The INSTITUTE of MARINE ENGINEERS
Transactions

1951, Vol. LXIII, No. 7

Marine Torsionmeters and Thrustmeters

R. COOK, M.Sc., M.I.Mech.E. (Member)

The general requirements of torsionmeters and thrustmeters for marine use are first discussed together with some of the fundamental difficulties encountered in their design.

A critical survey is then made of the various forms of torsionmeter which have been used or proposed for marine use and results are given of experimental investigations into the degree of accuracy obtainable with the Siemens-Ford torsionmeter. It is concluded that this instrument is capable of an accuracy of at least $\pm$ 2 per cent over a range of shaft stress from about 2,500lb. per sq. in. to 4,500lb. per sq. in. this range covering the full power condition in most merchant ships. In vessels with Diesel machinery aft the full power stresses are frequently lower and in these cases somewhat less accurate results are to be expected.

A similar survey is made in respect of thrustmeters and the results are given of calibration trials with a Michell thrustmeter from which it is concluded that an accuracy of at least $\pm$ 2 per cent is obtainable with this instrument under full power conditions.

INTRODUCTION

The correlation of tank test data with actual results obtained on the measured mile and in service is one of the most important items on the research programme of the British Shipbuilding Research Association. Early consideration was therefore given to the methods available for measurement of torque and thrust since accurate measurement of these quantities is vital to a proper analysis of trial and service data.

Since the beginning of the century, many forms of instrument have been developed for these two purposes and not a few of them have been actually tried out on board ship. Very little evidence was available, however, as to their probable accuracy and opinions were divided as to the reliance to be placed upon the results obtained with them. In these circumstances, a critical survey was made of the various forms of torsionmeter and thrustmeter which had been developed or proposed, with a view to selecting the more promising for trial and possible further development. In the present paper, the author proposes to deal briefly with this survey, to outline the considerations which led to the selection of certain of these instruments for trial, and to give some of the results so far obtained.

GENERAL CONSIDERATIONS

Before going on to discuss the various types of instrument which have been employed, it might be as well to consider briefly some of the general requirements of torsionmeters and thrustmeters for marine use.

So far as sea trials are concerned it may be assumed that expert personnel is available for operation of the instrument and it is suggested that the primary requirements are accuracy, ease and speed of installation, and applicability to a wide range of power and types of main propelling machinery; some complexity may be acceptable to obtain these requirements. The second and third of these requirements cannot be too highly stressed for it is of little use to have a first class instrument, capable of a high degree of accuracy, if its design is such as to call for special arrangements being made for its installation or if it can only be applied to certain types of machinery installation. The ideal torsionmeter or thrustmeter for trials work is one which can be fitted without disturbance or alteration to the normal machinery arrangements. If this is not the case the opportunities of using it will be severely limited.

In regard to service use a torsionmeter or thrustmeter must be suitable for operation by engine-room staff and it is considered that robustness, simplicity of use and reliability over long periods of time without expert attention are the primary requirements. Some slight reduction in accuracy may have to be accepted to obtain these requirements.

Consider now some of the fundamental difficulties in connexion with torsionmeters. In principle, torsionmeters measure the elastic twist of a given length of shaft, this twist being directly proportional to the torque and inversely proportional to the rigidity modulus and the fourth power of the diameter. The radial twist is a relatively small quantity (of the order of $\frac{1}{10}$ to $\frac{1}{3}$ of a degree per 10-feet length of shaft at full power) and is generally subject to cyclic variation so that its accurate measurement provides considerable scope for ingenuity. Moreover bending and centrifugal effects have to be taken care of. It is these considerations which have led to the multiplicity of torsionmeter designs. The accuracy of the results obtained depend upon the correctness of the zero-torque setting and upon the correctness of the multiplying factor in the twist measuring unit. With the majority of torsionmeters, the zero setting can be checked by rotating the engine by means of the turning gear, first in one direction and then in the other; if correct the readings should fall evenly on either side of the zero. Care should be taken to ensure that the propeller is in slack water whilst making these observations. Accuracy of the multiplying factor is best checked by calibrating the instrument, in the shop, assembled on the shaft at the point where it is to be used. It is not, however, always possible to carry out shop tests and where this is the case, the multiplying factor is arrived at by calculation from the known characteristics and dimensions of the torsionmeter and using an assumed figure for the modulus of rigidity of the shaft.

Because of variations in the modulus of rigidity of steel shafts the latter method of calculation may lead to inaccuracy in the multiplying factor and the question of the most suitable value of the modulus of rigidity to be assumed, together with possible errors likely to be introduced, has been the subject of considerable investigation.[1, 2, 3, 4]

The figures in Table 1, based upon data supplied by Messrs. Siemens Bros., illustrate the variation in modulus of some sixty-four mild steel shafts tested during the periods 1930-46 and 1946-50 respectively. These shafts, comprising both solid and hollow specimens, varied in diameter from $4\frac{1}{2}$ inch to $21\frac{5}{8}$ inch.

TABLE 1

Variation of modulus of rigidity from mean value of 12×10^6 lb. per sq. inch, per cent	1930-1946		1946-1950	
	No. of shafts falling within limits	Per cent of total shafts	No. of shafts within limits	Per cent of total shafts
$\pm\frac{1}{2}$	12	26·1	6	33·3
±1	23	50·0	10	55·6
±2	35	76·1	15	83·3
±3	40	87·0	17	94·5
±4	43	93·6	18	100·0
±5	45	97·8	—	—
±7	46	100·0	—	—

The figures for the forty-six shafts tested during 1930-46 show a rather wider deviation from a mean value of 12×10^6lb. per sq. in. than those for the eighteen shafts tested during 1946-50. Taking the figures for the latter period, as more representative of modern material, it will be seen that the modulus of only 55 per cent fell within $\pm$ 1 per cent and some 83 per cent within $\pm$ 2 per cent of a mean value of 12×10^6lb. per sq. in. To put the matter another way, if one assumes a mean value of 12×10^6lb. per sq. in. then there is only an even chance that one would be correct to within $\pm$ 1 per cent and about 4 chances out of 5 that one would be correct to within $\pm$ 2 per cent. These figures serve to emphasize the importance of carrying out a shaft calibration wherever possible.

So far as thrust measurement is concerned, three broad methods present themselves. The thrust may be counter balanced by weights or by hydraulic pressure; one can attempt to measure the deformation of an elastic member inserted in the line shafting or one can attempt to measure the elastic compression in the shafting by the use of strain gauges. The use of elastic members inserted in the shaft involves the grave disadvantage of alteration to the normal line shafting arrangement whilst the difficulty in the case of strain gauges lies in the very small strains involved (of the order of one to two thousandths of an inch per 10-feet length at full power). It is not always appreciated that the elastic strain in the line shafting of a merchant ship at full power is approximately the same as that due to a rise of 1 deg. C. in the temperature. It will thus be seen that with such a method the effect of variation in the temperature, to say nothing of bending, etc., may be of a magnitude comparable with the quantity one is trying to measure. In these circumstances it is not surprising that the most successful instruments to date have belonged to the first category referred to.

EXISTING TYPES OF TORSIONMETER

Torsionmeters can be divided into four groups according to the method adopted for measuring the angle of twist, thus mechanical, optical, acoustical and electrical methods have all been employed. Additionally, it is possible to arrive at shaft torque by measuring the elastic strain in the shaft by some form of electric strain gauge.

(i) Instruments which Measure Shaft Twist Mechanically

Many years ago, a spring torsionmeter was designed by Parsons[5] and fitted to a naval vessel, and although by modern standards this apparatus is now out of date, it is the forerunner of modern torsionmeters. The arrangement consisted essentially of a coupling in two sections placed in the shaft and having coiled springs between the face of the dogs. A bell crank pivoted on one half of the coupling had the extremity of one arm fixed to the other half of the coupling and so the movement of the extremity of the other arm of the bell crank from a fixed point gave an indication of the relative movement of the coupling sections and hence the torque. No mention is made in the published data of the method used in taking readings on the rotating shaft. This instrument appears to be mainly of historical interest.

A more recent design employing mechanical means is the White-Fox mechanical power meter.[6] Considerations of space forbid a full description of this instrument which has already been described elsewhere. Briefly, two geared up counter shafts are driven from the line shafting in opposite directions and an ingenious arrangement resembling a differential gear situated midway between them is used to record the twist in the line shaft between the points of the drive. This instrument has been used in a cargo vessel at sea but no reliable information is available regarding its performance. The chief advantages appear to be that it can be fitted without disturbance of the existing line shafting and that it gives direct reading of power. On the other hand, it appears that dynamic calibration would be necessary—a procedure which is only rarely possible—and moreover any differential wear between the forward and aft driving gear would tend to falsify the readings.

Another design employing mechanical means is the Fottinger torsionmeter[7] which was used on trials of German destroyers. It is reported to have lost favour, however, because of its bulk, the considerable time needed for installation, and errors due to friction, centrifugal force and lost motion.

(ii) Instruments which Measure the Shaft Twist Optically

The well-known Hopkinson Thring torsionmeter[3] has been used in both merchant and naval vessels for many years. In this instrument a sleeve and stump are attached rigidly to the shaft with approximately knife edge contact, the knife edges being a fixed distance apart. A small mirror is pivoted on the sleeve and an arm which is fastened to the mirror connects by link to the stump. Relative movement of the sleeve and stump causes the mirror to tilt in its pivots. A beam of light from a lamp is reflected by the mirror on to a fixed scale giving a spot of light once per revolution. The tilting of the mirror displaces the spot of light on the scale proportionally to the twist of the shaft. Since one mirror gives the reading for one point only during a revolution, multiple mirrors are required around the shaft if cyclic variations in torque are present.

One of the advantages of this instrument is that it can be fitted to existing shafting. On the other hand, each observation constitutes a mean over a number of revolutions and hence if there are marked variations in the instantaneous value of torque from revolution to revolution (such as will occur in all

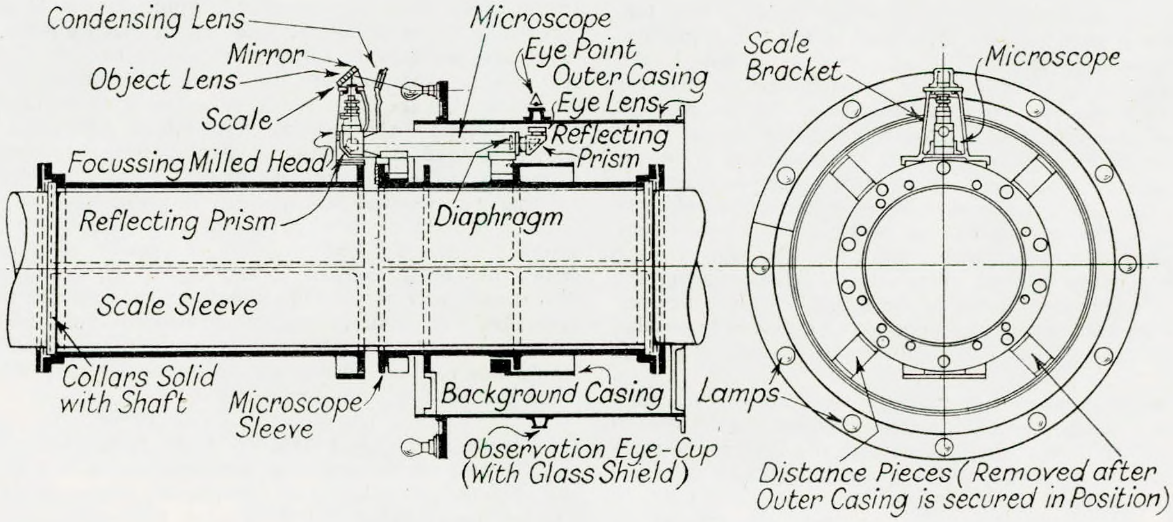


FIG. 1—*Denny microptic marine torsionmeter*

but very fine weather) accuracy of observation will be difficult if not impossible. Again, when the torque is not uniform a number of observations must be made at different points on the shaft circumference to obtain a true average; this renders the process of observation slow and tedious. A further point which can be made against this instrument is that the spigot arrangement which is employed to steady the sleeve introduces the possibility of error due to friction.

Rather similar arrangements are used in the Frahm[7] and Suyehiro[3, 8] torsionmeters. The former was extensively used in the trials of German naval vessels but suffered from the disadvantages already referred to.

The Denny Microptic Marine torsionmeter[3] has two stout sleeves clamped to collars turned on the shaft and spaced the required distance apart. A scale is attached to one sleeve and a microscope to the other (see Fig. 1). There is no contact between the sleeves. The body or tube of the microscope is parallel to the shaft axis, and prisms transmit radial readings from the object to the eye lenses. The scale is finely divided to 1,000 divisions per inch for direct measurement of the small movements that occur during torsion. Surrounding the sleeve which carries the microscope is a stationary casing provided with a number of openings around the periphery, so that an average reading of torque can be obtained.

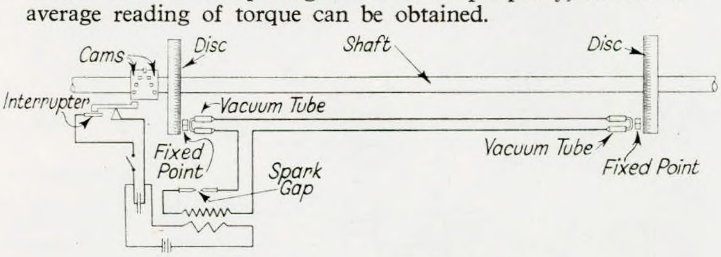


FIG. 2—*Stroboscopic torsionmeter*

It appears that with this torsionmeter accurate measurements of average torque should be possible, provided that there are no marked variations in torque from revolution to revolution, i.e., in reasonably calm weather. Under less favourable conditions, however, such as are encountered in service, some difficulty might be experienced in obtaining an accurate value. This instrument also suffers from the disadvantages regarding slowness and difficulty of operation already mentioned in connexion with the Hopkinson-Thring type. It should also be noted that the use of collars turned integral with the shaft, whilst making for rigidity of attachment and enhanced accuracy, has the disadvantage that it necessitates the use of a special length of shaft.

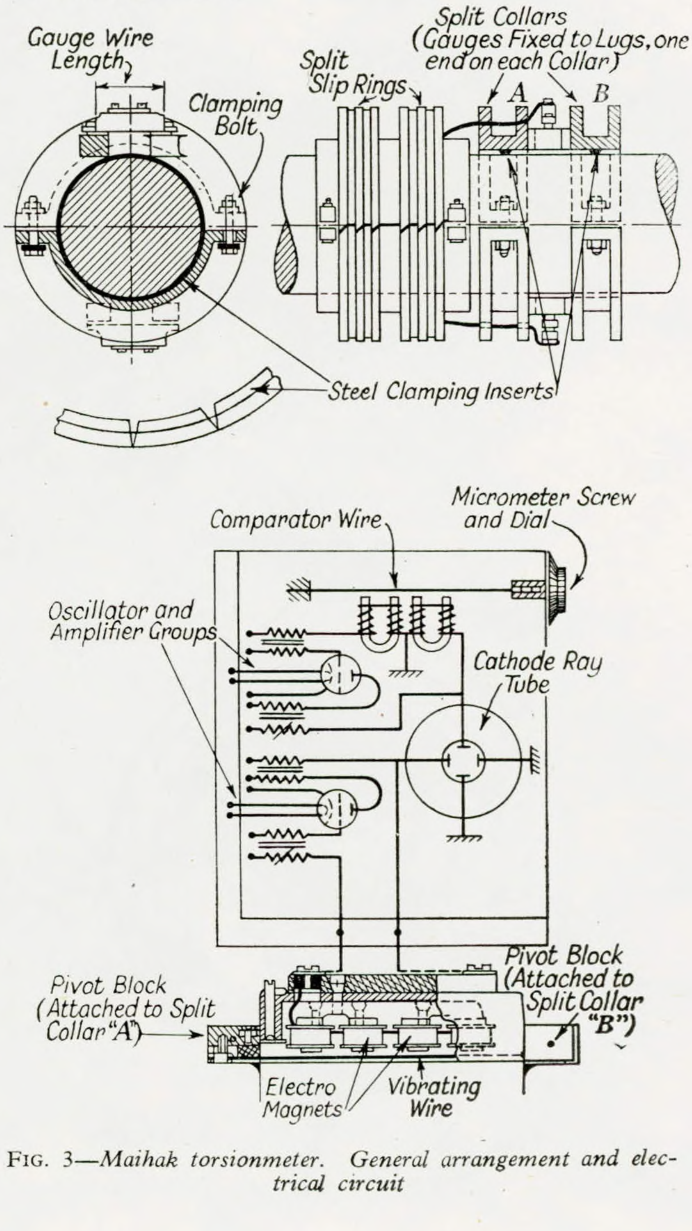


FIG. 3—*Maihak torsionmeter. General arrangement and electrical circuit*

The Muller Stroboscopic torsionmeter[9] is a long-base torsionmeter whose principles are illustrated in Fig. 2. A graduated disk is placed on the shaft at each end of the base length and a fixed point is taken adjacent to each disk so that the position of each disk relative to the fixed point can be observed. Simultaneous observations of the two rotating disks is made during a number of revolutions by using a stroboscope of simple design. A neon or other gas discharge tube is used to illuminate each disk at the datum. The secondary circuit of a transformer is connected to the two vacuum tubes which are in turn connected in series with an adjustable spark gap, the latter serving as a damping device for the electrical oscillations during the discharge. The primary circuit of the transformer is connected to an adjustable interrupter and a condenser is placed in parallel with the interrupter. The source of supply is a cell or storage battery.

Shaft rotation causes simultaneous operation of the lamps at one point in each revolution, and an observer placed at each disk can read the disk position relative to the datum, the rotating shaft giving the impression of being at rest. By having a series of cams around the shaft periphery and using them in turn to operate the interrupter, an average reading can be determined. The shaft disks are made as large as practicable, and it is claimed that it is possible to obtain an accuracy of 0·05 deg.

The advantages of the instrument, in comparison with the optical types previously described, would appear to be firstly that the long base avoids the necessity for magnification of shaft twist, and secondly that the provision of the cam interrupter gear obviates the necessity for taking observations in difficult positions. On the other hand, to arrive at an accurate multiplying factor for the instrument introduces difficulties. If this is to be arrived at by static calibration there is the problem of fitting up in the shop the long length of shafting (60-120 feet) together with its attendant bearings and couplings; on the other hand, if this factor is to be obtained by calculation using an assumed figure for the modulus of rigidity then additional errors may be introduced in arriving at the equivalent shaft. A further source of inaccuracy with this instrument lies in the possibility of relative movement between the fixed points adjacent to each disk. These datum points are attached to the ship's structure and some relative movement may be expected when the vessel is under way.

(iii) Instruments which Measure Shaft Twist Acoustically

At least one torsionmeter has been used which operates on the acoustic principle. This instrument, known as the Hoppe-Maihak torsionmeter[3, 10, 11] is of small bulk and can be fitted to the shaft in a few hours.

The principle employed is that the pitch of the note emitted by a vibrating wire depends upon the tension in the wire. A steel wire is attached tangentially to the shaft and put

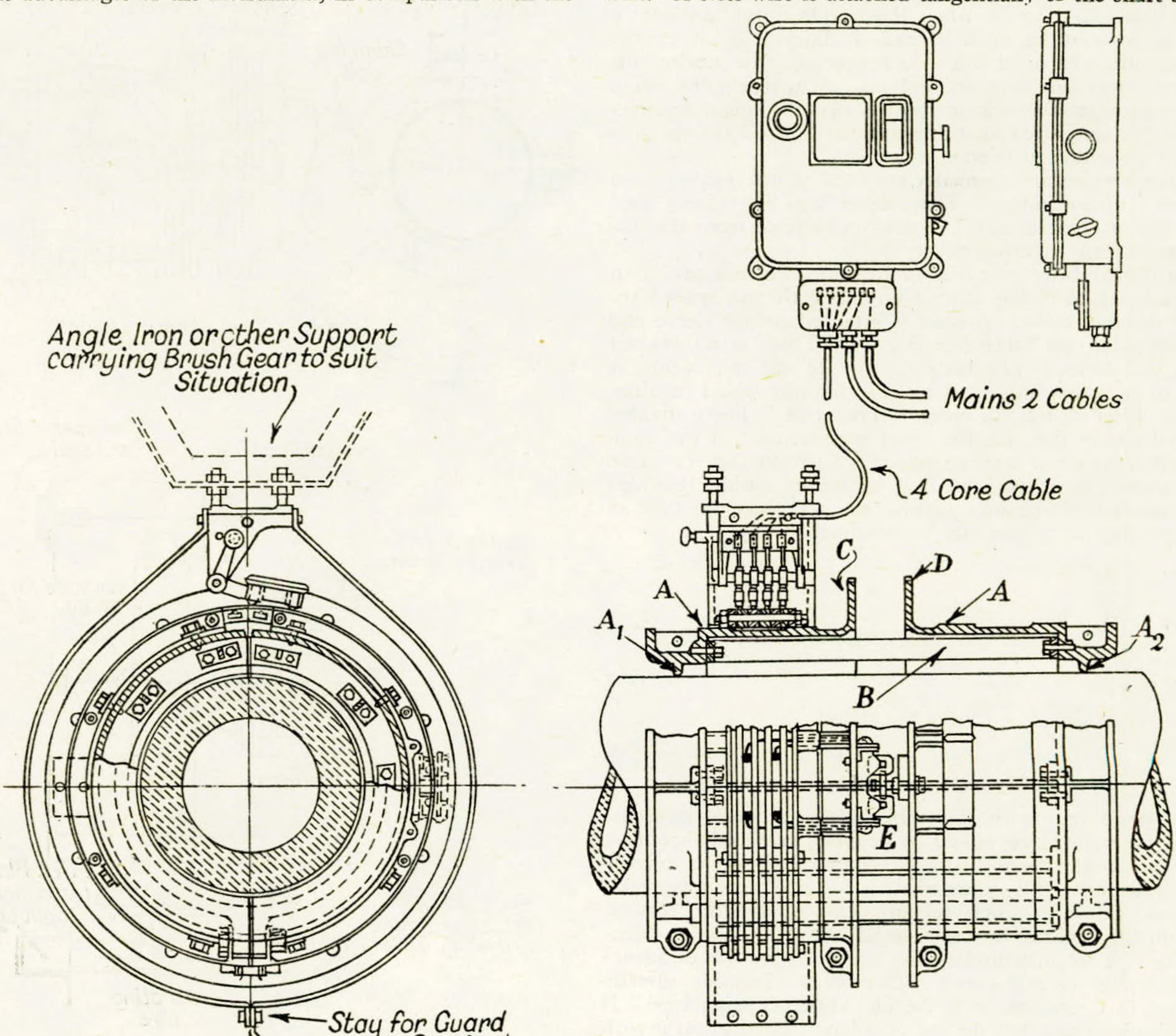


FIG. 4—*General arrangement of Siemens-Ford torsionmeter*

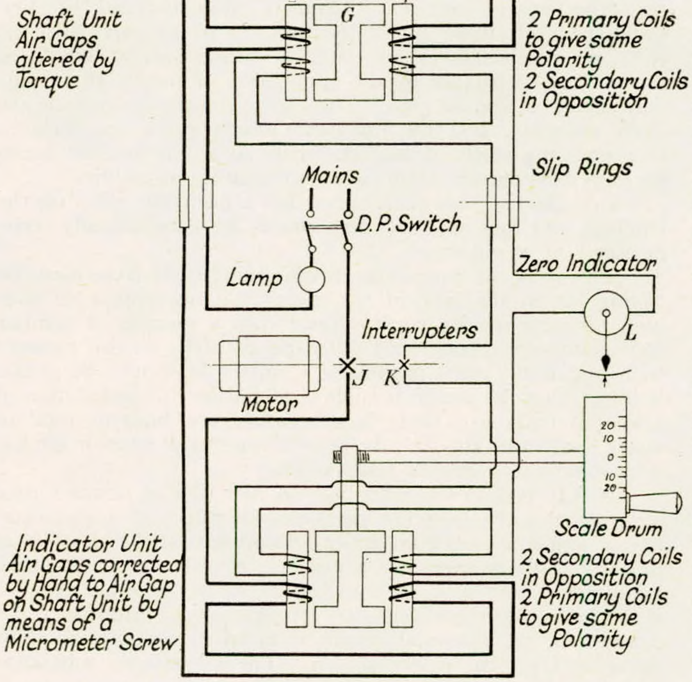


FIG. 5—*Schematic arrangement of circuit—Siemens-Ford torsionmeter*

under tension between two flanges, these flanges being fastened on the shaft a definite distance apart and relatively close together. Twist between the flanges increases the wire tension, and this can be measured by the change in pitch of the tone emitted by the vibrating wire. It is claimed that the change of pitch due to an elongation of one ten-thousandth of a millimetre (corresponding to a shaft twist of the same amount) can be heard clearly.

The wire is set vibrating by an electromagnet, and the frequency of vibration of a similar wire is brought into resonance by altering the tension of the latter wire by varying amounts. Thus the extension of the torsionmeter wire can be found and the torsion calculated.

In more recent patterns of this instrument, electronic means are employed for indicating when the frequency of the comparator wire is equal to that of the test wire on the shaft. A general arrangement of this later design is shown in Fig. 3. Each wire acts as an armature between two electromagnets connected in series. Vibration of the wire induces in the magnet coils a small current, of the same frequency as the wire, which after amplification is led to one of the plates of a cathode-ray tube. When the frequency of comparator wire and test wire are equal, the Lissajou figure produced on the screen becomes stationary. It is claimed that this instrument offers a better and simpler method of frequency determination. Little was known of the performance of this instrument except that it had been extensively used in merchant and naval ship trials in Germany, but it was suspected that cyclic variations in torque might render observation difficult.

(*iv*) *Instruments which Measure Shaft Twist Electrically*

In the Siemens-Ford torsionmeter[3] the twist of the shaft alters the air gap of a small differential transformer mounted on the shaft and this alteration is measured by electrical means.

A general arrangement is shown in Fig. 4. Two sleeves A are mounted on the shaft and clamped at A_1 and A_2 respectively. A series of steel strips B is used to keep the sleeves rigidly at right-angles to the shaft. Twist of the shaft causes displacement of flanges C and D, and this in turn alters the air gaps

cores F and F_1 of the transformer are both fixed to one of the flanges, whilst the H-shaped iron piece G is fixed to the other flange. An exactly similar transformer, the air gaps of which are alterable by means of a screw micrometer, is situated in the indicating instrument.

The primary coils on the cores of both transformers are excited in series by current from the ship's mains, a motor interrupter J in the primary circuit being used to produce an alternating e.m.f. in the secondary coils of the transformers. The secondary coils of each transformer are connected in opposition to each other, so that if the two sets of air gaps of each transformer are equal no current will flow in the secondary circuit. This is the condition with no torque.

When the shaft is twisted the air gaps of the transformer become unequal and current flows in the secondary circuit. This current is taken to the galvanometer L through the medium of the interrupter K. If now the air gaps of the indicator transformer are altered by means of the micrometer screw to produce an equal and opposite e.m.f. to that given by the shaft transformer, current will cease to flow and the galvanometer pointer returns to zero. Since the two transformers are matched, the alteration of the gaps of the indicator transformer will be the same as that of the shaft transformer and thus the movement of the micrometer screw is a measure of the twist of the shaft.

Since the two sets of transformers are connected in series, the accuracy of the results does not depend on a steady flow of current, and is therefore not affected by variation of voltage or by the condition of the slip rings or brushes provided that a current is flowing; the sensitivity, however, will be affected.

In the later models of this instrument, two shaft transformers at 180 deg. to each other are employed. The object of this arrangement is to eliminate the effects of any lack of concentricity not already taken care of by the steel strips and also of any bowing of the shaft.

At the time the survey was undertaken little impartial information was available as to the accuracy of the latest models of this instrument. Users' opinions as to the value of the earlier models were somewhat conflicting. On the other hand, points in its favour are that (a) it has been widely used, with the result that the makers have had considerable experience and opportunity for its development; (b) no special shaft section is required, since the sleeves clamp on to the existing shaft; (c) the indicator unit can be fitted in any convenient position away from the shaft; (d) no special skill is required to use the apparatus once it has been calibrated and installed; (e) readings are easily and rapidly obtained; and (f) the instrument is of rugged construction.

The Moullin torsionmeter[12, 13] is of the sleeve type and depends upon the measurement of the displacement of two sections of the shaft 4·5 feet apart. The inductance of choking

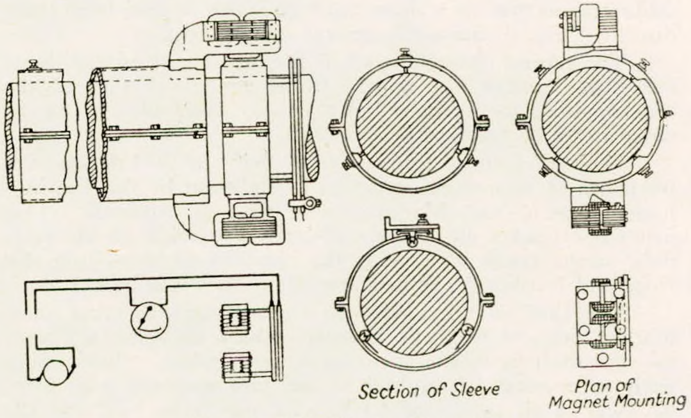


FIG. 6—*Moullin torsionmeter*

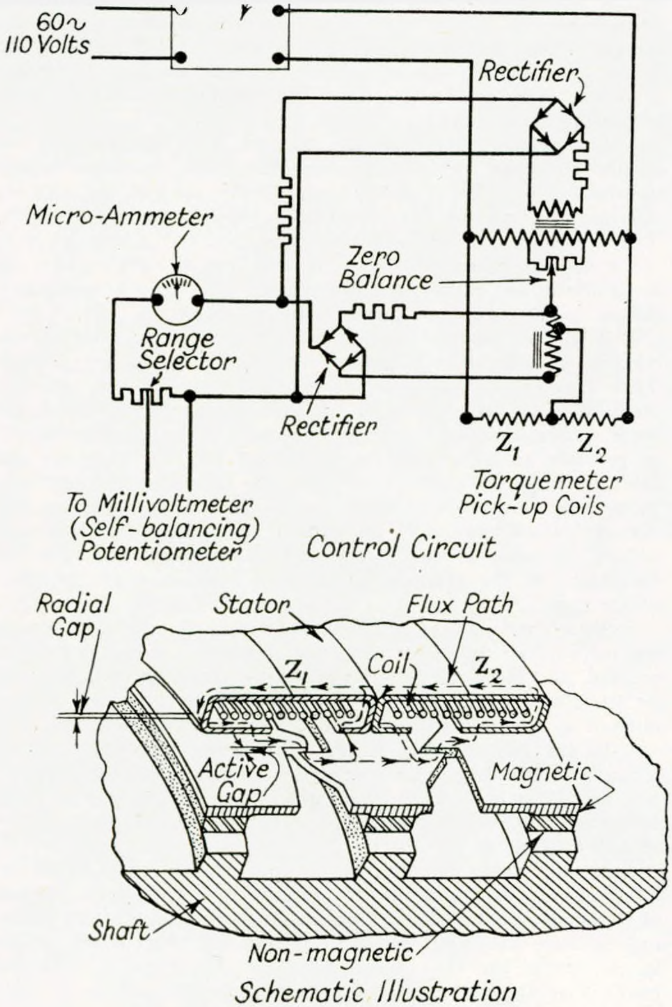


FIG. 7—*Magnetic coupled torquemeter*

coils fixed to the shaft is altered by varying the gap between two parts of their cores, this gap varying with the twist of the shaft. Alternating current is supplied to the coil from a small alternator which is driven either by the propeller shaft or by a separate electric motor, the current being led to the coils via brushes and slip rings. The virtual value of the current depends on the self-inductance of the coil and is therefore a function of the shaft twist. Ammeters placed in series with the coil are calibrated, so that the transmitted torque can be read from them directly. Fig. 6 shows the general arrangement.

The general characteristics of the instrument as regards its merits for marine use appear to be similar to that of the Siemens-Ford torsionmeter but it has not been subjected to the same degree of practical development.

Magnetic coupled torquemeter.[14, 15, 16] In 1944 details were published of an interesting design brought out by the Westinghouse Electric and Manufacturing Co., of America. This instrument makes use of the well-known principle of the magnetic strain gauge but avoids the necessity of employing slip rings and brushes. The arrangement is shown in Fig. 7.

The shaft is provided with three flanges carrying three toothed rings of magnetic material, which are separated from the steel shaft by means of non-magnetic spacers. Overlapping teeth or projections from each of the three magnetic rings form two sets of air gaps, one set between one outer ring and the middle ring and a second set between the middle ring and the other space increases. Magnetic flux induced by two stationary coils flows across the two sets of air gaps, and the coils are connected in an electrical bridge circuit which is thrown out of balance by the application of torque and resultant change in the air gaps. These coils completely encircle the shaft assembly, and the magnetic return paths are through their encasing shells of magnetic material and across the radial air gaps between the stationary and rotating assemblies.

It is claimed that shaft thrust has a negligible effect on the readings and that shaft bending effects are automatically compensated by symmetry.

This type of torsionmeter has been used in aero-engine testing but at the time of the survey did not appear to have been used for marine work. Since then a number of similiar instruments have been tried out experimentally in this country with apparently very promising results. It should be noted, however, that the design is such as to call for the installation of a special length of shaft incorporating the built-in torsionmeter and whilst this may be possible in special cases it renders it unsuitable for general trials work.

N.R.L. type Powermeter.[17, 18] A new type of power-torque meter capable of registering instantaneous values of shaft torque, power, and speed has been under development at the U.S. Naval Research Laboratories. It consists essentially of an electrical micrometer tube used in conjunction with the usual type of short base-length torsionmeter (Hopkinson-Thring type). A diagram of the electrical circuit is given in Fig. 8; the shaft sleeve arrangement is not shown. The micrometer tube con-

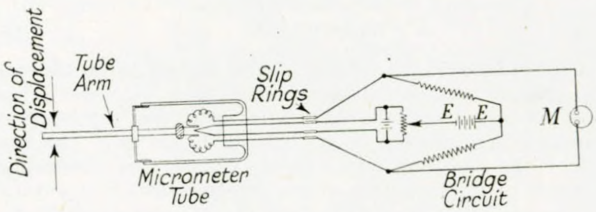


FIG. 8—*Micrometer tube (N.R.L.) torsionmeter*

sists of a vacuum tube with a hot cathode which is surrounded by a space charge of electrons, and two insulated plates or anodes to which electrical connexions can be made from outside the tube. Both anode plates are mounted on a moveable arm whose position relative to the hot cathode can be adjusted from outside the tube. Thus there is a variable plate resistance between each plate and the cathode. The change of resistance due to operation of the moveable arm is measured in a Wheatstone bridge circuit. The micrometer tube transfers mechanical displacement of the tube arm into proportionate electrical current in the bridge. Two constant-voltage supplies are necessary, one to heat the cathode, the other for bridge voltage.

By using a sleeve and stump over the shaft similar to the Hopkinson-Thring torsionmeter, the relative displacement of the flanges under torque can be transferred mechanically to the tube arm mounted on the shaft.

It is possible to use this instrument to give a direct reading of power, i.e., product of torque and shaft speed. In this arrangement the bridge supply is generated by a shaft-driven tachometer generator which gives a voltage proportional to shaft speed. Since the micrometer tube has a property that the sensitivity (i.e., meter reading for a given displacement of the arm) is proportional to the bridge voltage, the current meter indicates a quantity bearing a direct ratio to shaft power.

The outstanding advantage claimed for this instrument is the ability to expand the scale and give accurate readings at low power, a feature which would, however, appear to be more advantageous in the case of naval vessels than in merchant ships. Another feature is that by introducing a cathode ray oscilloscope the instrument may be used for studying vibration and also transient conditions during stopping and backing lished regarding its further development.

The principle of utilizing shaft twist to introduce phase difference between two generators has been used in the Drysdale[19] torquemeter, which was developed by the R.A.E. for aircraft work. The same principle has been used by the General Electric Co. of America[20] in the design of a torsion-meter for use in ships. This arrangement is shown in Fig. 9.

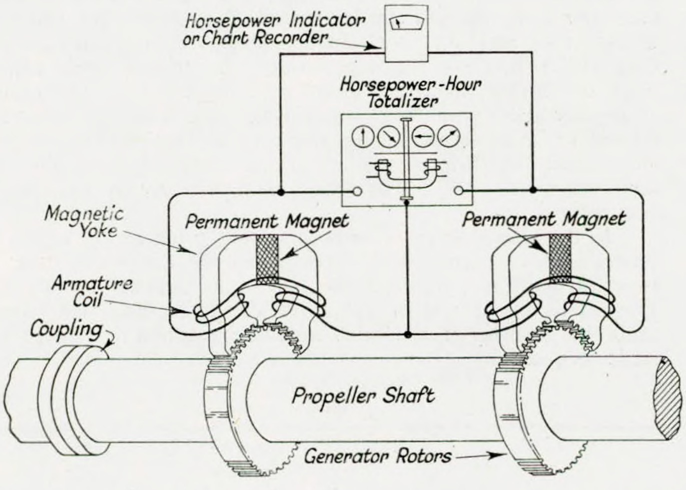


FIG. 9—*General Electric of America phase difference generator powermeter*

The meter consists essentially of two electric generators mounted on the propeller shaft and connected to remotely positioned indicators. The generator rotors are made in halves and assembled on the already installed shaft, and the stator structure is then assembled on the rotor structure. Although the stators are held to prevent their rotation, vertical or lateral motion of the shaft can occur without relative positional change between rotors and stators. The initial adjustment of the stator of each generator is such that at no torque on the revolving shaft the voltages generated by the two units are exactly 180 deg. apart in phase and balanced. Under load, the shaft twist alters the phase difference between the generators, and the resultant voltage, being proportional to both shaft twist and shaft speed, registers as an indication of horse-power.

The advantages of this scheme are that no special shafting or external source of power is required. Against this there is the necessity for dynamic calibration and the author suspects that it is difficulties in this connexion that have led to the abandonment of the design by the manufacturers. This company have recently brought out another design[21] employing electro-magnetic gauges which are mounted on clamping rings and measure the elastic twist of about 4 feet of shafting. A small a.c. generator geared directly to the shaft gives a voltage proportional to speed and by combining this with the electrical output of the gauges a direct indication of horse-power is obtained.

(*v*) *Instruments employing Electric Strain Gauges*

Strain gauges of small bulk, which can be applied to the member under investigation and which are suitable both for steady and rapidly varying strains, have been the subject of much experiment in recent years. Those types of most importance for the measurement of shaft torsion are the electrical capacity and electrical resistance types.

Electrical capacity strain gauges[16] can be used for torque measurement. Here an air-dielectric capacitor is formed of two groups of plates or conducting surfaces, insulated from each other and so positioned that when the assembly is subjected to strain the plate spacing or area changes in proportion to the

but space and weight restrictions are likely to limit the capacity to a few micro-farads, and so, even with an alternating current of several hundred cycles, amplifiers are required with the usual attendant difficulties.

Various types of capacity strain gauge developed for aircraft work have been described[22] and the serrated condenser pick-up (twist type) has been developed for fitting into the pinion-driving shaft of an aero-engine.

It must be appreciated that the use of a capacity-type pick-up involves special care with details of the circuit and that the gauges must be hermetically sealed to exclude oil and dirt; also, the stray capacities of electric connexions and mechanical supports are major difficulties.

These considerations have tended to restrict the application of this type of gauge to certain special cases, where the temperature conditions are such as to preclude the employment of wire resistance gauges or where the conditions are so severe that wire resistance gauges would be overstrained. Neither of these difficulties apply to the measurement of torque in marine installations, but the general characteristics of this type of instrument seem to make it more suitable for use in the laboratory than on board ship.

Electrical-resistance wire strain gauges[23] have been widely used in recent years for the measurement of both steady and rapidly varying strains, since their simplicity and small bulk render them attractive for many engineering purposes. The principle employed is the change of resistance of a conductor with change of strain.

A flattened coil of wire is cemented to the surface of a sheet of thin paper, which serves to insulate it from the test surface and to hold the wire in its correct position. The gauge is glued to the surface under investigation. Strain of the test object is then transmitted almost completely to the gauge wire and causes a change of electrical resistance. Except where thermal-setting cements are used, a gauge cannot be removed from a surface and re-cemented. Practical application depends therefore upon their uniform and reproductive properties or a calibration *in situ*.

One arrangement of strain gauge suggested[14, 24] for torque measurement on a rotating member is shown in Fig. 10. Four

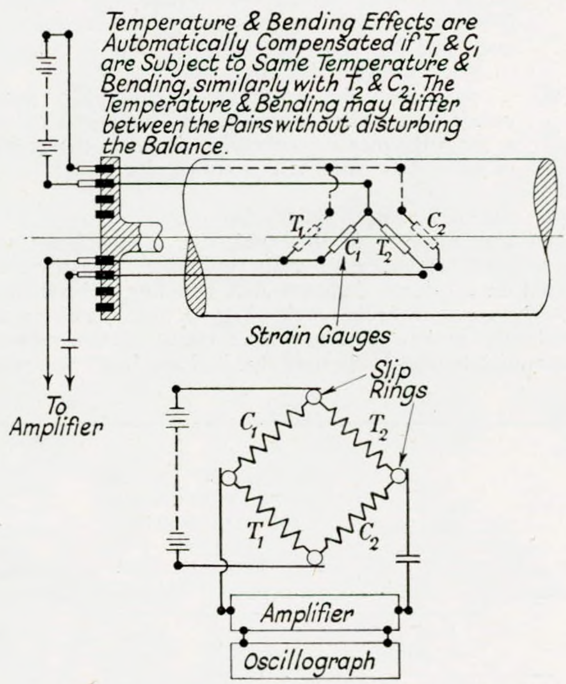


FIG. 10—*Wire resistance torsional strain gauge*

so that symmetry exists relatively to both the shaft axis and a plane perpendicular to the axis. These gauges form the four arms of a Wheatstone bridge. Connexion from the gauges to the bridge circuit is made through slip rings and brushes. This arrangement of gauges gives four times the output of a single gauge and thus reduces the effect of variable and erratic brush-contact resistance; it also eliminates the effect of bending or thrust in the shaft.

At the time the survey was undertaken (1946) this method of measuring torque did not appear to have been tried out on board ship.

TESTS OF SIEMENS-FORD TORSIONMETERS

As a result of the foregoing survey it was felt that the general characteristics of the Siemens-Ford torsionmeter were such as to merit further investigation particularly as to the accuracy which might be expected under sea trial and sea service conditions. It was considered that such tests should be particularly designed to show whether the torsionmeter gave the true average of a fluctuating torque and also whether any shift of zero was likely to occur when the instrument was subjected to high values of angular acceleration such as are encountered with marine oil engine installations when passing through critical speeds.

The tests were carried out by B.S.R.A. in collaboration with Messrs. Siemens Bros., and the necessary strain gauge and Geiger torsiograph measurements were made by the Research Department of Lloyd's Register.

Three groups of tests were undertaken, viz:—

(i) Tests on a small shaft transmitting power from a medium speed oil engine to an hydraulic brake. This arrangement meant waiving the use of large size torsionmeters but it had the advantage that the shaft system could be specifically designed to give the desired dynamic conditions.

(ii) Tests on full size intermediate shafts inserted between slow speed marine Diesel engines and hydraulic brakes. Three such tests were made. Two of these were similar and produced conditions where the shaft stress ranged from about 2,500lb. per sq. in. to about 4,500lb. per sq. in., a range which covers the full power condition in most marine installations. In vessels with Diesel machinery aft, the full power stresses are frequently lower and the third test was arranged with such low-stressed shafting.

(iii) Tests during the sea trials of electrically propelled vessels where shaft horse-power could also be arrived at from the measured electrical input to the propulsion motor. Two such tests were carried out.

(i) *Test bed trial using a small sized shaft*

For the purposes of this trial, one of the latest pattern Siemens-Ford torsionmeters with twin shaft transformers was mounted on a 4-inch diameter shaft coupling a medium speed eight-cylinder four-stroke cycle single-acting Diesel engine to an hydraulic brake. The two mass elastic system represented by the engine flywheel, auxiliary flywheel and brake was specially

TABLE 2

Siemens-Ford torsionmeter full scale reading	Angular acceleration at critical speed	Alteration of zero after running on critical speed for 2 to 3 minutes		Alteration of zero after load tests involving passing eight times through the critical speed
		1st period	2nd period	
	Rad. per sec. per sec.	Scale divisions	Scale divisions	Scale divisions
100 scale divisions	98	$\frac{3}{4}$	$\frac{1}{10}$	nil

values of angular acceleration and also so that the meter could be tested under conditions of torque fluctuation comparable with those encountered in Diesel installations.

Two main series of tests were carried out. In the first of these the torsionmeter was subjected to several periods of operation at the critical speed of the shaft system and examined for consistency of zero. The results are given in Table 2.

It will be seen that a slight zero shift was experienced during the first 2-3 minutes running under these conditions and that after a second and similar period of operation the further alteration of zero was negligible. Finally, during subsequent tests, which involved passing through the critical speed eight times, no further alteration of zero could be detected. The value of angular acceleration imposed during these tests may be considered to be as severe as any likely to be encountered in the intermediate shafting of marine Diesel installations in all but very exceptional cases. The results may therefore be considered satisfactory.

In the second series of tests the values given by the torsionmeter were compared with those given by the brake over a range of loads at a constant speed of approximately 190 r.p.m. The results are shown in Table 3. The meter and shaft were calibrated together in the usual manner by applying a range of static torques.

TABLE 3

Nominal load	Shaft speed, r.p.m.	B.h.p. from brake	Mean skin stress in shaft, lb. per sq. in.			Per cent variation of stress (from electric strain gauges)
			(a) From brake	(b) From torsion-meter	Per cent difference $=\frac{(b)-(a)}{(a)}\times 100$	
Full	191·0	93·6	2,575	2,601	+1·0	±51
	190·5	93·3	2,575	2,593	+0·7	±50
$\frac{3}{4}$	190·5	70·5	1,945	2,037	+4·7	±58
	191·0	70·6	1,945	2,025	+4·1	±53
$\frac{1}{2}$	188·5	47·2	1,313	1,403	+6·8	±86
	190·0	47·5	1,313	1,403	+6·8	±80
$\frac{1}{4}$	188·8	22·65	631	679	+7·6	±167
	189·3	22·7	631	679	+7·6	±167

It will be seen that comparison has been made upon a basis of mean skin stress. This basis has been chosen since the accuracy of a torsionmeter is closely related to skin stress. Instrumental errors tend to remain relatively constant in magnitude irrespective of stress and consequently, when expressed as a percentage, may be expected to vary inversely as the stress. An examination of the shaft stresses in a number of typical vessels has given values varying from 1,000lb. per sq. in. to 4,700lb. per sq. in. Thus, the mean skin stress value of about 2,600lb. per sq. in. obtained in these tests at "full load" is considerably lower than would be encountered in some vessels and higher than in others.

The values given by the Siemens-Ford torsionmeter are higher throughout than those given by the brake, the difference varying from approximately 1 per cent at nominal full load to 7·6 per cent at $\frac{1}{4}$ nominal full load. It was considered that these differences could be largely accounted for partly by possible inaccuracies in the brake and partly by frictional losses in the bearings which had to be employed to support the shafting.

It will be noted that the dynamic fluctuation of stress during the tests varied from approximately ± 50 per cent at nominal full load to approximately ± 170 per cent at $\frac{1}{4}$ nominal full load; such conditions may be taken to be at least as severe as those encountered on intermediate shafting on board ship in all but exceptional cases.

TABLE 4

Nominal load	R.p.m.	Shaft horse power			Skin stress in shaft (from electric strain gauges)		
		(a) From brake	(b) From torsionmeter	Per cent difference $=\frac{(b)-(a)}{(a)}\times 100$	Steady stress, lb. per sq. in.	Dynamic stress, lb. per sq. in.	Per cent variation of stress
$\frac{1}{2}$	97	3,298	3,283	−0·45	2,872	± 669	±23
$\frac{3}{4}$	102	4,417	4,409	−0·2	3,672	±1,163	±31
Full	116	6,183	6,291	+1·75	4,504	±2,608	±58

(ii) *Test bed trials on full size intermediate shafts*

Two trials were carried out during the shop tests of marine Diesel main propelling engines on which the shaft stress at full load was approximately 4,500lb. per sq. in. In each case the torsionmeter was of the single transformer type and was mounted on a $15\frac{3}{4}$-inch diameter shaft interposed between a 6,500 b.h.p. opposed-piston engine and an hydraulic brake. Observations were made over a range of engine speeds adjusted in approximate accordance with the simple propeller law:—torque = constant × speed^2 assuming constant slip. In each case the meter and shaft had previously been calibrated together over a range of statically applied torques. The results are shown in Tables 4 and 5 respectively.

TABLE 5

Nominal load	R.p.m.	Shaft horse power			Per cent variation of stress in shaft (from electrical strain gauges)
		(a) from brake	(b) from torsionmeter	Per cent difference $=\frac{(b)-(a)}{(a)}\times 100$	
$\frac{1}{2}$	93·8	3,305	3,357	+1·6	±17
$\frac{3}{4}$	111·3	4,903	4,880	−0·5	± 5·3
Full	120·7	6,445	6,494	+0·7	±10·2

It will be seen that the brake and torsionmeter readings agreed to within ± 2 per cent and indeed the differences in many cases were within the probable limits of accuracy of the brake. It is also of interest to note that during the first of these tests appreciable vibration stresses were present at full load due to a sixth order vibration.

The third trial, with low stressed shafting, was carried out during the test bed trials of a four-cylinder 4,500 b.h.p. opposed-piston oil engine. The torsionmeter, which was of the latest twin transformer type, had previously been calibrated *in situ* on the $19\frac{1}{2}$-inch diameter shaft which was subsequently inserted between the engine and brake. In cases where it is known that the shaft stresses are low it is the practice of the makers to fit oversized armatures to the shaft transformers which has the effect of reducing the air gap and increasing the sensitivity and this arrangement was accordingly employed. A galvanometer of higher sensitivity than that normally employed was also used during some of the tests.

TABLE 6. STANDARD 0·050 INCH PITCH MICROMETER, OVERSIZE ARMATURES, HIGH SENSITIVITY GALVANOMETER

Shaft speed, r.p.m.	Torsionmeter scale reading	Mean skin stress in shaft, lb. per sq. in.		
		(a) From brake	(b) From torsionmeter	Per cent difference $=\frac{(b)-(a)}{(a)}\times 100$
114·5	56·0	1,683	1,671	−0·6
113·5	50·0	1,517	1,495	−1·4
117·0	43·5	1,335	1,300	−2·6
115·0	37·6	1,135	1,105	−2·6
115·0	32·8	1,015	983	−3·1
115·0	28·0	865	834	−3·5

Observations were made at a constant speed of about 115 r.p.m. at shaft stresses ranging from 865 to 1,683lb. per sq. in. A test was also carried out to determine the effect of speed on the meter reading. The results are shown in Tables 6, 7 and 8.

TABLE 7. STANDARD 0·050 INCH PITCH MICROMETER, OVERSIZE ARMATURES, MEDIUM SENSITIVITY GALVANOMETER

Shaft speed, r.p.m.	Torsionmeter scale reading	Mean skin stress in shaft, lb. per sq. in.		
		(a) From brake	(b) From torsionmeter	Per cent difference $=\frac{(b)-(a)}{(a)}\times 100$
116·5	32·8	1,015	983	−3·1
115·5	28·0	865	834	−3·5

TABLE 8. STANDARD 0·050 INCH PITCH MICROMETER, OVERSIZE ARMATURES, MEDIUM SENSITIVITY GALVANOMETER

Shaft speed, r.p.m.	Torsionmeter scale reading	Mean skin stress in shaft, lb. per sq. in.		
		(a) From brake	(b) From torsionmeter	Per cent difference $=\frac{(b)-(a)}{(a)}\times 100$
116·5	32·8	1,015	983	−3·1
69	32·6	1,015	977	−3·7
41	32·6	1,015	977	−3·7

Considering first the results obtained when employing oversized armatures, it will be seen from Table 6 that as the load was reduced the percentage difference between torsionmeter and brake values increased steadily from − 0·6 per cent to − 3·5 per cent, the former value corresponding to shaft stress of 1,683lb. per sq. in. and the latter to 865lb. per sq in. These results were obtained using a high sensitivity galvanometer but if they are compared with those given in Table 7 it will be seen that identical values were obtained using a medium sensitivity galvanometer. Although the galvanometer sensitivity had thus no apparent effect upon accuracy, it was observed that the increased "liveliness" and improved pointer design of the higher sensitivity galvanometer made for easier and quicker null-point determination.

Examination of the results given in Table 8 shows that the effect of speed is small, a reduction from 116 r.p.m. to 41 r.p.m. leading to an increase of error of only $\frac{1}{2}$ per cent. This may have been due to difficulty in observing the null-point at the lowest speed, cyclic torque fluctuations causing pointer movement.

(iii) *Tests during sea trials of electrically propelled vessels*

During an extensive series of sea trials of a turbo-electric tanker, the opportunity was taken to obtain a comparison between the values of shaft horse-power given by a Siemens-Ford torsionmeter and those arrived at from the electrical power input to the propulsion motor.

TABLE 9.

Run No.	Mean r.p.m.	Mean Siemens-Ford scale readings	(a) S.h.p. by propulsion motor (assumed efficiency 97 per cent)	(b) S.h.p. by torsion-meter (assumed value of G=11·95×10⁶ lb. per sq. in.)	Per cent difference $=\frac{(b)-(a)}{(a)}\times 100$
1	95·3	42·0	3,420	3,520	+2·9
2	96·1	42·5	3,530	3,560	+0·9
3	104·7	51·4	4,630	4,707	+1·7
4	103·7	51·5	4,631	4,660	+0·6
5	105·5	52·0	4,684	4,780	+2·1
7	105·3	51·5	4,702	4,740	+0·8
8	105·5	51·0	4,683	4,700	+0·4
9	116·0	64·3	6,324	6,500	+2.8
10	116·0	63·3	6,304	6,418	+1·8
11	121·5	70·9	7,376	7,510	+1·8
12	121·8	70·5	7,340	7,470	+1·8
13	90·4	38·6	2,941	3,048	+3·6
14	90·8	38·0	2,940	3,020	+2·7
15	106·2	53·0	4,790	4,910	+2·5
16	106·7	52·0	4,786	4,860	+1·5
17	116·6	62·0	6,170	6,310	+2·6
18	117·3	61·4	6,146	6,280	+2·2
19	125·8	72·0	7,726	7,900	+2·3
20	125·8	73·4	7,852	8,058	+2·6
21	95·3	44·4	3,642	3,683	+1·1
22	96·0	44·4	3,660	3,685	+0·7
23	104·0	53·0	4,750	4,810	+1·3
24	104·1	51·8	4,680	4,710	+0·6
25	114·2	64·0	6,150	6,390	+3·9
26	114·1	63·2	6,180	6,290	+1·8
27	122·2	74·4	7,748	7,938	+2·5
28	123·0	74·3	7,756	7,970	+2·8

The Siemens-Ford torsionmeter was of the latest twin shaft transformer type and had been installed by the makers while a special sub-standard Weston kilowatt-meter had been installed by the propelling machinery contractors.

The results of twenty-seven runs over the measured mile at various powers under three conditions of loading are shown in Table 9.

The shafting stress during these trials varied over the range 2,100 to 4,100lb. per sq. in. It will be seen that the values of shaft horse-power registered by the torsionmeter were throughout higher than those calculated from the electrical input to the propulsion motor and the known efficiency of this motor; the differences ranged from 0·4 to 3·9 per cent and had no direct relation to the power. Two facts have to be borne in mind in connexion with these differences. In the first place the measurement of electrical input to the propulsion motor is subject to possible error estimated by the makers at ± 2 per cent; in the second place, it was not possible to calibrate the Siemens-Ford torsionmeter on the shaft against static torques and in consequence the modulus of rigidity had to be assumed at $11·95 \times 10^6$lb. per sq. in. and the meter constant calculated accordingly. With these facts in mind it will be seen that the results are not inconsistent with an accuracy of ± 2 per cent with the Siemens-Ford instrument.

During another series of sea trials on a Diesel-electric tanker a Siemens-Ford torsionmeter of the latest twin shaft transformer type had been installed and a special sub-standard kilowatt-meter of known accuracy had been installed by the manufacturer of the propelling motor.

Table 10 shows the values of shaft horse-power obtained from the two sets of instruments during eight runs over the measured mile in fine weather and with a calm sea.

As in the case of many vessels with machinery aft, the intermediate shafting was of such a size that skin stresses were relatively low, the estimated skin stress at full power being 1,720lb. per sq. in. Moveover, the oversize armatures were not fitted so that the meter scale readings were small. It should be noted further that facilities for calibrating the meter on the intermediate shaft against known torques were not available and in consequence the modulus of rigidity has had to be assumed at $11·95 \times 10^6$lb. per sq. in. and the meter constant calculated accordingly.

It will be seen from the table that the values of shaft horse-power as registered by the torsionmeter were lower throughout than those calculated from the electrical input and motor efficiency, the difference amounting to 4 per cent at half power, to 2½ per cent at three-quarter power and to 0-2 per cent at full power. In view of the considerations enumerated above, this must be considered reasonable agreement.

CONCLUSIONS

On the basis of the trials described, it was concluded that the Siemens-Ford torsionmeter is capable, in marine installations, of an accuracy of at least ± 2 per cent over a range of shaft stress from about 2,500lb. per sq. in. to about 4,500lb. per sq. in. This range covers the full power condition in most marine installations. In vessels with Diesel machinery aft, the full power stresses are frequently lower and in these cases somewhat less accurate results may be expected.

It is also concluded that the clamping arrangements of the meter are adequate to withstand the acceleration forces encountered in marine Diesel installations in all but very exceptional cases. The latter conclusion has been reinforced by recent experiences with this torsionmeter during acceptance trials.

EXISTING TYPES OF THRUSTMETER

(i) *Instruments in which the Thrust is Measured by the Counterbalancing of a Weight*

Thrustmeters in which the thrust is measured by the counterbalancing of a weight are fairly simple and relatively inexpensive for small thrusts, and they frequently find application in model test work. When designed for large thrusts, however, they tend to become very complicated and expensive. Moreover, to be certain that the calculated linkage multiplication and freedom from frictional effects are obtained calls for

TABLE 10

Run No.	Mean r.p.m.	Mean Siemens-Ford scale reading	Propulsion motor efficiency	(a) S.h.p. by propulsion motor	(b) S.h.p. by torsionmeter (assumed value of G=11·95×10⁶-lb. per sq. in.)	Per cent difference $=\frac{(a)-(b)}{(a)}\times 100$
1	96·2	17·0	96·0	1,867	1,793	−4·0
2	96·9	17·0	96·0	1,874	1,805	−3·7
3	96·0	17·0	96·0	1,852	1,788	−3·5
5	111·0	23·5	96·8	2,933	2,860	−2·5
6	112·5	23·4	96·8	2,966	2,886	−2·7
7	119·0	27·4	97·0	3,642	3,575	−1·8
8	119·0	27·4	97·0	3,571	3,575	negligible
9	120·5	28·1	97·0	3,753	3,712	−1·1

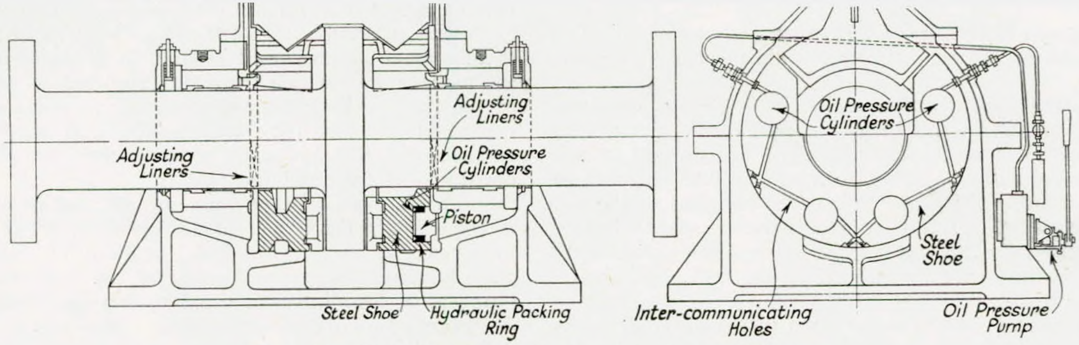


FIG. 11—*Michell thrust block with thrust indicator*

calibration—a process which is more difficult and expensive than that involved in calibrating torsionmeters and hence very rarely carried out. These considerations, together with the development of the relatively simple hydraulic type of thrustmeter, are no doubt responsible for the fact that in recent years the gravity type of thrustmeter appears to have been abandoned for ship use.

(ii) *Instruments in which the Thrust is Measured Hydraulically*

Instruments in which the thrust is opposed by a force due to hydraulic pressure have been the subject of a good deal of experimental and development work in recent years and at least one thrustmeter of the type is commercially available, namely the Michell thrustmeter.[3] This consists of a modification to the standard Michell thrust block, which involves the replacement of the usual cast-iron thrust shoe by a special forged-steel shoe of the same size and the fitting of a hand-operated oil-pressure pump, calibrated pressure gauge, relief valve, and oil piping. The arrangement is shown in Fig. 11.

On the back of the forged-steel thrust shoe are the usual adjusting liners, and the front is machined to take the thrust pads. Holes are machined in the back to form oil-pressure cylinders which are coupled together by a series of drilled holes. Under pressure from the hand pump, the thrust pistons are forced forward and gradually transfer the thrust load from the liners to themselves. The thrust shoe carrying the ahead pads then floats on the thrust pistons and the direct thrust is indicated on the pressure gauge. To guard against over-pumping, a special relief valve in one piston acts when half the axial clearance between the ahead and astern shoes is traversed. Astern thrust can be measured similarly by inserting an astern-indicating shoe.

This instrument is simple and robust and easily operated by engine-room staff. Moreover, the components can readily be transferred from one vessel to another employing the same size of thrust block. As regards applicability to various types of machinery installation, there are limitations. It can readily be fitted to machinery installations incorporating a separate Michell thrust block; and in turbine installations having the thrust block incorporated in the gear casing, it can also, in the majority of cases, be accommodated. In oil-engine installations, however, it is becoming increasingly common practice to incorporate the thrust block in the after end of the engine, and it is then not usually possible to install the Michell thrust-measuring elements.

The only criticism that has been levelled against this type of meter is that friction between the pistons and cylinders may give rise to erratic readings. On the other hand, there is little evidence of this in the records given in the reports of the Marine Oil-Engine Trials Committee dealing with the three trials in which this type of thrustmeter was fitted; indeed the sensitiveness of the hydraulic system is well illustrated by the

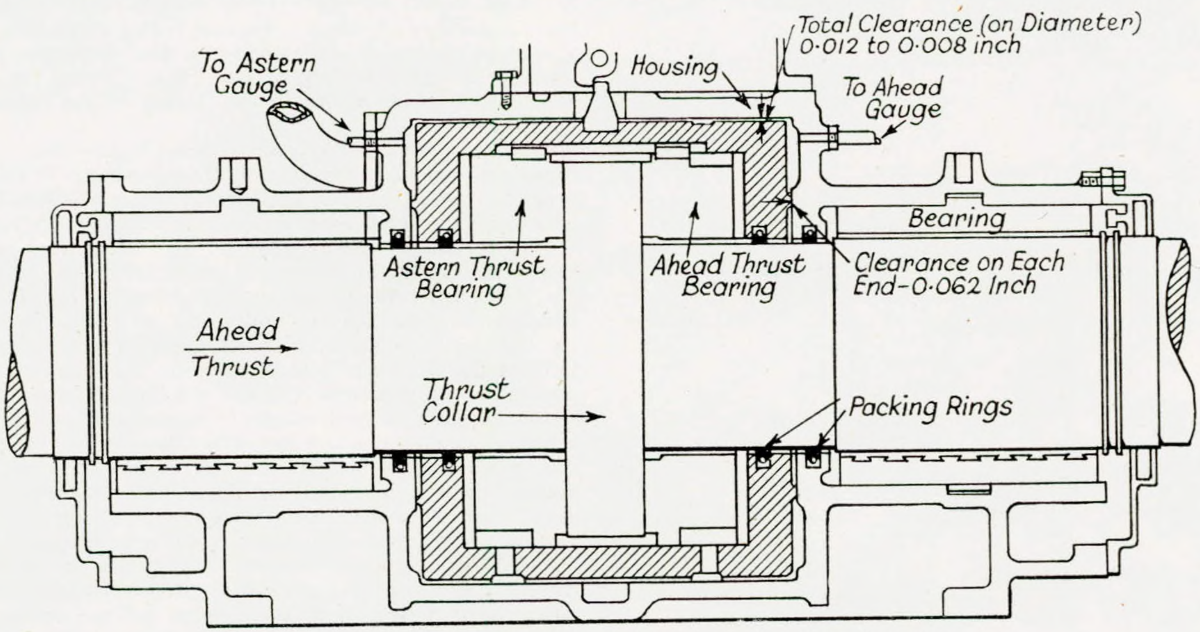


FIG. 12—*Westinghouse Electric and Manufacturing Co. thrustmeter*

The German "Simplex" thrustmeter,[7] which is of a similar design to the Michell, has been used in German merchant vessels.

The Westinghouse thrustmeter[3, 25, 26] is shown diagrammatically in Fig. 12. In this meter the true thrust block is the inner cage which carries the thrust pads. The outside of this cage is cylindrical and acts as a piston which slides in an outer cylinder or housing. Oil under pressure is supplied to the two ends of the housing, through control valves operated by the cage and arranged so that the cage is always being forced off the stops. The difference in pressure at the two ends gives a measure of the thrust.

A thrustmeter of this type was fitted in the s.s. *Clairton* by the United States Shipping Board for special trials which were conducted with that vessel. It was found that the meter functioned well during the trials, but it was clearly demonstrated that in its form at that time it was not a sea-going unit. Firstly, the oil supply through the double control valves could not be supplied at a sufficiently rapid rate to prevent the cage from bumping the ends of the housing cylinder when the ship

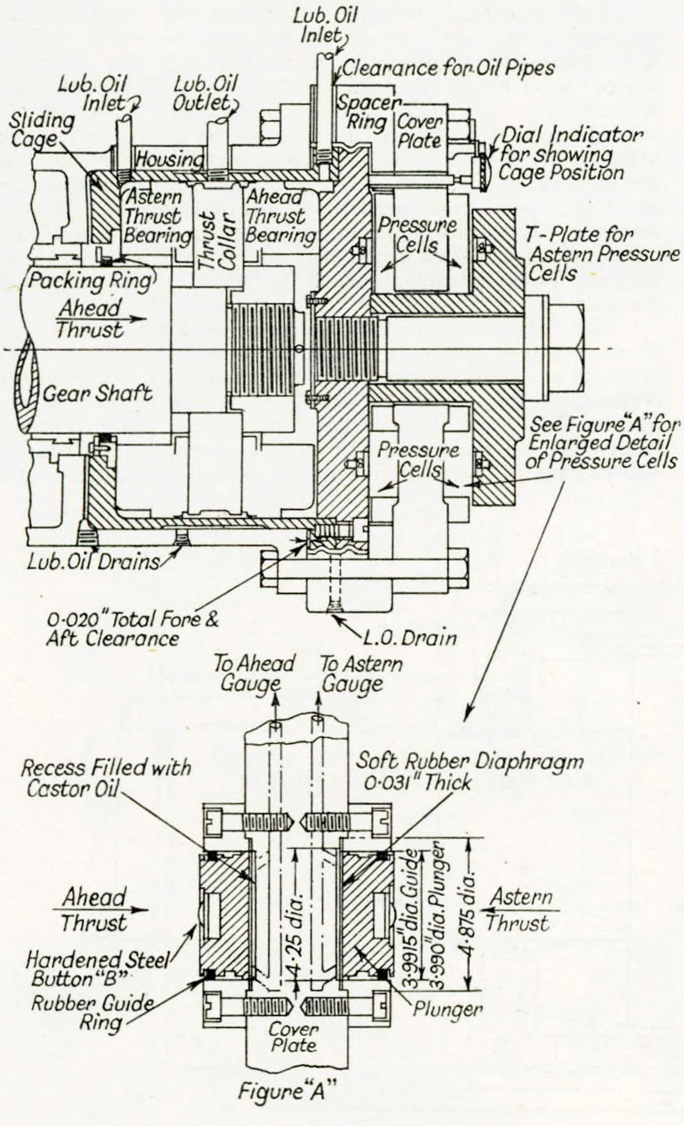


FIG. 13—*Kingsbury pressure cell thrustmeter*

pattern is that it eliminates the possibility of error from friction of piston packings. On the other hand, it would seem that there is an equal chance of binding in this type of meter due to the pins or lugs which must be employed to prevent rotation of the cage. Moreover, the design is more complicated, lacks the adaptability of the Michell type, and involves the maintenance of oil-tight seals between the stationary cage and housing and the revolving shaft.

The Kingsbury type pressure cell thrustmeter,[3, 25, 27] used in the trials of the U.S.S. *Hamilton* is shown in Fig. 13. The thrust bearing proper is housed in a sliding cage and a heavy plate forming the forward cover of the cage bears against the plungers of four interconnected hydraulic pressure cells which are mounted on a fixed cover plate. Movement of the plungers deflects thin rubber diaphragms which form the walls of the oil cells. For astern-thrust measurement, a similar arrangement of cells is mounted on the other side of the fixed cover plate and a T-plate, connected to the cage by a bolt, acts on the plungers of these latter cells. Relative cage position is read on a dial micrometer, and when the cage is forced to the mid-position the pressure reading is a measure of the thrust.

Although this design eliminates the closely fitting pistons of the Michell type, it introduces the necessity for a lug to prevent rotation of the cage, so that on the score of sticking there would seem to be little to choose between them. Moreover, the possibility of deterioration of the diaphragm cannot be ruled out, and although this is not perhaps a serious objection for trial purposes where calibration can be made before and after use, it certainly constitutes a serious drawback for continuous service.

This type of thrustmeter lends itself to ease of adaptation with the particular design of thrust bearing in which it was employed but not to the widely used Michell thrust block.

(iii) *Instruments in which the Thrust is Measured by the Deformation of an Elastic Member inserted in the Shafting*

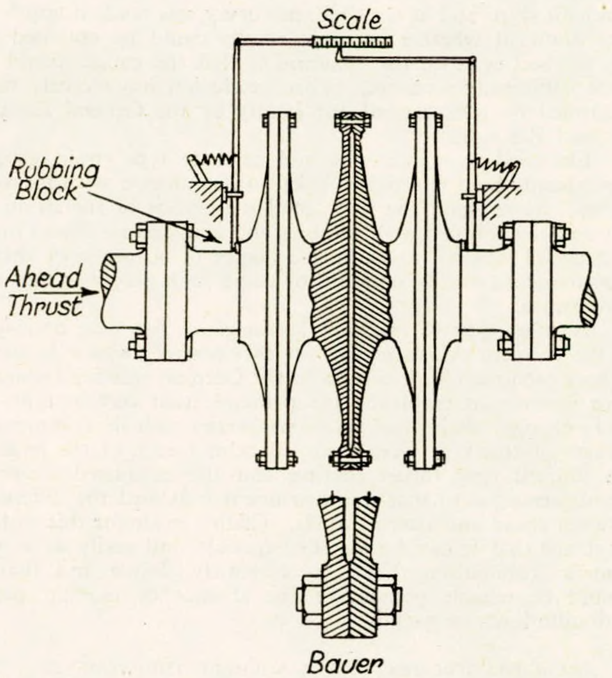
Bauer, Denny and Edgecombe, and Parsons and Cook,[3, 25] among others have suggested or designed instruments using a bellows arrangement. Two of the types mentioned are shown in Fig. 14. In each case the principle is the same; only in detail is there any difference in design. The Bauer and Denny and Edgecombe thrustmeters consist of several pairs of flexible disks or plates bolted together, the movement when under thrust being a measure of thrust. Because bolted disks have the disadvantage of introducing a varying and uncertain degree of elasticity at the boundary of each disk, Parsons patented the solid forged design shown in Fig. 14 but it does not appear to have been tried out.

None of these bellows types seems to have been successful because of various factors, mainly the uncertainty of their elastic characteristics, difficulties of manufacture, and the difficulty of measuring the small axial movements during shaft rotation.

It should be noted, moreover, that these types of thrustmeters involve the insertion of a special length of shafting.

The Denny marine thrustmeter[3] shown in Fig. 15, uses the deflexion of coiled springs under load as a basis for measurement of thrust. Two circular steel frames attached to the inner faces of the flanges of a shaft coupling carry the springs, and the deflexion is measured optically at a number of points spaced round the shaft circumference. Observation at each point gives a mean value over a number of revolutions for the thrust at that point, and the mean of the mean values obtained gives the average thrust. The coil springs are in compression and in equilibrium when the shaft is at rest, and this position is taken as zero. Any deflexion on rotation is due to thrust.

As the instrument has to transmit torque while simultaneously allowing no frictional constraint to axial motion, an arrangement is used of rollers inserted between the dogs alternately attached to the forward and aft frames. The steel frames form a spigot between the two shaft lengths.



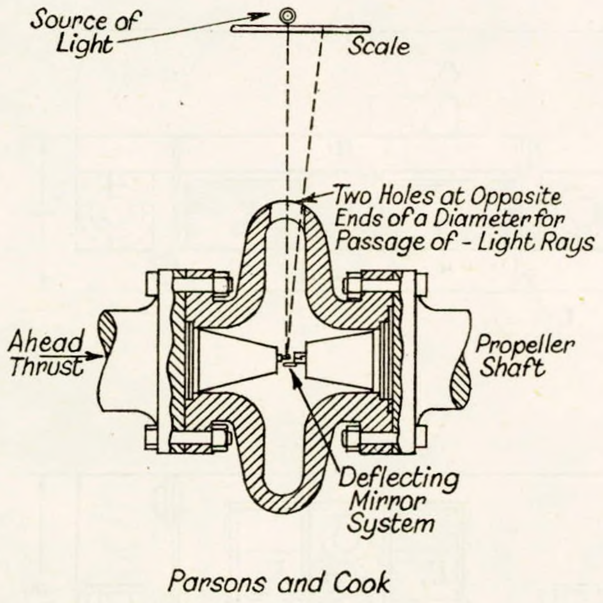


FIG. 14—*"Bellows" type thrustmeter*

The chief disadvantage of this thrustmeter for trial purposes would seem to be that its installation calls for a special length of shaft; for permanent installation this objection does not of course apply.

(iv) *Instruments Employing Electrical Strain Gauges*

Operating on change of inductance, the Westinghouse Magnetic Strain Gauge[28] shown in Fig. 16, has been used on the trials of an American Coast Guard cutter,[29] for simultaneous torque and thrust measurement. Two laminated iron cores are rigidly attached to a small frame and a laminated iron armature is attached to another, the frames being connected by leaf springs so that the armature is interposed between the cores. Clearance allows armature movement. When motion of one frame occurs relative to the other, one air gap between armature and core increases while the other decreases. This changes the reluctance of the magnetic paths and consequently the impedance of coils on the cores, these coils and an adjustable centre-tap inductance being connected in a bridge circuit energized by alternating current. When unbalancing of the

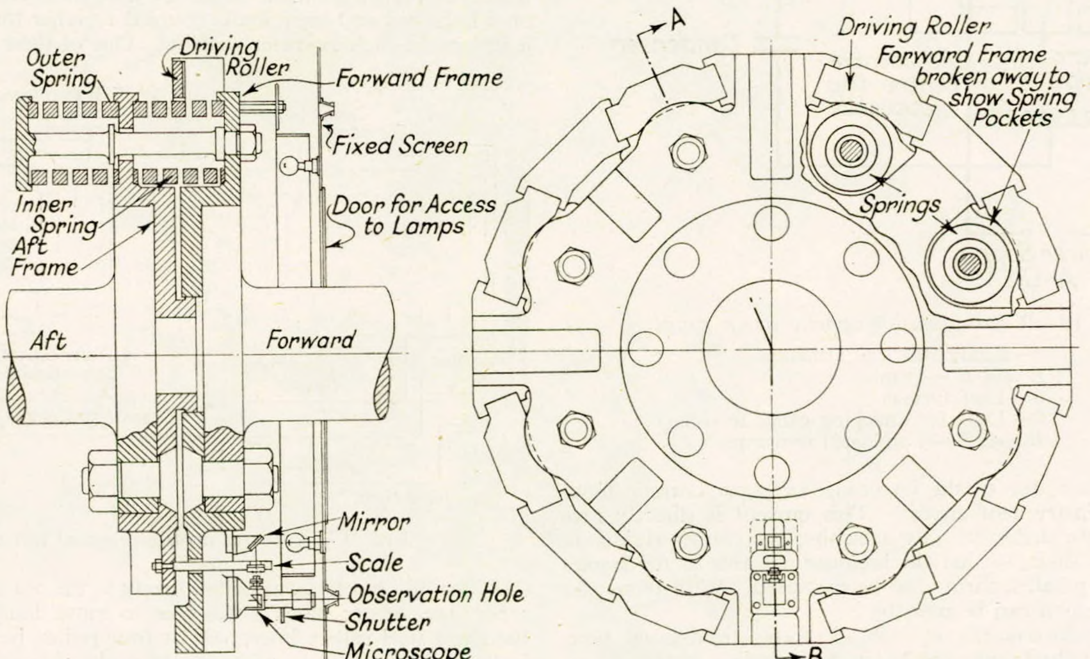


FIG. 15—*Denny marine thrustmeter*

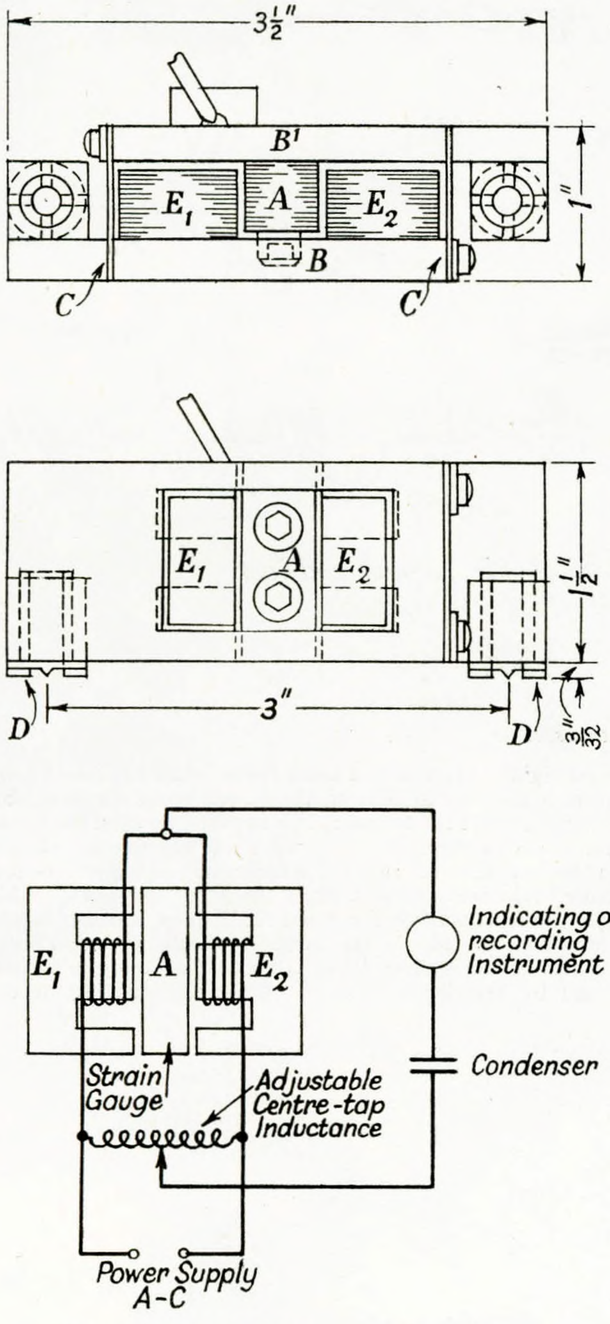


FIG. 16—*Westinghouse magnetic strain gauge*

A—Laminated iron armature
B and *B*¹—Frames
C—Leaf springs
D—Lugs for attaching gauge to collars
E_1 and E_2—Laminated iron cores

coils takes place due to the changing air gaps, current flows through the instrument circuit. This current is directly proportional to the deflexion. By attaching the frames rigidly to collars on the shaft, so that the longitudinal axes of the frames and shaft are parallel, thrust can be measured. With these axes at 90 deg., torsion can be measured.

With this instrument, as with all of the strain-gauge type, the greater the shaft stress the better the operation of the instrument; and in the case quoted,[29] a special hollow shaft section was installed for the trials, so that the compressive stress might be favourable for operation of the gauge.

Reference has already been made to the extremely small magnitude of the compressive strain in the line shafting of merchant ships and at the time the survey was made it appeared very doubtful whether accurate results could be obtained by this method even on the assumption that the gauges could be made sufficiently sensitive. This conclusion has recently been confirmed by tests carried out jointly by the General Electric Co. and B.S.R.A.

Electrical resistance wire and capacity type strain gauges have already been referred to briefly in connexion with torsionmeters. Here again the very small magnitude of the strain to be measured coupled with bending and temperature effects make it doubtful whether a reasonable degree of accuracy of thrust measurement could be obtained by using such gauges to measure shaft strain.

Another type of magnetic strain gauge has been employed in the Siemens Magnetostriction thrustmeter[7] which is stated to have been used with success in the German steamer *Potsdam*. This instrument employed the principle that certain types of steel change their magnetic properties when compressed. Gauges of this type were mounted behind each of the pads of the Michell type thrust bearing and the associated electrical circuit arranged so that the instrument indicated the difference between ahead and astern thrusts. Claims made for this instrument are that it can be installed quickly and easily as a permanent installation, that it is extremely simple and that it should be reliable because of the absence of moving parts, hydraulic leads or packing.

CALIBRATION TESTS OF A MICHELL THRUSTMETER

Following the survey already outlined it was decided to carry out some test bed trials with a Michell thrustmeter. The object of these trials was to ascertain the accuracy of this form of thrustmeter under conditions approximating as closely as possible to those encountered in service and they were carried out by B.S.R.A. in collaboration with Michell Bearings, Ltd., and the Research Department of Lloyd's Register.

Description of Test Rig

The layout of the test rig is shown in Fig. 17 and in Figs. 18 and 19 (Plates 1 and 2). Two Michell No. 2B type thrust blocks, each fitted with a 6¼-inch diameter thrust shaft and incorporating thrust-meter elements, were mounted back to back on a lathe bed and their shafts coupled together through a short length of 6¼-inch diameter shafting. One of these thrust blocks

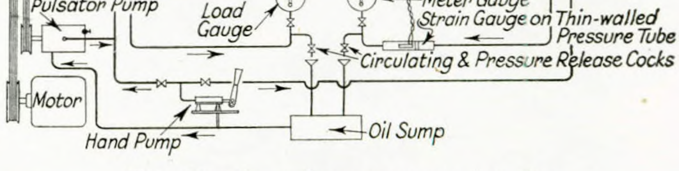


FIG. 17—*General arrangement of testing*

(the "loading block") was bolted firmly to the test bed whilst the other (the "meter block") was free to move longitudinally on hardened steel rollers interposed at four points between the test bed and the machined surface on the under side of its base. The casings of the two thrust blocks were connected together by two steel tie rods on which were mounted electric resistance wire strain gauges.

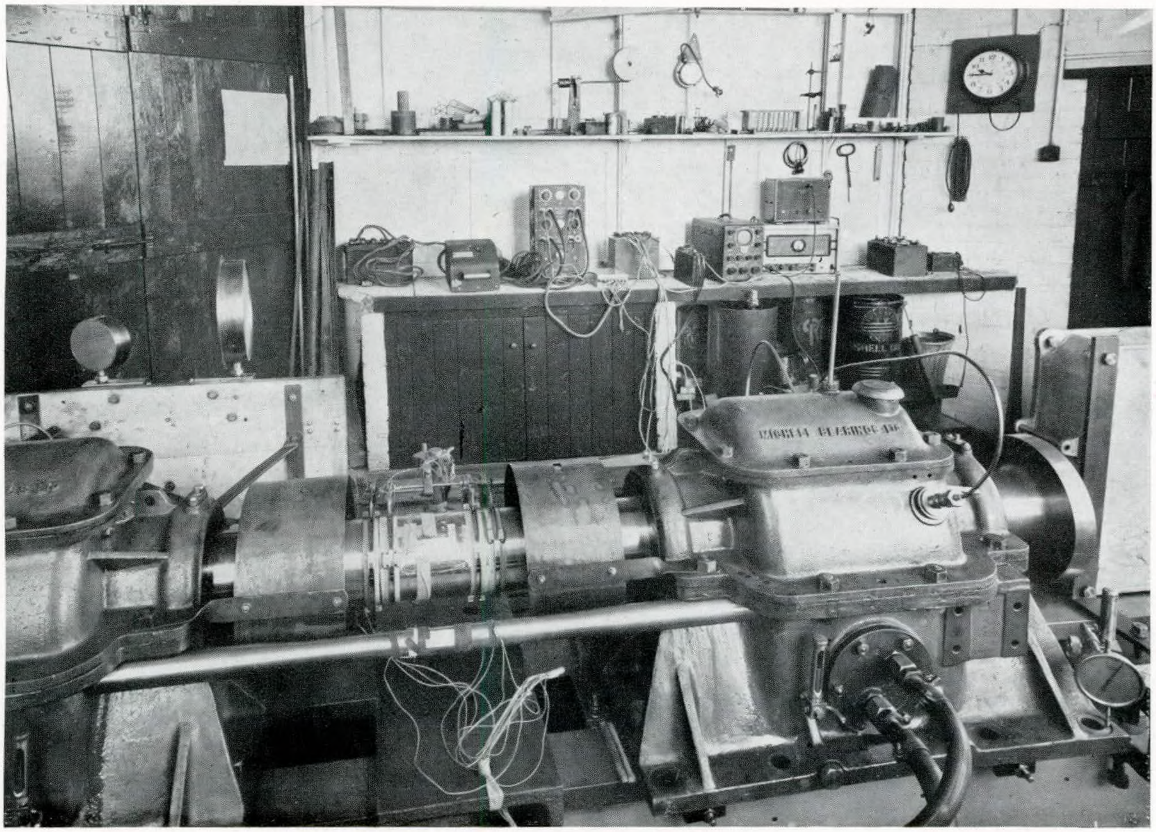


Fig. 18

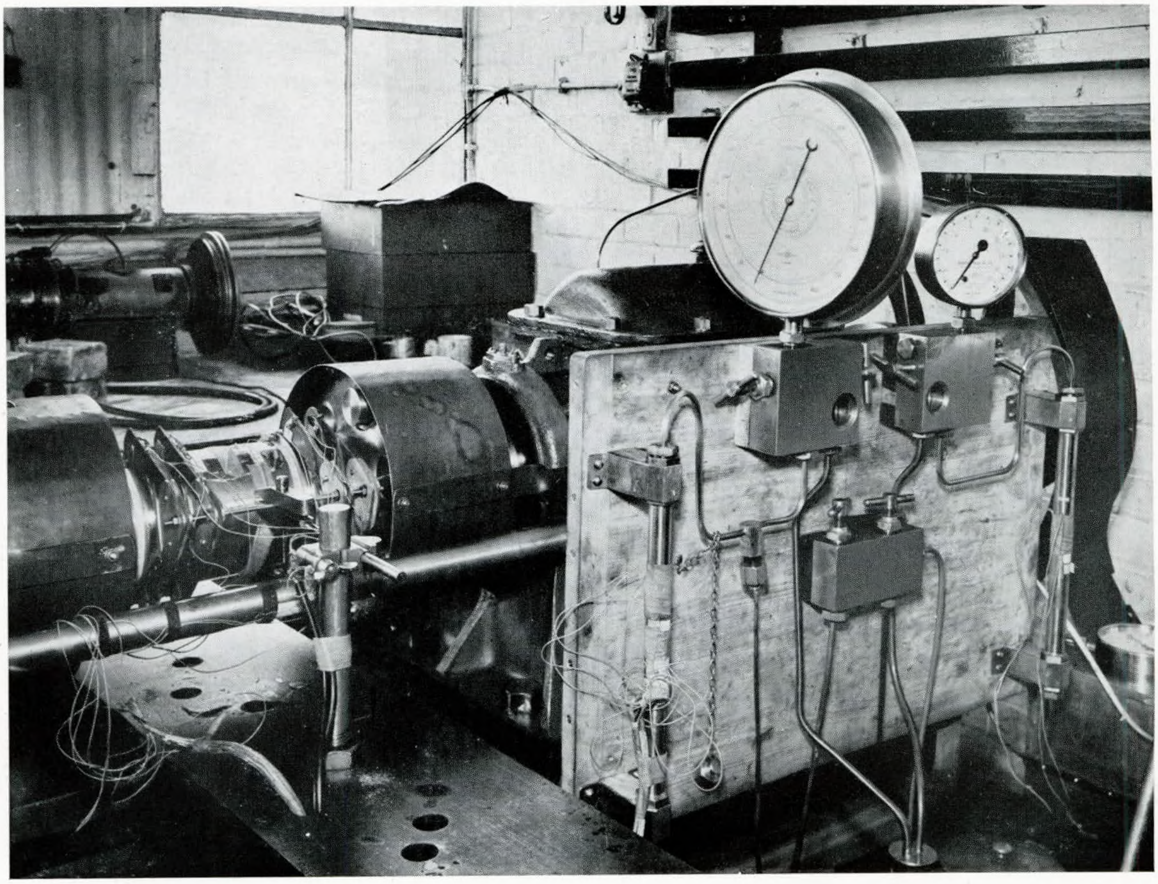


FIG. 19

first place it served, through the medium of its thrust elements, as a means of applying the required thrust loading to the meter block which contained the thrust meter under test; secondly, it provided an anchorage for the tie rods the tensile stress in which would be a measure of the applied thrust.

The steel tie rods were located at the vertical centre of pressure of the thrust pads so as to avoid bending in the vertical plane. Absence of bending in the horizontal plane was ensured by careful initial adjustment of tie-rod length so that each rod was loaded equally; this was checked by clock gauges placed at each side of the meter block.

The thrust-block shafting was fitted at the loading-block end with a pulley, driven by a constant-speed electric motor, at 240 r.p.m.

The hydraulic systems of each meter were cross-connected so that a common pressure could be applied by a hand-operated pump. In addition, a pulsator was coupled to the loading block system to enable a fluctuating pressure to be superimposed on the steady pressure. This pulsator consisted of a variable-stroke, single plunger driven at four times shaft speed from the thrust shaft through pulleys and belting, corresponding to the impulses of a four-bladed propeller. A thin-walled pressure tube, on which electric resistance wire strain gauges were mounted, was placed in the hydraulic system of the meter block to enable fluctuation of pressure to be recorded.

At their centres of length the two tie bars were turned down to 1⅝-inch diameter so as to provide highly stressed portions suitable for accurate strain measurement. On the reduced portions of both of these rods, two sets of electric resistance wire strain gauges were fitted. One set comprised Baldwin Southwark S.R.4, 120-ohm type A-1 gauges; these were coupled directly to a Baldwin Southwark strain indicator. The other set, comprising British Thermostat 2,400-ohm type SEA 11 gauges, were coupled in an a.c. bridge circuit fed with 200 c.p.s. carrier current at a supply voltage of 50 r.m.s. supplied from a low-distortion amplifier and stable oscillator. A four-channel M.I.T. recording oscillograph was used for photographically recording the dynamic strains registered by the British Thermostat gauges. Each type of resistance strain The reason for employing two entirely separate sets of electrical strain gauge elements lay in the fact that it was intended to carry out tests under both steady and fluctuating thrust conditions; the Baldwin Southwark equipment was considered to be the most accurate for the former condition and the M.I.T. equipment for the latter.

Mechanical extensometers of the Ewing type were made and fitted by Michell Bearings, Ltd., to each tie rod for the measurement of strain during the tests under static loading conditions.

Prior to the trials the strain gauges were fixed to the tie rods together with the mechanical extensometers and a static calibration was made in a 50-ton Avery testing machine over a range of load well in excess of that subsequently to be encountered in the actual tests. With all three types of strain recording equipment the individual points obtained lay very close to a straight line drawn between them and the origin.

Test Results

Two series of trials were carried out:—

(1) With a series of steady thrusts applied to the meter block.

(2) With a series of pulsating thrusts applied to the meter block.

(a) *Static load tests.* The results obtained under steady conditions of thrust are shown in Table 11. In the first column are given the hydraulic pressures registered by the standard pressure gauge attached to the meter block. The second and third columns give the tensile stress in the tie bars and the total thrust respectively as registered by the Baldwin Southwark indicator. The corresponding values of total thrust registered by the Michell thrustmeter, M.I.T. oscillograph, and mechanical extensometers are given in columns 4, 6 and 8 respectively. To facilitate comparison, the percentage difference between the thrust given by these three instruments on the one hand and the Baldwin Southwark indicator on the other, are given in columns 5, 7 and 9 respectively. The Baldwin Southwark values have been taken as a standard of reference since this instrument is

TABLE 11. RESULTS UNDER STEADY THRUST CONDITIONS

Standard pressure gauge	Stress in tie bars relative to B.I.	Baldwin indicator	Michell thrustmeter		M.I.T. oscillograph		Mechanical extensometer	
lb. per sq. in.	lb. per sq. in.	Tons thrust	Tons thrust	Per cent difference from B.I.	Tons thrust	Per cent difference from B.I.	Tons thrust	Per cent difference from B.I.
From readings taken before pulsating test :—								
0	0	0	0	—	0	—	0	0
200	1,210	2·22	2·32	+4·5	2·09	−6·0	2·06	−5·2
400	2,390	4·40	4·64	+5·5	4·34	−1·3	4·25	−3·4
800	4,930	9·00	9·28	+2·4	8·96	−1·0	8·97	−1·0
1,200	7,420	13·67	13·92	+1·8	13·66	0	13·30	−2·7
1,600	10,000	18·37	18·57	+1·1	18·35	0	18·03	−1·6
1,200	7,660	14·08	13·92	−1·0	13·93	−1·0	13·76	−1·6
800	5,100	9·39	9·28	−1·2	9·33	−0·7	9·27	−1·3
400	2,595	4·76	4·64	−2·5	4·52	−5·0	4·55	−4·4
200	1,275	2·34	2·32	−0·9	2·34	0	2·20	−5·0
0	0	0·07	0	—	0·05	—	0	—
From readings taken after pulsating test :—								
0	0	0	0	—	0	—	0	—
200	1,240	2·24	2·32	+3·5	2·21	−1·2	2·10	−5·4
400	2,420	4·44	4·64	+4·5	4·42	−0·5	4·32	−1·8
800	4,960	9·12	9·28	+2·0	9·03	−1·0	9·02	−1·1
1,200	7,400	13·60	13·92	+2·5	13·15	−3·2	13·55	−0·4
1,600	9,900	18·22	18·57	+1·8	18·11	−0·5	18·16	−1·1
1,200	7,600	13·99	13·92	−0·5	13·82	−1·0	13·95	−0·3
800	5,050	9·29	9·28	0	9·22	−0·7	9·45	+1·9
400	2,530	4·64	4·64	0	4·53	−2·5	4·66	0
200	1,280	2·37	2·32	−2·0	2·26	−4·5	2·32	−4·2
0	0	0·11	0	—	0·11	—	0	—

High readings relative to Baldwin Southwark indicator are regarded as positive differences.

stroke	pressure	oscillograph	Michell thrustmeter		Baldwin indicator		Pulsation
Inch	lb. per sq. in.	Tons thrust	Tons	Per cent difference from M.I.T. oscillograph values	Tons	Per cent difference from M.I.T. oscillograph values	±per cent
$\frac{5}{16}$	0	0	0	0	0	0	0
$\frac{5}{16}$	400	4·48	4·64	+3·5	4·40	−1·7	8·90
$\frac{5}{16}$	800	9·07	9·28	+2·0	9·04	−0·5	4·00
$\frac{5}{16}$	1,200	13·83	13·92	+0·5	13·60	−1·7	3·75
$\frac{5}{16}$	1,600	18·27	18·57	+1·5	18·30	0	1·80
$\frac{5}{16}$	0	0	0	0	0·09	—	0
$\frac{1}{2}$	0	0	0	0	0	0	0
$\frac{1}{2}$	400	4·44	4·64	+4·5	4·22	−5·0	13·5
$\frac{1}{2}$	800	9·21	9·28	+0·5	9·08	−1·5	8·0
$\frac{1}{2}$	1,200	13·92	13·92	0	13·62	−2·5	4·6
$\frac{1}{2}$	1,600	18·38	18·57	+1·0	18·40	0	2·4
$\frac{1}{2}$	0	0	0	0	0	0	0
$\frac{3}{4}$	0	0	0	0	0	0	0
$\frac{3}{4}$	400	4·51	4·64	+2·5	4·59	+1·5	20·0
$\frac{3}{4}$	800	9·26	9·28	0	9·08	−2·0	11·5
$\frac{3}{4}$	1,200	14·07	13·92	−1·0	13·73	−1·5	6·5
$\frac{3}{4}$	1,600	18·62	18·57	−0·5	18·50	−0·7	5·9
$\frac{3}{4}$	0	0	0	0	0·14	0	0
1	0	0	0	0	0	0	0
1	400	4·44	4·64	+4·5	4·61	+4·0	26·0
1	800	9·12	9·28	+1·5	9·23	+1·0	13·5
1	1,200	14·08	13·92	−1·0	13·77	−2·2	11·5
1	1,600	18·85	18·57	−1·5	18·31	−2·7	8·5
1	0	0	0	0	0	0	0

High readings relative to the M.I.T. oscillograph are regarded as positive differences.

considered to be more accurate than the M.I.T. oscillograph for the measurement of steady thrust.

It will be seen that in general, the tendency is for the M.I.T. oscillograph and the mechanical extensometers to give somewhat lower values than the Baldwin Southwark indicator, whilst the Michell thrustmeter gives somewhat higher values when the hydraulic loading is being progressively increased and somewhat lower values when it is being progressively reduced.

The Michell thrustmeters at present in use employ hydraulic pressures at full load varying from about 1,000lb. per sq. in. to 2,500lb. per sq. in., the tendency now being towards employment of the higher of these two values. It will be seen from Table 11 that as the hydraulic pressure is increased the percentage difference between the values of thrust given by the four thrust-measuring instruments becomes progressively smaller. Thus, with hydraulic pressures up to 400lb. per sq. in., differences up to about ± 5½ per cent were recorded between the Michell thrustmeter, M.I.T. oscillograph, and mechanical extensometers on the one hand and the Baldwin Southwark indicator on the other. Over the range 800-1,200lb. per sq. in. hydraulic pressure, these differences do not in general exceed ± 2½ per cent, whilst at 1,600lb. per sq. in. the differences are reduced to ± 1½ per cent.

It will also be noted that the differences between the values given by the Michell thrustmeter and Baldwin Southwark indicator, expressed as percentages, are of the same order of magnitude as the corresponding differences between the two forms of electric strain gauge equipment. A further point to be observed is that the values given by the mechanical extensometers are in reasonable agreement with those given by the other three instruments.

(b) *Pulsating-load Tests*. The results of the pulsating-load tests are shown in Table 12. In this case the M.I.T. oscillograph values have been chosen as a standard of reference since this instrument is considered to be more accurate than the Baldwin Southwark indicator for measurement of fluctuating strains.

It will be noted that with the pulsator set to maximum stroke, the alternating component of thrust, as measured in the hydraulic system of the meter block, amounted to about ± 10 per cent of the mean at the higher loadings. This condition is probably at least as severe as those commonly encountered in service.

The general character of the results is much the same as was observed under steady thrust conditions. Here again, the difference between the recorded values of thrust, expressed as percentages, are smallest at the higher loadings. Thus, with an hydraulic pressure of 400lb. per sq. in., differences of up to about ± 5 per cent were recorded between the Michell thrustmeter, Baldwin Southwark indicator, and M.I.T. oscillograph whereas at loadings in excess of this figure, the maximum differences recorded were only approximately ± 2½ per cent.

It will be observed that the values given by the Michell thrustmeter at the higher loadings are in close accord with those given by the M.I.T. oscillograph. Thus, with an hydraulic pressure of 800lb. per sq. in. the maximum difference between the values given by these two instruments does not exceed 2 per cent, whilst with hydraulic pressures of 1,200 and 1,600lb. per sq. in., the maximum recorded difference does not exceed ± 1½ per cent.

Comparison of the results obtained with different settings of the pulsator does not reveal any significant relationship between the magnitude of the alternating component of thrust and the differences between the values registered by the Michell thrustmeter and the M.I.T. oscillograph. Comparison with the results obtained in the static load tests, however, shows that the observed differences are in general slightly less under pulsating conditions than under steady conditions of thrust.

Conclusions

It is important, when considering these results, to bear in mind the probable accuracy of the instruments employed for checking the Michell meter. Experience suggests that the probable accuracy of the M.I.T. oscillograph strain gauge equipment, which has been adopted as the reference standard for the pulsating load tests, is of the same order as the observed differences between the values of thrust given by it and by the Michell thrustmeter. So far as the static-load tests are concerned a somewhat higher degree of accuracy is to be expected from the Baldwin Southwark instrument and the observed differences between the values given by it and by the Michell thrustmeter, are probably somewhat greater than those attributable to error in the former instrument. It is significant that at the higher values of both steady and pulsating thrust, all the instruments employed agreed with one another to within $\pm$ 2 per cent. It would therefore seem reasonable to conclude that the values given by the Michell thrustmeter under such conditions are accurate to within these limits.

ACKNOWLEDGMENTS

The author's thanks are due to the Council and Director of Research of The British Shipbuilding Research Association for permission to publish this paper and also to his colleagues who have assisted in the work.

REFERENCES

1. DENNY, A. 1907. Trans.Inst.N.A., Vol. 49, p. 121. "Torsionmeters as Applied to the Measurement of Power in Turbines and Reciprocating Engines".
2. HOPKINSON, B. 1910. Trans.Inst.N.A., Vol. 52, p. 184. "Notes on the Measurement of Shaft Horse Power".
3. TOWNS, V. W. H. 1935-36. Trans.Inst.Mar.E., Vol. 47, p. 281. "Developments in the Measurement of Torque and Thrust in Ships".
4. COFFIN, H. L. 1944. Trans. American Society of Naval Engineers, Vol. 56, p. 49. "Accuracy of Shaft Horse Power Data".
5. STONEY, G. 1937. *Engineering*, Vol. 144, p. 695. "Scientific Activities of the late Hon. Sir Charles A. Parsons, O.M., F.R.S.".
6. "Continuous Direct-Reading Power Meters". 1939. *Marine Engineering*, Vol. 62, p. 23.
7. "Full Scale Ship Trials in Germany with Special Reference to Torsionmeters, Thrust Meters and Speed Indicators". U.S. Naval Technical Mission in Europe. B.I.O.S. Misc. Report No. 86, H.M.S.O.
8. SUYEHIRO, K. 1914. Trans.Inst.N.A., Vol. 56, p. 167. "A Torsionmeter with Visible Scale".
9. MULLER, W. J. 1935. Trans.S.N.A.M.E., Vol. 43. p. 241. "Stroboscopic Measurement of Torque and Torsional Vibrations in Ships Propeller Shafting".
10. KEMPF, G. 1932. Trans.S.N.A.M.E., Vol. 40, p. 45. "Measurements of the Propulsive and Structural Characteristics of Ships".
11. Report on Visit to H. Maihak, A.G., Hamburg. B.I.O.S. Final Report No. 149, H.M.S.O.
12. MOULLIN, E. B. 1924. Proc.I.Mech.E., Vol. 2, p. 974. Appendix to First Report of the Marine Oil Engines Trials Committee. "A Distant Indicating Torsionmeter".
13. "The Moullin Torsionmeter". 1924. *Engineering*, Vol. 117, p. 764.
14. LANGER, B. F. 1945. Jl. Applied Mechanics, Vol. 12, p. A-39. "Measurement of Torque Transmitted by Rotating Shafts".
15. LANGER, B. F., and WOMMACK, K. L. 1944. Proc. Soc. Exp.Stress Anal., Vol. 2, No. 2, p. 11. "The Magnetic-Coupled Torquemeter".
16. GODSEY, F. W., and LANGER, B. F. 1944. Trans. American Society of Electrical Engineers, Vol. 63, p. 686. "Aircraft-Engine Torque Instruments".
17. HALL, W. C. 1940. Trans.S.N.A.M.E., Vol. 48, p. 320. "A New Type of Power-Torque Meter".
18. GUNN, R. 1940. Jl. Applied Mechanics, Vol. 7, p. A-49 "A Convenient Electrical Micrometer and its Use in Mechanical Measurements".
19. DRYSDALE, C. V. 1938. *Engineering*, Vol. 146, p. 69. "The Measurement of Mechanical Power".
20. "New Meter for Measuring Shaft Horsepower". 1939. *Marine Engineering*, Vol. 62, p. 208.
21. "Marine Shaft Horse Power Meter". February 1950. *Marine Engineering and Shipping Review*, Vol. 55, No. 2, p. 76.
22. CARTER, B. C., SHANNON, J. F., and FORSHAW, J. R. 1945. Proc.I.Mech.E., Vol. 152, p. 215. "Measurement of Displacement and Strain by Capacity Methods".
23. AUGHTIE, F. 1945. Proc.I.Mech.E., Vol. 152, p. 213. "Electrical Resistance Wire Strain Gauges".
24. RUGE, A. C. 1943. Proc.Soc.Exp.Stress Anal., Vol. 1, No. 2, p. 68. "The Bonded Wire Gauge Torquemeter".
25. SAUNDERS, H. E. 1934. Trans.S.N.A.M.E., Vol. 42, p. 128. "Measurement of Propeller Thrust on Shipboard".
26. ADAMS, R. R. 1930. Trans.S.N.A.M.E., Vol. 38, p. 245. "Thrust Measurements on the s.s. *Clairton*".
27. SAUNDERS, H. E., and PITRE, A. S. 1933. Trans. S.N.A.M.E., Vol. 41, p. 243. "Full-Scale Trials on a Destroyer".
28. LANGER, B. F. 1943. Proc.Soc.Exp.Stress Anal., Vol. 1, No. 2, p. 82. "Design and Applications of a Magnetic Strain Gauge".
29. CURRY, H. H., and FINK, J. C. 1941. Trans.S.N.A.M.E., Vol. 49, p. 300. "Test of Transient Conditions in Manœuvring a Coast Guard Harbour Cutter".

Discussion

MR. T. W. BUNYAN, B.Sc. (Member) said that the paper was indicative of the thoroughness with which the British Shipbuilding Research Association had tackled the problem of surveying the very wide range of instruments employed in this most important matter of torque measurement. He understood that in several instances in the past it had been the occasion of heated and sometimes very expensive argument between the shipbuilder, the engine builder, the propeller manufacturer, the purveyor of oil fuel, and so on.

Mr. Cook had dealt with the Siemens-Ford torsionmeter and the Michell thrustmeter in detail—and rightly so, because these instruments were the most popular in use in this country.

Table I contained some extremely valuable data. Divergencies in values of the torsion modulus were understandable, because the table covered a large number of shafts varying from 4½ inch to 21⅝ inch diameter, solid and hollow. This implied variations in forging reduction, heat treatment, and distribution or absence of the major segregate zones. He wondered whether the table could be extended in such a way as to give values that were more comparable; solid shafting, for example, being divided into categories of 16 inch diameter and over, 12 inch to 16 inch, 8 inch to 12 inch, and under 8 inch. This information, which might not indicate appreciable divergence in the stiffness modulus, would be very valuable to engineers who were compelled to spend so much of their lives in dealing with shafting problems of one sort or another.

The figure of $\pm$ 2 per cent accuracy with a torsionmeter which was mentioned in the conclusions on page 124 should, he thought, be qualified by the statement that it would only apply where a static calibration had been carried out with the shafting.

Results during the measured mile trials had been obtained, as far as he could see, in fairly smooth and sheltered water. Such conditions would exist, of course, in a large number of cases where an accurate torque or thrust measurement was necessary, but he was somewhat dubious as to whether such a high degree of accuracy could be obtained from the instrument when operating in a swell or in rough and choppy water. He would like to hear the author's views on this; he probably had other information.

Lloyd's Register had been chasing heavy weather stresses for some years now, not with overwhelming success. They had come to the conclusion that to get records of alternating and varying torques one needed some sort of recording torsionmeter. A voyage had recently been made for the purpose of studying propeller performance and excitation; the method employed used a modified Siemens-Ford torsionmeter feeding into an attenuator and magnetic tape recorder. The results had not been 100 per cent but they had been extremely encouraging. The technique was very simple and worthy of further development. The record was played back into a monitoring C.R.O., and any specially interesting parts of the record were shot on to the screen of a long persistence tube. In this way it was possible to study the results of a continuous record lasting up to an hour or more. The method was cheap and most convenient.

In conclusion, he would like to speak of an unusual but severe use of a Siemens-Ford torsionmeter. There had been occasion about a year ago to arrange some back-to-back tests on two large gear units, loading up the units by putting a controlled torque in the shafting. A Siemens-Ford torsionmeter was used to indicate continuously, over the whole trial, that in fact the specified torque was continuously maintained. The shafting was subjected to severe vibration at high frequency. After the first preliminary run, the zero was amazingly consistent, which suggested itself as a very convenient way of ageing the castings used for the sleeves of torsionmeters which he believed had a direct bearing on the stability of the apparatus.

MR. R. J. B. KEIG, B.A. (Associate) said that it was interesting to note that the extensive tests described had indicated that something a little better than ± 2 per cent of full power torque or thrust appeared to be the best accuracy obtainable with modern instruments. It was perhaps unlikely that a greater accuracy could be achieved in the near future with instruments designed for operation over long periods on shipboard. On the other hand, there was a very great need for instruments of much greater accuracy for trials purposes, and it was important to remember that all torsionmeters which measured the twist in a length of shaft were, in effect, a form of spring balance. It might well be that hysteresis of the shaft material would limit the further development of these instruments. Table I in the paper served as a valuable reminder of the variations in elastic properties to be found in typical shafts and emphasized the necessity for calibration of the shaft. It was possible that greater accuracy of torque measurement would only be achieved by some method which enabled the torque reaction to be weighed. The introduction of epicyclic reduction gearing would assist in this connexion.

It was important, however, to bear in mind the difficulties which might be encountered when using instruments of high sensitivity, due to slight variations in torque caused by the motion of the ship even in relatively calm waters, and it was suggested that some form of integrating meter would be required. At the Admiralty Engineering Laboratory some consideration had been given to the development of a counter device for this purpose. An instrument of this type should make it possible to evaluate the mean torque (or thrust) over a period in much the same way that mean shaft speed was obtained with a revolution counter and a stop-watch. It would be interesting to hear Mr. Cook's views on this suggestion.

It might be of interest to mention that at the A.E.L. some use had been made of the Maihak instrument and the magnetic-coupled torquemeters described in the paper. It was confirmed that the Maihak was capable of great sensitivity, but in practice some difficulty had been experienced with it and if torsional vibrations of any magnitude were present, it was extremely difficult to obtain satisfactory readings from it. The magnetic-coupled torquemeter had been used in small sizes with considerable success, particularly in applications where working conditions would have made an instrument fitted with slip-rings somewhat impracticable. It was unfortunate that the present design of this instrument did not lend itself to fitting to an existing installation without the insertion of a special length of shaft.

In conclusion, it might be mentioned that an extensive series of trials had recently been carried out in one of H.M.

ships, during which arrangements were made for recording relatively rapid changes of torque and thrust while the ship was manœuvring. This had been effected by fitting special inductive units and additional slip-rings to the Siemens-Ford torsionmeters and by fitting inductive pressure pick ups to the Michell thrustmeters. The inductive units were connected into a bridge circuit energized with 1,100 c.p.s. a.c. The output of the bridges was fed to meters on a panel and recorded photographically. The bridge output was also fed to an amplifier and a pen recorder. Calibration was effected by reference to the mean torque and thrust readings obtained in the usual manner from the Siemens-Ford torsionmeters and the Michell thrustmeters. The accuracy of the recording was limited by the scale length of the meters and the paper width in the case of the pen recorder, but suitable range change switches had been incorporated to improve the sensitivity for low power operation. It was considered that this method of recording transient and fluctuating torque and thrust should be capable of an over-all accuracy of about ± 5 per cent of full scale deflection, except at high sensitivity when drift due to temperature changes at the unit could be troublesome. The equipment had operated satisfactorily under somewhat arduous conditions over quite long periods.

Dr. J. F. Allan said that the subject of the paper was of considerable interest to himself and to his colleagues, and the justification and reason for that was quite clearly stated in the first clause of the introduction. The correlation of tank test data with actual results obtained on the measured mile and in service was one of the most important questions before the shipbuilders and the engineers at the present time.

Those present were probably well aware of the very particular attention being given to the subject in general at this time, and he need not elaborate that aspect of the importance of power measurement on board ship.

Mr. Cook had given a detailed account of the work which the British Shipbuilding Research Association had done in order to ascertain the probable degree of accuracy of the torsionmeters and thrustmeters which were most generally available for use in this country to-day, and they had gone to very great trouble to obtain good data on that subject. It was disappointing to find that they had come to the conclusion that there was a liability to error of the order of $\pm$ 2 per cent, even when every precaution had been taken.

He had been associated for many years with a firm who took very keen interest in trial work and in the measurement of torque when the steam turbine came into use in the early years of the century. In the early twenties they were using a meter known as the Denny-Edgecombe meter, a direct-vision torsionmeter which gave very satisfactory results on many trials. It was a straightforward stump and sleeve type with mechanical multiplication and visual observation of the angular movement and was succeeded later by the microptic meter described in the paper. These meters had tests against Siemens-Ford meters and they were at least as accurate as the Siemens-Ford meter at that time.

He did not say this in any competitive sense, but he was forced to the conclusion that the modern meter was not spectacularly more accurate than the meter of thirty years ago. It might be more easily fitted and handled but the increase in accuracy was not outstanding. It seemed probable that the limit of accuracy had been reached in the development of this type of meter. He would go further and would suggest that there was a case for the development of two types of meter—first, the robust type which could be left on the ship and used for service work; and secondly, a laboratory type of instrument which would be useful for trial trip work.

He would strongly suggest that an effort should be made to bring the accuracy of $\pm$ 2 per cent down to something of the order of $\pm$ $\frac{1}{2}$ per cent for trial trip work. The question of accuracy and liability to error had an important bearing on the whole question of ship model comparison. Without going into detailed figures, but making reasonable assumptions of the accuracy of measurement of speed and torque and revolutions first of all in the tank and then at the ship end, and accepting the figure of $\pm$ 2 per cent (which was for a calibrated shaft so that it would be at least $\pm$ 3 per cent for a meter fitted to a non-calibrated shaft) one arrived at a figure of about $\pm$ 5 per cent for the liability of error between the prediction from the model test and the result measured on the measured mile—and that was assuming ideal conditions there. It was worth while reflecting on that figure, which inferred that if every error went the wrong way, the model prediction could disagree with the ship result on the measured mile to something of the order of 10 per cent and still be within the limits of error of all the factors involved.

Thrust measurement was extremely valuable, and the work done by the British Shipbuilding Research Association in developing the use of the Michell type of meter was leading to the accumulation of very valuable data. The same remarks on accuracy applied more or less to the accurate measurement of thrust. Knowledge of thrust was more useful from the analytical point of view than knowledge of the power, because it by-passed scale effects which existed in the propeller performance and in the wake and brought one to a straight measurement of the force which was pushing the ship along. By its use one could learn a lot about the resistance of ships and the effect on resistance of various surface finishes.

Mr. J. M. Ford said that the author had made a very useful survey of the subject of torsionmeters and thrustmeters which included not only descriptions of various types of instrument but also most interesting tests. The latter were really interesting, and he was sorry he had not had time to study them more closely. In addition, the paper contained a lot of what might be called incidental information. Table I, for instance, was quite revealing and emphasized the need for caution in interpreting results obtained from non-calibrated shafts. It might be thought that the examination of a test piece cut from the shaft forging would give a reliable figure for the rigidity modulus, but he understood this was not so, since at times the modulus could actually vary from one end of the shaft to the other. He would be glad if the author could throw more light on this question, because anything which would help to avoid the expensive and time-consuming full-scale calibration and give a lead towards greater accuracy would be well worth while.

He did not propose to deal with the tests at all, but would like to discuss one or two minor points in connexion with torsionmeter design.

He realised that there was a limit to what could be dealt with in a paper of this kind, but it was a pity that the author, when dealing with torsionmeters, should have confined his descriptions almost exclusively to the actual measuring devices. This was only half the story. Some of these measuring devices were extremely interesting. For instance, the Maihak device in particular enabled torques to be measured on an 18-inch diameter shaft with a length of something like 4 inches. Whether such a very short length would give adequate accuracy under ordinary conditions was another question.

The sensitive element was only half the story because the engineering and mechanical design of the torsionmeter body was an equally important factor. Unless the body had the requisite rigidity, permanence of form and handlability, especially for lifting on to a shaft in an awkward position, and unless it could normally be fixed immovably to the shaft, there was no point in providing a precision micrometer device. In the past a number of torsionmeter designs had failed simply because the emphasis had been placed on some cunning scientific measuring principle, while the engineering of the rest of the meter had been left to chance.

For example, there was the method whereby the two halves of a torsionmeter, such as the two sleeves A in Fig. 4, were kept in their correct relative position. In this particular meter many problems of location were eliminated by the use of a simple elastic construction which avoided all rubbing supports and all wear and which required no adjustment of any kind. The design led to high rigidity of the meter in all directions, except where real flexibility was wanted—that was, torsionally; the two halves could not get out of square with the shaft, and the measuring length was positively determined. The general idea was pretty well known but in case anyone was interested, he had brought a model which could be examined afterwards. It consisted of two rings joined together by a series of steel strips of narrow rectangular section, the major axes of these strips being radial. The structure was extremely flexible torsionally, but laterally, regarded as a beam in any direction, it was very stiff.

A further example of the kind of thing that was important in the design of a torsionmeter was the attachment of the instrument to the shaft at the measuring planes. Unless the mechanical design of the clamping arrangement was correct, slip and zero wander could easily occur.

It was perhaps not always appreciated how much care was required to clamp a heavy body to a shaft so that it did not slip under any conditions. It would be seen from Table II on page — that in one of the tests the angular acceleration of the shaft at critical speed was 98 radians per sec. per sec. If one took the shaft which was the subject of this test, with its meter in place, and suppose one could by some means run this shaft up to a speed of 1,000 r.p.m. in a time interval of one second, one might say, in ordinary language, that "something might go". The accelerations involved in so speeding the shaft were almost exactly the same as the accelerations corresponding to the 98 radians per sec. per sec. Perhaps that would help to demonstrate what was involved in making a meter stick on the shaft without slipping.

In short, although the author had not dealt with these aspects of the torsionmeter, they were really extremely important and correct design was as important here as the correct design of the measuring device itself.

COMMANDER(E) J. I. T. GREEN, R.N. (Associate Member) said he had only one criticism and that related to Table IX. The assumed constant motor efficiency of 97 per cent did not appear to be supported by any observations. This contrasted oddly with the care taken to measure the torsional rigidity assumed in the same table. Was the figure a maker's guaranteed efficiency?

He also suggested that as accuracy improved it might become possible to observe the cyclic variation in torque due to propeller wake effects by direct torsionmeter reading.

MR. J. F. R. ELLISON (Member) said that few people realised that there were so many types of torsionmeter and thrustmeter, as only three of the former had been used to any great extent in this country. It was probably due to the fact that few of them produced the desirable features mentioned on the first page of the paper; namely robustness, simplicity and reliability. These qualities were essential if the instrument was to be fitted permanently in a marine installation.

It was only during the last five years that a serious effort had been made to measure the actual shaft horse power output of Diesel engines under seagoing conditions. The number of motor vessels having torsionmeters permanently fitted was steadily increasing.

In this respect, the company with which he was associated had made it a standard practice to have torsionmeters of the Siemens-Ford type fitted to all vessels, and they had twenty motor vessels so fitted.

One development of the "Ford" torsionmeter not mentioned in the paper was for use with alternating current. This had, of course, only a limited application, as comparatively few vessels had an a.c. supply available, but it did eliminate the need for a motor-driven interrupter which, with its cam-operated contacts, required adjustment and maintenance from time to time. He had just returned from the trials of a vessel fitted with a.c. torsionmeters, and their performance from the point of view of sensitivity had been most satisfactory.

With regard to the Michell thrustmeter, which appeared to be the most popular in this country, his experience of this instrument had shown that it was difficult to obtain an average reading except in very calm weather, and it would appear almost a necessity, to be able to take continuous readings over a period of time. This was, of course, impossible with the present arrangement of handpump and pressure gauge, but it might be achieved by means of a motor-driven pump and pressure recorder.

In conclusion, he considered that the research being carried out by the British Shipbuilding Research Association in connexion with the measurement of shaft horse power, both on the test bed and afloat, was most valuable, and he looked forward to their further findings with keen interest.

Correspondence

DR. T. W. F. BROWN (Member) wrote that the author was to be congraulated on giving actual figures for accuracy experimentally recorded in two commercially available instruments for measuring torsion and thrust. It was regretted, however, that the paper did not record any advance in the development of marine torsionmeters or thrustmeters as a result of research, and the accuracy in the instruments on which measurements were taken was so low that small improvements caused by research cannot be accurately recorded. Observing that further errors could be introduced into the use of torsionmeters by the difficulty of clamping the torsionmeters on the shaft and by shift of zero error on the torsionmeter after going full astern, the actual accuracies achieved in many cases were lower than those mentioned by the author.

The author did not fully make his point in specifying that a torsionmeter should go on an existing shaft. Providing robust, accurate and not too expensive torsionmeters could be developed, there seemed no objection to the torsionmeter having a special shaft as part of the instrument. Already the line shafting was divided at various couplings for ease of shipment or removal without introducing any difficulty.

A torsionmeter should not be regarded only as a tool on which figures of horsepower developed were measured purely during trial runs. In turbine engined vessels, if an accurate robust torsionmeter could be developed, it should form part of the equipment of the vessel as it was even more important during the life of a ship to prove that deterioration in the machinery had not occurred, and to enable the machinery to be maintained at full efficiency.

In trials at Pametrada utilizing the large brake, a standard Siemens-Ford instrument had been used through a range of shaft stress varying from 650lb. per sq. in. to 8,000lb. per sq. in., the torsionmeter having corresponding readings from 9-110 divisions. The torsionmeter did not require static calibration as the brake horsepower was used as the standard. The maximum error varied from slightly less than 2 per cent. at full power to about 12 per cent at the lowest power (2 per cent of full power) but the accuracy of the brake was naturally low at such a very low proportion of full power and the above error of 12 per cent might include a proportion of brake error.

It was noted that in the author's second series of trials, the shaft stresses were near the lower limit mentioned in the note above but torsionmeters of greater sensitivity were used and the scale reading for a given shaft stress was more than twice that of the instrument used at Pametrada. A maximum error of 3½ per cent was obtained, which was lower than the value obtained on a brake at the lowest power mentioned above.

In any large improvements the accuracy of marine torsionmeters and thrustmeters was of great importance to the marine industry and it was hoped that the examination of the errors measured in the commercial instruments mentioned by Mr. Cook would lead to further improvements in the future.

MR. A. M. RIDDELL (Member) wrote that the author mentioned in the introduction to his paper the importance of co-relation of tank data with actual results on trials and in service. Whilst torsion and thrust readings were essential for such co-relation, it was also essential to be able to measure speed. With information on this point and on torsion and thrust, all that was necessary for comparison to tank trials, was available. Methods of measuring speed were beyond the scope of the paper, but they did form an essential complement and therefore it would be valuable if the author could, very briefly, give particulars of the available methods of measuring speed and the limits of errors to be anticipated. Speed measurements were at least as important as thrust and torsion, since at normal speed/length ratios, thrust and torque varied as something more than the square of the speed at given displacements and trim. Conventional "over the mile" trials were often suspect, since the rate of increase of speed of tide could give an untrue picture. Where speed/length ratios were about unity or in excess of unity, a 3 per cent error in observed speed "over the mile" could result in a 15 to 25 per cent error in power credited for such incorrect speed.

Since permanent torsionmeters were often fitted, it would be appreciated if the author could give an idea of any likely alteration to modulus after, say, fifteen or twenty years in service. Designed torque stresses would be well within the elastic limit, but some form of fatigue might eventually be set up as a result of momentary building up of stresses due to reversing. Apart from torque stresses, bending moments had often to be accepted in tunnel shafting, due to weardown, motion of hull and machinery, or loading. Had any research been carried out on the difference between modulus of a new shaft and the same shaft after a number of years in service? With regard to loading, the writer was for some years in a cargo vessel which loaded manganese ore; from bitter experience he was able to predict which tunnel bearing would give trouble as a result of concentration of cargo at various points in the after holds.

Since most modern vessels were fitted with Michell thrust, a permanent thrustmeter could be incorporated at slight expense. A permanent torquemeter, however, would be more expensive. Nevertheless, if a temporary torquemeter were fitted for the trials it would be possible, with knowledge of torque and revolutions, to calculate within close limits the torque for any given revolution after the temporary torquemeter was removed. As part of the equipment, the builders could supply graphs based on information lifted from the temporary torquemeter which could indicate the revolutions and torque to be anticipated at various trims and displacements. These graphs would permit of the engineering staff on board being able to nominate power for any revolutions. This would give an excellent commercial check up on fuel per unit of power or fuel per unit of carrying capacity at a given speed.

The foregoing would be true provided no change of propellers occurred. Where there was a change of propellers a very close assessment of the difference in torque and thrust could be made by the use of the very complete and excellent standard propeller charts at present available.

Where tank trials were not carried out and where a torsionmeter was not fitted, it would be possible, with knowledge of the propeller, to make a very close prediction of thrust and torque on any given revolutions, provided a close assessment of wake could be made. With regard to wake, sufficient data existed to arrive at a reasonably close assessment in normal cases. Van Lammeran and Bragg both covered this point very well but there were, of course, other presentations. Since Michell thrusts were in such common use, if thrustmeters were fitted it would be possible to check up on an assessment of wake by checking back through the thrust recorded. The Wageningen Tank authorities had recently published a series of charts showing the results of model screws. The Ks-Km chart was particularly suitable for the above operation.

With the present numbers of standard propeller charts, he wondered why certain shipping companies still religiously logged apparent slip. With information lifted on trials using torsion and thrustmeters, wake could be established and thereafter true slip could be quoted rather than the very misleading apparent slip. One of the factors which controlled apparent slip was the observed day's run which might be, in part, with or against a tide or current. Also surprising was the fact that if torsionmeters were fitted for the trials, indicator cards were still called for. The errors to be anticipated in predictions of power from known propeller revolutions, were very much less than were to be anticipated from cards. Perhaps the author could be persuaded to express his opinion as to the relative merits of power indications from cards or propeller charts.

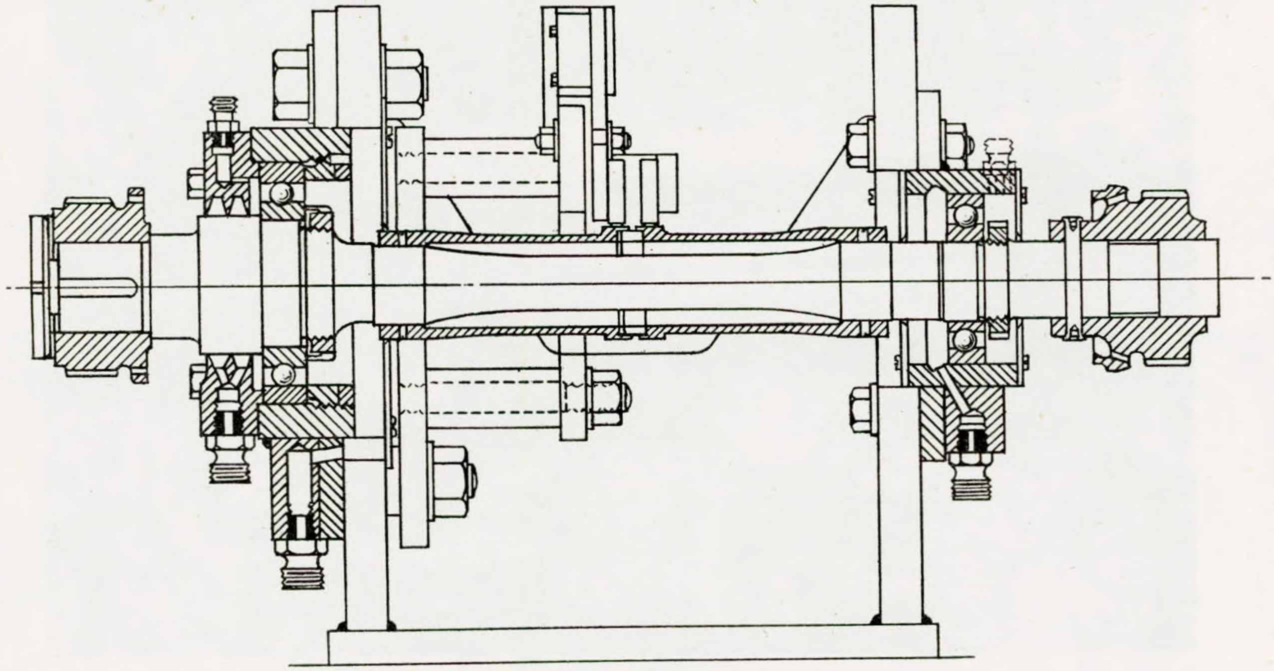


FIG. 18

Mr. G. Standen, A.M.I.Mech.E., wrote that the firm with which he was associated had had considerable experience with torsionmeters, particularly for high speed shafts (5,000-12,000 r.p.m.). Early tests were made using wire resistance strain gauges with stainless steel slip rings and silver morganite brushes. Experience showed that the collector gear was a constant source of trouble and required frequent attention, particularly in cases where the torsionmeter was in close proximity with a high speed bearing.

As a result, further work was limited to designs which did not require any electrical connexion with the rotating member and the method of measuring the electrical phase change between two electromagnetic generators as shown in Fig. 9 of Mr. Cook's paper was adopted. The electronic equipment for this type of measurement was developed during the war and one company now marketed a finished electronic equipment. The mechanical and electrical design of the phonic wheels and pick-ups was not standardized and it was left to the individual user to develop his own design.

For test bed work a design as shown in Fig. 18 had been produced suitable for speed from 500 to 12,000 r.p.m. and for horse powers up to 2,000. The unit was complete with its own bedplate and could be adapted for various powers by changing the torque shaft. It had been our practice to stress the shaft up to 20,000lb. per sq. in., using a steel with 60 tons per sq. in. U.T.S. and 45 tons per sq. in. yield. The teeth were of rectangular form and $\frac{1}{10}$in. pitch. A torquemeter of this design made to give full scale deflexion at 525 h.p. at 5,000 r.p.m. was checked against a Heenan and Froude No. DPX3 water brake. Agreement between the torsionmeter and brake readings were within $\pm$ 1 per cent of full power load over a load range of 5 to 1.

This method of torque measurement had also been developed for use on large diameter slow speed shafts. The torsionmeter shown in Fig. 19 was developed to give a full scale reading for a skin stress of 7,000lb. per sq. in. at a speed of 450 r.p.m. Fig. 20 showed the latest development of this unit. At speeds below 200 r.p.m. it was necessary to use a more sensitive meter, shunted to give the same sensitivity as that normally fitted in order to prevent a once per revolution oscillation of the pointer. This oscillation was caused by small manufacturing errors in the phonic wheels and errors in joining the two halves together in cases where it was necessary to manufacture them in two halves because of large diameter flanges on the shaft. The first of these torsionmeters was tested against two German water brakes and an accuracy to within $\pm$ 1 per cent was obtained over a load and speed range of 4 to 1.

The smallest torquemeter of this type so far manufactured by his firm had an overall length of $2\frac{1}{2}$ inch over the wheels and was capable of measuring 50 h.p. at 13,000 r.p.m. One objection to this type of torquemeter mentioned in Mr. Cook's paper was that dynamic calibration was necessary. We had found that this was not so though it was necessary to set the zero dynamically. Our standard procedure had been to obtain first the torque-angle of twist relationship over the measured length by static calibration, and then to obtain the meter reading—phase angle calibration from a simple calibrator. The calibrator consisted of a single phonic wheel which could be rotated at various speeds. Two pick ups were used, one fixed, the other movable by means of a micrometer. The two calibrations could be combined to give a torquemeter reading calibration. Zero setting could be done by rotating the shaft under no load condition at the speeds required and adjusting the meter reading electronically. This backing-off current could be checked at any time by means of a single switch. Where dynamic calibration was thought advisable or it was impossible to run the shaft under zero load, the pick up assembly was carried on the shaft itself by means of a specially supported sleeve and the pick up

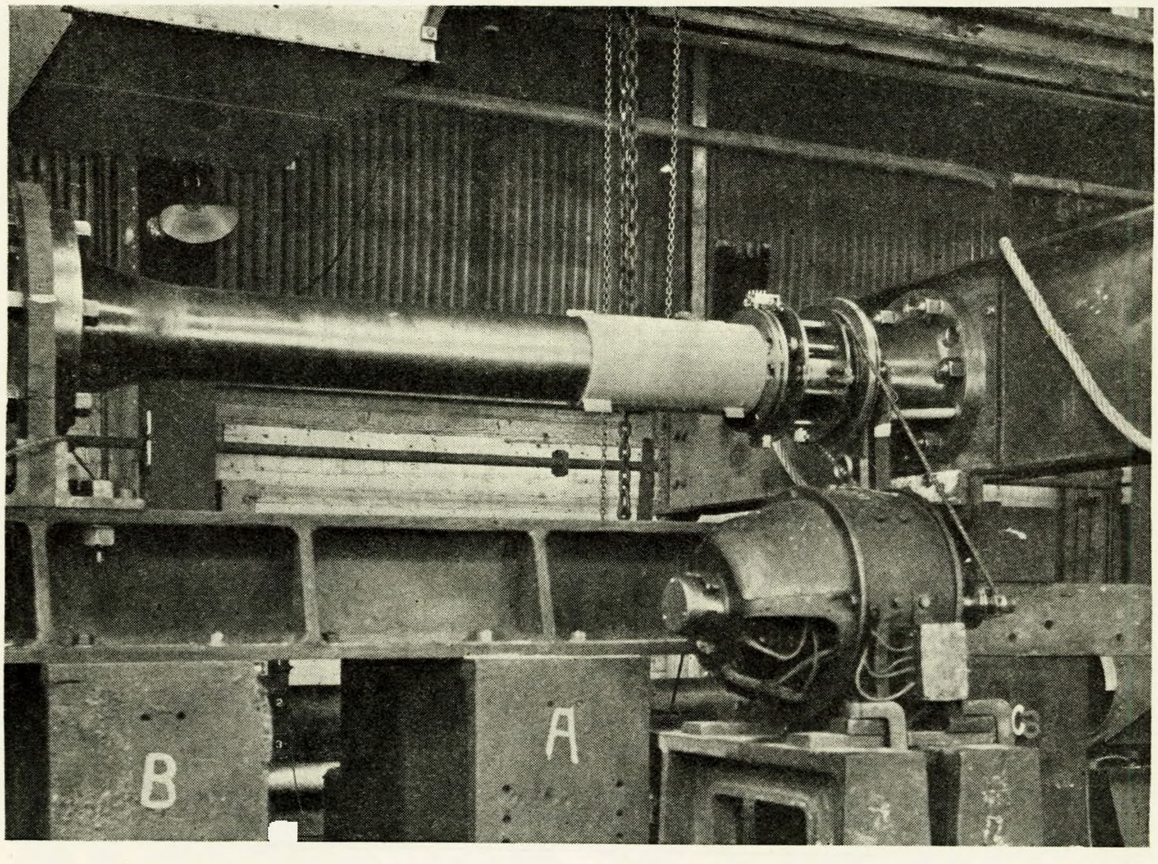


Fig. 19

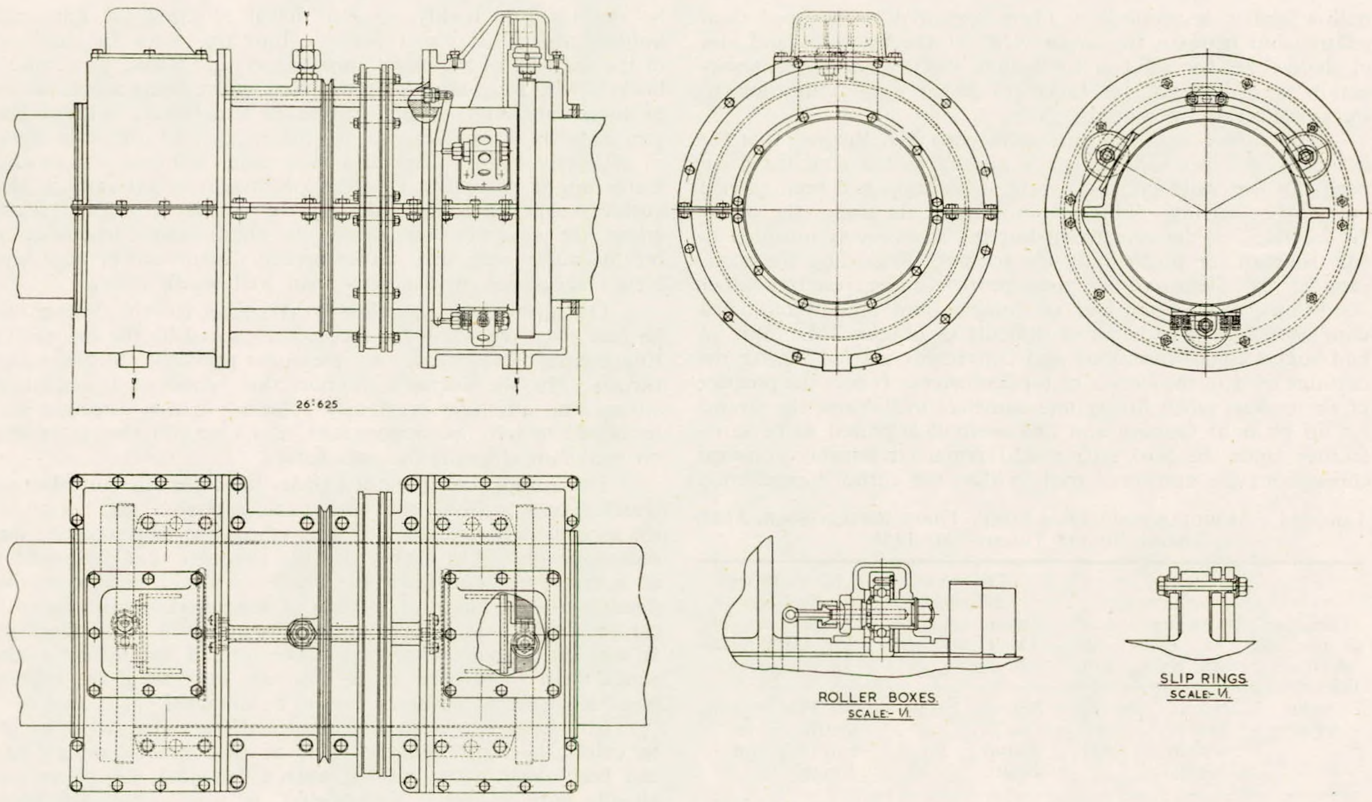


FIG. 20

output taken through slip rings. As, however, the voltage output from the pick ups was high (1 or 2 volts) and the rings were only in use in the initial setting, no trouble with slip rings had been encountered.

To date they had made no tests on large diameter shafts of 14 inch to 18 inch diameter with skin stresses below 5,000lb. per sq. in., but accuracies of the order of those quoted in the paper should be obtainable by careful mechanical design. It was estimated that the overall length of such a torquemeter would not exceed 22 inch.

Author's Reply

MR. COOK, in replying to the discussion, said that several contributors to the discussion had expressed interest in the analysis given in Table 1 illustrating the variation in modulus of rigidity of shafting. Since writing the paper the author had, through the courtesy of Siemens Brothers, been able to obtain further data permitting a more extensive analysis. For sixty-nine British mild steel shafts tested during the period 1945-1951, the mean value of the modulus was $11{\cdot}92 \times 10^6$lb. per sq. in. Of these sixty-nine shafts, some forty-three were solid, having a mean value of $11{\cdot}93 \times 10^6$lb. per sq. in. The remaining twenty-six were hollow shafts intended for naval use and the mean value of the modulus was $11{\cdot}90 \times 10^6$lb. per sq. in. Table 13 showed the probable error, based on these tests, if an assumed mean value were taken for the modulus of rigidity. It would be seen that the author's previous conclusions were fully borne out by this new data. Thus, if a mean value of $11{\cdot}93 \times 10^6$lb. per sq. in. were assumed, then there was only one chance in three that one would be correct to within $\pm \frac{1}{2}$ per cent and very slightly over one chance in two that one would be correct to within $\pm$ 1 per cent. These conclusions held irrespective of whether the shaft was solid or hollow.

Mr. Bunyan had suggested that further analyses on the basis of shaft size might be valuable and the author had accordingly done this for the sixty-nine shafts already referred to. Tables 14 and 15 showed the results of this analysis for the solid and

TABLE 13. MODULUS DATA FROM SIXTY-NINE BRITISH MILD STEEL SHAFTS TESTED 1945-1951

Variation of modulus of rigidity from mean value $11{\cdot}92\times10^6$ lb. per sq. in. per cent	Solid shafts		Hollow shafts		Solid and hollow shafts	
	No. of shafts within limits	Per cent of total	No. of shafts within limits	Per cent of total	No. of shafts within limits	Per cent of total
$\pm\frac{1}{2}$	14	32	9	35	23	33
±1	27	63	15	58	42	61
±2	40	93	20	77	60	87
±3	43	100	26	100	69	100

hollow shafts respectively. There appeared to be no logical relationship between the mean value of the modulus and size of shaft either for solid or for hollow shafts. Moreover, segregation according to size failed to reveal any diminution in the variation of the modulus.

The author is in full agreement with Mr. Bunyan that the figure of ± 2 per cent accuracy quoted in the conclusion on page 124 was valid only if a static calibration had been carried out on the shafting. This was implied by the use of the words "is capable" in the conclusion but the author was indebted to Mr. Bunyan for underlining the matter. Regarding the accuracy of the Siemens-Ford torsionmeter under rough weather conditions, the author had no information, and indeed such data would seem to be most difficult to obtain. Mr. Bunyan had suggested an ingenious and convenient way of ageing the castings used in the sleeves of torsionmeters. It was the practice of the makers when fitting torsionmeters to disperse the strains set up by light tapping and this method appeared to be satisfactory since the zero setting had remained sensibly constant throughout the numerous trials within the author's experience.

TABLE 14. MODULUS DATA FROM FORTY-THREE BRITISH SOLID MILD STEEL SHAFTS TESTED 1945-1951

Variation of modulus of rigidity from mean value per cent	8 in.-12 in. diameter (mean value of $G=11 \cdot 94\times10^6$ lb. per sq. in.)		12 in.-16 in. diameter (mean value of $G=11 \cdot 96\times10^6$ lb. per sq. in.)		Over 16 in. diameter (mean value of $G=11 \cdot 90\times10^6$ lb. per sq. in.)	
	No. of shafts within limits	Per cent of total	No. of shafts within limits	Per cent of total	No. of shafts within limits	Per cent of total
$\pm\frac{1}{2}$	2	100	3	21	12	44
±1	—	—	5	36	19	70
±2	—	—	12	86	25	93
±3	—	—	14	100	27	100

TABLE 15. MODULUS DATA FROM TWENTY-SIX BRITISH HOLLOW MILD STEEL SHAFTS TESTED 1945-1951

Variation of modulus of rigidity from mean value per cent	Under 8 in. outside diameter (mean value of $G=12 \cdot 07\times10^6$ lb. per sq. in.)		12 in.-16 in. outside diameter (mean value of $G=11 \cdot 88\times10^6$ lb. per sq. in.)		Over 16 in. outside diameter (mean value of $G=11 \cdot 85\times10^6$ lb. per sq. in.)	
	No. of shafts within limits	Per cent of total	No. of shafts within limits	Per cent of total	No. of shafts within limits	Per cent of total
$\pm\frac{1}{2}$	1	25	4	29	2	25
±1	2	50	8	57	8	100
±2	3	75	10	72	—	—
±3	4	100	13	93	—	—
±4	—	—	14	100	—	—

Mr. Keig's remarks were specially valuable in view of his practical experience of the difficulties involved in torsionmeter design and operation. The author was inclined to agree with Mr. Keig that an accuracy of slightly better than ± 2 per cent was unlikely to be improved upon in the near future with torsionmeters designed for operation over long periods on shipboard, but he felt that somewhat greater accuracy might be achievable for trials purposes, particularly in that rather exceptional class of trials where adequate facilities for calibration, etc., could be obtained. Mr. Keig suggested that hysteresis of the shaft material might limit the further development of this instrument, but the author felt that whilst this might be the case in highly stressed naval shafting, it appeared unlikely that this would prove a limiting factor in the case of the shafting of merchant ships where the stresses were much lower. Mr. Keig also suggested that when using instruments of high sensitivity, trouble might be experienced with slight variations in torque caused by the motion of the ship even in relatively calm waters and that some form of integrating meter might be required. This phenomenon did not, in the author's experience, arise during the course of normal trials where the objective was merely to obtain shaft horse-power but it might well arise under special circumstances and Mr. Keig's suggestion would thus seem well worth trying.

The author was indebted to Mr. Keig for the description he had given regarding the method employed by the Admiralty Engineering Laboratory to measure transient torques and thrusts. In the author's opinion this represented a distinct advance on anything previously achieved in this direction and he hoped to have an opportunity of trying out this technique on merchant ships in the near future.

The author was grateful to Mr. Ford for his contribution which, coming from one who had originated the design of the most widely used of all forms of marine torsionmeter, was most valuable. The author did not consider that examination of a test piece cut from the shafting would give a reliable figure for shaft modulus in view of the variations in mechanical properties which were known to occur in large forgings. It was the author's view that calibration of the meter on the actual shaft to be used in the ship was essential if the highest possible degree of accuracy was to be obtained.

Mr. Ford expressed his disappointment that what might be called the "ironmongery" side of torsionmeter design had not been dealt with, but the author felt that the paper was already long enough. Nevertheless, he was in full agreement with Mr. Ford regarding the very great importance of such things as method of attachment to the shaft, rigidity of sleeves, and so on. Indeed, the long experience of the makers with the Siemens-Ford torsionmeter was one of the factors which led B.S.R.A. to investigate the accuracy of this instrument. The author was indebted to Mr. Ford for his picturesque description of what was involved in an acceleration of 98 radians per sec. per sec. It showed that the design of the clamping gear of the Siemens-Ford torsionmeter was extraordinarily good, although on the other hand, there had been occasions on turbo electrical installations where it had not been good enough and the zero had shifted. It must be admitted, however, that these failures were under conditions which no torsionmeter could be expected to withstand.

Commander Green raised the question of the motor efficiency adopted for calculation purposes in Table 9. The figure of 97 per cent was assumed by the manufacturers of the propelling motor for calculation of shaft horse-power during the trials and in view of the 2-1 power variation the author saw fit to query this with the makers. It appeared that efficiency tests had not been carried out on this particular motor but subsequently, as a result of tests on a similar motor, the makers supplied the author with figures showing a variation from approximately 96-97 per cent. In view, however, of the fact that no test had been carried out on the actual motor employed, the author did not feel justified in introducing this refinement into Table 9.

The author shared Dr. Allan's disappointment that the accuracy of torsionmeters and thrustmeters most generally available for use in this country was not greater than ± 2 per cent, and he fully agreed that the matter should not be allowed to rest there. Dr. Allan's target of an accuracy of ± ½ per cent might be desirable but it remained to be seen whether such accuracy could be obtained under the difficult conditions existing on shipboard. The author could not agree, however, that there had been so little progress in these matters as Dr. Allan's contribution would suggest. The modern torsionmeter and thrustmeter was certainly more reliable and accurate than their counterparts of thirty years ago.

Dr. Allan contended that when allowance was made for errors in ship speed, torque and revolutions, there was the possibility of a 10 per cent discrepancy between model prediction and ship result on a measured mile. The author would merely point out that if torsionmeter errors were entirely eliminated there would still be, on Dr. Allan's reckoning, a possible difference of 5 per cent, and the author found it difficult to believe that errors of speed and revolution could be responsible for such a large discrepancy.

Mr. Ellison's statement that his company already had twenty motor vessels fitted with Siemens-Ford torsionmeters was a sufficient answer to those who contended that this class of instrument was not robust enough for use in service. In the author's opinion, service shaft horse-power figures obtained in this manner were much more accurate than those arrived at from indicator cards and an assumed mechanical efficiency, and he felt that the practice of fitting torsionmeters would continue to grow.

The author's experience with the Michell thrustmeter did not quite tally with that of Mr. Ellison's since no trouble had been experienced in obtaining steady readings in the course of many trials. Pressure recorders had been fitted to Michell thrustmeters with satisfactory results but there seemed to be little point in going to this elaboration unless a study was to be made of transient conditions.

The author was grateful to Dr. Brown for his contribution and for the figures he had given regarding trials with a Siemens-Ford torsionmeter at Pametrada. The author ventured to think that these results were not inconsistent with those given in the paper.

While noting Dr. Brown's views regarding the possibility of using a torsionmeter requiring a special shaft as part of the instrument, the author still contended that for the general run of trials work such a procedure was virtually impossible. In these days shipowners were usually anxious to get their vessels away to sea immediately after the trials and few would be willing to accept the delays imposed by the removal of the special length of line shafting; it was only in exceptional cases that the fitting of such a torsionmeter could be contemplated.

Mr. Standen's contribution formed a valuable addition to the paper and the details he gave of the various phase shaft torsionmeters developed by his firm were most interesting. Elimination of slip rings and collector gear was a very desirable objective, particularly for high speed shafts, and Mr. Standen and his colleagues were to be congratulated on the degree of accuracy they had achieved. As regards application to the slow speed, low stress shafts encountered in merchant ship practice, however, the author was a little doubtful whether this type of torsionmeter would show a marked advantage over commercially available instruments, particularly when the difficulties of obtaining calibration facilities were taken into account.

Mr. Riddell raised the question of the accuracy obtainable in the measurement of ships' speed over the measured mile. Such considerations were outside the scope of the paper and the author would content himself with merely observing that Mr. Riddell's assumption of a 3 per cent error in observed speed over the mile seemed unduly pessimistic. Few would admit the possibility of such a large error in any well-conducted trial.

The author regretted that he had no information regarding alteration of modulus of a shaft after a long period of service.

In regard to Mr. Riddell's suggestion for the temporary fitting of a torsionmeter and the subsequent use of the relation between propeller revolutions and torque so established at various trims and displacements, the author did not consider that this would lead to an accurate assessment of service power since so much would depend upon the condition of the hull and upon weather conditions.

The author shared Mr. Riddell's opinion regarding the superiority of torsionmeter assessment of power over that obtained with indicator cards; he did not feel competent, however, to comment upon the relative merits of indicator cards and propeller graphs for this purpose.