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Electro-Magnetic Slip Couplings for use with Geared Diesel Engines for Ship Propulsion.

READ

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Synopsis.

THIS paper describes the electro-magnetic slip coupling and its application to geared diesel-engine-driven ships. The function and characteristics of the coupling are described and the electro-magnetic principles on which the design of the coupling is based are outlined.

The application of the coupling to geared diesel engines for ship propulsion is examined and a number of installations at present in service are described. A theoretical analysis of the transmission properties of the coupling is appended.

Introduction.

The diesel engine has held an established position as a power unit for the propulsion of ships for several years. In the early installations the conventional steam reciprocating engine was merely replaced by a diesel engine running at a comparatively slow speed which was coupled direct to the screw of the ship and no technique of motor ship propulsion could be said to exist. These installations proved to be well suited for ships of comparatively slow speed such as certain classes of cargo ships, coasters, trawlers, and tugs.

Many improvements have been made to diesel engines since the first installations were put in. Engine sizes have tended toward standardisation, engine speeds have increased and the cost, size and weight per horsepower have been reduced. The desirability of using these improvements in marine applications was quickly recognised. To-day the so-called high-speed geared diesel engine has shown itself to be a reliable power unit which has a smaller initial cost, maintenance, size and weight than its older slow-speed counterpart and to be a power unit eminently suited for the propulsion of ships. It is also an important consideration that when geared diesel engines are used the power required for propulsion can be supplied by two or more engines, enabling any single engine to be

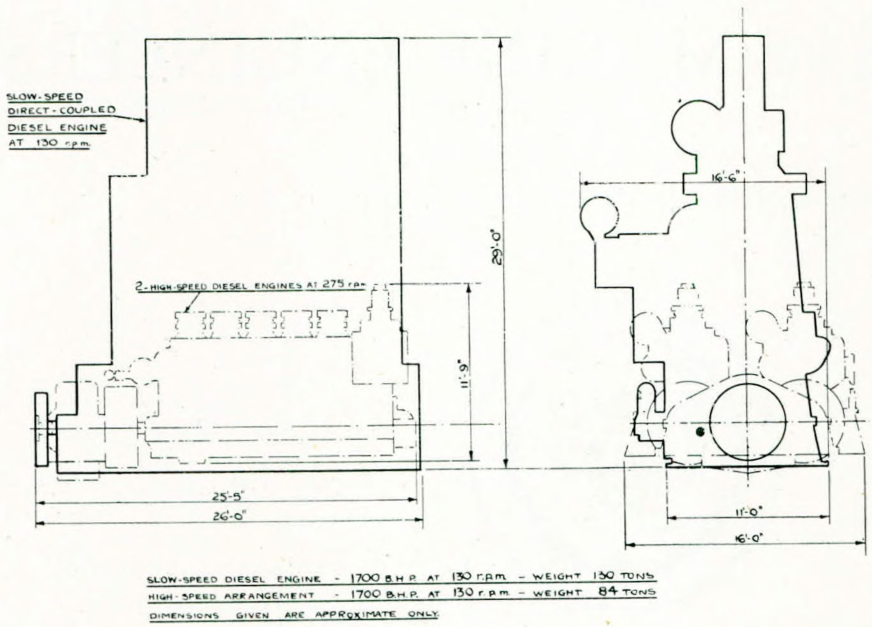


FIG. 1.—Comparison of slow-speed diesel engine and high-speed engines with geared drive.

shut down for repairs or for economic cruising at slow speed.

A comparison of the space requirements of a direct-coupled slow-speed diesel engine installation with a high-speed geared diesel engine arrangement is shown in Figs. 1 and 2. Fig. 1 compares a 1,700 b.h.p. 130 r.p.m. diesel engine direct-coupled to a propeller with two 850 b.h.p. high-speed geared diesel engines running at 275 r.p.m., and connected through electro-magnetic slip couplings and mechanical gearing to a propeller running at 130 r.p.m. The high-speed geared arrangement shows a saving of over 17ft. or 56 per cent. in headroom and a saving of 46 tons or 35 per cent. in weight. Fig. 2 compares a 3,800 b.h.p. 130 r.p.m. diesel engine direct-coupled to a propeller with four 950 b.h.p. high-speed geared diesel engines at 260 r.p.m. connected through electro-magnetic slip couplings and mechanical gearing to a propeller running at 130 r.p.m. The high-speed geared arrangement shows a saving of over 17ft. or 56 per cent. in head room and a saving of 70 tons or 28 per cent. in weight. These figures illustrate the possibilities of the high-speed geared diesel engine.

The diesel engine, in common with all reciprocating engines, has an uneven turning moment and is liable to transmit undesirable vibrations to the gearing and shafting connected to it. This possibility is furthermore accentuated when the working pressure, number of cylinders and speed of the engines are increased. It follows that the full advantages of comparatively high-speed geared diesel engine sets for marine propulsion can only be obtained by the use of some form of flexible coupling between the engines and the gear which will prevent the transmission of torsional vibrations from the engine crankshaft to the gear and permit two or more diesel engines to be simply

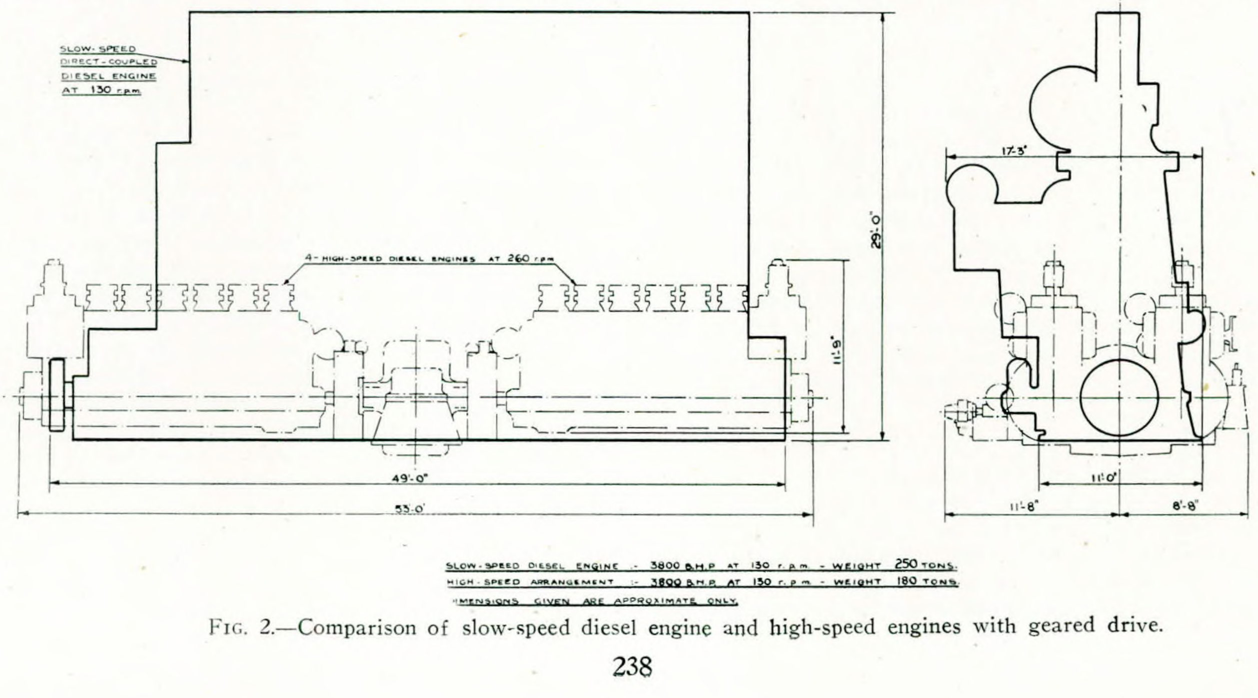


FIG. 2.—Comparison of slow-speed diesel engine and high-speed engines with geared drive.

and flexibly coupled and uncoupled through helical gearing to the propeller shaft. This problem has been solved for the special-duty ship by the use of an electrical equivalent of a mechanical gear, but diesel-electric propulsion is often too costly for normal-duty ships.

This paper describes the Asea electro-magnetic slip coupling as distinct from the diesel-electric drive and hydraulic coupling. The Asea coupling enables high-speed diesel engines of low cost, size and weight to be used with mechanical reduction gearing and solves the problem as far as normal-duty ships using high-speed geared diesel engines are concerned.

The Function of the Coupling.

A typical installation using two high-speed diesel engines connected through electro-magnetic slip couplings and mechanical reduction gearing to a slow-speed propeller is show in Fig. 3.

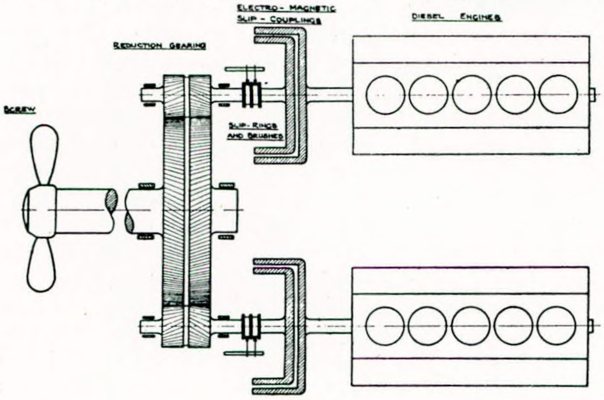


FIG. 3.

The functions of the coupling are :—

(*a*) To provide a flexible coupling between the diesel engines and the gear, which prevents torsional vibrations and shocks being transmitted from the diesel engine crank shaft to the gearing and propeller shaft;

(*b*) To permit two or more high-speed diesel engines to be simply and flexibly coupled and uncoupled through mechanical gearing to the propeller shaft.

Many important practical advantages arise directly from the fulfilment of these functions, viz.,

1. (1) the wear and maintenance of the mechanical reduction gear is reduced and a quieter running gear is obtained;
2. (2) the power required for propulsion can be supplied by two or more high-speed diesel engines of small size, weight and head room;
3. (3) increased reliability follows from having more than one engine;
4. (4) any single engine can be shut down for overhaul without stopping the ship;
5. (5) economic cruising at slow speed by shutting down one engine and proceeding at reduced speed on the other is rendered available.

The electro-magnetic slip couplings are designed in such a manner that

1. (1) the couple can be made and broken by the closing or opening of a single electric switch;
2. (2) the torque is transmitted through the coupling across an air gap so that the coupling is not subjected to wear;
3. (3) the torque transmitted by the coupling does not depend on the speed of the engine;
4. (4) the maximum torque that can be transmitted by the coupling is approximately twice the normal full load value. This protects the gearing against excessive stresses which might otherwise arise if some part of the machinery became locked;
5. (5) the inherent flexibility of the coupling permits a certain degree of misalignment of the two halves of the coupling and is conducive to increased reliability and simplifies the erection of the plant.

Construction and Electro-Magnetic Principles of the Coupling.

The general construction of the coupling is shown in Fig. 4. The construction is simple and comprises two electro-magnets; the primary electro-magnet (Fig. 5) is mounted on one shaft and excited from the auxiliary d.c. ship's supply, and the secondary electro-magnet (Fig. 6) is mounted on another shaft and excited inductively. The primary part is actually a multi-polar magnet ring and is connected to the high-speed gear pinion-shaft, and the secondary part is provided with a short circuited winding and is connected to the diesel engine (Fig. 7). The two parts of the coupling are thus overhung on their respective shafts and are separated by a radial air gap (there is no mechanical connection between them)

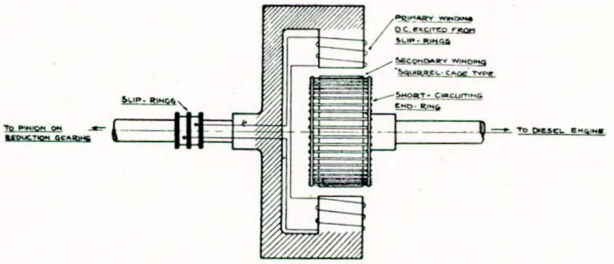


FIG. 4.—Electro-magnetic slip couplings.

The size of the air gap separating the two parts of the coupling depends upon the value of the torque to be transmitted and the speed of the

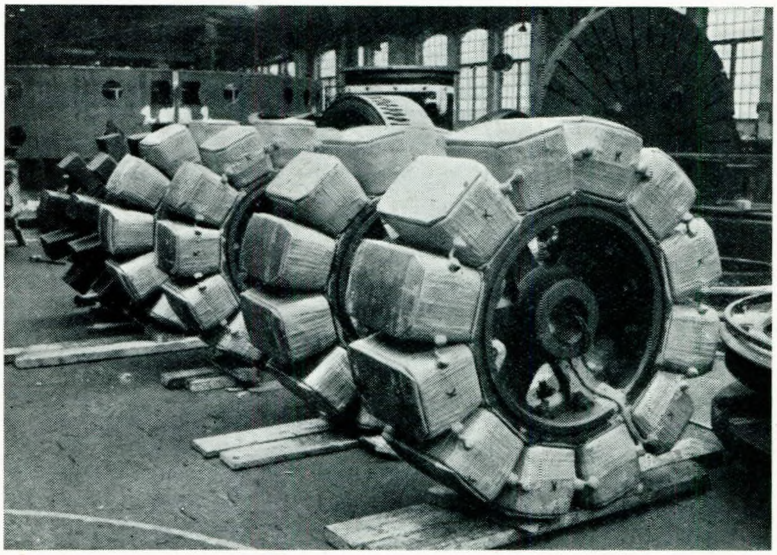


Fig. 5.

engines; it varies between 0·2″ and 0·4″. For constructional reasons the primary part of the coupling is generally placed on the inside and the secondary part on the outside, as shown in Fig. 7, but these positions can be reversed without affecting the characteristics of the coupling.

The primary electro-magnet is excited at any available d.c. voltage (generally 110 or 220 volts) and the actual windings themselves are very simple and are insulated from the iron magnet frame by means of mica. The short circuited secondary winding consists of heavy copper rods lying in uninsulated slots; the ends of the copper rods are welded into copper short circuiting rings. The coupling is designed so that in cases of emergency such as an electrical breakdown in the coupling or failure of the excitation supply, it is possible to couple mechanically the two halves of the coupling, thus transforming it into an ordinary flexible coupling for transmitting the drive direct from the engines to the propeller. The mechanical connection of the parts of the coupling can be made by means of rubber-cushioned bolts or similar means. Provision was made against this eventuality on the first two couplings built so that there was no fear of the ship being brought to a standstill by the electrical failure of the couplings. Subsequent experience has shown this precaution to be unnecessary as the use of two or more diesel engine units working on the same propeller has been found to be a sufficient standby. The outer part of the coupling is fitted with teeth for turning the diesel engine with screw gear.

The design of the Asea electro-magnetic slip coupling is based upon a fundamental law of electro-magnetism which states :—

> "The force exerted upon a current-carrying conductor when placed in a magnetic field is proportional to the conductor length, the current flowing in it, the intensity of the magnetic field and the sine of the angle made by the directions of the current and the flux cutting it".

It follows directly from this law that if a secondary system of current-carrying conductors is placed in a magnetic field as shown in Fig. 8 any movement in the magnetic field relative to the system of current-carrying conductors will result in a force tending to produce a similar movement in the conductors, and vice versa. When the primary part of the coupling is excited and rotated a magnetic field is produced and the lines of force pass from the poles across the air gap cutting the conductors of the secondary part of the coupling and inducing a current in them. A torque results from the interaction of this current and the magnetic field which causes the secondary system to rotate. Powerful currents are induced in the secondary windings with quite small relative differences in speed between the primary and secondary parts of the coupling. The interaction of these currents with the magnetic field produces a force which tends to eliminate the difference between the speed of the two parts of the coupling, and thus effects the transmission of torque across the air gap of the coupling.

The operation of the coupling is shown in Fig. 9, in which curve "a" is the no-load excitation curve of the coupling. This magnetisation curve is of the conventional form, the flux at first

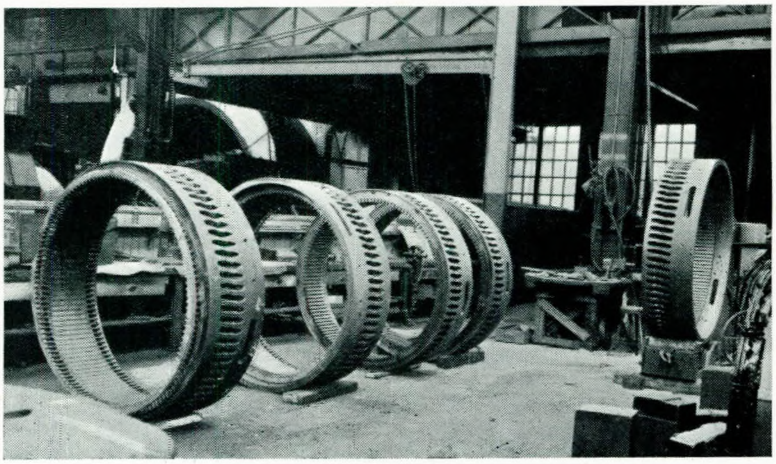


Fig. 6.

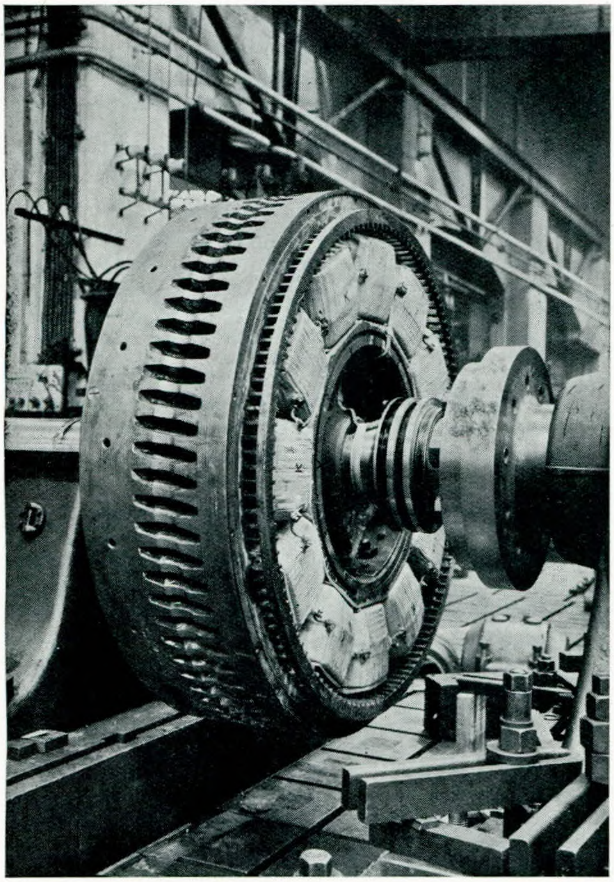


FIG. 7.

increasing in direct proportion to the excitation current and falling off as saturation is approached. It can be seen that the magnetic system of the coupling operates at a comparatively high induction density, enabling the size and weight of the magnetic system to be kept small. The torque transmitted by the coupling is proportional to the product of the magnetic flux and the current in the secondary system of conductors. This torque causes the speed of the secondary part of the coupling to increase until it reaches a value below that of the primary part that is just sufficient to circulate the secondary current required to transmit the torque. The small difference in speed between the two parts of the coupling is termed "the slip".

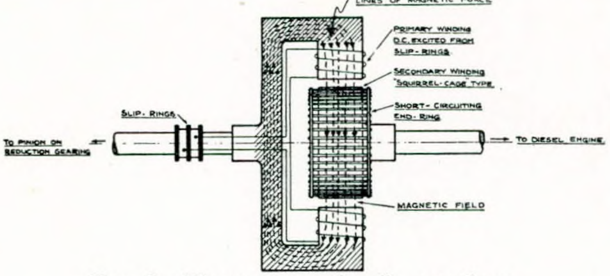


FIG. 8.—Electro-magnetic slip couplings.

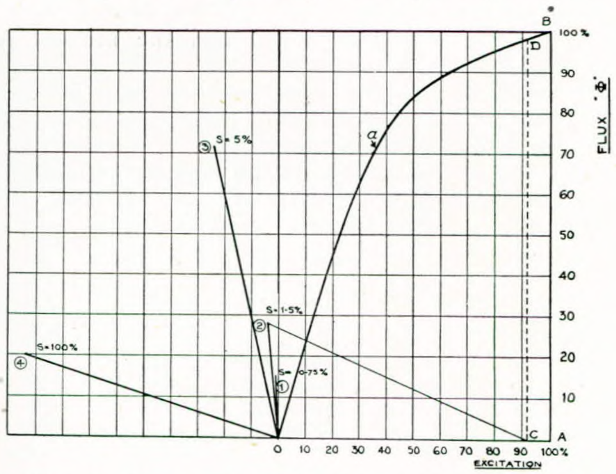


FIG. 9.—Electrical characteristics of the slip coupling.

The alternating currents induced in the secondary windings increase with the torque; the currents corresponding to four different values of torque are shown in Fig. 9 by 0-1, 0-2, 0-3 and 0-4. There is a different slip between the respective coupling halves corresponding to each torque; this slip is shown in the diagram for the four cases taken. The curve 0-2 corresponds to the transmission of 100 per cent. torque through the coupling, and at this torque the slip is shown to be 1·5 per cent. The currents induced in the secondary windings have a phase displacement relative to the magnetic field. This displacement is shown in the diagram where the angle separating the current curves from the vertical corresponds to the angle by which the current lags behind the flux. This phase displacement depends upon the slip; with small values of slip corresponding to the transmission of a small torque the phase angle is practically zero, and with large values of slip (current vector 0-4) the angle approaches 90°. For the normal full torque corresponding to curve 0-2 the displacement angle is about 5·5°, as can be seen in the figure.

The currents in the secondary system tend to demagnetise the magnetic field and therefore reduce the effective flux when the coupling is transmitting torque. The magnitude of this reaction can be determined by a simple geometrical construction, which is given for the normal loading corresponding to curve 0-2 in Fig. 9. From the point 2 the line 2-C is drawn equal to OA. The three sides of the triangle 0-2-C then represent the magnetising forces produced in the coupling. The line 0-C gives the resultant magnetising force which produces the working magnetic field; in the case shown in the figure, 0C (the magnetising force) is 92 per cent. and the corresponding field CD is 98 per cent. When the coupling is transmitting its rated torque corresponding to vector 0-2 the demagnetising effect of the secondary currents is small because the coupling is working well above the knee of the magnetisation curve. The torque which can be transmitted by the coupling is proportional to the magnetic flux ϕ, the secondary current 0-2 and the cosine of the angle of phase displacement θ. The conditions existing in the coupling when transmitting the torque values shown in Fig. 9 are summarised in Table I, from which it will be seen that the highest secondary current value which can be reached (at 100 per cent. slip) is about 3·3 times normal.

TABLE I.

Slip. %	Secondary current. %	Cos θ.	Field strength. %	Torque. %
0	0	—	100	0
0·75	51	—	99	51
1·5	103	0·99	98	100
5	277	0·94	82	213
100	329	0·23	31	25

Characteristics of the Coupling.

The coupling has well-defined characteristics which are described below.

The Torque Curve of the Coupling.

The electro-magnetic slip coupling has characteristics similar to those of the alternating-current squirrel-cage induction motor. The output torque is equal to the input torque and the slip can be adjusted if required to give characteristics similar to the slip-ring induction motor with rotor resistance. The variation of torque with slip for the example dealt with in Fig. 9 is plotted in Fig. 10. The curve in the figure shows that the torque increases with the slip until a maximum value equal to about 215 per cent. normal torque is reached at about 6 per cent. slip. The torque then decreases with increasing slip until finally with 100 per cent. slip, i.e. full speed on the engine with stationary propeller, a torque value equal to about 20 per cent. normal torque is reached.

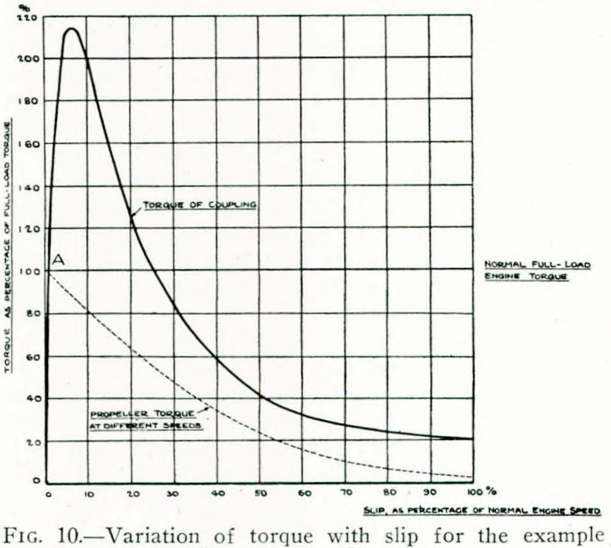


FIG. 10.—Variation of torque with slip for the example illustrated in Fig. 9.

The full load torque of the coupling is transmitted at 1·5 per cent. slip and the normal working range of the coupling lies between the points 0 and A; the coupling normally runs with a very small slip. If for any reason the torque to be transmitted exceeds the maximum value for the coupling, the coupling will "pull out of step" and the load on the engine will fall to the value corresponding to 20 per cent. torque. This characteristic is very valuable to the diesel engine by reason of its small overload capacity. It is well known that a diesel engine cannot in general exert a torque appreciably greater than its normal full load value, consequently any momentary demands for excess torque can only be supplied by the energy stored in the rotating parts. The coupling therefore protects the engine and shaft against excess stress which might occur if the propeller became locked, as for example when the vessel was navigating in ice.

The coupling can transmit momentary torques somewhat greater than the maximum value shown in Fig. 10 because the self-induction prevents the demagnetising effect of the secondary current from coming into effect immediately. The coupling is therefore somewhat less flexible in its reaction to momentary increases in torque and can transmit a torque up to three times normal for short periods of say ½ to 1 second. This permits the flywheel effect of the rotating parts to take effect and is not injurious to the engine. The dotted curve in Fig. 10 gives the torque required by a normal design of ship's propeller over a range of speed from 0 - 100 per cent. The torque of the coupling lies well above the torque curve of the propeller over the whole working range of speed, consequently the coupling is capable of transmitting all the torque required by the propeller under normal conditions and at all speeds.

It is rather difficult to make an exact determination mathematically of the torque for different conditions because part of the secondary current passes through the iron circuit and all higher harmonics of the field curve contribute to increase the transmission effect. A test has therefore been made on a coupling designed for an input of

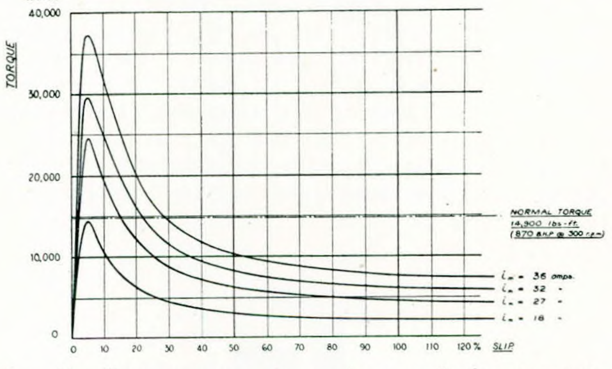


FIG. 11.—Torque curves for various excitation currents.

870 h.p. at 300 r.p.m., and the measured torque at various values of excitation current and slip up to 120 per cent. are given in Fig. 11. These curves show that the torque at standstill is rather larger than the calculated figure of 20 per cent.

The variation of torque with slip at various engine speeds is shown in Fig. 12. This figure shows that the torque that can be transmitted by the coupling is not dependent on engine speed but is determined only by the value of actual slip, i.e. the difference between the speeds of the two parts of the coupling. The coupling can therefore transmit the maximum torque exerted even at the lowest diesel engine speeds. This characteristic is particularly advantageous in the case of tugs, where it is necessary for the propeller to exert its most powerful propulsive efforts at low speeds.

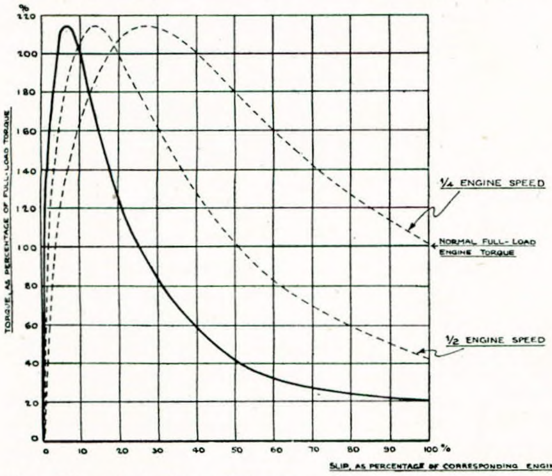


FIG. 12.—Variation of torque with slip at various engine speeds.

Efficiency.

The efficiency of power transmission through the coupling is very high, being of the order of 98 to 99 per cent. The small losses which occur in the secondary electro-magnet appear in the form of heat and are dissipated by the fan blades attached to the coupling. In addition there is a small amount of energy required from the ship's auxiliary supply system to excite the primary electro-magnet; this excitation energy is of the order of 1 - 2 per cent. of the energy transmitted.

Torsional Vibrations.

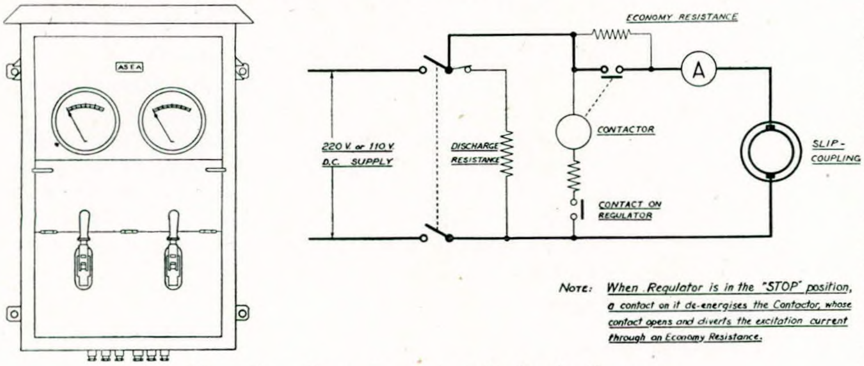
A large number of practical tests have been made to confirm that the Asea electro-magnetic slip coupling reduces to negligible proportions the cyclic irregularity and shock caused by the torsional oscillations from the engine crankshaft. These tests have been highly satisfactory and reference is made here to the mathematical analysis of the transmission properties of the coupling given in Appendix I. This analysis shows that the coupling reduces the torsional vibrations and shock imparted to it to a remarkable degree, and also reveals the fact that the greater the oscillation the greater is the damping effect of the coupling.

The torsional oscillations on the secondary side of the coupling correspond to those on the primary side but have a greatly reduced amplitude. It is impossible for resonance phenomena to occur in the coupling, which provides a practically smooth torque protecting the gearing and shaft from torque variations emanating from the Diesel engine and promoting a silent running gear. The analysis also shows that when calculating the torsional vibrations of the diesel engine shaft it is unnecessary to consider any masses beyond the coupling half actually fastened to the engine shaft. The effect of the other coupling half, gear and propeller can be neglected for all critical frequencies met with in practice.

System of Control and Operation.

The couple is made and broken in the coupling by the closing and opening of the switch which controls the supply to the excitation circuit. In practice the coupling is generally kept excited all the time the engines are in use and the various manœuvres are carried out on the engine controls. The engines are therefore started, run and reversed just as in the case of a direct-coupled diesel drive and work as if they were merely flexibly coupled to the propeller. The propeller speed follows the engine from moment to moment without any greater difference in speed than 1 or 2 per cent. (i.e. the slip required to give effective protection to the gearing against shock due to speed fluctuations) and there is no time lag between the control of the engines and the control of the propeller.

The speed of the engines is controlled by varying the fuel supply in the usual way, and no special automatic governor is required beyond the usual overspeed device which is in any case part of all standard marine diesel engines. When two or more



NOTE: When Regulator is in the "STOP" position, a contact on it de-energises the Contactor, whose contact opens and diverts the excitation current through an Economy Resistance.

FIG. 12A.—Control panel and circuit diagram.

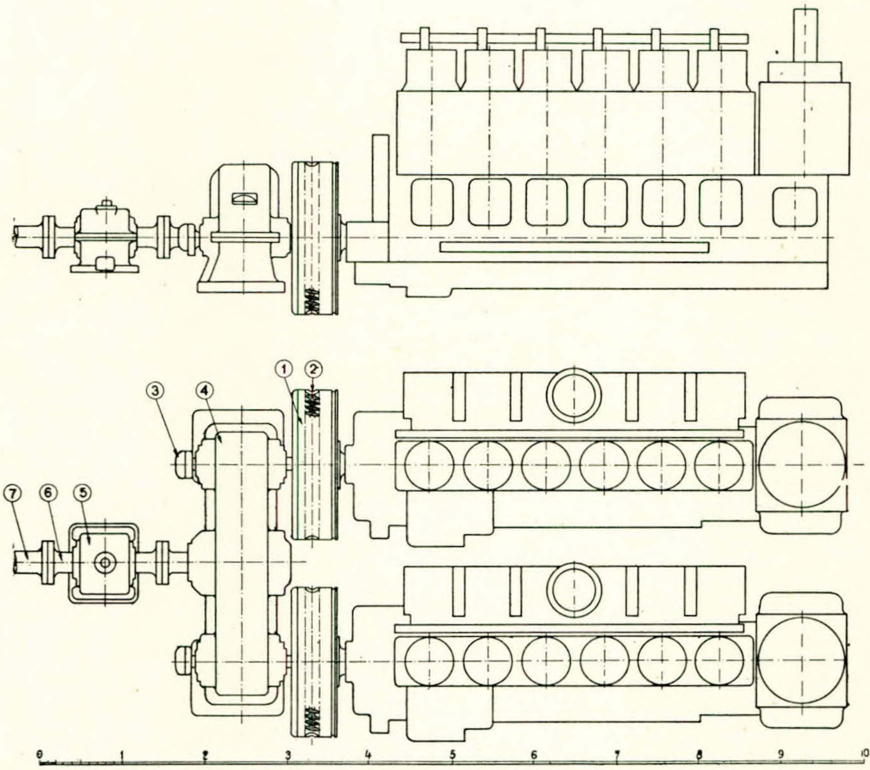


FIG. 13.—Typical single-screw two-engine installation.

engines are used the load can be controlled so that it is evenly divided among the diesel engines and there is no risk of any one engine being overloaded. This is done by adjusting the fuel control of each engine so that the couplings each run with the same value of slip. The slip is shown by a stroboscope, the operation of which is described in Appendix II. The coupling can facilitate manœuvring to a marked degree if needed; at slow ship speeds and where there is frequent starting and stopping it is possible to run the engines continuously at slow speed and control the stopping and starting of the propeller by opening and closing the excitation switch, using the coupling as a clutch. This method of operation gives a very rapid and smooth control of the propeller, facilitates manœuvring and economises starting air. It is further possible to keep one engine running ahead and a second engine running astern and so propel the ship ahead or astern without reversing the engines. This advantage can be of importance when manœuvring in crowded and narrow channels.

The electrical circuits associated with the couplings are very simple, as can be seen in Fig. 12A, which shows the excitation circuit and a view of the control panel.

Installations.

It will be clear from the development of the subject so far that the coupling finds its fullest application in ships using more than one engine per screw. Installations can be made on single and multi-screw ships and typical installations are briefly described below.

Single-Screw Two-Engine Installation.

The arrangement and overall dimensions of a typical installation using two 1,050 b.h.p. diesel engines running at 300 r.p.m., coupled through slip couplings and single reduction gearing to a propeller running at 130 r.p.m., is shown in Fig. 13. The propulsion unit shown was installed in the m.s. "Astrid Thorden" built by Chrichton-Vulcan, Åbo, using diesel engines built by Messrs. Atlas Diesel, Stockholm.

An installation of a similar equipment installed in the m.s. "Astri" built by Lindholmen A/B, Gothenburg is shown in Fig. 14. This installation comprises two 870 b.h.p. Atlas diesel engines coupled through slip couplings and single reduction gearing to a slower speed propeller. A more detailed view of the actual reduction gear and the primary part of the couplings is shown in Fig. 15. In this figure it can be seen that the enclosed slip-rings for the excitation supply to the coupling are placed on the after end of the pinion shaft and the leads between the slip-rings and the primary winding are brought through the centre of the shaft. This arrangement is used to make the slip-rings more accessible and to reduce the axial length of the coupling. The diesel engines are controlled by means of the two operating handles shown in Fig. 14 and may be controlled singly or together. One of the operating handles is used for starting, stopping and reversing and the other handle is used for regulating the oil supply.

Single-Screw Four-Engine Installation.

The arrangement of a typical installation using four 1,100 b.h.p. Atlas diesel engines running at 300 r.p.m., coupled through slip couplings and single reduction gearing to a propeller running at 90 r.p.m. is shown in Fig. 16. The installation shown was installed in the m.s. "Dagmar Salen" built by Oresundsvarvet A/B Landskrona. This ship underwent sea trials on September 8th and went into regular service the same day.

A more detailed view of the slip couplings and reduction gearing is shown in Figs. 17 and 18. In this installation it can be seen that the slip-rings are mounted between the primary part of the coupling and the gear.

Twin-Screw Four Engine Installation.

A twin-screw four-engine installation is now in the course of manufacture, but unfortunately arrangement drawings will not be ready in time for inclusion in this paper.

Conclusion.

The so-called high-speed diesel engine is now accepted as a power unit eminently suited for the propulsion of ships, and experience with installations at sea has shown that the couplings are extremely reliable, simple to install and operate, and fulfil their function in every respect. The authors consider that the Asea electro-magnetic coupling has proved itself a most reliable and simple medium for combining high-speed diesel engines with slower-speed propellers and are of the opinion that it would be very difficult to design a coupling of any other than electro-magnetic form to give such a high performance so simply. The coupling by reason of its great flexibility enables very satisfactory running of the gears to be obtained since it eliminates practically all dangerous mechanical stresses, including those caused by slight movements of the engine bed. The saving in size and weight has already been referred to but it is interesting to note that standard single-screw two-engine transmission units can be built with a weight of only 22lb. per engine b.h.p. This weight includes the propeller, thrust bearing, reduction gear and two couplings and is based on the assumption that for a 600 h.p. installation two 300 h.p., 600 r.p.m. engines are used, for a 1,200 h.p. installation two 600 h.p., 400 r.p.m. engines are used, and for 3,000 h.p., two 1,500 h.p., 300 r.p.m. engines are used.

Even those who consider that the use of one large diesel engine for ship propulsion leaves nothing to be desired will admit that the advantage of shutting down one engine for overhaul or for cruising at slow speeds can in many cases be very considerable. The flexibility of the whole installation also makes it ideal for fitting into existing ships and bringing them up to date as far as the propulsion equipment is concerned.

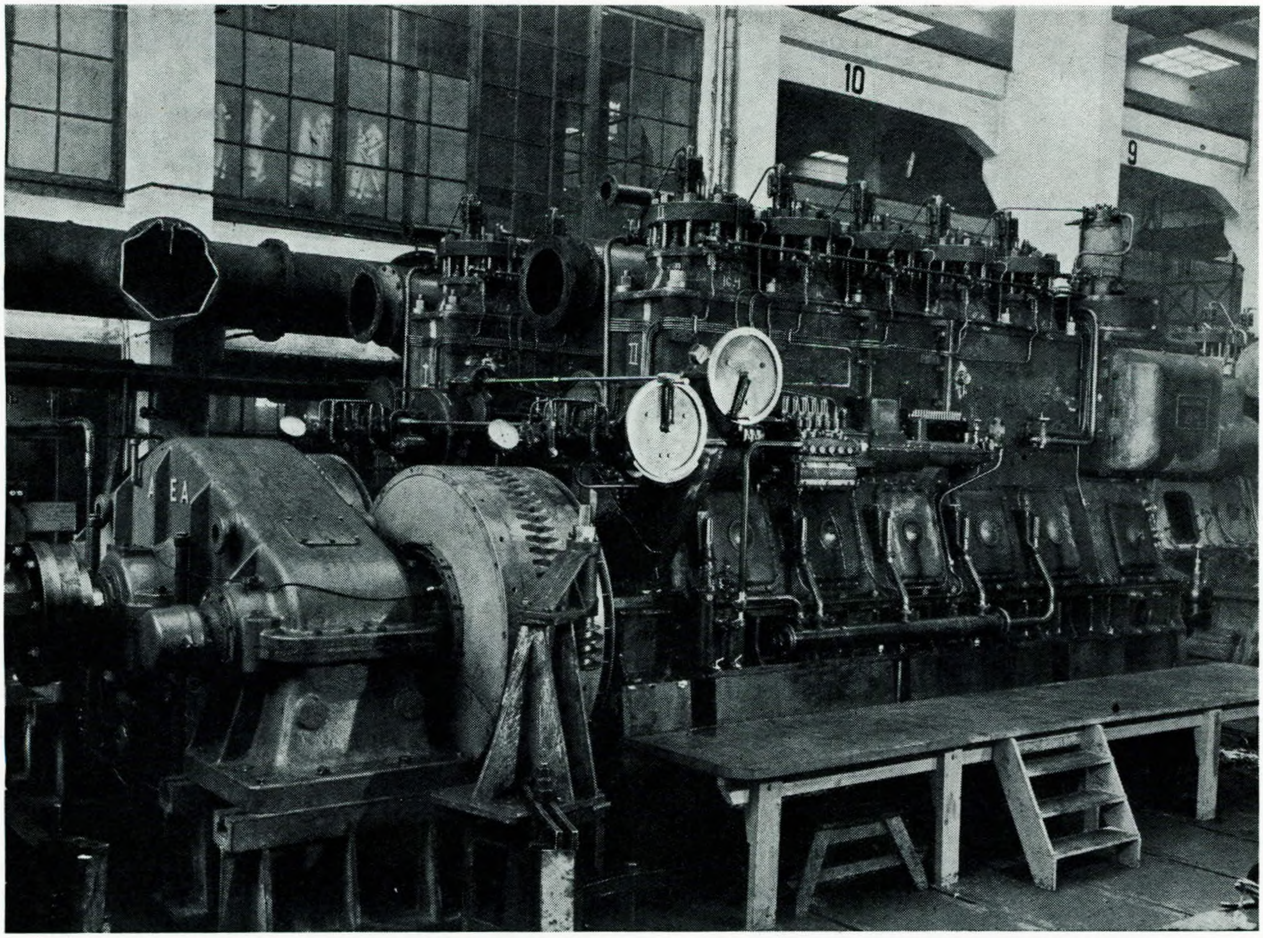


FIG. 14.—Equipment installed in the m.s. "Astri".

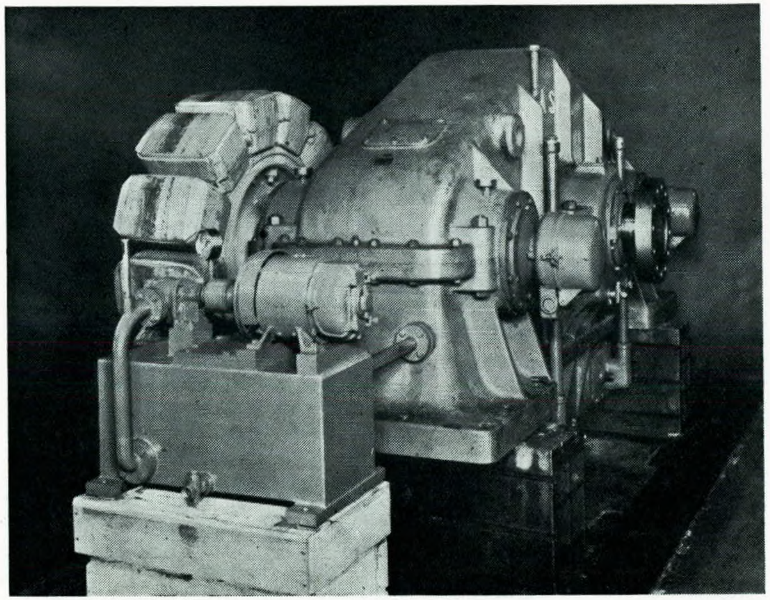


Fig. 15.--Detailed view of reduction gearing and primary part of the coupling fitted in the m.s. "Astri".

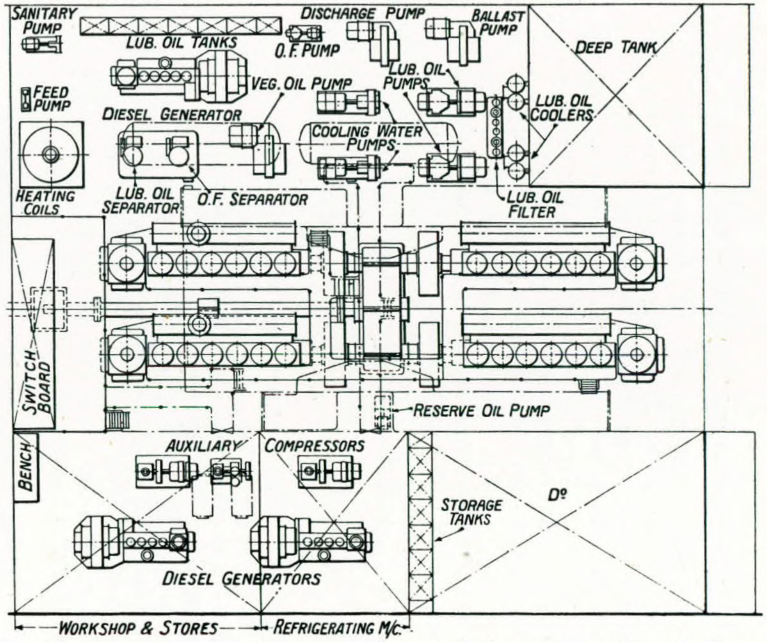


Fig. 16.—Arrangement of diesel engines and electro-magnetic couplings in the "Dagmar Salen".

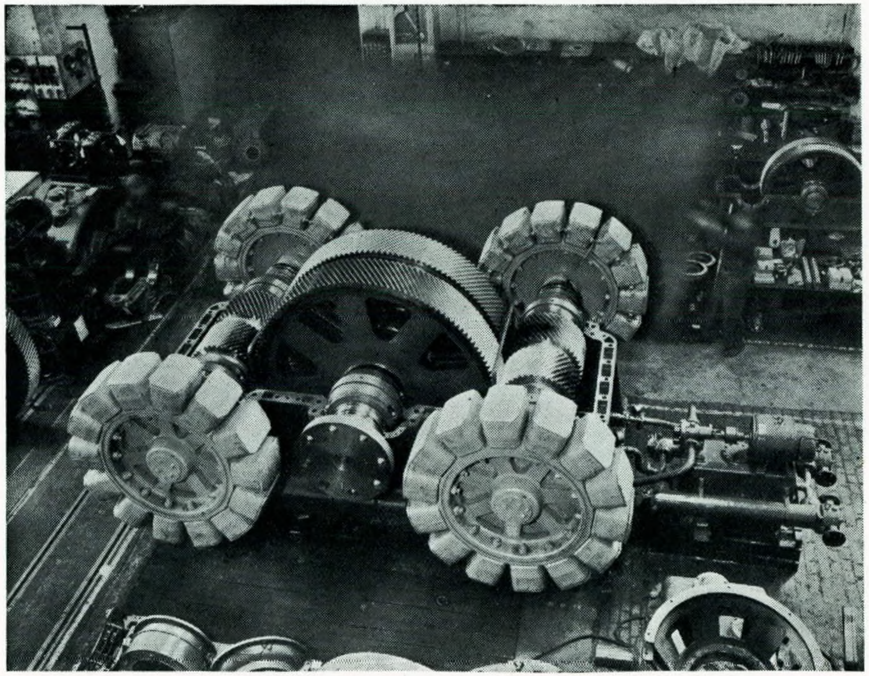


Fig. 17.—The couplings, pinions and main gear wheel as fitted in the "Dagmar Salen".

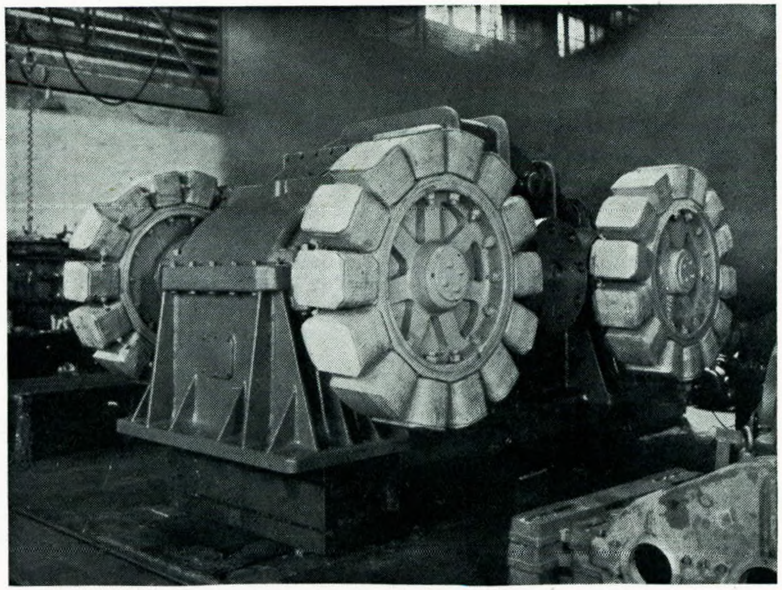


Fig. 18.—Slip coupling and gearing installation in the "Dagmar Salen".

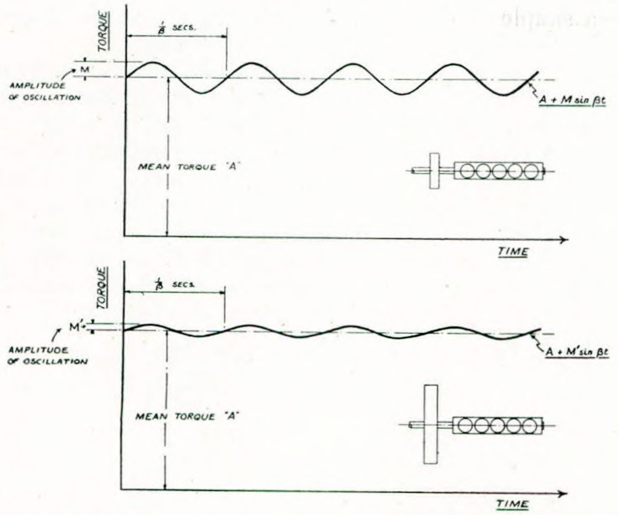


FIG. 19.—Crankshaft torque with various flywheel sizes.

In conclusion the authors wish to express their thanks to Allmänna Svenska Elektriska A.B. (ASEA) for permission to publish this paper and to record their appreciation of the expert advice on diesel engines given them by Mr. B. J. O. Stromberg and Mr. R. E. Ellis of The Atlas Diesel Co., Ltd., and for the assistance and help given by Mr. E. W. Goodman, B.A.(Cantab.) in the general preparation of the paper.

APPENDIX I.

Analysis of Transmission Properties of Couplings.

This investigation analyses the process by which the torsional oscillations produced by the fluctuating torque of the diesel engine crankshaft are damped to negligible proportions by the couplings. The fluctuating torque of a diesel engine crank shaft is a cyclic function and can, therefore, be resolved by means of the Fourier Theorem into a constant torque and a number of simple sinusoidal components of differing frequency. Each of these simple harmonic components contributes to the torque and the total of these components is equal to the torque delivered by the engine crank shaft to the primary part of the coupling (*see* Fig. 19). The calculations which follow are based on a torque exerted by the engine of the form :—

$$T = M \sin \beta t + \quad \ldots \quad \ldots \qquad (1)$$

The reaction of the coupling to this simple harmonic torque will be similar in form to its reaction to all the other simple harmonic components which go to make up the torque pulsations of the normal diesel engine; the results obtained from the investigation will therefore be equally true for the train of simple harmonic torques which form the total torque as they are for the simple case taken in this hypothesis.

In the hypothetical case shown in Fig. 20 let

I_1 = Moment of inertia of primary system (i.e. of engine and primary coupling half).

I_2 = Moment of inertia of secondary system (i.e. of propeller, gears, etc. + secondary coupling half).

ω_1 = Instantaneous angular velocity of the diesel engine crankshaft (i.e. of primary system) due to the torque of the engine.

ω_2 = Instantaneous angular velocity of gear pinion shaft (i.e. of secondary system).

Then $\omega_1 - \omega_2$ = Instantaneous value of slip.

m = a "constant" for the coupling which determines the magnitude of the torque transmitted in terms of slip. Thus for any slip $(\omega_1 - \omega_2)$ the torque transmitted is m $(\omega_1 - \omega_2)$. Reference to the typical torque/slip curve in Fig. 10 will show that "m" is practically constant over the normal working range of the coupling because the torque/slip curve is almost a straight line; "m" is therefore a measure of the rigidity of the coupling. For higher frequencies of pulsation "m" decreases above a certain value, making the coupling more flexible at higher frequencies.

The torque input T_1 applied to the air gap of the coupling is given by the torque from the engine crank shaft less the torque absorbed in accelerating the masses of the primary system I_1.

Thus

$$T_1 = M \sin \beta t - I_1 \frac{d\omega_1}{dt} \quad \ldots \quad \ldots \quad \ldots \qquad (2)$$

The torque output of the coupling available for accelerating the masses of the secondary system I_2 is equal to the torque input and can be written

$$T_1 = m\,(\omega_1 - \omega_2) \quad \ldots \quad \ldots \quad \ldots \quad \ldots \qquad (3)$$

$$\text{and } T_1 = I_2 \frac{d\omega_2}{dt} \quad \ldots \quad \ldots \quad \ldots \quad \ldots \qquad (4)$$

These three differential equations may be solved simultaneously, and give ω_1 and ω_2 in terms of sin β and cos β. These represent sinusoidal fluctuations of ω_1 and ω_2 of amplitudes (for ω_1)

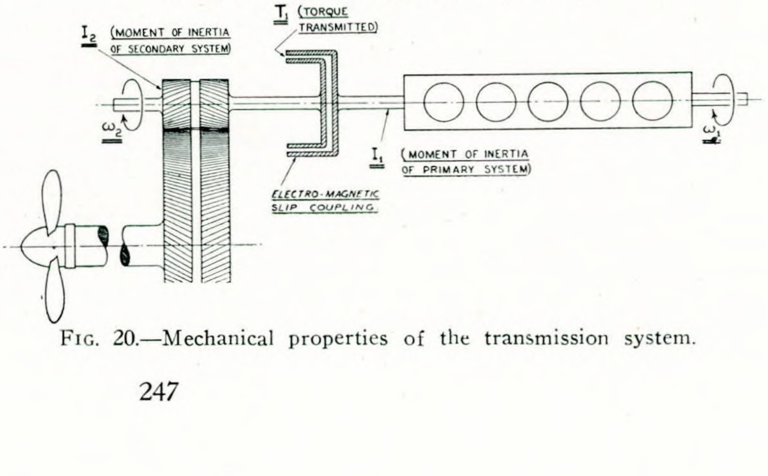


FIG. 20.—Mechanical properties of the transmission system.

$$\frac{M}{\beta}\cdot\frac{\sqrt{1+\left(\frac{\beta I_2}{m}\right)^2}}{\sqrt{\left(\frac{I_1 I_2 \beta}{m}\right)^2+(I_1+I_2)^2}}$$

and (for ω_2)

$$\frac{M}{\beta}\cdot\frac{1}{\sqrt{\left(\frac{I_1 I_2 \beta}{m}\right)^2+(I_1+I_2)^2}}\quad\ldots\qquad(5)$$

It can be seen from equation (5) that the amplitude of "ω_2" the angular velocity of the secondary system decreases as "m" the rigidity of the coupling decreases or as β the frequency of the applied torque component increases. An increase of "m" is equivalent to making the coupling more rigid (*see* definition of "m" above). When "m" tends to ∞ the amplitude of the angular velocity of the secondary system given in equation (5) simplifies to

$$\frac{M}{\beta}\cdot\frac{1}{I_1+I_2}\quad\ldots\quad\ldots\qquad(6)$$

This is the condition existing when a rigid coupling is used but for all definite values of "m" expression (5) will be less than expression (6) showing that the flexibility of the coupling does actually reduce the amplitude of the oscillations transmitted. An increase of β has a similar effect showing that the coupling damps out higher frequency oscillations even more readily than low frequency oscillations.

APPENDIX II.

Stroboscopic Device for Adjusting the Torque Transmitted by the Couplings.

The operation of the coupling is based on the existence of a small but continuous slip between the primary and secondary parts which is practically proportional to the value of torque transmitted over the working range. The slip therefore presents a simple method of measuring the power developed by the diesel engine driving the coupling.

For measuring the slip the couplings are provided with a simple stroboscopic arrangement consisting of two thin metal discs one of which is fixed to the primary part of the coupling and the other to the secondary part of the coupling. Each disc is provided with a number of holes in the periphery; one disc runs inside the other and a small lamp is placed behind them as shown in Fig. 21. Each time a hole in one disc coincides with a hole in the other disc a sharp light impression of the lamp appears. The number of light impressions per minute gives an exact measure of the slip. This method of measuring the slip has a high degree of accuracy and enables the power developed by the engines to be compared and adjustments made so that the load is equally shared by the engines.

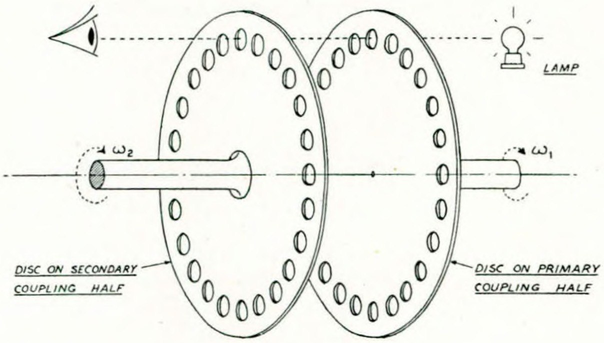


FIG. 21.—Stroboscopic device.

The engineer is also provided with a table which enables the b.h.p. developed by each engine to be found, provided the excitation current given on the ammeter and the number of light impressions per minute (obtainable from the stroboscope) are known.

Discussion.

Mr. G. O. Watson (Member), opening the discussion, said that in this country they had not had very much experience of geared Diesel drives, as it was a development which was peculiar to the Continent where it was favoured. He believed there were between 40 and 50 motor ships fitted with various types of couplings and gearing. These vessels ranged from 1,000 to 12,000 h.p., and about half of them originated in Germany; most of them, however, were between 3,000 and 6,000 h.p. in power.

In the paper read by Mr. Shoosmith before The Institute last month the advantages of the slow-speed propeller were stressed, and he thought it was generally admitted that, with regard to propeller speed, modern Diesel drives were more or less a compromise. Whereas it was more economical to run the propeller at slow speed, it was necessary to run the Diesel engine at a slightly higher speed; so if gears were used, some form of flexible coupling was necessary to damp out the fluctuations in torque due to cyclic irregularities and torsional vibrations.

The only parts of the coupling described subject to wear were the brushes on the slip ring. As was the case with most devices, there was a weak link somewhere in the chain, and in this instance the vulnerable feature was the d.c. field coils. The manufacturers of this coupling would be well advised to pay great attention to the field coils and to design them with a wide factor of safety, i.e., not to run them too near the limit for temperature rise and to ensure that they were mechanically sound. Every a.c. electric propulsion equipment depended on a revolving d.c. field, and they knew from the reliability of a.c. propulsion drives that these were remarkably free from failure from every point of view. This coupling, however, involved using slightly smaller copper sections than was usual in a.c. synchronous motor drives, so that, as he had said before, the manufacturers would be well advised to pay a lot of attention to the field coils.

It was stated in the paper that it was possible to couple mechanically the two halves of the coupling by means of rubber-cushioned bolts in cases of emergency. That had not been provided for in the more recent equipments, so that if the coupling failed without these bolts on board, nothing could be done.

It was also stated in the paper that there could be a certain amount of mal-alignment of the two halves, but there was a limit to this. If there was any eccentricity in the air gap an unbalanced pull was produced in the direction of the smallest air gap. In this particular coupling the gap was of the order of 10mm. and the unbalanced pull was of the order of 1,000kg. per mm. So if there was as much as 1mm. error in the air gap this would result in a big unbalanced pull. It was necessary from time to time to check the air gap to ensure that no change had taken place. From the point of view of the engine it was advisable to see that all wear was taken up.

Finally, he would draw attention to an instance which demonstrated one of the features of multi-engine schemes using this form of coupling, i.e., reliability. An outward bound ship which had two engines suffered an ordinary engine failure, but it was able to carry on with one engine and to return to a European port to have the trouble remedied. This was a good example of the reliability obtained by using this coupling.

Mr. J. H. Wheadon (Member) said that the description given of the construction and working of this coupling was very complete and they would probably agree that both electrically and mechanically it was a very robust and simple device.

Looking at the paper from the shipowners' point of view, however, several very important points were not quite clear. Fig. 2 compared a high-speed geared installation with a slow-speed direct-coupled engine. This comparison in his opinion did not give sufficient credit to the modern direct driving Diesel engine, and in order that the comparison might be more complete would the authors state the type of engine they chose as a comparison? He had before him the dimensions of a double-acting two-stroke engine, many of which type were being built at present. This engine developed the same power at the same propeller speed as those shown in Fig. 2, but the overall length was only 35ft. as compared with 53ft. for the geared drive, i.e., a saving of 18ft. in length of engine room. He agreed the geared installation showed to advantage so far as head room was concerned, but he had been told by naval architects that space gained by shortening the engine room was of more value than a cargo space over the engine room. Particularly was this so in the case of insulated vessels and vessels for special cargoes. Furthermore, to deal with cargo in such a space over the engine room would necessitate extra derricks and winches on the boat deck.

With regard to breadth, no advantage appeared to be gained by adopting the geared machinery—in fact in the case of a twin-screw vessel of ordinary dimensions there appeared to be very little space left in the engine room wings for auxiliaries. However, as the authors referred to such an installation on page 245, they might be able to enlarge on this.

So far as weight was concerned, there seemed no doubt that upon the same basis of propeller speed, the geared system showed to advantage. Against the saving in engine weight, however, there would probably be extra fuel to carry and an additional generator for excitation purposes.

The authors, so far as he could see, did not claim any saving in fuel costs, excepting perhaps on ballast voyages when certain units might be cut out. As during the greater part of a vessel's life, however, it was operating at full power or nearly so, it would be of value to owners to know the probable overall fuel consumption.

The authors referred to transmission losses—slip, excitation and gearing—totalling about 4 to 5 per cent. No mention was made of the greater specific fuel consumption of the small high-speed Diesel engine. This would possibly amount to another 5 per cent. and should be added to the transmission losses. Would the authors state the specific fuel consumption based upon power transmitted to the propeller shaft?

As the lubricating oil bill for Diesel vessels was a very important item in the running costs, it would be interesting to know if the authors considered this would be higher with the high-speed engines with direct-Diesel propulsion.

Finally, a word with regard to costs. The authors stated that the cost of this type of machinery compared favourably with the cost of Diesel-electric plant. It would be of perhaps greater interest to owners to know how the cost compared with the direct-Diesel propulsion.

Mr. F. Johnston (Visitor) said that he did not like the name "slip" coupling because it conveyed to him the idea of getting variation in speed by coupling slip. Actually it only slipped because it happened to be an induction wound motor.

In Figs. 1 and 2 the authors compared a 1,700 b.h.p. 130 r.p.m. direct-coupled Diesel engine with two 850 b.h.p. geared Diesel engines running at 275 r.p.m. and a 3,800 b.h.p. 130 r.p.m. direct-coupled engine with four 950 b.h.p. high-speed geared engines running at 260 r.p.m. It seemed to him that just double speed for the smaller engine was hardly high enough, and it would be noted that on page 245 the authors referred to very much higher relative speeds, i.e., two 1,500 h.p. 300 r.p.m. engines.

It was mentioned that the two parts of the coupling were overhung. They had to be, but on reference to Fig. 7 he was not sure that he liked this. Had the authors thought of spigoting these two parts together to take some of the weight? The relatively big air gap was possibly due to the fact that they were overhung.

In Fig. 9 illustrating the electrical characteristics of the coupling, the lengths of the vectors (1), (2), (3), and (4) presumably represented the strength of the current. No. 3, which had a 5 per cent. slip should be compared with the length of No. 4 with a 100 per cent. slip and it would be found that No. 4 was less than No. 3, whereas on turning to Table I it would be noted that with the 5 per cent. slip the current value was about 20 per cent. greater. It looked to him as if vector (4) should be much longer.

It was stated in the paper that the coupling could transmit the maximum torque exerted even at the lowest engine speeds. That required explanation because although it did exert the maximum tractive effort at low speeds, how long could say a 5 per cent. slip be maintained without overheating?

An economy resistance was shown in the diagram of connections, Fig. 12a. Did that mean that when the regulator was put to "STOP" there was still a current flowing through the field?

It seemed to him that having gone so far it was a pity not to go a step further and have a generator on each engine and couple them up with solid bars each to its own motor. It need not be a purely electric drive and all the reversing could still be done on the engines, so eliminating all the electrical safety requirements usually associated with that form of drive.

He thought that the facility of being able to run the engines while the ship was against the quay was an advantage which should have been stressed. Everyone knew what an advantage it was to warm up a motor car engine before starting, and this was equally applicable to a big Diesel engine.

Mr. A. C. Basebe (Visitor) asked the authors to give the overall mechanical efficiency of the coupling, including such items as increased bearing loss due to the weight of the rotor on the main bearing.

Windage would include the cooling effect obtained by the rotating portion. This windage, of course, was governed to a certain extent by the volume of air required to dissipate the heat losses, and by the slip and excitation power consumption.

In a twin-engined vessel, if one engine failed then the speed of the propeller would fall due to the fact that one engine could not maintain the torque required to keep the propeller at full speed. Presumably, in this case, the engine speed would be pulled down, but at the same time the torque of the propeller would fall until a natural balance was established. Owing, however, to the fact that the coupling would not slip beyond 6 per cent. until approximately 200 per cent. torque was applied thereto, it would mean that the one remaining engine would be overloaded. In the case of the propeller becoming choked with ice then (as the propeller speed was brought down) the torque would rise, but the engine would not be able to develop twice full load torque which would be necessary to cause the coupling to get up to the full torque portion of its slip curve. Thus the engine speed would be progressively brought down, and ultimately the engine would stall due to its inability to develop the necessary twice full load torque to get it over the peak, as indicated in Fig. 10 in the paper.

Mr. S. N. Kent (Member of Council) said that the paper dealt with slip couplings for use with geared Diesel engines for ship propulsion and it was suggested that high-speed Diesel engines were, from the point of view of weight and economy, likely to be most useful for special-duty ships.

The speaker assumed the gearing to be any type or make, provided the ratio of the wheels was to requirements, and presumably the lower half main bearing of the secondary winding shaft was made so that it could be adjusted.

In a four-engine equipment, was it right to assume that the four small pinion wheels were separate one from the other?

When he visited the makers' works in the latter part of 1936 he saw the manufacture of this coupling in its various stages and the making of the gearing, and he was much impressed by the care that was exercised in gear-cutting and the maintenance of the plant for this, as well as the care and skill exhibited in the manufacture of the electrical parts.

One could not help but be impressed by the fact that this was a coupling of which the two sections were distinctly separate and could not come into mechanical contact with one another. The flexibility of such a coupling was manifest and its application to marine work should be particularly attractive. Its cost in consumption seemed low and the loss altogether was small, but even were the losses slightly greater, seeing that there was no wear and tear, the upkeep and maintenance must be small.

Furthermore, the authors had pointed out that this coupling could facilitate manœuvring to a marked degree if needed, as at slow ship speeds and during frequent starting and stopping it was possible to run the engines continuously at slow speed and control the stopping and starting of the propeller by opening and closing the excitation switch, using the coupling as a clutch. Was this practice recommended and, in such circumstances, did it involve an additional engineer officer in attendance at the controls when manœuvring?

He had seen no reference to any temperature rise in the electrical part of the installation, but no doubt special arrangements had been made to ensure perfect ventilation of the space in which the coupling was placed, so as to control the temperature and to keep such space free from oily vapour, etc.

What overhauling was likely to arise with regard to the coupling, etc., and what facilities were provided to gain access to the various clutch parts?

One other point to which his attention was attracted was that the combined turning wheel, etc., was of large diameter and the tank top would have to be penetrated for this housing. Could this be avoided by increasing the length of the clutch and reducing its diameter relatively?

He also desired to know if any torsional vibrations had been taken on any vessel fitted with this installation, and if so with what result.

A vessel pitching in a high sea would normally cause considerable racing of the main engines. What was the effect in the case of a vessel fitted with this coupling?

The question of multi-small engines as a substitute for the large engine was eminently worthwhile considering for other than tramp practice, but this seemed to be beyond the scope of the paper. It must, however, be taken into consideration where the combination of engine and coupling was a necessity.

To what further purpose was this coupling being applied, and what were the smallest and largest powers for which its use was contemplated?

Dr. R. G. Jakeman (Visitor) said that the authors had explained the action of this coupling and had pointed out that the torque characteristics were practically the same as for an induction motor. He wished to put forward a rather different explanation which appealed to him as an electrical designer. The ordinary three-phase induction motor had three-phase current supplied to its stator windings and that produced a rotating magnetic flux which actually rotated at a speed corresponding to the frequency of the supply and the number of poles. That rotating flux, by cutting the bars of the squirrel-cage winding, induced a voltage which sent a current through them and produced torque, so that the rotor ran at a speed which was slightly below this rotating flux.

The main components of an induction motor were therefore the rotating flux and the rotor turning at a speed slightly lower. This flux could be produced in any desired way and the slip coupling was one way of doing it. The coupling had poles excited by direct current which were rotated mechanically and that clearly produced a rotating flux. From this point the performance was exactly the same as an induction motor. It would be seen from the diagram, Fig. 9, that the ends of the current vectors lay approximately on a circle and this corresponded to the ordinary circle diagram of the induction motor.

In an induction motor energy was supplied electrically, converted into magnetic energy in the air gap and again converted into mechanical energy. In a slip coupling the energy was supplied mechanically, converted into magnetic energy in the air gap and then converted into mechanical energy. The performance was otherwise exactly the same as that of the induction motor. The coupling could never run at synchronous speed since there would be no relative motion between the two parts and therefore no current in the squirrel-cage winding. It was clear from this that there must always be some slip.

He was sorry that the authors perpetuated the expression "to fall out of step", because with an induction motor characteristic the coupling, never being in step, could never fall out of step. The expression normally used for induction motors was that the motor "stalls".

Lieut.-Comd'r. (E.) C. M. Hall, R.N. (Engineer-in-Chief's Department, Admiralty) said that the flexibility, the elimination of torsional vibrations and the ease of manœuvring were attractive features of the coupling, but it seemed rather bulky for the horse-powers and speeds quoted.

He would draw attention to one engine of 1,050 h.p. output at 300 r.p.m. referred to in the paper. What would be the reduction in size of the coupling by running the engine at 500 instead of 300 r.p.m., and what would be the size of the coupling for a 2,000 h.p. engine running at 400 r.p.m. and also a 1,200 h.p. engine running at 1,000 r.p.m.?

They had been told that the torque transmitted depended on the primary current and the flux density, and it seemed this was limited by the size of the rotor; also presumably there was, from the centrifugal point of view, a limiting speed on the winding. Therefore, was there not a limit to the h.p. which the coupling could transmit? He also desired to know if the temperature rise was a serious consideration.

A further question was, could the coupling be designed to slip at quite a small overload, say 10 or 20 per cent.?

With regard to the ingenious method of measuring the h.p. by the stroboscope, it was an excellent diagram but he would like to know how it was fitted up in practice.

Mr. J. A. Jaffrey, M.Sc. (Member) said that the secondary part of the slip coupling was usually overhung from the Diesel engine shaft, and he asked the authors whether they had any information comparing the weights of the secondary part of the slip coupling with the standard flywheels usually adopted when the engine was driving a generator, and also when the engine was the prime mover for a system not requiring so small a cyclical irregularity as the usual electrical case.

Would the authors also state whether the slip coupling eased the situation with regard to torsional stresses as a result of the short shaft and stiffness of the secondary part.

The stroboscope actually used presented various difficulties, as the number of holes in the stroboscope was about 24 on an engine running at 300 revolutions per minute, and with a slip of 1¼ per cent. corresponded to 90 flashes per minute. Was it not possible to use some form of indicating instrument on which the slip was shown directly? Alternatively, he suggested the number of holes should be such that the number of flashes on full load full speed should not exceed 60. After all, the correct sharing of the load at lower loads was not so important as the prevention of overload on the Diesel engine.

The question of interchange of power between the slip couplings had been mentioned and it should be pointed out that the stroboscope as fitted was incapable of indicating instantaneous values. From personal observation on board he could say that the gear noise and tachometers did not indicate any interchange of power under straight conditions. The only time he noticed any indication of power interchange was during the starting period when the telegraph was moved from stop to full ahead.

The operation of the couplings on board gave an impression of elasticity. No sudden shocks were either heard or felt when the field switches of the primary parts of the couplings were opened or closed under any conditions of load. One always had the impression of a gradual change, and no sudden shocks were transmitted to the engines. In fact, if the fuel was cut off when running full speed, the engine on coming to rest oscillated quite freely on the compression stroke, and the primary parts of the coupling actually oscillated momentarily in opposite direction to the secondary part.

The coupling inherently was unaffected by sudden load changes, and tended to change relatively slowly to a new set of conditions; thus it tended to smooth out the variation in turning moment of the engine. He could only say that at sea on full speed he could not detect any variation in gear noise; a steady note existed. On a single-screw ship with two engines, no beat note was apparent in the gear noise, although the two engines were not in step, and the vibration in the engine room was similar to that of a twin-screw ship.

Dr. Jakeman had compared the slip coupling to a squirrel-cage induction motor in which the rotating field was produced by mechanical instead of electrical means. The speaker thought it well to point out that the slip coupling had not the small air gap usually associated with induction motors. Moreover, the current induced in the secondary part of the coupling was not reflected back into the primary part. The excitation of the primary part of the slip coupling was constant and unaltered at all speeds and varying slip, but, as stated by the authors, the current induced in the secondary part increased with the slip.

The coupling might also be compared with a short-circuited alternator of special construction. In the normal alternator the resistance of the stator copper was low in order to reduce the copper loss and increase the efficiency.

The reactance limited the current on short circuit, and in consequence the current induced in the stator copper on short circuit was more or less independent of speed and frequency. If, however, the resistance of the stator winding was increased to such a value as to be appreciable with reference to the reactance of the winding, then the current would vary with the frequency when on short circuit.

Broadly speaking, the slip coupling might be compared to a short-circuited alternator of the latter type.

Mr. C. Wallace Saunders (Member) pointed out that all the diagrams showed the squirrel-cage winding as connected to the Diesel engine, whereas the actual photographs showed the salient poles with their windings connected to the engine. They had heard a lot about the torsional oscillations effect. That would be transmitted to these salient poles and their windings and would be to their detriment eventually, but if the squirrel-cage part of the coupling could always be connected to the engine it would be a better mechanical job.

Mr. R. E. Ellis (Visitor) considered that the paper showed that the scope of application of the higher speed Diesel engines in marine work had been considerably widened. The utility of this application had been, from the outset, recognised by the firm with which he was associated and which, so far, was the only one as yet having made such installations. There were at present in service five vessels with this class of machinery, these being the m.s. "Werna" of 1,400 b.h.p., the m.s. "Wiros" of 1,400 b.h.p., the m.s. "Astri" of 1,700 b.h.p., the m.s. "Astrid Thorden" of 2,040 b.h.p., and the m.s. "Dagmar Salen" of 4,290 b.h.p. The machinery in all these ships was functioning to the fullest satisfaction of all concerned.

It was of particular interest to note that in the case of four of these vessels there were repeat ships now on order. There were two sister ships of the "Werna" and "Wiros" now building, whilst the same owners had also two larger vessels on order, all being equipped with Asea couplings and gear. A sister ship of the "Astri" was now nearing completion and was expected to be in service early next year.

The "Werna", which was in service on the Swedish coast, generally made about 13 knots, some passages being as high as 14 knots, depending of course on cargo, tides, weather conditions, etc.

The average consumption of fuel for all purposes worked out at 5·9 tons per 24 hours, whilst the lubricating oil consumption was 4 gallons per day. This vessel made a round trip of about 10 to 20 days' duration from Gothenburg round the south and east Swedish coast and back to Gothenburg, the length of the voyage being conditional on the number of ports visited—this varying from possibly 10 to 20, depending of course on the cargo offering. The average time the vessel was at sea was about 50 per cent.

He noted that one speaker in the discussion commented on the possibility of the air gap in the coupling being reduced considerably due to wear down of bearings. In the type of engine used so far with these couplings the amount of main bearing wear recorded over long periods did not amount to more than a few thousandths of an inch. Wear down of as much as 1mm. was unthinkable.

The point was also raised as regards load on bearings. There was a very large factor of safety allowed—in many cases a flywheel of over four times the weight could be used if necessary.

On the proposal of **Mr. J. Calderwood, M.Sc.** (Member of Council), seconded by **Mr. A. C. Hardy, B.Sc.** (Associate Member of Council) a cordial vote of thanks was heartily accorded to the authors.

By Correspondence.

Mr. A. F. Evans (Member) wrote that he considered that the authors' proposals were of greater value than the electrical transmission schemes now so much to the fore. As the torsional vibration on the propeller shaft generated by the propeller itself might be much greater than that of the engines, it would appear that such a clutch was also required between the propeller shaft and the gear wheel.

Some years ago Messrs. Blohm & Voss built some vessels with two shafts and two engines geared to each. They provided, in the first instance, a flexible coupling, but this was found to be redundant. From the writer's own observation these gears were wearing well and there was no noise from the gear case. Why was this arrangement departed from?

What was the cost and weight per h.p. of these clutches?

Would it not be better to devote more thought and expenditure to the elimination of the fundamental fault and then to insulate the propeller shaft from the gear by means of a simple spring coupling with dash-pots?

Mr. F. O. Beckett (Member) wrote that on his ship some 47 years ago the pilot, near Gravesend, put the engine room telegraph to full speed astern to avoid a collision and possible loss of life. The result was that the propeller brought up in the mud and stopped the engines with all the possible full load of steam on them. This caused the l.p. crank pin to fracture. Though they continued the voyage they had to renew the whole crankshaft on its completion.

If such an order were carried out and similar circumstances arose in the case of a vessel fitted with these couplings, would they heat up and burn out the windings and thus become inoperative, or, if not, what would happen?

Mr. G. Ridley Watson, B.Sc. (Member) wrote that there were one or two points which he thought would add to the value of the paper if further information could be given.

The electro-magnetic coupling layout was compared in each case with single-screw machinery. As a shipbuilder the writer regularly found that draught was the determining factor so far as propeller diameter was concerned. Therefore in many cases it was necessary to adopt a twin-screw arrangement and he would have no hesitation in driving such a ship with the so-called high-speed engines the authors suggested. It would be appreciated therefore if an overall figure for losses due to the electro-magnetic coupling and gearing (which must be at least 5 per cent.) could be given, together with a statement as to what savings it was considered would result from the adoption of the slower running propeller.

After carefully studying the paper and listening to it being read, the writer had got a very definite impression that the great advantage of the electro-magnetic coupling was to enable the present standard range of Polar engines to be adopted for ships where greater power was required than could ordinarily be provided through single or twin screws.

Having regard to the fact that in many twin-screw vessels revolutions from 250/400 per minute were not now uncommon, it appeared to him that to get the full advantage of the electro-magnetic couplings it was really necessary to go up to much higher engine revolutions. He stated this because the electrical equipment and the gearing were going to add considerably to the cost compared with a twin-screw installation, and unless a substantial saving in the first cost of the Diesel engines could be shown, he could not see what was going to be gained from a commercial point of view.

There was no doubt of course that where electrical control of the machinery from the bridge was desirable, the electrical couplings had many advantages usually claimed for Diesel-electric propulsion, which recently had not found favour in many instances solely owing to high first cost of the electrical equipment.

In view of the loss at the coupling and the gearing, the fuel consumption must of necessity be higher than in a direct-drive proposal, and also there must be a higher rate of consumption of lubricating oil due to the losses, and also due to the oil required in the gear case.

He thought the low head room which resulted from the adoption of the couplings, particularly in the case of a four-engine arrangement, would be of great advantage in the case of tug-boats and also train ferries or ordinary cargo vessels with machinery aft, as in the latter case it would be possible to get much greater space for officers' and crew's accommodation than was at present possible with engines requiring greater head room.

The authors' remarks regarding the alignment of the machinery had been noted, but in view of the possibility of wear down on the bearings, and also the fact that the ends of the couplings both had considerable overhang, the writer would have thought that great care was still necessary in aligning the machinery and also in watching the alignment subsequently when the vessel was in service.

It was noted that for excitation the energy required was 1 to 2 per cent. of the total energy transmitted. Had this necessitated the fitting of an additional Diesel generator on the vessels now at sea with electro-magnetic couplings? This information would be useful in assessing the total additional cost involved in the Asea couplings installation.

He presumed there would be no difficulty on a vessel fitted with Asea couplings in using the main engines for driving large suction or salvage pumps through a coupling at the forward end, as if this was practicable there would be a field for the adoption of the couplings in modern dredger craft of the suction type.

The Authors' Reply to the Discussion.

The authors fully supported Mr. Watson's views regarding the advantages of the slow-speed propeller, and considered that the advantage of being able to select the most efficient propeller and engine speeds was alone sufficient to justify the use of couplings and gears.

The question of the mechanical and electrical design of the excitation coils was one of importance and had received very careful attention. The windings were very strongly braced mechanically and were carefully insulated. The temperature rise of the windings with continuous full-load was very small, and furthermore an economy resistance was automatically inserted in the field circuit of the coupling when the engine was stopped, which reduced the excitation of the coupling. The temperature rise was thus kept at a very low level when the coupling was stationary, but the coupling was still capable of transmitting torque and consequently prevented the propeller being rotated by currents in the water surrounding it. Actually, Mr. Watson had raised this question with the authors before and had assisted them very materially by his suggestions.

It had been the authors' experience that the coupling was extremely reliable and not prone to failure. For this reason the owners of several vessels already in commission had agreed that emergency coupling bolts would be redundant. This was, however, a very small point and one which could easily be arranged if required when the coupling was built.

As far as mal-alignment of the coupling halves was concerned, it was true that there would be an out-of-balance magnetic pull, as Mr. Watson had stated. This, of course, should be taken into account when the engine and pinion bearings were designed. Nevertheless, little trouble need be anticipated as regards bearing wear for, as Mr. Ellis had stated, this would never amount to more than a few thousandths of an inch, so that the extra pull due to wear would be quite negligible.

One of the strongest points in favour of the use of multi-engined vessels was the increased reliability, and the case mentioned by Mr. Watson, when the ship was able to complete her voyage with one engine uncoupled, demonstrated vividly the possibilities of the coupling in this direction.

Mr. Wheadon stated that Fig. 2 did not give sufficient credit to the direct-driving Diesel engine. The engine taken for comparison was a Doxford type but the authors agreed that certain other makes might possibly show to better advantage in such a comparison. Nevertheless, the high-speed engine still showed a vast saving of space—particularly in head room. Further, the spares to be carried in the multi-engined ship would be smaller in size and would occupy less space. The saving in head room moreover, was a point to be stressed. It was of the utmost importance in the case of such vessels as tugs, and in vessels designed for normal cargo this space was employed with advantage for engineers' and crew's quarters, thus rendering available valuable space elsewhere for cargo. Vessels designed for special cargo, however, would always be the subject of special consideration, and generally came outside the scope of the paper.

There was at the moment under construction a twin-screw vessel, and although the authors regretted that no layout drawings were as yet available, they were assured that there was ample space—especially as regards breadth—for all the normal engine-room auxiliaries.

Mr. Wheadon also raised the question of a separate generator for excitation purposes. This amounted, at most, to 10-15 kW. for two couplings and could be easily taken from the usual ship's generator, which would normally have ample capacity. The additional fuel consumption would be quite negligible.

Mr. Wheadon put certain questions with regard to the relative consumption of fuel and lubricating oil, and also to the relative costs of high-speed geared Diesel and of the direct-Diesel drive.

It was, of course, very difficult to give comparative figures, as these varied considerably with the type of direct-drive engine chosen. But the authors would refer to Mr. Ellis' figures given in the course of the discussion, in which he gave performance and consumption figures for the geared high-speed Diesel drive on the m.s. "Werna", and it was thus possible to make a comparison with any make of direct-driving Diesel. The authors would draw attention in particular to the very low lubricating oil consumption.

With regard to the costs, there again one could not give comparative figures unless a definite equipment be named. In general, however, it might be said that the Diesel-electric drive was by far the most expensive, apart from having the highest installation charges. It was the authors' definite opinion that the geared high-speed Diesel drive with electro-magnetic slip couplings was the cheapest of all the arrangements, and in particular compared more than favourably with the direct-driving Diesel installation.

The authors appreciated Mr. Johnston's remarks as to the term "slip" coupling. It was necessary, however, to distinguish between this form of *slip* coupling (with a continuous relative movement or "slip" between the two parts) and the ordinary flexible kind, where there was only a *displacement* between the two halves, but no continuous movement or slip. In the couplings described in the paper the amount of slip was a fixed function of the torque and could not be controlled externally, but there was a design where the slip could be controlled by means of inserted rotor resistance, as in the slip-ring induction motor. Perhaps, therefore, the term "slip" coupling best described the apparatus and distinguished it from other forms of coupling.

It was true that Figs. 1 and 2 showed engines with speeds of 275 and 260 r.p.m. respectively. There was no reason why higher speeds should not be used. Several couplings had been manufactured for 300 r.p.m. and designs made for speeds up to 600 r.p.m.; there was no reason why the limit should be placed there. In general, the development of higher speeds must rest with the engine designers.

It was also true that the coupling halves were overhung, and the authors considered that to spigot them would complicate what was now a very simple arrangement. Furthermore, the property of the couplings, whereby a certain amount of misalignment would be allowed, would thereby be lost.

With regard to Fig. 9 it was true that the vector 0-4 should be greater than 0-3, and the authors regretted that it was not so shown in the original figure.

It was stated in the paper that the torque exerted by the coupling was independent of the speed, and, in fact, full torque or indeed any torque from zero to maximum (or "pull-out") torque could be exerted at standstill. Mr. Johnston was therefore not quite correct in saying that the coupling exerted its maximum tractive effort at low speeds. He was thinking, no doubt, of the characteristics of the d.c. series traction motor. The torque exerted by the coupling was quite independent of speed, and depended only upon the slip. The 5 per cent. slip mentioned would correspond to about a 200 per cent. torque (see Fig. 10), which could not be maintained by the Diesel engine without stalling it. Therefore, the question of the coupling maintaining a 5 per cent. slip continuously, or even for a short time, could not arise.

The economy resistance shown in Fig. 12A was inserted by the regulator via a contactor whenever the regulator was in the "STOP" position. This maintained a reduced field through the primary windings, which would prevent self-rotation of the propeller through currents in the water and enable the coupling to transmit torque even with the engine at standstill; the current would not be sufficient to overheat the windings, although there would be no ventilation action with the coupling in a stationary state.

Mr. Johnston's suggestion that a generator be placed on each engine and that these should be coupled solidly to their own motors amounted in effect to a Diesel-electric drive without the flexibility of that arrangement. Apart from other considerations, such an equipment would be considerably more expensive than, and would have no advantages over, the couplings and gear.

The authors realised the great advantage of warming-up and turning-over the engines when alongside, and thanked Mr. Johnston for raising this point.

The authors regretted that it would not be possible to give overall mechanical efficiencies, as Mr. Basebe requested. The increased bearing loss depended almost completely upon the main bearing design and lubrication, and this in turn was a question for the engine designers and came rather outside the scope of the paper. As far as the remaining items were concerned, however, the efficiency of a well-designed precision gear should be between 98·8 per cent. and 99·0 per cent., and the losses in the coupling due to slip (about 1½ per cent. at full load) would make the full-load efficiency of the coupling about 98·5 per cent. The excitation losses (about 1½ per cent.) did not strictly affect the mechanical efficiency, as they would be supplied from the separate ship's generator and not from the main engines. Windage losses were exceptionally low as there was hardly any relative movement between the two coupling halves, and consequently there were practically no eddies.

Mr. Basebe had raised two interesting points in connection with the possible stalling of the main engines under certain conditions. At the outset, however, the authors must repeat that the function of the coupling was not to prevent the engine from stalling, but one of its functions was to prevent damage to the main engines and gearing due to severe overload. It was quite correct to say that if one engine failed its speed, as well as that of the propeller, would be simultaneously brought down until a balance was reached at a lower speed, due to the fact that the coupling would not, in general, pull out or stall under those conditions. The overload on the engine would pull it down just as if no coupling were there, but the coupling of the faulty engine could be at once de-energised, and its dead load taken from the sound engine. The faulty unit was then quite free for immediate overhaul. Such a temporary and relatively gentle overload on the main engine would do it no harm whatsoever, and its speed would thereby be reduced until a balance was reached.

If the propeller became gradually choked with ice, but not sufficient to make the coupling pull out or stall, the speed of the engines would be progressively brought down until they stalled. *But at no time would the overload on them have exceeded about 200 per cent. torque* and they would have suffered no harm; neither would the gears have been damaged. But if the propeller had struck a hard object, such as large ice or driftwood, or perhaps have become stuck in the mud on the bottom, as one contributor had described, the sudden stop would normally result in very severe and dangerous overloads on the gear and engines due to the inertia of the moving masses. This overload would far exceed 200 per cent. torque and the coupling, as soon as the torque had risen above 200 per cent., would immediately stall and relieve the engines and gears of all load except about 20 per cent. torque. In practice, the sudden stalling point would be nearer 300 per cent., due to the magnetic induction of the coupling (see page 242, col. 2, line 20). In fact, instead of being jerked to a standstill with consequent severe danger of damage, the engines would be relieved of load and, in consequence of the full fuel supply, would start to race until the overspeed device had closed the fuel valve.

In general, it should be realised that the coupling would protect the engine from damage due to severe overload, and would allow the engines to remain coupled during slight overloads, so that they might have a chance to overcome obstacles if these were within their power.

The authors were not quite clear as to the exact meaning of the first point raised by Mr. Kent. The couplings were normally delivered with the primary (inner) part arranged to be pressed on to the pinion shaft of the gear casing, and with the secondary (outer) part fixed to a short shaft, which was provided with a flange for bolting to the corresponding flange on the Diesel engine crankshaft. So far as the authors could see, there was therefore no need for any adjustment of the bearings.

The couplings could be used with any make of gear, although it was naturally preferred to manufacture gear and couplings together, so that they might be assembled and tested as a complete unit.

In the four-engine arrangements so far designed (such as that shown in Fig. 17) the four pinions were arranged on two shafts, each pair of pinions being *solid* with their shaft and not separate. This was necessary, for it would be seen on reference to Fig. 17 that in the case of each pair of pinions one was cut with a positive "skew" and one with a negative. When a torque was applied to the pinion shaft through either or both couplings, the end thrusts due to the opposite skew gearing cancelled out. But if one engine were shut down, say for overhaul, and the four pinions were separate, the one pinion transmitting the drive from the sound engine would produce an unbalanced end thrust. Further, as all the four pinions were fixed relative to each other through the gear-wheel teeth, there would be no point in having them separate on their shafts. Any engine could be uncoupled simply by de-energising its coupling.

The authors fully agreed with Mr. Kent when he stressed the simplicity and sound construction of the coupling. As he correctly stated, there could be no wear whatsoever between the two halves, and the upkeep costs were negligible.

The Asea coupling did not offer an alternative method of manœuvring a vessel, but merely extended the manœuvring range well into the lower speeds. Indeed, the authors had repeatedly stressed the fact that the control of the ship was carried out on the Diesel engines as usual, but at low speeds, where the engine was near its limit of stable running, it was possible to continue the control by coupling and uncoupling the propeller as required. No additional engineer would be needed as the control cubicle would be placed near the engine regulators and all the engineer would have to do would be to close or open a switch. Furthermore, starting air could be economised due to the fact that continuous stopping and starting of the engines would not be necessary, and the speed with which the vessel could be put ahead or astern by running one engine in each direction and coupling in either one or the other, could not be too much stressed. It should be realized, too, that for slow running, when both engines were running fairly light, it would often be more economical to shut one engine down completely and to run on one only. Due to its increased load, the combustion and efficiency would be better, and a lower stable speed would be possible.

In the authors' opinion there would be little likelihood of the coupling needing overhaul owing to the fact that it was essentially simple and robust. The coupling had really no parts, the windings being placed on the outside, and being thus readily inspected.

The diameter of the coupling would not be greater than that of the normal flywheel and should not offer any additional difficulties in accommodating. It would not be desirable to increase the length at the cost of diameter, as such an arrangement would be heavier for a given speed and power, and the distance between its two bearings would also be increased. This would raise the moments of the overhung weights and should be avoided where possible. In addition, the flywheel effect of the outer (secondary) half would be greatly diminished.

As far as torsional vibrations were concerned, the authors would point out that the couplings had been installed in no less than fourteen ships, many of which were now at sea. and the results of these installations showed that the couplings were equal to all the claims that had been made for them. There was no question, therefore, of them still being in an experimental stage, and it was rather interesting to note that the data obtained from the installations that had been made fully confirmed all the operating characteristics that had been predetermined in design.

When a vessel pitched in a high sea and the propeller came out of the water the shafting would be relieved of load, as would also the couplings and engines. The engines therefore would gather speed just as if the couplings were not there, until the overspeed governor checked the fuel supply. The coupling slip would drop from about 1½ per cent. to zero, but this would not be noticeable. However, the shocks caused by the propeller blades reentering the water would not be transmitted to the engine shaft and the propeller shafting would thus be greatly relieved of shock stresses.

Consideration was always being given to further development and uses for the coupling and, in this connection, it might be of interest to mention a scheme for a rotary dredger. This vessel had two Diesel engines, coupled through slip couplings at the after end and reduction gearing to a single screw, according to the normal arrangement. Two further couplings, however, were placed at the forward end of the engines, and to these the dredger pumps or propeller, or indeed both, could be coupled and uncoupled merely at the touch of a switch.

At the moment, the smallest couplings designed were for 210 b.h.p. at 600 r.p.m. and the largest 1,500 b.h.p. at 300 r.p.m. There was no reason, however, why smaller or still larger units could not be designed if required.

The authors were grateful to Dr. Jakeman for his very lucid analogy between the operation of the slip couplings and the squirrel-cage induction motor. They agreed that the expression "to stall" would be better than "to fall out of step", and would endeavour to overcome the rather conservative use of the existing phraseology.

Commander Hall raised the question of the bulkiness of the couplings, but in the authors' opinion it is not more so than any other similar form of coupling, especially when it was recalled that no other auxiliary plant was required, for the excitation was taken from the ship's d.c. supply. It must also be borne in mind that the use of high-speed Diesel engines more than compensated for this space, as shown in Figs. 1 and 2, not to mention the overall saving in weight.

As far as the actual sizes mentioned by Commander Hall were concerned, a coupling transmitting 1,050 b.h.p. at 300 r.p.m. would have a diameter of about 6ft., whereas a similar power at 500 r.p.m. would require only about 5ft. 4in. There would be little difference in the length. A coupling transmitting 2,000 b.h.p. at 400 r.p.m. would have a diameter of about 6ft. and a length of about 2ft. 3in. These figures would be approximate only, as it would be necessary to design the couplings in detail to arrive at definite figures. As regards the other size mentioned, namely 1,200 b.h.p. at 1,000 r.p.m., this was rather more difficult as such a high-speed coupling had not, as yet, been designed. It might be expected, however, that the diameter would be in the neighbourhood of 4ft. or somewhat less, with a length around 2ft. 6in. These figures were given, however, with great reserve.

As regards the limits of power which the coupling could transmit, it should be clearly borne in mind that the torque was proportional to the product of the *secondary* (not primary) current and the flux, which was itself a function of the primary current. The secondary current was proportional to the product of flux and slip. Therefore, the torque could be said to be a function of the $(\text{flux})^2$ and slip. It was true that for any given coupling and excitation plant there was a limit to the maximum flux and to the speed of the rotor (or primary winding) and so a limit to the horsepower which that coupling could transmit. That was the "rating" of the coupling. For higher horsepowers or lower speeds a different set of quantities must be chosen and another size of coupling with a higher rating would be arrived at. There was no theoretical reason why there should be any limit to the power which could be transmitted, it being remembered that the designer could vary the diameter, excitation or length at will.

The temperature rise had, of course, been taken into account but had been quite successfully dealt with. The coupling could transmit full torque down to half speed continuously without any undue temperature rise, although in practice at these lower speeds the continuous (as distinct from the intermittent) torque was seldom at its full rated value.

The coupling could, of course, be designed to pull out or stall normally at any overload desired up to the limit of about 200 per cent. torque, although it would not be necessary to make special designs. If reference be made to Fig. 11 it would be seen that, by reducing the excitation current to about half normal, the coupling could be made to stall at just under full load. Intermediate values of excitation current would, of course, give intermediate stalling points.

The stroboscopic method of measuring the slip was shown schematically only in Fig. 21 for the sake of clearness. In practice, however, the discs shown were replaced by two concentric cylindrical sheets, the inner one being fixed to the primary (inner) coupling half and the other, which was concentric with it, was fixed to the secondary (outer) part. The lamp was placed inside the inner ring and the two cylindrical rings rotated relatively, the one inside the other, much like a sleeve valve, but with continuous rotation. Fig. 14 showed the device installed complete with the small lamp in place. (The lamp could be seen by following the connecting flex lying across the gear casing).

Mr. Jaffrey asked for particulars concerning the weights of the secondary parts of the slip couplings and the corresponding usual flywheels under different conditions of drive. It was rather difficult to answer this point generally, as it would be readily appreciated that the design of each equipment took into consideration the peculiar requirements of the case under consideration. In general, however, it would be reasonable to say that where a Diesel engine was driving a generator, little or no additional flywheel capacity was required owing to the high moment of inertia of the generator rotor itself. A small flywheel was generally used with less than six cylinders, however, and sometimes with more than six, although it served more for barring the engine round than for real flywheel capacity. With the engine as prime mover for other cases, however, the coupling secondary half would be somewhat heavier than the corresponding flywheel but clearly it would not be possible to give any definite figures. The authors stressed that each design would be considered on its merits and it was possible that there might be, in special cases, some variation from the above general statements.

As far as torsional stresses were concerned, the short shaft and stiffness of the secondary coupling half merely replaced the usual flywheel. The system comprising engine, crankshaft extension and coupling secondary half was the only part that need be considered from a torsional oscillation point of view. The torsional stresses in that system would depend on the engine designers.

The actual design of the stroboscopic device was a small matter and could easily be arranged to suit the requirements of the individual as regards number of holes and so on. It would, of course, be quite possible to arrange some indicating instrument, perhaps of a differential pattern, but the stroboscope described had been chosen as standard owing to its simplicity and its positive manner of measuring slip, not to mention its greater cheapness. There was no necessity, however, to use any one particular system.

Mr. Jaffrey had stated that the current in the secondary windings was not, as in the induction motor, reflected back into the primary part. That was true in a sense, although not quite in effect. Actually, the secondary current did tend to demagnetise the primary magnets and, although the primary d.c. excitation *current* remained constant throughout, the actual *induction* became gradually reduced. Up to full load, the reduction was only about 2 per cent. (see Fig. 9), but as the overload increased, the demagnetisation became progressively greater until, at about 200 per cent. torque, it finally caused the coupling to pull out or stall.

The comparison between the slip coupling and the short-circuit alternator with a higher internal resistance was an interesting one and the authors were grateful to Mr. Jaffrey for mentioning it.

Mr. Jaffrey recorded his impressions of a voyage on a vessel equipped with Asea slip couplings and reduction gears and the authors noted with satisfaction his remarks with regard to vibration and gear noise. This bore out in full the first claim for the coupling, namely, that it obviated all shocks and vibration from the engines and shafts system.

The authors feared that Mr. Saunders had not quite correctly understood the photographs. All the photographs (Figs. 7, 15, 17, 18) showed the magnet wheels with their salient poles connected to the *reduction gear,* not to the Diesel engine. This was perhaps not so clear from Fig. 14, but there also it was actually so. In this case, the end cover fixed to the outer (squirrel-cage) part might seem to make it appear fixed to the pinion shaft. Actually, however, it was possible to see a small gap between them, through which cooling air was drawn by means of the fan blades on the outer coupling half, fixed at its forward end. Thus the large moment of inertia of the heavy inner magnet wheel was applied to the pinion shaft and not to the Diesel engine—as Mr. Saunders suggested.

It should be noted that in the diagrams in Figs. 4 and 8, the secondary part was shown as the inner portion for the sake of simplicity of drawing, whereas in practice the secondary part was outermost. Nevertheless, in all cases the secondary part was fixed to the Diesel engine and the heavy primary magnet wheel to the pinion shaft.

The authors were grateful to Mr. Ellis for the figures which he gave concerning fuel and lubricating oil consumption. He also made comments with regard to the wear and load of the bearings and the authors were fully in agreement with them. These remarks answered certain questions raised by the speakers and in the communicated contributions. There was therefore no need to enlarge upon them here, and reference would be made to Mr. Ellis' comments where the need arose.

The authors were not aware that in general the torsional vibrations from the propeller itself were much greater than those of the engines. Further, in the case of the propeller shaft there was the very large inertia of the main gearwheel which absorbed a considerable amount of the oscillations coming up the propeller shaft. But if, in any particular case, such oscillations were found to be serious, there was no reason why a coupling should not be placed aft of the gears. This coupling would, however, be very large and somewhat expensive.

Mr. Evans gave no details as to the vessels built by Messrs. Blohm & Voss, so that it was difficult to comment on them without some knowledge of the engine and operating conditions

Engine designers were doing everything possible to eliminate the fundamental cause of torsional oscillations, but these had not been as yet completely removed, and the authors' Company had recognised this fact in producing its electro-magnetic slip couplings. The "simple spring coupling with dash-pots" mentioned by Mr. Evans would appear to be a more complicated form of quill drive used in electric locomotives, and the authors feared that such an arrangement would be anything but simple, and certainly not cheap.

It should be realised that the ordinary flexible coupling gave no protection against severe overload and could not be coupled and uncoupled at will. It had been found necessary with modern so-called high-speed Diesel engines to provide a flexible buffer between the engines and gears and in the authors' opinion the electro-magnetic slip coupling offered the simplest and most practical solution to the problem. Its many other advantages, apart from the absorption of high-frequency oscillations, rendered it even more suited to modern requirements.

It would not be possible in this paper to give details of prices of the couplings and, of course, the weight was not a function of the horsepower only. Speed was just as important and the couplings for the slowest speeds were, in general, the largest in size for any given power. One might say that the weights varied from about 1½ tons for a 200 b.h.p. coupling at 600 r.p.m. to about 7½ tons for 1,500 b.h.p. at 300 r.p.m. Intermediate powers and speeds could be reckoned very roughly proportional, although naturally use would be made of standard frames where these existed.

Mr. Beckett gave a most interesting description of an accident to a steamship many years ago, when the propeller brought up suddenly in the mud with a full load of steam on the engine. If such an order had been signalled on a vessel equipped with Diesel engines, slip couplings and reduction gears, and the propeller had become firm in the mud, the load on the engines would rise to about 200 per cent. (possibly as high as 300 per cent., but not more) momentarily and would then fall to about 20 per cent. This overload would be very light as far as the mechanical parts were concerned and there would be no danger of fracture such as in the case narrated by Mr. Beckett. The engines, relieved of load, would rise in speed until the over-speed device shut down the fuel supply. The heating of the couplings at this load would be very slow and, in any case, the engineer would soon have stopped the engines, which automatically inserted an economy resistance into the coupling field windings. He might even uncouple the engines by opening the coupling switch, but this would not be necessary. There would certainly be no danger of the couplings overheating and they would completely protect the engines and gears from damage.

The authors appreciated Mr. Watson's remarks and the confidence he expressed in the electro-magnetic couplings for use in a twin-screw vessel. It might be of interest to record that, at the moment, there was a twin-screw four-engined vessel building in Malmö, with a total shaft output of 5,000 s.h.p. at 130 r.p.m., the engine speed being 300 r.p.m.

In the authors' reply to Mr. Basebe the overall efficiencies had been given in some detail, and perhaps Mr. Watson would refer to these. There was no reason why a slower-running propeller should not be adopted from the coupling point of view, and the efficiencies of the coupling would not be affected. The gearing might of course, be a little larger, but that would be a question for the designer to settle.

It had been stated elsewhere that there was no reason why the couplings should not be used for higher speeds, and indeed designs had in many cases been worked out. The authors were of the opinion that there was a saving in first cost by using couplings and gearing and a great increase of reliability, together with greater degree of control and possibility of more economic running.

One or two propositions had been put forward where emergency bridge control had been incorporated. Opinions differed as to the desirability of any form of bridge control, but in the cases mentioned the vessel was to be designed for service around Vancouver where there was a danger from floating logs and drift-ice. When an obstruction was seen, it was arranged that the couplings could be tripped instantly from the bridge by means of a simple push-button. Re-closing of the couplings was effected through the normal engine-room telegraph in the usual way. The additional cost of this feature was quite negligible and the only extra equipment required would be a push-button and two contactors.

Mr. Ellis had given in the discussion figures for fuel and lubricating oil consumption to which Mr. Watson would perhaps refer. It should also be borne in mind that very economic running might be obtained when cruising at slow speed or making voyages in ballast.

The authors agreed that the additional head room was a great advantage, especially in tugs, train-ferries and certain cargo-vessels. Officers' and crew's accommodation was a point which was sometimes rather overlooked.

With regard to bearing wear and alignment of the coupling halves, the authors would draw attention to the remarks by other speakers and to the replies thereto, and in particular to Mr. Ellis' remarks in the discussion. Wear down of the bearings was rather over-estimated and no trouble whatever had been experienced in this direction with the vessels already in service.

It was stated earlier on (in the reply to Mr. Wheadon) that the excitation for the couplings would normally be taken from the ship's supply generator which would in general have ample capacity. As far as the vessels now in service were concerned, the authors were not aware that any additional generators had been fitted for excitation purposes.

Mr. Watson also suggested the use of the main engines for dredger use, and the authors would draw attention to their reply to Mr. Kent in which they described briefly a scheme already proposed. In their opinion, there was a large field for development in the direction of modern dredger craft of the suction type.

In conclusion, the authors wished to thank all the speakers for the very interesting points they had raised and for the assistance that they had given in bringing forward new aspects of the whole problem of marine couplings.

The Annual Conversazione.

Another memorable achievement has to be credited to the Convener, Mr. Alfred Robertson, and the Social Events Committee in that all previous records were eclipsed on the occasion of the Annual Conversazione. This ever attractive function was held at Grosvenor House on Friday, 26th November, 1937, and was attended by 1,032 members and guests. Even this number—185 more than last year's attendance—would have been exceeded had the capacity of the Great Hall at Grosvenor House permitted the issue of a further 50 tickets to a number of late applicants, whom the Committee regret having had to disappoint.

The President, Mr. Stephen J. Pigott, and Mrs. Pigott received the members and guests, the company including the following distinguished guests of the President:—Sir Percy E. Bates, Bart., G.B.E., Chairman of Cunard White Star, Ltd., Sir Thomas Bell, K.B.E., Director of Messrs. John Brown & Company, Ltd., A. J. Grant, Esq., Managing Director of Messrs. Thomas Firth and John Brown, Ltd., and Mrs. Grant, Eng. Vice-Admiral G. Preece, C.B., Engineer-in-Chief of the Fleet, and Mrs. Preece, H. J. F. Mills, Esq., Manager, New Zealand Shipping Company, Ltd., and Mrs. Mills, J. Austin, Esq., Superintendent Engineer of Cunard White Star, Ltd., and Mrs. Austin, S. T. Pigott, Esq., and Miss Maureen Pigott. Dinner was served immediately after the reception, and was followed by half an hour of variety entertainment. The artistes included Deveen, "The Distinguished Deceiver" and his New York Blondes, The Bega Four, "The Laughing Dare-devils of Dance", and The Viennese Singing Sisters.

Dancing ensued till midnight, being followed by a "Cavalcade of Cabaret", under the direction of Gordon Marsh, comprising his famous young ladies, also Dawnya and Petrov in "the Act Supreme—Youth: Beauty: Personality"; The Four Synnecks in "Athletics Extraordinary", and The Andos Family, "The Famous Japanese Acrobats and Entertainers". Musical direction throughout was by Donald Hatton. The Silver Masked Caricaturist was a centre of attraction during the reception.

Sydney Jerome and his Dance Orchestra rendered a delightful programme of music throughout the dinner and dancing. The proceedings ended at 2 a.m. with a grand rendering of "Auld Lang Syne", followed by rousing cheers for the President and Mrs. Pigott.

During the dinner the toast of "The President and Mrs. Pigott" was submitted by Mr. R. Rainie, M.C., Chairman of Council, in the following terms:

"Mr. President, Ladies and Gentlemen,

The Chairman of Council of this Institute has many duties and very few privileges. I, however, have two privileges to-night.

The President has given me his permission to make an appeal, which has his approval and support, to the members of our Institute present, to help me. That is one of my privileges. I would ask those of you who are guests to bear with me for a minute or two while I speak to my brother members of The Institute; my reason for asking your indulgence is that this is the last occasion this year when such an opportunity occurs.

Gentlemen of The Institute, on the 7th of May of this year I addressed some 3,800 letters to individual members of The Institute. I asked them to help to make Coronation Year a memorable one by joining our Guild of Benevolence, whose work in helping members of our profession who have fallen on evil days through no fault of their own is known to you. The result of that appeal to date is 54 new members. That makes Coronation Year memorable, but not in the way I and your Council anticipated. I want your help. Coronation Year has still some four weeks to run. Your help can yet make it a truly memorable year. The Secretary's address in the Minories is known to you; the Secretary's table to-night is No. 6 and he is ready to take your names down now.

THE PRESIDENT replying to Mr. Rainie's Toast. Mrs. Pigott is on his left, and Sir Percy Bates on his right.

MR. R. RAINIE (Chairman of Council) proposing the Toast of The President and Mrs. Pigott.

THE PRESIDENT'S TABLE.

THE ANNUAL CONVERSAZIONE.

THE ANNUAL CONVERSAZIONE.

To those of you who are not in the Guild I would say "Trust your instinctive goodness of heart and come in". To those of you who are in the Guild I would say "Go out into the highways and byways and compel others to come in". I want to put this to you—if the success of this party to-night depended upon your spending another 10s. 6d. would you hesitate? I leave that thought with you. Gentlemen, we are here in Grovenor House. All I want from our members is 10s. 6d. a year to help those of our colleagues who never can come here. Will you help?

Now, ladies and gentlemen, I am going to exercise the privilege for which I do not need to ask our President's permission; I am going to offer you the toast of his health. Members of The Institute do not need to be told about Mr. Pigott and his work, and for those others—our guests—who are with us to-night, it should suffice that we think so much of him that we elected him to the Presidential Chair, and so I would ask you to rise and drink to the health, prosperity and happiness of our President and his lady".

The toast was enthusiastically accorded with musical honours and loud and prolonged applause.

The President, in replying, said:

"Mr. Rainie, Ladies and Gentlemen,

The Institute of Marine Engineers, in choosing me for their President for the current year, has indeed conferred upon me a very great honour. Also, I feel great pride in having qualified some years ago for admission to this Institute whose members cover such a wide field of action. It would seem that marine engineering employs in some degree every phase of the engineering profession, from men associated with scientific research to those who actually go down to the sea in ships. To Mr. Rainie I extend my fullest thanks for his proposing a toast of health, happiness and prosperity to my wife and myself, and to you, ladies and gentlemen, for so heartily responding to this toast". (Renewed applause).

Members who, until reading the above report of Mr. Rainie's appeal, have withheld their support from the Guild are urged to come in without further delay as subscribing members. Those whose applications are received before the date of the next meeting of the Executive Committee, January 5th, will be counted as having enrolled in Coronation Year, except as regards the renewal date of their subscriptions, which will be entered as January 1st, 1938.

INSTITUTE NOTES.

ELECTION OF MEMBERS.

List of those elected at Council Meeting held on Monday, 6th December, 1937.

Members.

George Alan Broadbridge, 39, Byron Mansions, Corbets Tey Road, Upminster, Essex.

William Edward Bruce, 8, Derwent Drive, Wallasey, Ches.

George Crawford, Port and Marine Office, Dar-es-Salaam.

John Osmond Middleton Fisher, 279, Christchurch Road, Newport, Mon.

John Knight Gibbon, Woodbank, Moneyclose Lane, Heysham, Lancs.

Robert Hay, Standard Vacuum Oil Co., Hankow, China.

Alfred Edgar Jones, Brunlea, 2B, Thorburn Road, New Ferry, Ches.

Griffith Fitzhowell Jones, Llansay, Velmore Road, Chandlers Ford, Hampshire.

Charles Gray Lowery, 3, Handsworth Dr., Great Barr, Birmingham.

John Edmund Calder Macdonald, 19, Niton Street, Fulham, S.W.6.

George Valentine Norledge, 14, Park Road, Meols, Wirral, Cheshire.

James Arthur Porteous, Malayan Collieries Ltd., Batu Arang, Selangor, F.M.S.

Isaac Leonard Pullen, 41, Newmarket Street, Georgetown, Demerara, B.G.

James Chang Ling Wong, 14, Stafford Road, Kowloon Tong, Hong Kong.

Associates.

Arthur William Adams, 203, Abbott Road, Poplar, E.14.

Patrick Dalrymple Davidson, 47, Hosack Road, Balham.

Reginald Bruce Godfree, 19, Paget Street, Gillingham, Kent.

Nicholas Hall, 164, Strathnairn Street, Cardiff.

Robert Joseph Reed Hancock, 24, Aberfeldy Street, Poplar, E.14.

Stanley Thomas Ogilvie, Glen Wood, 19, Hodder Drive, Perivale Park, Greenford.

David Wood Paterson, 85, Duff Street, Macduff, Banffshire.

James Stretton Pryde, 38, Francis Street, Stornoway, Isle of Lewis.

John George Robinson, Rosedale, Holmside Avenue, Dunston, Gateshead, 11.

Student.

Ralph Albert Collacott, 235, City Way, Rochester, Kent.

Transfer from Associate to Member.

Arthur Stanley East, 35, Westcombe Park Road, Blackheath, S.E.3.

Transfer from Student to Associate.

Henry Noel Buddle, 36, Marlborough Road, Falmouth.

John Vernon Vincent, 7, Marine Crescent, Falmouth.

ADDITIONS TO THE LIBRARY.

Purchased.

"Very Low Temperatures—Book Three", by T. C. Crawhall, M.Sc., H.M. Stationery Office, 1s. 3d. net.

Contributory Pensions—Mercantile Marine Order, 1937. H.M. Stationery Office, 1d. net.

Merchant Shipping (Superannuation Contributions) Bill. H.M. Stationery Office, 1d. net.

Treatment of Chimney Gases at the New Battersea Power Station of the London Power Co., Ltd. (Further Report of the Committee Presided over by the Government Chemist). H.M. Stationery Office, 1d. net.

Presented by the Publishers.

The following British Standard Specification:

No. 754, 1937. System for the Direction of Rotation of Machine Tool Handwheels and Levers relative to Movement Produced.

Technical Report for 1936 of the British Engine Boiler and Electrical Insurance Co., Ltd. 7s. 6d. net.

Bulletin de L'Association Technique Maritime et Aéronautique, No. 41, Session 1937.

Proceedings for 1935 and 1936 of the Victorian Institute of Engineers, containing the following papers:—

- "An Insight into Modern Methods of Manufacture of Glass", by Simcock.
- "Prevention and Mitigation of Flood Damage", by Anderson.
- "Bagasse Fired Boilers", by Gamble.
- "The Story Behind the Notice to Airmen", by Pyke.
- "The Uses of Glass in Modern Building Practice", by Simcock.
- "Glycerine", by Naismith.
- "Transport Compression-ignition Engines", by Street.
- "The Engineer in Modern Architecture", by Michaelson.

"Gearing for Cranes", by E. G. Fiegehen. Emmott & Co., Ltd., Manchester, 20pp., illus., 1s. net.

This monograph has been prepared as an aid to the selection of machine-cut spur reduction gearing for cranes to comply with British Standard Specification No. 436 and to resist satisfactorily the starting overloads of which modern crane motors are capable.

It will be found that the classification of pinions upon a "permissible torque" basis avoids the usual preliminary calculation of pitch-line loads and speeds and of zone factors. Examples are given to indicate the use of the tables. A chart is included indicating the approximate inertia characteristics (Wr^2) of cast steel wheels as used in crane construction, and an example is given of its use.

"Arc Welding Handbook", by Karl Meller. Hutchinson's Scientific and Technical Publications, 210pp., 85 illus., 8s. 6d. net.

The historical portion of this book is commendably brief, dealing only with salient facts. Passing on to present-day problems the author covers an immense amount of ground in some 200 pages of condensed matter. It would take some time and study to confirm many of the statements made and on some of these there is definitely a difference of opinion, but the writer courageously states his views and in most cases gives his reasons. The book is rather in the nature of an epitome and must be read with understanding, but in these days of specialization it is quite useful for the worker in any particular section to have a volume which at any rate indicates the salient features in the many forms of arc welding. The writer advocates the use of bare wire electrodes on account of cheapness except where alternating stresses are present, but his recommendation must be accepted with the greatest care as alternating stresses and suddenly applied stress are often difficult to foresee and estimate. There are, as said before, matters upon which experts differ, such as the protective glass employed. The modern view is that the composition and thickness of the glass as cutting off the harmful infra-red and ultra-violet radiations are of more importance than its colour. That is to say, a glass of greenish neutral tint may give much better protection than a densely coloured glass. Again many people consider that cast-iron aluminium and copper are much better welded with oxy-acetylene than with the electric arc owing to the lower temperature of the former, but perhaps it is hardly fair to expect a reference to this in an arc welding handbook. The book is well written and put together, and contains much of value to the welder. It has been admirably translated by Mr. Ginger.

"Brown's Flags of All Nations". Brown, Son & Ferguson, Ltd., 52-58, Darnley Street, Glasgow, S.1, 1s. 6d. net, postage 2d.

There have been so many changes in the world's flags within the last few years that another edition giving the alterations in their original colours and in a form that may place them at a glance is welcome. An eminent cartographer has put his best work into this production, which should have a very wide appeal.

For use at sea or in the home, the edition published at 1s. 6d. is recommended; this is folded to booklet form 9½in. by 6½in. The sheet form, 36in. by 23in., cloth mounted, varnished and on rollers for hanging, is recommended for offices, schools, libraries, etc. The price of this is 10s. 6d. plus 6d. postage (postage 2s. abroad).

Th flags of the British Admiralty, government and public departments, and special British section amount to 70 in number, while the section relating to India, the Dominions, Colonies, Protectorates and Mandated Territories, contains 114 flags. The third section deals with foreign flags and ensigns, which number 132. In section four the pilot flags of all nations amount to 34 in number. A further section includes the new International Code of Signals, while another shows 76 flags flown by the leading yacht clubs in Britain and abroad.

"Hydraulics—A Text on Practical Fluid Mechanics", by R. L. Daugherty, A.B., M.E. McGraw-Hill Publishing Co., Ltd., 4th edn., 460pp., 318 illus., 21s. net.

This is the fourth edition of a book first published twenty-one years ago. The subject matter has been brought into line with the gradual development of fluid mechanics and the presentation generalised so as to apply to all fluids whether liquids or gases.

The book starts in the usual way with chapters on properties of fluids, intensity of pressure, and hydrostatic pressure on areas. The chapters on the kinematics of fluid flow and dynamics of fluid flow are well written, the development of the general energy equation for steady flow of any fluid being very explicit.

Stream line and turbulent flow are treated in the chapter dealing with friction in pipes and the practical use of Reynolds' critical number is indicated. In the opinion of the reviewer the important section on dynamical similarity could have been treated at greater length and further illustrated by examples of dynamically equivalent systems, to show the universal application of the principle.

The chapters dealing with flow through pipes and open channels treat the subject matter very thoroughly, while the chapter on applications of hydrokinetics is excellently planned and set out. The section of the book devoted to hydraulic machinery—turbines, pumps etc.—follows along the usual lines and some excellent diagrams, graphs and well selected photographs are given. The book is not without blemish, however, for it is unfortunate that the author has not included a fair number of worked examples. This seems to be a common fault with many American textbooks. At the end of each chapter, however, problems are given along with answers. For the degree student, for the teacher, and for the practical designer, this book can be recommended.

"Low Temperature Physics", by M. and B. Ruhemann (of the Physico-Technical Institute, Kharkhov). Cambridge University Press, 313pp., illus., 18s. net.

In this book of over 300 pages and 130 diagrams, the authors have described the various methods and experimental apparatus built and used by physicists in their investigations and endeavours to reach the absolute zero of temperature.

The book is divided into four parts, the first of which deals principally with the liquefaction of oxygen, nitrogen, and the rare gases from the atmosphere, and their purification and rectification, with a brief description of the different types of liquefiers used commercially and their relative efficiencies. Section four of this part is devoted to the means of measuring the extremely low temperatures produced.

Part two deals with the determination of crystal forms which are stable at low temperatures, together with their thermal energy and their specific heat. Nernst's Third Law on the inaccessibility of absolute zero, suggested in 1905, is dealt with at some length in chapter three of this section.

Part three deals with degenerate states, ortho and para hydrogen, and the behaviour of gases at low temperature in a magnetic field; also magnetic cooling and the problems connected therewith.

Conductivity at low temperatures, electrical resistance and supra-conductivity are dealt with in part four.

A very comprehensive bibliography of nearly four hundred references is given, down to the end of 1935 when the book was written, and a further list of about 90 references has been added, bringing it to May, 1937.

To students and physicists who are interested in the development and application of extremely low temperatures the book can be strongly recommended.

"Design and Cost Estimating of all Types of Merchant and Passenger Ships", by A. Kari, M.Sc., The Technical Press, Ltd., 4th edn., 396pp., illus., 42s. net.

The latest edition of Mr. Kari's work is a great improvement on his earlier efforts, but the book is still one that should prove more valuable to those already engaged in ship design than to the student. To write a treatise such as this on the basis of personal experience demands some courage, and naturally there is a tendency to over-elaborate special subjects of which the author has most experience. For example, too much stress is laid on dredgers, ice-breakers, and other special craft, but on the whole the book has better balance than the original edition.

The ground covered is very extensive, and in places the subjects are treated exceptionally well. Allowing for the fact that a great deal of the data has been culled from particular ships, there is much in this book to stimulate thought. The remarks on various types of ships' forms are well written, and the chapters on design of passenger ships are good, provided that the reader is prepared to modify some of the statements by applying his own experience.

There is a tendency to rely too much on the use of formulæ and tables, and the estimator who blindly followed the book in this respect would be rather misled.

The book is a very useful form of ready reference, and will probably be more generally used by designers as a guide to the formulation of their ideas and experience, rather than as a source of exact data.

"Projective Geometry", by B. C. Patterson, Ph.D. Chapman & Hall, 276pp., 113 illus., 17s. 6d. net.

The author points out that the production of his book is based on the outgrowth of a set of notes used for a number of years in a semester course in projective geometry offered to juniors and seniors.

Commencing with a lucid elementary conception of projective or synthetic geometry, the treatise is carried to a comprehensive discussion of the subject consistent with an introductory course. Among the outstanding departures from the usual treatment of this study are the following: A careful distinction is made between projective and non-projective ideas. The principle of duality—a reciprocal relation existing among the axioms—is developed in some detail and is emphasized throughout the book. The proofs of some of the earlier theorems are separated into steps, a procedure familiar to the student from his study of elementary geometry. The proof of the fundamental theorem is built on the concept of a net of rationality on a line. The theory of the imaginary elements is introduced in some detail and is applied to various construction problems. A "focus" is introduced by the Plücker definition, i.e., as an ordinary point of intersection of tangents from the circular points.

The chapters on Desargues', Pascal's and Brianchon's theorems are exhaustive. In the final chapter there is a brief discussion of the general projective group of transformations on a plane and of some of its sub-groups, of which the most notable is of course the metric group.

The exercises are grouped together at the ends of the chapters and sufficient material is contained therein for a full year's course in the subject. The diagrams illustrating the text are neatly prepared and the whole clearly printed and carefully produced.

"In Sail and Steam", by Captain V. L. Making. Sidgwick & Jackson, Ltd., 320pp., illus., 8s. 6d. net.

This book provides very interesting reading and is well put together. It is valuable as an authentic account of life at sea by one who has entered into all the experiences of both sail and steam. The subject matter is compiled largely from letters written at the time and not from memories many years afterwards when the mind of the writer has perhaps acquired a different perspective of the facts. The earlier part of the book, dealing with the writer's sailing ship days, reminds the reader very forcibly of Dante's Inferno and the passage of a lost soul through the ice fields of cleansing "until the foul deeds of his youth are melted away". Relief of the tortured soul is experienced when the Elysean fields of steam are reached and a warmth is comprehended by the penitent in his voyage through Purgatory till he reaches the haven which he has been seeking—an appointment as a brass bound commander of a liner.

There is, however, a sad side to the story, that relating to the treatment of seagoing men by those who exploit them and their families for gain. To one who does not know the law on the point it seems incredible that a sailor's pay should stop the moment his ship has been abandoned through no fault of his own and that his allotment to his wife at home also ceases, he becoming a "distressed British seaman" and she a seeker for parish relief. It should not be possible for a man of integrity and ability to give the best of his life to employers only to be cast aside when he no longer serves the purposes required. It is perhaps a sign of the times that shipping interests are waking up to the fact that the labour market can no longer be continued on these terms and that better conditions must be and are being offered.

The author's effort should receive acknowledgment by the large circulation which his book deserves and he would do well to give the general public further food for thought in later works. This book is well written and well worth reading.

"Hydraulics and Mechanics of Fluids", by E. H. Lewitt, B.Sc. Sir Isaac Pitman & Sons, Ltd., 5th edn., 409pp., 188 illus., 10s. 6d. net.

The first edition of this work was published in 1923, and now appears revised and enlarged in its fifth edition. It covers fully the syllabuses of the B.Sc. (Engineering) and the Institutions of Civil and Mechanical Engineers, as well as that of The Institute, in this subject, and has long been recognised as a standard text book on hydraulics.

As in the earlier editions the book deals with the properties of fluids, buoyancy, flow of liquids, orifices, notches and weirs, friction and flow through pipes and channels, pumps, turbines and hydraulic machinery, but new chapters on the aerofoil and its applications and the boundary layer are now included.

The author deals thoroughly with fundamental principles and, by keeping mathematical work as elementary as possible and adopting an easy rate of progress, achieves his aim of treating a somewhat intricate subject in a manner well within the capabilities of the average student.

Each chapter is concluded by a number of set problems so that the student may at once test his grasp of the subject. This is a commendable practice in all text books, but where, as in the case of this book, the answer follows immediately upon the question, it is probable that much of the value of the test is lost, since knowing the answer in advance, the student tends to make his treatment of the problem suit the result. From this point of view, therefore, it would seem desirable to adhere to the established custom of grouping the answers to all problems in a separate section at the end of the final chapter.

The book is well produced and handsomely bound, and merits the close attention of all who are interested in hydraulics and kindred subjects.

EDUCATION GROUP.

The Rôle of the Technical College in the Training of Marine Engineers.

An address on the above subject was delivered to a meeting of the Education Group at The Institute on Thursday, November 25th, 1937, at 6 p.m., by Dr. F. T. Chapman, one of His Majesty's Inspectors, Board of Education and co-opted representative of the Board on the Executive Committee of the Group. Mr. F. H. Reid, Chairman of the Executive Committee, presided. A short discussion followed the address, and the cordial thanks of the meeting were expressed to Dr. Chapman by the Chairman.

It is hoped to publish a summary of the address in the January Transactions.

The half-yearly meeting of the Group followed at 7 p.m., during which the Chairman reported the decision of the Council to approach the authorities concerned with a view to establishing a Marine Endorsement of the National Certificates in Mechanical Engineering.

JUNIOR SECTION.

The Future of the Diesel Engine.

The annual meeting of the Junior Section with the Students of the L.C.C. School of Engineering and Navigation, Poplar, took place on Thursday evening, the 18th November, 1937, when a large audience was privileged to hear Mr. W. S. Burn, M.Sc. (Member) deliver a thoughtful and intriguing discourse under the above title. Mr. J. Paley Yorke, O.B.E., M.Sc. (Principal of the School) again honoured the occasion by occupying the Chair.

A series of about seventy slides illustrated the lecture and made clear the grounds on which Mr. Burn based his conclusion that the present trends indicated that the Diesel drive of the near future was destined to take certain forms which he specified. Into what these would ultimately be resolved was a problem, he felt, on which it was inadvisable at present to venture an opinion.

On the proposal of Mr. T. A. Bennett, B.Sc. (Head of the Marine Engineering Department of the School) a vote of thanks was warmly accorded to Mr. Burn, and on behalf of the Council Mr. B. C. Curling (Secretary of The Institute) thanked Mr. Paley Yorke and his Staff for the enthusiastic co-operation which they continued to give in arranging these meetings.

Compressors and Blowers for Marine and Industrial Use.

On Thursday, December 2nd, 1937, a very successful joint meeting was held at the Central Polytechnic, Croydon, when Mr. E. Markham, Wh.Ex. (Member), delivered a most instructive lecture under the above title to a large audience. Mr. A. C. West, B.Sc. (Principal of the College) occupied the Chair.

Mr. Markham engrossed his audience for about two and a half hours with an outline of the fundamental principles of air compression, followed by a comprehensive description of compressors and blowers made by various manufacturers in this country, and illustrated his remarks by lantern slides of which he had a notably valuable collection. Indeed these were so greatly appreciated that, although the discussion and questions which usually form one of the most useful features of these lectures had to be omitted on this occasion through lack of time, general satisfaction was felt that the lecture was in no way curtailed.

At the conclusion of the proceedings Mr. Markham was accorded a hearty vote of thanks on the proposal of the Principal.

ABSTRACTS OF THE TECHNICAL PRESS.

Development of Hull Form of Merchant Vessels.

Brief reference is made to the influences which must have had their effect in determining the shape given to the earliest built ships. The effect of form of motive power is also discussed. In the beginning only "oar" power was available, and the great growth of this during the period 400-100 B.C. is shown. The changes in proportions and dimensions of large sailing ships from the time of Henry VIII through the "clipper" period to the present day, and the relative water resistance of the earlier and later hulls are given. The method of developing hull forms of large sailing ships in the 17th century is given in some detail, and a set of lines based on Deane's instruction is given. Details of some resistance experiments with various shaped bodies made by Fortree at about this time are given with his conclusions. Reference is made to the early theoretical work of Euler, Romme and others in the period 1750 to 1800, and to the conclusions as regards form of hull which were formulated in France by Romme and in England by Beaufoy as a result of their experiments. The changes due to the introduction of steam as a motive power and iron as a material are discussed. Scott Russell's method of drawing hull level lines based upon his wave-line theory is given and compared with Rankine's "lissoneoid" hull lines derived from the stream lines around various ovals. The next great advance in both theory and methods of experimental research, due to W. Froude, is discussed together with the recent dependence on model experiments. Some general changes in hull form are referred to and consideration is given to the possibility of further increase in ship speed.—*Dr. G. S. Baker, Trans. of the North-East Coast Institution of Engineers and Shipbuilders, November, 1937.*

Modern Motor Coasters of Restricted Draught.

The author deals with modern motor coasters of restricted draught which, during the last few years, have been in increasing numbers making their appearance on the British coast. Representative types are described and illustrated and the special features affecting successful operation of these vessels are dealt with. The general lay-out of this type of ship is discussed with particular reference to ballast arrangements, cargo spaces, cargo handling gear, accommodation and steering arrangements. The engine-room arrangement, including main and auxiliary machinery, is dealt with and details of typical installations are given. Some notes are included on the subject of machinery maintenance, and a suggested programme of overhauls based upon the author's personal experience in the operation of such ships is submitted.—*C. H. D. Rogers, Trans. of the North-East Coast Institution of Engineers and Shipbuilders, December, 1937.*

Marine Engine Lubrication Practice.

That the principles and design of the lubricating arrangements in marine engines are effective and are rightly carried out is made evident by the freedom from heated or excessively worn bearings and the concern caused when such do occur. That the oil industry has done its part in supplying oil of the right characteristics for the different uses involved is equally obvious. A number of marine superintendent engineers have been consulted and the general experience seems to be that they have had in recent years little serious trouble with lubricating oils. This is the more remarkable since the more modern types of machinery, the geared turbine and the oil engine, impose much heavier duties on the oil than the old reciprocating steam engines. The high temperatures in the superheated steam cylinder and the oil engine cylinder, the severe conditions in reduction gearing, and the long periods during which the oil is in use in a closed system are all more difficult conditions than obtained in general marine practice until after the War. Few of the old school of marine engineers were trained to appreciate abstract friction problems of power transmission, although they all rapidly acquired intimate experience of marine lubrication practice and realized that methods of application are as important as the quality of the oil. The majority accepted the oil purchased for their use with complete reliance on the supplier and hoped for the best, but a more critical generation is now arriving, which wishes to know more about the lubricating oil and its suitability for the part of the engine for which it is intended.

Oil Engines.—The arrangement for the return of bearing oil to the tank in the double bottom, and the course of the oil in that tank, apply to the oil engine as to the turbine. There is often a tendency to over-lubricate rather than to under-lubricate, especially in oil engine cylinders and compressors, and standard figures should be worked out for each type of engine. If these are exceeded an explanation should be required and found.

At the end of the compression stroke in the cylinder the piston rings are being forced out at a maximum pressure, say, 400-600lb. per sq. in., against the liner wall; the gas temperature is at a maximum, say, 1,100-1,400° C.; and the piston comes momentarily to rest. It is obvious that under these conditions boundary lubrication will be in force and that an ample supply of oil of a good load-carrying quality is needed. On the other hand, there is the danger that solids may build up in the piston ring grooves if too much or too unstable an oil is used; which points towards the use of a pure mineral oil with high stability to heat and oxidation, good oiliness and small frictional loss and high viscosity to effect sealing adequately.

The lubrication of piston and liner is carried out by mechanically operated lubricators delivering oil to two or more points on the cylinder liner in the scavenging area. The feed should be timed to inject the oil as the piston reaches the bottom dead centre after firing, and between the second and third rings from the top of piston. There should be one lubricator pump for each point of injection of each cylinder; the number of points will vary with the size and type of engine. In oil engines where the cylinders open into the crankcase, the oil film formed on the cylinder walls is swept down into the crankcase, and all the oil is eventually exposed to heat sufficient to cause oxidation. Part of the oil is decomposed, resulting in formation of carbon, giving a black appearance. Separation and filtration remove the solids, leaving the oil darker than before, but the viscosity is not materially altered. With trunk engines a marked saving of lubricating oil has been effected by fitting efficient skirts to the pistons. One of the worst causes of rapid wear is "blow past". The piston rings, if of the Ramsbottom type, should be located in their grooves in such a manner that exhaust gas must take a labyrinth passage to pass the piston. If the gaps in these rings come into line so that gas can blow past the rings, the oil film is destroyed and wear is rapid. The modern sealed rings obviate this defect. The piston ring wear is more serious in that the vertical wear makes the piston grooves deeper and ultimately ruins the initial uniformity of depth; a cast-iron ring must then be fitted in the grooves to save the piston.

Oil Engine Bearings.—Shaft journals, pins, etc., of oil engines are lubricated by a forced circulation system at a pressure of about 25lb. per sq. in. The oil, as a rule, is led through the crankshaft, crank webs and pins to all the bearings. The same lubricating oil is employed in many engines to cool the main pistons and is discharged to the same tank to be cooled, filtered, and returned to the circuit. During circulation under elevated temperatures, the contact with air, water, and metallic oxides tends to oxidise the oil, some of the resulting products being deposited where temperature is low and movement lessened. The effectiveness of oil coolers is lessened by sludge deposited in this way on the cooler tubes. The oiling system must be thoroughly clean before the engines are set to work. It is amazing how much rubbish comes out of the system when first scoured out by the oil.

In Diesel engine crankcase systems the oil should have the characteristics set out in the Appendix. The sludge value clause was introduced into specifications for Diesel crankcase systems on the basis of work which had already been carried out on transformer oils. Cylinder bearing and compressor lubrication are separate problems and should be dealt with separately. There is an inclination to use one type of oil throughout, but it is doubtful whether the best results can be obtained in this way.

Mechanical Lubricators feed the exact amount required while the engine is running. Feed sights enable control to be kept of their working and the amount being fed. The speed of the engine determines the amount and timing of the oil feed, and the supply of small quantities can be kept up with regularity, uninfluenced by temperature or other altering circumstances. Each point can be dealt with separately. These lubricators can pump against high pressures. It is a great help to have a small motor-driven oil pump and connections fitted so that the oil reservoirs on these mechanical lubricators can be quickly and cleanly filled with the right oil without possibility of mistake or waste. Two-stroke engines have suffered from fires in the scavenge belts which in some cases have been traced to excess cylinder oil carried into these spaces by the scavenge air. In the hope of effecting a cure the amount of oil has been reduced, while ensuring that sufficient oil is injected to keep the piston rings

OIL SUPPLIES FOR MARINE ENGINE LUBRICATION.

Dimensions. No. of Cylinders.	Bore mm.	Stroke mm.	S.h.p.	Speed r.p.m.	Type.	Remarks.	Oil per 1,000 h.p. per day, gals. Engine.	Cylinder.	Compressor.	Total oil per 1,000 h.p. per day, gals.
MAIN ENGINES :—										
8	740	1,150	3,000	115	Four-stroke single-acting	Unsupercharged air injection	1·84	1·08	0·19	3·11
8	630	1,100	1,850	125	Do.	Do.	1·94	0·99	0·11	3·04
6	620	1,300	2,750	140	Do.	Supercharged air injection	0·77	1·08	0·05	1·90
8	740	1,500	4,300	110	Do.	Do.	0·86	0·89	—	1·75
6	450	1,200	3,300	120	Two-stroke double-acting	Airless injection Do.	0·48	1·25	—	1·73
AUXILIARY ENGINES :—										
3	320	350	160	300	Four-stroke single-acting	Air injection	8·0	3·4	3·2	14·6
3	12·8	16·0	160	300	Do.	—	3·9	0·80	0·80	5·5
3	330	600	250	300	Four-stroke single-acting	Airless injection	7·3	*	1·2	8·5
4	220	370	250	400	Two-stroke single-acting	Airless injection	17·7	0·89	0·41	19·0

* Compressor oil used for cylinders to prevent wrong oil being used on compressor.

in a free condition to prevent the passage of exhaust gases.

In the older type of engine the fuel is injected by means of blast air. This blast air is delivered by three-stage air compressors at a pressure of 800-1,200lb. per sq. in. These air compressors need special lubrication which acts as a piston seal and protects the cylinder walls. This is effected by means of mechanical lubricators of similar type to those used for the main engine cylinders. As there is always a certain amount of water vapour present in the compressors, a saponifying type of oil must be used, as set out in the Appendix.

Condition and Cleaning.—The full flow of oil is passed through strainers, and a portion, say about 2 per cent., is by-passed into a centrifugal separator, which is started as soon as the oil system pumps and the clean oil is returned to the system. The correct size of diaphragm should be used to make certain of completely purifying the oil passed. For good separation a temperature of about 180° F. is desirable.

In a typical installation of a six-cylinder two-stroke supercharged double-acting engine and three three-cylinder auxiliary engines, the separator passed 110 gals. per hr. and extracted on an average 0·057 per cent. of sludge. This was sufficient to keep the system in a sweet condition. Make-up oil amounting to about 20 per cent. of the bulk is introduced every four to five months, so that 100 per cent. is renewed in, say, two years' time. Any additions of new oil should be made gradually, as there is a tendency to form sludge if a large amount of new oil is introduced into the system too rapidly. Oil which has become more or less contaminated in use is dealt with by a settling tank with an internal steam coil. The bulk of the water and heavier material is thrown down and the oil is drawn through a high suction valve to a heater, whence it passes through a centrifugal separator back to the oil sump. Unfortunately these separators can only extract matter with a different specific gravity from the oil. When going through the centrifugal separator it has been found good practice to allow a small amount of hot water at a temperature of 180-200° F. to mix with the oil before it is separated. Some oil will be lost in the form of sludge from the resulting emulsion, but once the oil has all been passed through the washing process the consumption will be restored to normal. The broad fact is now realized that lubricating oil cannot be injured by mechanical use, and the old belief that oil "lost its nature" through punishment of this kind is disappearing. Loss in lubricating value is caused by impurities formed in or picked up by the oil and the only reason for which oil is now condemned as unfit for use is that the contamination by salt water or fuel oil is so great that it would be uneconomical to attempt to purify it.

The service of an oil specialist to deal with the physical and chemical qualities of the oil purchased are essential if high lubricating efficiency is to be obtained and maintained. His work falls into three classes: (1) the examination of new oils, (2) the examination of the oil in use in systems, and (3) post-mortems.

The tests of new oils are intended to cover general quality and to decide whether or not satisfactory service can be expected in specific cases. The examination of oils actually in systems is made to ensure that such oil is in a satisfactory condition, that its breakdown by oxidation in service is not taking place at a faster rate than is made up by fresh supplies, and to keep a careful eye on dilution by fuel or water, sea or fresh. The third class—post-mortems—explains itself. Such extraordinary happenings have occurred as the accidental introduction of linseed oil and of bean oil into lubricating systems, leading to curious results and lengthy and costly periods of elimination.

Lubricants.—In practice the limits between which oils must lie to give efficient lubricating service in any given type of engine are wider than were formerly believed. The principle governing lubrication is that suitable oil must be available at the points in the machinery where it is wanted, when it is wanted. Heavy sludge in the crankcase oil may ultimately result in the oil not arriving at the lubrication points when it is required, and mechanical breakdown is the result. If sludge is not removed its presence leads to the production of more sludge on a sort of compound interest law; hence the advisability of continuous separation of the oil under suitable conditions. Provided that oils have the same broad general characteristics, no bad results will be found in practice on mixing oils from different suppliers.

After examining a large number of samples of various lubricating oils, it is difficult for the purchaser to understand the difference in the prices of these oils. It would be of much interest if those who control the supply and sale of lubricating oil would indicate whether, in their view, we are faced with a shortage of supplies in this area of oil production. In view of the fact that oil is not easily destroyed, although it may become contaminated, has the time arrived to add largely to our resources in the matter of oil saving and reconditioning plant? The amount of oil that is allowed to run away after being used once must be considerable, and much of it cannot be greatly the worse for having passed through a bearing before being lost.

APPENDIX.

Lubricating Oil Specifications.

Diesel Engine Crankcase Oil.—To have the following characteristics:—

Flash point (Pensky Martens), above 400° F.
Viscosity (Redwood) at 70° F., 1,600-2,000 secs.
Viscosity (Redwood) at 100° F., 525-625 secs.
Viscosity (Redwood) at 140° F., 170-185 secs.
Viscosity (Redwood) at 180° F., 80-85 secs.

This sludging value (Michie) as determined by the standard I.P.T. test is not to exceed 0·8 per cent.

Diesel Engine Air-Compressor Oil.—To have:

Flash point (Pensky Martens), above 400° F.
Viscosity (Redwood) at 70° F., 1,800-2,200 secs.
Viscosity (Redwood) at 100° F., 575-675 secs.
Viscosity (Redwood) at 140° F., 180-200 secs.
Viscosity (Redwood) at 180° F., 85-90 secs.

To be slightly compounded (say, 3 per cent.) with neutral fatty oil.

Diesel Engine Cylinder Oil.—To have approximately the following characteristics:—

Specific gravity at 60° F., 0·897.
Flash point (Pensky Martens), above 500° F.
Viscosity (Redwood) at 140° F., 575 secs.
Viscosity (Redwood) at 180° F., 220 secs.

Abstract in "The Motor Ship", November, 1937, p. 284, of paper contributed by S. B. Freeman to the discussion on "Lubrication and Lubricants" before Institution of Mechanical Engineers.

The Lubrication of Bearings of Internal-combustion Engines.

Oil Film Thicknesses.—Theoretical and experimental studies of film lubrication now enable the coefficient of friction, eccentricity ratio, and minimum film thickness to be calculated within a tolerable margin of error, for bearings in which the load is constant in magnitude and direction. These two conditions are not fulfilled in most internal-combustion engine bearings and it is recognized that variation in the direction of loading is beneficial in preserving the integrity of the oil film.

It is well known that wear of the crankpins of four-stroke internal-combustion engines is greatest on the side of the crankpin nearest the centre line of the shaft. This effect is particularly easy to observe on high-speed engines. A similar effect is noticeable on the main bearings of both four-cycle and two-cycle engines when the bearing lies between two unbalanced cranks on the same centre or with an angle of 90 deg. or less between them. In these instances the maximum wear is in line with the resultant centrifugal force which lies in the same direction as that of the maximum inertia in the neighbourhood of both top and bottom dead centres.

In such cases three conclusions seem legitimate:—

(1) The more intense but briefly sustained pressures due to combustion are of small importance compared with the less intense but longer sustained pressures due to centrifugal force and inertia.
(2) The wear (such as it is) is incurred during full-speed running and not when starting or stopping.
(3) Since continuous running under conditions of boundary friction is out of the question, the wear is presumably due to abrasive particles passing through the oil film at its thinnest part.

Whatever means are provided for the filtration or centrifugal separation of lubricating oil should be capable of eliminating solid particles smaller in diameter than the minimum film thickness of the bearings.

Conclusions (1) and (2) suggest that for purposes of calculation the resultant pressure on bearings most subject to wear may be represented by a steady load equal to the centrifugal force of the revolving weight plus half the reciprocating weight of one set of parts multiplied by a factor taking into consideration the angle of spacing of the cranks each side of the bearing. This factor would be $\sqrt{2}$ for two cranks on the same centre and $\frac{1}{2}\sqrt{2}$ for two cranks at right angles and so on. On this basis the film thickness has been worked out for a number of typical instances.

In the first six examples the crankshaft diameters vary from 11 to 29 cm., the speed of revolution from 1,200 to 300 per minute, and the equivalent centrifugal bearing pressures from 61 to 25 kg. per sq. cm. The minimum film thicknesses all lie between the limits of 0·5/1,000in. and 1·25/1,000in. In the last two examples of crankshafts 34 and 43·5 cm. in diameter respectively, running at 170 and 120 r.p.m., on account of the use of balance weights the centrifugal pressures work out to low figures, namely, 15·5 and 12·7 kg. per sq. cm. and the corresponding film thicknesses are 1·3/1,000in. and 1·5/1,000in. In the last two examples the loading due to gas pressure is probably more important than that due to centrifugal force and inertia.

Cylinder Lubrication.—For trunk engine types it is the usual practice to use only straight mineral oil. Although this may not be exactly the best lubricant for cylinders in this type of engine, it is found preferable to use straight mineral oil, to avoid complications due to contamination. For engines in which the cylinders are separate from the crankcase, and in which it is now customary to use a straight mineral oil, it will no doubt be found that a compound oil is much more suitable, because in cylinders especially boundary lubrication is probable. Recent investigations show that a straight mineral oil, compared with a compound oil, is very deficient under these conditions, and no matter how finely the metal surfaces are finished it seems to be impossible with high pressures and temperatures to avoid boundary lubrication near the combustion space, resulting in the wear and tear which takes place with the materials used for liners and piston rings.

Bearing Lubrication.—For bearings it is usual only to use a straight mineral oil, supplied under pressure through filters. Filters, however, can only collect the sludge and abraded particles as they are carried along with the oil stream, after they have wrought their harmful influence in the engine

Film Lubrication of Two-Cycle Trunk Engine Main Bearings.

Example.	1	2	3	4	5	6	7	8
Diameter of crankshaft, cm.	11·0	12·5	15·0	18·0	22·0	29·0	34·0	43·5
Length of bearing, cm.	6·2	6·6	8·0	9·6	18·0	21·2	23·8	30·0
Speed, r.p.m.	1,200	1,000	800	600	375	300	170	120
Equivalent centrifugal bearing pressure, kg. per sq. cm. ...	53·0	56·5	61·5	38·3	25·9	24·6	15·5*	12·7*
Calculated coefficient of friction	0·0067	0·0058	0·0046	0·0053	0·0045	0·0041	0·0038	0·0034
Calculated minimum film thickness, thousandths of an inch	0·55	0·53	0·51	0·72	1·07	1·2	1·3	1·5

* Balance weights fitted.

bearings. So it does not seem possible to diminish to any great extent the usual wear and tear of bearings by any method of filtration.

For an enclosed crosshead engine it may be possible in the future to lubricate all the bearings with water or a water-oil mixture when all the bearings are filled with a composite material suitable for this type of lubrication. After all, in ordinary pressure oil lubrication the oil is used more for cooling than for ideal lubrication, and therefore gives rise to more waste.

Up to the present, experiments have only been carried out with lower surface speeds and pressures than those which are commonly used in actual practice, so it would be of considerable help to carry out further research and experiments on points such as the following:—

(1) Constant speeds with variable pressures for the same number of revolutions compared with constant speed and constant pressure.

(2) Influence of oil grooves at variable speeds and pressures.

Abstract in "The Motor Ship", November, 1937, p. 285, of paper contributed by V. Mickelsen to the discussion on "Lubrication and Lubricants" before Institution of Mechanical Engineers.

Wear of Cylinders and Piston Rings.

Bearings.—Any engine bearing works badly when faulty in (1) design, (2) oil supply, (3) lubricant, or (4) quality of bearing metals. Regarding the latter, it has been shown (Young, 1935) that certain reciprocating bearings, namely, the top end brasses of marine engines, which gave constant trouble over many years, would function successfully on substituting a good bearing metal in place of one which was not good. Further, the main difference between the bad and the good metals was one of *structure,* the successful structure being that which was able, under working conditions, to form and maintain a surface easily wetted by oil and retaining an oil film upon it.

A piston ring and a cylinder wall form a reciprocating bearing which functions satisfactorily from one end almost to the other end, at which extremity it wears. If the rate of wear was no greater than it is over the major portion of the surface covered, this paper would be unnecessary. Cylinder wear, then, occurs at its maximum on that area of the cylinder wall exactly facing the uppermost free piston ring when at its top position nearest to the compression zone. The effect is less opposite the ring immediately below and fades away to little of consequence beneath the lowest compression ring at its topmost position. The circumstances under which, in the author's experience, the rate of wear at these points is more, or less, will be considered.

Cylinder Design.—A greater thickness of that part of the cylinder wall facing the top position of the upper rings, when causing increased heat sufficient to harm the oil film, leads to a higher rate of wear. Engines of some vessels passing through the Persian Gulf, where the cooling system is probably 30-50° F. above normal, have suffered more rapid wear than when operating in temperate climates.

Any factor causing excessive heat at the top of the cylinder increases wear by destroying the oil film; and undue wear, apart from that caused by abrasive dust, occurs when contact is established between the material of the ring and that of the cylinder wall. The rate of wear is low because moments of contact are rare, otherwise excessive wear would happen in a few hours.

Stopping, Starting, Manœuvring, Idling.—The total number of times the piston reciprocates over a given number of hours or miles of running provides no indication of the total cylinder wear experienced. The cylinder is not worn at a rate governed by the *amount* of use. Services involving many stops of long duration suffer a rate of wear higher than those engines having stops which are no fewer but are of short duration, insufficient to allow the oil film to drain or the system to cool or, if the engine idles, to permit of local overheating.

Speed of Piston Travel.—High speed affects the whole area and both extremes of the area travelled and, by itself, does not cause increased cylinder wear. Very low speeds may not be so innocuous.

Surface Finishes.—When the Brinell hardness number of the metal of the cylinder or liner is below about 850. it forms in time its own working skin; hence the running-in of such irons is a period when their surface is forming. Bearing metals come out of service with a skin totally different from their machine-shop finish; they possess the power to form (and to re-form, when damaged) a surface such as is wetted by oil as an unbroken film. The ideal finish for a piston ring would consist of an amorphous skin. In large engines extremely rapid bore wear, ring wear or breakage occurs, if, on reboring the cylinder, ferritic iron is exposed (Young, 1935).

Corrosion by Oil.—The author has shown (1927) that used oil from motor ships and automobiles sometimes is corrosive and that chemical tests fail to detect it. This finding was unpopular, though Taub has stressed its importance recently (1937). Diesel exhaust gases (from fuel containing 1 per cent. of sulphur) contain about 0·04-0·05lb. of sulphuric acid per 1,000 cub. ft.; moreover, if the air (from a "wet" compressor using compound lubricant) passes through copper inter-cooler tubes it gathers corrosive copper salts.

The degree of attack on liners by contaminated oil is infinitesimal, but it lowers the fatigue resistance of the skin of the iron. Wear starts with the weakening and loosening of one particle, causing overstress of adjacent particles, fatigue sets in, the particles loosen, disappear and the iron has "worn". Machine-shop finishes stress the surface layers, affecting the "life" of the liner to a degree yet unexplored. Prof. Sawin (1937) of Skoda Works states that wet and dry grinding decrease resistance to wear at a depth up to 0·05 mm. and that wear resistance bears no known relationship to chemical analysis or to hardness of materials not of the same origin and structure, all of which the author's experience confirms.

Brinell Hardness and Tensile Strength.—The best-wearing grey iron liners which have come before the author have been neither hard nor very high in tensile strength. When sheer hardness is relied upon it should exceed a Brinell hardness number of 850 to be efficacious.

Grain Size.—"Close-grained" cast iron is a fetish. Very good liner irons, including "Lanz Perlit", have larger grain than ordinary irons. Little is known about the effect of grain size; it has been a superstition like that concerning sulphur.

Heat.—Fluctuations of heat entering a liner affect, as the sun's warmth affects an ocean, the surface layers only. Those of the order of 11° C. at 0·5 mm. depth become of the order of 3° C. at 1·5 mm. and disappear altogether at 5 mm. The temperature and the fluctuations in the skin of the iron, say, at 0·001 mm. depth are unknown quantities; the author believes they are great and produce stresses leading to fatigue breakdown of the skin, particularly just after an engine is started from cold and while no heat gradient to the water exists.

Sulzer (1926) showed that the piston rings of a two-cycle marine Diesel engine were cooler than those of a steam-jacketed steam engine, the top ring of the former attaining a maximum of only 126° C. (259° F.) when the liner had a maximum temperature of 253° C. (487° F.) and the piston crown (midway between centre and circumference) had 211° C. (412° F.), all measurements being at 0·5 mm. depth. Over-heating vitally concerns two areas, namely, the surfaces of the upper part of the liner and of the top piston ring grooves, one effect being always destruction of the oil film followed by wear of liners or grooves, stuck rings, etc.

Conclusions.—There are no mysteries about bore wear, only difficulties. Every bearing requires oil and a bearing metal; and none, especially a reciprocating one, can be expected to wear normally when one portion of its travel sometimes lacks oil film or a film free from powders and is exposed always to fluctuations of heat together with superheated steam or the gases of combustion. Much can be done to get oil to the top position, but this, to be successful, also needs improved metals for the barrels, improved bearing metals with, also, the power to resist abrasive action whether caused by dust, by the destruction or absence of the oil film, or the action of corrosive oil and gases.

Skin fatigue, the author believes, is the cause of the start of particles disappearing, i.e., of wear; this fatigue is not only induced by scoring during dry or semi-dry moments of contact, but by etching attack, heating and cooling stresses, and stresses left by machine-shop finishes.—*Abstract in "The Motor Ship", November, 1937, p. 286, of paper contributed by H. J. Young to the discussion on "Lubrication and Lubricants", before Institution of Mechanical Engineers.*

Shop Work in Welded Ship Construction.

The writer draws attention to a description of the welding plant of the Sun Yard of Chester, Pa., which was presented to the American Society of Naval Architects and Marine Engineers in a paper read by Messrs. John W. Hudson and T. M. Jackson. This related to the welding of a length of 117ft. of the middle body of a bracketless tanker, consisting of three tanks, each 35ft. in length, and a pump room 12ft. in length. All the parts of the structure were assembled in one tank length, the sequence of operations being so arranged as to reduce hand welding on the slip to a minimum. The machine welding plant of the Sun Yard consists of a tack-welding table, a tilting table for long straight welds, and a butt welding machine. With a plate on the tacking table, the stiffeners are placed in position and tack welded. The plate is then passed to the tilting table and tilted to the required angle to enable the welding head, which is secured vertically and traversed automatically on a runway, to deposit a symmetrical fillet between the stiffeners and the plate. The plates with stiffeners attached are then rolled to the butt welding machine where they are welded by an automatic traversing head. Finally the completed section is lifted and the underside is reinforced by a light run along the joint. Departures from normal ship construction methods further include the elimination of all rolled sections, for which heavy flat bars are substituted and the use of heavy wrapper plates at the bulkheads, on the decks and on the bottom and side shell plating. Commenting in this connection on

British welding practice, the writer considers that the shipyards of this country have not been so backward in the use of welding as the publicity enjoyed by some foreign applications of the process might appear to indicate; and while he acknowledges the inherited tendency of British practice to rely upon the man rather than on the machine, he urges that the industry cannot allow it to be supposed that the further development of tools and technique is entirely in the hands of its overseas competitors.—*"Engineering", 15th October, 1937, p. 431.*

The Engineer as an Expert in Law Suits.

The author observes that the need for technical experts in juridical matters is rising parallel with the ever-increasing importance of engineering technology in the life of the community. After enumerating the various functions which the engineer may be called upon to fulfil in the courts, the systems of appointing engineering experts in Sweden are discussed. The present system, introduced in 1935 and designed to eliminate the danger of excessive partiality on the part of opposing experts, gives the Court the right to appoint its own technical adviser, but first allows the litigants the opportunity of agreeing as to the suitability of the nominee, the final decision remaining always with the Court. Various weaknesses in this system are indicated and it is urged that the disputing parties should still be afforded the right to call in their own technical representatives or advisers as formerly. It is suggested that the experts called in by the disputing parties should not be substituted by a Court-appointed expert, but should rather be complementary to him. The writer also suggests that considerable time, energy and temper might be saved if interested parties resorted more to arbitration outside the Courts, appealing for this purpose to authorities whose competence and impartiality had previously been accepted. Some remarks are made regarding the various factors which should guide the engineer in submitting his evidence or opinion and a warning is given of the danger of attempting to assume the role of the advocate, also of advancing his principal's interest at the expense of his own professional conscience. He quotes in this connection, "Never express an opinion which you are not prepared to defend before any technical body whatsoever". Emphasis is laid on the need for clarity of expression, and it is asserted this is, or should be, the natural atmosphere of the engineer. Facts and personal opinions should be strictly separated and the basis for the latter given wherever possible. Admitting in conclusion that even the cleverest and most conscientious technician is liable to err, the writer is nevertheless confident that the engineer in his position of responsibility in the Law Courts will justify more and more fully the trust invested in him, as the demands made upon his specialised knowledge by the ever-advancing tide of engineering technology become increasingly heavy.—*Teknisk Tidskrift, October 9th, 1937, pp. 401-5.*

Boiler Maintenance and Repairs in Cargo Steamers.

The author places on record the most usual and ordinary methods employed at the present time in the management and maintenance of the coal fired cylindrical boilers fitted in coasters, colliers and tramp steamers. He briefly describes the conditions in which the boilers of these different types have to be operated and considers the subject from the point of view of feed water, superheat, troubles and methods of repair with special reference to the effect of higher steam pressures and temperatures. Discussing the supply and treatment of feed water, the author recommends a method customary with many engineers when taking away new ships. This consists in washing out and filling boilers with clean sea-water until "bleeding" of the boiler material ceased. A slightly alkaline content is then maintained by the use of common soda while using salt make-up. A thin coating of white scale is formed in these conditions, but the author states that the coal required to evaporate extra-make-up feed would greatly exceed the amount required to overcome the heat resistance of the scale formed. Referring to superheat the author stresses the desirability of using a pure mineral oil as a cylinder lubricant, as blended oils are liable to form fatty acids at high temperatures and pressures, while a pure oil on reaching the boiler will rise to the water surface where it can be "scummed" off. The author considers that the value of adequate feed heating from a boiler maintenance point of view is still not sufficiently appreciated. Not only are the steaming conditions improved by using hot feed at a temperature near the boiling point, but the air is driven out of the feed water before it enters the boiler and its very active corrosive effect is stopped. Referring to the failure of boiler materials by "caustic embrittlement", i.e., by chemical intercrystalline fractures as distinct from fatigue fractures, the author considers that the extent of this trouble in relation to marine boilers has been considerably exaggerated, although its seriousness cannot be over-emphasized. Should it really become general and should it be proved beyond doubt to be due to the use of high pressure, it is fairly certain to sound the death knell of such higher pressure boilers in cargo vessels, boilers requiring special treatment being unsuitable for cargo ship use where in the engine room one man is frequently on watch by himself.—*E. P. Wilson, Trans. of the North East Coast Institution of Engineers and Shipbuilders, November, 1937.*

Testing Installation for Ships' Fans.

The author describes a testing installation designed to enable shipyards to carry out acceptance trials of the fans employed in ventilation, forced draught, and oil burning systems in an expeditious manner. The installation consists of an electric driving motor running on rollers and a tubular measuring channel which is suspended from an overhead crane. For the testing purposes the fan runner and casing are separately mounted on the shaft of the motor and the end of the channel and aligned on the test bed, the same installation being thus adaptable to either axial or radial flow fans. The measuring channel consists of a pressure chamber of rectangular cross section in which the static pressure of the air delivered by the fan is recorded at the inlet and checked at different points of the cross section of the flow by three movable pressure indicators mounted near the middle of the chamber, so that a mean static pressure can be determined by graphic methods. From the pressure chamber the air passes through a tube of smaller circular cross section at the end of which the volume delivered is determined from the head of water maintained by the dynamic pressure of the flow through the orifices of the interchangeable shutters. Uniformity of the velocities in front of the shutter is obtained by fitting a Venetian blind damper at the end of the pressure chamber, by accelerating the air in the tapered connecting piece joining the rectangular and circular portions of the channel and, finally, by passing it through a rectifier at the forward end of the circular portion. The indicators, gauges, etc. of all instruments employed for the measurement of the power output, air delivery, static pressures and revolutions are mounted in a closed carriage, so that in taking readings the observers are not disturbed by the noise of the shop. The installation is built by the makers in four sizes for maximum air deliveries, respectively, of 40, 400, 1,600 and 4,000 cub. metres per minute (abt. 140, 1,400, 6,400 and 14,000 cub. ft. per min.), and the author states that the maximum error in the determination of the fan efficiency amounts to −2·2 per cent.—*Dr. Ing. Busemann, Werft, Reederei and Hafen, 1st November, 1937, p. 308.*

New Krupp Two-stroke Diesel Engines and Archaouloff Injection.

The author describes the construction of the latest standard design of the Krupp Diesel engine, which is of the single-acting two-stroke crosshead type with solid injection arranged on the Archaouloff system. In this design the cylinders are cast in pairs and bolted together with vertical flanges. The cylinder liners are of cast iron and project for a short distance above the upper edge of the cylinders in such a manner that in way of the combustion chamber an excessive accumulation of material is avoided. The space between the cylinder and the liner narrows from the bottom to the top so that in the zone of the higher temperatures the speed of flow of the cooling water is increased, a maximum of speed round the combustion zone being obtained by forcing the water through cast-in narrow channels. Scavenging is arranged on the transverse flow principle, the main scavenge ports being situated opposite the exhaust ports. Auxiliary exhaust ports situated above the ends of the rows of main scavenge ports provide an ejector effect whereby the stream of scavenge air is forced to rise vertically to the cylinder cover, thus thoroughly scavenging all corners before forcing the remaining exhaust gases through the exhaust ports. A double acting scavenge pump is provided for each pair of cylinders, the scavenge valves being arranged horizontally, so that they respond to the smallest pressure variations. The working pistons conform with the maker's established design practice. The cylinder covers of the latest type of engine which are of large internal capacity are fitted on the under side with thin walled water chambers. In these, narrow channels are formed by means of cast-in ribs; the resulting high speed of flow of the water provides a strong cooling effect and at the same time prevents scale deposits. The Archaouloff solid-injection pump which has been adopted in many of the new engines contains a "gas piston" actuated by the maximum compression and combustion pressure together with a fuel pump piston. The piston areas are fixed in accordance with the injection pressure desired, thus at about 300 atm. (abt. 4,300lb. per sq. in.), the area of the gas piston is about 10 to 13 times that of the fuel pump piston. The commencement of the injection is determined by the tension of the fuel valve spring. As the pump is not mechanically driven the direction of rotation of the engine does not affect its operation. The sense of rotation of the engine is controlled by the starting mechanism, the pump being self-adjusting. A reversible engine fitted in this manner thus does not require any reversing arrangement in the fuel pump and the camshaft is dispensed with. The author states that this system of injection was originally devised for the purpose of transforming existing air-injection engines into solid-injection engines. It has been successfully applied to a whole series of Krupp and other Diesel engines, the greater number of which were fitted in tankers, the existing fuel pumps being utilised as supply pumps conveying the oil to the Archaouloff pump during the low pressure phases. Experience has further shown that with the Archaouloff system the shape of the combustion chamber does not affect the injection so that no modification of cylinder covers and piston, which were originally designed for air injection, is required.—*S. Bock, Schiffbau, 1st November, 1937, p. 345.*

Special Steel in Relation to the Problem of Corrosion and Shipbuilding Technology.

The author gives a detailed survey of the behaviour of various kinds of steel under marine conditions, and arrives at the following conclusions:—

(1) Special steels—nickel, chromium-nickel

and chromium steels—are not generally used in the construction of vessels on account of the cost.

(2) Manganese steel may be adopted for shipbuilding purposes, but it is necessary to produce manganese steel with carbon contents lower than at present in order to obtain a steel of good welding quality.

(3) Steel with low carbon contents (0·2-0·3 per cent.) may be recommended as a material of higher resistance to corrosion in comparison with ordinary mild steel.

(4) Special attention should be paid to mild steel with 1·0 to 1·2 per cent. of copper as a steel of high mechanical properties which can be produced at a moderate price.

(5) Rivet steel at present in use is unsatisfactory and should not be used if the material contains sulphur. In addition to the existing tests for rivet steel a Bauman test should be made.

(6) It is essential to introduce a rivet steel of higher mechanical properties in comparison with the existing material.

In the paper there is also a table showing results of corrosive experiments carried out at Cronstadt and Sevastopol on ordinary manganese and copper steel.—*Prof. Voskresensky, Trans. of the Scientific Technical Society of Shipbuilding and Marine Engineering, U.S.S.R., 1937, p. 5.*

The Application of Special Steel in the Construction of Vessels in Relation to the Technology of Electric Arc Welding.

The author notes that the trend of the development of modern shipbuilding in the U.S.S.R. yards shows a continuous progress of the application of electric arc welding. This welding has become a predominant method in the construction of vessels. The wide application of electric arc welding has already suggested that the present requirements as regards special steel for shipbuilding should be revised, especially for manganese steel. Not more than 0·25 per cent. of carbon content should be permitted, and the yield point of the material should be kept to the lowest limit, also all thin plates should be annealed at the steel works.—*Prof. Vologin, Trans. of the Scientific Technical Society of Shipbuilding and Marine Engineering, U.S.S.R., 1937, p. 11.*

Low Alloy Steel for Shipbuilding.

This paper deals with the consideration of the application of high tensile steel for ship construction and the merits of a low alloy steel. The author suggests the introduction of the chromansil type of steel containing carbon not exceeding 0·3 per cent. for plates and rolled sections, and 0·18 per cent. for rivets. The mechanical and thermal properties as well as the welding qualities are given in the paper according to information already published and the author's own investigations. The technical considerations are supported by calculation of saving in weight, which may be obtained by the use of this steel for the building of river craft. Taking into consideration the behaviour of specimens of chromansil steel under various heat treatments, the author also proposes that it should be adopted for the machinery of vessels in lieu of chromium-nickel, chromium-vanadium steel, etc. and also for castings.—*D. N. Shvetzov, Trans. of the Scientific Technical Society of Shipbuilding and Marine Engineering, U.S.S.R., 1937, p. 15.*

A New Method of Calculating Wave Resistance of Ships.

This paper gives a practical method of calculating the wave resistance of ships based on the application of Mitchell's theoretical formula. The new method of integration of function, in the form ${}_{a}\int f\,(x)^{\sin}_{\cos}\,(ax)$ of Mitchell's formula is developed by the author by graphic representations of ships' waterlines by a system of harmonic coordinates. In comparison with the other existing methods of calculation of integrals containing Furje functions this method has the general character of being suitable for every value of argumentation (a). A numerical example of calculation of wave resistance is given for a ship the model of which was tested by Taylor. (Transactions of the Society of Naval Architects and Marine Engineers, 1908). The calculated resistance is about 13 per cent. higher than that obtained by the tank test.—*R. I. Antimonov, Trans. of the Scientific Technical Society of Shipbuilding and Marine Engineering, U.S.S.R., 1937, p. 63.*

Ship Forms of Least Resistance.

The author analyses various formulæ for the theoretical calculation of hull forms of least resistance (Newton, Joukovsky, Mitchell, Weinblum) and finally gives a method of constructing curves of sectional areas for Froude numbers varying from 0·325 to 0·430.—*Prof. Pavlenko, Trans. of the Scientific Technical Society of Shipbuilding and Marine Engineering, U.S.S.R., 1937, p. 28.*

Effect of Viscosity on the Wave-making of Ships.

The author states that the wave-making resistance of a model deduced by subtracting the estimated frictional resistance from the measured total resistance does not agree closely with the value calculated by the Michell theory. In particular, the humps and hollows are less accentuated and tend to occur at higher speeds in the observed curve than in that calculated. The flattening of the humps is most marked at low speeds. By means of an empirical correction to the calculated resistances, he accounts for a part of the discrepancy in the case of seven models. The correction consists in applying a reduction factor to the portions of the resistance due to the afterbody and to interference between the fore and after bodies, the forebody resistance remaining unchanged. This is equivalent to assuming a reduction in the effective slope of the afterbody waterlines, together with damping of the forebody wave before it reaches the afterbody, effects which may be attributed to viscosity, neglected in the ordinary theory. The factor is zero for values of the Froude number ($v/\sqrt{g}L$) less than 0·15 and tends to unity at high values of the order of 0·5. At these speeds the theory holds, whilst at low speeds all wave-making is assumed to be due to the forebody. The models used in deriving the correction have prismatic coefficients ranging from 0·636 to 0·727, length/beam ratios from 6·4 to 16, and beam/draft ratios from 1 to 3·75. The correction accounts for part of the difference in amplitude of humps and for the difference between resistance ahead and resistance astern in the case of unsymmetrical models, but does not affect the calculated position of the humps.—*W. C. S. Wigley, Trans. of The Institution of Engineers and Shipbuilders in Scotland, November, 1937.*

The Influence of Curvature on Friction.

This paper deals with the results of investigations on the influence of alteration in the length of the ship's model, on the distribution of velocity, pressure, depth of boundary flow and intensity of friction of turbulent flow around the model. The solution of the problem is obtained in a simple form and is applied for the determination of the viscosity of the fluid at the movement of a two-angle body on the basis of the Prandle-Carman theory. The general conclusion relates to the effect of curvature on the friction coefficient at the length-breadth ratio greater than 8.—*I. G. Hanovitch, Trans. of the Scientific Technical Society of Shipbuilding and Marine Engineering, U.S.S.R., 1937, p. 79.*

The Determination of the Spot of Abruption of Laminar Boundary Flow.

The authors give an approximate method of the evaluation of the Polghausen integral equation adopted in order to ascertain the spot of abruption of the laminar boundary flow. The calculated results are in substantial agreement with those obtained by a more exact method.—*Prof. Krilov and I. G. Hanovitch, Trans. of the Scientific Technical Society of Shipbuilding and Marine Engineering, U.S.S.R., 1937, p. 98.*

Nomogram Calculations to Determine the Strength of Propellers.

The author gives various nomograms for ascertaining the ratio of the elements of ships' propellers, and a general rule for calculating the strength of propellers.

The following nomograms are given:—

(1) Nomogram for shaft horsepower, number of revolutions and optimum diameter of propeller.

(2) Nomogram for ascertaining the blade area for a given width of blade and vice versa. The coefficient for correction for extremely wide blades are given separately in the body of the paper.

(3) Nomogram for obtaining the weight of propellers, the weights being given separately for boss and blades of ordinary and aerofoil section.

The general expression for calculating the strength of propellers applies to the root of propellers. The first part of the formula gives the bending stress of the blade on account of thrust and torque, the second part gives the stress owing to bending of raked propeller blades by centrifugal force, and the last part the tearing stress on blades due to the same force.—*A. I. Amaev, Trans. of the Scientific Technical Society of Shipbuilding and Marine Engineering, U.S.S.R., 1937, p. 118.*

A Method to Determine the External Forces Acting on the Hull of a Vessel in Icefields.

The author proposes a method for determining the external forces acting on the hull of a vessel under compression of an ice-field by assuming that these forces are equal to the crushing load of ice. An analysis of this problem establishes theoretically the relation between the thickness of ice, the angle of inclination of the vessel's sides, and compression forces of ice acting on the hull.—*A. I. Maslov, Trans. of the Scientific Technical Society of Shipbuilding and Marine Engineering, U.S.S.R., 1937, p. 129.*

Marine Alternating Current Installations.

The author points out that in land installations alternating current is now used in about 90 per cent. of all electric motors built, a.c. motors, especially those of the short-circuited rotor type, having established themselves, apart from other considerations, because of their very simple arrangement and of the reduced amount of switchgear which permits of direct throwing-in without the use of a starter. In marine installations the successful application of alternating current in turbo- and Diesel-electric propulsion opens up further possibilities of a more extensive adoption of a.c. motors as, particularly in Diesel-electric vessels, alternating current for the auxiliaries and for lifting and heating can be obtained by means of stationary transformers from the main generators, in which it is produced at a low consumption of fuel. The author discusses the advantages associated with the use of alternating current for auxiliary purposes, viz. its easy transformability to any desired voltage by means of simple and stationary apparatus which is not subject to wear and tear and does not require constant attendance, the complete absence of any compass disturbance, as there are no high tension lines, and the reduced disturbance of wireless reception owing to the absence of commutators on the generators and the motors. A comparison of weights and costs indicates that a.c. motors of the short-circuited rotor type are substantially lighter and cheaper than drip-water protected d.c. motors. Thus, for an output of 5kW. at 1,000 r.p.m. the a.c. unit is 30 per cent. lighter than the d.c. unit, and as its price per unit of weight is also 25 per cent. less, the price of an a.c. motor of this size will only be 53 per cent. of that of the corresponding d.c. motor. In vessels equipped with a.c. installations the adoption of a.c. motors of the short-circuited rotor type for the drive of as many auxiliaries as possible is therefore indicated. Attention must, however, be given to the circumstance that the speed of these motors is definitely fixed by the number of cycles of the supply system together with the number of the poles. Thus, at 50 cycles per sec. the no-load speed will be 3,000 r.p.m. for motors having two poles, 1,500 r.p.m. for four poles, 1,000 r.p.m. for six poles, and so forth, the full-load r.p.m. being about 2 to 5 per cent. less on account of the slip, which is approximately proportional to the load. At heavy overloads the so-called "tilting moment" will ultimately be reached, when the motor will stop. It follows that the speed of a.c. motors of this type is limited to a very narrow range and that no subsequent adjustment by the insertion of resistances, as in d.c. shunt motors for pump and fan drives, is possible; nor can such a.c. motors be constructed for any given speed. Thus at 50 cycles it would not be possible to supply a 2,000 r.p.m. unit; at full-load this would have to be designed either for 2,900 r.p.m. with two poles or for 1,450 r.p.m. with four poles. As the output of pumps and fans can thus not be increased by a slight increase of the revolutions, such pumps and fans must be carefully dimensioned to give an adequate output at the exact motor speed, throttling being the only means of regulation. The author points out, however, that apart from the possibility of gearing the simple a.c. motor of the short-circuited rotor type to driven units having different revolutions, the adoption of a.c. current does in itself preclude speed governing. This can be obtained by employing a.c. motors with changeable pole connections, a.c. motors of the slip-ring type, a.c. commutator motors, variable speed a.c. sets, by the Ward-Leonard system, or by means of grid controlled mercury vapour rectifiers, the principal design features and possible applications on board ship of which he discusses in detail.—*Dr. Ing. J. Bahl, "Schiffbau", 17th November, 1937, p. 377.*

Recent Experiments in Connection with the Spraying of Metal.

The author observes that the spraying of the various carbon steels, and also, more recently, of alloy steels for the purpose of re-surfacing and building up worn machine parts has now been accepted as standard practice in many engineering works, and he describes some experiments conducted with a view to improving the quality of deposits made in this kind of work when using the wire-fed type of pistol. In this process each sprayed particle is enveloped by a skin of oxide, which is to a large extent responsible for preventing the resolution of the sprayed mass into a normal structure, thus causing the deposit to be of poor quality metallurgically. In addition the oxide content of the sprayed steel artificially hardens it to the point of brittleness and tends to produce low tensile strength. The spraying method was, therefore, examined experimentally in its entirety from the nozzle functions of the pistol to the actual deposition of the "atomised" metal, in order to ascertain where oxidation occurs and what benefits could be expected from its elimination. In summarising the results obtained, the author states that the main cause of oxidation is the air blast at the nozzle and that beneficial results may be expected if dissolved acetylene is used as the fuel gas in conjunction with an inert gas or de-oxidised air as the impelling medium coupled with suitable heat treatment after spraying. Thus, if the oxide inclusion in the deposit is sufficiently reduced, diffusion of carbon from the core to the coating will not be impeded and all traces of the junction between the two metals will disappear. Cast iron, therefore, when coated with a steel deposit and heat treated should give so much carbon to the steel as to make it extremely hard without further treatment, thus providing a means of covering frictional or the bearing surfaces with a hard and homogenous layer of wear resisting metal.—*Richard R. Sillifant, "Engineering", 5th November, 1937, pp. 526-527.*

Power and Consumption Measurements on the Machinery of a 250 i.h.p. Steam Tug.

The author gives detailed particulars of the steaming and coal consumption trials of a tug, fitted with a compound engine exhausting to the atmosphere, in which an exhaust steam heated evaporator of the Meyer and Oestreich system was added to the existing installation for the purpose of distilling the boiler feed water, and of thus eliminating the formation of scale and deposits of mud associated with the direct feed of river water in non-condensing or jet-condensing installations. In the Meyer and Oestreich system the quantity of exhaust steam admitted to the funnel is limited to about 40 per cent. or less, the amount required to maintain sufficient draught in the boiler. The remaining exhaust steam of 19 to 22lb. per sq. in. pressure is utilized as heating steam in an evaporator working in conjunction with a superimposed condenser, the cooling water of which in turn forms the supply to the evaporator. As 1·25lb. of exhaust steam will generate 1lb. of steam in the evaporator the condensate of the heating steam will in combination with the condensate obtained from the evaporator steam yield sufficient distillate to ensure a pure feed water supply of 203° to 208° F. The tug on which the trials were carried out was a vessel of 69ft. length and 18ft. beam by 6ft. 11in. draft, built in 1911 and fitted with a 14in. by 23in. by 15in. stroke compound engine capable of developing about 250 i.h.p., together with single furnace cylindrical boiler of 860 sq. ft. heating surface and 17·2 sq. ft. grate area. The measurements, of which detailed particulars are reproduced in the form of tables and graphs, were taken on three short runs of about 20 mins. each and on a five hours steam and coal consumption trial, using coal of about 13,000 to 13,500 B.Th.U.'s per lb. The results obtained indicated a steam consumption of 10·35kg. (22·8lb.) per i.h.p. per hour together with a steam consumption of 8·64lb. of steam per lb. of coal, corresponding with a coal consumption of 2·69lb. per i.h.p. per hour, and the author estimates on the basis of these results that a steam consumption of 9·9kg. (21·8lb.) per i.h.p. per hour and a coal consumption of 1·145kg. (2·52lb.) per i.h.p. per hour would be obtained with modern machinery of the same type. *—Dipl. Ing. K. Wetjen, "Schiffbau", 17th November, 1937, p. 381.*

The Cathode Ray Indicator in Service.

The writer describes the Standard-Sunbury Cathode Ray Indicator and discusses its employment in the works of the National Gas and Oil Engine Co., Ltd. In this instrument, which is primarily used for experiments and research, the indicating element is an evacuated glass bulb—the cathode tube—at one end of which a heated filament throws off the cathode ray, i.e. a stream of electrons which is focussed by a control-grid through a small hole of an anode. Issuing from the hole in the anode as a fine beam, this travels down the tube and produces a spot of light on the fluorescent coating on the inside of its end wall. The beam, which is invisible, together with the spot of light can be deflected horizontally and vertically by applying an electric potential to guide plates arranged in the tube. If then the spot of light is made to travel horizontally at a speed corresponding with the engine speed by means of a contact breaker on the crankshaft, and vertically in correspondence with the cylinder pressure, the result will be an "indicator card" on a time base. For the latter purpose the Sunbury indicator is fitted with an electromagnetic pick-up. This instrument contains a diaphragm communicating with the cylinder, the deflection of which by the pressure induces an electrical impulse in a pole piece with surrounding coil which is amplified and applied to the deflecting plates. The magnitude of the impulse depends on the speed at which the diaphragm is deflected, i.e. on the rate of change of the cylinder pressure, not on the pressure itself. To obtain deflection of the cathode ray proportional to pressure the impulses must therefore be integrated by means of an electrical circuit in the amplifier. Either diagrams of pressure or rate of change of pressure can, however, be thrown on the screen. The instrument has practically no inertia lag and it is accurate up to revolutions far in excess of any engine speed possible. The diagrams are diagrams of single cycles, so that variations between adjacent cycles can be observed, and as indication is continuous the effect of variations can be observed while the engine is running. Pressure variations down to a few inches of mercury can be recorded by the use of suitable diagrams and any part of the cycle can be examined by magnifying the vertical or horizontal scales (electrically). Finally, exact timing of any phenomena can be obtained, so that the instrument is now used by fuel experts for the determination of the delay periods of different oils.—*"Gas and Oil Power", November, 1937, p. 271.*

EXTRACTS.

The Council are indebted to the respective Journals for permission to reprint the following extracts and for the loan of the various blocks.

Water Circulation in Boilers.

"The Engineer", October 8th, 1937.

The designer of water-tube boilers never admits any doubt as to the way the water circulates inside them. He will, if you ask him, be pleased to draw little arrows on the tubes pointing the way for the water, and he will add other curly ones here and there for the guidance of the flue gases. Everything is then perfectly clear. Down goes the water from the drum by these tubes, across by those, and up by the others, round and round like the music and with equal persistence. The scheme would be convincing if only we could be sure that the water knew what the arrows meant. For the flue gases this does not matter, because they are bound to travel from the furnace to the stack by the only path open to them. They are not required to circulate; a single passage and their work is done. Circulation of the water, however, is essential to the proper working of an ordinary boiler. Not only is it conducive to rapid transmission of heat from the tubes to the water, but dead water in any tube is almost sure to lead to trouble. When stagnation does occur it is not necessarily because the water has no means of escape. It could leave freely by either end of the tube, but if the conditions are such that there is no inducement for it to prefer one direction of flow to the other, it just stays where it is. In a thick bank of tubes, for example, when those nearest the fire are designed to act as risers and those on the far side as downcomers, the water in the central rows may be doubtful as to which way to go, and consequently may be reluctant to move at all. The same state of affairs may be found in the intermediate tubes of a bank intended by the designer to consist of risers only, for it is common knowledge that on such banks there may be at times a reverse flow in the cooler tubes, owing to a circulation being set up within the bank itself. The evidence of defective circulation is severe pitting in the guilty tubes. Whatever the actual cause of the pitting may be, there is no doubt that in many types of boiler an experienced engineer can foretell almost exactly which rows of tubes are likely to suffer most from pitting, and it will always be found that these are the rows where inadequate circulation may be expected.

The causes of natural circulation are very simple and are, of course, well understood. The cooler portions of the water, being denser, tend to sink, displacing the more highly heated portions, the up-flow of which is considerably aided by the bubbles of the steam entangled with them. The action has been demonstrated by a thousand models and the mental picture of the process is always in the background of the designer's mind. But a glass model, working at or about atmospheric pressure, is a very different affair from an actual boiler. The one thing common to both, which is important, although not directly obvious, is that the rate of circulation is kept in check by its own consequences. Every increase in its speed tends to make the temperature of the contents of the boiler more uniform, while every approach to uniformity of temperature diminishes the forces on which circulation depends. There is thus a limit to the rate of natural circulation in any boiler, and this will be the lower the more nearly every part of the boiler is fulfilling its real function of producing steam rather than heating water. But there is another factor antagonistic to a brisk circulation which is becoming more evident than formerly. A modern power station boiler is often supplied with feed water preheated to a very high temperature. In the days when boilers worked at about 200lb., and were fed with water at 212° F. or so, the temperature range in the boiler may have been nearly 180° F. In a modern boiler working at 600lb. pressure and supplied with feed water at 340°, the temperature range will only be 150°, with a corresponding reduction in the forces tending to promote natural circulation. More significant, however, is the effect of the increase in pressure. The bubbles constituting 1lb. of steam at 600lb. pressure will displace less than 40 per cent. of the water displaced by the bubbles of steam at 200lb. pressure, and their effect in causing circulation will be reduced in the same proportion. With boiler pressures of 1,200lb. and over, which are now common in the United States, the bubbles will displace only about half as much water as at 600lb., so that their effect is again reduced by half.

It would appear, therefore, as if the tendency of modern boiler practice was to reduce those factors which make for brisk circulation, while demanding at the same time average evaporation rates three or four times as great as engineers used to be contented with. Boilers are also becoming highly complex structures, with water tubes all round the furnace, and slag screens in addition, imposed on the original tube systems. When tubes exposed to furnace temperature are expected to evaporate 50lb. or more of water per sq. ft. per hour, it is obvious that a proper circulation must be maintained through them, and circumstances are combining to make it more and more difficult to ensure this by natural means. It is hardly surprising therefore to find schemes for producing a forced circulation through the tube system attracting the attention of many designers. The idea is by no means a new one. In the 'sixties of the last century quite a crop of forced circulation water-tube boilers was developed in the United States. The period was one of great activity in the development of water-tube boilers, and many of the designs were such that the water could hardly be expected to traverse the many sharp corners provided in its path unless it were positively pushed along. It may also be significant that the practice of pumping the water round the boiler coincided with a general increase on steam pressures, such as we are again witnessing to-day. But it soon fell into disuse, being found a needless complication when the boilers were so designed as to make natural circulation more easy. Whether history will repeat itself in this respect it is impossible to say. No one would want a circulating pump attached to his boiler if he thought he could do without one, for though the power it absorbed might not be worth considering, the need for its continuous operation at full boiler pressure and temperature would not add to the charms of the boiler house, even if it did not detract from the reliability of the plant. It is true that there are certain boilers now in use or in course of development in which there is no circulation at all in the true sense of the word, the water passing once only through the tube system as it does in an economiser. In such boilers the movement of the water is, of course, maintained by the feed pumps. They should be free from any difficulties due to stagnant water, but experience only will show whether a final solution is to be found in this direction. At present the overwhelming majority of boilers in service depend upon internal circulation, and the question is whether in the future this will be kept up by a natural or artificial means. If the tube system, including the water walls, can be so simplified that an adequate and continuous flow through all tubes can be assured without recourse to excessively high drums, there is little doubt that this is the better plan. Otherwise, the alternative of pumping the water round will have to be adopted.

Smoke Indicators.

"Shipbuilding and Shipping Record", October 14th, 1937.

One of the surest methods of estimating the efficiency of combustion in the boilers of a steamship is to study the nature of the flue gases emerging from the funnel. In addition, not only can it be affirmed that where there is black smoke there is unburned carbon, and therefore wasted fuel, but particularly on passenger ships, this unburned carbon may be deposited on the decks causing considerable annoyance. Once the boilers are thoroughly under way, the elimination of smoke is largely a matter of controlling the draught produced by the fans, but unfortunately the reading of the draught gauge is not enough to tell the engineer below the conditions at the funnel outlet, since the direction and strength of the wind has a profound influence on the draught. It is apparent, therefore, that some form of smoke indicator is desirable as a means of giving visual information of the condition of the gases issuing from the funnel. Indicators of this type are now available, their action being based upon the variation in the amount of light from a small electric lamp passing across the uptake, which varies the discharge from a selenium cell. This in turn operates the needle of the instrument which ranges from clear shown as white on the dial to thick smoke shown in black with the intermediate position shown by a gradation between the two. A recording instrument gives the smoke condition throughout the watch.

Elastic Diesel Engine Foundations.

"Shipbuilding and Shipping Record", October 28th, 1937.

The design of the foundations for the diesel engines installed on board ship, both for propulsion purposes and for driving the auxiliary generators, is a problem which has formed the subject of much research. In the early days of the marine diesel engine it came to be regarded as one of the most difficult problems with which the designer was faced, and if, to-day, the problem has lost something of its urgency, it is due to the fact that better balanced engines are employed and that these are usually carried on foundations of extraordinary strength and rigidity, rather than that the problem of designing a theoretically perfect foundation has been satisfactorily solved. It may be noted in passing that this problem is almost of equal urgency with the land engineer, particularly in view of the wider adoption of the high-speed diesel engine for driving electrical generators. Hence, the paper by Dr.-Ing. Kurt Klopstock, which was read on Thursday last, before the Diesel Engine Users' Association, aroused considerable interest. Taking as his subject "The Design of Elastically-Supported Foundations for Reciprocating Engines", the author gave credit at the outset to the novel researches of Dr.-Ing. H. Hartz, upon which the paper was based, and briefly it may be stated that the fundamental idea involved is the employment of a foundation which, while rigid in itself, is supported in such a way that its movement balances the movements of the engine. In this way it is claimed that the oscillating foundation solves the problem of preventing the transmission of perceptible disturbances to the surroundings. Emphasis is, however, laid upon the fact that this is an entirely different principle to the interposition of an elastic shock-absorbing medium between a rigid foundation and ship's hull or the building as the case may be.

It is known that in any system of moving masses, the centre of gravity tends to remain stationary. Hence, if a piston of mass m_1 moves through a distance r_1, then the engine frame, together with its foundation of total mass m_2, should be free to move through a distance r_2 such that $m_1 r_1 = m_2 r_2$, the movements of m_1 and m_2 being in counter-phase. The same law applies to moments as well as to forces, so that if the moments of inertia of the two masses m_1 and m_2 are I_1 and I_2, then the angular displacements must be ϕ_1 and ϕ_2 such that $I_1\phi_1 = I_2\phi_2$. With solid or absolutely rigid foundations the centre of gravity cannot remain fixed, but undergoes periodic displacements for which force is required, and this is noticeable as vibrations in the vicinity of the foundation. If, on the other hand, the entire system is "freely floating", there would be no such disturbances. The calculation of the required total foundation mass and inertia m_2 and I_2 depends upon fixing the maximum permissible movement of the foundation, and it is suggested that a translatory amplitude of 0·15mm. and a rotatory angular amplitude of 0·0001 (radian) can be permitted. In this way the total foundation weight, including the framing, etc., of the engine, can be determined. The author then gives a series of curves, from which, what he terms, the tranquillizing mass can be determined for different values of the forces and moments producing vibrations in the engine.

Finally, the question arises as to the manner in which the suspension of the total foundation weight must be made in order to approach the ideal condition of a freely-floating system. Here the problem becomes one of the application of the laws of oscillation, and as the mathematical treatment is very involved, the results only are given in the paper. It is found that neither cork nor india-rubber can be employed to give the desired freedom to the foundation, but only a system of springs so designed as to give six degrees of freedom, viz., translation along three axes and torsional movement around these axes. The desired system of suspension has been patented and it has been applied to a number of installations with marked success. It has already been mentioned that the problem of engine vibration applies to both land and marine engines, and the majority of the installations referred to by the author were in conjunction with engines in difficult situations on land, such as adjacent to other machinery, in a residential area, and on the upper floor of a building. Particulars are, however, given of the adoption of the elastic foundation system to the main engines of a shallow draught motorship. In this instance the Voith-Schneider system of propulsion was adopted with twin propellers, so that the engine speeds and propeller speeds were different, while the ship's hull in its different parts had various natural frequencies, mostly low. Moreover, since a hull of this type is extremely sensitive towards vibration influences, the test of the elastic foundation system was a very severe one. The arrangement has the merit of being relatively simple.

Metal-sprayed Journals.

"Shipbuilding and Shipping Record", October 28th, 1937.

In a paper read before the Association of Metal Sprayers in Reynolds Hall, Manchester, on 20th October, by Harry Shaw, some interesting facts were given regarding the building up of worn bearings by metal spraying.

Tests carried out by the author to investigate the frictional and load-carrying properties of metal-sprayed steel shafts running in white-metal bearings on a friction testing machine showed that the coefficient of friction of the sprayed steel is lower than that of the ordinary steel at all loads (except below 400lb. at 261ft. per minute) by amounts varying from 5 to 30 per cent. The addition of colloidal graphite to the oil still further reduces the coefficient by similar amounts.

Despite the increased load capacity and reduced friction of the sprayed metal in the laboratory tests it was thought that the sprayed metal might not stand up under conditions such as exist in an engine where loads are erratic and pulsating, so tests were carried out on crankshafts from 22 different engines, of which five were private car engines.

The crankshafts were reduced on half the journals to about $\frac{1}{8}$in. below size, built up by metal spraying with steel, and finished to size, this giving a sprayed metal thickness of about $\frac{1}{16}$in. One crankshaft from each class of engine was given a thickness of sprayed metal of only $\frac{1}{32}$in. Checks of the wear on the metal-sprayed journals and on the ordinary journals were made over running periods of 1,000, 5,000, 10,000 and 15,000 miles and in no case did the wear on the sprayed journals exceed that on the ordinary journals. The difference in the thickness of metal spraying of from $\frac{1}{32}$in. to $\frac{1}{16}$in. appeared to have no influence on the wear.

On the private car engines, commercial vehicle engines and petrol bus engines the wear on the metal-sprayed journals was, on an average 15 per cent. less than on ordinary journals, the vehicles used for the greatest stop-and-start work showing the metal-sprayed surfaces up to greatest advantage. In the diesel bus engines the wear on the metal-sprayed journals was but 50 per cent. of that on the plain journals.

The most remarkable effect of the metal-sprayed journals was that the wear on the contacting bearings was much less than on those contacting with the plain journals.

A Dynamometer Barge.

"The Engineer", 29th October, 1937.

In order to ascertain the towing power of tugboats when actually in service, the Dravo Corporation, a United States shipbuilding company, has built a dynamometer barge. The barge, which is placed between the tow and the tug, is a standard 100ft. by 60ft. river craft, and on its deck is fitted a welded steel frame riding on rollers which give free movement fore and aft, but not athwartship. Four sets of rollers are provided at each of the three bearing points which support the floating frame. The top and bottom units consist of two rollers and the side units of three rollers. The frame has a width of 20ft. from centre to centre of bearing supports, which are at the extreme end of the barge. Adjacent to and overhung from these bearing supports is a heavy beam, by which the tug is attached to the frame. The tugboat does not come into contact with the dynamometer barge, its entire thrust or pull being exerted on the frame which operates two hydraulic rams, each of which has an area of 100 square inches. The two rams are placed so as to take the thrust or pull in either the forward or astern direction. These register the amount of thrust or pull upon recording and indicating gauges provided for each ram and located in a cabin set between the rams.

A Bow Propulsion Vessel.

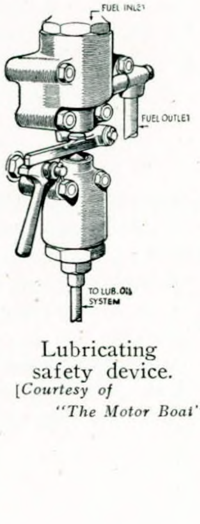
"The Engineer", 29th October, 1937.

What is claimed to be the first vessel to be fitted with bow propulsion has been built for canal service in Germany by the Mannheimer Schiffs und Maschinenbau. As described in "Shipbuilding and Shipping Record" propulsion is effected by propellers mounted at each side of the bow in Kort nozzles with the object of suppressing the bow wave and resultant wash which necessitate strict limitation of speed on canals in order to avoid flooding and scouring of the banks. Driven, by two 135 h.p. oil engines, the ship has a length of 220ft. 4in. and a breadth of just under 27ft. During trials in the Neckar Canal at a speed of 12 kiloms. per hour, it is reported that not a wave was produced by the boat.

A Safety Device.

"Petter's Monthly News", October 15th, 1937.

The small sketch illustrates a device used in a Petter engine for automatically stopping the engine in the remote eventuality of the supply of lubricating oil failing. The lower chamber houses a plunger which is forced up by the pressure of the lubricating oil. This plunger holds open, against the pressure of a spring, a valve contained in the upper chamber and allows the fuel oil to pass to the engine. Should the lubricating oil pressure fail the plunger would immediately fall and the valve in the upper chamber would be closed by the spring pressure. This would cut off the fuel and



Lubricating safety device.
[*Courtesy of "The Motor Boat"*

cause the engine to stop before any damage was done to the bearings.

The lever and arm shown on the left are for use when starting the engine. As soon as the lubricating oil pressure reaches normal the lever falls, and thus sets the safety device.

Boiler Water Problems.

"The Syren and Shipping", November 24th, 1937.

On Friday, 12th November, an address was given to the members of the South Wales branch of the Society of Consulting Marine Engineers and Ship Surveyors by Mr. J. S. Merry, A.I.C., the subject being "Boiler Water Problems". The meeting was held at the South Wales Institute of Engineers, Park Place, Cardiff. Some examples of corroded boiler plates and tubes were exhibited, together with various scale formations. The lecturer illustrated by means of lantern slides the different forms of corrosion and scaling met with in various types of marine and land boilers, and also showed charts indicating the scale and acid-forming salts and dissolved gases contained in the waters commonly used in boilers. He demonstrated by chemical experiment how it is possible to treat the feed water by a colloidal process which neutralises and deposits the harmful impurities in the form of a sludge that cannot form scale on the heating surfaces. Another form of corrosion dealt with was that due to electro-chemical action, and it was shown experimentally how easy it is for the condition to arise in a boiler for this action to take place, with consequent wastage of the electro-positive surfaces. He showed that when two similar pieces of metal are immersed in a weak salt solution no electrical action takes place, but if one of the plates is coated with a wash of lime scale a difference of potential of 20/30 millivolts is set up, causing a flow of current from the coated to the clean plate and thereby causing corrosion of the coated plate. This explains why it is very often found on removing scale from a boiler plate that the surface underneath is corroded and soft. Another experiment was made with two exactly similar pieces of boiler tube, one clean and slightly polished and the other with a film of rust on the surface. When these were immersed in the salt solution an electrical action was set up, causing damage to the rusted tube.

The Apprentice Question.

"The Syren and Shipping", November 17th, 1937.

The negotiations on the engineering side in connection with the claim of the trade unions to negotiate for apprentices was carried a stage further last week, when the engineering employers and representatives of the joint trade union engineering movement met in London to consider the proposal in greater detail. It will be remembered that a month ago the employers agreed in principle to concede the right of the unions to act for apprentices, but this concession was stated to be subject to such reservations of machinery and scope as might be mutually agreed upon. This left some doubt as to how much the employers had actually conceded and it was partly to consider what reservations are to be made that the parties met again. The meeting was private and a statement issued afterwards merely said that good progress had been made. At any rate, after a conference which lasted some three hours the talks were adjourned for about a fortnight, and we are still left in doubt as to the scope of the agreement which may ultimately be arrived at. Meantime the employers are to meet the Amalgamated Engineering Union to-day (Wednesday) on the same subject. This separate meeting is necessary because of the withdrawal of the A.E.U. from connection with the joint engineering trades movement, a state of affairs which seems rather unfortunate, as it cannot but be more difficult to negotiate separately with two individual bodies on the same points and go over all the ground twice.

Performance of the Reheater Engine.

"The Marine Engineer", November, 1937.

It may be recalled that some months ago we described and illustrated the North Eastern reheater engine—one of the most important technical developments in the marine steam engineering field to be announced in recent years. The two vessels in which this reheater machinery was then fitted, with the latest type of North Eastern superheater in the boilers, are the Chambers-owned vessels "Lowther Castle" and "Lancaster Castle", single-screw cargo ships of about 9,300 tons deadweight. In view of the considerable technical interest attaching to this development, particulars of the service behaviour of the "Lancaster Castle" will be of general interest.

The attached data relates to a 12,000-mile round voyage, fully loaded in both directions, which the vessel has recently completed, and shows that something like 600 tons of cargo can be carried per day at a speed of 10 knots for the consumption of one ton of coal. A statistically-minded correspondent advises us—and we see no reason to dispute the conclusion—"that the corresponding performance of an ordinary motorcar carrying four average passengers would be 1,850 miles per gallon". The average speed on the outward voyage was 9·95 knots on an all-purposes daily coal consumption of 15·3 tons, which is little more than half of what would have been considered a good performance a few years ago. During the outward voyage a series of 24 trials was run at various powers from half to full normal output, when the following data were obtained :—

Ship's speed, knots	8·7	10·12	10·76	11·39
R.P.M.	52	58	62·5	66·5
Apparent slip, per cent. ...	5·5	2	6·6	3·6
I.H.P.	903	1220	1510	1790
All-purpose coal per day, tons	9·4	13·3	17·5	19·6
All-purpose coal per i.h.p., lb.	0·97	1·01	1·08	1·02

Copper-alloy Cylinder Heads.

"Shipbuilding and Shipping Record", November 25th, 1937.

It is well known that copper possesses a very high thermal conductivity, and it follows that the use of this material for the construction of the cylinder heads of high-compression internal-combustion engines would lead to very efficient cooling with all the advantages which that entails. In a paper presented to the Institution of Automobile Engineers which has recently been issued as a pamphlet, the author mentions the fact that these cylinder heads have been developed in the United States and have been found to be particularly suitable for the engines of high-speed motor boats. The alloy employed is pure copper with an admixture of 0·5 per cent. of chromium, and the result is a metal which possesses a tensile strength of 22 tons per sq. in. with an elongation of 25 per cent. and a Brinell hardness number 110. The material can be readily cast and has proved very suitable for the cylinder heads of petrol engines in which a high-compression ratio is employed. Tests have shown that this alloy has a thermal conductivity twice as great as that of high-tensile aluminium alloy and almost seven times as great as that of cast iron, and tests on similar engines fitted with cast-iron aluminium-alloy and copper-alloy cylinder heads showed that the petrol consumption per b.h.p. per hour was 0·74, 0·70 and 0·62lb., respectively, while the b.h.p. developed at a given engine speed increased in the ratio of 53, 58 and 64. Confirmatory tests in this country have indicated an all-round improvement in performance with copper-alloy heads.

Fishing Vessels With Kort Nozzles.

Performance results and savings effected by German craft.

"Shipbuilding and Shipping Record", November 18th, 1937.

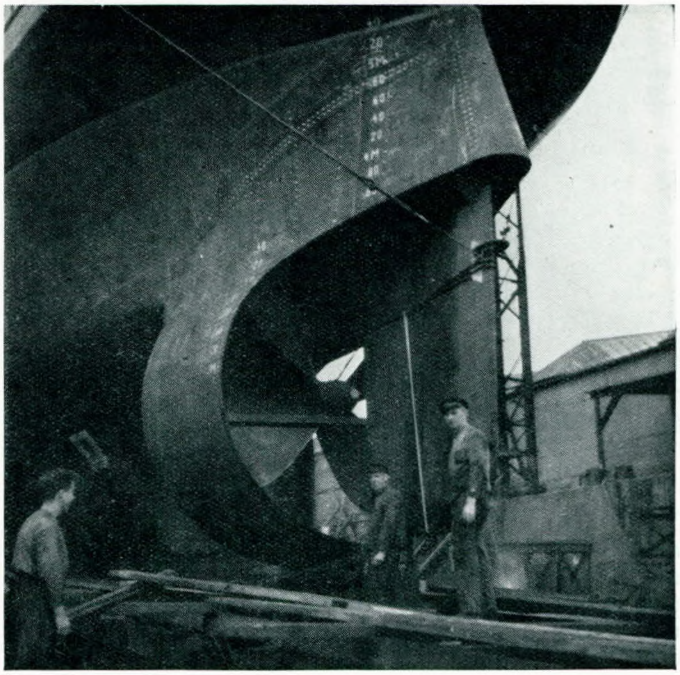
The deep-sea fishing vessels "Alemania", "Teutonia" and "N. Ebeling", built by the Howaldtswerke A.G. Hamburg, for the Hochseefischerei N. Ebeling, Bremerhaven, embody a number of innovations and are regarded as constituting a new and improved type likely to be well represented in the extensive programme of new constructions for German deep-sea fisheries. The principal particulars are:—

Length, b.p.	163ft. 6in.
Breadth	27ft. 3in.
Draught, mean	13ft. 11½in.
Propeller diameter	10ft. 4in.

These dimensions are normal for the 50m. (164ft.) class of vessel now so popular in Germany. The relation between speed and horse-power, for vessels without nozzles and in smooth water, as determined from model tests, is shown in Table I. Even with maintained high values of engine and propeller efficiency, the additional power (i.h.p.) for higher speed rises rapidly, as follows:—

Speed increase	Additional I.H.P.	Speed increase	Additional I.H.P.
8 to 9 knots	67	11 to 11½ knots	106
9 „ 10 „	104	11½ „ 12 „	196
10 „ 11 „	164	12 „ 12½ „	428

To increase the speed from 11½ to 12½ knots would necessitate practically doubling the horse-power. With this form of hull, 11½ knots is about the highest speed economically practicable in fine weather, and the available horse-power in excess of 630-650 i.h.p. should be regarded only as a reserve for bad weather conditions and as some assistance in racing to market, though it may be doubted whether the time casually saved on a day's or half a day's steaming by increasing speed from 11½ to 12 knots justifies the expenditure of an additional 250 to 350 i.h.p. In place of the usual installation of 900 to 1,000 i.h.p., the new vessels are fitted with a standard type of triple-expansion engine using superheated steam, but without exhaust turbines.



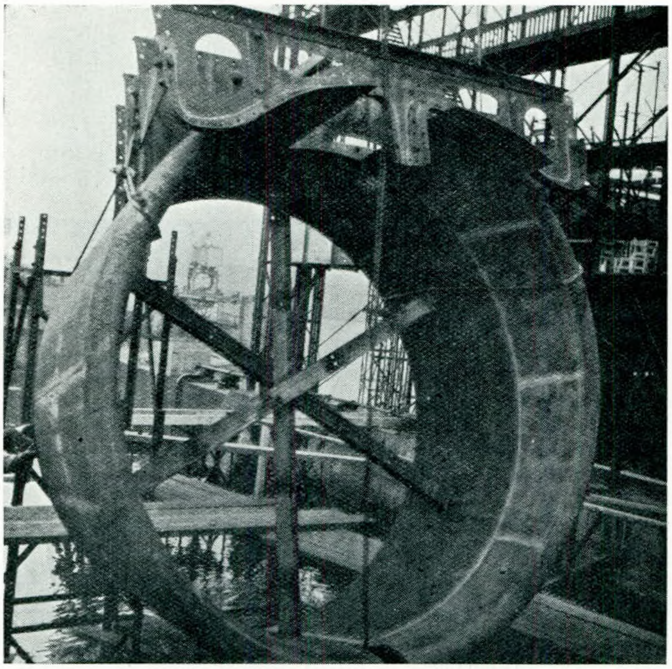
Stern of the "Alemania" showing Kort nozzle.

TABLE I.—SPEED AND HORSE-POWER OF FISHING VESSELS WITHOUT KORT NOZZLES, IN SMOOTH WATER.

Knots.	Effective H.P.*	S.H.P. (to propeller).	I.H.P. of propelling machinery	Propulsion efficiency (Effective H.P./S.H.P.) per cent.	Engine efficiency S.H.P./I.H.P. per cent.
8	92	125	215†	73·3	58·0
9	137	183	282†	74·8	65·0
10	201	271	386†	72·2	70·0
11	296	398	550‡	74·5	73·0
11·5	369	517	656‡	71·4	79·0
12	504	724	852‡	69·6	85·0
12·5	715	1,110	1,280†	64·4	87·0

* Net power required to move hull, without screw, at stated speeds.

† Estimated. ‡ Observed in the sister vessel "Ernst Flohr" without nozzle on the measured mile.



One of the nozzles ready for mounting.

The normal output is 700 to 750 i.h.p., with 58 per cent. cut-off; up to 850 i.h.p. is obtainable as a maximum. Instead of providing a reserve of 250 to 350 i.h.p. above the 650 i.h.p. good weather requirement, the policy adopted in the "Alemania" class vessels has been to provide 700 to 750 i.h.p. with Kort nozzles as the equivalent of a power reserve. This, says Dipl.-Ing. E. K. Roscher, in "Hansa", results in steam economy, simplified operation, and reduced constructional costs. The decision to use Kort nozzles was based on favourable experience with four of the owners' older vessels converted to nozzle propulsion. Better performance is to be expected from the new vessels, as the after part of the hull is designed in full conformity to the requirements of the nozzle. The performance of the three new vessels has already been so satisfactory that nozzle propulsion has been adopted in a further two constructions of the "Alemania" type.

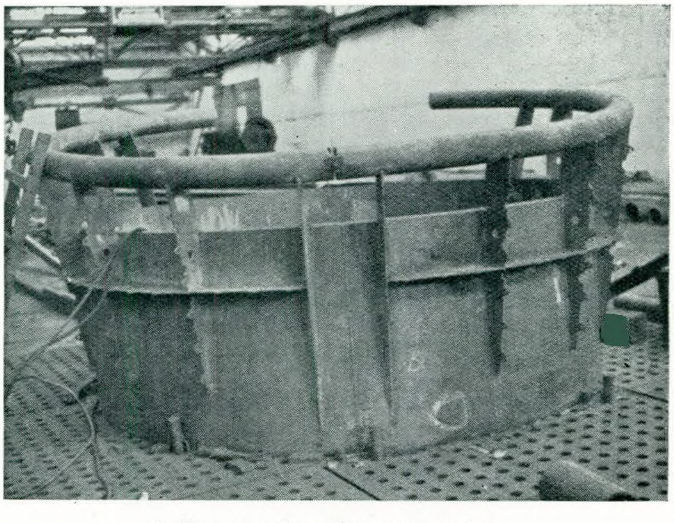
It is claimed that, apart from the 9 to 10 per cent. saving in power effected by the nozzle in good weather, and the satisfactory manœuvring characteristics ahead and astern, the new vessels are able to keep up with ordinary vessels of 950 to 1,000 i.h.p. under free-running conditions and for long voyages. Table II compares the measured-mile performance of the "Alemania" with that of the sister-vessel "Ernst Flohr" (without nozzle). Allowing only 4 to 5 per cent. more power for the additional form-resistance of the 10 per cent. higher displacement of the "Alemania", the figures in Table II justify the claim that the nozzle saves 9 to 10 per cent. in the fair-weather power requirements. Allowing the same engine efficiency (85 per cent.) and the same effective power requirement (504 h.p.) in both cases, as in Table I, the effective percentage of the s.h.p. is 76·4 per cent. in the "Alemania", compared with 69·6 per cent. in the "Ernst Flohr".

The results in Table II indicate a clear saving in favour of nozzle propulsion under smooth water conditions, but the major advantage makes itself felt when the ship is pitching in a seaway, i.e., when the ship's speed, due to an increase in resistance, decreases, and when, in the case of a normal open propeller, the direction of incident flow into the propeller blades continually changes, and when, due to both of the foregoing causes, the proportion of shaft horse-power converted to effective thrust may quickly fall from 70 to 30 or 20 per cent., or even lower. Of what use is it to install an additional 250 to 350 i.h.p., i.e., about 50 per cent. of the economic basic power, as a reserve for use in bad weather, when under bad weather conditions, just when such reserve is required so very little of it is transmitted into useful power?

When a Kort nozzle is fitted the normal effect is that, with an added resistance and a a greater slip, the reaction from the nozzle becomes greater, so that under conditions of bad weather there is an increase in thrust from the nozzle itself; in addition to this benefit there is the

TABLE II.—TRIAL DATA FOR FISHING VESSELS WITH AND WITHOUT KORT NOZZLES.

	"Alemania".	"Ernst Flohr" (without nozzle).
Draught, forward ...	11ft. 6½in.	11ft. 0in.
„ aft	16ft. 0in.	15ft. 6in.
„ mean ...	13ft. 9½in.	13ft. 3in.
Displacement	1,110 tons	1,000 tons
Vessel speed	12 knots	12 knots
R.p.m.	123·5	125
I.H.P.	812	852



A Kort nozzle under construction.

fact that when the vessel is pitching the correct direction of incident flow to the propeller, i.e., parallel to the shaft, is maintained as indicated by the drawing below. The net result is a higher voyage-speed and much better towing force, for given power and fuel consumption.

Tests at the Hamburg Tank with a model of a 164ft. fishing vessel with a Kort nozzle closely resembling that of the "Alemania", show that under the most unfavourable conditions, with wave length equal to hull length, 300 s.h.p. (350 i.h.p.) maintains the same vessel speed as 700 s.h.p. (830 i.h.p.) where no nozzle is used. All practical cases must lie between this extreme of 100 per cent. gain in thrust, and the 10 per cent. gain under smooth water conditions. Thus, 30 per cent. is a conservative estimate for the equivalent reserve of power obtained from the Kort nozzle at no additional operating cost.

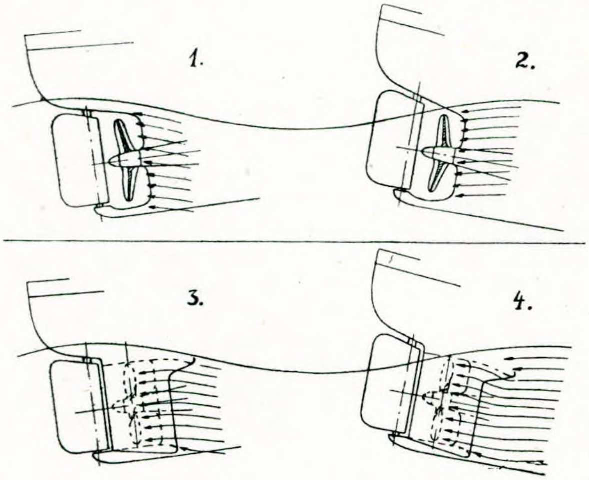


Diagram showing water flow to propeller when vessel is pitching. Figs. 1 and 2 without nozzle and Figs. 3 and 4 with Kort nozzle.

About 500 to 600 i.h.p. is required to tow at 3½ knots the nets now used by German vessels of the 164ft. class without nozzles. On the other hand, tests with the "Jeverland", which uses a Kort nozzle, show that only 330 i.h.p. is required to tow nets at 3½ knots with a pull of 4·5 to 4·6 tons. The saving of 200 i.h.p. corresponds to about 265lb. coal per hour, and this—for four months' herring fishing with 400 hours' towing per month, and seven months' fresh fishing with 135 hours' towing a month—effects a saving of 300 tons of coal per annum, worth, say, £270. The 30 per cent. gain in power when running free amounts to a further saving of about 200 i.h.p. for, say, 160 days per annum, worth another £400 or so. It is claimed that with regard to towing these estimated savings are confirmed by the savings actually effected by more than 200 nozzle-tugs now in service, and the performance of the 750 i.h.p. "Alemania"-class vessels to date indicates that they will keep pace with 950-1,000 i.h.p. vessels of the same length without nozzles under all circumstances, and outpace them in bad weather.

Marine Engineers' Examinations.

"Shipbuilding and Shipping Record", November 18th, 1937.

The Departmental Committee set up by the Board of Trade to inquire into the existing methods of conducting the examinations for marine engineers' certificates have now concluded their labours and presented a report for the consideration of the Board. It is understood that this report will be published shortly. The Committee was set up as the result of representations that the examinations for second and chief engineer were of such a nature as to act as a deterrent to those who might seek the avocation of marine engineering. We hear that the Committee have been able to devise a scheme which, without lowering the standard of the examination, should remove some of the obstacles which have resulted in such a high percentage of rejections. This proportion, we believe, has been rather formidable in recent years, and the Committee, in their investigations, have come to some surprising conclusions on this aspect of their investigations, which will, no doubt, appear in the report. We hope that the evidence on which they based these conclusions will also be available in some form, because we believe it will be found that it dissipates allegations of the harshness of the examiners and will show where either the educational system is at fault or the Board of Trade regulations need some relaxation. There are some interesting innovations in the proposals involving questions of principle which the Board of Trade may, or may not, see fit to adopt. Meanwhile, we await the publication of the details with interest.

Vacuum-operated Filters.

"Shipbuilding and Shipping Record", November 11th, 1937.

The flow of a fluid depends primarily upon a pressure difference, and it follows therefore that a vacuum on the delivery side is just as capable of operating a filter as is a pressure on the input side. The vacuum-operated lubricating oil filter has certain advantages over other types and a well-known British firm specialising in the design of filters has produced a range of units operated upon this principle in which a vacuum pump operated by an electric motor is used to draw the dirty oil through the filter and to deliver it to a clean oil receiver from which it can be drawn off as required. Electric immersion heaters are employed to reduce the viscosity of the oil, these being located in a water jacket surrounding the filter, although steam heating can be arranged for if desired. The vacuum pump is also designed to supply compressed air, for cleaning the filter, two movements of a control handle being sufficient to first of all drain the filter while the air pressure builds up in an air bottle carried at the back of the unit, and second, to release the compressed air through the filter packs, discharging the sludge through a cock provided for the purpose. In the smaller units, the apparatus, including dirty and clean oil tanks, is entirely self-contained, but in the larger sizes a separate dirty oil tank for mounting on the bulkhead is supplied thus giving a gravity feed to the filter itself.

Contact Corrosion.

"Shipbuilding and Shipping Record", November 11th, 1937.

The use of high-pressure and high-temperature steam on board ship has introduced a number of difficult problems, notably in connection with the design and maintenance of the various flanged and bolted connections on the pipe lines and the fittings. These problems, it may be recalled, are under investigation at the National Physical Laboratory in connection with an extensive research being undertaken on behalf of the Pipe Flange Research Committee of the Institution of Mechanical Engineers, and attention was drawn to the progress of this investigation in the recent annual report of the N.P.L. In particular, attention was drawn to the phenomenon of contact corrosion which takes place between materials held in surface contact under local pressure. We recently had an opportunity of studying the conditions on a liner equipped with high-pressure, high-temperature steam generators, and found that a very fertile source of trouble was the various pipe joints. It would appear as though contact corrosion between the threads of bolts and nuts is very liable to occur, since not only does the slight relative movement tend to cause a gradual leakage at the joint, but it is found that disintegration of the material of the bolts and nuts occurs at the surfaces of contact. It is often a matter of extreme difficulty to disconnect the bolts and nuts, although various patented solutions are available, the use of which may permit of an easier separation.

Scaling Evaporator Tubes.

"Shipbuilding and Shipping Record", November 11th, 1937.

In modern steam installations on board ship it is not considered sufficient that fresh water, as obtained from the main supply at the ports of call, should be used for feed make-up purposes, but this water is first of all evaporated in order to eliminate, as far as possible, all scale-forming substances, before it passes to the boilers. As a consequence, the steam coils in the evaporators are liable to become rapidly coated with a hard scale and it becomes a problem of some importance to remove this in order that the efficiency of the evaporator shall be maintained. What is known as "flashing" the evaporator, i.e., subjecting the tubes to a rapid change of temperature by heating with steam and then cooling with fresh water, has the effect of cracking the scale due to the expansion and contraction of the tube so that it falls to the bottom of the evaporator casing, whence it can be easily removed when opening up. Unfortunately, however, the process of flashing gradually tends to become ineffective due to the hardening of the metal. When this stage has been reached, it is necessary to remove the scale by hand and then carefully soften the tube by annealing before replacing it. On the argument that prevention is better than cure, there are certain compounds available with which the surface of the tube can be coated and upon which the scale does not deposit. When these are used it should be remembered that the coating gradually disappears and periodical recoating is required.

Lubrication Research.

"The Engineer", October 22nd, 1937.

Of all the natural forces with which the engineer has to deal friction is at once the most commonplace, one of the most important from the practical standpoint and one of the least completely understood scientifically. After generations of study we can still only guess at its cause, and the "laws" which it obeys are still not fully and accurately defined. Morin's laws of "solid" friction, set forth with such assurance in the older textbooks, are purely empirical approximations and are employed more for their convenience than for their accuracy. It is in fact no exaggeration to say that "solid" friction of the kind which Morin imagined he was measuring when he formulated his laws never occurs in Nature as the mechanical engineer knows it. Some degree of lubrication is always present. As soon as a lubricant is deliberately introduced the complexities of the subject become greatly enhanced. On top of the uncertainties concerning friction, as such and by itself, there are added fresh uncertainties regarding the action of the lubricant. Modern developments in the direction of heavily-loaded bearings, running at high speeds and sometimes—as in steam turbines and internal combustion engines—at high temperatures have augmented the complexities of the subject still further by imposing upon the lubricant the duty of acting as a cooling medium in addition to requiring it to serve as a means of reducing friction. Our uncertainty and lack of complete understanding of the phenomena of friction and lubrication are not confined to the generalities of the subject, but persist throughout almost all its details. For example, it is known that the viscosity of the lubricant enters as a critical factor into the problem in various ways. Viscosity as a physical conception can be defined in absolute terms, the unit being the "poise" of dimensions M/LT. Nevertheless no practical method for the measurement of absolute viscosity has been developed. To arrive at it we must deduce it with the help of an approximate formula from the non-absolute measures provided by the Redwood, Engler, Saybolt, or other form of practical viscometer. In view of all these uncertainties on the scientific side it is perhaps a cause for wonderment that the practice of lubrication should to-day be as nearly perfect as it is.

It was an admirable scheme on the part of the Council of the Institution of Mechanical Engineers to arrange a symposium of papers on lubrication and lubricants and to invite contributions to it from leading authorities on the subject residing or carrying out their investigations in the principal engineering countries of the world. At the four meetings held last week at the Central Hall, Westminster, the large audiences which assembled for the occasion had before them for their consideration and discussion about one hundred and twenty-five papers on the design and behaviour of bearings, on the lubrication of engines and other industrial applications and on the properties and testing of lubricants. Of necessity there was a certain amount of duplication among the papers, but taken as a whole they provide a very complete collective statement of existing knowledge on the subject. If it does nothing else the symposium reveals the magnitude of the efforts now being made in this country and abroad to study all aspects of the lubrication problem. It may be that the symposium will provide the inspiration to fresh work which will yield important results of wide-reaching significance. For the time being, however, practical engineers must be warned against expecting too much from it. In particular the plea made on the closing day by Mr. Reavell for increased simplicity in lubrication research and for the presentation of the results in a manner which will facilitate their practical application is not likely to be met for some time to come. In many respects the physicists, chemists and others engaged on lubrication research are not ready to pass on their results to the engineer. They are still employed on the investigation of fundamentals, and until they have determined what these fundamentals really are and have devised means for their study we cannot reasonably expect to derive much practical advantage from their work. On the theoretical side the symposium does little to encourage the hope that any great illuminating principle which will guide engineers towards improved practice is on the point of emergence. With the brilliant exceptions of the hydrodynamical theory developed by Reynolds from Tower's experimental results and of the mathematical investigations on which Michell based the design of his tilting pad bearings, practical lubrication owes very little to theory. It is to be hoped that present-day investigators, both on the experimental and theoretical side, will constantly bear in mind the fact that lubrication is purely a practical problem and that they will not allow their attention to be deflected—however great the temptation—to academic aspects. It would be regrettable if they followed the example of the elasticians who by deliberately restricting their outlook to idealised bodies and conditions have divorced their subject from practical reality and value.

Engineers made bearings and discovered the value of lubricants long before scientists turned their attention to the subject. To-day in spite of the uncertainties which crowd upon us from the scientific side and largely, if not wholly, as a result of practical trial, we seem, in at least some important instances, to have reached a stage in the evolution of the lubricated bearing which nearly approaches perfection. What farther practical developments, it may be asked, can we expect from the intensive prosecution of lubrication research? Sir Nigel Gresley has stated that at the most two or three hot axle-boxes occur per month on the carriages of the London and North-Eastern Railway. As a means of preventing hot axle-boxes there appears therefore to be little further opportunity for the application of fresh discoveries in the field of lubrication. Again as a means of improving the efficiency of power transmission gearing, which already may reach a figure of 98 per cent. or over, the maximum possible benefit to be obtained by any lubrication development is clearly limited to a small figure. In some other applications, as, for instance, turbine and rolling mill bearings and the gudgeon pins of internal combustion engines, the field open for possible improvement is perhaps greater. In these and some other similar cases, however, the improvement mainly required is not so much a reduction in the friction loss as an increase in the capacity of the bearing and its lubricant, or lubricant-coolant, to dissipate the heat which flows to the bearing from its surroundings. The problem is therefore more closely linked with bearing design than with lubrication research. This conclusion is supported by the fact that in a number of rolling mills the traditional type of bearing is being replaced by bearings of plastic or other non-metallic material lubricated and cooled with water. It would perhaps be cynical to say that the object of the present activity in lubrication research is to discover why modern bearings run so well. A justifiable suspicion does, however, exist that a good deal of it is directed towards academic ends and that some of its results will be of much more interest to the atomic physicist than to the practical engineer. We must, however, hesitate to criticise any scientific investigation on the ground that no immediate practical application for its results is visible. The elucidation of the phenomena of friction and lubrication in all their aspects cannot but add greatly to our understanding of Nature, possibly in directions which, at the moment, we are unable to foresee.

Pilot Fuel Injection.

"Shipbuilding and Shipping Record", November 4th, 1937.

The high-speed heavy-oil engine is of particular interest to the marine engineer, as apart from its possibilities for main propulsion in conjunction with some form of speed-reduction gearing, it represents a very attractive proposition for driving the auxiliary generators on account of its low weight-power ratio. One of the great disadvantages of this type of generator, however, is the "diesel-knock", which is dependent upon what is termed the delay period of the fuel, i.e., the delay between the injection and the explosion of the more volatile fractions of the fuel. A contemporary publishes an interesting account of an investigation carried out on a heavy-oil engine having a normal speed of 1,200 r.p.m., in which an endeavour was made to eliminate the diesel-knock by introducing a small quantity of the fuel before the top dead centre position. The engine was run at a constant fuel injection, but both the quantity of oil and the timing relatively to the top dead centre of the pilot injection were varied, and the diagrams showing the combustion process were taken on a cathode-ray indicator. From an analysis of the results obtained it is suggested that pilot injection does undoubtedly lead to the elimination of diesel-knock, and therefore of smoother running, and that the best results are obtained when the pilot injection represents about 21 per cent. of the main charge, the timing being between 40 and 50° before the top dead centre. The pilot injection was effected by a special design of cam.

T.B.D.'s and C.M.B.'s.

"Shipbuilding and Shipping Record", November 4th, 1937.

The various types of high-speed craft, as used in the Royal Navy, are of particular interest to the naval architect as, apart altogether from their peculiar merits as vessels of offence and defence, they form the prototypes of larger ships used both for naval purposes and in the mercantile marine. For example, in referring to a recently reconditioned passenger liner to which certain structural modifications were also made, it was stated that "she now has the bow of a destroyer", indicating that the experience gained in the design of low-resistance hull forms for torpedo-boat destroyers had been utilised in reducing the resistance of the vessel in question. Similarly, in the development of types of propelling machinery of low specific weight, success obtained in naval vessels may have a profound influence on the equipment of merchant ships. The water-tube boiler leaps to the mind as an illustration of this phase of the question. Hence, apart altogether from its intrinsic interest to the student of naval history, the presidential address, which Sir John E. Thornycroft gave before the Institution of Mechanical Engineers, is of great value as indicating the reactions which have occurred in the past and may occur in the future between naval architecture and marine engineering as applied to naval and mercantile vessels.

Acting upon suggestions made to him, Sir John took as the subject of his address "torpedo boats", and from his great personal knowledge of the development of the many types of high-speed craft with which the world-famous firm bearing his name has for so long been associated, he was able to weave a vivid story, starting from his recollection of being taken on a trial run of H.M.S. "Lightning", which was officially No. 1 Torpedo Boat of the British Navy, some sixty years ago, when she came back to her builder's yard on the Thames for some alterations. The earliest types of torpedo boat were of very small dimensions and could be produced in large numbers, so that when, in 1892, it was realised that certain foreign navies were concentrating on the production of this form as a weapon of offence, the British Admiralty decided to build a class of vessels to combat them. Hence was born the "torpedo-boat destroyer", which was in reality a larger and faster torpedo boat, armed with guns. Other navies soon began to construct similar craft, and the smaller torpedo boats were no longer built, but the name "torpedo-boat destroyer" remains to this day, sometimes abbreviated to "destroyer" without qualification, or more briefly still "T.B.D." In addressing an audience of engineers, Sir John, as might be expected, dealt more particularly with the development of the propelling machinery, and he gave an interesting story of the early troubles with locomotive boilers and high-speed reciprocating engines, notably the trouble which was experienced in detecting hot bearings in the latter, which, since they were not forced lubricated, had to be felt by hand when running at speeds up to 400 r.p.m. In spite of all these difficulties, the development of this type of craft was such that in 38 years from H.M.S. "Lightning", a vessel 81ft. long, of 400 i.h.p., of 30 tons displacement and a speed of 18½ knots, to 1914, on the eve of the Great War, the destroyer had become a vessel of about 1,000 tons and 20,000 s.h.p., with a speed of 36 knots or more. The use of turbines, of speed reduction gearing, of oil fuel, and so on, while it may be regarded as a recent story, loses none of its interest when told, as it was by one who was so largely responsible for the success of these innovations.

The coming of the internal-combustion engine, particularly as developed for the surface propulsion of submarines, led to the investigation of the possibilities of their use in torpedo-boat destroyers, but whereas during the war period steam turbine machinery gave 60 h.p. per ton of weight, the lightest submarine engines gave only about 20 h.p. per ton. It happened, however, that owing to the existence of minefields the war-time destroyer was often precluded from reaching its objective. Hence, a small type of shallow-draught torpedo boat was developed, driven by petrol engines similar to a type used in certain aeroplanes. These vessels, originally called motor torpedo boats, but subsequently named coastal motor boats or C.M.B.'s were originally of just sufficient size to carry a single 18in. torpedo, but they proved so successful that larger sizes were built, some having a length of 55ft., carrying a load of more than two tons of torpedoes and other ammunition, and developing a speed of about 40 knots, while towards the end of the war the length had reached 70ft. and the load of torpedoes or mines, five tons. The C.M.B.'s were responsible for a stirring chapter in the naval history of the war with which the name of Thornycroft will ever be associated. In concluding his address, Sir John ventured upon a glimpse into the future, and commenting upon the tendency towards the introduction of complicated mechanisms and appliances, he suggested that the work of the engineer should be in the direction of simplification. The more complicated the apparatus the greater the chance of its being put out of action, and since simplification and reliability go together, he considered it to be all-important that an endeavour should be made towards simplification. Altogether, a most interesting presidential address and one that will be long remembered by those privileged to hear it.

The B. and W. Two-stroke Single-acting Engine.

The Latest Design.

"The Motor Ship", December, 1937.

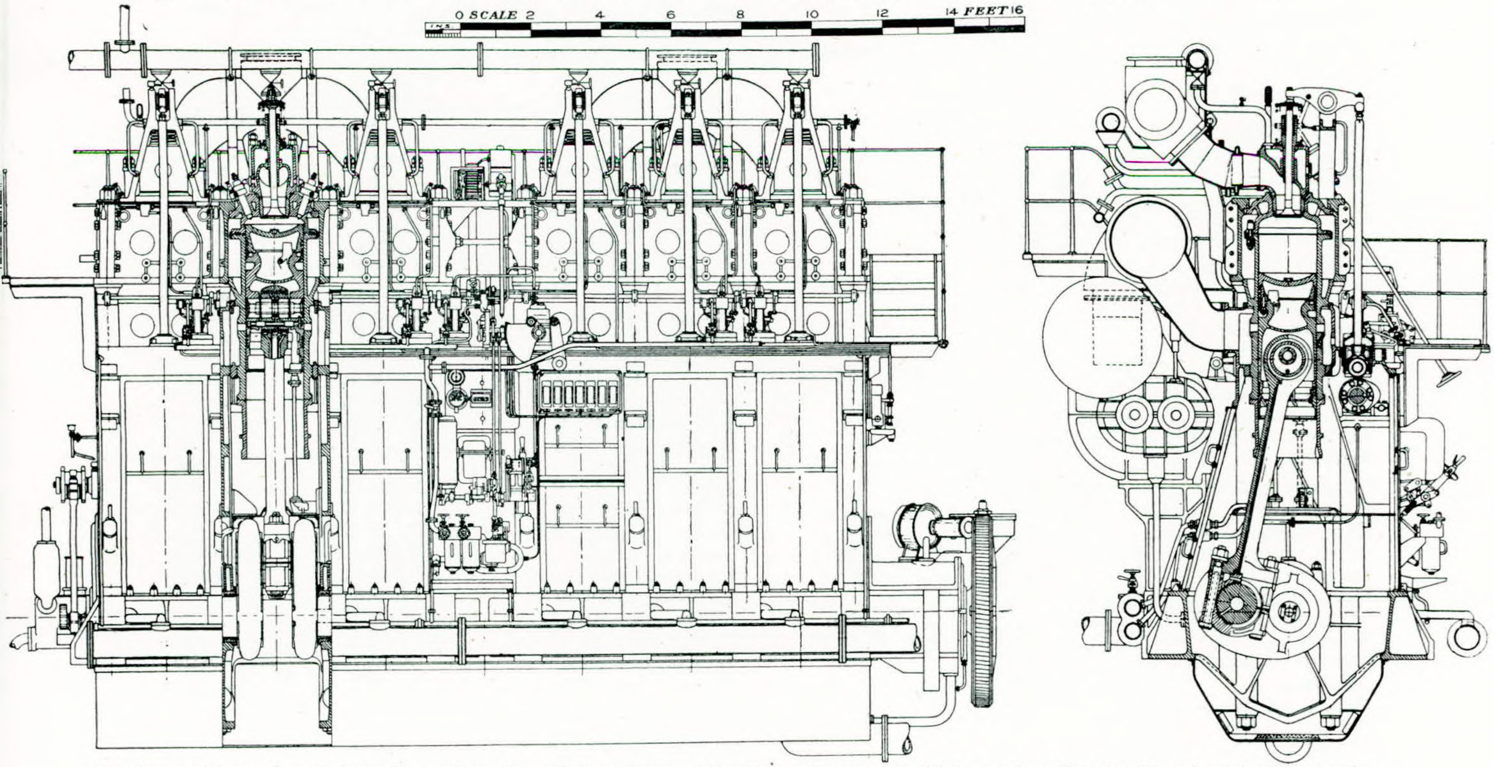
When Dr. Blache read a paper on the Burmeister and Wain two-stroke engine at a meeting of the Institute of Marine Engineers last year (we published an extract of the lecture in our November, 1936, issue) he drew attention to the fact that the exhaust piston valves originally adopted for the single-acting, B. and W. two-stroke trunk-piston engine had given place to exhaust valves of the poppet type. A cross-sectional elevation of a typical unit was included among the illustrations of the machinery Dr. Blache described.

The drawings reproduced on this page, however, represent the engine in its latest form. The modifications are not of any major importance, the general interest in the illustration being that for the first time in this journal there is shown a side elevation of the engine with a cylinder in section, apart from the new end elevation to which reference has been made. The ports for the entry of scavenging air will be observed at approximately half the height of the cylinder wall. The exhaust gases have a straight passage through the cylinder and are discharged through the poppet valve at the top, this valve being located centrally in the cylinder head. The camshaft is located within the crank-chamber, above the control station, and the poppet valves are operated through vertical push-rods and rocking levers. It should be noted that the same cams are used for ahead and astern operation of the engine, as the camshaft is not moved in a fore-and-aft direction for reversing. Scavenging air is supplied by rotary blowers.

BOARD OF TRADE EXAMINATIONS.

List of Candidates who are reported as having passed examinations for certificates of competency as Sea-Going Engineers under the provisions of the Merchant Shipping Acts.

Name.	Grade.	Port of Examination.
For week ended 7th October, 1937 :—		
Kerr, John	2.C.	Belfast
Foster, Joseph G.	2.C.M.	"



Sectional side and end elevations of a six-cylinder two-stroke Burmeister and Wain engine fitted with exhaust poppet valves.

Name.	Grade.	Port of Examination.
Archdeacon, Denis J. R.	1.C.	London
Fee, Thomas H. C. H.	1.C.	"
Fraser, Alexander D.	1.C.	"
Haig, Lionel C.	1.C.M.	"
Marshall, David	1.C.	Glasgow
Paterson, David W.	1.C.	"
Livingstone, Donald	1.C.M.	"
Clensy, Stanley	1.C.	Liverpool
Ewing, John C.	1.C.	"
Hamer, Edward V.	1.C.	"
Nestor, John F.	1.C.	"
Williams, Charles E.	1.C.	"
Chubb, Nigel E.	1.C.M.	"
Scaife, Roy	1.C.	Newcastle
Martin, William S.	1.C.M.	"
Crowther, Harry	1.C.S.E.	Liverpool
Williamson, Robert J.	1.C.S.E.	"
Stevens, Alfred E.	1.C.S.E.	Newcastle
Johnson, Albert E.	1.C.M.E.	"
Stewart, Robert	1.C.M.E.	"
Maxwell, James	1.C.M.E.	Liverpool
Bowman, Joseph	1.C.M.E.	Newcastle
Black Robert	1.C.M.E.	"
Crick, Percy J.	1.C.M.E.	Liverpool
Brainiff, John F.	1.C.M.E.	"
Fawcett, William	1.C.M.E.	Newcastle
Hall, William G.	1.C.M.E.	"
Passman, Henry	1.C.M.E.	"
Falconer, John	1.C.M.E.	Liverpool
Morton, James R.	1.C.S.E.	Glasgow
Logan, Stanley C.	1.C.M.E.	London
Martin, Harry S. E.	1.C.M.E.	"
Maxwell, Robert G.	1.C.M.E.	"
Stewart, Leonard A.	1.C.M.E.	"
Riley, Edward H.	1.C.M.E.	"
Millar, John H.	1.C.M.E.	Glasgow
King, Thomas J.	1.C.M.E.	"
Henderson, William H.	1.C.M.E.	"
Duncan, David	1.C.M.E.	"

For week ended 14th October, 1937:—

Name.	Grade.	Port of Examination.
Ferguson, Thomas	2.C.M.	Newcastle
Victory, Gordon	2.C.M.	"
Barnes, Arthur S.	2.C.	London
Christensen, Stanley G.	2.C.	"
Shirley, Sidney D.	2.C.	"
Findlay, Francis	2.C.M.	"
Blundell, George E.	2.C.	Liverpool
Stephenson, Henry	2.C.	"
Colwell, John McM.	2.C.M.	"
Drummond, William	2.C.M.	"
Hughes, George	2.C.M.	"
Gibbeson, John C.	2.C.	Glasgow
Giffin, John A. MacK.	2.C.	"
Stewart, Arthur J.	2.C.	"
Porteous, James S.	2.C.M.	Glasgow
Briffett, Albert S. G.	2.C.	Cardiff
Davies, Clement H. A.	2.C.	"
Harvey, Norman J. M.	2.C.	"
Turner, William L.	2.C.	"
Fenwick, John H.	2.C.M.	"

For week ended 21st October, 1937:—

Name.	Grade.	Port of Examination.
White, John	1.C.	Liverpool
Kerr, Rodger N.	1.C.M.	"
Evans, Ivor J.	1.C.	Cardiff
Jenkins, Charles A.	1.C.	"
Jones, Thomas O.	1.C.	"
Abraham, Leslie T.	1.C.	London
Bremner, Robert C.	1.C.	"
Goodier, James B.	1.C.	"
Lee, John S.	1.C.	"
Mackavoy, James J.P.	1.C.	"
Duckett, Alfred M.	1.C.	Newcastle
McFadyen, Donald	1.C.	"
Brown, Thomas A.	1.C.	Glasgow
Craig, Joseph R.	1.C.	"
Lowes, Robert H.	1.C.S.E.	Newcastle
Qualters, Thomas A.	1.C.M.E.	Liverpool
Smith, Charles H.	1.C.M.E.	Liverpool
Jacob, Mervyn C.	1.C.M.E.	Cardiff
Tippett, Keith F.	1.C.M.E.	"
Adams, Frank	1.C.M.E.	London
Berlie, Roland St. C.	1.C.M.E.	"
Clargo, Randolph P.	1.C.M.E.	"
Holman, William F.	1.C.M.E.	"
King, Robert H.	1.C.M.E.	"
Hodgkinson, Cecil F.	1.C.M.E.	Liverpool
Holker, James	1.C.M.E.	"
Adams, James T.	1.C.M.E.	Glasgow
Cragie, Christopher R.	1.C.M.E.	Newcastle
Graham, Robert R.	1.C.M.E.	"

For week ended 28th October, 1937:—

Name.	Grade.	Port of Examination.
Hall, Leonard	2.C.	Glasgow
Milligan, Peter	2.C.	"
Bell, James F.	2.C.M.	"
Laing, John	2.C.M.	"
Cromey, William R.	2.C.	Liverpool
Crosby, Frank	2.C.	"
Vincent, John V.	2.C.	"
Young, William A.	2.C.	"
Garden, John	2.C.M.	"
Davy, Robert E.	2.C.	London
Morse, Frank E.	2.C.	"
Simpson, Jack	2.C.	"
Angus, James MacD.	2.C.M.	"
Jackson, William T. H.	2.C.	Newcastle
Russell, Archibald E.	2.C.	"
Glass, David A.	2.C.M.	"

For week ended 4th November, 1937:—

Name.	Grade.	Port of Examination.
McLean, Peter W.	1.C.	Glasgow
Turnbull, George	1.C.	"
McGuffie, David D.	1.C.M.	"
McCintock, John S.	2.C.M.	Belfast
Beavis, Jack C.	1.C.	London
Carnaghan, James T.	1.C.M.	"
Lewthwaite, Harry R.	1.C.	Liverpool
Weaver, Douglas R.	1.C.	"
Jones, Richard	1.C.M.	"
Holme, Thomas	1.C.	Newcastle
Lewis, Frank	1.C.	"
Ridley, Arthur	1.C.	"
Story, Thomas W.	1.C.	"
Jackson, William G.	1.C.M.	"
Little, Thomas R.	1.C.M.	"
Robinson, John G.	1.C.M.	"
Chubb, Nigel E.	1.C.S.E.	Liverpool
Kerr, Rodger N.	1.C.S.E.	"
Haig, Lionel C.	1.C.S.E.	London
Brunton, Arthur J.	1.C.S.E.	"
Pratt, William L.	1.C.M.E.	Liverpool
Trowin, Ernest V.	1.C.M.E.	London
Jones, Alexander B.	1.C.M.E.	"
James, Donald K.	1.C.M.E.	"
Reading, John S.	1.C.M.E.	"
Cook, Robert S.	1.C.M.E.	Liverpool
Jones, Thomas R.	1.C.M.E.	"
Nestor, John F.	1.C.M.E.	"
Owen, Stephen R.	1.C.M.E.	"
Savage, William	1.C.M.E.	London
Buchan, John	1.C.S.E.	Glasgow
MacKay, William G.	1.C.S.E.	"
McDougall, Alexander	1.C.M.E.	"
McLaren, John W.	1.C.M.E.	"
Willcock, James	1.C.M.E.	Newcastle
Nixon, Benjamin M.	1.C.M.E.	"
Samuel, Henry A.	1.C.M.E.	"
Henry, Peter	1.C.M.E.	"
Henderson, Robert	1.C.M.E.	"
Allinson, George F.	1.C.M.E.	"
Langley, William A.	1.C.M.E.	