HYDROFOIL CRAFT AND THEIR MARINE ENGINEERING ASPECTS

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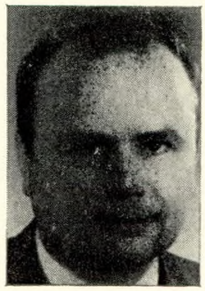

Upon consideration of a general design philosophy for commercial hydrofoil craft, the percentage weight of some structural groups, published by various authors, are discussed and some data are added according to the present state of the art. Design and analysis problems for the layout of long, inclined propeller shafts, as well as for cardan shaft V-drives, are dealt with. A general view is given for power transmission by means of bevel gears. Finally, a brief prospect for assumed future tendencies is given.

In his lecture, Mr. Crewe[1] expressed his opinion that, in future, a number of hydrofoil craft up to 50 tons displacement and a speed of about 40 knots could be built. In fact, 11 years after this excellent lecture was presented, about 125 hydrofoil craft built according to Supramar's designs, were launched. This fleet, spread all over the Western world, represents an accomplishment rarely found in shipbuilding. The fleet not only distinguishes from the conventional displacement, or planing craft due to the foils which reduce drag and increase seaworthiness, but especially because these 125 hydrofoil craft are composed of two or three types. Although the superstructures have been adapted to the present style, the structure design of the hull, foils and especially the engine plant have remained unchanged. A conception found already in 1954 for the propelling unit and the elements used are still today considered as proper and are still manufactured. A number of improvements have doubtlessly been made on hull, foils and engine plant. A few examples are given below:

Hull
The fittings for the foils and the propeller shaft brackets have been redesigned to a more suitable construction for continuous running. Parts of the hull and superstructure have been redesigned from a riveted to a welded construction.

Foils
The design of the foils has been considerably simplified, difficult welding designs have been improved and special attention has been given to the design of the underwater parts, such as rudder and flap bearings.

Engine
The high thermal engine load at take-off and the vast demand made regarding continuous running at a high load under sometimes unfavourable maintenance conditions in the operational areas necessitated various modifications. For example, the support of the cylinder liner in the light-alloy crank case has been reinforced and the cylinder head gasket has been improved.

Gear
Improvements of the solenoid operated clutch and the sliprings became necessary for the aforementioned reasons.

Electrical Plant
The increasing comfort of the passengers, the higher standard of navigational equipments and the operation at night necessitated a replacement of the originally applied D.C. plant with shaft-driven generator and battery by a modern A.C. plant.

Ventilation and Air Conditioning Plant
The all year round operation of hydrofoil craft requires some modifications of the ventilation and heating plants. For certain operation areas, air conditioning plants had to be installed.

Navigation Equipment
The all year round operation of hydrofoil craft, that is also under poor visibility, requires a completion of the navigating aids by a radar plant, a gyro compass and a hyperbolic navigation system (Decca navigator).

Ship's Safety
The Classification Societies and Solas requested for the light-weight design in aluminium an effective fire insulation and the dimensions of the evacuation passages and the capacity of the life-saving appliances had to be considerably increased.

* * *

All the measures mentioned contributed to an increased safety for the passenger, the passenger's riding comfort, an increased operational readiness of the craft and a reduction of the maintenance expenditure. Although these factors resulted in a considerable additional weight, the basic design and its main performance factors, such as speed range, disposable load and price remained constant in principle. Hydrofoil craft equipped with all these improvements obtain

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today an operational performance which can be compared with the operational performance of a Diesel-express-locomotive. Today, a hydrofoil craft covers a distance of 80 000 to 130 000 nautical miles a year, which is equivalent to about 2500 to 4000 operating hours a year.

The daily economical service by about 125 hydrofoil craft, whereby quite a number of craft have been in operation for the past ten years, presupposes not only experience in design and analysis, but also a consideration of a certain design philosophy. This design philosophy originated from observations made with many hydrofoil craft in operation and experiences exchanged with various shipowners. Some of these aspects are:

a) the application of high quality and special material must be kept to a minimum. A decisive criterion when choosing material is that repair work can be executed by yards of small and medium capacity;
b) a light-weight design of high degree should only be applied when the production tolerances do not become disproportionately small.

For both principles, the criterion is "economically acceptable". This partly subjective criterion is formed by the following influences:

1) non-standard materials or special production methods result in an unjustified high price, which is due to the relatively small number of components required.
2) the application of extreme light-weight designs with corresponding low safety factors require a considerably more thorough supervision and results in a respective higher wastage quota; both factors influence the price;
3) the application of non-standard, special material and extreme light-weight designs increases the number of specialists necessary for the manufacture, thus not only influencing the cost, but also the time schedule.

No doubt, these criteria do not represent any definite limits, because they are influenced by the state of the art, the intended use of the craft and the deadline date. Therefore, they must be rechecked continuously and adjusted accordingly. Their basic principle must be maintained. Simply said, it is the hydrofoil designer's task to always find a compromise between aircraft design and shipbuilding design. This problem cannot always be solved easily, as often there arises a conflict between the designer's preference for aircraft design and the possibilities in shipbuilding.

The publications[1,2,3,4] regarding the structure weights are commented on in the following section.

STRUCTURE WEIGHTS

Those structure weights which are missing in the displacement or planing craft will be discussed first (Fig. 1). In this

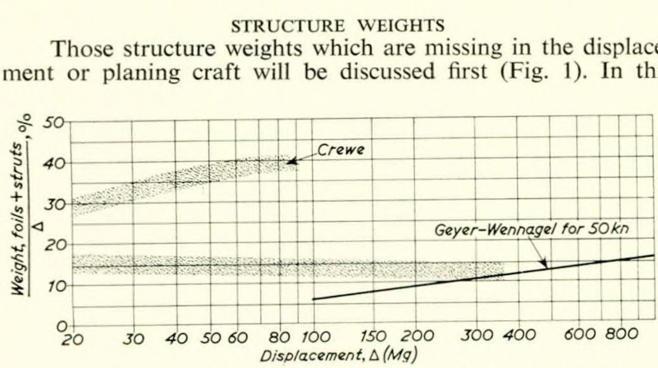


FIG. 1—*Percentage weight for foils and struts*

connexion, it may be mentioned that the figures[1] regarding the percentage weight of the foils and their struts are lower in practice. It can be considered as safe that, without any extraordinary design work and material used, this percentage will be half of the figures expected[1]. Also, in larger displacements, the expected increasing tendency to percentage weight of foils and struts, cannot be confirmed. Contrary, there is rather a tendency to a slight decrease of percentage weight of foils and struts to be observed. In contrast with the experiences gained, information published by other authors[2,3,4] stress values for percentage weights for foils and struts which probably can only be achieved by good design work and a careful choice of material for navy craft. The percentage weight for the hull and superstructure, including its equipment[1], increases by factor 1·1 only (Fig. 2). On the other hand, the figures for the engine plant and

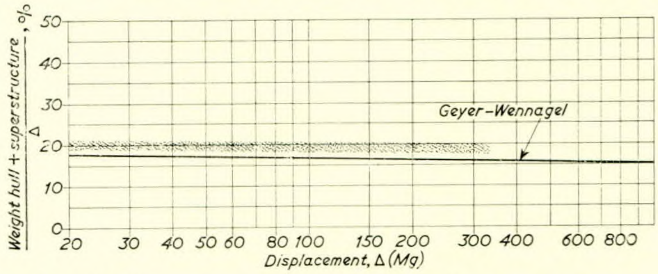


FIG. 2—*Percentage weight for hull and superstructure*

its equipment, valued at 28 per cent[1], correspond with those attained in practice. Unfortunately though, this percentage weight shows a tendency to increase with increasing displacement. The reasons for this phenomenon are set forth in an analysis of this percentage weight. This analysis shows that, at an increased displacement, the percentage weight of the electrical plant, including the Diesel generator sets, increases considerably as well. On the other hand, the percentage weight of the engine and hull pipework remains almost the same. The most important percentage weight, i.e. the disposable load, has been estimated at 24 per cent. In practice, values of only 20 to 25 per cent can be attained. It is not surprising that the authors[2,3,4] publish considerably higher values, up to 55 per cent. The reasons are due not only to a different design philosophy, but also lie in the question whether these values which mostly originate from project studies only, consider the requirements of the Classification Societies and Solas. The published data produce the impression

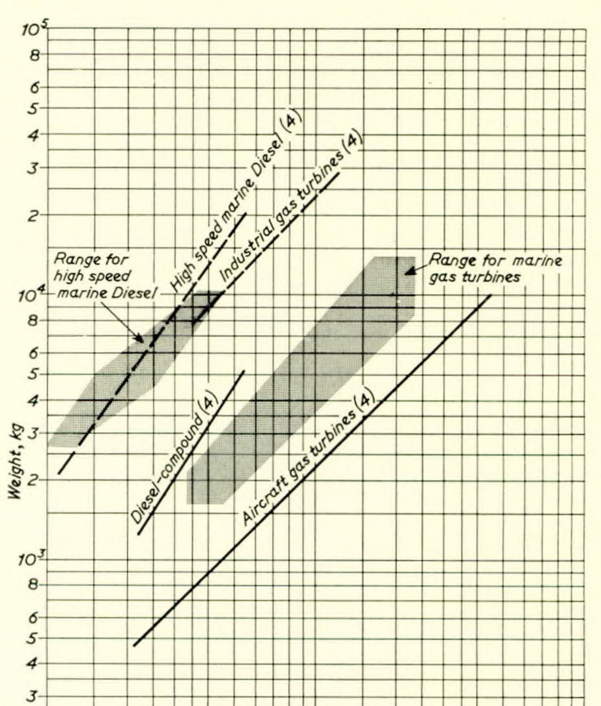


FIG. 3—*Engine weight versus horsepower*

that this is not the case. Apart from that, it appears that for the engine, for example, weights were evaluated which, at the present state of the art, cannot be obtained, as can be seen from Fig. 3.

In practice, the assumed value[4] of 0·32 kg/hp for power transmission system has not yet been attained either. Considering all elements, this value rather comes to about 0·6 kg/hp, whereby it is to be said, that this kind of coefficient is rather questionable, as the latter refers to the power and not to the torque and gear reduction ratio. The weight for the auxiliary systems is set by the other authors[3,4] at 0·1 Δ, which is rather too small. These discrepancies explain partly only the difference between practical values and those of other projects. Obviously, the remaining discrepancies are due to the fact that the passenger equipments, life-saving equipments, and fire protection equipments have been calculated as disposable load. This presumption is confirmed by the missing data on published weight break-downs. The summary of the percentage weight, divided into the following three main groups:

1) hull, equipment and foils;
2) engine plant and equipment;
3) disposable load;

gives a relatively small scatter range for this percentage weight at a suitable plotting (Fig. 4). A shifting of this percentage

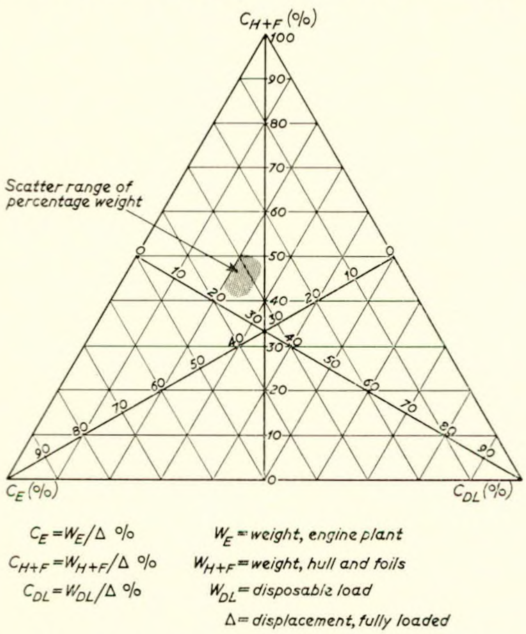


FIG. 4—*Percentage weight for hull and foil, engine and disposable load*

weight of these three groups should only be possible by another design philosophy, which is entirely different from the one from which these data were calculated.

STRUCTURAL GROUPS

The marine engineering for hydrofoil craft differs from that for displacement or planing craft not only by its relatively light-weight design. Problems have to be solved which do not appear at all in displacement or planing craft design or else can be avoided, as such problems are unconventional. Some interesting examples are given later. The reason for starting a reflection on power transmission of a hydrofoil craft with inclined, long propeller shaft is, that about 95 per cent of all craft presently in operation are equipped with this power transmission system. This does not mean that this particular power transmission system represents the best and universal solution. It rather appears that the application limit for inclined, long propeller shafts lies at approximately 5000 hps. This is not only due to the retractability of foils and propeller necessary for larger craft, but also to the fact that the ratio power rev/min at rising power is increasing considerably. For a power of 1350 hp, this ratio is approximately one, increasing to approximately three at 3500 hp and attaining a value of approximately 12 at a power of 20 000 hp. The torque increases accordingly, and the shaft diameter requested by the Classification Society is, at the same time, increased, too. The resulting hydrodynamical situations and respective weights of the power transmission plant set the aforementioned limits.

The following explanations regarding the problems of power transmission by means of an inclined, long propeller shaft should not ignore the fact that this type of power transmission is the cheapest system and has itself proved to be very reliable in operation.

Propeller Shafts

In comparison to the propeller shafts of displacement or planing craft, the propeller shafts of hydrofoil craft have certain peculiarities, referred to later in the paper.

These differences in the hydrofoil propeller shafts arise because the distance between propeller and hull of hydrofoil craft is several times greater than that of displacement or planing craft. This peculiarity, which is determined by the special system, requires the leading of the propulsive power from the engine to the propeller over a considerable distance outside the hull. Therefore, the propeller shaft is for the most part situated externally, in contrast to displacement or planing craft, where most or a considerable part of the propeller shaft is arranged inside the vessel.

For calculation, design, manufacture and mounting of propeller shafts for displacement or planing craft, there exist extensive experimental reports, rules, as well as specifications. These particulars are not applicable at all or only to a certain extent for propeller shafts on hydrofoil craft. First of all, applicable parameters had to be drawn up and had to be discussed with the Classification Societies. Only wide limits are set to diameter, weight and bearing distance of propeller shaft for displacement or planing craft. For hydrodynamical reasons and in consideration of the weight/power ratio, the propeller shaft's diameter and bearing distance of hydrofoil craft are quite limited. Also, hydrofoil craft propeller shafts have to operate under far more disadvantageous ambient conditions than those of displacement or planing craft. These more unfavourable conditions arise from the relatively large inclination of the rotation axis, compared to the flow direction of the water and due to the considerably higher speed of craft. Consequently, the quasi-statical force of the shaft is differing as to quantity and also quality. In addition, the relatively large bearing distances, which meet the hydrodynamical requirements, result in dynamic propeller shaft properties, qualitatively different to those of displacement or planing craft already known. There are these groups of problems to be considered:

1) stress analysis;
2) material properties;
3) vibration analysis.

The following explanations can be given with regard to these three main factors:

Stress Analysis

Apart from the torque moment and propeller thrust which also occur on displacement and planing craft, a force perpendicular to the thrust direction with centre of effort on the propeller is applied on the under-water parts of the hydrofoil propeller shaft (Fig. 5). Also, a hydrodynamic force occurring along the shaft between the bearings is being applied. These two kinds of forces either do not occur at all on normal displacement or planing craft, or they are of no great importance, so that they are usually not considered at all. Both forces are of fundamental significance for the propeller shaft stress analysis of hydrofoil craft. From the force diagrams shown a combined buckling and flexural stress for the propeller shaft results. Before going into more detail regarding the practical analysis method the causes of these forces will be investigated. The variation in speed and consequently the angle of attack, at full revolutions of a propeller blade, results in an asymmetrical force distribution, which generates transverse force F_y. Aerodynamical test results[5] concerning the values of these forces exist, which do not offer any reliable particulars. This is due to the higher pitch and surface ratios usual in shipbuilding. The measurements made on full-scale hydrofoil craft in operation were not in conformity with those values which were theoretically attained. These values were determined by means of the impulsion theory on jet deflexions[6]. In the year 1964 only, the published complete tank tests[7] made a reliable sizing of the force F_y. The ratio of this force F_y to the propeller thrust is in the range of the usual surface ratio of 0·8–1·0 of hydrofoil propellers about 30 to 35 per cent of the propeller thrust, whereby the smaller value for a pitch ratio is 1·1 and the greater value for a pitch ratio is 1·3. The eccentricity of the thrust centre of effort which arises from the same causes is about 15 per cent of the propeller diameter.

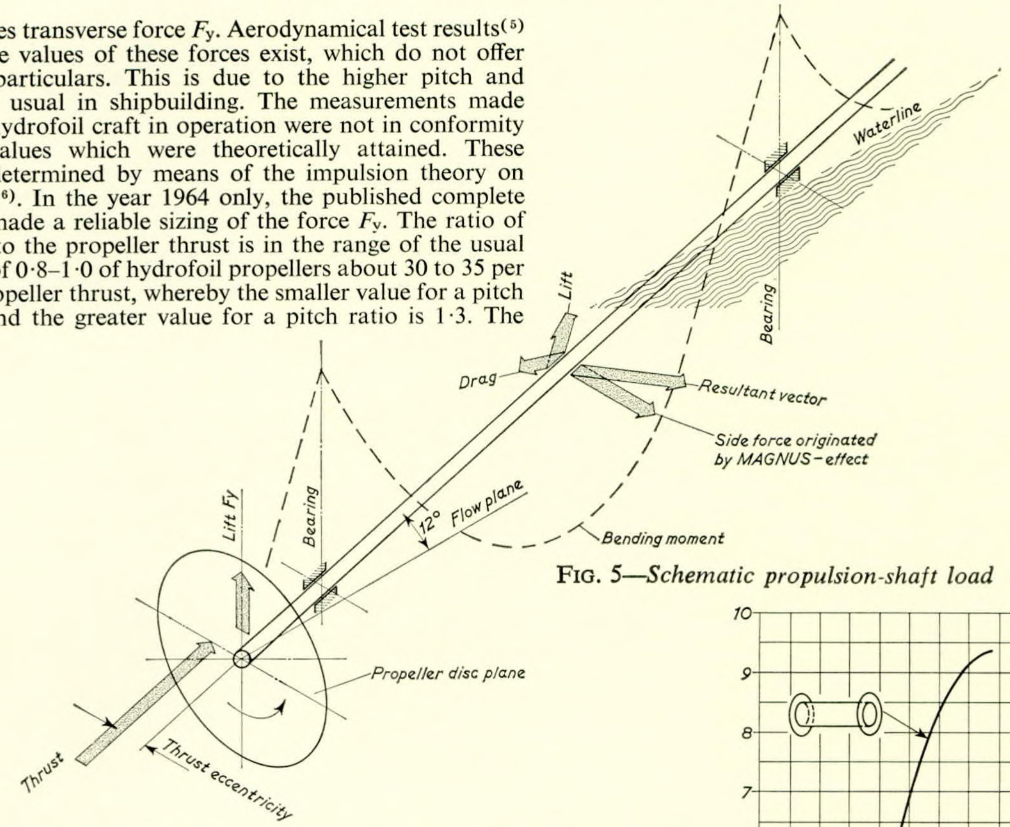


FIG. 5—*Schematic propulsion-shaft load*

The hydrodynamic force along the propeller shaft is composed of lift, resistance and the so-called Magnus effect. The latter influences the quantity of the resulting force most of all. The mean value generated by the Magnus effect is dependent on the mean immersion of the propeller shaft. A mean immersion of the propeller shaft means the ratio between the overall length of an elliptical cross-section of the shaft on flow level and the distance to the water surface, measured on half immersed length. The experimentally determined coefficients at an immersion ratio of about 1·0 quite coincide with the coefficients of a rotating cylinder without end discs[8], known from hydrodynamical experiments. At an immersion ratio of about 2·0, the coefficients determined by experiments, compared with those of full-scale craft, do not correspond (Fig. 6). Here it is encountered as a similar phenomenon as is known from the hydrodynamical immersion effect. The coefficient calculated from the measured stress on the shaft, is of the same magnitude as that of a rotating cylinder with end discs. In view of the simplifying assumptions that the load distribution along the shaft between the bearings would be constant, this consistence can be considered as favourable.

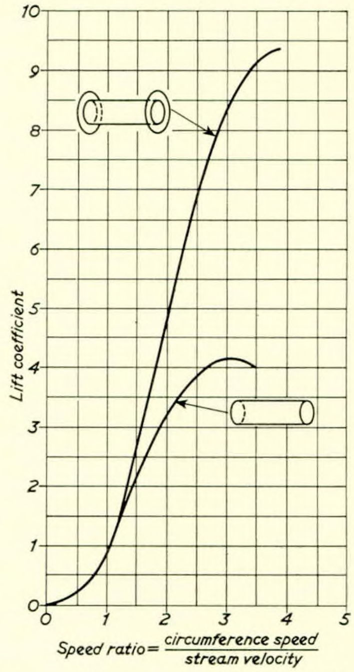


FIG. 6—*Lift coefficient originated by Magnus-effect*

Propeller Shaft Material

Many materials have been tried as protection for non-corrosion resistant, heat-treated steels for the propeller shafts. These steels have been coated by means of paint, plastic lamination, or sprayed on by corrosion-resistant metals. Although the use of such material, with the resulting protection, will be less expensive than the application of heat-treated, stainless steels, this solution has not prevailed on the market. This can be explained by the difficulties which arise when one tries to protect the region of the propeller boss or the shaft bearings, in order to guarantee a longer operation period. Even if this problem could be solved properly, durably and without having to consult special workshops, there would still remain the problem concerning the damage of protective coatings by drift wood during operation. Today's most commonly applied materials for propeller shaft are solid stainless steels adequate to the following standard designation, listed in frequency order of popularity application:

X 22 CrNi 17
17–4 PH
X 35 CrMo 17

Material 17–4 PH is a maraging-steel which is exclusively used in Japan and aged for the following qualities:

a) tensile strength 100 kg/mm^2;
b) yield point 75 kg/mm^2;
c) elongation 22 per cent.

The first material mentioned is used exclusively in Europe; the last-named material represents an alternative to the maraging-steel. The material used in Europe is tempered at extremely high temperatures (Fig. 7), in order to reduce internal stresses to a large extent. This measure gives better fatigue strength in sea water environment.

Experience gained from damage investigations has shown the necessity of limiting the forging ratio to six maximum for a 5 m long stock as is laid down by the Classification Societies

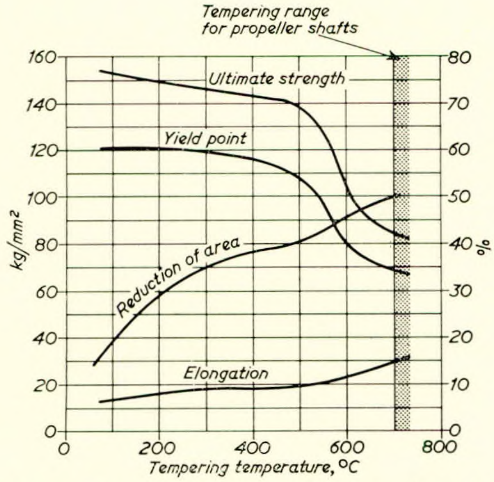


FIG. 7—*Tempering diagram for shaft material X22CrNi17*

for crankshafts, whereby this limit is set at five. Apart from that, it has proved necessary not only to determine the notch impact value in the longitudinal direction at minimum 4 mkg/cm^2, but also to require a minimum notch impact value of 2·5 mkg/cm^2 in transverse direction. These tests must be made three times on each of the two shaft ends and they serve as a control over the occurrence of solid non-metallic inclusions or an imperfect ferrite-containing structure, which produces the same effects as non-metallic inclusions. The extremely important flexural fatigue strength required by the forces on the propeller shaft, comes to about 47 kg/mm^2 in the air. Only incomplete data[9–14] are available regarding the flexural fatigue strength in sea water; they all point to a reduced strength despite the sea water resistant qualities of the material (Fig. 8). But the influence of sea water has not

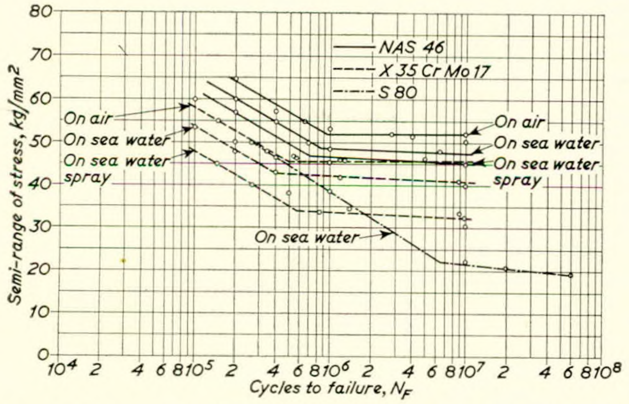


FIG. 8—*Flexural fatigue strength of different stainless steel*

only quantitative, but also qualitative effects, because a fatigue limit is no longer defined. Contrary to practice in aircraft design, where one reckons with load cycles of 5×10^3 to 10^5, fatigue limit is within the range of 6×10^8 for a desirable lifetime of the propeller shaft of 10 000 operational hours. The lack of a defined fatigue strength in sea water environment is, at these high load cycles, most important.

In view of the few available particulars on corrosion fatigue, Hitachi have made trials with X 35 CrMo 17, and it was found that a reduction of 11 per cent of values measured in air, occurs after 10^7 load cycles. In sea water spray, this reduction comes to 31 per cent. At the same time, Hitachi's tests with the material NAS–45, corresponding roughly to type 17–4 PH, showed a reduction of 12 per cent. On the other hand, tests made by Armco Steel Corporation[15] resulted in a reduction of up to 40 per cent of those values measured in air, another author stated higher effects of sea water[16]. According to tests made by Hitachi, the quality of 17–4 PH is considerably better than that of X 35 CrMo 17; in other words, there is a smaller sea water influence on the fatigue strength when 17–4 PH is used. As a comparison, there exists a different test result[15]. It is of interest to see that the curve mentioned in *Stainless Iron and Steel*[16] is considerably lower, i.e. below those measured by Hitachi and, apart from that, this curve shows a sharper drop following the knee than can be observed from the Hitachi values.

In the view of these uncertainties (Fig. 9) and the defects on the surface to be expected from transportation and sea damage by driftwood, and short period operation with an unbalanced propeller resulting from damage, the nominal reverse flexural

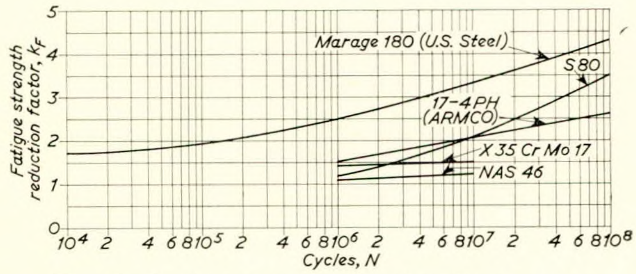


FIG. 9—*Fatigue strength reduction by seawater*

stress is normally limited on design to 6 kg/mm^2. In comparison with these very low values obtained through the *Stainless Iron and Steel*[16], the foregoing results in a calculated security factor against fatigue failure of about two, reaching a minimum after 10 000 operational hours.

Vibration Analysis

A final determination of the flexural frequency of the propeller shaft is made more difficult for the following reasons:

1) non rotary-symmetrical inclination and displacement stiffness of the bearings;
2) bearing clearance;
3) buckling and torsional load.

The measurement on full-scale construction, showed the possibility of checking, and of correcting the previously assumed figures for the inclination and displacement stiffness. The determination of flexural natural frequency of the elastically bedded shaft is calculated by a computer programme, which applies the determinant matrices. For each flexural natural frequency found, the respective form of displacement is also calculated.

As a common result of a number of calculated shaft arrangements with different bearing distances, bearing stiffness and bearing diameters it has been determined that the first degree of flexural natural frequency under certain conditions is always below the idle running speed of the propeller shaft. The difference between flexural natural frequency of first degree in horizontal and vertical directions can be limited to about ten per cent, despite the non-rotary symmetrical stiffness. The second-degree of flexural natural frequency lies within a speed range which is run through during take-off at normal operation. For all calculated shaft arrangements, the third-degree of flexural natural frequency lies above operational range.

The full-scale measurement (Fig. 10) showed flexural natural frequencies which exceeded the calculated values by approximately ten per cent. The checking of the rubber bearing hardness in the full-scale construction, from which the measurements had been taken, revealed that it was of harder quality than that usually applied and calculated. A new calculation with values adequate to the harder rubber bearings have shown not only excellent agreement with practical results, but also the influence of the hardness of the rubber of the bearings on the flexural natural frequency of the shaft.

According to calculation and measurement results of the full-scale construction, the propeller shaft of a hydrofoil craft runs "overcritical". Although two natural frequency values have

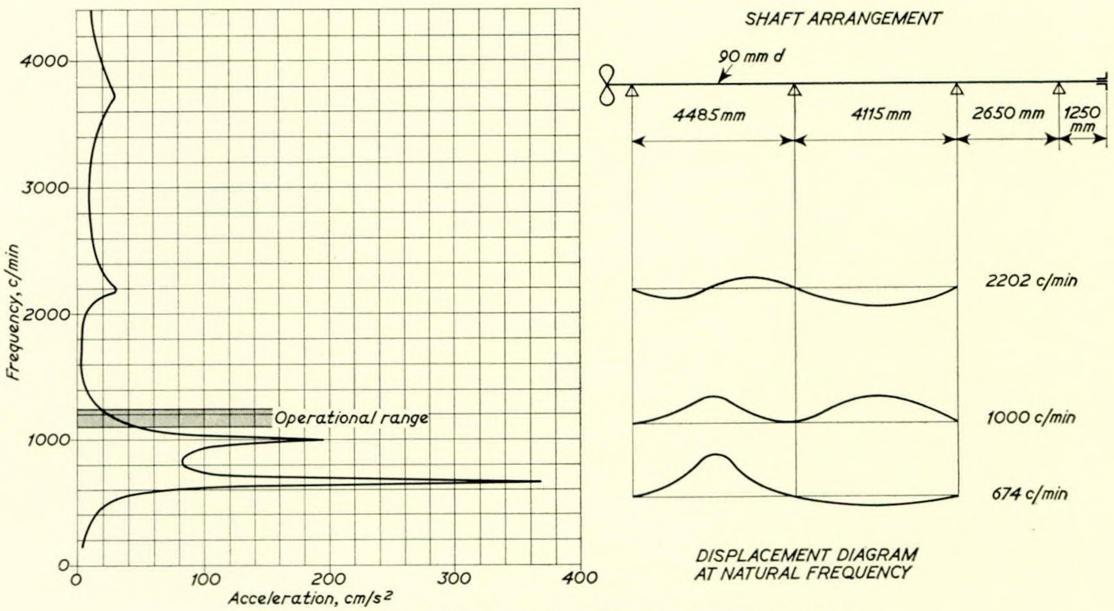


FIG. 10—*Flexural natural frequency of propeller shaft*

to be run through, no important difficulties arise in practice. This is not only a subjective observation derived from the lifetime of a propeller shaft; it is rather proven by measurements on the bearings and on the underwater rotating shaft itself. Despite theoretically determined and measured proven flexural natural frequency values on the shut down shaft arrangement, when mounted on shore, no distinctive resonance phenomenon could be measured.

The explanation for the non-appearance of a distinctive resonance phenomenon for the first degree natural frequency lies foremost in the fact that this frequency range is run through at high acceleration by the propeller shaft during engagement of the coupling in the reverse gear. The non-appearance of the distinctive resonance phenomenon at a second degree of flexural natural frequency cannot be easily explained. Strictly speaking, the deflexion value with which the shaft, in case of resonance, rotates around its theoretical axis of revolution, is independent of damping, because no vibration arises. If the shaft were statically deflected in a determined direction and to a multiple of the eccentricity of the centre of gravity before a resonant condition is attained, a vibration arises here as well because the shaft deflexion changes during each shaft revolution and thus damping forces are overcome.

Unfortunately, insufficient data are available on the damping coefficient. According to measurements, the damping coefficient is 6·0 to 8·5 per cent of the critical damping, calculated of the amplitude ratio of the measured flexural natural frequency of first degree. The damping value calculated from the relative half-value-width is five per cent of the critical damping, at a frequency of 10 Hz, and approximately seven per cent, at a frequency of 37 Hz. These values are normal for vibrating structures, although they have also been measured underwater. The amplitudes on which these damping values have been measured, were rather small. The structural damping, is a function of the vibration amplitude[17,18,19], and, therefore, ought to be greater for the amplitudes to be expected in practice. Additionally, the damping force vector is in phase with the motion vector of the vibrating structural part, a fact which raises special theoretical questions, not discussed here.

Universal Joints

The distance between propeller and hull of a hydrofoil

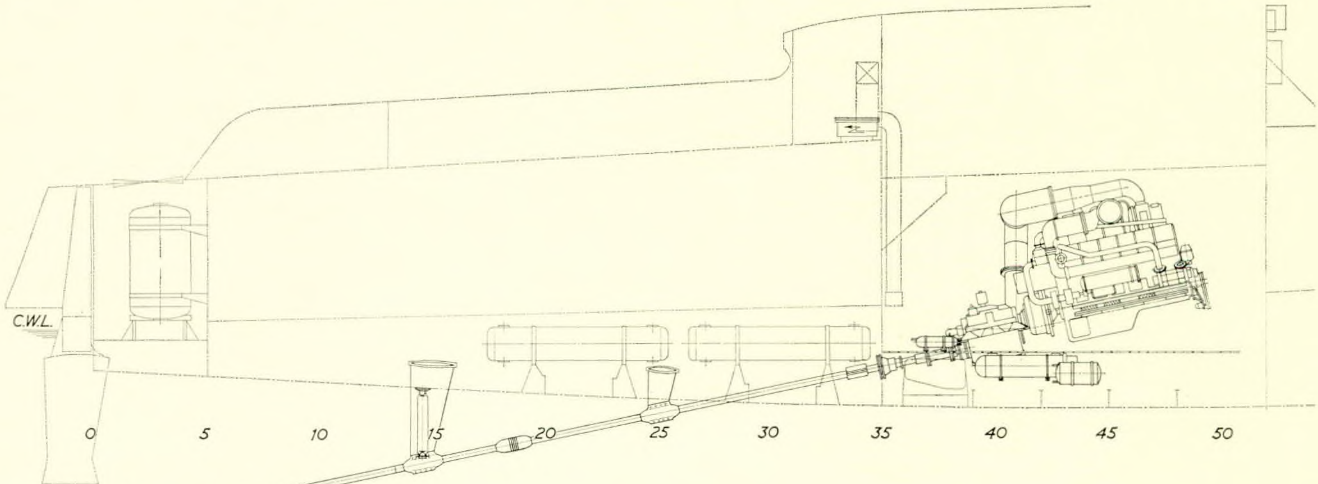


FIG. 11—*Main propelling machinery straight drive*

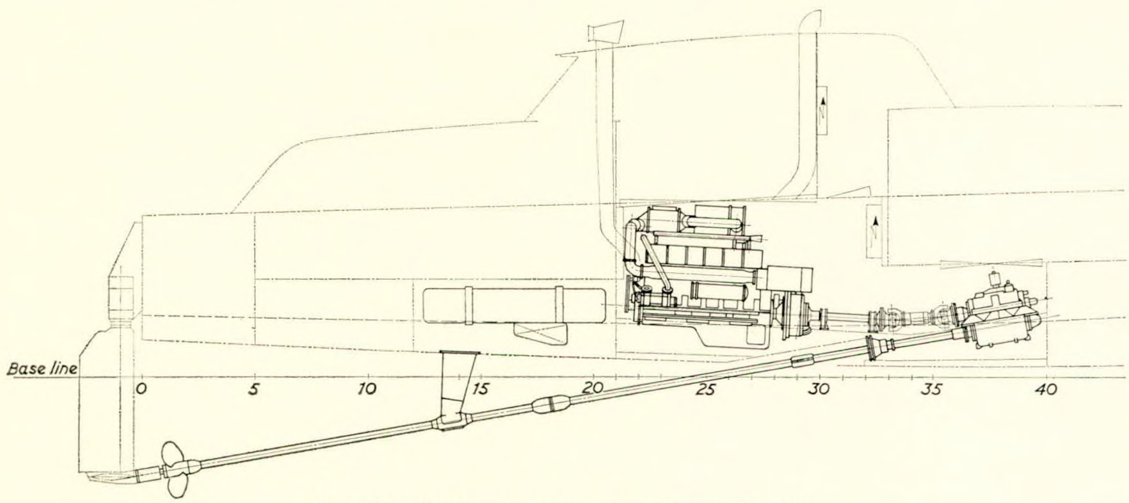


FIG. 12—*Main propelling machinery, V-drive*

craft does not only require a relatively long propeller shaft, situated outside of the hull, but there is also a relatively large inclination of same; however, the shaft should not be unnecessarily long. This inclination of the shaft against the horizontal, and therefore propeller flow, is limited by the variation of incidence of the propeller blade elements during revolution[20]. At an oblique flow of the propeller of 10° and a number of advance of 1·0, the incidence of a blade element at 0·6 R reaches to 4·5°. At an increase of the oblique flow to 15°, this variation of incidence is 7°. These variations are considerably higher in the region of the boss. In consequence of this variation of incidence, not only do thrust and torque fluctuations occur, but also cavitation, especially in the region of the propeller boss. Therefore, the inclination of the propeller shaft of hydrofoils rarely exceeds 12° in practice. This inclination measured in relation to the base line of the craft is increased by trim and by the downwash of the bow foil, so that an oblique flow of the propeller of about 15° has to be reckoned with.

In consequence of an inclination of the propeller shaft of about 12° in relation to the base line of the craft, the engine room is normally situated in the fore end of the craft and the engine has to be mounted to suit this inclination (Fig. 11). This requires a specially designed elastic mounting and careful alignment. Also, the manufacturer of the engine has to pay careful attention to ensure that lubricating oil system functions under such an inclination, which increases during take-off and when operating in a seaway. Although all problems in this respect can be solved, it is frequently necessary to arrange the engine in horizontal position. Neither the position, height of engine room, resulting from the inclined arrangement, nor the centre of gravity of the engine plant situated in the foreship, are in conformity with the hull design requirements.

Shifting the engine's centre of gravity towards the stern of the craft, with the engine inclined at a smaller angle than the propeller shaft requires a so-called V-drive (Fig. 12), which can be realized by means of bevel gears, or by universal joint coupling. However, the latter is possible only if angles up to 13° are considered. A V-drive by means of bevel gears is considerably more expensive than that of universal joint couplings. For a transmission power of 4000 hp, the bevel gears, including reverse gear, are four times more expensive than a universal joint coupling and reverse gear. Apart from that, there are only a few such specially designed bevel gears available which have the required angle. The propeller shaft inclination often requires certain corrections after tank tests have been carried out, as, for example, when the position of the propellers in relation to the rear foil, has to be improved. This subsequent angle correction does not present a great problem in the V-drive arrangement with a universal joint coupling, as long as the admissible deflected angles are not exceeded. However, a design with bevel gears will necessitate considerable foundation alterations, because the angle of the bevel gear is fixed and thus cannot be adapted to the separate requirements of special designs for economic reasons. As opposed to a design with bevel gears, a V-drive with universal joint coupling and a reverse gear of series production is, therefore, less expensive, because the universal joint coupling, as well as reverse gear, has a considerably wider range of application and is thus manufactured in series.

The pulsatory forces (Fig. 13) occurring are not compatible with a V-drive with universal joint couplings which require certain precautions with regard to a light-weight design. These precautions need to be taken at the design stage and require an arrangement which allows sufficient distance between both joints. The greater the distance, the smaller become the side forces, pulsating with a frequency equivalent to a double rotational frequency, and which are in a plane perpendicular to the angularity plane. Unfortunately, there exists no absolute limit of these admissible side forces. This depends on the design of the bedding and on the foundation possibilities. These possibilities are rather limited for shipbuilding and especially for hydrofoil craft building; among other things, there is a lack of mass in the foundation. Additionally, there should not be too great a distance between the joints, because the increased risk of shaft whip and the critical speed have to be considered.

The product rev/min angle of one joint yields an approximate value for the control of the shaft whip between the two joints. The admissible limiting value can only be determined by test runs in the manufacturer's workshop. It is known from experience that there exists no firm limit, but a scatter band which still allows operation.

From experience gained it can be considered as safe to use universal joint couplings economically in shipbuilding and especially on hydrofoil craft, provided the characteristics are considered. It would be desirable to arrange elastic bedding of the driving and the driven parts, connected by means of a universal joint coupling, instead of oversized foundations. For the layout of such elastic beddings, the forces and moments generated by a universal joint coupling, working under a defined angle, must be considered. The forces, working on the bearing of the input shaft (Fig. 14), respectively output shaft, resulting from these forces and moments, could be represented by an auxiliary system which is composed of a static force vector D and a force vector E, rotating round an imaginary centre of rotation. The rotation frequency is double that the rotation frequency of the input shaft. The size of these force vectors is substantially influenced by the reciprocal of the joint spacing.

If the universal joint coupling is directly connected to the elastically mounted engine or to the gear flanged to this engine, elastic bedding of the driving part of the universal joint coupling is automatically achieved and special precautions will not be necessary. The strength of the shaft bearing in the engine or the gear flanged to it has to be rechecked for the forces and moments

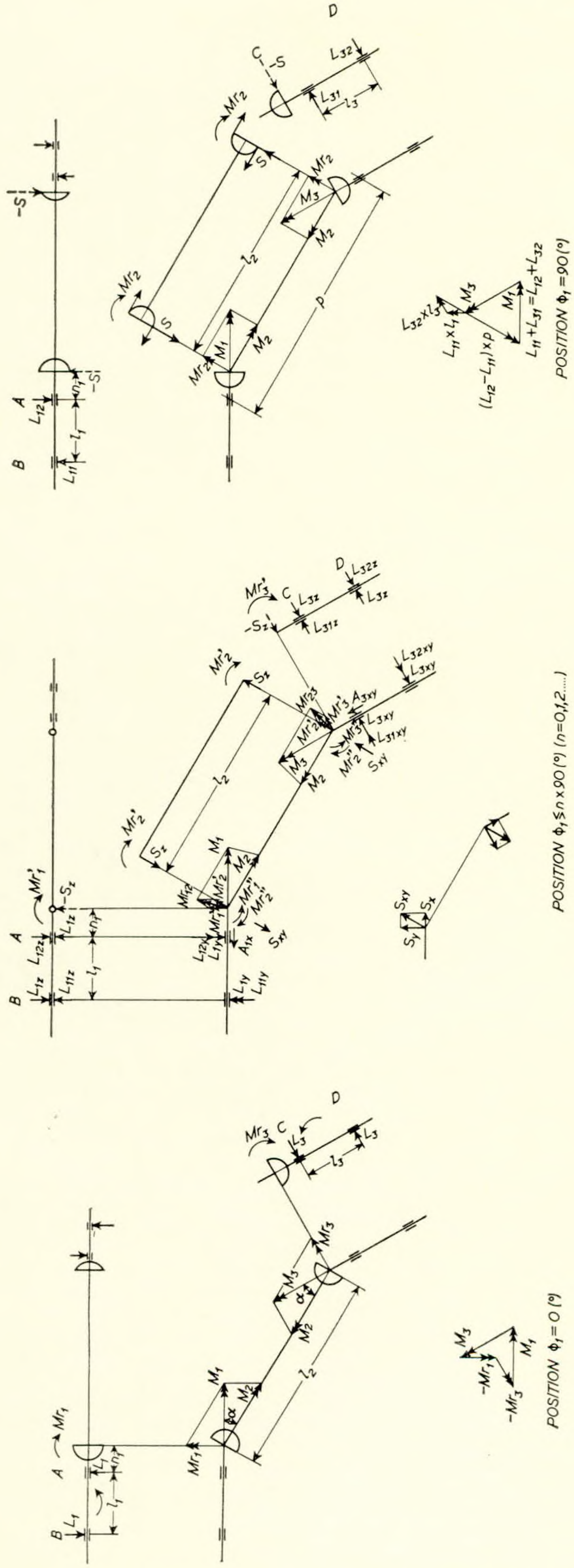


FIG. 13—*Forces and moments by cardan shaft*

Illustration from reference (21)

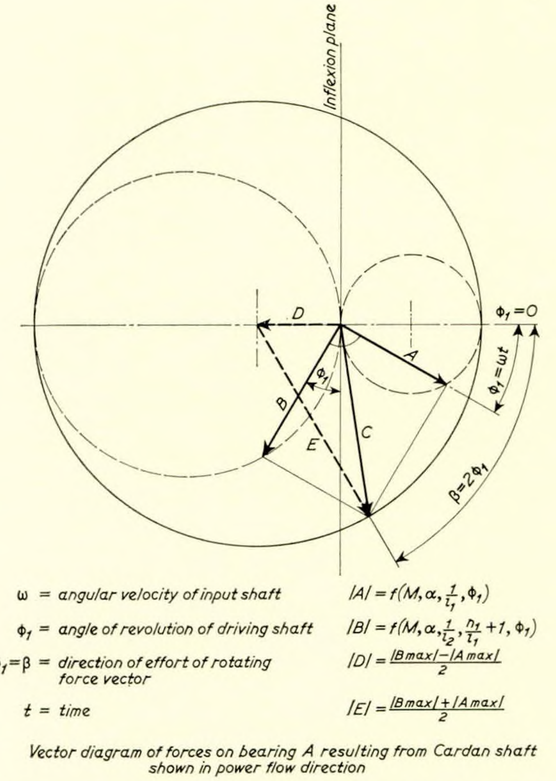


FIG. 14—*Load vector diagram for universal joint bearing*

generated by the universal joint coupling. For the elastic bedding of the reverse gear, which is driven by the universal joint coupling, some care concerning design and calculation of natural frequency become necessary. The constructional arrangement of the elastic bedding of the reverse gear demands consideration of the fact that the latter picks up propeller thrust, simultaneously, which requires, in its direction of action, a considerably higher rigidity of the elastic bedding than is necessary in the main direction of the forces and moments, generated by the universal joint coupling. A reliable analytical treatment of the vibration behaviour of an elastically installed reverse gear requires the experimental determination of the centre of gravity and the mass moment of inertia, in relation to the three main axes. It is important for the lowest possible value of both coupled natural frequencies in the perpendicular plane to the propeller thrust, that the ratio of stiffness between the horizontal direction and the vertical direction of the elastic bedding be as small as possible. The ratio of gear height to bearing distance is rather limited constructionally. The influence of this value at a small value of the aforementioned stiffness ratio to the value of both coupled natural frequencies is small.

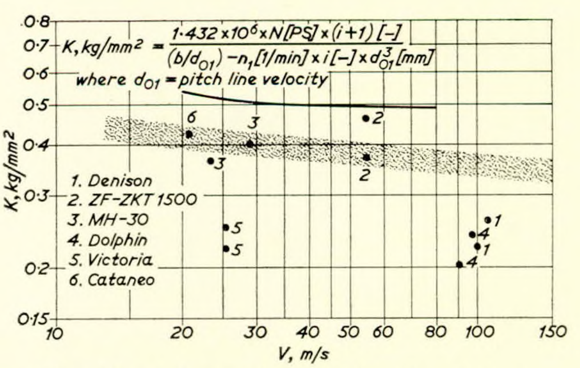


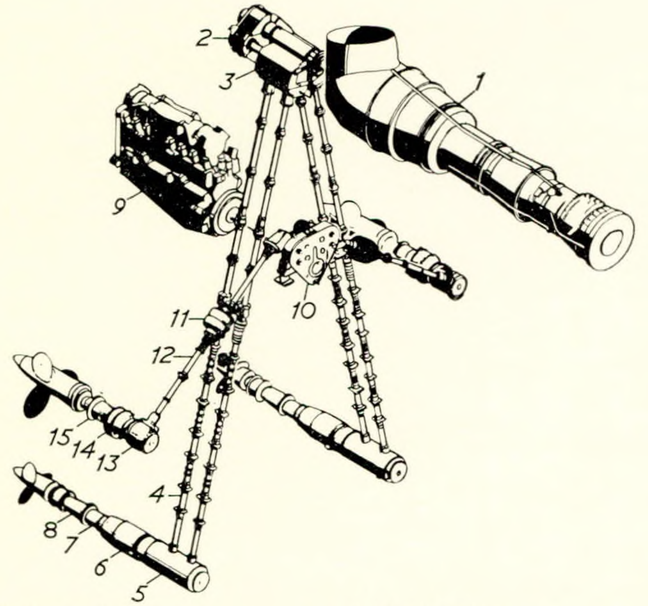
FIG. 15—*K-factor diagram for Bevel gears*

Bevel Gears

Already in 1943 the propulsion plant of a hydrofoil craft of the Schertel-Sachsenberg System through bevel gearing had been achieved. This solution has shown some disadvantages, as the design work involved and the production expenditure for such a special construction are quite considerable with a corresponding effect on overall costs. If, despite this effect, bevel gears are used, this is not only because this kind of power transmission is more elegant than the inclined, long propeller shaft, but because the technical requirements, i.e. the retraction device of foil necessitates this kind of power transmission.

With the present state of the art, spiral bevel gears are practicable up to a root-strength of teeth of 21 kg/mm^2 at a pitch line velocity of 92 m/s. The actual limit for such case-hardened gears lies at approximately 750 mm in diameter. These limits show that a transmission of approximately 12 000 hp per meshing is possible. The attainable values obtained up to now at continuous running with the Gleason tooth system are plotted in Fig. 15 as a K-factor used in the U.S.A.(22,23,24,25,26). The plotted band represents the range recommended by Gleason as the limit. The curve plotted on top of it is quoted from the American study(4) on large hydrofoil craft, reproducing values which would seem to be valid for short period operation.

The Denison craft of the US Maritime Administration is the best known hydrofoil craft which has been successfully equipped with a bevel gear for transmission of a relatively high output. Quite a number of troubles developed during the trials of this new design. Upon close consideration of the sources of trouble, it can be stated that these were not peculiar to the bevel gear. Moreover, it seems that they are caused by the too optimistic design for underwater operation, which, amongst other



1) Gas turbine
2) Foilborne inboard helical gear-box
3) Foilborne inboard bevel gear-box
4) Foilborne down shafting
5) Foilborne outboard bevel gear-box
6) Compound star gear
7) Automatic clutch
8) Foilborne propeller cartridge
9) Diesel engine
10) Displacement inboard gear-box
11) Air clutch
12) Displacement down shafting
13) Displacement outboard bevel gear-box
14) Planetary gear
15) Displacement propeller cartridge

FIG. 16—*Power transmission system of FHE 400 hydrofoil*

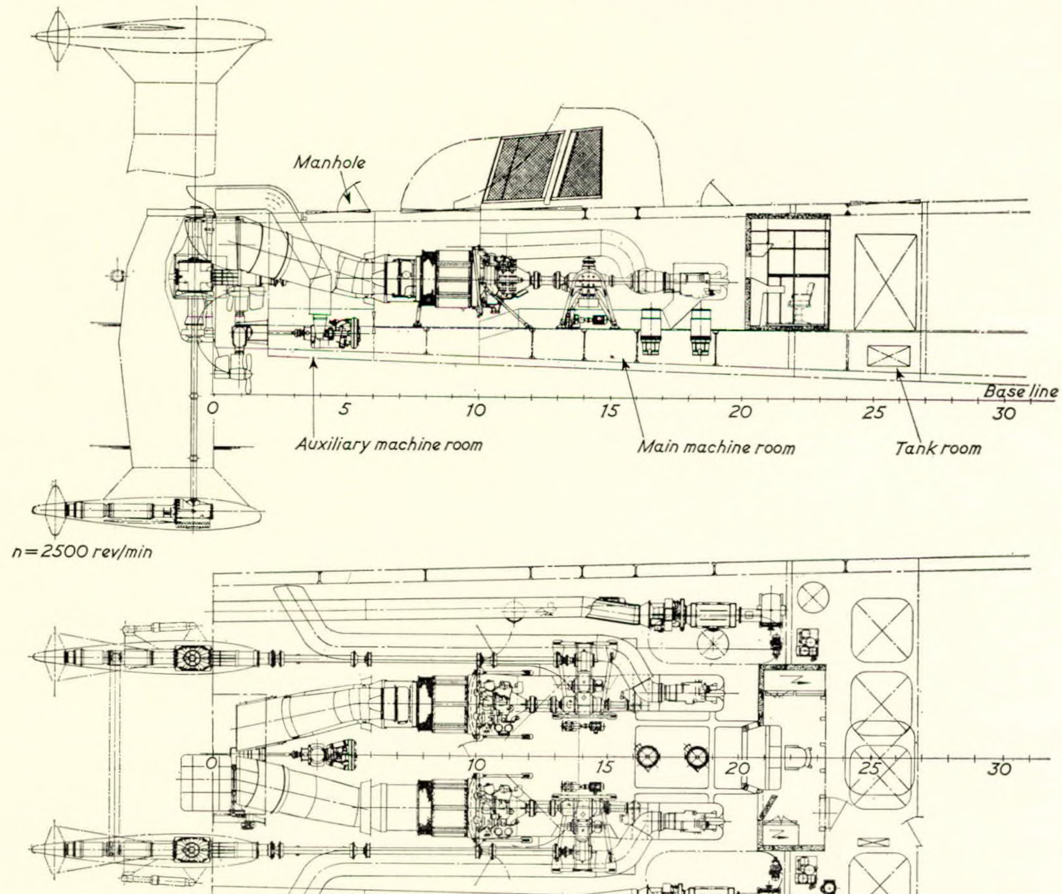


FIG. 17—*Main Cogag propelling machinery for hydrofoil FPB*

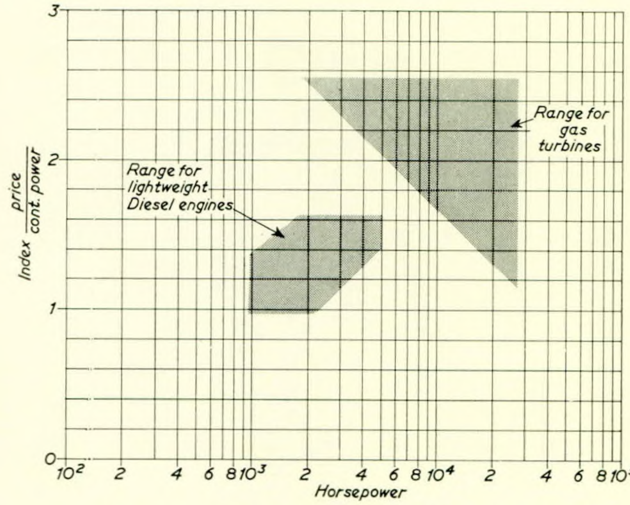


FIG. 18—*Engine price versus horsepower*

things, should have made possible the achievement of extremely low weight.

Bevel gears put successfully into operation in Japan have shown that this type of power transmission can be reliable under service, provided a certain weight is accepted, and those built recently in the U.S.A. have shown that experience gained with the Denison craft was valuable. It has proved expedient to add a planetary gear to the bevel gear, placed between the lower bevel gear and the propeller. Investigations have shown the feasibility of arranging such a planetary gear in the cross section area of a heavy-duty bevel gear. Thus it will be possible to take advantage of high speed and smaller bevel gears and the consequent smaller vertical shaft diameter without having to lose the efficiency of low propeller speed. No doubt, such a solution implies further difficulties and increases in price.

Fig. 16 shows the schematic arrangement of the propulsion system of a hydrofoil craft now under test for the Canadian Navy. This craft is designed for an output of 22 000 hp, transmitted through a gear-box by two inclined bevel gears and a planetary gear to the propellers.

FUTURE TENDENCIES

It can be assumed that for power transmission systems the long, inclined propeller shaft, as well as bevel gears, will be adopted, each system in accordance with its specific advantage. Complete replacement of the Diesel engine by the gas turbine is not expected. It is rather expected that the Diesel engine for an output of up to 5000 hp will be used as a main propulsion engine in hydrofoil craft. This is in consequence of the lower price and also because it is now and will be in the near future, easier for the shipowner to find a skilled crew for the operation and maintenance of Diesel engines, rather than for gas turbines.

However, for navy, different aspects have to be considered, as fast patrol boats in Fig. 17, which shows a CoGAG plant with 4300 hp twin-main gas turbines and 1000 hp twin-boost gas turbines. For higher outputs, a tendency is shown for combined propelling plants, i.e. CoDAG plants. In this connexion, it is interesting to observe that a plant consisting of several Diesel engines is in the same price range as a gas turbine plant (Fig. 18). On the other hand, a plant planned for a navy craft (Fig. 19) is also of the CoDAG type, whereby 12 000 hp twin-main gas turbines and 4000 hp twin-boost Diesel engines, the latter provided for hullborne cruising, have been provided for. This combination also seems to be suitable for civil use in a somewhat modified configuration.

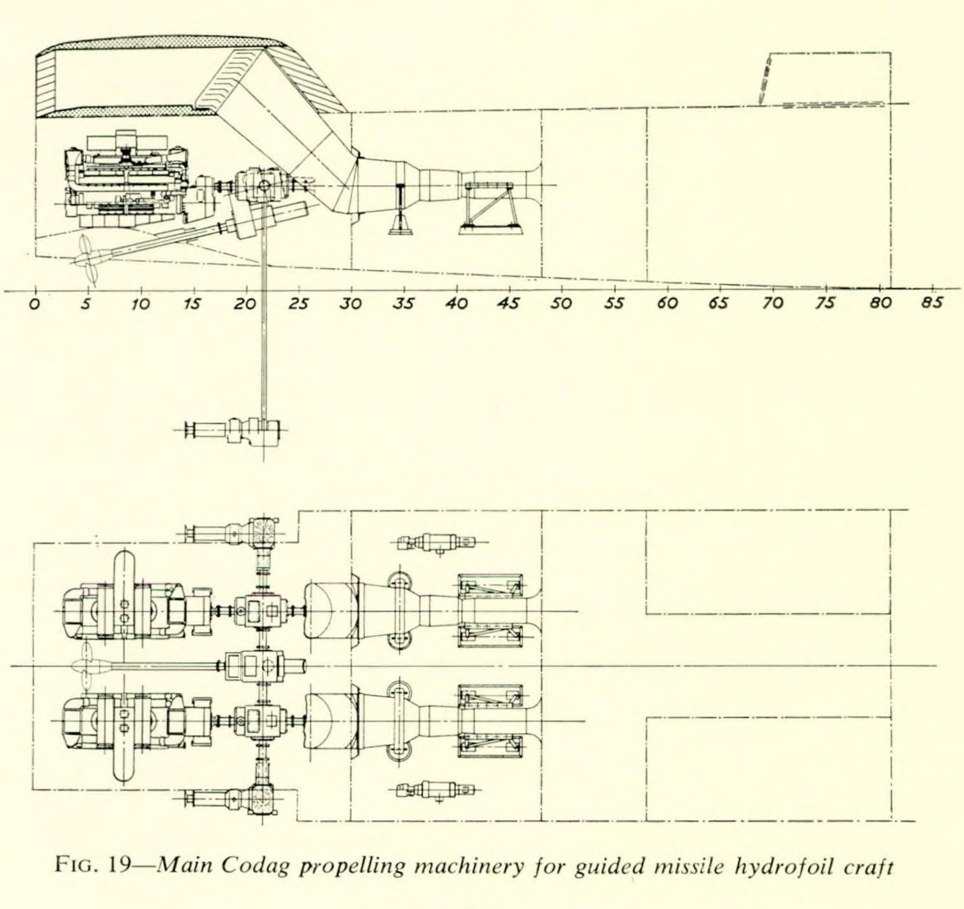


FIG. 19—*Main Codag propelling machinery for guided missile hydrofoil craft*

ACKNOWLEDGEMENTS

The author wishes to thank Supramar Ltd., Lucerne for permission to publish this information. The author also owes a personal debt of gratitude to a number of Supramar Ltd.'s staff who helped in the preparation of this paper and the illustrations.

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Discussion

MISS JUANITA KALERGHI opened the discussion by saying that the author was to be congratulated on presenting a paper on hydrofoils which dealt with craft built with their commercial operation and exploitation in mind.

Following so many various and conflicting published statistics concerning weight break down, disposable loads and weight fractions of hydrofoil systems, it was good to see published figures with an indisputable background of solid experience. This practical experience was reflected in the type of modification made to successive craft, built by the author's company. It was also refreshing to see, in the paper, a list of those factors considered in the design philosophy. Those criteria could well be more frequently applied to other aspects of marine engineering, outside the hydrofoil field.

The influence of SOLAS equipment requirements on hydrofoil weights were almost universally found to have a very adverse effect on the operational economics of commercial hydrofoils. Could the author enlarge on this point?

Regarding structure weights, it was interesting to note that, as a percentage of total weight of the craft, the percentage for foil and strut assemblies was stated to be lower in practice than was predicted. Could the author confirm that his comments applied there only to craft produced by his own company?

She was surprised at the statement that this percentage did not rise for the larger craft. Up to the present it had been widely accepted that such a rise would form a likely barrier for large hydrofoils of the surface-piercing type. Could the author say if that statement referred only to existing craft? Or would he expect this to hold true for much larger vessels, in excess of the size of a PT 150, for example? Could the author also say if the reduction in the percentage figure for foils and struts which was mentioned was influenced by the fact that the larger Supramar craft were fitted with an air stabilization system?

A point to bear in mind, when considering weight breakdown figures, was that they were based on the use of existing materials. By the time that large hydrofoil craft, which one hoped to see, were being built, probably the use of new materials such as carbon fibre for such highly stressed structures would be commonplace and any limitation on craft size due to what might be called "engineering difficulties" would be less severe.

In the text there was very little mention of the transmission installations aboard PT150DC. Could the author enlarge on this and the sort of performance which had been achieved in practice? One feature unique to this craft was the use of a torque converter in the transmission to provide more popular propeller torque during take-off. Could the author explain further the purpose of this device and give information on its value in practice, its reliability, etc.

In studying the last section of the paper she had been struck by the fact that the author had not mentioned water-jet propulsion at all. For simplicity and reliability this system had much to recommend it and, although she realized that no large commercial craft yet used this method of propulsion, it could well be that future developments in this field would make it an attractive proposition. Would the author care to comment on the use of this system?

As regard the statement that skilled crew for the operation and maintenance of Diesel engines were easier to find than for gas turbines, in respect of operation, she did not feel this was a valid point in favour of the Diesel engine, as, with the exception of naval craft, which were a special case, there were very few occasions when any work was carried out at sea on a hydrofoil's engines, whatever their type. They were invariably operated and monitored from the wheelhouse. Considering maintenance, the point was valid although, here again, only very simple maintenance was usually carried out as, due to the light weight and compactness of the gas turbine, repair or overhaul by replacement was an accepted practice.

It was mentioned that for the same financial outlay an operator could choose a multiple Diesel engine arrangement or a gas turbine plant. Under such conditions it was to be remembered that while, with the multi-engine installation, greater reliability was implied, maintenance and weight would probably also be greater. It had to be conceded, however, that greater flexibility in the production arrangement was possible using several Diesel engines.

This situation regarding the factors influencing the choice of prime mover could well be changed with the advent, on a large scale, of the industrial gas turbine, giving a greater range to choose from. This would be particularly so if first costs and fuel consumption were significantly lower than for currently available units.

Finally, could the author say how he would modify, for civil use, the CODAG plant illustrated in Fig. 19?

MR. C. HOOK asked, what the loss of speed of the 150 ton had been due to. Had it been extra weight or faulty calculation or had it been due to some disappointment in the air bleed system, which he understood had been installed on the tail foil? Was it to be expected that the air bleed control increased the drag or not?

Referring to Z-drives he said he carried out a conversion in Haifa, consisting of taking the existing overdrive, cutting it in half and inserting an extension of about 18″, to give a clearance for the 22 ft boat of about a metre over the water. It was an extremely simple Z-drive with two main castings, a top and a bottom casting. There was a boring only in one direction, and a boring at 90° in the other direction in each casting.

There were various diameters. Once the two castings were put on to the machine one could finish all the machine work in one operation.

The outdrive cost roughly £600 to £700 and, in quantity, it could be produced for about £450.

This had been developed from the Cattaneo drive. All the equipment for the small craft had been made of light alloy cast material.

In Miami in 1952 the techniques of the first Z-drive had been implemented by Miami Shipbuilding. It was mounted on a pantograph arrangement but it was not movable on the small craft. First, they had used a Johnson outboard; next they had an interior engine driving out through the shaft and universal joint.

The reason for the pantographs was that everything had to retract in full flight; as one went towards the beach one had to press a button and commence retraction operations. Roughly, $300 000–$400 000 had been spent on it. The only thing it did was to put everybody off the idea of mechanical controls.

He suggested that Mr. Faber, in conclusion, had omitted to mention the question of the hydrostatic drive which was a very good alternative to the bevel drive.

MR. B. TODD (Associate) expressed interest in the materials of the hydrofoil construction. Whilst Mr. Faber had given a lot of useful information on shafting, he had not said anything about materials for foils and struts. The rapid flow of seawater over these parts suggested they should be made of corrosion-resistant materials and, particularly, cavitation-resistant materials.

Could Mr. Faber give any information on the material used in commercial hydrofoils?

In this connexion, he had seen at a recent exhibition, a Russian hydrofoil used in fresh water in which the foils appeared to have been made of stainless steel, or a weld deposit of stainless steel on a carbon steel structure. The Russians had great experience of hydrofoils. They had used these materials in fresh water and it seemed that they too recognized this problem. No details of the steels used had been given.

Although an austenitic stainless steel, such as AISI type 316 would give attractive corrosion and cavitation resistance, the mechanical properties might be too low, therefore a compromise of a strong-base material and a cladding of corrosion resistant alloy would seem attractive from an economic point of view for commercial vessels.

Stainless steels were subject to localized pitting in seawater and this could be very intense and it varied from alloy to alloy. If the corrosion fatigue tests mentioned by Mr. Faber were run under conditions where there was very little probability of pitting one would expect a high value for fatigue strength, whereas in conditions which would produce pitting, such as one might predict in service, then one would predict a much lower result.

A report* gave a figure for 17–4 pH stainless steel heat treated to a similar strength to that given by the author, equal to 16–17 kgf/mm^2 in seawater of 10^7 cycles. This was much nearer the value than the 12 per cent reduction in the air fatigue strength suggested by the author.

If corrosion fatigue properties were a limit on the design of shafts one could make out a case for using material such as alloy 625, a nickel chromium-molybdenum alloy. This had about the same corrosion fatigue strength in air and in seawater. It was extremely resistant to seawater; it had the highest corrosion fatigue strength of any material which had been tested in his company's laboratories. The other would be alloy 500, commonly known as K-Monel; it had a very good history of usage in this type of shafting. Although the mechanical properties were similar to 17–4 pH stainless steel, its corrosion fatigue strength would be considerably higher.

If corrosion-fatigue strength were a limiting factor, then the use of these stronger materials would presumably allow one to use smaller shafts.

The author had mentioned the shafts were designed for 10 000 hours life, i.e., 2½ to 4 years. Were the shafts renewed after this or were they inspected and returned to service? Had any fatigue failures developed in service?

MR. D. H. L. INNS (in a contribution read by Mr. Franklin) said that the paper referred more than once to the involvement of the Classification Societies in hydrofoil construction and operational activities. Although not openly critical, the references, taken in context, indicated that the author felt the Classification Societies generally exerted a retarding influence on hydrofoil development.

Considering firstly the requirements relating to safety of life and prevention of fire hazards, the role of the Classification Societies was mainly to give advice and to administer the requirements. The requirements themselves were rather the outcome of international agreement.

Where hydrofoils competed with conventional displacement craft and provided similar services, it was reasonable that comparable standards of safety should apply to both types. If this philosophy adversely affected the weight-sensitive hydrofoil more than the displacement craft, this represented an inherent disadvantage of the former concept rather than an imposed penalty.

On the other hand, uniform standards of safety for similar services should apply equally for hydrofoils and air cushion vehicles and present trends indicated that this was not so.

Secondly, with reference to the size of propeller shafting, it was stated that the power/revolutions per minute ratio increased rapidly with increased power in typical installations. Presumably this was due to an upper limit on propeller speed, or due to considerations of shaft whirling. Thus, since torque would increase with power, obviously larger shaft diameters would be

* Tschentke, G., August 1962. "Seawater resistant alloy steel castings for shipbuilding." *Konstruktion*, pp. 313–318.

required for acceptable stress levels. Classification Societies did not impose a size factor on shaft over and above this and, therefore, could not be blamed for the increase in shaft diameters. It might be that the author intended to make the point that Classification Societies were over-conservative in that respect. However, since he later described the difficulties in dealing with the complexities of the stressing problems and the variability of the imposed loadings, coupled with determining acceptable fatigue limits for the materials, the Classification Societies' approach could not be considered to be over-cautious.

Turning to the treatment of shaft whirling, could the author give more information? It was not easily determined whether propeller gyro effects had been taken into account or whether they had been found negligible. Also the critical speeds quoted were presumably for first order excitation, once per revolution, and were not at propeller blade order exciting frequency.

Finally, with reference to the CODAG type of installation, illustrated in Fig. 19, it was stated that this arrangement might also be suitable for civil as well as naval applications. This remark appeared surprising. The adoption of such a complex system, involving two different types of propulsion unit, was understandable for naval use where the more economical Diesel unit could be used for prolonged patrolling in the displacement mode. In civil use, however, in the type of service where hydrofoils were likely to be used, the periods of hull-borne manoeuvring would surely be too short to warrant the additional complexity of a separate Diesel unit, and the gas turbine could serve in both modes of operation.

MR. J. W. G. MEYER said that since high lift forces were generated on propeller shafts, and as these forces were associated with pressure reductions, it was possible that cavitation erosion occurred on the shafts. Had Mr. Faber had any experience of this?

MR. S. W. HOBDAY drew the meeting's attention to the fact that carbon fibre had been mentioned. This was something his company had discussed with the RAE, who had raised the point that carbon fibre structures would be electro-positive to about $1\frac{1}{2}$ V with respect to metals. The insulation of the structure would require very careful attention if the metal parts bolted to them were not to disappear under seawater corrosion conditions. In the design of his company's hydrofoil boats this had led them to "shy away" from carbon fibres at the present time.

Correspondence

MR. J. A. COX, in a written contribution, said that his experience with regard to surface piercing foil hydrofoils revealed that one of the main problems which was continually present was the effect of cavitation on the propellers. In view of this it would be appreciated if the author could give his opinion on whether or not:

a) the use of the conventional propeller was inefficient at speeds attained or
b) if, in the not too distant future higher speeds of these craft were envisaged and the use of a water jet was contemplated.

As regards structural strength since it was extremely important in these craft to have a small weight per horsepower ratio, had thought been given to the use of carbon fibre in association with main structural members?

Author's Reply

Mr. Jost, replying to the discussion, on behalf of Mr. Faber, thanked all the contributors. Regarding Miss Kalerghi's contribution, he said that the foil weight or percentage of the foil weight could be assumed to be correct up to approximately 500 tons full loaded. Regarding foils made of carbon fibre, he suggested the author should reply in writing since this was a problem which had been partly investigated. It was not intended to substitute steel by other materials, i.e. carbon fibre within the next two years. Carbon fibres might come within their range during the next five/ten years.

Referring to the principle layout of the PT150 engine arrangement with a torque converter and its advantages, this called for written explanation or preferably should be dealt with in a separate lecture.

The author's company had very carefully investigated the question of water jet propulsion, during the past two years. Although water jet propulsion had great advantages, it also had disadvantages. The water pump proved no difficulties; this could be obtained. Some water pumps were already on the market. The difficult part however was the water inlet, which had to be designed and tested in full scale for each individual craft. The finances which had to be made available for such full scale tests were so high that the company, at the present time, had decided not to be involved with merely own financial resources.

Turning to Diesel engines and gas turbines and their application to hydrofoils, and the skilled personnel and crew necessary to both propulsion systems he referred to the Institute's forthcoming lecture, where consideration was to be given to this problem. He added that written reply would be given on this question.

Fig. 19 showed the CODAG arrangement. Regarding its application to commercial hydrofoil craft and necessary modifications which would have to be carried out a reply would be given in writing.

In replying to Mr. Hook's point on the speed of the PT150, the displacement had been increased by the owners and the Swedish authorities who had to approve the vessel in question. It had been the first hydrofoil of this size which had come up for their approval and, at the same time, the new recommendations from IMCO had been published. This dealt only with hydrofoils up to 200 passengers but his company had decided on a PT150 with a capacity of 250 passengers. They now had to cope with SOLAS with very few exemptions only and the special regulations for hovercraft and hydrofoils up to 200 passenger capacity. Figures could be produced in writing on the safety figures for the PT150.

The foil system which had been applied to the PT150 prototype was of the air stabilized Schertel-Supramar type, and of almost identical configuration of the first air stabilized PT50 test craft, the so-called *Flipper*. The results of the extensive trials, carried out on the south coast of Norway, were unfortunately so delayed that the necessary amendments could not be fully incorporated in the design and construction of the PT150 foils. Due to this and the weight increase, the contractual speed of 39 knots could not be realized. The maximum speed obtained was 37·3 knots and the cruising speed 34·5 knots. It should also be noted that the air-fed stabilization system had its greatest advantages in higher speeds. When operating in smooth water the drag of this system would be slightly higher compared with the conventional Schertel-Sachsenberg system, but operating the same craft in a seaway the average speed would be higher due to the extremely better sea capability.

Due to the low speed obtained, the PT150 had an unfortunate speed range where the stabilization effect was not quite as high as it had been calculated. The next two PT150 under construction at present would have a slightly modified although principally identical foil system as the prototype. The speed would be 361/2 knots. Extensive tank tests with a new foil configuration for the

PT150 would reduce drag to an extent that all future craft of this type would have a speed of approximately 39 knots. This new configuration would also give full advantage to the air-fed system.

As regards the Z-drives a reply would be given in writing.

To Mr. Todd the author replied that for the conventional foils they had so far used MSt 52–3 steel (German standard) with an ultimate strength equal to 52 kg/mm^2 and yield point equal to 36 kg/mm^2. They had applied stainless steel for foils to a number of boats for special purposes and the specifications could be given in writing. Normally they used a steel having an ultimate strength equal to 52 kg/mm^2.

For the PT150 they had applied another steel, a heat-treated low alloy higher in ultimate strength, with a yield point equal to 50 kg/mm^2. This enabled them to obtain the same percentage of foil weight as with previous craft. All steels mentioned had good weldable properties and a repair job could be undertaken at any yard in the world.

The prime task they faced was to ensure that the boats could be maintained everywhere not using special materials which were not available on the spot and which imposed repairs to be carried out under special conditions.

They had not experienced pitting on the shafts even after 10 000 operating hours. The shafts having a total running time of 10 000 hours normally would be replaced by new shafts. At 10 000 hour range the shafts were sometimes damaged on the surface and had to be replaced to meet the Classifications Societies' regulations. Inspection had been carried out a number of times during the 10 000 hour operation and after that period it was considered that a new shaft had to be installed.

Regarding the special alloys there was firstly the question of price, secondly that of delivery times and thirdly the approval by the Classification Society. The design had to meet the demand of various Classification Societies as it was not known which Classification Society had to approve the design and the material used for the actual construction purposes. Only in very small cases special designs could be made for special ships.

To Mr. Inns a reply would be given in writing.

To Mr. Meyer the author said that they had never experienced cavitation on the shafts.

Written Reply

To Miss Kalerghi the author replied that the quoted percentage relative to the weights of the design section, especially for the foil sections did mainly meet the types of vessels designed by his company. It could, however, be assumed that these figures had a general validity if the specialist found that the design characteristic of another design allowed a comparison of the percentage weights or whether important deviations in the design were recognizable thus prohibiting the use of the percentage dealt with.

Further on it could be confirmed that the stated percentage weights did not only refer to the existing vessels but also originated from smaller or bigger projects than the PT150.

The author thought that the effect of the air stabilization system on the weight of the foils was so slight that it fell into the scatter range of accuracy of the project data and therefore could be neglected.

In the present paper no reference had been made to any special type of vessel, this also included the PT150 and its characteristics. The author had meant to give a general summary and a retrospective view of the fields which, in his opinion, had not been covered extensively by technical literature. If the Institute were interested in an extended lecture to discuss the questions and the problems of propulsion with torque converter, its advantages and disadvantages compared with variable pitch propeller, his company would welcome this opportunity. Therefore, it would be advisable to wait for additional operation experience and the relative evaluations in respect of the two vessels at present under construction. The same applied to the subject of water jet propulsion, which was not the subject of this lecture.

On principle it seemed to be correct that for the crew there was no difference between running gas turbines and Diesel engines at sea. But this could be questioned when, in such a comparison, idealized conditions were put too much in the foreground. Neither the Diesel engine nor the gas turbine could be expected to run without any servicing for small mechanical deficiencies between the scheduled times for overhaul. Up to now the author had not seen any published data which could prove that a gas turbine with a power range of about 4000 hp had been running under absolute identical operating conditions and had achieved similar running hours as generally expected by a Diesel engine today. It had to be stressed that by identical operating conditions was meant the use of Diesel fuel and the servicing of the engines without specialist from the turbine manufacturers on board and/or ashore. Therefore, the shipowner had to employ personnel capable to handle a device originating from the aircraft industry under nautical conditions. It might be assumed that the handling of gas turbines was an integral part of the education programme of the marine engineer and consequently the ship's manning regulation considered the installed H.P. and not the type of internal combustion engine used. However, due to the problems occurring in actual practice, with both the servicing and the maintenance of high speed light weight Diesel engines, the author appreciated the hesitation of experienced hydrofoil operators to use lightweight gas turbines. This might change in future but being already a pioneer in a new means of transport, i.e. hydrofoil, he would not like to pioneer a new propulsion unit too. This point of view seemed very conservative but the manufacturer was not the user and had to meet the demands of the existing market which did not seem ready yet for the use of gas turbines instead of Diesel engines on passengers hydrofoils.

Without contradicting this opinion the author thought, regarding the modification of the CODAG plant shown in Fig. 19 that there was a power limit factor beyond which the discussion of the use of Diesel engines or gas turbines was useless, i.e. engine plant with power requirements for which numerous Diesel engines became necessary. For such a craft the choice of the gas turbine would be the only solution, as it offered an essentially higher concentration of power than the multiple Diesel engine arrangements, due to the output limit, now 5000 to 7000 hp per unit. Considering the project in question it would be advantageous for price reasons to prefer a single gas turbine, as the main propulsion unit for both main propellers. But it would not be suitable for alternative operation of the auxiliary propulsion. Auxiliary propulsion was considered absolutely necessary for hull borne operation which, contrary to the general opinion, took a higher percentage of the total operating time than normally expected, when considering estuary trading and operation in extreme sea states which did not allow the use of main propulsion engines due to engine noise in harbour and exhaust smoke when running the engine on low power. A second propulsion engine for estuary trading and as a reserve at sea for hull borne operation was not new. The necessary power output for this second propulsion unit depended on the intended condition in estuary trading and at sea. In the past, the separate hullborne propulsion units had been underpowered. However, if a sufficiently rated hullborne drive was installed, it was advisable to connect this plant alternatively to the main propulsion plant thus reducing the weight/power ratio of the whole engine plant. For price reasons a Diesel engine should be used as the second propulsion unit. Consequently this would mean a CODAG plant. The additional arrangements on the gearing were relatively small in relation to the whole plant.

The author thought that this explanation would also answer a question raised by Mr. Inns.

In reply to Mr. Hook, it had been extremely interesting to see how simply the conception of the presented Z-drives had been solved. It might not be ignored, however, that the price for one propulsion unit with bevel gears was about four times the price of that with conventional inclined propeller shaft. Therefore, propulsion units with bevel gears had been applied, until today, mostly to navy vessels only. The author thought that the hydrostatic propulsion should be given more attention but here too there seemed to exist limits. These limits made it necessary to instal several units under water acting on one propeller. This would lead to the use of gears with all resulting problems which might also reduce the advantage in price compared with the bevel gears.

To Mr. Todd the author replied that the materials used were:

a) foils: sheet MSt 52–3 DIN 17 100

or alternatively in stainless steel on request: X 4 CrNiMoNb 25·7 W.No. 1·4582

b) propeller shaft X 22 CrNi 17 W.No. 1·4057.

The author would be interested to see duplicates of the slides shown by Mr. Todd as a number of questions might arise which could lead to a direct fruitful correspondence.

To Mr. Inns the author replied that the rules of the Classification Societies as they were set up today would become conservative if not out-dated tomorrow owing to the nature of the subject. It depended on the subject in question, how strongly it was felt or had been felt for these rules to be conservative.

Mr. Inns had just hinted at the problem of unified safety regulations for hydrofoils and hovercraft and the author could only endorse these remarks. If Mr. Inns had gathered, from the author's explanations, that there existed different view points between the Classification Societies and the designers with regard to their judgement of a new design, he could say that these differences were mainly related to what was the "acceptable stress level", to use Mr. Inn's terminology.

As an example, taking the dimensions of the propeller shaft the author didn't think that the known formula dimensioning the diameter took into account the actual loadings. If the dimensions obtained from this formula coincided with the dimensions calculated when applying the actual loads this was only mere chance, since there might be design conditions where this would not occur. A formula which merely considered one of the various load components and compensated all other possible components by setting the acceptable stress level to comparatively low figures would have its application limits and could certainly not be valid when dimensioning propeller shaft for hydrofoils.

On the question of how to calculate the natural frequency of the propeller shaft it might be said that this was regarded as a through going beam with several supports. Each support had its specific inclination and displacement stiffness which in most cases was different in the two main axes. The gear shaft was not included in the calculations. Moreover, comparison calculations had shown that it was sufficient to consider only the stiffness of the clamp on the gear output flange. Studies of the gyro effect of the propeller had not been made yet. The third order excited from the propeller blades had not been checked closely since the the vibration measurements in hand did not give any reason to take up this problem.

In replying to Mr. Cox, the use of carbon fibre had already been suggested and the author was grateful for the interesting hints especially from Mr. Hobday which showed that precautions were needed when introducing new materials.

Propeller cavitation was a problem but solutions had been found, by a systematic full scale testing, that produced operating hours acceptable by shipowners. This method of achieving optional propeller design regarding cavitation had to be re-elaborated for each type of vessel, since for propellers on inclined axis this problem, theoretically and experimentally, was only partly solved.

The use of water-jet propulsion for higher speeds as an alternative to the propeller was uncontested although the development expenses involved were quite important. Therefore, on a commercial basis propellers were to be preferred for hydrofoils as long as this was justifiable.