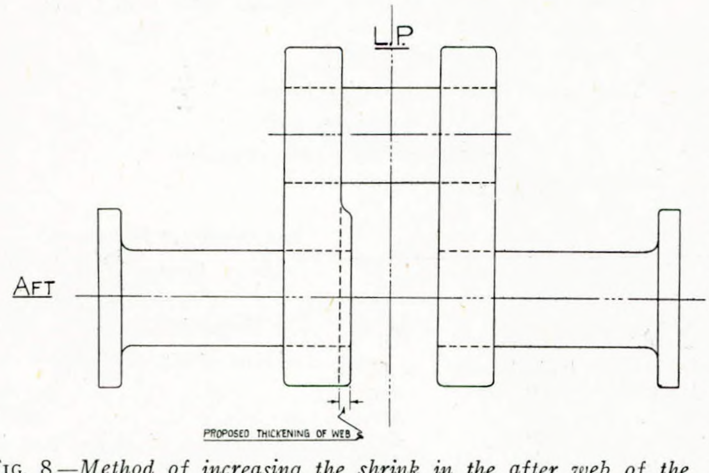


FIG. 8.—*Method of increasing the shrink in the after web of the L.P. crank.*

thsi paper invariably occurred in the *after* web of the L.P. crank. This is what might be expected from consideration of the cause of the failures as now ascertained. Both the vibration torque and that due to the general speed surge is reduced at the forward L.P. web by an amount corresponding to the torque required to accelerate and decelerate the heavy reciprocating L.P. masses including the pump gear driven from the crosshead. In view of this, it may be stated that for the machinery under investigation the after web of the L.P. crank has been stepped as shown in Fig. 8, thus adding considerably to the strength of the shrink grip.

From the foregoing it is obvious that speed surges alone, with engines fitted amidships, do not produce unduly high torques; in other words, the propeller and long length of intermediate shafting are "kind" to the engine crankshaft during racing. The major cause of the excessive torsional loading during racing comes from violent torsional oscillation in the absence of propeller damping. So far as steam reciprocating machinery is concerned, the aim should be to prevent speed surges taking the machinery into speed ranges corresponding to the 2nd and 3rd orders. Where this cannot be achieved, even by the fitting of a governor, care should always be taken to trim the ship by the stern to maintain, so far as possible, propeller immersion. For the case dealt with in this investigation, the frequency of 339 v.p.m. is higher than usual for this type of installation; even so, the 2nd order vibration was excited during racing as shown in Fig. 3. Had a governor been fitted to prevent speed surges in excess of, say, 100 r.p.m., no damage would have occurred to the crankshaft. A more usual value of 1-node natural frequency for steam reciprocating machinery with engines amidships would be about 240 v.p.m., giving 3rd and 2nd order criticals at 80 and 120 r.p.m., respectively. In such cases, with normal running speeds of 70 to 80 r.p.m. it would be difficult to avoid the 3rd order by the fitting of a governor, but a governor would prevent the higher torsional stresses due to the 2nd order, which would otherwise be fully excited at 120 r.p.m. The results of this investigation indicate that the 3rd order torsional loading, whilst high, should not cause damage during racing, and it is likely that with proper attention to trimming the ship aft, the loading would be less than indicated in Fig. 3.

With aft-end steam reciprocating machinery the 1-node frequency will be about 500 v.p.m. Under these conditions only the higher orders of vibration would be excited and, since the magnitude of the resultant harmonic of the T.M. diagram for these orders is corresponding less, violent torsional oscillation would not develop during racing. On the other hand the torques resulting from the general speed surge would in all probability be higher, due to the reduction in resilience as compared with that present in the shafting of the amidships installation. This is a matter requiring further investigation. Regarding oil-engine installations with machinery fitted amidships, common values of 1-node frequency would be 200-350 v.p.m., and the lowest order of major vibration would be the 3rd order with a normal running speed of 100-120 r.p.m. Under these conditions it would not be possible always to avoid the major order under conditions of racing, but provided the major order is lower than the normal running speed, which incidentally will generally be the case, the propeller would be immersed during transient passing of this critical, and propeller damping would prevent the development of excessive torques.

With aft-end installations the natural frequency will be about 500 v.p.m. with the major critical generally above the normal running speed. In this case there is a risk of the development of violent torsional oscillations as described in this paper, unless care is taken to place the critical well above the normal running speed and governing arrangements are related to the margin provided, in order to avoid working on a high part of the flank of the critical. It may be added that this theorising from the results described in this paper fits likes a jig-saw into practical experience with many cases of trouble in aft-end installations. Experience indicates that where the governing arrangements are satisfactory, the major critical, if above the normal running speed, should be placed not less than 30 per cent. above.

GOVERNING ARRANGEMENTS.

A common form of governor fitted to steam reciprocating and oil engines is of the ordinary inertia type fitted to pump or special governor levers. The nominal increase of speed for which these can be adjusted to act is 4-5 per cent. above the speed at which the engines are required to run. These governors will, however, cut-out on any sudden acceleration, in spite of the fact that the average speed of 4-5 per cent. above the running speed has not been reached. In addition, an emergency governor is fitted which comes into operation in the event of prolonged or excessive engine racing due to a fractured shaft or lost propeller. This emergency governor does not allow of the admission of power after such speed surges have ceased.

In the case of steam reciprocating engines running at nominal speeds up to 85 r.p.m., the emergency governor locks up at twice the running speed, or even higher, this being necessary because of the sudden accelerations possible due to the low inertia of these engines. At higher speeds a slight reduction in the margin mentioned above is made, *e.g.* at 95 r.p.m. (normal) the emergency speed might be fixed at 165 r.p.m., and with 100 r.p.m. (normal) at a speed of 180 r.p.m. For oil engines the margin for the emergency governor may be closer, being usually 6-10 per cent. above the speed specified for the engines.

However, the important point is the functioning of the main governor, which under conditions of rough weather should prevent excessive speed surges. It is claimed by one well-known firm of manufacturers of governors that had a governor been fitted to the engines on which this investigation was carried out, the speed rise would have been no greater than that shown in Fig. 6, Curve (3). In these circumstances, reference to Fig. 3 indicates clearly that no trouble would have been experienced with the shafting of this vessel.

The author believes that the importance of engine governing has not been fully realized in the case of the steam reciprocating engine, nor has that of the necessary margin between normal running speed and a major critical in the case of many oil-engined vessels.

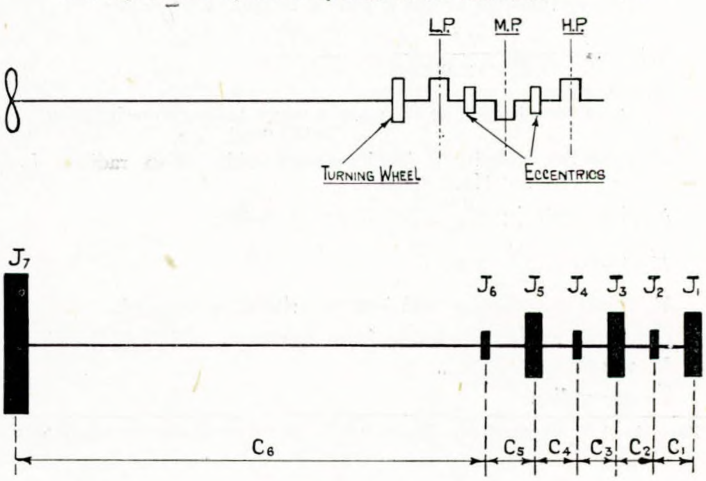
ACKNOWLEDGMENTS.

The author wishes to thank the Committee of Lloyd's Register of Shipping for permission, kindly given, to publish this paper, also to acknowledge help given, both on trials and in the preparation of the results for presentation, by his colleague, J. W. Burrill, B.Sc.

APPENDIX I.

Natural Frequency of Engine System.

The equivalent dynamic system is as follows:—



Where inertias and stiffnesses are:

Inertia.	Stiffness.
$J_1 = 2{,}890$lb. in. sec.^{2}	$C_1 = 603\times10^6$lb. in./radian
$J_2 = 457$ „ „ „	$C_2 = 603\times10^6$ „ „ „
$J_3 = 3{,}162$ „ „ „	$C_3 = 603\times10^6$ „ „ „
$J_4 = 234$ „ „ „	$C_4 = 445\times10^6$ „ „ „
$J_5 = 4{,}160$ „ „ „	$C_5 = 480\times10^6$ „ „ „
$J_6 = 500$ „ „ „	$C_6 = 10{\cdot}4\times10^6$ „ „ „
$J_7 = 37{,}700$ „ „ „	

N.B.—J_7 (propeller inertia) includes 25 per cent. for entrained water.

$p^2 = 1{,}150$. 1-Node Frequency Tabulation.

No. of Mass.	$\frac{J}{g}$ lb.-inch-sec.^{2}	$\frac{J}{g}p^2$ lb.-inches.	θ Radians.	$\frac{J}{g}p^2\theta$ lb.-inches.	$\sum\frac{J}{G}p^2\theta$ lb.-inches.	C lb.-inch/rad.	*Twist* Radians.
1	2,890	$3{\cdot}33\times10^6$	1·0000	$3{\cdot}33\times10^6$	$3{\cdot}33\times10^6$	603×10^6	·0055
2	457	·526 „	·9945	·52 „	3·85 „	603 „	·0064
3	3,162	3·64 „	·9881	3·60 „	7·45 „	603 „	·0123
4	234	·27 „	·9758	·26 „	7·71 „	445 „	·0173
5	4,160	4·78 „	·9585	4·58 „	12·29 „	480 „	·0256
6	500	·575 „	·9329	·54 „	12·83 „	10·4 „	1·2315
7	37,700	43·40 „	−·2986	−12·94 „	−0·11 „	—	—

p^2 = (circular frequency in radians per sec.)$^2 = 1{,}150$.

$F = \frac{60}{2\pi}\sqrt{p^2} = 324$ vibrations per minute.

where $\frac{J}{G}$ = Mass moment of inertia in lb.-ins.-sec.^{2}

θ = Amplitude of swing at mass in radians.

C = Stiffness of shafts in lb.-ins. per radian.

N.B.—Without allowance for entrained water, *F = 332 vibrations per min.*

Giving for 1-node vibration:

2nd order occurs at 162 r.p.m.
3rd „ „ „ 108 „
4th „ „ „ 81 „
5th „ „ „ 65 „
6th „ „ „ 54 „

APPENDIX II.

Estimated Stresses.

By Calculation.

In order to estimate the stresses set up in the shafting, the combined turning moment curve shown in Fig. 7 was derived from actual indicator cards taken at 66 r.p.m., the i.h.p. being 652.

By harmonic analysis, using 24-ordinates, the combined harmonic torque co-efficients for various orders were obtained, as under:—

T_n for 2nd order harmonic co-efficient $= 118{,}000 \sin.(2\theta+65^\circ\ 20')$ lb. in.
„ „ 3rd „ „ „ $= 118{,}400 \sin.(3\theta+307^\circ\ 30')$ „
„ „ 4th „ „ „ $= 26{,}800 \sin.(4\theta+72^\circ\ 48')$ „
„ „ 5th „ „ „ $= 4{,}975 \sin.(5\theta+21^\circ\ 6')$ „
„ „ 6th „ „ „ $= 39{,}400 \sin.(6\theta+20^\circ)$ „

Where θ = h.p. crank angle from t.d.c.

From *"Strength of Marine Engine Shafting".

Maximum value of actual vibration torque $= C\times\frac{L}{\overline{L}}$ lb.-ins.

$$\text{Where } \frac{L}{\overline{L}} = \frac{N_1^2}{Q_1\overline{E}^2\times p\times28{\cdot}6\times H}\times\frac{T_n\times\overline{L}_m}{N}$$

$N_1 = 66$ r.p.m.

Q_1 = propeller torque at 66 r.p.m. = mean torque × mech. effcy. = 557,000lb./ins.

$\overline{E}$ = propeller amplitude in frequency table = ·2986 radians for 1 rad. at No. 1 mass.

p = phase velocity $= \frac{324}{60}\times2\pi = 33{\cdot}9$ radians/sec.

$$H = 1+0{\cdot}2\frac{(\overline{L}_m)^2}{(\overline{E})^2} = 3{\cdot}16.$$

T_n = total harmonic co-efficient at critical considered.

$\overline{L}_m$ = mean engine amplitude from frequency table = ·9822 rads.

L = actual „ „

$\overline{L}$ = one radian.

* Paper by S. F. Dorey, D.Sc., Trans. N.E.C. Institution of Engineers and Shipbuilders, 1939.

C = maximum torque given in frequency table with 1 radian at engine $= 12{\cdot}83\times10^6$ lb. ins.

N = r.p.m. at critical considered.

Then actual value of vibration torque $= C\times\frac{L}{\overline{L}}$

$$= \frac{12{\cdot}83\times10^6\times(66)^2}{557{,}000\times(\cdot2986)^2\times33{\cdot}9\times28{\cdot}6\times3{\cdot}16}\times\frac{T_n\times\cdot9822}{N}$$

$$= 364\times\frac{T_n}{N}$$

2nd Order (at 162 r.p.m.).

Max. torque $= \frac{364\times118{,}000}{162}$ = 265,000lb./ins.

Stress in 10in. diam. intermediate shaft $= \frac{265{,}000\times16}{\pi\times(10)^3} = \frac{265{,}000}{196{\cdot}3}$

Stress = 1,350lb. per sq. in.

3rd Order (at 108 r.p.m.).

Max. torque = 400,000lb. ins.
Stress = 2,035lb. per sq. in.

4th Order (at 81 r.p.m.).

Max. torque = 120,500lb. ins.
Stress = 614lb. per sq. in.

5th Order (at 65 r.p.m.).

Max. torque = 27,900lb. ins.
Stress = 142lb. per sq. in.

6th Order (at 54 r.p.m.).

Max. torque = 265,000lb. in.
Stress = 1,350lb. per sq. in.

Estimation of Stresses from Torsiograph Records for 1-node Vibration.

From Figs. 4 and 5.

Max. double amplitude from record at 2nd order = 32mm. plus*
„ „ „ „ „ „ 3rd „ = 21·2 „
„ „ „ „ „ „ 4th „ = 6·3 „
„ „ „ „ „ „ 5th „ = 9 „
„ „ „ „ „ „ 6th „ = 8·7 „

Diameter of shaft at forward station = 10in.

2 mm. of double amplitude corresponds to ±0·00788 radians at point of taking records, *i.e.*, 5ft. aft of thrust block.

From frequency table when $\theta = 1$ radian at free end of system, single amplitude at point of taking records = ·8074 radians.

Maximum torque for amplitude of 0·8074 radians from tables $= 12{\cdot}83\times10^6$ lb.-ins.

Then torque for 1 radian at point of taking records

$$= \frac{12{\cdot}83}{\cdot8074}\times10^6 = 15{\cdot}9\times10^6 \text{ lb.-ins.}$$

Stresses in Intermediate Shaft for Amplitudes listed above.

2nd Order (at 169·5 r.p.m.).

Max. double amplitude recorded = 32 plus* mm. with instrument on stops.

Torque at 154 r.p.m. $= \frac{32}{2}\times\cdot00788\times15{\cdot}9\times10^6$ lb.-in.

Stress, 10in. dia. shaft = 10,230lb. per sq. inch.

Similarly,

3rd Order (at 113 r.p.m.).

Max. torque = 1,330,000lb. in.
Max. stress = 6,780lb. per sq. in.

4th Order (at 84 r.p.m.).

Max. torque = 395,000lb. inch.
Max. stress = 2,010lb. per sq. inch.

5th Order (at 68 r.p.m.).

Max. torque = 565,000lb. inch.
Max. stress = 2,900lb. per sq. inch.

* It will be noted from Fig. 5 that the instrument was hitting the stops on this order, and the revolutions per minute were 154 instead of 166 for maximum resonance with the propeller out of water. In spite of the instrument hitting the stops, the 2nd order frequency was maintained, and the vibration torques and stresses have been calculated assuming resonant conditions as given by the frequency table in Appendix I.

6th Order (at 56·5 r.p.m.).
Max. torque = 546,000lb. inch.
Max. stress = 2,780lb. per sq. inch.

APPENDIX III.

Torques arising from Speed Surges during Racing.

Mean torque from turning moment diagram = 619,000lb. in.

Propeller moment of inertia = $\frac{J_p}{g}$ = 37,700lb. in. sec.²

Total engine moment of inertia = $\frac{J_e}{g}$ = 11,400lb. in. sec.²

Assume that T_m (the mean torque at 66 r.p.m.) remains constant when the engine is racing, *i.e.* there is no wire-drawing of steam supply as engine speed increases, the engine settings and throttle valve openings being unaltered.

Let T_p be the torque in the intermediate shafting.

„ α be the angular acceleration of engine.

Neglecting friction, losses, etc.

Then $T_p = T_m - \frac{J_e}{g}\alpha$ during acceleration.

Also $T_p = T_m + \frac{J_e}{g}\alpha$ during deceleration.

From speed surge diagrams Fig. 6.

Acceleration during period of propeller *leaving* water
α = *5·5 rads./sec.²

Torque transmitted by shaft during acceleration $= T_m - \frac{J_e}{g}\alpha$
= 556,000lb. ins.

With 10in. dia. shaft, Stress $= \frac{556{,}000 \times 16}{\pi \times (10)^3}$ = 2,830lb. per sq. in.

Max. acceleration during period of propeller *entering* water
α = 11·3 rad./sec.²

Torque transmitted by shaft during deceleration $= T_m + \frac{J_e \cdot \alpha}{g}$
= 748,000lb. in.

With 10in. dia. shaft, Stress = 3,810lb./sq. in.

*Note.—It is interesting to compare this figure of 5·5 rad./sec.² with the acceleration which would result if the mean turning moment acted on a rigid system without any propeller resistance torque, then,

$$\alpha = \frac{T_m}{\frac{(J_e + J_p)}{g}} = \frac{619{,}000}{49{,}100} = 12{\cdot}6 \text{ rad./sec.}^2$$

ADDITIONS TO THE LIBRARY.

Purchased.

Modern Marine Engineer's Manual, Vols. I and II. Edited by A. Osbourne. Cornell Maritime Press, New York, $6.00 and $4.00 respectively.

Practical Ship Production. By A. W. Carmichael. McGraw-Hill Book Co., New York, 2nd edition, $3.00.

Boatbuilding—A Handbook of Wooden Boat Construction. By H. I. Chapelle. W. W. Norton & Co., New York, $5.00.

Marine Electrical Installation. By J. F. Piper. Cornell Maritime Press, New York, $2.50.

Marine Pipe Covering. By W. W. Godwin. Cornell Maritime Press, New York, $2.00.

Practical Principles of Naval Architecture. By S. S. Rabl. Cornell Maritime Press, New York, $2.00. 2nd edition.

Handbook of Ship Calculations, Construction and Operation. By C. H. Hughes. McGraw-Hill Book Company, New York, 3rd edition, $5.00.

Modern Marine Electricity. By P. de W. Smith. Cornell Maritime Press, New York, $2.50.

Ship Repair and Alteration. By G. V. Haliday and W. E. Swanson. Cornell Maritime Press, New York, $2.75.

Modern Marine Pipefitting. By E. M. Hansen. Cornell Maritime Press, New York, $3.00.

Pumping and Flooding of Ships. By E. W. Ansell. The Draughtsman Publishing Co., Ltd., 53pp., 18 illus., 2s. net.

This useful pamphlet sets out the general principles and details of fittings, and gives some idea of the methods employed in arranging the various services required for the pumping, flooding and draining of varying types of ships. It is primarily intended for the perusal of junior draughtsmen and others not acquainted with the subject, but in view of the lack of literature on the subject with which it deals, its appeal to a much wider field is apparent.

Descriptions of various types of pump are given followed by a section describing the types of valves used for the efficient control of the pumping and flooding system. Deckplates, sounding, etc. are then dealt with, and a short section on hose connections precedes a much longer one on pipes and fittings. Illustrated descriptions of some typical arrangements for the pumping and flooding of various types of ships are then given. After some general notes on pipe arrangements and the classification societies' rules for pumping and flooding, the pamphlet concludes with examples of two methods of calculation for flooding a magazine.

Presented by the Publishers.

"Radicon" Worm Reducers. Messrs. D. Brown & Sons (Huddersfield), Ltd.

The following British Standard Specifications:—

B.S. 970B.	Memorandum to Consumers and Producers regarding the Standardisation of Alloy Steels, with the object of Alloy Conservation.
B.S. 1099-1943.	Small Fusion-welded Steel Air Receivers.
B.S. 1101-1943.	Pressure Paint Containers.

Abstracts of the Technical Press

Ships' Lifeboats and Davits.

The author points out that this subject has not been discussed by the technical societies for nearly 20 years, during which time the conditions governing the provision of life-saving appliances have undergone important changes. Under pre-war conditions rules were framed on the assumption that when a casualty occurred at sea, near-by ships would be summoned to assist by W/T, and the life-saving equipment was so designed as to make it possible for the boats to be launched safely into the water within an assumed reasonable margin of time, the lifeboats then remaining in the vicinity of the casualty until rescue was effected. War conditions have made it necessary to deal with totally different circumstances, and lifeboats must now be capable of covering hundreds of miles under their own power in order to preserve the lives of their occupants. After reviewing pre-war requirements, the author deals with a number of important points relating to the design and equipment of lifeboats, including their capacity, motor-boats, sails, lifting hooks, disengaging gear, mechanical means of propulsion, equipment, metal boats, availability of lifeboats, davits, the embarkation of passenger and boat drills. He then gives a brief account of what has been done to meet the conditions created by the war, and describes the special provision made for lifeboats in tankers. Other points dealt with in the paper include a note on the equipment of lifeboats with a portable apparatus for converting salt water to fresh drinking water, and the supply of additional equipment in lifeboats.—*Paper by E. W. Blocksidge, "Transactions of the Institute of Marine Engineers", Vol. LV, No. 1, February, 1943, pp. 1-13.*

Steel Lifeboat of Welded Construction.

An old-established firm of boatbuilders and welders in Chiswick have developed an improved design of ship's lifeboat which is capable of being pre-fabricated and mass-produced, and which conforms to the requirements of the Ministry of War Transport. The hull, of 14-gauge C.R.C.A. steel plating, is arc-welded throughout, no rivets of any kind being used, and the only bolts employed being those holding the engine to the bed. The keel is of a special rolled-steel T-section and is bent round the forefoot, forming part of a single component with the stem and stern-post. The hull plating is butt-welded, all external welding being subsequently ground smooth. No stringers or frames of any kind are employed, as the buoyancy tanks from the bilges up virtually form an inner skin with partitions of great stiffness which are claimed to give ample strength without the added weight of framing. It is estimated that this form of construction produces a steel hull weighing 5 per cent. less than a wooden hull. The ring bolts for the lifelines are welded to the hull, and strong and effective grab rails are fitted under the turn of the bilge. These rails add very little resistance to propulsion, provide an easy method of climbing into the boat and constitute a ready means of righting the hull if the boat capsizes and is floating keel uppermost. The rails also act as bilge keels, thus reducing the rolling of the boat in bad weather. They are strong enough to take the weight of the boat when backed. The only timber used in the construction is for the thwarts and side benches, and the boat's equipment includes built-in fresh-water tanks and watertight food lockers. The hull plating is zinc-sprayed by the Schori process and the zinc is given an additional coating of water-resisting melamine resin mixture. The boat's engine is of the standard lifeboat type, and is mounted on a seating built up of steel plates and angles, forming an oiltight tray under the engine, which can be pumped dry.—*"The Journal of Commerce" (Shipbuilding and Engineering Edition), No. 35,895, 25th February, 1943, p. 8.*

Rejection of Erren System of Submarine Propulsion by British Admiralty.

In reply to a question raised in the House of Commons concerning the rejection by Admiralty experts of the Erren system of propulsion for submarines, the First Lord of the Admiralty recently stated that close attention had been given to the Erren invention since 1930 and that various proposals were submitted to the Admiralty by Erren or his associates between 1932 and 1938. Admiralty representatives visited Germany to inspect a high-pressure electrolyser plant which had been developed by the Erren Engineering Company, and went to Messrs. Beardmore's works to see an engine running on the Erren system. In 1938 it was finally decided, after very thorough investigations conducted with the assistance of an eminent outside engineering authority, that none of the proposals made to the Admiralty by the Erren Co. could with advantage be adopted for submarine propulsion, and the Erren Engineering Co. were so informed in August, 1938. In view of the number of highly technical considerations involved it would be neither desirable nor feasible to discuss the reasons which caused the Admiralty to come to such a decision. Press reports to the effect that the Erren system is being employed in German submarines and that a captured U-boat was so equipped, were entirely incorrect.—*"The Journal of Commerce and Shipping", No. 35,889, 18th February, 1943, p. 3.*

"Powder Metallurgy: Its Products and Their Various Applications".

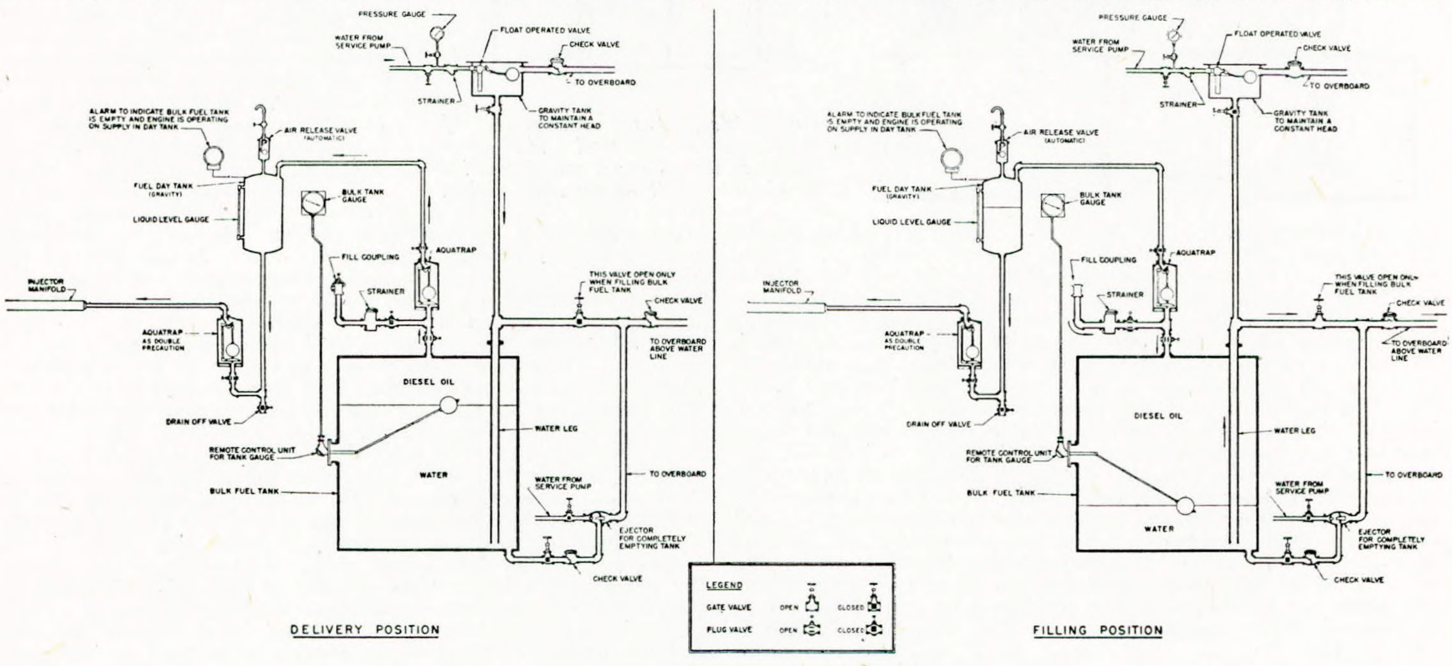
The author of the paper bearing the above title defines powder metallurgy as the art of making metallic objects or masses by moulding powdered metals. Its increasing popularity depends both upon the possibility of manufacturing metal parts having properties which cannot be achieved in any other manner, and by the fact that it is often more convenient or cheaper to use the technique than other methods of manipulation. The best known article manufactured from powders at the present time is the porous bronze bearing made by compressing a mixture of copper and tin powders and heat-treating the compacts in a protective atmosphere. Other well known materials made by powder metallurgy include the hard metal carbide cutting tools, electrical contact materials, tungsten filaments and the Alnico magnet. In America, various iron parts, chiefly for use in motor-cars, are made by cold-pressing and hot-pressing techniques. The various advantages of powder metallurgy are described and considered by the author, who concludes his paper by citing a number of examples in which the technique would be of interest in the field of marine engineering. He suggests that it could be applied to the manufacture of piston rings for feed pumps, Diesel engines and reciprocating steam engines, as well as for packing rings, etc., for valves and seatings for feed pumps, etc., for safety valves, steam traps and similar fittings; for turbine blades; and for the bushings and bearings of light auxiliary machinery, small cams, tappets and pinions. The electrical field embraces many possibilities, such as commutator segments, brushes, contacts, etc., but the largest field is, perhaps, in the manufacture of instruments where numerous small components are required, cams, tappets, screws, bearings, magnets, etc. The radio and telephone trades already utilise considerable quantities of powders in the manufacture of cores and transformers.—*Paper by W. D. Jones, M.Eng., Ph.D., read at a meeting of the N.-E. Coast Institution of Engineers and Shipbuilders, on the 19th February, 1943.*

Electrode Economy.

Figures recently published in the United States show that failure to use the last available 4in. of a welding electrode stub can mean the loss of 25 ships a month. Although every 18-in. electrode can be burned down to a stub of 2in. or less, welders sometimes throw away 3¾in. and some even discard as much as 6in. The use of merely another 1¾in. on the average stub would be sufficient to add another 5½ ships to the 60 or so cargo vessels of arc-welded construction now being being turned out every month. Each arc-welded cargo vessel requires approximately 175,000lb. of electrodes, and if this figure is multiplied by 60 (the approximate number of arc-welded ships launched every month), the resulting figure is 10,500,000lb. In other words, every inch of welding electrode is vital to the Allied shipbuilding programme. If the welder uses only 14¼in. of an 18-in. electrode, the remaining 3¾in. is thrown away as unusable, but it has been found that he can very well use another 1¾in.—enough to do all the welded construction on 5½ vessels. Similarly, if he discards a 6-in. stub, he is throwing away enough electrode material to weld 25 vessels. It cannot be over-emphasised that every welder should use electrodes right down to the last 2in., and that his slogan should be "Burn Another Inch and Build Another Ship". If electrodes are not used down to a 2-in. stub, the number of interruptions in the actual welding time is greater, and the operating factor is reduced, while the labour cost per lb. of electrode deposited is increased. Recent surveys indicate that the change from one electrode to another requires an average of 21·6 seconds when the welder does his own cleaning of the deposited bead.—*"Shipbuilding and Shipping Record", Vol. LXI, No. 8, 25th February, 1943, p. 188.*

Transfer of Fuel by Water Pressure.

The use of sea water for "pressing up" fuel-oil tanks has, so far, met with little favour in merchant vessels, despite the fact that this method of regulating the supply of fuel from the tank or tanks in actual use has been employed in submarines for many years past. A fuel-oil transfer installation of this type for Diesel-engined merchant vessels and tugs has, however, now been developed in America, where it is known as the Aqua System. The use of sea water in this system is not in any way similar to a condition of D.B. or deep tanks partly filled with fuel and contaminated by a relatively small amount of sea water inadvertently admitted to the tank through leaks or possibly left there after the tank has been filled with sea-water ballast. In the partly filled tank the fuel and water swirls and lashes its way through orifices in baffle plates, over and around the vari-shaped structures of steel inside the tank and in this great commotion, which prevails as long as the ship is rolling, such water as is present becomes thoroughly mixed with the fuel oil. A similar tank containing the same amount of water *under* the fuel, with the upper surface of the latter pressed "hard up" against the top of the tank, will follow the ship's movements with the same gentle rolling motion without agitation. The water will remain at the bottom, there will be no stratum of emulsion above it and no water will go into suspension in the fuel. More water can be pumped in and clean fuel will overflow at the top of the tank until water has completely displaced the fuel. The Aqua System can, it is claimed, be appied to all grades of fuel sufficiently fluid under all normal operating conditions to flow freely without heating. The system is, therefore, not suitable for use with the bunker oil commonly burned in steamships or with the low-grade fuel oil employed in certain motor vessels. In the U.S. Navy a modification of the Aqua System is also used in handling petrol. The sponsors of this method of handling fuel claim that it possesses special advantages for tugs, as it enables the trim to be maintained constant for optimum propeller efficiency without regard to the location of the fuel tanks or the amount of fuel remaining on board. In passenger and cargo vessels similar control of trim has its advantages. As the tanks once empty of fuel can then be emptied of water, it is clear that the use of the system does not affect the cargo-carrying capacity of the ship. Although the usual fuel strainers and filters are retained with the Aqua System, the latter tends to clean the fuel of impurities and keeps it in better condition for Diesel consumption than a conventional storage tank. Most fuels contain impurities in the form of solids or liquids heavier than the fuel, and these impurities settle out of a relatively quiet body of oil floating above an equally quiet body of water; they then remain in the latter and do not mix with the fuel. On the other hand, impurities in agitated fuel tend to mix with it. A further advantage of the Aqua System is the simpler pumping equipment required. Since water is pumped in to displace fuel, a centrifugal pump is all that is needed for fuel transfer, in fact a jacket-cooling pump or sanitary service pump can be employed to provide the requisite water pressure for fuel transfer, thereby eliminating the need for special fuel-transfer pumps. A constant head can be maintained on the fuel tanks by automatic control of the water pressure, thus permitting direct connections from the main fuel tanks through filters to the fuel pumps on the engines of all motorships not employing centrifugal purifiers to clean the fuel. Leaking rivets in fuel tanks would permit escape of the water under the fuel, whereas in the conventional tank arrangement such leaky rivets allow water to leak into the tanks and result in water-fuel mixtures reaching the engine. A schematic arrangement of an installation employing the Aqua System is shown in the accompanying diagram.—*"Motorship", Vol. XXVIII, No. 3, March, 1943, pp. 202-203.*



Schematic drawing of Aqua system showing delivery and filling positions.

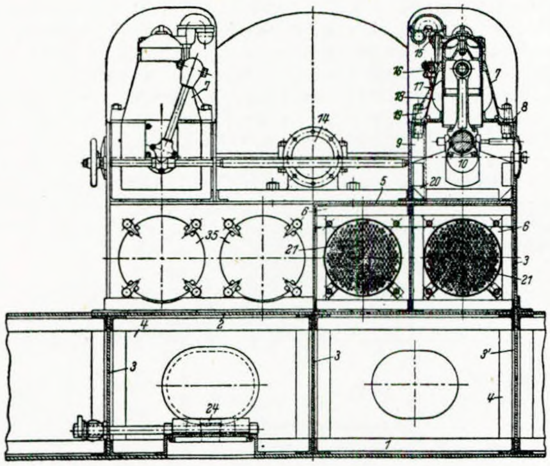
"Fast or Less Fast Ships".

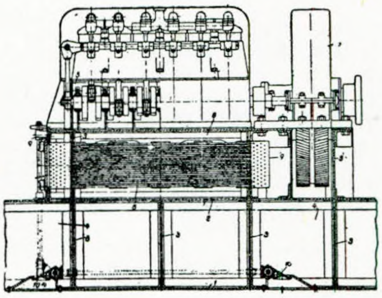
The author of the short paper bearing the above title referred to his presidential address in the course of which he had dealt with the time and labour required to build (*a*) motorships of 10,300 tons d.w. having a service speed of 11¾ knots on a fuel-oil consumption of 10 tons/24 hrs., and (*b*) motorships of the same dimensions, but with finer lines and heavier machinery, having a d.w. capacity of 8,300 tons and a service speed of 15 knots on a fuel consumption of 27 tons/24 hrs. Carrying the comparisons a stage further, he now assumed that each type would be six months on the stocks, but that the 11¾-knot ships would take two months to fit out, and the 15-knot ships three months. At the end of three years from the inception of the alternative programmes, 58 vessels of 10,300 tons d.w. would be delivered, as against 23 ships of 15 knots and 8,300 tons d.w. During a similar period the slower vessels would carry 6,162,200 tons of cargo as against the 2,142,800 tons which the faster ships could transport, assuming round voyages of 10,000 miles, or 4,019,500 tons against 1,344,700 tons on 20,000-mile voyages. Summarising his calculations, the author showed that at the end of two years 26 ships of 11¾ knots and eight ships of 15 knots would have been delivered instead of 48 ships of 11¾ knots under the present programme. At the end of three years there would be 26 ships of 11¾ knots and

18 of 15 knots instead of 72 ships of 11¼ knots.—*Paper by J. Ramsay Gebbie, O.B.E., B.Sc., read at a meeting of the N.-E. Coast Institution of Engineers and Shipbuilders, on the 26th February, 1943.*

Modern Geared Steam Reciprocating Machinery.

An article in *Schiffbau* states that a recent German patent granted to Dr. Lentz covers the application of high-speed, geared, multi-cylinder-in-line steam engines in the space-and-weight saving lay-out shown in the accompanying diagrams. The latter are drawn to different scales and are obtained from different sources, but appear to indicate that the new machinery arrangement is really an alternative to Dr. Lentz's advanced star or radial uniflow engine. The cylinder castings in the new arrangement are carried on extensions of the frames (*3*) of the double bottom (*1* and *2*), these extensions serving to form a box-shaped structure which houses the twin-engine crankcase (*5*), the condenser body (*6*), and the reduction-gear casing (*7*). The engine exhausts are conveniently carried down into the integral condenser, which comprises the tube nests (*8*) and water chambers (*9*), these latter being connected to the sea inlet and discharge valves (*10*) by pipes (*4*) passing through the double-bottom spaces. It is claimed that the employment of high-speed, reversible, uniflow steam engines ensures good starting and manœuvring, as well as freedom from vibration. Each group of cylinders can be used independently, so that the breakdown of one group does not put the entire propelling-machinery installation out of action. Provision is also made for isolating damaged portions of the condenser plant, if required.—*"The Marine Engineer", Vol. 66, No. 788, March, 1943, p. 73.*





Dr Lentz's proposed arrangement of multi-cylinder vertical high-speed steam reciprocators is an alternative to his now well-known star engine.

Geared Turbine Production in America.

In view of the statement made by Admiral Land in the course of his presidential address to the American Society of Naval Architects and Marine Engineers that marine turbine reduction gearing was likely to prove a "bottleneck" in the U.S. Maritime Commission's programme for installing exhaust turbines in the improved EC-2 cargo vessels, it is interesting to note that the Westinghouse Company's new marine steam turbine works (which have special equipment for cutting gears) were built, equipped, and put into operation in about ten months at a cost of some $26,000,000. The first set of geared turbines from this new factory is now about to be delivered, some four months ahead of schedule. It is a set of two-casing high-pressure geared turbines. The works are expected to attain their contract output capacity four months after the delivery of this set of turbines, and it is anticipated that that output will be exceeded by one-third by next September. The works are producing turbine machinery for various warships and auxiliaries, as well as for merchant vessels. A number of land turbines are also being constructed.—*"The Marine Engineer", Vol. 66, No. 787, February, 1943, pp. 31 and 50.*

Vibration of Turbine Plant.

Vibration in a turbine is often due to the uneven deposit of dirt on the moving blades or to partial stripping of the blading or shrouding caused by priming of the boilers or to a failure to function of the automatic trip gear. A more insidious form of vibration is that which develops slowly due to excessive clearance in the bearings or to excessive movement of the entire bearing in its housing. The necessary adjustments must, of course, be carried out with great care, and any reduction of clearance should be followed up by a similar adjustment of the pedestal cap. Coupling wear is a common cause of turbine vibration and should, therefore, be looked for. Any shaft revolving at high speed will vibrate heavily if retarded in any way, as, for instance, by a segment of packing which has stuck solidly in its housing, or if unequal expansion of the turbine casing has caused the latter to buckle and the diaphagms touch the rotor. The utmost care must be taken to ensure that any pipes coupled to the turbine casing are adequately supported and cannot distort the relatively flimsy casing by reason of their inherent tendency to expand and contract while the turbine is in operation. One of the most formidable types of vibration to which turbines may become liable is of a recurrent nature and requires treatment on its merits, due regard being paid to the evidence which is always present if only sought with sufficient diligence. The writer experienced such a case with a 1,500-kW. mixed-pressure direct-coupled turbo-alternator in which a heavy thudding noise and vibration sometimes developed when it was being run up to speed. After a few hours on load the noise completely disappeared, but at times it came on when the plant was changed over from using L.P. steam to using H.P. steam. Then it would die away, but was invariably present when steam was shut off and the turbine allowed to come to rest. An examination of the turbine rotor showed that all the blading was in good condition, the alignment correct and the coupling in good order, but a prolonged search revealed that some of the wheels were loose on their keys; when the plant was hot they tightened up, but any change of load caused them to "bump" on their keys. In another case, trouble was given by a turbine-driven geared generator after the plant had been on load a few hours. The independent balance of the turbine rotor, generator rotor and gears were all checked statically and independently, but the combined unit would not run more than a few hours before a terrific vibration arose in the gears. After many exhaustive experiments had been made and many theories tried and proved useless, it was at last found that the whole trouble was caused by a relatively simple fault. The main reduction gear wheel, of fairly large diameter, was a spoked drum keyed to a steel shaft. After a period of running the wheel accumulated a large quantity of lubricating oil inside its periphery in one particular place where the casing formed a natural reservoir. Draining holes were thereupon drilled to clear this area and the trouble disappeared. Alignment is in itself an important factor, and any search for vibration should always include a check of the alignment and level to make perfectly sure than no foundation settlement has developed. Most modern turbines drive their governor gear—and sometimes their lubricating-oil pumps—by means of a reduction gear from the main shaft, and shaft vibration is occasionally set up by bad contact of the gears or by worn teeth.—*W. W. Alcroft, "The Power and Works Engineer", Vol. XXXVIII, No. 440, February, 1943, p. 38.*

The Effects on Twin-screw Propulsion of Some Changes in Bossing Design.

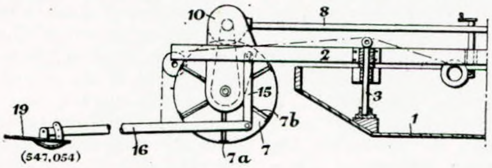
The paper gives some further results obtained in a research on the propulsion of twin-screw ships which is being carried out in the William Froude Laboratory. One of the chief results described in a paper presented to the I.N.A. in 1941 (*see abstract on p. 116 of* TRANSACTIONS, *September, 1941*), was the marked loss of propulsive efficiency at relatively small immersion of the shaft centres when using normal bossings compared with values obtained when using A-brackets. A part of this paper deals with new experimental work on this problem, and it is shown that such loss can be reduced by lowering the forward end of the bossing. The rest of the paper relates to conditions in which the bossings were well immersed and describes the effects on propulsive efficiency of considerable changes in the form and extent of the bossing. It is shown that only small improvements in efficiency are likely to be obtained compared with that given by normal bossing designed to follow the stream flow determined by stream vane tests with the naked hull.—*Paper by G. Hughes, B.Sc., Ph.D., "Bulletin of the Liverpool Engineering Society", Vol. XVI, No. 7, February, 1943, pp. 4-26.*

Determination of Sea Speed.

The U.S. War Shipping Administration recently issued some new regulations for the determination of sea speed on the basis of the method outlined in Admiral D. W. Taylor's *Speed and Power of Ships,* which is the standard American work on the subject. The regulations lay down a series of arbitrary factors for determining the relation between the effective horse-power—*i.e.,* the h.p. required to tow a ship—and the shaft horse-power, the difference between these powers being the power lost by the propeller and in other ways. In the case of a "Liberty" ship with engines of 2,500 i.h.p., the effective h.p. calculated from these formulæ is 1,070, which corresponds to a speed of 10·5 knots if Admiral Taylor's curves are used, whereas in the case of the "Ocean" class ships built for the British Government, which are of similar dimensions, an effective h.p. of 1,070 corresponds to a speed of 10·9 knots. This difference is due to the improvements which have been made in the design of the lines of ships' hulls in the 30 years which have elapsed since Admiral Taylor carried out his systematic experiments. The 2,500 i.h.p. of the British vessels suffices to give them a trial speed of 12 knots, so that the American formula, in this instance, allows for a reduction of 1½ knots for the average sea speed.—*"Fairplay", Vol. CLX, No. 3,120, 25th February, 1943, p. 266.*

Water-weed-cutting Launch.

A launch designed for cutting weeds in shallow and narrow waterways has been developed and patented by an engineering firm in the Midlands. Referring to the accompanying diagram, the hull



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of the launch has a flat bottom *(1)* and a shallow draught of about 4in. The mechanism for propulsion, steering and weed cutting is mounted on a separate structure independent of the hull, and consisting of two parallel bearer beams *(2)* which carry a depending pivot spindle *(3)* engaging two bearing blocks fixed to the hull. The rear end of this longitudinal bearer carries an I.C. engine, while the forward end, overhanging the bows of the craft, carries the propelling and cutting mechanism *(7)*, comprising a combination of cutter blades *(7a)* and paddle blades *(7b)* for cutting the weeds. The paddle is driven by a shaft *(8)* mounted above the longitudinal frame *(2)*, from the power unit. The shaft drives through a reversing gear-box *(10)*, from which a chain drives a sprocket wheel on the paddle shaft. An eccentric is mounted on the other end of the paddle shaft and oscillates a lever *(15)* pivoted on the bearer and connected to the arm *(16)* of an oscillating weed cutter. The paddle blades *(7b)* are set at an angle of 30° to the axis of the shaft, so that the weeds cut will be ejected outwards to the sides of the waterway. The oscillating cutter blades *(19)* are arranged centrally in advance of the paddles, so that the weeds between the latter are cut by them. The tips of the paddle blades sweep to a variable depth below the bottom of the boat. The oscillating cutters *(19)* retard the boat and prevent the paddle from merely riding on the bottom of the waterway, thereby ensuring that the paddles turn at a peripheral speed greater than the forward speed of the boat. Any loose silt lodged in the weeds will also be removed laterally by the paddle blades. The power unit is mounted in a cradle at the rear of the longitudinal bearer, and is supported by a wheel which rides on a runway at the bottom of the boat, the position of the engine being at about the centre of the hull to give a balanced weight distribution. A steering handle is fixed to the rear of the longitudinal frame, which is swung on the vertical spindle *(3)* for steering the boat. The cutters therefore always follow the steering. A rudder is also fitted at the opposite end of the hull and can be fixed in a central position, but when the steering is controlled by this rudder, the frame *(2)* is locked in a central position. The rudder and the propelling unit can also be ganged together by means of cable interconnectors.—*"Engineering", Vol. 155, No. 4,031, 16th April, 1943, p. 320.*

Types of Ballast.

It is reported that the additional top weight, due to the armament, of the American "Liberty" ships, has reduced the stability to such an extent as to necessitate the carriage of permanent ballast. In future designs the beam is to be increased to compensate for these additions. According to the New York *"Journal of Commerce",* ballast in the form of cobblestones, rock, sand, gravel and poured cement, was tried by shipbuilders, but the stowage factor of these materials was so high that much of the ship's cargo capacity was wasted. As it is not possible, under present conditions, to use ballast in the form of pig iron or pig lead, the Maritime Commission and certain private concerns carried out experiments which have resulted in the production of ballast in the shape of a dense cement aggregate in a block or monolithic form. This material has a density of about 200lb./cu. ft., *i.e.,* about twice that of ordinary cement, and is poured directly into the ship's holds, tanks or bilges, after insulation has been provided to prevent adhesion to the ship's structure. The poured material is divided into separate sections, so that it can be easily removed when repairs have to be made to the ship's structure. This is an important precaution, as anyone who has had the misfortune to be associated with the cutting-out of cement before repairs could be made will bear out. In some cases it has proved necessary to resort to blasting before the cement which had been used as a temporary repair could be removed. Ballast of the kind described above is sometimes made up in the form of bricks which can be closely stowed and are easily removable, and in this country a type of ballast with properties similar to that now evolved in America has been available in the form of bricks for some time past.—*"Fairplay", Vol. CLX, No. 3,120, 25th February, 1943, pp. 266 and 268.*

Extra Ballasting.

In view of the more arduous service which the standard tramp ships built in this country are being called upon to carry out at the present time, particularly voyages in waters that would not normally be entered in winter time, it has been found necessary to make provision for the carriage of increased water ballast. In some of these vessels, the increased ballast capacity amounts to as much as 1,000 tons or more. The orthodox way of providing the extra space necessary for this purpose would have been to increase the capacity of the double-bottom tanks by raising the tank top. This, however, would have meant a stiff ship, and the extra ballast capacity was therefore provided for in side tanks, where, incidentally, it is more convenient to stow liquid than dry cargo, and the opportunity was taken to strengthen the structure of the ship as well as ensure a smoother performance in the ballast condition. Adequate ballasting is of vital importance for safe convoy operation, and in the case of some vessels of pre-war construction, it has been found necessary to supplement the water ballast with sand ballasting, an arrangement wasteful of time and man-power. Although no information is available concerning the feasibility of improving ballast performance by adopting a propeller specifically designed to increase the draught aft when working, there is no doubt that the extra water-ballast capacity provided in the new cargo ships by ensuring complete immersion of the propeller at all times (except in very heavy seas), enables these vessels to maintain an appreciably higher speed in the light condition —and ballast voyages outwards from this country, under present conditions, are far more common than in times of peace. Some owners building new tonnage under licence were, at first, somewhat perturbed regarding the commercial aspects of an increase of 60 to 70 per cent. in the normal ballast capacity, but their subsequent experience with the ships has caused them to take a more favourable view of this innovation.—*"The Shipping World", Vol. CVIII, No. 2,591, 10th February, 1943, p. 161.*

The Effect of Machinery Torque Variations on Propulsive Performance.

The author explains how variations of the torque of a reciprocating engine cause the propeller to rotate with an irregular motion which is a combination of the torsional vibration and the motion with the shaft assumed rigid. He points out that the fluctuations of propeller speed combined with the wake may cause the blade angle of attack to reach dangerous values, and that the efficiency value is reduced by damping, this reduction being appreciable near the critical speeds. A considerable thrust fluctuation results from the irregular speed of rotation, and this may, in some cases, account for the axial vibration of the shaft. Certain relations are obtained, giving the speed and thrust fluctuation and loss of energy for a simple two-mass system, with damping applied at the propeller and the engine. The author describes how any system of concentrated masses can be reduced into an equivalent two-mass system, and gives a method for obtaining the damping factor from published model experiment data. A number of examples are worked out for the purpose of comparing various types of engines. The concluding portion of the paper is devoted to an explanation of the momentum theory with a fluctuating thrust; although this is only of theoretical interest, the author suggests that it may be found useful in understanding how energy is dissipated by damping at the propeller.—*Paper by Lieutenant (E) E. Panagopoulos, B.Sc., R.H.N., read at a meeting of the N.-E. Coast Institution of Engineers and Shipbuilders, on the 5th February, 1943.*

Increasing Refrigerating Capacity.

An article by T. Mitchell in a recent issue of the American technical periodical *Power* deals with the various factors influencing refrigerating performance and the extent to which existing plant can be increased to meet present-day demands. The fact that refrigerators operate by continual recirculation means that balance must be maintained, any major alteration to one part affecting the operation of others. Weak spots in the plant will limit its overall performance, *e.g.*, coils and coolers may be unable to absorb heat rapidly enough; wasteful clearance, poor valve action or low suction pressure may reduce compressor efficiency; the condensing pressure may run too high; or there may be unseen leaks. These and other defects are discussed and remedies suggested. The capacity of a refrigerating machine increases rapidly as the suction pressure is raised, *e.g.*, a 66·5-h.p. machine producing 25 tons at zero suction pressure can deliver 38·5 tons at 20lb./in.^{2} suction pressure with only an extra 27·5 h.p., so that a 54 per cent. increase in output is obtained with an increase of only 42 per cent. in the power absorbed. Old-fashioned coils were too long and often involved an excessive pressure drop before the suction gas reached the compressor. A coil giving a heat transfer of 2 to 3 B.Th.U./ft.^{2} per °F./hr. with natural air circulation can be made to transfer up to 18 B.Th.U. under the right conditions with forced air flow. Attention is drawn to the importance of keeping the coil surfaces free from frost, and to the economy of pre-cooling the material to be refrigerated. Coils used to cool water must be kept fairly free from ice between the rows of pipes for maximum performance. Every additional 10lb./in.^{2} of head pressure reduces the capacity of an ammonia machine with a 20lb./in.^{2} suction by about 1 per cent. and raises the motor load by about 3 per cent. Hence the importance of looking to the discharge pressure. Inadequacy lies more often in the condensers than in the compressors connected to them. The effective area of a shell condenser can often be increased by running water over the outside of the shell as well as through the tubes. The author stresses the importance of using a purger to eliminate air and other non-condensable gases from the system. The clearance of horizontal machines should be checked carefully. Automatic controls promote economy, as steady operation is an important factor in saving power. It is more economical to run one compressor continuously with a high suction pressure than to operate two machines at low temperature in the daytime only.—*"The Power and Works Engineer", Vol. XXXVIII, No. 440, February, 1943, pp. 46-47.*

An American Floating Dry Dock.

An improved design of floating dock has recently been developed in America and patented in this country. Referring to the accompanying diagrams (Fig. 4), the pontoon of the dock has an arcuate or sloping bottom and a longitudinal airtight buoyancy chamber. The dock comprises a pontoon *(2)* supporting side walls *(3)* and having an air chamber *(4)*. The weight of water displaced by the chamber is slightly less than the weight of the non-buoyant dry-dock structure itself when under water, thus providing a small margin, sufficient for submerging the dock. The pontoon section, having a bottom *(6)* as shown in the diagram, allows of cut-away corners, thereby reducing the bending moments set up as the ship is lifted to clear the water, while the central buoyancy chamber reduces the combined stresses when the wings are first pumped out. The outer shell plating is supported by channel stringers *(7)*, some of which are shown. These in turn are supported by the top and bottom members *(8, 9)* and by interior bracing connecting transversely spaced points in the pontoon. The water pressure is transmitted with a resultant arch compression to the members *(9)* and the pontoon structure. When the dock is in its fully raised position, the water level *(10)* outside is just below the level of the dock floor. When the ship is clear of the water and the outside level is that indicated *(11)*, the water in the pontoon is at the level shown by the dotted line *(12)*.—*"The Motor Ship", Vol. XXIV, No. 279, April, 1943, p. 34.*

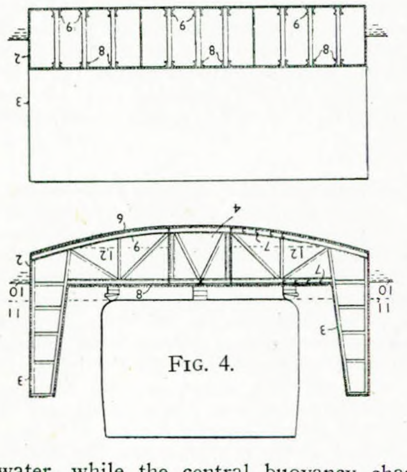


Fig. 4.

Nickel-zinc Galvanizing.

A new process for covering steel sheets and wires, known as corronising, is claimed to possess advantages over ordinary zinc galvanizing. The process, which was recently described in a paper presented at a meeting of the American Electro-platers' Society by R. Rimbach, consists of the electrical deposition of thin successive layers of nickel and zinc on the surface of the article to be covered and then annealing it for several hours at a temperature of 680° F. in order to effect a fusion between adjacent layers of the deposited metals. It is usually found that the innermost surface of the combined deposit remains as pure nickel and that various zinc-nickel alloys are superimposed on it. The process is applicable to both ferrous and non-ferrous articles and is said to afford protection from ordinary atmospheric corrosion quite as efficiently as ordinary zinc galvanizing. The coating is uniform in thickness, adherent, smooth and ductile, whilst the saving in the amount of zinc required is no less than 87 per cent. No mention is made of the resistance of a corronised surface to the corrosive effect of sea air or sea water, nor are any figures given regarding the cost of the process as compared with that of ordinary zinc galvanizing.—*"Shipbuilding and Shipping Record", Vol. LXI, No. 6, 11th February, 1943, p. 123.*

The McKee Wear Gauge.

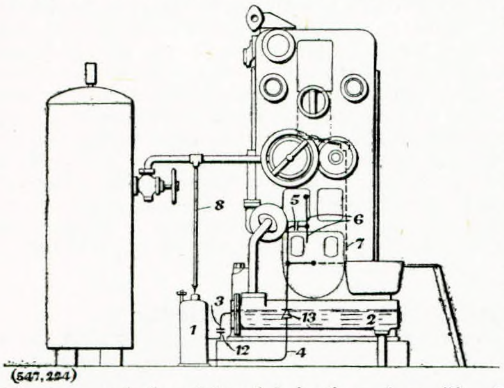
The McKee wear gauge, which is claimed to be capable of measuring wear of as little as 15 millionths of an inch on metallic surfaces, is a recent development of the American Instrument Company, Washington, and forms the subject of a descriptive article in *The Iron Age*. The gauge is primarily designed to measure the amount of metal removed during grinding, honing or lapping, or as a result of ordinary wear, and it is already being used in the manufacture or for rechecking the sizes of cylinders, crankshafts, bearings and other components. The operation of the instrument is based on the principle of measuring the width of a precision-impressed, diamond-shaped, pyramidal indentation, the depth of which is a function of the peak angle of the indentor point; consequently, the depth of the indentation can be determined by measuring the width of the base before and after wear has occurred, and the amount of wear calculated. In some cases, it is desirable to know when an exact amount of wear has occurred, as, *e.g.*, during a grinding or honing process. For this, the instrument is used to make an indentation of predetermined depth calculated from the amount of metal to be removed; the surface is then ground or honed only until the indentation is entirely removed, which indicates that the predetermined depth has been reached. The particular advantage of this method is that it provides an indication of wear only, whereas the usual measurements of changes in dimensions of pistons or cylinders, etc., do not distinguish local wear or the differential between wear and such distortion as may have occurred during the test. The instrument itself is not only accurate within a maximum measuring limit of 0·0014in. and a minimum of 15 millionths in., but is also flexible and capable of application to wear measurement on the inside of cylinders at any place within 10in. from the ends, and at any accessible spot on the outer surfaces of flats, cylinders or spheres.—*"Foundry Trade Journal", Vol. 69, No. 1,381, 4th February, 1943, p. 99.*

An American Marine Engineer's Gallantry.

The 63-year-old chief engineer of an American oil tanker, Mr. A. Friberg, was recently awarded the Merchant Marine Distinguished Service Medal for courage and devotion to duty when his ship was torpedoed by an enemy submarine in the Atlantic. The tanker was struck by two torpedoes on the 8th January, as the result of which a number of fires broke out on board. The chief engineer was at the entrance to the W/T room at the time and immediately realised that unless the fires could be brought under control by means of the steam-smothering fire-control system, the cargo oil tanks were likely to explode, with disastrous consequences for the vessel's crew. The steam-smothering system could, however, only be put into operation from the engine room, where the master valve was located, but the deck between Mr. Friberg and the E.R. entrance was swept by flames feeding on spreading oil. It was impossible to walk through the raging fire, so the chief engineer swung himself over the rail and worked his way aft along the wallowing ship's side by clinging to the rails. He reached his goal despite the flames, smoke and intense heat and opened the valve. When the flames had been sufficiently checked the entire crew was able to leave the doomed ship. Mr. Friberg sustained severe injuries and has, in consequence, had to retire from sea service.—*"Motorship", Vol. XXVIII, No. 2, February, 1943, p. 150.*

Lubricating-oil Priming System for I.C. Engines.

A well-known British firm of oil-engine manufacturers have developed and patented a system for priming with lubricating oil internal-combustion engines which are started by compressed air. Referring to the accompanying diagram, the priming oil container (*1*) is permanently connected to the engine sump (*2*) by a sump pipe (*3*) led from below the oil level in the sump to a lower level in the container. A priming oil pipe (*4*) is led from near the bottom of the container (*1*) through a N.R. valve (*5*) to the main lubricating system (*6*) leading to the filter box and



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bearings and the piston lubricating pipes (*7*). The container (*1*) is filled up to the sump oil level at atmospheric pressure in the container. A branch (*8*) of the pipe supplying air from the starting-air receiver to the engine applies pressure to the top of the oil in the container when the receiver outlet valve is opened. A partial N.R. valve (*12*) in the pipe (*3*) reduces the pipe area when the pressure in the container (*1*) is greater than that over the oil in the sump. This valve has cuts across the seat to allow a small quantity of oil to be forced back into the sump, followed by compressed air which tends to keep this oilway clear. The pressure of the air in the oil container is thus reduced, and this prevents the oil from being entirely exhausted from the container and air entering the lubricating system. As a further precaution, a pressure-reducing valve (*13*) is fitted in the priming oil pipe, so that unless the pressure in the priming oil container is above the setting of the reducing valve, oil will not flow into the lubricating-oil system and air below this pressure will be exhausted into the sump. When the receiver outlet valve is closed, the air pressure in the supply line leaks away and the priming oil container is re-filled with oil by gravity through the sump pipe and is thus ready for a fresh engine start. Air is not actually admitted to the engine cylinders until the engine starting valve is opened, so that during the short period between the opening of the receiver outlet valve and the operating of the engine starting valve priming of the engine is taking place. By these means the pistons are automatically primed for a short period before the engine actually starts and for a short period after starting.—*"Engineering", Vol. 155, No. 4,030, 9th April, 1943, p. 300.*

Cargo-hold Dehumidification.

The three main items of a present-day cargo-hold dehumidification system are: (*1*) an air-drying unit consisting of a silica gel moisture-absorbing plant located in a compartment adjacent to the main engine room of the ship; (*2*) a separate fan-and-duct system for each cargo hold; and (*3*) a set of recording instruments for registering the dew point and temperature both of the outside atmosphere and of the air inside the holds. The two operational phases of the system are: (*a*) ventilation with outside air as long as the atmospheric dew point is lower than that of the air in the holds; and (*b*) recirculation of air in the holds supplemented by the injection of dry air from the air-drier unit at all other times. As regards (*1*), the silica gel unit employed in American practice possesses outstanding merits, insofar as it is easy to operate and takes up little space. Owing to the use of a dry moisture-absorbing agent, the danger of spilling, necessarily ever-present in liquid absorber units, is entirely absent. As salt spray and dust are injurious to the silica gel absorber beds, special protection is provided for these in the form of filters both for the air supplied for reactivating the beds and for that entering the absorbing beds. The absorbing power of silica gel depends entirely upon the internal surface of the material, and if the apertures and interstices leading to these internal surfaces are allowed to become choked with dust and salt particles, the absorbing power will be greatly reduced. A special type of high-efficiency filter is, therefore, employed with silica gel units, and these filters must usually be renewed twice a year. It is reported that when using such filters, no diminution of the absorbing power of the silica could be observed after two years of operation at sea. The employment of a central dehumidifying plant must necessarily involve the installation of an expensive system of dry-air mains extending over roughly two-thirds of the length of the ship. An adequate system of air trunking must also be provided in the holds *without making excessive inroads upon the cargo space.* The design of such a system of air trunking must, therefore, involve a number of difficult problems. Attempts to improve air circulation by the expedient of a special storage arrangement of the cargo in the holds, have proved impracticable. Much, therefore, remains to be done before dehumidification systems can be inexpensively and yet effectively adapted to existing ships. The use of separate dehumidifying units for the various holds might prove feasible in some cases, but prospects in this direction do not appear particularly favourable, as the multiplication of air-drying units may easily lead to an increase in the cost of the plant out of all proportion to the advantages gained by the elimination of the dry-air trunking. Furthermore, the increase in the amount of attention required in the case of individual plants would weigh heavily against their use, unless it were possible to overcome this drawback by the employment of fully-automatic control equipment. The designers of the American central unit system of cargo-hold dehumidification have expressed doubts concerning the feasibility of operating ventilating and recirculating systems by such means, because it is frequently necessary for the officer in charge to exercise his own judgment in anticipating future climatic conditions, sometimes weeks in advance, and this cannot be done by recording instruments alone. It is too late to start the dehumidification system when the critical time arrives, because the capacity of the plant is not large enough for this, while if the machines were made large enough to deal with the situation when it actually arose, the cost of the whole installation would become prohibitive both as regards the capital expenditure entailed, and the space and weight requirements involved. The above remarks, however, apply only to ships in transit and not to cases in which a dehumidification unit is to be used to ensure permanent dryness in vessels laid up for prolonged periods, and where automatic operation is practicable. A small portable unit of this type has been developed for use in the U.S. Navy. It can be passed through a standard W.T. hatchway of a destroyer and is fitted with a special timing device which automatically changes the operation of the machine from a heavy-duty drying-out cycle during the initial running period to the light-duty cycle required to maintain the dryness achieved. In this way the power requirements of the air-drying unit are kept at a minimum. It has been found desirable to equip units of this type with acoustic or visual alarm devices which operate in the event of a breakdown of the apparatus. Another type of portable unit has been developed for use in warships. These units are small enough to pass through a manhole and are fitted with removable silica gel containers which are removed for reactivation after maximum absorption has been attained.—*"The Shipping World", Vol. CVIII, No. 2,598, 31st March, 1943, pp. 309 and 311.*

Multiple Diesel Engines for High Powers.

The idea of using small multiple Diesel engines for high-powered marine installations is by no means new. The Diesel-electric drive for the 1,000-ft. White Star liner, which was laid down in Belfast some 15 years ago, but on which construction was stopped before much progress had been made, was to have been on this basis. The total power of 200,000 s.h.p. for four screws was to have been provided by 47 six-cylinder Diesel engines coupled in pairs to generators. A similar design, though on a much smaller scale, was subsequently adopted for the German excursion vessels "Wilhelm Gustloff" and "Robert Ley". In the first-named ship, which had a tonnage of 25,500 gross, four 8-cylr. Diesel engines, developing about 9,000 b.h.p. were geared to two propeller shafts, whilst in the 27,300-ton "Robert Ley" Diesel-electric drive on the same lines was utilised. There were some obvious advantages for these systems of propulsion as regards lay-out in these particular vessels. For instance, the small amount of head-room required for the machinery made it possible to extend the accommodation decks to a lower level than is usual in passenger ships, in addition to which the multiple-engine design made it practicable to carry out machinery overhauls while the vessels were in service. Alternatively, a defective engine could be removed for overhaul ashore and replaced by a complete new unit at any convenient port of call. The chief disadvantages of the geared multiple-Diesel system, whether the gearing is mechanical or electrical, are the transmission losses at the couplings and gearing, and the greater fuel consumption of the high-speed engines. This may amount to anything from 10 to 20 per cent. more than for a similar installation with direct-drive slow-running oil engines, and although this increased fuel consumption might not be important at the present time, it would certainly have a bearing on the choice of machinery in normal circumstances. Another point which has to be considered is that the complication of the engine room increases with the number of engines, and the ancillary gear necessary would, on the whole, tend to offset any reduction in weight in the main machinery obtained by the installation of small fast-running engines. —*"Fairplay", Vol. CLX, No. 3,121, 4th March, 1943, p. 291.*

Atlas Diesel Scavenging Fan Blower Control.

In scavenging fan blowers the air delivery increases with a falling pressure, so that at low loads a considerable effect takes place in the engine, while the fuel consumption rises on account of the increased power absorbed by the blower. With the arrangement recently patented by the A.B. Atlas Diesel, of Stockholm, and illustrated in Fig. 2, controllable dampers are fitted on each side of the blower. When the engine is started, one damper (*2*) is kept more or less closed by a hand-operated device until the full speed of the engine is reached. With a diminishing load, the deflection of the engine governor (*16*) increases and the lever (*15*) is turned to the position shown by the dotted lines. The piston (*11*) then moves into the cylinder (*9*) and a portion of the recess (*13*) registers with holes which communicate with the pipes (*8*). Air pressure is then supplied to the cylinder (*7*) from the scavenging blower (*23*). Thus, the piston (*6*) moves to the right and by reason of the conical shape of the recess (*13*) the supply of air takes place progressively. The damper (*2*) throttles the air suction pipe (*24*) to correspond with the decrease of the load. When the load increases, the governor makes a smaller deflection and the lever (*15*) turns to the left-hand position, this movement being transmitted to the fuel pump through the rod (*18*) and causing the fuel supply to be increased. The piston (*11*) then moves to the left, so that the recess (*13*) ceases to register with the pipes (*8*) and no air is supplied to the cylinder (*7*). The piston (*6*) is then moved back by the spring (*19*). The diagram shows an additional damper (*2'*) in the air pressure pipe. With the arrangement shown in the lower view the lever (*4*) of the damper (*2*) is connected by a rod (*20*) to a lever (*21*) actuated by the governor. The lever (*21*) is pivoted at the point (*22*) and an attached link (*8*) extends to the fuel pump.—*"The Motor Ship", Vol. XXIV, No. 279, April, 1943, p. 34.*

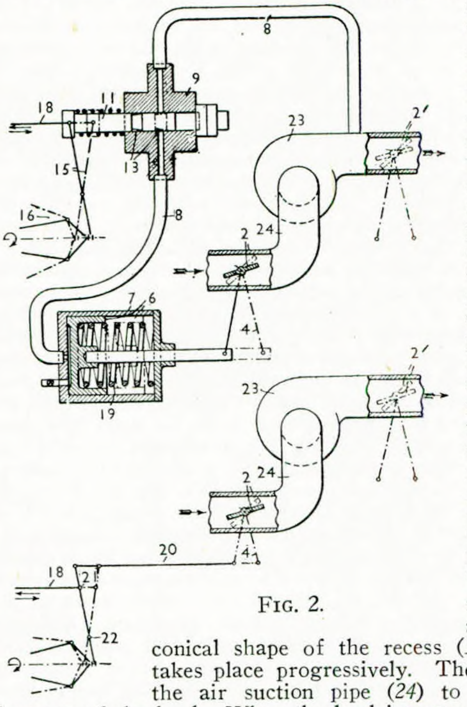


FIG. 2.

Tugs Built in Sections.

The application of prefabrication methods to the building of a large number of small tugs in this country has involved a radical departure from standard practice in regard to hull form, details of design and methods of construction. The tugs are steel-built vessels 70ft. in length, with reciprocating steam engines of about 220 i.h.p., giving them a maximum speed of 7 to 8 knots. Thirteen contractors are engaged on the work of building the hulls, each firm taking one or more sections 10ft. long. Each hull is divided up into eight such units, the maximum size of which is 10ft. by 17ft. by 13ft. These units are too large for rail transport, and are therefore taken to the shipyard by lorry. A reception space has been laid out in 80ft. of storage space in the yard, in line with the assembly berth. As no lifting facilities were available, a rough gantry for hand operation was erected and has proved adequate for the purpose. The gantry slides over the reception space and carries a 10-ton lifting beam. Each hull section is allotted its correct position on the floor of the reception space, being lifted off the lorry on arrival by the gantry and dropped on a pair of joists laid across portable keel blocks at either end of the unit. When a complete set of units has been received it is transferred from the reception space to the assembly berth by means of a low trolley traversed by a hand winch and equipped with hydraulic jacks and portable wooden blocks located at the same centres as the cross joists. The sections are then assembled by welding, and it is reported that no appreciable distortion takes place during this process, the completed hull being quite as good as an ordinary riveted structure. The entire cycle of operations from the time the units are received at the assembly berth to the launching of the hull can easily be carried out in seven days. A second assembly berth is shortly to be put into service in order to increase the output to two tugs a week. The engines and boilers are installed after the hulls are launched. As the drawings ordinarily used by constructional engineers differ in many important respects from those prepared for use by shipbuilders, and as the sections are built by men with no previous knowledge of shipbuilding practice, it was found necessary to include many additional details in the original working drawings. Down to the delivery of the first unit more than 1,400 prints of drawings were supplied. Besides simplifying construction and tapping a source of labour new to shipbuilding technique, this new method of building tugs is saving over 20 per cent. of steel.—*"The Times", No. 49,508, 31st March, 1943, p. 8.*

Tugs of Standard Type to be Used on Italian Canals.

The Ansaldo Company of Genoa have won a competition for a design for a standard tug for service on the Pavia-Po-Venice section of the Milan-Po-Adriatic inland waterways system. The design provides for a vessel of about 66×16½×7½ft. with twin screws driven by two Ansaldo producer-gas engines, each developing 85-90 h.p. at 400-420 r.p.m. The estimated fuel consumption is from 1·9 to 2·13lb./h.p.-hr. and sufficient fuel would be carried for a period of 40 hours' service. The designed speed is 3¾ m.p.h. when towing five 100-ton barges along a canal 66ft. in width, with no current. The gas-producer plant is located alongside the engines and consists of two wood-burning generators, one washer and two driers, although only one of the latter would be in use at a time. The generators can be tended without stopping the engines, but if one generator is out of action, the remaining one will suffice to drive both engines at reduced power.—*"Lloyd's List and Shipping Gazette", No. 40,026, 3rd March, 1943, p. 10.*

Inspection and Mishaps.

At a recent informal discussion held by the Institution of Mechanical Engineers on mechanical mishaps, Mr. G. E. Windeler stressed the preventive value of regular inspection by persons external to the establishment. A number of mishaps were detailed, many of which would not have occurred if there had been regular inspection by a disinterested person. It does not follow that the internal inspector is either dishonest or negligent, but he is often unconsciously influenced by the authoritative views which he hears expressed around him so constantly that mere opinions about the condition of a machine appear to him in the light of facts. When the external inspector hears these views, as he sometimes does, he usually makes a point of putting them to the proof. A case of this kind occurred in connection with a 100-ton crane which had just begun to hoist a load of 60 tons when a 5-in., second-motion shaft failed and the load fell to the ground, fortunately a matter of only a few inches. A new shaft made locally was fitted and the incident dismissed as due to the shaft being old and the bearings slack. The new shaft very soon failed in exactly the same place as the first one. The minor officials mostly concerned attributed this to some inferiority of the local product and made no great fuss about it, as here again there had been no serious damage or personal injury. It was only when a third failure occurred, with the loss of a dozen lives and injuries to a large number of men, that an immediate investigation by a Government officer took place. From such evidence as remained available it was put practically beyond doubt that all three accidents were due to the same combination of worn gear teeth (exercising a strong radial thrust upon the shaft) and misaligned bearings, aggravated in the second case by the locally-made shaft being of lower tensile strength and very much lower elastic limit than the cranemakers' shafts. Each shaft had, therefore, been continuously subjected to greater alternating stress than it could bear indefinitely, and failed from fatigue. Competent inspection would have scotched the trouble immediately after the first failure, if not before it.—*"The Syren", Vol. CLXXXVI, No. 2,427, 3rd March, 1943, pp. 327 and 335.*

Structural Design of Oil Tankers.

In this paper the author (who is the Chief Surveyor of the British Corporation Register) makes a critical examination of the various structural arrangements adopted in modern tankers, in order that the relative merits and demerits may be assessed. He expresses the hope that by doing this the causes of certain troubles which develop with different types will be understood more clearly, and claims to show what should be done to remove these troubles. The author then puts forward a new arrangement, which, while it does not involve the introduction of any new principle, combines in one structure the various features which have proved satisfactory in service.—*Paper by J. L. Adams, read at a meeting of the N.-E. Coast Institution of Engineers and Shipbuilders, on the 12th March, 1943.*

Bolinder Engine Circulating Pump.

A gear-wheel pump for circulating water contaminated by mud or sand forms the subject of a new British patent granted to the A.B. Bolinder-Munktell, Eskilstuna, Sweden. The mechanism of the pump, shown sectionally in Fig. 3, includes a pair of gear-wheels (*14*) for water and another pair (*13*) for the circulation of oil. The oil pump is located between the bearings (*16, 17*) of the driving shaft (*15*), which actuates the shaft (*18*). Soft packing (*28*) is fitted at the contact surfaces between the covers and the pump housing, and the shoulders (*30*) are arranged to bear against each other in order to ensure the correct clearance when the bolts (*27*) are tightened. As the water in the compartment (*24*) must be prevented from entering the bearings (*17, 20*) and as any leakage of oil from the oil pump into the water pump is also undesirable, suitable packing (*32*) is provided between the gear-wheels (*14*) and the housing (*11*). The hubs (*33*) are connected with the shafts so as to be fluid-tight, screws (*35*) expanding the sleeves (*34*) and sealing the space between the hubs and the shafts. Accordingly, water is unable to penetrate to the bearings (*17, 20*), leakages being drained through the passages (*41*). The teeth of the gear-wheels (*14*) are covered with a layer of rubber and are machined angularly to a greater width than those of the gear-wheels (*13*) of the oil pump. Water is drawn into the pump through an inlet (*46*) and discharged through a conduit (*49*), a partition (*51*) being provided.—*"The Motor Ship", Vol. XXIV, No. 279, April, 1943, p. 34.*

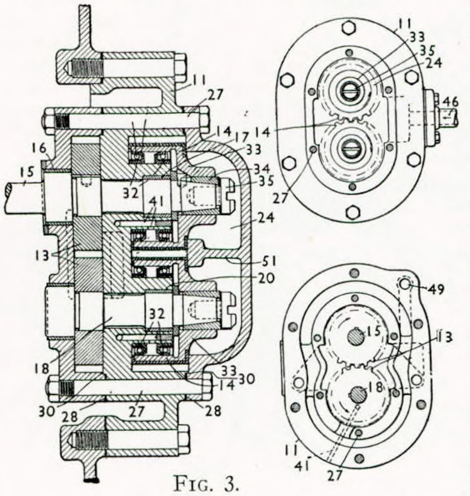


FIG. 3.

Modern Stems and Stern-frames.

There have, in the last few years, been many changes in the form and construction of stems, stern-frames and rudders, steel castings having replaced forged steel or iron, and cast steel having now been replaced by components fabricated by welding from mild steel plate. The change of form and section, with the snubbed shape at the upper part of the stem required to overcome the too prominent snout from the V-shaped forward sections of the so-called Maierform, and the general adoption of the cruiser stern have contributed largely to the popularity of fabrication. In some cases the lower part of the stem has been of cast or forged steel, as well as the stern-post and lower part of the stern-frame, but latterly the whole of both stem and stern-frame have been entirely fabricated from mild steel plate, the lower parts becoming in effect a continuation of the foremost and aftermost keel-plates. In the case of a fabricated stem the plates are bent cold into a U-form to sections taken from the mould loft by a knife tool under a hydraulic press, or from the plate furnace, in conveniently long lengths, and welded into a single piece. A central web is welded into the lower part of the stem to which the centre keel can be riveted or welded, also transverse webs for connection to the fore-peak floors and at the height of the decks, and stringers at the intermediate points necessary to ensure rigidity. The lower end of the stem is lap-welded to the forward butt of the foremost keel-plate, or perhaps more conveniently flush-butted with an internal butt strap. Similarly, the sole-piece of the fabricated stern-frame forms a continuation of the after keel-plate, the lower part of the stern-post of bent plate, similar to the stem, being butt welded to the top edge of the sole-plate as well as continuing down inside the latter to the bottom and being secured by welding. The forward edges of the stern-post are curved towards the bottom to give a longer weld to the upper edges of the sole-plate. Webs are welded into the sole-plate for connection to the centre keel and floors, as in the case of the stem. The part above the boss, made in a similar manner to the rest, runs from the boss over the propeller aperture, with webs for the attachment of the floor plates of the after frames, and with a central web where necessary. The rudder post, of bent steel, is also butt-welded to the top of the sole-plate and continued down to the bottom, the upper open side of the sole-plate being closed by a plate welded also to the after edge of the stern-post and forward edge of the rudder post. The gudgeons are of cast steel, the lowest one being welded into the after end of the sole-plate, whilst the intermediate ones are welded into the open part of the rudder post, the space between them being closed by a welded-in plate. The rudder post extends above the screw aperture and is connected to the transom floor as usual. The boss is simply a thick ring of cast steel of the required dimensions for the stern-tube, to which the upper and lower parts of the stern post are securely welded, with vertical stiffening webs welded both above and below the boss casting. It is perhaps too early to weigh up definitely the advantages and disadvantages of fabricated stems and stern-frames. The fabricated pieces are somewhat more susceptible to local damage than solid castings or forgings, but they are also much more easily and cheaply repaired, and the work of repair can be more readily carried out due to the avoidance of the delays so often associated with the delivery of large castings and forgings. With stern-frame castings, the damage frequently involves the renewal of the component forming the sole-plate and lower parts of the stern and rudder posts, and the new component may not always prove to be a perfect fit, whereas with a fabricated component the undamaged part can remain and it is merely necessary to cut out and remove the damaged portion for renewal by welding. In some cases the damage can even be made good in place by local heating and fairing. Another advantage of the fabricated stem is that the hooding ends of the bow plates, lapping inside the stem, are protected from the erosion produced by cutting through the water. The connection of a solid rolled-bar stem and solid stern-frame to the keel-plates involves long riveting through the thick forging or casting, with the usual percentage of bad holes, and is not very satisfactory. The rudder, of streamlined form, consisting of a shell ribbed internally, both vertically and horizontally, and incorporating steel castings in way of the pintles and coupling flange, must of necessity be fabricated. Rudder stocks are commonly specified to be of forged ingot steel, an expensive forging due to the large dimensions of the flange, but in foreign shipyards the flange is cast with a clump attached, and this clump is then forged down to the diameter of the stock, the result being quite satisfactory and far less expensive. There is nothing new in the adoption of U-shaped sections for stems and stern-frames instead of solid sections, as hollow sections were used by the Admiralty for warships when castings were commonly employed and electric welding was unknown.—*"The Journal of Commerce" (Shipbuilding and Engineering Edition), No. 35,901, 4th March, 1943, p. 1.*

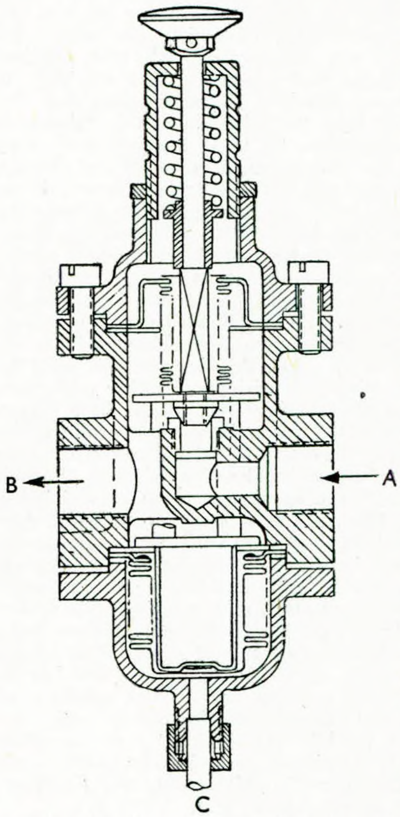
Low-temperature Welding.

A new welding process known as "Castolin Eutectic Low-Temperature Welding" has been developed in the United States. It utilises the usual heating methods, *i.e.*, gas torch, electric arc, induction furnace, salt bath, etc., but the fusion temperatures of the parent and weld metals are claimed to be far lower than those required for any other welding process. It is also claimed that the Eutectic welding method is so simple that it can be successfully employed by inexperienced welding operators. The process is stated to have been evolved by "the particular proportioning of a metallic alloy or solution which results in the lowest possible melting point", and there are special welding rods for every possible junction of metals, similar or dissimilar. Moreover, each type of weld metal has its own particular flux with a melting point which governs the amount of heat to be applied just prior to the application of the rod. The flux, in addition to deoxidizing the surfaces to be joined, has three main purposes. To promote the diffusion of the molten weld metal into the parent metal, to reduce the surface tension on the fluid alloy, and to lower the melting point of the alloy and produce more rapid flow. The Castolin Eutectic process can be used for all types of horizontal, vertical and overhead joints, whether butt, sleeve, lap or chamfered, and for dissimilar methods of different thicknesses. The process can be employed for joining, hard-facing, building-up pieces broken away by mishandling, the repair of castings and tool salvage. When properly used, the resulting weld deposit matches the colour of the parent metal. The manufacturers, Eutectic Welding Alloys, Inc., produce about 40 different types of welding rods and fluxes for cast iron; all types of steel; copper, brass and bronze;

nickel and nickel alloys; aluminium and aluminium alloys, and for zinc-base castings.—*"Motorship", Vol. XXVIII, No. 3, March, 1943, pp. 200-201.*

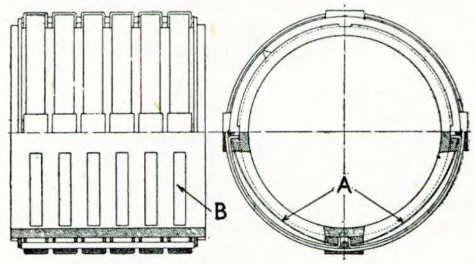
Lubricating-oil-pressure Safety Device.

Automatic shut-down devices for oil engines are usually of a fairly complicated nature, but a relatively simple appliance of this nature has now been developed and put on the market by a well-known firm of engineering accessory manufacturers in Middlesex. The device forms part of the fuel-supply line between the daily service tank and the injection pumps, and as it is only 8in. in height and 3in. in diameter, its weight can easily be supported by ⅜-in. gas or ½-in. fuel-oil piping. The construction of the cut-out device is shown in the accompanying sectional drawing, from which it may be seen that there are only three connections to be made when installing it, these being the fuel inlet (A), the fuel outlet (B) and the lubricating-oil connection (C). There are three main components, one of which is the bottom casting housing a bellows unit upon which the lubricating-oil pressure acts. This compartment is completely sealed from the rest of the device, so that lubricating oil cannot flow into the fuel, or vice versa. The central portion of the device, which is concerned with fuel oil, contains a mitre-faced valve and its seating, the valve being raised by the lubricating-oil bellows unit through the medium of two rods forming a stirrup member. Surrounding the spindle of this valve is a bellows-type seal, so that no fuel oil can seep out of the upper section, which is another casting secured by screws to the main body of the device, and containing a valve-operating spring and valve-spindle extension, surmounted by a knob. The effective strength of the spring can be varied by rotating its housing, so that the minimum safe pressure of the lubricating oil can be determined accurately after trial on the engine. Where the device is fitted to an engine having a priming pump in the lubricating-oil circuit, the operation of this pump, to give a pressure of some 5lb./in.², raises the cut-off valve and permits fuel to flow to the injection pump, so that the engine can be started in the usual way; but where no priming pump is fitted, the valve plunger is raised from its seat against the spring by means of the upper knob. Where there is only a single watchkeeper, arrangements can easily be made to hold the valve off its seating while the engine is being started up, after which the safety shut-down capacity of the device can be put into action.—*"The Oil Engine", Vol. X, No. 120, April, 1943, p. 325.*



The Circoflex Automatic N.R. Valve.

The Circoflex automatic non-return valve shown in the accompanying diagram has been developed for use in connection with the scavenging and pressure charging of two-stroke oil engines, as well as for pumps and compressors. The valve is claimed to be suitable for the control of air or liquids at either high or low pressures. The construction is extremely simple and consists of a central tube with slotted air passages, surrounded by stop plates that form abutments for thin metal strips which flex between the slotted tube and the stop plates. These are the only moving parts; they require no lubrication, and automatically spring inwards to close the air passages when the pressure falls. The assembly is dismantled by removing a spring ring at each end of the valve, after which all the parts can be taken out without special tools. The air flows through the open valve from the inside to the outside, and there is no return path. The metal strips are always under tension. It is claimed that this form of construction makes it possible to utilise the entire internal diameter of the valve, whereas with a disc valve only about one-fifth of the area is effective for air passage. The cylindrical form ensures rigidity, which is important where high pressures are used. The valve is particularly suitable for high operating speeds, the resistance to air flow being only 0·25lb./in.². A crankcase scavenge pump fitted with Circoflex valves has shown a volumetric efficiency of 90 per cent. No adjustments are needed in service, periodic cleaning being all that is required.—*"The Motor Ship", Vol. XXIV, No. 279, April, 1943, p. 27.*

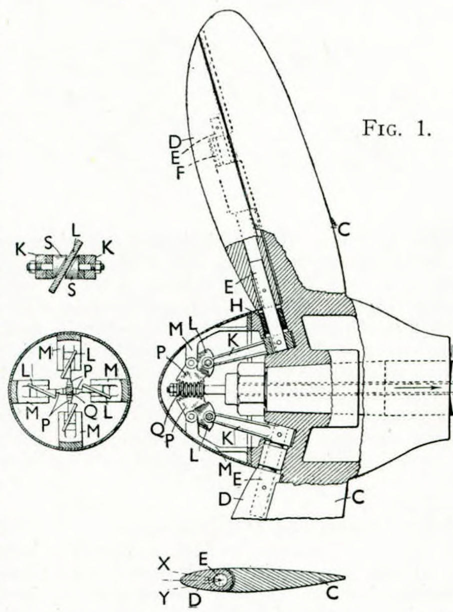


The Circoflex valve.
A.—Flexing strips. B.—Air passages.

Stone Variable-pitch Propeller.

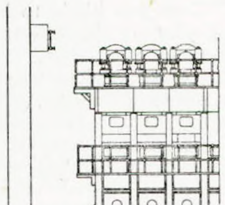
In an improved type of variable-pitch propeller, recently been patented by J. Stone & Co., Ltd., Deptford, the pitch is varied by the angular adjustment of the control blades (D), shown in Fig. 1, hinged at their leading edges on the trailing edges of the main blades (C). The hinged connection of the control blades is formed by inner and outer pintles (E) fitted in bearings (F, H) of hard rubber. The pivotal adjustment of the control blades is limited (X, Y). The arms (K) which move the control blades are actuated bv cams (L), each comprising a segmental plate disposed at an angle to the radius on which its supporting bracket (M) lies. The cams are attached to toothed segments (P) meshing with a rack rod (Q) which is moved longitudinally by a control spindle (R). When the spindle is moved, the toothed segments (P) are turned by the rack (Q). The cam segments (L) make contact with the jaws (S). The mechanical advantage due to the length of the levers make it possible to adopt a light construction and to obtain fine degrees of adjustment.—*"The Motor Ship", Vol. XXIV, No. 279, April, 1943, p. 34.*

FIG. 1.



Variable-pitch Propellers for Large Ships.

A 7,400-ton cargo liner building at the Götaverken for the Johnson Line, Stockholm, and due for completion this summer, is the first large vessel to be equipped with Kamewa reversible, variable-pitch propellers made by the Karlstadts Mekaniska Verkstad. The machinery installation consists of two sets of 6-cylr. two-stroke Diesel engines of the latest Götaverken type, each developing 3,500 b.h.p. at 125 r.p.m. and designed to give the ship a service speed of 16¾ knots in loaded condition. The cylinders are 630 mm. in diameter and have a piston stroke of 1,300 mm. The two engines are directly coupled to twin propeller shafts fitted with Kamewa v.p. propellers, as shown in Fig. 1. Oil for operating the propeller blades is normally delivered by the pumps (*6*) driven from the propeller shafts, but an electrically-driven pump (*7*) for use when starting, is also installed. The oil is supplied to the valve rod of each propeller through an oil-distribution box (*5*). The valve rod can be adjusted from the bridge by a telemotor (*10*) acting on a small servo motor located in the oil-distribution box. The servo motor actuating the blade is within the propeller hub and is not shown in section in the drawing. The control levers on the bridge are so arranged that predetermined positions of the propeller blades correspond to those of the control levers, and the latter cannot be moved more quickly than the movement of the blades. It takes about 13 seconds to move the levers from the full speed ahead position to full astern. A special indicator independent of the telemotor, shows the position of the blades at all times. The blade adjustment may be carried out at two control pillars on the bridge



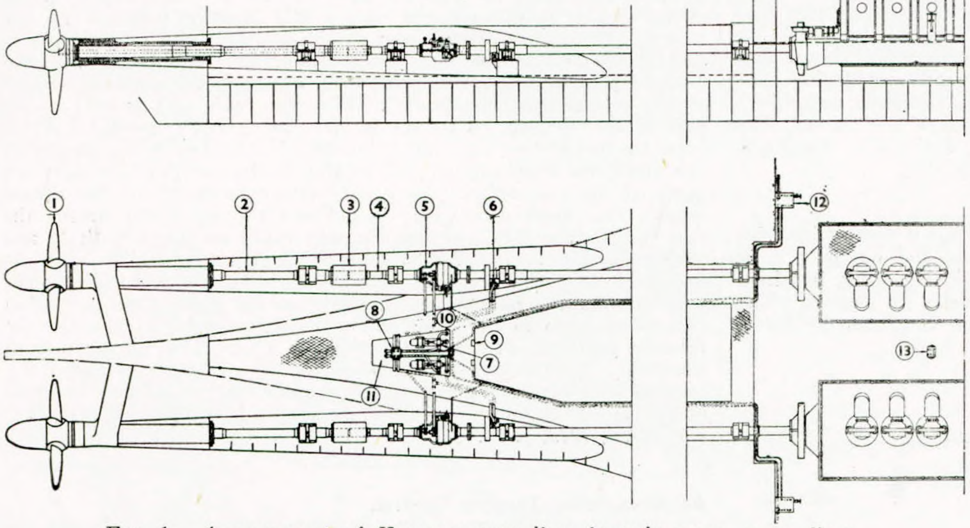


FIG. 1.—*Arrangement of Kamewa propellers in twin-screw cargo liner.*

1.—Kamewa propeller. 2.—Propeller shaft. 3.—Sleeve coupling. 4.—Intermediate shaft. 5.—Oil distribution box. 6.—Gear-wheel-driven pump. 7.—Motor-driven pump. 8.—Safety valve. 9.—Manœuvre-recording instrument. 10.—Telemotor. 11.—Lower oil container. 12.—Upper oil container. 13.—Control pillar in engine room.

as well as from a control pillar in the engine room. It can also be effected by hand, without using the telemotor, at the oil-distribution box. On one of the control pillars on the bridge are push buttons for starting and stopping the motor-driven pump and for adjusting the speed of the engines. Change-over valves for the emergency stopping of the Diesel engines are also fitted on the bridge. The E.R. pillar likewise makes it possible to adjust the propeller blades and the speed of the motor-driven oil pump from the engine room. As it is impossible in the engine room to hear or see the movement of the propeller blades when they are adjusted, each propeller is connected with a recording device, operated by clockwork, for recording the momentary position of the blades on a strip of paper, thereby eliminating the need for making entries in the E.R. log book. The propeller blades are of a stainless alloy containing 13 to 14 per cent. of chromium and various secondary constituents which make the blades resistant to corrosion in sea water. When getting under weigh, the motor-driven oil pump is first started and the propeller blades set in the zero position to facilitate a quick start of the main Diesel engines. As soon as the latter are running, the motor-driven pump is stopped and the speed of the main engines is adjusted to the normal, after which the signal "all clear" is given to the bridge. The captain then takes charge and operates the control levers on the bridge as required, the engine speed being maintained constant by the governor. A table on the bridge shows the captain what the fuel consumption per hour or per nautical mile will be with the propeller blades at different adjustments, and there is also a test curve and table indicating the correct blade adjustment for a given ship's displacement. Among the advantages claimed for the v.p. propeller is that all manœuvring can be carried out direct from the bridge, whilst the engines operate under more favourable conditions. Propeller vibration can be entirely eliminated, and if a blade sustains damage it is more easily replaced. Turbine ships equipped with Kamewa propellers do not require to have any astern turbines.—*"The Motor Ship", Vol. XXIV, No. 279, April, 1943, pp. 6-10.*

Lifting of Welded Ships' Ends.

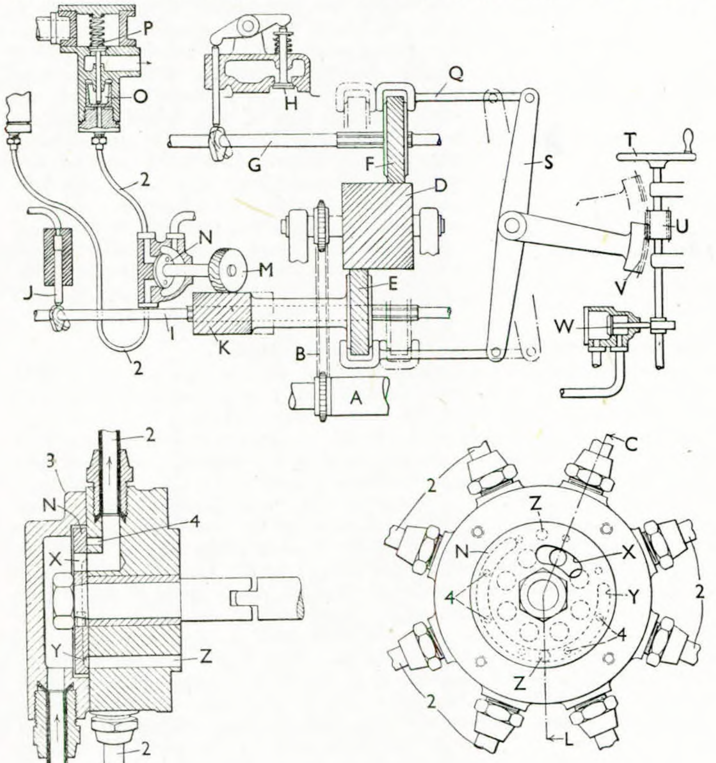
The tendency of the ends of welded ships' hulls to lift from the blocks during construction is caused by the length of the welded butts, the contraction forces at the upper part of the structure being greater than those at the bottom of the ship. This tendency can be corrected in two ways, *viz:* by arranging the sequence of welding in such a manner as to ensure that at no time is there an important excess of contraction forces tending to lift the structure over the similar forces tending to hold it down, or by cambering the line of keel blocks slightly at the ends of the ship and allowing the vessel to lift so that the final line of keel is straight. The first procedure may seem to be better than the second, but there is bound to be a residuary stress, whichever is adopted, as unless proper allowance is made for the contraction of any welded joint, trouble may be experienced on this account. It has been suggested that since the ends of the ship will drop as soon as she is afloat, the initial unfairness is of little importance. It must be remembered, however, that the dropping of the ends of the ship induces certain tensile stresses on the topsides which, in the case of a welded hull, will be additional to those locked up in the structure. The general opinion is that the combined effect of the various stresses on a welded ship due to the contraction of the weld metal and the external forces, is not serious, and it has even been suggested that provided the material is stressed sufficiently, plastic flow will take place and the stresses will be relieved automatically. A possible objection to this theory is that the stress structure may be subjected to repeated reversals of stress on the top of the initial stress, so that a plate may fail from fatigue without plastic flow having had a chance to develop. Although there is now sufficient successful experience of all-welded ships to remove any fears of possible failures of such a nature, there is still room for investigation and research into this particular matter.—*"Fairplay", Vol. CLX, No. 3,122, 11th March, 1943, p. 318.*

The Loss of the American Tanker "Schenectady".

Although there have been several failures in the all-welded vessels which are now being turned out from American shipyards in ever-increasing numbers, the loss of the oil tanker "Schenectady" was an occurrence quite out of the ordinary, and has evoked a good deal of comment in American shipping papers. The ship was an all-welded tanker built at the Swan Island Yard, Cal., an organisation run by the Kaiser Company. She was launched just seven months to the day after the laying-out of the yard had been commenced, the construction of the vessel having taken only 115 days, as compared with average times of 200 days on the East Coast, and 230 days on the Gulf Coast, for similar ships. While lying in the fitting-out basin after completion, the "Schenectady" broke in half and sank. No final statement has yet been made as to the reasons for this occurrence, and, according to the New York *Journal of Commerce* the cause of the casualty might have been anything from poor manning to inherent weaknesses in the methods of construction. The speed of construction may also have been a contributory factor to the accident and, judging by the nature of the trouble given by certain other all-welded ships built in record time, it is to be feared that many more weaknesses will be found in such vessels than in those built in a normal time. If this is really the case, then the matter may be remedied in two ways: either the ships must be built at a more leisurely pace or a greater degree of supervision must be exercised. In the case of riveted vessels, past experience indicates that workmanship did not necessarily suffer through speedy construction, or, if it did, that steps could be taken to remedy matters, whereas, in the case of ships of all-welded construction, the position is very different. The efficiency of welding depends so much on the conscientiousness of the workmen and the care with which the operation is carried out, that any undue speeding-up of the process is likely to lead to unsatisfactory results. The trouble which can arise in this way is accentuated by the fact that detection of faulty welding is not always easy; and, furthermore, that if welding is carried out in the wrong sequence, which could easily happen when an attempt was being made to beat the clock, abnormally high locked-up stresses would result. If the cause of the failure of the "Schenectady" was faulty construction, the matter may be easily dismissed, but if it can be proved that the accident was due to other circumstances, the casualty becomes of great importance. It may be assumed that, as in other cases where welded structures have failed, the fracture took place in the solid plate and not at the welded connections, in which event the "plastic strain" theory is likely to be upset. The supporters of this theory contend that, if the internal stresses in a welded ship structure becomes sufficiently severe, all that happens is that the plates, being stressed beyond the yield point, accommodate themselves by stretching in the plastic condition, and thus relieve the stress without any harm being done. It would appear that, in the case of the "Schenectady" conditions were ideal for this stress-relieving process. The ship was in the fitting-out basin, and, therefore, there could have been none of the dynamic effects apparent in a sea-way to complicate matters. The fact that the ship did not merely stretch, but broke completely, seems to invalidate the "plastic strain" theory. The occurrence shows that there is still much to be learned in regard to all-welded ship construction and it is to be hoped that when the American authorities have arrived at their conclusions as to the cause of the failure, they will permit these to be published.—*"Fairplay", Vol. CLX, No. 3,124, 25th March, 1943, pp. 372-373.*

Crossley Engine-reversing Mechanism.

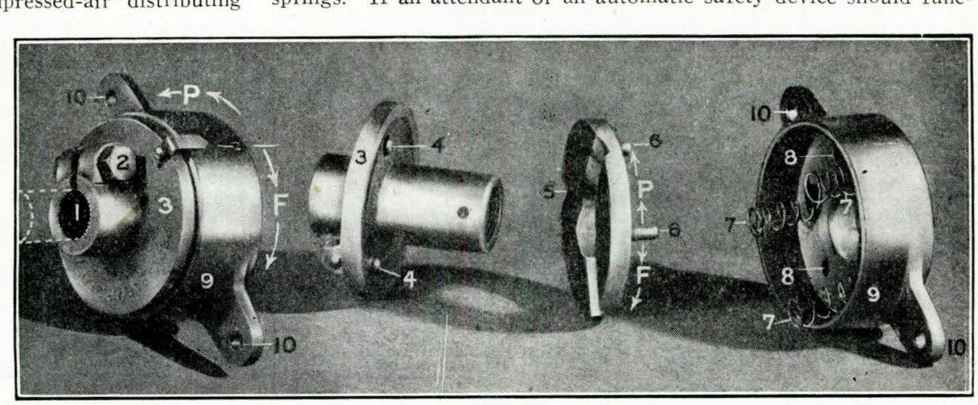
An improved form of starting and reversing mechanism for Diesel engines, by means of which the valves and pumps can be controlled by a single hand-wheel through helical gearing, has been developed and patented by the makers of Crossley oil engines. The arrangement illustrated in the accompanying diagrams is for a



Crossley two-stroke marine engine having exhaust valves in the cylinder heads. The complete gear is shown in the upper diagram, whilst the lower left-hand one is a section on the line C-L of the adjacent end view, both showing the compressed-air distributing valve. The main crankshaft (A) drives a comparatively long helical gear-wheel (D) through a chain (B). Two wheels (E, F) mesh with this wheel and both are splined to enable them to slide along and at the same time drive their respective shafts with an advanced or retarded motion. One of the exhaust valves (H) is represented in a cylinder head and the operating cams are fitted on the camshaft (G). The lower wheel (E) is combined with a helical wheel (K) driving another wheel (M) which rotates a disc valve (N). This valve controls the flow of air to and from the pistons (O), which open the valves (P) delivering the air to each cylinder in turn. In addition to driving the wheel (K) for the air distributor valve (N) the wheel (E) also operates the fuel-pump shaft (I) which carries the cams for working the pump plungers (J), only one of which is shown. A single hand-wheel (T) controls all the operating mechanism. Its shaft carries a worm (U) which raises and lowers a quadrant (V), while at the lower end of the shaft means are provided to control a master air valve (W). This valve allows compressed air to pass to the distributor. The disc valve (N) for distributing the air has a port (X) shown in the lower diagrams. When the valve turns, air passes through the port and reaches each of the branches (*2*) in succession, thus opening the valves (P) already referred to. A groove (Y) extends around part of the seating surface of the valve (N), thereby connecting the relief ports (*4*) of the branches with the exhaust passage (Z) after each period of supply and until a further supply is made to each branch. The disc valve (N) is held on its seat by the pressure of the air when starting, but normally it floats between the seating and the valve cover (*3*). When the reversing operation has been carried out by the handwheel (T), the different parts of the mechanism take up the positions shown by the dotted lines. The quadrant (V) is raised and the lever (S) moves the rod (Q) to the left, causing the wheel (F) to move with it and turn relatively to the helical gear-wheel (D). A similar effect is produced on the wheel (E), which moves to the right and turns relatively to the helical wheel, while at the same time the wheel (K) moves also to the right and turns the wheel (M) to alter the relative positions of the air distributor (N). The cams on both shafts (G, I) also turn to the positions shown by the dotted lines, thereby altering the timing of the exhaust valve (H) and the fuel pump (J) for running astern.—*"The Motor Ship', Vol. XXIV, No. 279, April, 1943, p. 26.*

An Over-riding Throttle Control.

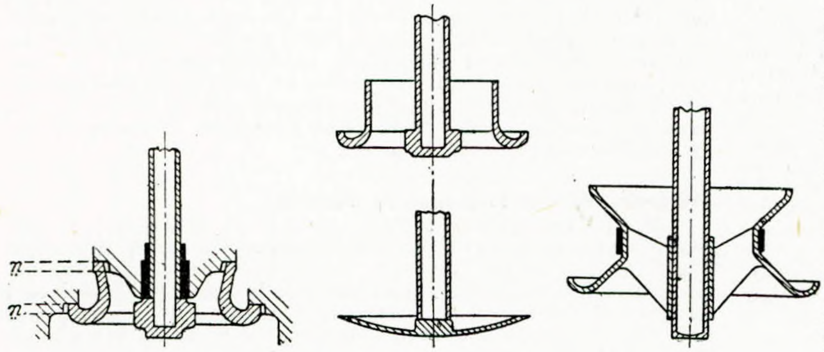
An improved form of inter-connection for the throttle and reverse gear controls of a motor-boat's engine has been developed and patented by John I. Thornycroft & Co., Ltd., and is shown in the accompanying illustration. The purpose of the device, which is purely mechanical, is to close the throttle valve automatically to a tick-over of the engine when the reverse gear is put in neutral, but in an emergency or if required for engine adjusting purposes, the interlocking control can be instantly put out of action by means of a small and very simple hand-operated clutch. The device can be used with either mechanical or hydraulic controls to provide a positive-motion transmission in one direction and frictional transmission in the other. Referring to the illustration, the hub of the device is fixed to the control shaft (*1*), which is operated by a master or remote control, the hub being clamped to the shaft by the bolt (*2*). The back-plate (*3*), in the working face of which are sunk steel balls (*4*), is an integral part of the hub. Mating with this component is a circular plate which can rotate upon the hub centre. In one direction, however, the cam grooves cut in this floating plate come up against the balls (*4*) and check its rotation positively, so that the plate (*5*) must move in company with the plate (*3*). In the opposite direction the grooves in the plate (*5*) are inclined and separate the two plates (*3* and *5*), allowing the latter to move in relation to the former. The frictional restraint against rotation in this direction is provided by the loading of three springs (*7*), which tend to press the plate (*5*) against the plate (*3*). A rotational connection is provided between the plate (*5*) and the outer member of the device (*9*) by pins (*6*), which register with holes (*8*) in the cover plate (*9*). In one direction, therefore, the latter is rotated positively by the shaft (*1*), but in the opposite direction the remote control moves the control linkage connected to the lugs (*10*) purely by means of the frictional effort which is dependent upon the strength of the springs. If an attendant or an automatic safety device should func-



tion to rotate the cover in the "frictional" direction, this is easily accomplished. The amount of such rotation for all practical purposes is unlimited, but under normal circumstances a few degrees should suffice. The cover member can easily turn back to its initial position, or a positive reconnection will take place if required after rotating in the "frictional" direction for 120°, where there are three cam grooves. The three main components of the unit illustrated are of Monel metal and therefore hard wearing and resistant to corrosion.—*"The Motor Boat", Vol. LXXVI, No. 1,906, April, 1943.*

Lighter Poppet-type Steam Engine Valves.

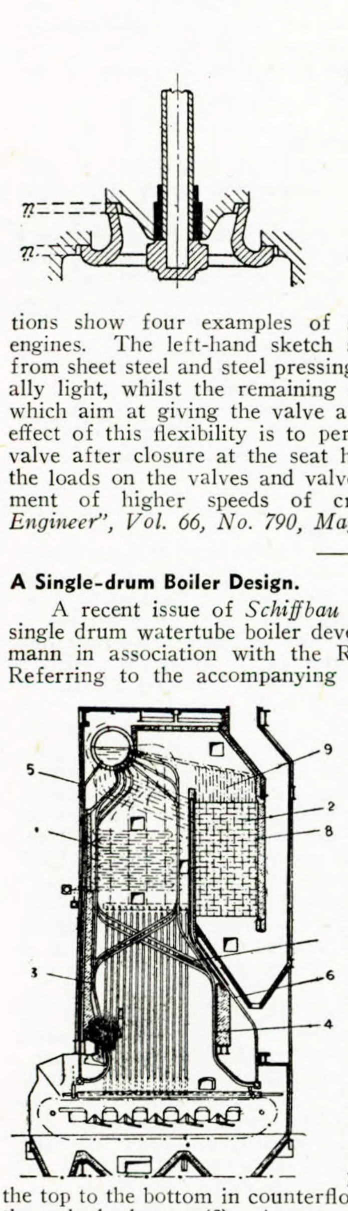
An article by P. Koessler which appeared in *Z.V.D.I.* some months ago described some recent improvements in the design and construction of valves and valve gear for steam engines. In order to reduce the inertia effect of poppet valves, it is proposed to make them lighter and more or less flexible. The accompanying illustra-



tions show four examples of such valves for Lentz high-speed engines. The left-hand sketch shows a double-seated valve made from sheet steel and steel pressings, which is claimed to be exceptionally light, whilst the remaining sketches show three Lentz designs which aim at giving the valve a certain degree of flexibility. The effect of this flexibility is to permit of slight further travel of the valve after closure at the seat has been attained, thereby reducing the loads on the valves and valve gear and facilitating the employment of higher speeds of crankshaft rotation.—*"The Marine Engineer", Vol. 66, No. 790, May, 1943, p. 115.*

A Single-drum Boiler Design.

A recent issue of *Schiffbau* contained a short description of a single drum watertube boiler developed and patented by W. Wippermann in association with the Rheinmetal-Borsig A.G., of Berlin. Referring to the accompanying drawing of the boiler, the superheater (*1*) is situated in the uptake above the combustion chamber at about the same level as the feed heater (*2*), which is in a second flue. Most of the vertical tubes (*3*) and (*4*), which form the front and back walls of the combustion chamber, serve to provide the radiant heating surface. The unheated downcomer tubes (*5*) and (*6*) are arranged so that the tubes in separate groups have approximately the same length. The crossing of the tubes as shown means that part of the radiant surface also acts as a conduction surface and serves to shield the superheater from the combustion chamber. The steam-generating tubes in the front and back walls discharge directly into the upper drum. The flue gases pass through the duct (*8*) from



the top to the bottom in counterflow to the passage of the feed water through the heater (*2*). Any steam generated in the upper parts of the heater tubes can pass to the upper drum by means of the tube (*9*). The superheater is suspended from the vertical tubes (*3*) and (*4*), burning of either tubes or supports being unlikely.—*"The Marine Engineer", Vol. 66, No. 790, May, 1943, p. 116.*

Electric Welding in Shipbuilding.

A series of instructional lectures delivered at Stow College, Glasgow, by Mr. R. B. Shepheard, B.Sc., who is the Superintendent of Welding Development (Merchant Shipbuilding) at the Admiralty, dealt with such points as design, practical considerations, supervision and inspection, training and repairs. He advocated that in order to achieve the greatest increase in output welding should be adopted to replace local concentration of rivets—*i.e.*, at butts, seams, for oil-tight and watertight work, and the like; such welding can be carried out effectively at the berths. Hand and pneumatic riveting should be replaced by welding before hydraulic riveting, the latter being a cheap and effective method of prefabricating work on the skids. The point was made, in the section dealing with design, that butt joints have a much greater fatigue strength than overlap joints or fillets, and should, therefore, be employed where possible. It was also pointed out that the shift of butts, normal in riveted work, is not essential with welded construction, provided joints in one line are properly disposed in relation to adjacent structure, and that special attention is paid to the welding procedure. The lecturer stressed the importance of stringent supervision and inspection of welded work, and declared that the general welding procedure should be thoroughly controlled by the management and not left to a foreman. The U.S. shipyard practice of having one supervisor for ten welders was strongly recommended.—*"Fairplay", Vol. CLX, No. 3,121, 4th March, 1943, p. 292.*

Arc Welding and Cutting Under Water.

An article by H. Schmidt in a recent issue of *Z.V.D.I.* describes the extensive use being made by the Germans of under-water electric welding and cutting for ship-repair and harbour work. Presumably the process should also find application in marine salvage operations, replacing to some extent the present tedious methods employed in patching and otherwise repairing wrecks for refloating. The commercial d.c. converter and the d.c. multi-point welding converter with resistances are suitable, provided rapid voltage return is available after short circuit, as *e.g.*, on starting up and with an excess flow of weld metal. Good arc continuity is essential, because the electrode is difficult to guide owing to the poor light and other characteristics of diving. The electrode holder and cable must be well insulated to prevent losses of current, with possible danger to the diver. The body of the holder is a special rubber moulding and the electrode is gripped by turning the head of the holder. The electrode itself is coated with an non-conducting lacquer which is resistant to sea-water and has been developed expressly for this purpose. The electrode must be able to withstand any tendency to contraction cracking in cold water. The lacquer, composed of hydrocarbon compounds, forms a gas envelope around the electrode as it burns and thus assists in keeping the arc steady. A satisfactory size of electrode for all welds is $\frac{3}{16}$in. diameter by 14in. long. The diving gear and diving dress may be of the usual standard pattern, but certain bare metal parts, such as the helmet and corselet must be insulated by rubberising or, temporarily, by means of a coating of insulating paint. The air-supply regulators of self-contained diving dresses must also be insulated. Under-water welding is confined to the making of fillet welds in all positions, horizontal, vertical and overhead. It requires a good deal of practice, because the electrode has to be guided almost entirely by the sense of touch. Vertical welds are always made downhand. The welding speed is about the same as in ordinary arc welding and the time taken to change an electrode is estimated at 1·16 min. An expert operator can produce welded fillet joints of an ultimate tensile strength of $26\frac{1}{2}$ to $28\frac{1}{2}$ tons/in.^{2} in steel plate, but the ductility is liable to be reduced appreciably by the rapid water quenching, which causes hard spots in the weld and may even result in longitudinal fracture of the weld fillet under test. Brinell hardness values of 180kg./mm^2 under water and 150 above are obtained, indicating a 20 per cent. increase in the former. An under-water weld acquires no beneficial nitrogen from the air. Under-water cutting demands a high overload capacity of the electrode and welding apparatus. The $\frac{3}{16}$-in. electrodes are satisfactory for cutting purposes because the heat generated by the passage of the current is completely dissipated by the water, and it has been found that an electrode can be loaded with 900 amp. with perfect safety. The cutting speed varies from 8in./min. for $\frac{3}{16}$-in. steel plate to 1in./min. for $\frac{3}{4}$-in. plate, using 450 amp. at 42 volts. The cut is naturally a good deal more ragged than when flame cutting is employed, but for the class of work for which under-water cutting is used this should not matter.—*R. P. Nott, "Shipping", Vol. XXXI, No. 368, March, 1943, pp. 20 and 22.*

Lubricating-oil Temperature Control.

The control of the lubricating-oil temperature of I.C. engines involves a number of conflicting requirements. Following a cold start, it is desirable to raise the oil temperature as quickly as possible, so that its viscosity may become suitable to give the best degree of "fling" and to form a freely flowing film. On the other hand, once the engine reaches normal operating temperature conditions, the oil should not get too hot. As the oil takes a great deal of heat away from the bearings and rotating and reciprocating parts of the engine, it is most desirable that this heat should be dissipated from the oil itself in order to prevent it from becoming too hot. This is an obvious application for thermostatic control, but as certain conditions have to be observed, such as prevention of back-pressure in the oil circuits and automatic adaptability to changes in operating conditions, many of the existing designs of thermostat intended for fluid control would be quite useless. A special design to meet the special requirements involved has been developed by a Gloucestershire firm of instrument manufacturers, the main function of the device being to by-pass the oil cooler when the engine is cold. When the engine is warmed up, the thermostatic device regulates the proportions of hot oil and cooled oil for readmission to the engine, so as to give an operating temperature of a desired value, plus or minus a few degrees, under a wide variety of engine load and speed conditions. The thermostat is mounted in the oil circuit and has two oil inlets and one outlet, plus an alternative outlet. Of the former, one deals with hot oil direct from the engine, and the other with oil which has passed through the cooler. The outlet is common to both streams, and the thermostatically controlled shuttle determines the proportions of the hot or cool oil which are admitted. The two inlets are connected to opposite ends of a tube passing straight through the body of the thermostat. At the centre of this tube is a solid barrier, so that there is no possible straight flow through. Each stream is deflected radially through 90° by toroidal surfaces on the barrier, so that it has to flow into the body of the thermostat, through a ring of ports in the tube. As the total area of these ports is larger than the cross-sectional area of the tube, no appreciable back pressure can be set up by the action of the toroidal deflector in changing the direction of flow of the oil. This arrangement allows oil to enter the thermostat body from both ends, the proportion admitted at each end being regulated by the controlling unit by means of a shuttle sliding on the tube and connected through a simple ball-and-socket mechanism to a lever forming part of an assembly containing a coiled bi-metallic thermostat element. The latter is located in the body of the instrument, immersed in the oil, and is therefore directly affected by the temperature. As the element expands or contracts in consequence of a rise or fall in the temperature, its effective length increases or decreases and so moves the arm and therefore the shuttle along the tube. If the oil becomes too hot, the thermostat expands and moves the shuttle so as partially to close the ports admitting hot oil and to open those admitting cool oil. The action is exactly opposite when the oil is cold, until it reaches the predetermined temperature. The selection of the working temperature is effected by the partial rotation of the thermostat element in relation to the cover of the device, a pointer on the spindle enabling the operator to observe the change in the setting with a high degree of accuracy. Although the instrument is not calibrated, this can easily be done by the user, if required. The makers claim that there is no possibility of any jamming of the shuttle through the passage of relatively large particles of foreign matter carried in the oil stream and that the operation of the lubrication circuit is not affected by mechanical damage of the device. In the event of a broken element, the setting can be arranged so as to give a fixed "full cool oil" position for the shuttle, thus entirely cutting out thermostatic control. Exhaustive tests of the device have shown that once the oil has reached working temperature, it does not vary more than ±5° F., and that the working temperature is reached from cold more quickly than when manually operated cooler cut-out valves are used.—*"The Oil Engine", Vol. X, No. 119, March, 1943, pp. 282-283.*

Diesel Engines in Torpedoed Ship Keep Running.

An American cargo vessel in the West Indies was recently struck by a U-boat's torpedo forward and caught fire. The vessel developed a heavy list, whilst the bow was submerged about 15ft. Under these apparently hopeless conditions, the crew were ordered to abandon the ship and reached port safely. Shortly afterwards an aircraft on patrol duty reported the ship still afloat, whereupon a salvage crew was sent out accompanied by the chief engineer, second engineer, second mate, and some of the crew. The salvage crew succeeded in extinguishing the fire in several hours. The second engineer, who was the first to board the ship, found the two 8-cylr. Cooper-Bessemer auxiliary Diesel engines still running, and the lubrication system of both units functioning perfectly after 40 hours, even though the engines were tilted at a steep angle due to the ship's list. The main and auxiliary engines in this particular vessel were run on Bunker-C oil, a relatively cheap, low-grade fuel containing considerable carbon and other impurities.—*"Motorship", Vol. XXVIII, No. 3, March, 1943, pp. 224-225.*

Hull Corrosion and Fouling.

The paper comprises a brief review of the work of the Marine Corrosion Sub-Committee of the Iron and Steel Institute and the British Iron and Steel Federation. The recently published *First Report* of this Sub-Committee contains an account of the work carried out up to the autumn of 1942, and includes full experimental details of many branches of the Sub-Committee's work. Some of the tables of data given in this paper are taken from the *Report,* and the author expresses the hope that the present paper may serve as an introduction to that more detailed document, and as a medium for the presentation of one or two additional topics. It is, however, mainly concerned with the fouling of ship's plates and methods of combating it.—*Paper by G. D. Bengough, M.A., D.Sc., F.R.S., read at a meeting of the N.-E. Coast Institution of Engineers and Shipbuilders on the 16th April, 1943.*

Admiralty View of Propulsion by Electricity.

In reply to a question recently raised in the House of Commons regarding the adoption of electrical propulsion for British-built ships, the First Lord of the Admiralty stated that electric drive was not acceptable for the propulsion of warships in view of the additional weight and volume of the machinery required, and because it was less efficient than the system of transmission used in the Royal Navy. The Admiralty were aware that electric drive was being adopted in a considerable number of smaller naval craft in the U.S.A., but the production and fitting-out facilities available in America were greater than those of this country. Electrical propulsion was, however, being utilised in a number of British submarines as well as certain merchant vessels building in U.K. shipyards where production could be linked up with the necessary fitting-out facilities.—*"The Journal of Commerce" (Shipbuilding and Engineering Edition), No. 35,043, 22nd April, 1943, p. 7.*

The Largest Self-propelled Floating Crane.

A self-propelled floating crane with a maximum lifting capacity of 350 tons, recently completed in Germany, is reported to be the largest of its type in the world. It was built by the Demag A.G., and is fitted with Voith-Schneider propellers at each end, driven by electric motors. The current for these, as well as for the operating motors of the crane mechanism, is furnished by three Diesel-driven generators. The hull, which was built by the Deutsche Werft, has a length of 205ft. b.p., a breadth over frames of 108·2ft. and a depth of 17ft., the mean draught without load being 10·1ft. and the corresponding displacement 5,000 tons. The total complement comprises 3 officers and 20 men, for whom cabin accommodation is provided on board. There are three 8-cylr. M.W.M. type 4-stroke Diesel engines, with a cylr. diameter of 375 mm. and a piston stroke of 480 mm., the m.e.p. at full load being 72·5lb./in.^{2} and the corresponding output 900 b.h.p. per engine at 375 r.p.m. Each engine is directly coupled to an 800-kVA three-phase alternator supplying current at 850 volts 50 cycles to the main circuit. Two Voith-Schneider propellers are arranged one at each side of the after end of the hull, whilst the third is amidships at the bow end. Each propeller is driven by a 730-b.h.p. motor running at 325 r.p.m. Current for the auxiliary machinery, and for heating and lighting, is provided by a 220-volt system through two 200-kVA transformers. This circuit is also supplied by a 225-kVA three-phase generator driven at 500 r.p.m. by a 275-b.h.p. 6-cylr. M.W.M. Diesel engine. There is, in addition, an emergency generator of 14-kVA driven at 750 r.p.m. by a twin-cylinder M.W.M. engine rated at 20 b.h.p. The crane's maximum lift of 350 tons can be exerted at a radius of 60ft. from the hull side, the maximum lifting capacity at a radius of 160ft. being 50 tons. Trials of several days' duration were carried out with the crane and it is recorded that all conditions of the specification were satisfactorily fulfilled. An average speed astern was 7·38 knots, the speed of the Diesel-engined alternatives in each case being about 330 r.p.m.—*"The Motor Ship", Vol. XXIV, No. 279, April, 1943, pp. 12-17.*