STUDIES IN MARINE

PROPULSION VIBRATION

by

D.H.NORRIE B.E.(HONS)., B.Sc.

APPENDIX V

ADDITIONAL MATERIAL SUBMITTED

AS EVIDENCE

This appendix comprises the following publications, submitted as additional evidence, in reprint, proof or manuscript form:-

(1)	NORRIE, D.H.	"The Towing of Small Ship Models" Bulletin of Mechanical Engineering Education, vol. 3, pp 1-7, 1964.
(2)	NORRIE, D.H.	"Marine Propeller Vibration Research at the University of Adelaide" Proceedings of the First Australasian Conference on Hydraulics and Fluid Mechanics. 1962, (Published by Pergamon Press).
(3)	NORRIE, D.H.	"The Research Water Tunnel Facility at the University of Adelaide" Journal of the Institution of Engineers, Australia, vol. 36, Nos 1 - 2, Jan-Feb.1964.
(4)	NORRIE, D.H.	"Strain Gauge Instrumentation for Simultaneous Measurement of Torque and Thrust" Australian Journal of Instrument Technology, vol. 20, No 1, Feb.1964.
(5)	NORRIE, D.H.	"Research in Propulsion Vibration at the University of Adelaide" Transactions of the Royal Institution of Naval Architects, vol. 106, No 1, Jan. 1964
(6)	HALE, M.R. and NORRIE, D.H.	"Hydrojet Propulsion Research at the University of Adelaide" Engineering (to be published July-August, 1964)
(7)	NORRIE, D.H. and SCHUMANN, R.	"The Development of a D.C. Strain Gauge System for Ship Tailshaft Measurements" Australian Journal of Instrument Technology (to be published November, 1964).

Norrie, D. H. (1964). The towing of small ship models. *Bulletin of Mechanical Engineering Education*, *3*, 1-7.

NOTE:

This publication is included in the print copy
of the thesis held in the University of Adelaide Library.

Norrie, D. H. (1962,). Marine propeller vibration research at the University of Adelaide. In R. Silvester (ed.), *Hydraulics and fluid mechanics: proceedings of the 1st Australasian Conference.* (p. 335-351). University of Western Australia.

NOTE:

This publication is included in the print copy
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Norrie, D. H. (1964). The research water tunnel facility at the University of Adelaide. *The Journal of the Institution of Engineers, Australia*, 35-38.

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Norrie, D. H. (1964). Strain gauge instrumentation for simultaneous measurement of torque and thrust. *The Australian Journal of Instrument Technology*, *20*(1), 25-32.

NOTE:

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of the thesis held in the University of Adelaide Library.

Norrie, D. H. (1964). Research in propulsion vibration at the University of Adelaide. *Transactions of the Royal Institution of Naval Architects*, *106*(1), 81-93.

NOTE:

This publication is included in the print copy of the thesis held in the University of Adelaide Library.

ENG—6—BEN—12.5.64

HYDRO-JET PROPULSION RESEARCH AT THE UNIVERSITY OF ADELAIDE

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Research has been ~~done~~ at the University of Adelaide on propeller vibrations since 1954, both in the model propeller research facility, and during full-scale ship tests. Beginning in 1962, an investigation started on the possible configurations of propulsion systems having inherently low vibration characteristics. carried out/

has been in progress/ &

If the energy transfer to the fluid in the propulsion unit is to be by rotodynamic means (by an impeller or propeller), then there are two basic requirements which must be satisfied if there are to be no fluctuating forces generated. In the first place, the fluid motion must be axially symmetric in the proximity of the impeller so that the velocity field relative to each blade is steady. The force on each blade will then be constant with time, and the torque, thrust, side load and bending moments on the impeller will thus be also constant. Secondly, the solid boundaries in the proximity of the impeller must be symmetrical around the impeller axis, so that the force due to the blade pressure fields on the boundaries is constant. will be/ If torque fluctuations in the drive shaft are acceptable, however, fluctuations in some of the other forces and moments may be kept small despite violation of the above conditions, provided the impeller is enclosed by a rigid duct of sufficient length.

& There are a number of ~~other~~ requirements which must also be satisfied by any propulsion system if it is to be suitable for large displacement vessels. & now/ Propulsive efficiency must at least be comparable with the efficiency achieved by the conventional screw propeller. Steering power must be adequate. Last, but not least, the ability to maintain satisfactory performance in varying conditions of seaway and the ability to withstand damage in excessive conditions, must be comparable with screw propellers.

The following configurations, which would have inherently low vibration characteristics, were considered:

1. Conventional screw propeller behind a hull with modified after-body.
2. Various configurations with the propeller mounted at or forward of the bow.
3. Various multi-hull configurations in which the propeller or propellers would operate in more uniform wakes than with the conventional screw system.
4. A submersible hull consisting of a cigar-shaped main body completely submerged with a narrow streamlined superstructure protruding above the waterline.
5. A hull with propulsive units in pods on outriggers.
6. Various forms of internal duct system (hydraulic jet propulsion or hydro-jet).

The preliminary study showed that although configurations 1 to 5 would be suitable for special-purpose vessels of various types, only configuration 6 showed possibilities as a replacement for the screw propeller on large displacement vessels. Attention has therefore been concentrated on this configuration.

The status of the hydro-jet project at present is as follows:

A general analysis of the hydro-jet system has indicated the requirements which must be satisfied to obtain high efficiency. To satisfy these requirements, a particular configuration must be adopted. A project study has been carried out for a 19,000 ton ore carrier which indicates that this configuration could be adapted satisfactorily to such a vessel. A model ducted-jet system is at present being constructed for testing in the 18 in research water tunnel ~~later~~ ~~in 1963~~, to determine the vibration characteristics of such a system and to provide experi- in/ mid 1964/

mental data on performance. If the results of the model studies are sufficiently encouraging, the construction of a 20 ft self-propelled model may be undertaken ~~incorporating the ducted system is contemplated for early 1964~~.

Vibration Characteristics of a Hydro-Jet System

One of the propulsive units for a dual "hydro-jet" system (one on each side of the ship) is shown in Fig. 1. Flow enters through the intake at station 1, passing along the duct, through the very high specific-speed impeller, through the stator vanes which remove the rotation imparted to the flow by the impeller, and out through the nozzle at station 4. The hub of the stator vanes contains the end-bearing for the tail-shaft. The thrust bearing is mounted inside the hull adjaçent to the point at which the tail-shaft passes through the plating. The flow in the upstream section of the duct near the hull splits to pass around the streamlined tail-shaft fairing before entering the impeller. A torsional isolater is mounted on the tail-shaft adjacent to the thrust bearing. If it is assumed that the duct is completely rigid and that the thrust bearing is fixed rigidly to the duct, then the net mean force which this closed duct system will impart to the hull will be equal to the momentum difference between the flow at stations 1 and 4. The flow at section 1 can be assumed to be steady although not necessarily uniform. Steadiness of the flow at station 4 will depend upon the distance between the impeller and the station. If this distance is sufficiently large, the trailing vortex sheets from the impeller and stator will have mixed to produce a fairly uniform turbulent flow. Although the momentum flux through station 4 will fluctuate about a mean value, fluctuations will be small in comparison with the mean momentum flux. The thrust force which the duct applies to a ship will therefore be a steady mean force on which is superimposed small fluctuations. There will also be small surface fluctuating forces in the radial directions. The level of these vibrations will be small, as compared with a conventional screw system. Fluctuating forces applied to the impeller may still be at an appreciable level, however. The use of a suitable torsional isolater in the tail-shaft might be used to reduce the torque fluctuations to a small level.

The above analysis assumes a rigid duct. In this case, the large surface fluctuating forces are to a large extent cancelled within the system. In practice, although the duct may be made very stiff with reinforcing rings, etc., the duct deflections will reduce the effectiveness of the internal cancelling.

Information on the vibration characteristics of a ducted system will be obtained in mid 1964 ~~later in 1963~~ from a pressure survey along the duct surface during the model studies. The torque and thrust fluctuations in the impeller shaft will be measured by the propeller dynamometer (described in Reference 1). As the duct is mounted on strain-gauged supports, the net force on the duct will also be measured.

Performance of a Hydro-Jet System

Consider the diagrammatic hydro-jet unit shown in Fig. 2. The hydrodynamic losses may be divided into two groups:

(*a*) Impeller and stator losses.

(*b*) The losses due to intake, duct, nozzle.

The losses associated with the impeller and stator are accounted for in the impeller efficiency, η_E. The intake plus duct plus nozzle losses, which will be termed the ductwork losses, may be expressed as a fraction of the intake kinetic energy, on the basis of a dimensional analysis, i.e.:

$$\text{Ductwork losses per unit mass flow} = \xi\left(\frac{V^2}{2}\right)$$

where ξ is termed the loss factor and V is the intake velocity.

The performance of the system can then be represented by curves such as shown in Fig. 3. This figure has been plotted using an impeller-stator efficiency of 0·90, as it has been estimated (see Reference 2) from high specific-speed axial-pump data that this efficiency should be obtainable for the large size impellers here

INSERT A from sheet attached.

considered. The adverse effect of increase in loss factor ξ on the ~~overall~~ η propulsive efficiency is clearly seen. On the diagram, for comparison, there has been plotted the characteristics of the standard B4-55 Screw Series for pitch diameter ratios of 1·0 to 1·4. The 0·80 × ideal (Froude) efficiency line for screw propellers has also been plotted on the diagram, as this is often taken in practice as representing the upper limit of efficiency for normal screw operation. It will be seen that the ducted system offers a potential increase in efficiency (particularly at the lower thrust load coefficients) as compared with the screw propeller, provided that the loss factor ξ is kept below about 0·10.

Actually, the performance of the hydro-jet system is more favourable than would appear at a first glance from the diagram. In order to compute the ~~overall~~ η propulsive efficiency η_p from the diagram, it is necessary to know the value of the thrust deduction factor t', ~~and~~ the wake fraction w' and the relative rotative efficiency, η_R. Values of thrust deduction and wake fraction will depend upon the ship/propulsion-system configuration. Values for screw propulsion are given in the literature. No information of this nature is available for the hydro-jet system. It can be deduced that the thrust deduction for the hydro-jet system will be smaller than for the screw configuration, and the wake fraction of the same order as ~~the~~ a single screw configuration. Insert B from attached sheet. This means that the hydro-jet efficiency will be more favourable than the shaded area would indicate on Fig. 3.

The reason why the thrust deduction is less for the hydro-jet unit is that the low-pressure region on the after-body of the ship caused by the flow being accelerated into the propeller disc does not exist in the hydro-jet system, outside the duct, to any appreciable extent. Also, any region of slightly reduced pressure forward of the intake is on a surface substantially parallel to the hull axis and hence has only a small component in the direction of motion. (Propulsive thrust is always larger than the resistance of the hull without the propulsion unit; this difference being expressed mathematically by the "thrust deduction factor." The additional resistance associated with the low pressure region is a major component of this difference, for screw propeller ships).

The reason why the wake fraction for the hydro-jet would be of the same order as ~~the~~ a single screw ~~screw~~ system is because the intake shape would be made such as to swallow as much low-velocity fluid in the boundary layer as possible.

The results of a more detailed analysis are shown in Fig. 4. In this analysis, the ductwork losses have been equated to the frictional loss in the equivalent straight duct. The equivalent straight duct has been defined as a circular duct of cross-sectional area equal to the hydro-jet intake area and whose length is such that the duct has a frictional loss equal to the ductwork loss of the hydro-jet unit being considered, at the same intake velocity. Only conditions of operation along the maximum efficiency line in Fig. 3 have been considered. The effect of variation of impeller efficiency, however, has been included in this diagram. Insert C from attached sheet. It will be seen from Fig. 4, for example, that in order to obtain a value of ~~$\eta_{p\,max}\left(\frac{1-t'}{1-w'}\right)$~~ $\frac{\eta_{p\,max}}{\eta_R}\left(\frac{1-w'}{1-t'}\right)$ equal to 0·75, with an impeller efficiency of 0·90, the length to diameter ratio of the equivalent duct must be about 5·0 for a duct of 15 ft diameter and an intake velocity of 26 ft per sec.

If the configuration of the hydro-jet system approximates to a straight parallel duct, and if the intake and nozzle losses are small, this means that the actual hydro-jet duct length will have to be about 5 impeller diameters if the 0·75 value is to be obtained for the diameter and intake velocity quoted. Achieving a value of 0·75 would mean (see Fig 3) that the hydro-jet efficiency would be comparable with that of a screw propeller, for the lower values of thrust load coefficient. The advantage of a plot such as Fig. 4 is therefore that it can be used to indicate very rapidly the approximate dimensions of a hydro-jet system for any specified performance. A study of Figs. 3 and 4 indicates that in order to obtain satisfactory efficiency from a hydro-jet system, the duct length must be comparatively short, less than approximately 5 to 6 times the impeller diameter.

Project Study

A project study has been carried out for a 19,000 ton ore carrier propelled by a hydro-jet system. The primary aim of this study was to determine whether a duct system could be incorporated in such a vessel without causing excessive difficulties in terms of shape, space, structural requirements, machinery layout, steering, etc. A number of configurations were considered, but that shown in Fig. 5 was the only one which appeared to satisfy all the basic requirements. Although it is difficult to make an accurate estimation of the thrust deduction and wake fraction which would be applied to such a configuration, conservative values of these factors were estimated and a value for the ~~overall~~ propulsive efficiency calculated. This indicated that an ~~overall~~ propulsive efficiency for the vessel of the order of 75 per cent might well be obtainable.

Consideration was also given to the problem of steering. Some form of swinging nozzle would appear to have advantages over the rudder immersed in the jet stream, both from the point of view of steering power and efficiency. The effect of fouling on the performance of the vessel was also considered. Although the fouling problem is by no means solved, the success of some of the treatments developed by the US Navy indicate that fouling in the duct may not present insuperable difficulties.

References

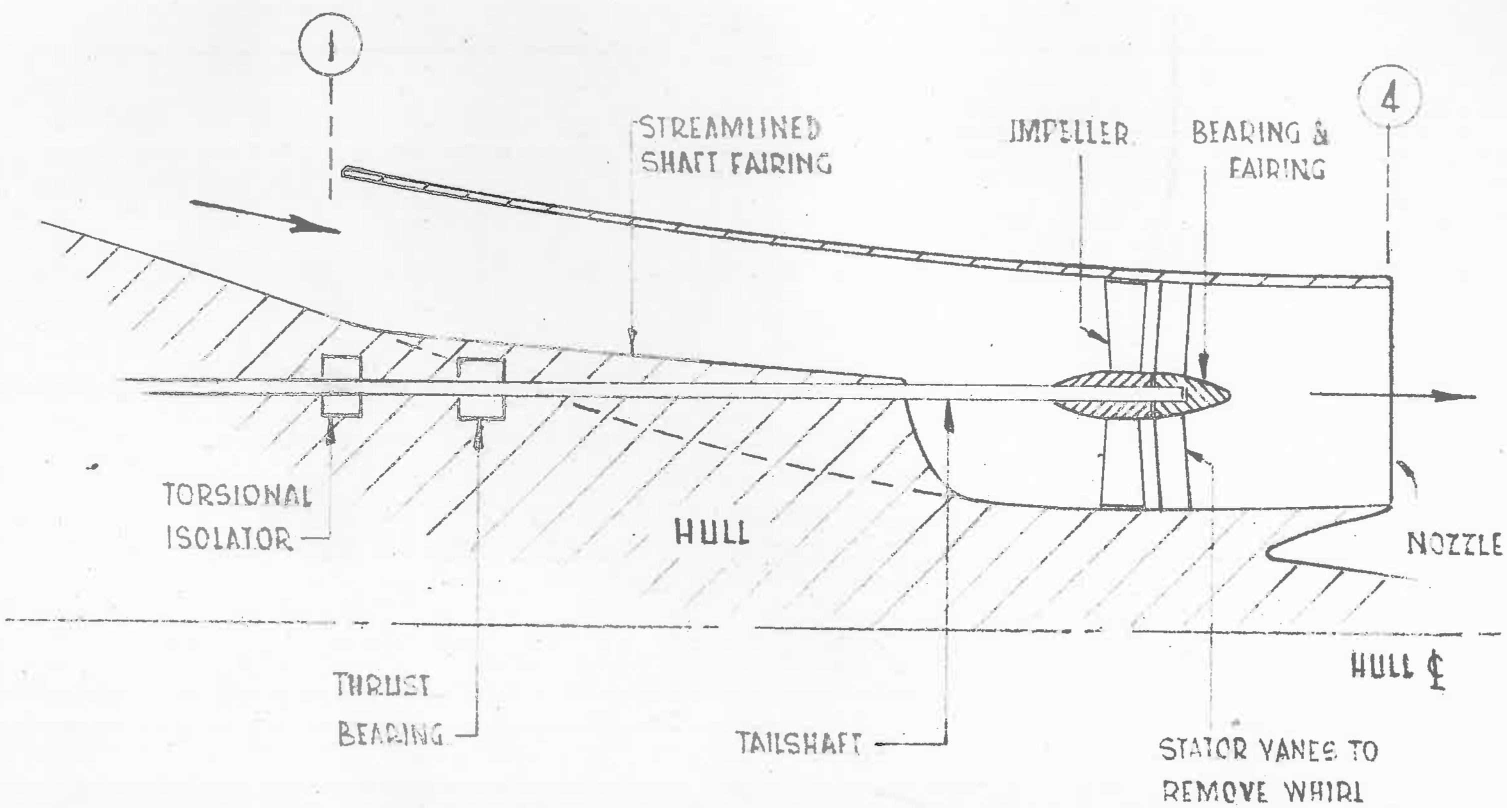
1. Norrie, D. H., "Marine Propeller Vibration Research at the University of Adelaide." Proceedings of the Conference on Hydraulics and Fluid Mechanics (Perth), 1963, Pergamon Press.

2. Van Manen, J. D., "Fundamentals of Ship Resistance and Propulsion," Part B, p. 120, Netherlands Ship Model Basin Publication 132a.

3. . . . .

insert D from attached sheet

Insert E from attached sheet



1
STREAMLINED
SHAFT FAIRING
IMPELLER
BEARING &
FAIRING
4
TORSIONAL
ISOLATOR
HULL
NOZZLE
HULL ℄
THRUST
BEARING
TAILSHAFT
STATOR VANES TO
REMOVE WHIRL

Fig. 1

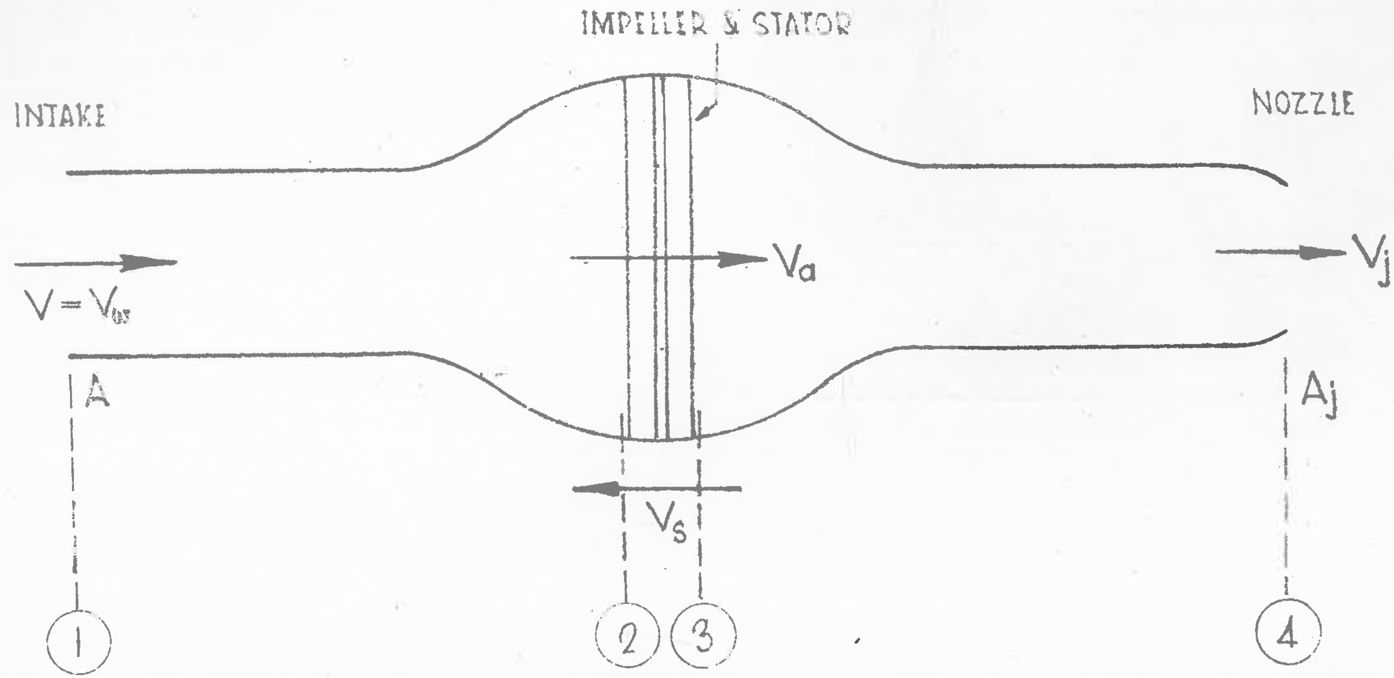


Fig. 2.

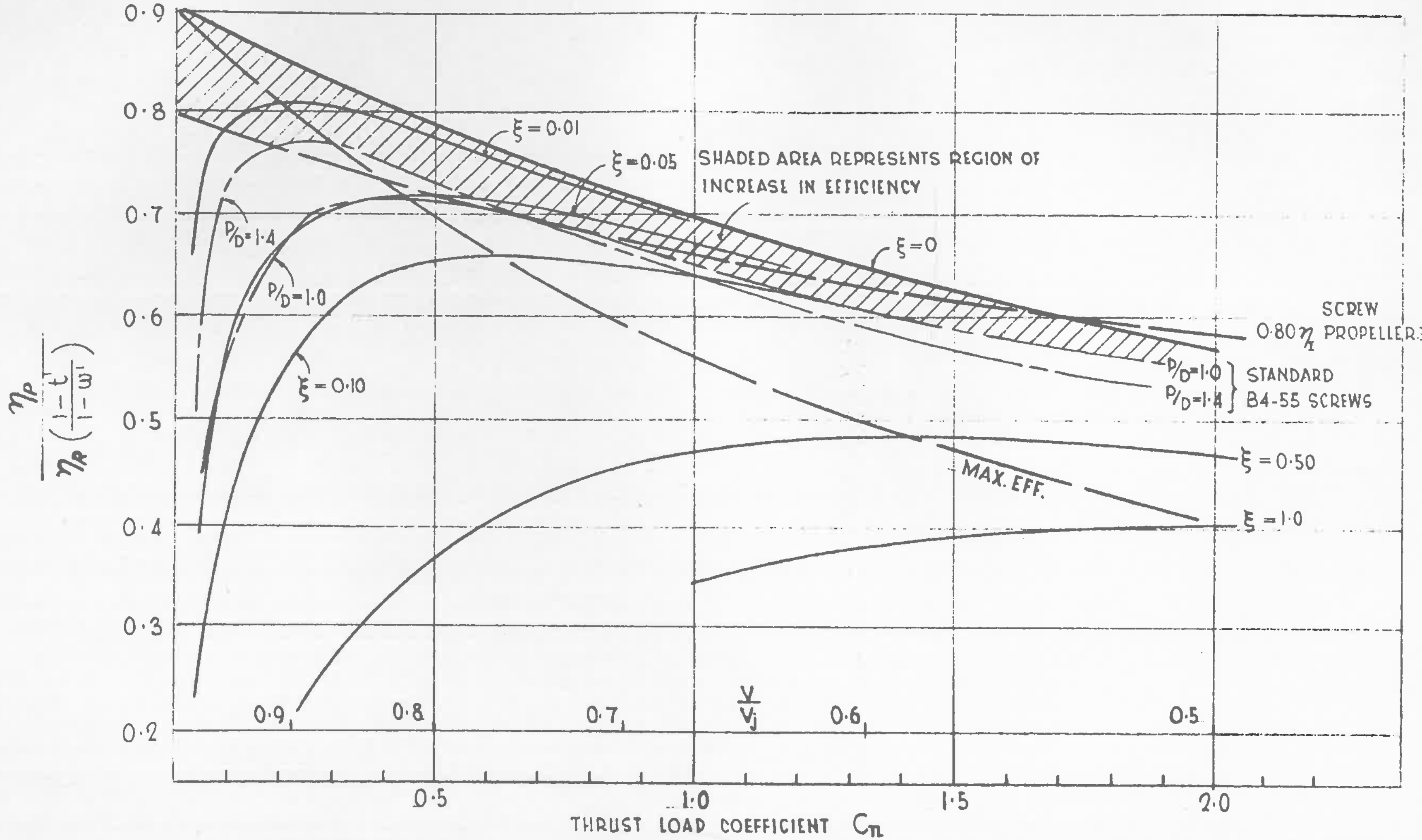


FIG. 3 ESTIMATED PERFORMANCE OF HYDRO-JET UNIT

(An impeller efficiency η_I of 0·90 has been assumed.)

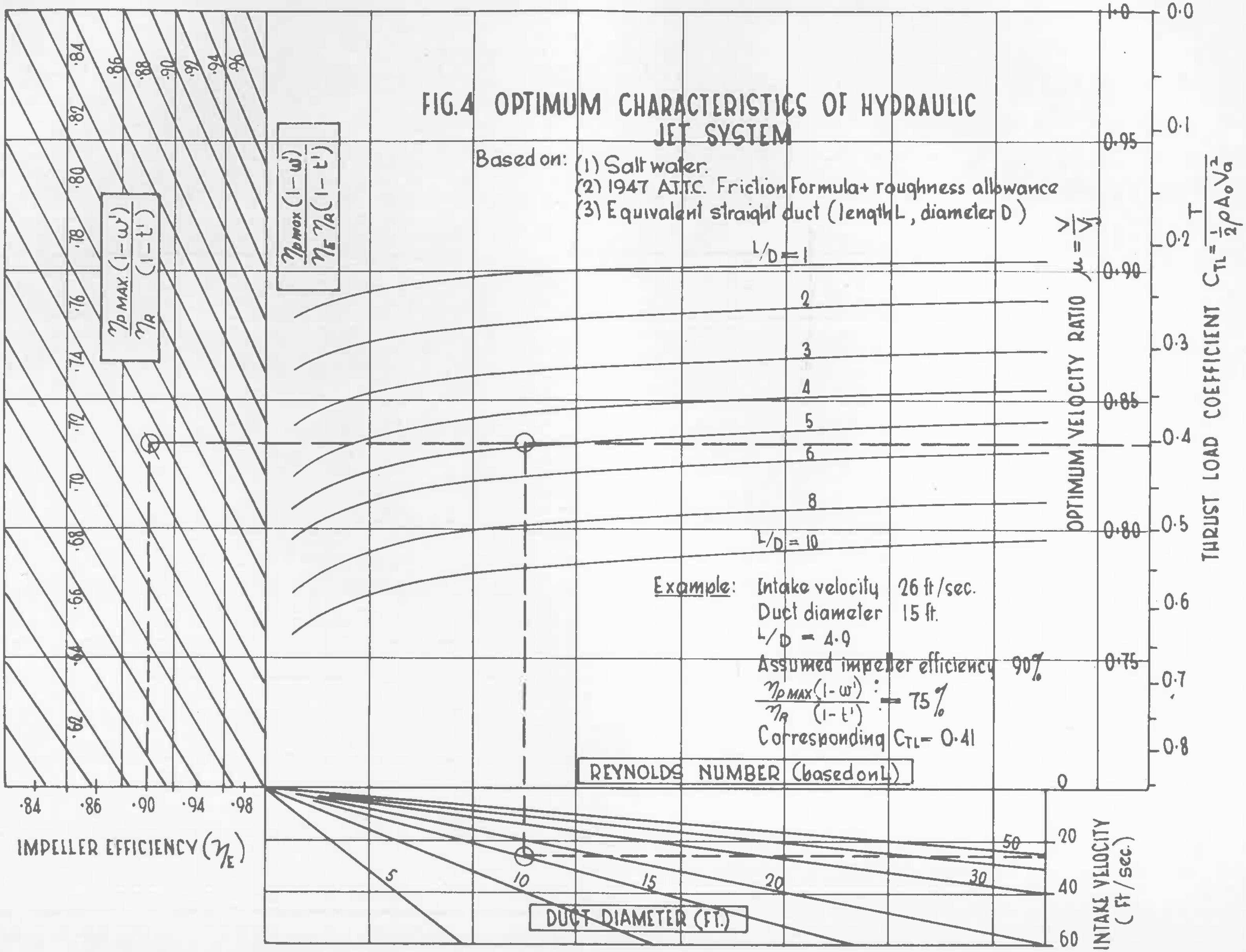
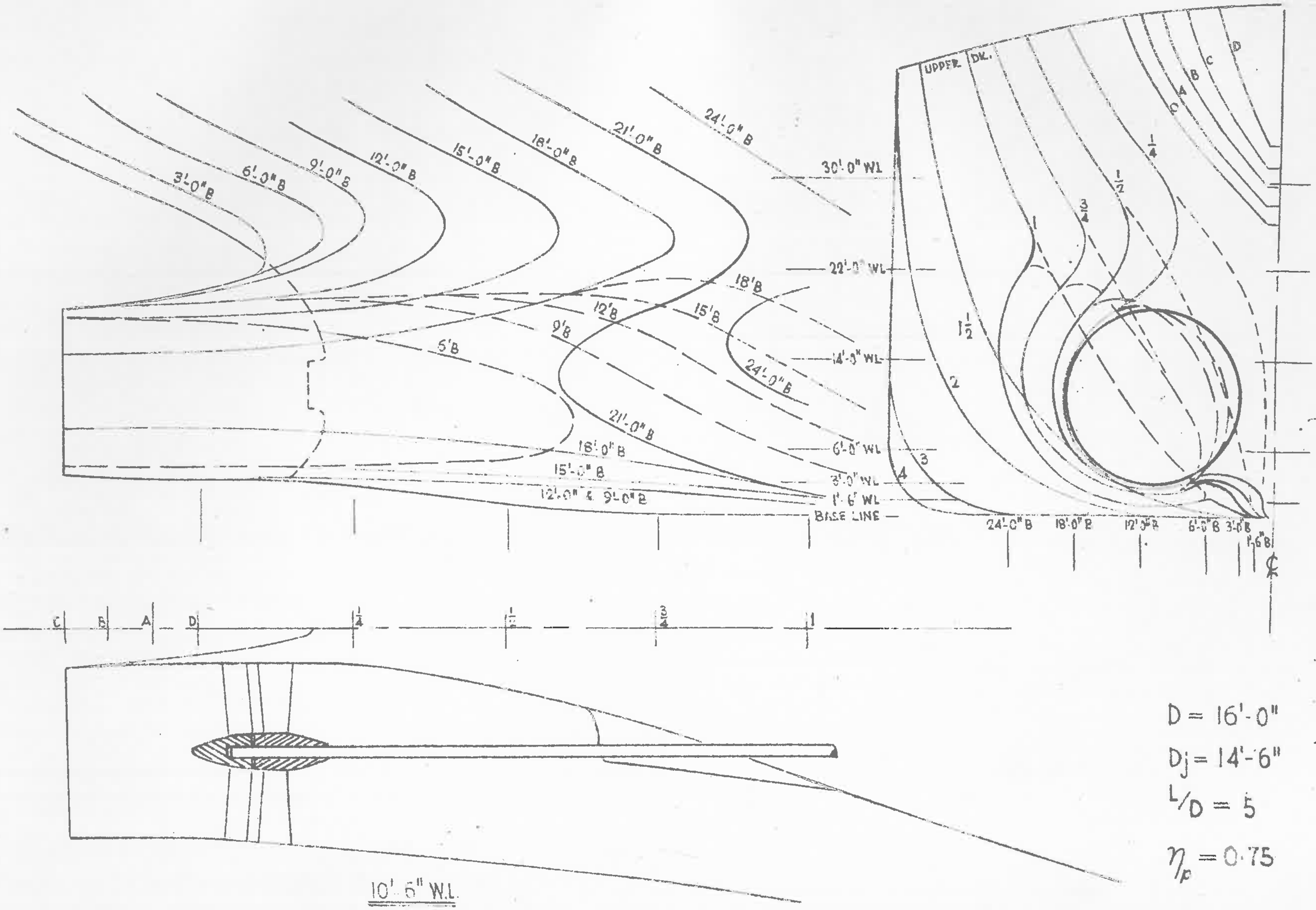


FIG.4 OPTIMUM CHARACTERISTICS OF HYDRAULIC JET SYSTEM
Based on: (1) Salt water.
(2) 1947 A.T.T.C. Friction Formula+ roughness allowance
(3) Equivalent straight duct (length L, diameter D)
$\frac{\eta_{p\,MAX}(1-w')}{\eta_R\,(1-t')}$
$\frac{\eta_{p\,max}(1-w')}{\eta_E\,\eta_R(1-t')}$
L/D = 1
2
3
4
5
6
8
L/D = 10
Example: Intake velocity 26 ft/sec.
Duct diameter 15 ft.
L/D = 4.9
Assumed impeller efficiency 90%
$\frac{\eta_{p\,MAX}(1-w')}{\eta_R\,(1-t')}$ = 75%
Corresponding C_{TL} = 0.41
REYNOLDS NUMBER (based on L)
DUCT DIAMETER (FT.)
IMPELLER EFFICIENCY (η_E)
OPTIMUM VELOCITY RATIO $\mu = \frac{V}{V_j}$
THRUST LOAD COEFFICIENT $C_{TL} = \frac{T}{\frac{1}{2}\rho A_0 V_a^2}$
INTAKE VELOCITY (ft/sec.)



$D = 16'\text{-}0''$

$D_j = 14'\text{-}6''$

$L/D = 5$

$\eta_p = 0{\cdot}75$

Fig. 5

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THE DEVELOPMENT OF A.D.C. STRAIN-GAUGE SYSTEM FOR SHIP TAIL-SHAFT MEASUREMENTS

by

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A.M.S.I.T.A.

1. INTRODUCTION.

The Department of Mechanical Engineering at the University of Adelaide has been carrying out full-scale investigations on propeller-shaft vibrations in ships since 1954. (Ref.1). All of these tests have been based on the electrical resistance strain-gauge measurement of surface stresses in the tail-shaft. The techniques and instrumentation have, however, progressively become more complex, due to the demand to obtain a wider range of data with a higher degree of accuracy. The corresponding development in the equipment has resulted in an increase in both accuracy and reliability. The description given in this paper is of the equipment used in the most recent series of tests, which were those on a 19,000 ton turbine-powered ore-carrier.

In the ore-carrier investigation, measurements of the instantaneous torque and thrust in the tail-shaft were made by using electrical resistance strain-gauges cemented to the tail-shaft surface. The output from these gauges was amplified by transistorised pre-amplifiers attached to the rotating shaft. The amplified output was then transmitted through slip-rings to the recording equipment.

2. THE STRAIN-GAUGES.

The strain-gauges which were used were a 600 ohm standard type

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of wrap around wire gauge, attached to the metal surface by epoxy cement. For the measurement of torque stresses, strain-gauges were cemented to the tail shaft surface at angles of 45° and 135° to the shaft axis, so that the principal tension and compression stresses, respectively, would be measured. A fully-active bridge circuit, with a symmetric arrangement of four gauges, was used, to ensure that signals due to axial and bending stresses would be cancelled within the bridge. Eight strain-gauges were used for the thrust measurement, in pairs, at angles of 0° and 90° to the shaft axis. A fully-active bridge circuit, with a symmetric arrangement of gauges, was also used, so that signals due to bending stresses would be internally cancelled.

The methods of measuring torque and thrust in a shaft, by strain-gauges, are well known. However, in the present application, there were a number of difficulties which required considerable development before they were overcome. Some of the most serious of these difficulties were associated with the strain-gauges. Errors in measurement due to this cause, may be grouped into two classes; errors due to gauge position, and errors due to gauge variation. Gauge misalignment errors were minimised by trimming the edges of each gauge in a guillotine-jig mounted on a microscope, with reference to the mean axis of the gauge-grid, as located by a scribed reference line on the perspex stage. Back-lighting was used through the transparent stage to make visible both the reference line and the wires of the gauge-grid. During cementing, the gauges were located on the shaft by means of a polytetrafluoroethylene jig. Errors due to gauge variation in resistance, gauge factor, and temperature coefficient were minimised by careful matching. Thermal insulation to reduce the temperature differential between gauges was also adopted. Further details of the above measures are given in Ref. 1 and 2.

It should be pointed out that it was then found that the most important residual error was associated with temperature differences between gauges. This error could have been further reduced if either temperature-compensating gauges or gauges with a lower temperature coefficient had been used. Due to difficulties of supply it was not possible to make use of these at the time. In work of this type where the stresses to be measured are very small and where high accuracy is required, it is strongly recommended that such special gauges be used.

3. THE ROTATING PRE-AMPLIFIERS.

In a D.C. strain-gauge system such as was used in the present application, errors due to temperature drift in the pre-amplifiers can be considerable. To obtain reasonable stability from a transitor pre-amplifier is in most cases not difficult, as there are now a variety of circuits available which will ensure low drift (Ref.3). It is, however, rather more difficult to obtain low drift with a simple reliable circuit suitable for a miniature pre-amplifier mounted on a rotating shaft. The pre-amplifier described below was developed in order to obtain a satisfactory compromise between stability and complexity.

The circuit used was based on the conventional differential type of transistor pre-amplifier, and is shown in Fig.1. The transitors used were carefully matched for both gain and leakage at room temperature. It was initially proposed to further match within these groups, for temperature-variation, but this would have required larger transistor stocks, and was in any event not found to be necessary. It appears that temperature-variation effects within the groups as selected, were of second order. Initially, low-leakage germanium transistors (Type OC44) were used. It was found, however, that an improvement in drift over the specified temperature range of the order of 2 times was obtained by replacing these in the first stage with audio-frequency silicon transistors (Type BCZ 10), and these were used in the final design.

To reduce temperature variations between transistors, they were mounted symmetrically within a circular aluminium block, whose function was to act as a heat sink of uniform temperature.

Initially, cracked-carbon high-stability resistors were used. It was later found that an improvement in drift of the order of 5 times could be obtained by replacing these in the first stage, with metallic-oxide-film high-stability resistors. Although the temperature coefficient of the metqllic-oxide resistors is not greatly different from that of the carbon resistors, their consistency in this regard is better, and appears to account for the improvement obtained.

The drift characteristic of the pre-amplifier is shown in Fig.2, which shows that the drift voltage after rising to a maximum at 88° then decreases with further increase in temperature. The magnitude of the drift voltage over the working range 60°- 100°F is, however small.

The gain of the amplifier was 2150. The input and output impedances were 9000 ohm and 800 ohn respectively.

Because the pre-amplifiers had to be mounted on a rotating shaft, it was important for size and weight to be kept small. The components were assembled on a section of matrix-board, and enclosed in an aluminium case of over-all dimensions 7" x 4" x 1¼".

4. THE SLIP-RING ASSEMBLY.

In the earlier investigations, both cylindrical - and disc-type slip rings were used. In the first type of slip-ring, each individual ring is a short length of cylinder mounted co-axially around the shaft, with the brushes mounted radially. In the second type, each "ring" consists of an annular disc mounted normal to the axis of the shaft with the outer and inner peripheries of the disc concentric with the shaft. The brushes are mounted parallel to the axis of the shaft and bear on the faces of the disc. The principal advantage of the disc-type slip-ring is that it is very easy to construct, the sheet-metal discs and insulating segments being fixed together with bolts, and attached to the shaft with a split clamping-ring. The disadvantages of the disc-type are, in the first place, the length of shaft required for a large number of rings (since sufficient space must be provided between the discs for the axially located brushes), and in the second place, the relatively high susceptibility to damage of the thin discs during transportation and assembly. Because of these disadvantages the cylindrical type of slip-ring was used in all the later investigations.

With both types of slipring assembly, care must be taken to ensure correct alignment on the shaft. Misalignment is not desirable, as it causes a varying pressure between the brush and slip-ring, resulting in variation of brush noise. In extreme cases of misalignment brush bounce may occur with the brush losing contact with the slip-ring surface intermittently.

In the most recent investigation, an assembly of 16 slip-rings (of the cylindrical type) were used, the rings being embedded in an epoxy-fibreglass annulus fitted to a duralumin clamping ring. The slip-rings were of square-section brass, and were curved to the appropriate radius by rolling. The rolled rings were cut in half and the resulting half slip-rings were cemented with epoxy cement into grooves turned in the epoxy-fibreglass annulus. The rubbing surfaces of the sliprings were then turned to size and polished. To enable the slip-ring assembly to be attached to the shaft it was constructed in two half sections (split on a diametral plane) and bolted together in position. The slip-rings were wired internally to a connecting strip mounted on the side of the duralumin clamping ring. Silver-carbon brushes were used to reduce brush noise. Two sets of brushes were used at positions approximately 120^{o} apart around the shaft. Each half-slip-ring was wired electrically to the complementary half-ring to also help reduce end-gap effects. An exception was the slip-ring used to measure angular position. In this case, the half-slip-ring was insulated from its complementary half-ring. This half-ring was supplied with an electrical voltage from an adjacent slip-ring. A single brush was then used to contact this half-slip-ring, so that a rectangular wave-form was obtained which indicated the position of the slip-ring end-gap. Since the relative angular positions of the end-gap and the propeller blades were knoen, the angular position of any blade at a given instant during rotation, could be determined.

Although brass was used for the slip-rings, because of its relatively low cost, previous experience in the Department of Mechanical Engineering had indicated that rather better results could be obtained by using monel-metal. One of the disadvantages of brass is that a thin oxide-film tends to form relatively rapidly and causes excessive brush noise, unless the slip-rings are regularly cleaned at intervals with 600-grit emery, followed by cleaning with a carbon tetrachloride impregnated cloth. Work carried out elsewhere indicates however, that the oxide trouble would have been prevented, had the brass slip-rings been plated with a precious metal.

5. THE DRIVER AMPLIFIERS AND THE RECORDER.

The driver amplifiers were of the push-pull transistor type, using matched germanium transistors. These amplifiers were developed commercially for this work, and were designed to operate from a standard 12 volt lead-acid accumulator. The specification called for input voltages up to ± 50 millivolts with corresponding maximum output voltages of ± 6 volts. (These values correspond to full-scale deflection of the recorder pens). The maximum amplifier gain was 120. Coarse and fine attenuation was available in 2 db. steps up to 40 db. The input impedance of the amplifiers was 10,000 ohms.

The output from the amplifiers was used to drive 30 ohm centre-tapped recorder pens. The recorder was of the ink-on-paper type, and was supplied commercially to specification. Eight recording pens were used, each with a maximum travel of ± 2 centimetres. Nine marker pens were also incorporated in the instrument. A range of paper speeds from 0.0125 centimetres per second to 50 centimetres per second was available. The frequency response of the pens was flat, up to 50 cycles per second.

6. THE IMPEDANCE CONVERTER.

Both the torque and thrust output from the slip-rings after amplification were recorded in two forms. The total output for either torque or thrust can be regarded as the sum of two components; a mean D.C. value (corresponding to the mean torque or thrust), and an A.C. component (corresponding to the fluctuating torque or thrust) superimposed upon the mean. The output from the slip-rings was, for both torque and thrust, recorded in three separate forms, the three recording systems operating in parallel;

(a) The total output was amplified by a driver-amplifier and recorded as an ink-on-paper trace.

(b) The mean value was recorded by a high-impedance voltmeter, which was heavily damped.

(c) The A.C. component was separated from the mean value by means of a specially designed impedance converted, then further amplified, and recorded as an ink-on-paper trace.

The design of the impedance converter posed the following problem. The A.C. component of the total output was comparatively small (of the order of 1 to 2%), of the mean D.C. component, under many operating conditions. To enable the A.C. component to be amplified and recorded, the large D.C. signal had to be completely blocked. The A.C. component however, had to be transmitted through the impedance converter with negligible phase change, and at a gain, preferably near 1, which was constant with both frequency and time. The use of a blocking condenser in each leg of the system was the obvious method of accomplishing the above requirement. However, the phase shift θ for such a circuit is given by $\tan\theta = \frac{1}{2\pi fRC}$ (1) where f = frequency, R = circuit resistance, C = ½ the capacitance of each condenser. The resistance R for the case being considered is the input impedance of the driver amplifiers to which the circuit is connected. The input impedance of these amplifiers was low, of the order of 10,000 ohms. The frequencies of the A.C. component were comparatively low. For example, the fundamental frequency in the torque or thrust wave-form at half of the maximum shaft speed was about 4 cycles per second. Equation (1) shows that for $\dot{\theta}$ to be kept small under these conditions would require extremely large values of C. In the present case, the values of C required were impractically large. Hence, between the capacitor and the driver input, a cathode follower circuit was inserted, as shown in Fig.3. The resistance R in equation (1) now became the input impedance of the cathode follower. This was a very high value, of the order of several megohms. Thus the value of C required was very much smaller than previously. It was found that θ could be kept below 1^{o} at 4 cycles per second with the use of quite small values of C. To cover a range of operation, it was arranged that R could be varied, as shown in Fig.3, by switching in an additional resistance into the circuit.

A disadvantage of the above circuit was that the time constants were comparatively large. Hence, during sudden changes of engine speed the variation of output voltage from the cathode follower circuit while the condensers were discharging, was found to be too great to be accommodated by the driver amplifier. A manually-operated bias voltage was therefore applied to the output of the cathode follower to enable the signals during such sudden changes to be kept within recording limits.

The input impedance of the impedance converter , when connected to the driver amplifier was 34.8 megohms maximum. The gain of the impedance converter was 0.85, constant over the range of frequencies encountered.

7. CALIBRATIONS.

Despite the measures taken in the design of the pre-amplifier and driver-amplifiers, it was to be expected that there would still be residual drift in these amplifiers due to temperature. By periodically switching off the power to the strain-gauges and measuring the amplifier outputs, it was possible to obtain a continuous record of amplifier drifts. These drift values were then used to correct the recorded data where necessary.

Although it was not expected that the over-all gain of the whole measuring system would vary appreciably with time, it was considered desirable to have a continuous record of the gain during the tests. To calibrate the overall gain of the system, a high-stability calibrating resistor could be switched into either strain-gauge bridge. The switching was carried out on the shaft by a relay operated by an external voltage transmitted through a pair of slip-rings.

In order to estimate drift due to local temperature variations of the strain-gauges, a log was kept of output voltages at the zero-stress condition, i.e. when the shaft was under no load, usually with the ship at the dock.

8. ASSOCIATED EQUIPMENT.

The power for the strain-gauges and the rotating pre-amplifiers was supplied from dry cells. In the case of the strain-gauge power, the voltage across the strain-gauge bridge was kept constant by rheostat control and manual monitoring. The power for the rotating pre-amplifiers was also transmitted through the slip-rings. It was found that temperature variations of the strain-gauges caused by the strain-gauges being supplied with power at intermittent intervals (i.e. only when measurements were required), was an appreciable source of error. To conserve the dry cells during a long voyage, alternating current of the equivalent voltage was passed through the strain-gauges during periods between readings, to maintain the same level of power dissipation, to prevent the undesirable heating and cooling effects. This alternating current was obtained by breaking down the ship's 32 volt A.C. with resistances.

It was essential that a time standard be recorded on the recorder paper, in case there was any variation of the paper speed. A clockwork-operated contactor was used for this purpose, to provide a rectangular wave-form of known period (a half-second per period).

A check on the contactor could be made by means of electronic counter.

Event marks were made on the recording paper by operating marker pens in various sequences, using a previously worked-out code.

During the investigation, much of the equipment was located in one of the noisiest sections of the ship's engine room. Other sections of the equipment were located at a considerable distance away. Communication between the three persons operating the equipment, was found to be impossible, without artificial aid. An intercommunication system was therefore built up, based on throat-microphones and ear-muffed headphones, incorporating a transistorised amplifier. This equipment was portable and worked well under conditions of very high ambient noise level.

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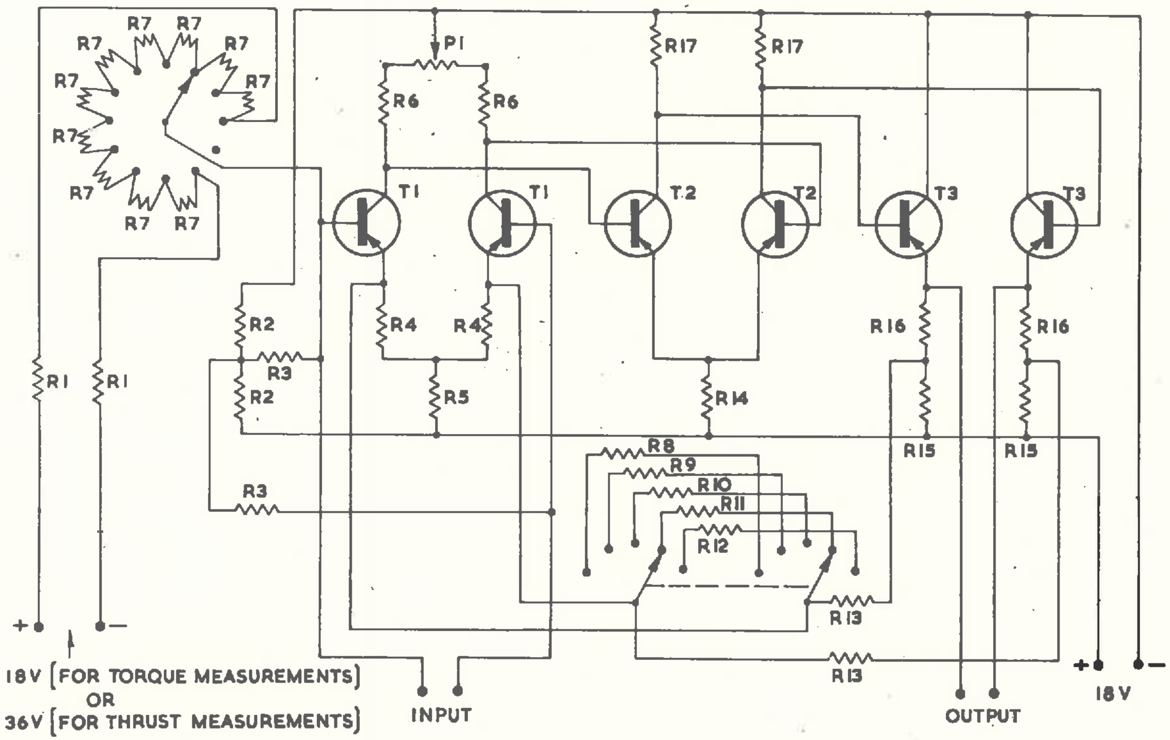
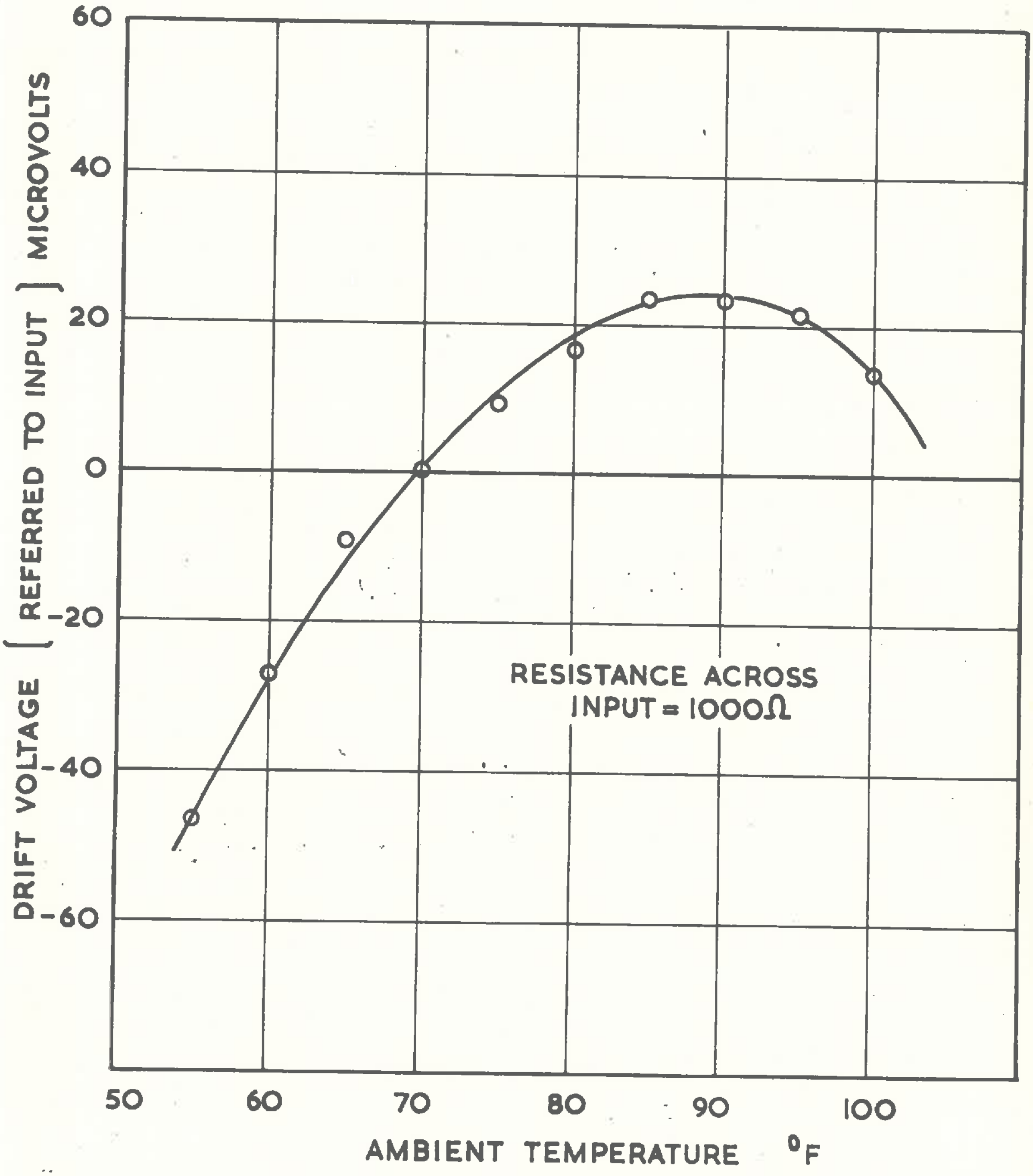
THE DEVELOPMENT OF A D.C. STRAIN-GAUGE SYSTEM FOR SHIP TAILSHAFT MEASUREMENTS

CAPTION LIST



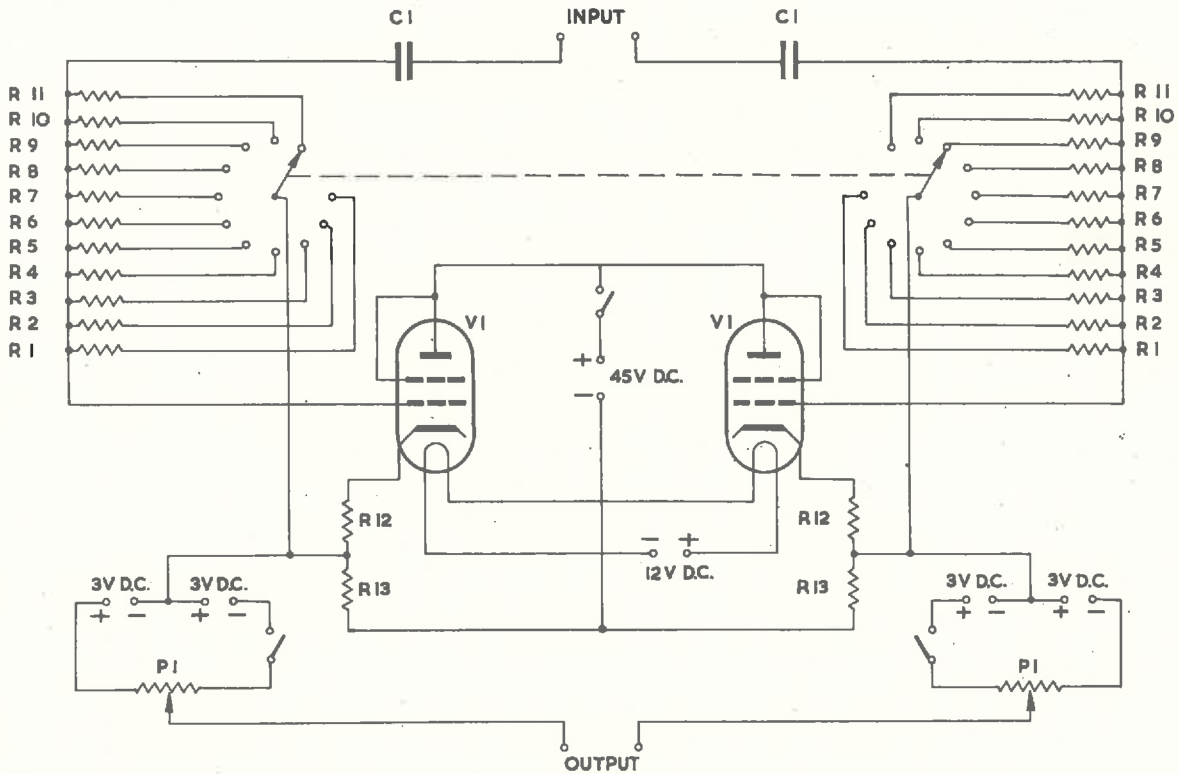
TRANSISTORS		
T1	TYPE	BCZ 10 SILICON
T2	TYPE	OC 44 GERMANIUM
T3	TYPE	OC 44 "
RESISTORS		
R1	100 kΩ	METALLIC OXIDE
R2	40 kΩ	" "
R3	40 kΩ	" "
R4	4·7 kΩ	" "
R5	6 kΩ	" "
R6	30 kΩ	" "
R7	100 Ω	" "
R8	70 Ω	" "
R9	150 Ω	" "
R10	240 Ω	" "
R11	470 Ω	" "
R12	1 kΩ	" "
R13	100 kΩ	" "
R14	5·6 kΩ	CARBON
R15	5·6 kΩ	"
R16	3·9 kΩ	"
R17	10 kΩ	"
POTENTIOMETERS		
P1	1 kΩ	WIREWOUND

Fig. 1



60
40
20
0
-20
-40
-60
DRIFT VOLTAGE [REFERRED TO INPUT] MICROVOLTS
RESISTANCE ACROSS
INPUT = 1000Ω
50
60
70
80
90
100
AMBIENT TEMPERATURE °F

Fig. 2.



VALVES	
V I	TYPE E80F
CAPACITORS	
C I	1 μF
POTENTIOMETERS	
P I	1 kΩ
RESISTORS	
R 1	180 kΩ
R 2	220 kΩ
R 3	270 kΩ
R 4	390 kΩ
R 5	560 kΩ
R 6	1·2 mΩ
R 7	1·5 mΩ
R 8	2·2 mΩ
R 9	3·3 mΩ
R 10	4·7 mΩ
R 11	6·8 mΩ
R 12	1·8 kΩ
R 13	33 kΩ

Fig. 3.