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THE ANALYSIS OF CONTROLLABLE PITCH PROPELLER CHARACTERISTICS AT OFF-DESIGN CONDITIONS

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Controllable pitch propellers present the propulsion engineer with special problems relating to both the calculation of the blade loadings at off-design operating conditions, and also with the integration of the c.p. propeller within the overall ship propulsion system. This paper endeavours to assist in the solution of these problems by considering the implications of both the practical and theoretical aspects of ship propulsion using controllable pitch propellers.

Initially, attention is focused on a series of cavitation tunnel experimental studies using model c.p. propellers working in both uniform and variable wake flows, and at different cavitation conditions. Consideration is then given to the geometry of the blade sections and in particular to the distortion of the blade at off-design pitch settings. The effects of both the section distortions and the various seagoing operational modes of a c.p. propeller on the calculation of the hydrodynamic and inertial blade loadings are discussed. A correlation of these calculated loadings with the experimental results is made and the effects of transient loadings are also mentioned.

The practical constraints imposed upon the design of the controllable pitch propeller blades are investigated and theoretical results are used to demonstrate the effects of changing some of the blade shape parameters such as skew, rake and surface area upon the required twisting moments. An estimation diagram for the calculation of the maximum non-frictional spindle torque is given for c.p. propellers of conventional blade shape and the effects on these loadings of placing a duct around the propeller are also mentioned. Finally, some discussion is given to the methods of estimating the propeller loadings encountered during manoeuvring conditions and also to the problems associated with constant shaft speed operation.

INTRODUCTION

Traditionally propeller design and analysis has for the most part been confined to conditions at or close to the prescribed operating conditions of the ship. Consequently, the majority of the propeller evaluation methods have been based upon the idea of a moderately loaded propeller and relatively few methods have attempted to predict the characteristics at off-design conditions.

The dilemma that has faced the designer of a controllable pitch propeller mechanism has been that of establishing a reasonable calculation procedure to enable him to predict the off-design loadings to which his design is likely to be subjected. In the past the estimation of these loadings has been achieved largely from the use of empirical methods, however, with the increasing use of c.p. propellers on high Power installations it was felt that a more rigorous basis for these calculations should be sought.

Fig. 1 shows the general arrangement of the forces and moments acting on the sections of a propeller blade. In tabular form the most important component of these loadings, as shown in Fig. 1 are as follows:

i) a radial distribution of thrust acting normal to the XY plane dependent upon the pitch and advance conditions and their respective rates of change;

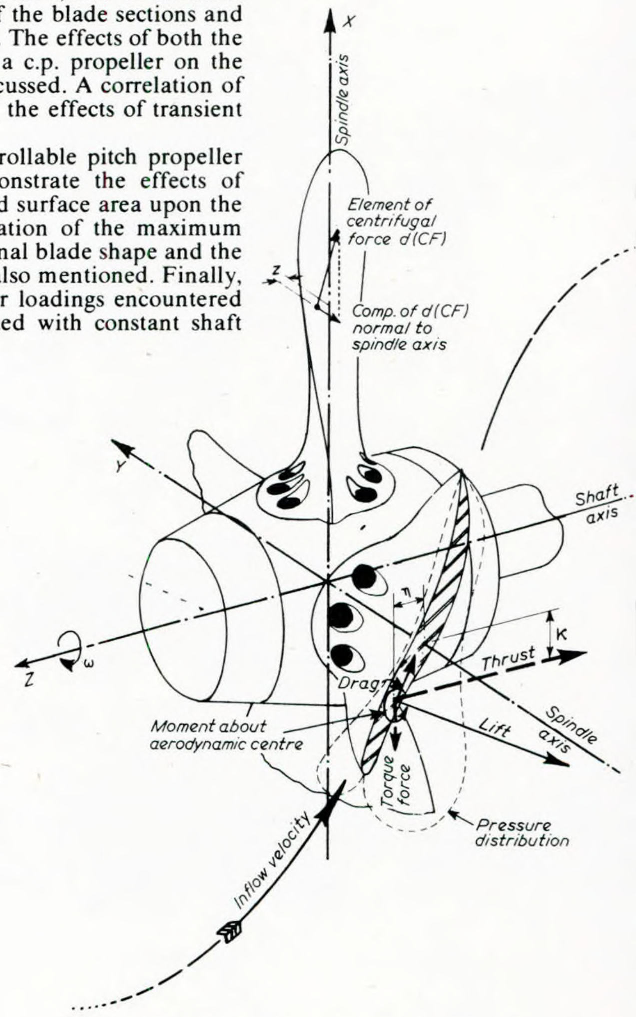

FIG. 1—*General arrangement of forces acting on a c.p.p. blade*

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ii) a radial distribution of torque acting normal to the YZ plane, also dependent upon the pitch and advance conditions and their respective rates of change;

iii) an inertial spindle torque in the ZY plane due to the centrifugal, coriolis and other transient accelerations about the shaft and spindle axes OZ and OX respectively;

iv) a hydrodynamic spindle torque in the XZ plane due to the pressure field around the propeller blade for a particular pitch setting, advance coefficient and the respective rates of change of pitch and advance;

v) an inertial force acting normal to the YZ plane due to the steady state and transient centrifugal effects.

This study endeavours to focus attention on each of the types of loading delineated above by discussing methods of calculation and also the relative significance of the loading components. At the outset of this work it was felt important that the accuracy of these methods should be verified, therefore, concurrently with the theoretical approach a representative range of model controllable pitch propellers were manufactured and tested in the cavitation tunnel of the University of Newcastle upon Tyne. The results of these tests have been used throughout this work as a basis for comparison.

Although the primary aim of this work was to examine the off-design blade loading problem, a final section dealing with the applications of this study to practical problems has also been included. In this latter section some of the difficulties, and the way in which these difficulties have been overcome, of determining the manoeuvring forces, spindle torque, blade design and the influence of ducts on the loading of the propeller blades are discussed in relation to practical ship propulsion problems.

With regard to the frictional spindle torque, in addition to being dependent upon both the hydrodynamic and inertial loadings transmitted from the propeller blades, it is also fundamentally influenced by the mechanical arrangements of the hub. Consequently, since the prime consideration here is of the design and analysis of c.p. propeller blades, the frictional component must be regarded as outside the scope of this work, due to its dependence on the physical aspects of the hub.

EXPERIMENTAL STUDIES

Model Propeller Experiments

Measurements of the forces and twisting moments acting on the blades of a controllable pitch propeller have been obtained using a dynamometer which was designed and manufactured in the Naval Architecture and Shipbuilding Department of the University of Newcastle upon Tyne.

The dynamometer comprises of a steel strain gauge beam of cruciform section which is located diametrically inside the propeller boss such that two flat surfaces of the beam are parallel and two normal to the shaft axis, see Fig. 2. The base of the strain gauge beam is rigidly secured to the propeller boss whilst one blade is locked to the opposite end of the beam by means of a collet; this arrangement having been incorporated to allow for the adjustment of the blade pitch angle. A clearance has been given around the blade flange and is designed to permit the attainment of the maximum deflexion under load. The remaining blades of the propeller are locked in recesses around the boss at their appropriate pitch angles.

The measurements of the blade loadings are realized *via* twenty-four electrical foil type strain gauges which have been bonded to the beam at three sections in its length as indicated by Fig. 2. The cable leads from the gauges are taken along the inside of the model propeller shaft to a set of silver sliprings, and from there on to form a system of bridges and half bridges from which the following set of force and moment measurements can be taken:

a) thrust;
b) torque;
c) axial bending moment;
d) torque force;
e) blade twisting moment.

The calibration of the strain gauge bridges was accomplished by applying individual known forces and moments to an arm fixed to the strain gauge beam so as to simulate the various types of blade loading. Care at this stage was taken to ensure that only one type of load or moment was applied at any time to the beam. During the calibration it was found that interaction effects between the bending moments took place, these effects being due to slight inaccuracies incurred during the manufacture of the beam coupled with errors in the positioning of the strain gauges.

The centrifugal forces acting on the blade elements cause a twisting moment about the spindle axis. Consequently measurements were taken with the propellers operating in air over the range of revolutions intended for the water tests so that the centrifugal effects could be isolated for each pitch setting. The experimental results derived from these tests were recorded and at the same time the interference values combined with the thrust and torque bending moments were also noted, these latter quantities being relatively small due to the density of air. The twisting moments thus obtained were corrected for interference effects by a computer based iterative procedure, consequently, the resulting twisting moments obtained from the air test results are considered to be a fairly accurate measurement of the centrifugal spindle torque. The centrifugal twisting moments were subsequently deducted from the water test results, which were similarly

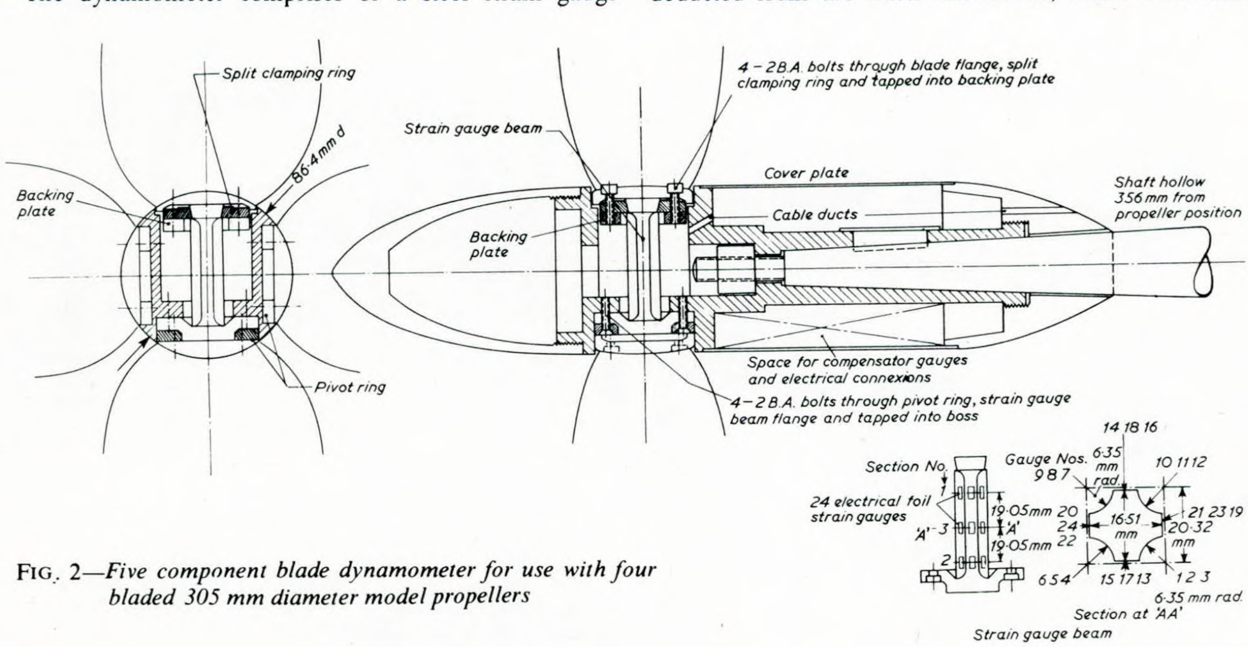


FIG. 2—*Five component blade dynamometer for use with four bladed 305 mm diameter model propellers*

corrected for interference effects, to give the net hydrodynamic results.

During each cavitation tunnel test the values of the thrust and torque were also measured by the standard shaft dynamometer for the purpose of comparison. In the analysis of the test results the value of the advance coefficient has been corrected for the tunnel wall interference using the Wood-Harris method to determine the equivalent free stream velocity.

Sign Convention

The following sign conventions have been adopted for the description of the propeller forces, moments and direction of advance:

1) thrust — a positive value indicates an ahead thrust;
2) propeller torque — a positive value indicates the condition where the propeller is being driven by the engine, while the negative sense indicates the propeller is acting as a turbine;
3) spindle torque — a positive moment is one such that if the blade of a right handed propeller is viewed from above it would tend to rotate in a clockwise direction;
4) advance — a positive value indicates forward advance.

Description of Model Propellers and Test Conditions

The experiments were conducted in the University of Newcastle upon Tyne Cavitation Tunnel, with 305 mm diameter, four-bladed model propellers as listed in Table I.

TABLE I

Model No.	Design P/D	B.A.R.	Comments
117	0·794	0·472	
118	0·797	0·479	
119	0·982	0·529	
120	1·063	0·665	
121	0·661	0·463	
143	0·961	0·782	Run with and
144	0·964	0·650	without duct

Boss diameter ratio $r_h/R = 0{\cdot}283$ for all model propellers.

In all cases the blades were initially set at the design pitch before fitting the propeller into the tunnel, subsequent changes in blade angle being carried out *in situ* by means of a jig at 0·7 radius.

With each propeller seven blade pitch settings were examined, namely at design pitch, +3, −5, −10, −20, −30 and −40 degrees.

The model propellers were tested in a uniform stream with the cavitation tunnel open to the atmosphere. At pitch settings at or near the design setting the advance coefficients were restricted to the forward direction only, in this case the tunnel stream velocity was maintained constant and the slip varied by changes in shaft revolutions. At certain negative pitch settings the tests were conducted with constant shaft revolutions and the stream velocity varied from forward to astern advance.

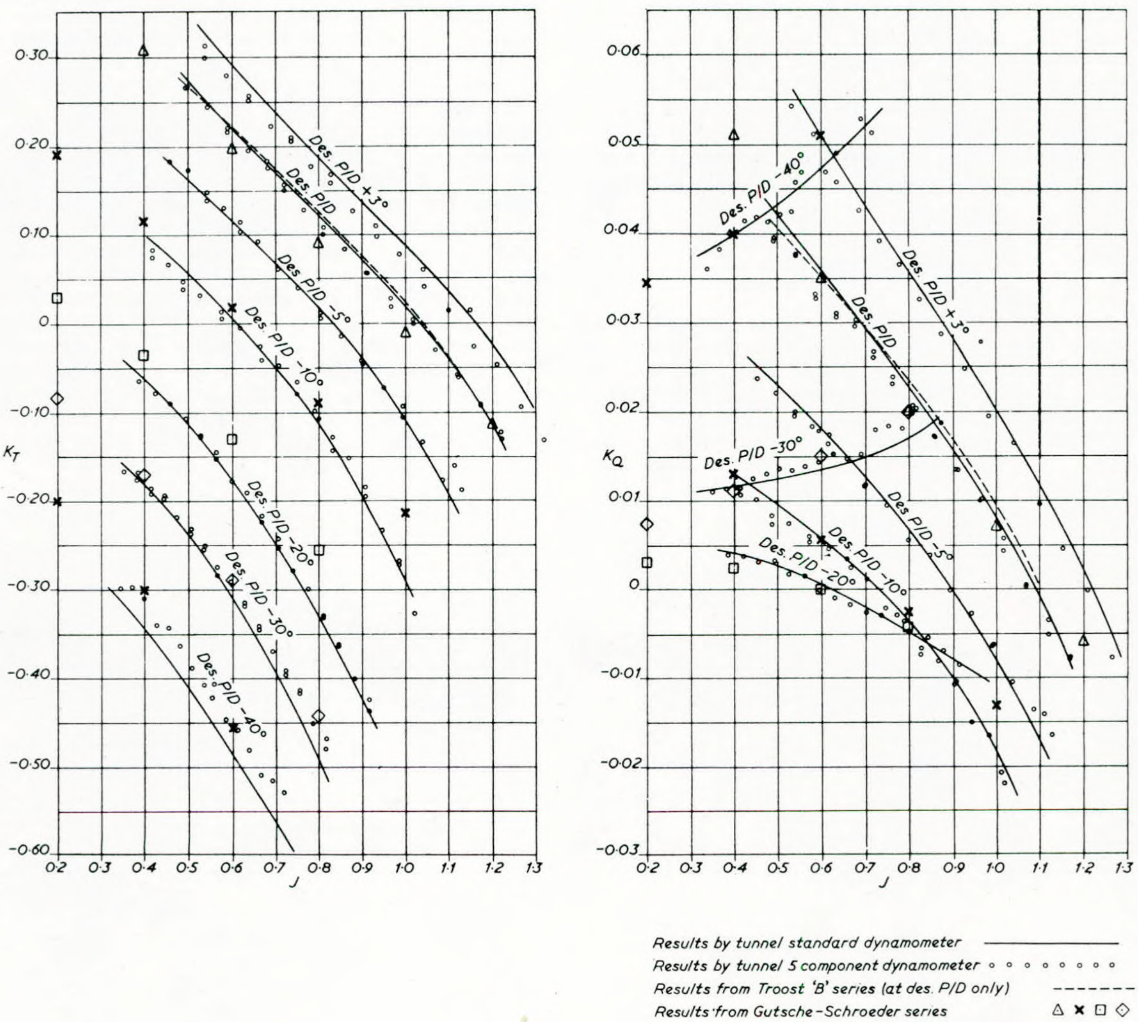


FIG. 3—*C.P.P. model no. 119 — Comparison with other empirical data*

Accuracy of Results and Correlation with Existing Data

Because of the relatively unique nature of the experiments it was considered desirable to examine the results for compatibility with existing data. Since the majority of the available standard series data is applicable to fixed pitch propellers with the exception of the work of Gutsche-Schroeder and also Yazaki, the more detailed correlation has been confined to the design pitch setting. At this condition the thrust and torque coefficients obtained from the cavitation tunnel experiments have been correlated with values derived from the Troost B-4 series B_p–δ diagrams. The values lifted from the Troost charts have been corrected for variations in power absorption and pitch ratio from the basic propeller using the procedure proposed by Burrill[1]. For the off-design settings an orthogonal analysis technique based upon the Gutsche-Schroeder series[2] was used for the comparison. A typical set of results for c.p. propeller model no. 119 are compared with the cavitation tunnel experimental values in Fig. 3.

The five component dynamometer measures the forces on one blade only, so that the total thrust and torque has been obtained by multiplying the measured result by four. Consequently, experimental errors due to inaccuracy in blade pitch settings, are enlarged by this factor. Despite this fact, a reasonable agreement is seen to exist between the tunnel results and the results from the equivalent Troost and Gutsche-Schroeder propellers.

Influence of Cavitation on Blade Loading

The tests described in the section dealing with the description of model propellers and test conditions were conducted in the cavitation tunnel at atmospheric pressure, consequently giving high values of cavitation number which resulted in the suppression of most cavitation. To supplement the earlier results, additional tests were carried out on c.p. propeller no. 120 at the following conditions:

i) With the blades at design pitch + 0·42 degrees setting and working in a uniform stream (forward advance only), varying propeller revolutions and reduced static pressures, such that cavitation numbers σ = 13, 7, 5, 4 and 3 were obtained.

ii) With the blades at design pitch −30·87 degrees, the propeller turning with constant revolutions and the water speed in the tunnel varying from ahead to astern advance. These tests were carried out by setting an initial value of tunnel static pressure to give cavitation numbers of σ = 65, 20 and 10 respectively and allowing the static pressure in the tunnel to automatically change with the variation in stream velocity, so that as $V_a \rightarrow 0$ then $\sigma \rightarrow \infty$.

Curves of K_T, K_Q and hydrodynamic K_{QSH} plotted to a base of J, for the tests i) and ii) are shown in Fig. 4. The results from tests i) show the usual effects of cavitation on the values of thrust and torque coefficients. At high slip conditions the development of cavitation is seen to cause the absolute value of the twisting moment coefficient K_{QSH} to reduce from that obtained at σ = 13·0.

When working at negative and low slip conditions, the propeller blades experience an increase in twisting moment as face cavitation is developed, i.e. reducing cavitation number, without any change in the value of thrust or torque. This implies a shift in the position of the centre of pressure due to face cavitation distorting the non-cavitating pressure field around the blade sections.

The results from tests ii) conducted with the same propeller in an astern pitch setting show that the thrust is negative for the major part of the test range and is unaffected by the change in cavitation number considered. In this pitch setting, the blades were almost normal to the tunnel stream velocity, consequently severe eddy shedding developed in the flow from the propeller which resulted in forces and moments of an unsteady nature.

The value of the torque coefficient increased with reducing cavitation number, over the complete range of the tests. This can be attributed to a change in the effective camber, caused by the growth of leading edge face cavitation, resulting in an increase in the section drag. In the range of the test conditions, the hydrodynamic pitch angles were small and hence the drag force constituted the major part of the total tangential force on the blade. A small increase in drag will therefore produce a significant change in propeller torque.

In the forward advance range of these tests the twisting moment increases with the suppression of cavitation, while in the astern running conditions the values of K_{QSH} at the various cavitation numbers converge to ultimately attain a common value.

Influence of a Wake Field on Blade Loading

It was realized that considerably higher local values of thrust, torque and spindle torque would be experienced with the propeller working in a variable wake stream. Tests were therefore conducted with the blades at design pitch and the propeller working in an arbitrary variable wake stream of w_t = 0·46, at four different shaft speeds, i.e. rev/s = 7·69, 10·00, 13·33 and 20·00 respectively.

For these experiments it was necessary to ensure the highest possible response from the five component dynamometer, so that the variation of forces and moments which occur during each revolution could be measured. The con-

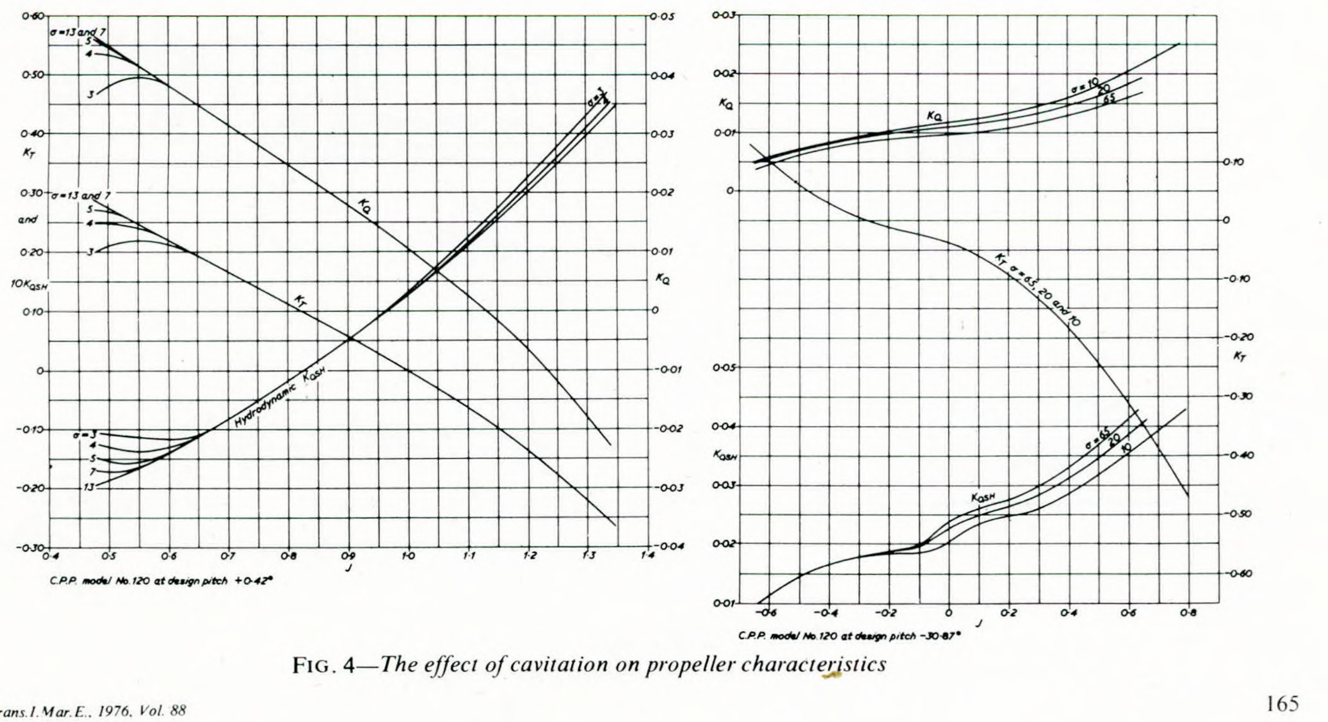


FIG. 4—*The effect of cavitation on propeller characteristics*

densers in the bridge circuit for "noise" suppression were removed and results obtained first with the propeller working in a uniform stream, as a basis for comparison with the variable stream results. At each of the four settings of propeller revolutions, the tunnel stream velocity was adjusted to give a thrust identity with the uniform flow results.

The circumferential distribution of velocities in the variable wake stream are given in Fig. 5 and curves of

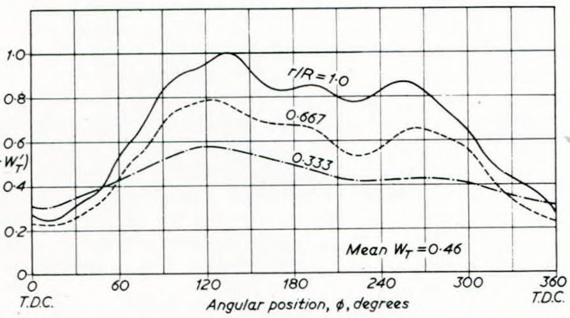


FIG. 5—*Circumferential distribution of* (1 − W′T)

circumferential variation of K_T, K_Q and hydrodynamic K_{QS} for mean J values of 0·485, 0·74, 1·00 and 1·31 shown in Fig. 6.

Considering the case of mean J = 0·74, which is close to the service operating condition for this particular screw, it is seen that the spindle torque values vary considerably from that of the mean value obtained in the uniform stream tests. The maximum value of the hydrodynamic twisting moment is developed when the propeller thrust and torque are a maximum at the position 40–50 degrees from T.D.C. The wake distribution given in Fig. 5 shows that the minimum local velocity occurs at φ = 10–15 degrees, there is therefore a phase shift in the development of the maximum lift coefficient from the position of the minimum local velocity.

C.P.P. BLADE GEOMETRY

Section Deformation at Off-Design Conditions

If the blade of a controllable pitch propeller is rotated about its spindle axis through some angle $\Delta\theta$ from its design condition, then it is found that helical sections at any given radii are subjected to a distortion when compared to the original design profile.

To illustrate this point further, consider a blade in the designed pitch setting together with a section denoted by a projection of the arc ABC at some given radius r, (Fig. 7).

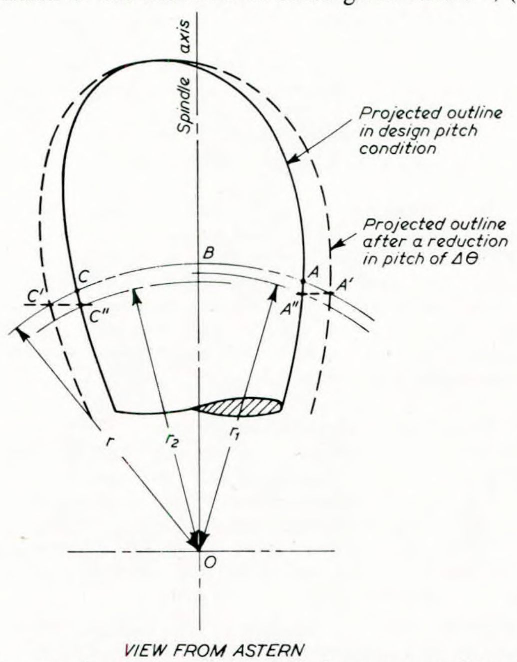
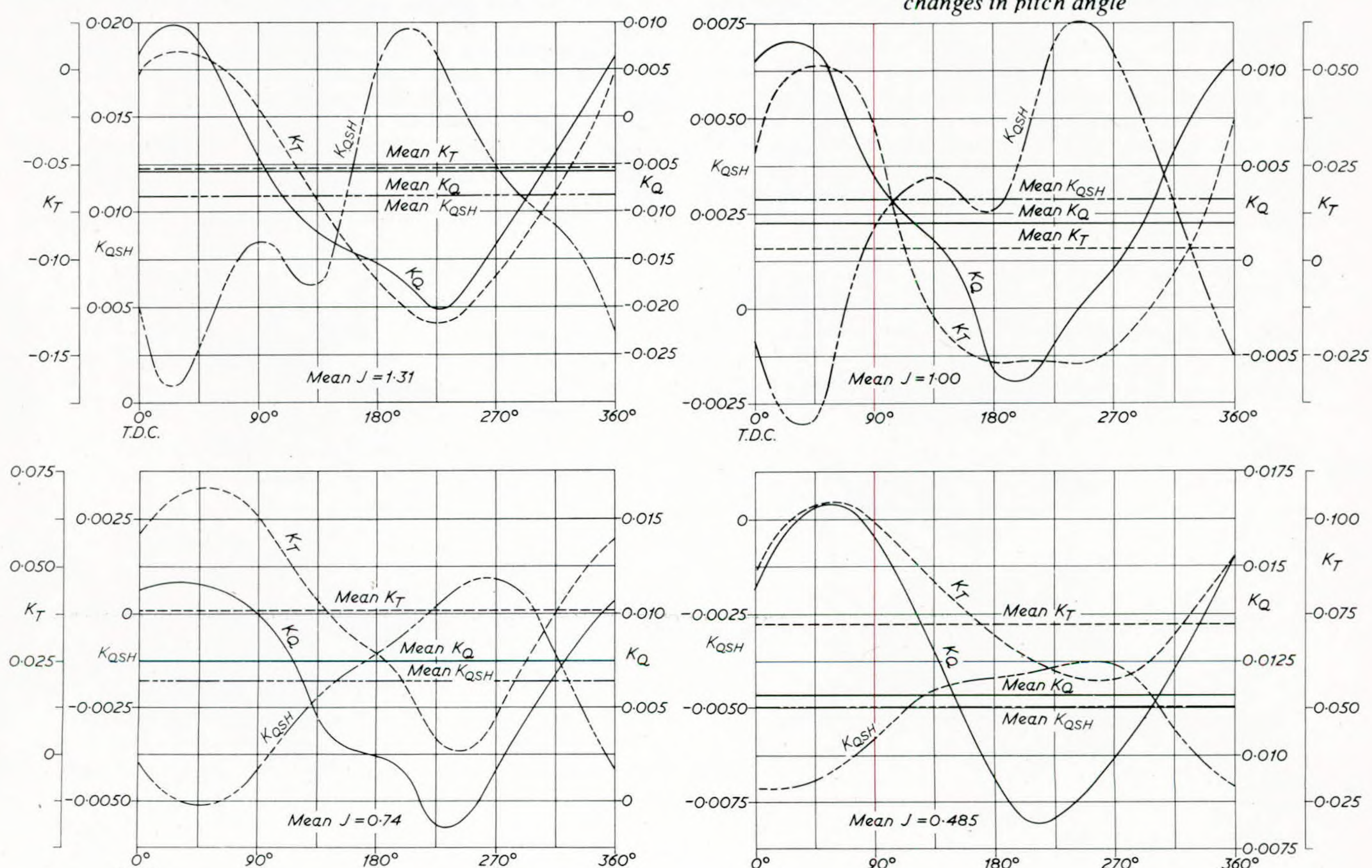


FIG. 7—*Geometric effects on blade section resulting from changes in pitch angle*



Mean values of K_T, K_Q and K_{QS} are obtained from 'open' water test results

FIG. 6—*C.P.P. model no. 120 at design pitch — Tests in a variable wake stream* W_T = 0·46

When the blade is rotated about its spindle axis, through an angle $\Delta\theta$, such that the new pitch angle attained is less than the designed angle then the blade will take up a position illustrated by the hatched line in the diagram. Therefore, at the particular radius r chosen the above helical section is now to be found as a projection of the arc $A'BC'$. However, the point A' has been derived from the point A'' which with the blade in the design setting was at a radius r_1, ($r_1 < r$). Similarly with the point C', since this originated from the point C'' at a radius r_2, ($r_2 < r$). Consequently the helical section $A'BC'$ at radius r becomes a composite section containing elements of all the original design sections at radii within the range r to r_2 assuming $r_1 > r_2$. These distortions are further accentuated by the radially varying pitch angle distribution of the blade causing an effective twisting of the leading and trailing edges of the section. A similar argument applies to the case when the pitch angle is increased from that of the design value. This latter case, however, being of a fairly trivial nature from the section definition viewpoint since the pitch changes in this direction are seldom in excess of four or five degrees.

It is, therefore, to this problem of evaluating the induced deformations of the helical sections that attention has now been focused since in any study of the action of a c.p. propeller it is of fundamental importance to appreciate the effect of these variations.

Although the solution of the induced deformation problem has been considered using three quite distinct approaches, the first being an approximation proposed by Rusetskiy *et al.* (3), the second being a geometric drawing technique developed for this purpose and finally a numerical analysis method for use on a high speed digital computer, the latter approach has been adopted in this present work.

Numerical Analysis Method

This, the third and final approach to the problem of "off-design" section definition, has been developed in order to provide both a more exact and also faster solution of the distortion effects. However, due to the extensive amounts of numerical manipulation inherent in this type of method, it is suitable only for use in conjunction with modern high speed digital computers.

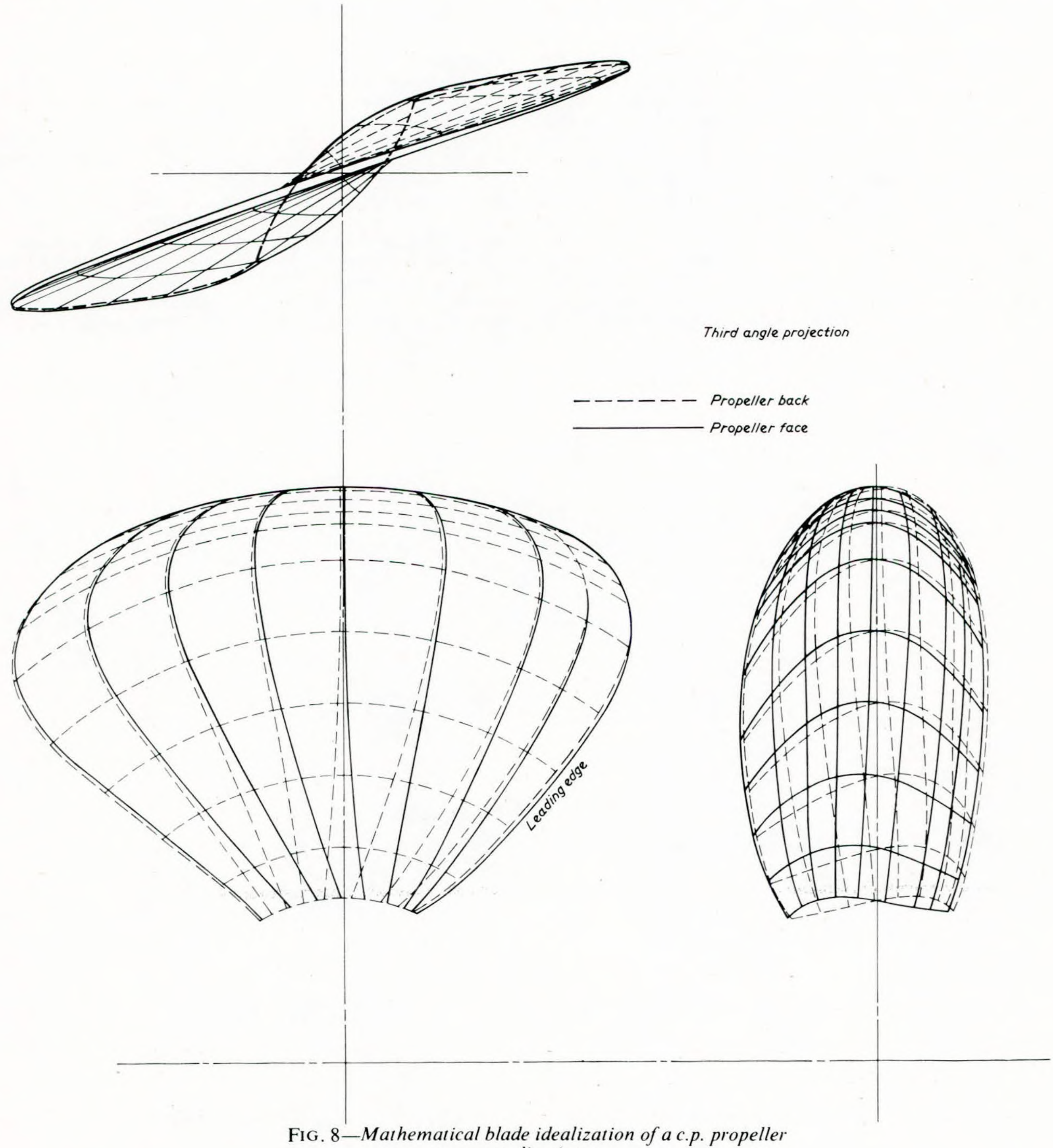


FIG. 8—*Mathematical blade idealization of a c.p. propeller application*

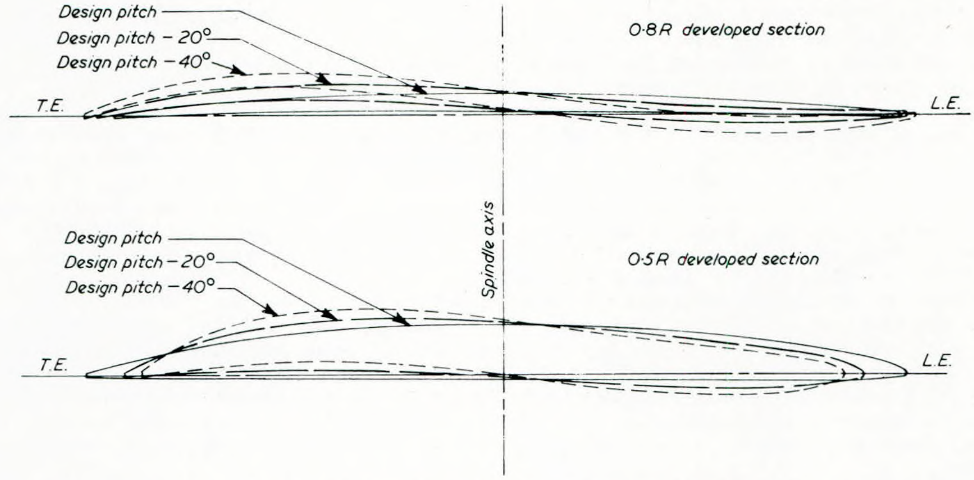


FIG. 9—*Section distortion due to changes of pitch angle*

The method employed with this solution is to define the back and face surfaces of the propeller blade in a mathematical form about a set of three mutually perpendicular cartesian axes. The actual surface definition is obtained by fitting to the surface a numerically defined patchwork based upon a set of third order polynomials such that their partial derivatives and cross curvatures, in both the chordal and radial directions are continuous over the surface of the blade. A typical example of a propeller idealization resulting from this form of analysis is shown in Fig. 8.

Having satisfactorily established a numerical analogue of the actual blade in the designed pitch condition it then remains to determine the form of the helical sections at the appropriate radii resulting from a change in pitch angle. This is achieved by fixing the blade in its design pitch setting relative to the co-ordinate system and intersecting it with a "cutting" cylinder of the appropriate radius, which is constrained to rotate about a diameter coincident with the spindle axis, through the required pitch change angles. The chordal spacing of the intersection points between the blade and cylinder is a variable, depending in the first instance on the accuracy of definition required and secondly on the relative position of the point of the section. For example, near the leading and trailing edges the degree of definition is considerably increased from those of the more mid-chord stations. The resulting section ordinates then have to be converted using a compound transformation technique from their three dimensional system to one comprising only two

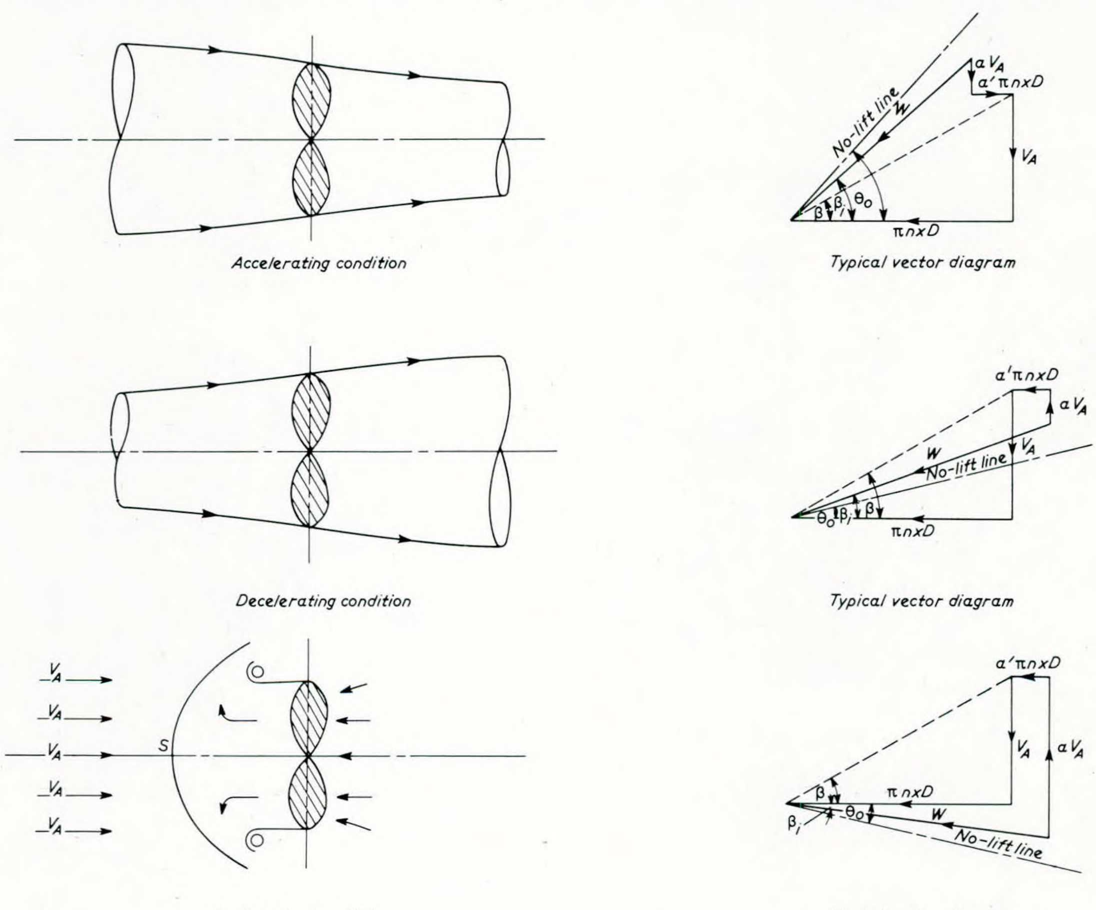


FIG. 10—*Idealized flow configurations*

dimensions thereby giving normal representations of the pitch and geometric characteristics at the desired "off-design" condition.

Fig. 9 illustrates a typical example of the distortion to which the sections at 0·8R and 0·5R of a fairly wide bladed propeller might be subjected when undergoing a reduction in pitch angle of up to 40 degrees.

CONTROLLABLE PITCH PROPELLER HYDRODYNAMICS

Modes of Operation

When working at off-design conditions the controllable pitch propeller induces flow conditions of many different types, the analysis of which poses an exacting challenge.

For the purposes of discussion the various operating conditions may be grouped into two basic types, the first of these being where the flow of water is passing through the propeller disc in the same direction as that of the relative advance velocity. In this mode of operation the slip-stream can be either undergoing an expansion or contraction and such conditions as these would typically occur during a mild braking or accelerating manoeuvre. The second of the two basic flow types is characterized by the induction of water back through the propeller, in the opposite direction to the oncoming advance velocity, so as to form a stagnation point ahead of the propeller, whereupon the water then passes back over the blade tips. Conditions such as these would result from quite severe manoeuvring operations, for example, an emergency full astern pitch change from full ahead. The principal features of the above flow configurations are outlined in Fig. 10, which despite an over-simplification demonstrates the general nature of the flow.

Naturally there are several other not so clearly defined modes of operation in which the resultant flow comprises a mixture of the types described above, this being especially true of the transient manoeuvres. These mixed mode conditions are considerably affected by the effects of cavitation, viscosity and flow separation and, consequently, impose severe limitations on the inviscid mathematical modes discussed in this paper. Consequently, it is proposed to leave these conditions for discussion at some future date and regard them as outside the scope of this present work.

Aerodynamic Properties of the Blade Sections

The theoretical zero lift angle is primarily a function of the centre-line camber distribution of the aerofoil, although in practice there are found to be secondary effects resulting from the section thickness distribution and cascade conditions.

Previously the calculation of the theoretical no-lift angle had been carried out using the numerical integration procedure proposed by Burrill[4], however, in order to deal successfully with the distorted sections shown in Fig. 9 it was felt that a revision of Burrill's method should be undertaken so as to make provision for the high trailing edge curvatures of the section camber lines. Consequently, the following numerical relationship was developed by employing a second order approximation to the camber line near the trailing edge of the section.

$$\alpha_0\,(\text{Theoretical}) = \sum_{k=1}^{19} (M_k . y_{ck})\ (\text{degrees})$$

the coefficients m_k are given in Table II below:

TABLE II

Theoretical Zero Lift Angle Multipliers

k	*Fractional chord from L.E.*	M_k
1	0·05	5·04
2	0·10	3·38
3	0·15	3·00
4	0·20	2·85
5	0·25	2·81
6	0·30	2·84
7	0·35	2·94
8	0·40	3·10
9	0·45	3·33
10	0·50	3·65
11	0·55	4·07
12	0·60	4·65
13	0·65	5·46
14	0·70	6·63
15	0·75	8·43
16	0·80	11·40
17	0·85	17·02
18	0·90	−22·82
19	0·95	310·72

This method has been applied to a range of mathematically defined camber lines ranging systematically from a parabolic form to a symmetrical "*S*" shape and the results have been found to agree to within 0·5 per cent when compared to those derived from thin airfoil theory for all forms within the range. However, the results derived from the earlier work of Burrill, whilst being in reasonable agreement for the more conventional sections, show a deviation of about 8 per cent from the zero lift angle obtained from thin aerofoil theory for the "*S*" shape type sections.

An alternative solution for the theoretical zero lift angle can be obtained from the work of Riegels[5] and Truckenbrodt[6]. This method is based upon the representation of the camber line by a Fourier cosine series,

$$viz. \qquad y_c = 2\sum_{k=1}^{\infty} a_k \cos(k\varepsilon)$$

However, if for practical calculations the infinite series is replaced by the following finite series where the camber ordinates are situated at fixed chordal stations defined by

$$x_{cm} = \left[1 + \cos\frac{\pi m}{N}\right]/2$$

Then

$$\alpha_0\,(\text{Theoretical}) = \frac{2}{N}\sum_{m=1}^{N-1}\left[\frac{1-(-1)^m}{1-\cos\frac{\pi m}{N}}\right] y_{cm}$$

Comparisons have been made between the zero lift angles calculated using both of the foregoing methods, from which it has been demonstrated that good agreement exists between the methods for the aerofoils tested, consequently, either method may be used according to one's own preferences.

A further property which is of interest in the design and analysis of controllable pitch propellers is the pitching moment of the section about the aerodynamic centre or moment at zero lift. Thin aerofoil theory defines the aerodynamic centre of the section at a distance of a quarter of the chord from the leading edge and evaluates the magnitude of the pitching moment at this point, this value being independent of angle of attack, and solely dependent, in the first order, upon the shape of the camber line. The magnitude of this moment is given in non-dimensional form as

$$C_{Mc/4} = 2\int_0^1 \frac{y_c\,(1-2x_c)\,dx_c}{\sqrt{x_c\,(1-x_c)}} - \frac{1}{2}\int_0^1 \frac{y_c\,dx_c}{(1-x_c)\sqrt{x_c\,(1-x_c)}}$$

However, for practical calculation purposes use can be made of an approximation proposed by Pankhurst which takes the form,

$$C_{Mc/4} \simeq -\sum_{k=0}^{N} P_k [y_{bx} + y_{fx}]$$

The coefficients P_k are tabulated in Ref[7].

Although the accuracy of this approximation is limited since it is based upon the assumption that the camber ordinates are given by half the sum of the face and back ordinates of the section, it has been shown by making comparisons with the results derived from thin aerofoil theory to be sufficiently accurate for the majority of purposes considered here.

Alternatively, as in the case of the zero lift angle, the moment at zero lift may be calculated by appealing to the work of Riegels and Truckenbrodt in which the following relationship was derived for the moment coefficient,

$$C_{Mc/4} = -\frac{\pi}{N} \sum_{m=1}^{N-1} \left[\frac{1-(-1)^m}{1-\cos\frac{\pi m}{N}} + 2\cos\frac{\pi m}{N} \right] y_{cm}$$

Whilst the correction factors to the angle of zero lift and pitching moment coefficients for standard types of aerofoil are well established, this is not the case for the distorted sections of the controllable pitch propeller working at off-design conditions. Wind tunnel tests have been carried out for a series of distorted profiles from which it would appear that for the outer sections the zero lift angle correction is sensibly constant. However, for the thicker inner sections, it is observed that the zero lift angle correction factor reduces quite considerably with increasing amounts of distortion. Similar work has also been carried out for the quarter chord pitching moment.

Pressure Distributions around Distorted Sections

The pressure distribution around the surface of the helical sections is of considerable importance when trying to evaluate the section loadings. Consequently, a series of wind tunnel experiments were commissioned in order to assess the effects of section distortion on the flow characteristics.

Fig. 11 shows a typical set of viscous pressure distributions obtained from the wind tunnel studies for section shapes of the type shown in Fig. 9. Correlation work has also been undertaken between the results of this experimental work and the invicid and viscous pressure distribution methods due to Wilkinson[8] and Firmin[9]. From this work it has been shown that Wilkinson's method will give reasonable correlation at the design pitch and design pitch −20 degree aerofoils, however, for the more heavily distorted profiles at design pitch −40 further work will be necessary to properly account for flow separation and transition. This is especially true of section shapes developed from high blade area ratio propellers, of the type used by warships and container ships.

At the present time little work has been done on trying to assess the effects of cavitation on the pressure distributions surrounding the distorted sections, and for the moment some of the more general conclusions put forward by Balhan[10] are introduced into the c.p.p. analysis procedure.

Calculation of Blade Loadings

A propeller blade for the purposes of analysis can be replaced by a line of radially varying bound vortices, together with a system of helical free vortices emanating from each radial station whose strength is given by,

$$\Gamma_F = \left(\frac{\partial \Gamma_B}{\partial x}\right) dx$$

The optimum propeller design problem lends itself readily to this type of approach since the resulting equations describing the system can be combined with a minimum energy loss criteria, thereby enabling a complete solution to the system to be obtained.

However, in the case of a controllable pitch propeller when operating at arbitrary settings of pitch and advance coefficient the solution becomes extremely complex since the optimum conditions no longer apply. Rusetskiy[3] in endeavouring to overcome this difficulty, suggested a simplified mathematical model in which the rate of change of bound vorticity along the blade length was zero, consequently, restricting the formation of free vortices to a semi-infinite helical vortex of variable radius from each of the blade tips and a semi-infinite line vortex along the polar axis of the propeller. Although this method depends upon a gross simplification of the physical problem, when used in a suitably modified form gives surprisingly good results for the majority of pitch settings. Concurrently, with the above approach a parallel study was undertaken based upon the earlier work of Burrill. This work, however, is fully reported in Ref. [11].

Having estimated the inflow angles using the simplified lifting line model outlined above the elemental thrust and torque coefficients may then be deduced from the following relationships respectively,

$$dK_T = \frac{\pi^2 x^2 z c}{4D} (1-\alpha')^2 \sec^2 \beta_i (C_L \cos\beta_i - C_D \sin\beta_i) dx$$

and

$$dK_Q = \frac{\pi^2 x^3 z c}{8D} (1-\alpha')^2 \sec^2 \beta_i (C_L \sin\beta_i + C_D \cos\beta_i) dx$$

In these expressions, the lift coefficient can be either determined from the integral of the pressure distribution around the section or from thin aerofoil theory as described earlier. If the resulting radial thrust and torque distributions are integrated across the blade length for each of the blades

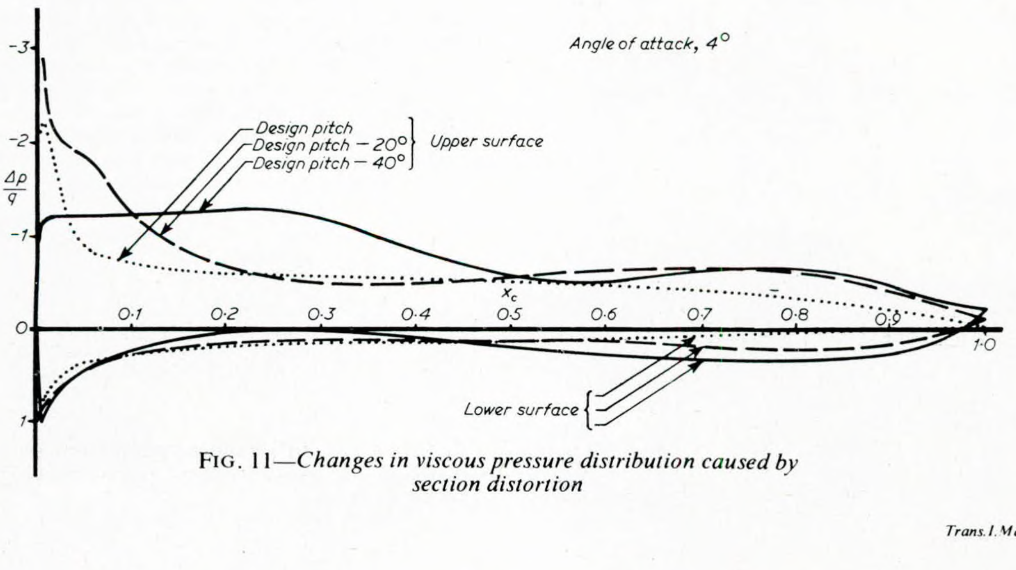


FIG. 11—*Changes in viscous pressure distribution caused by section distortion*

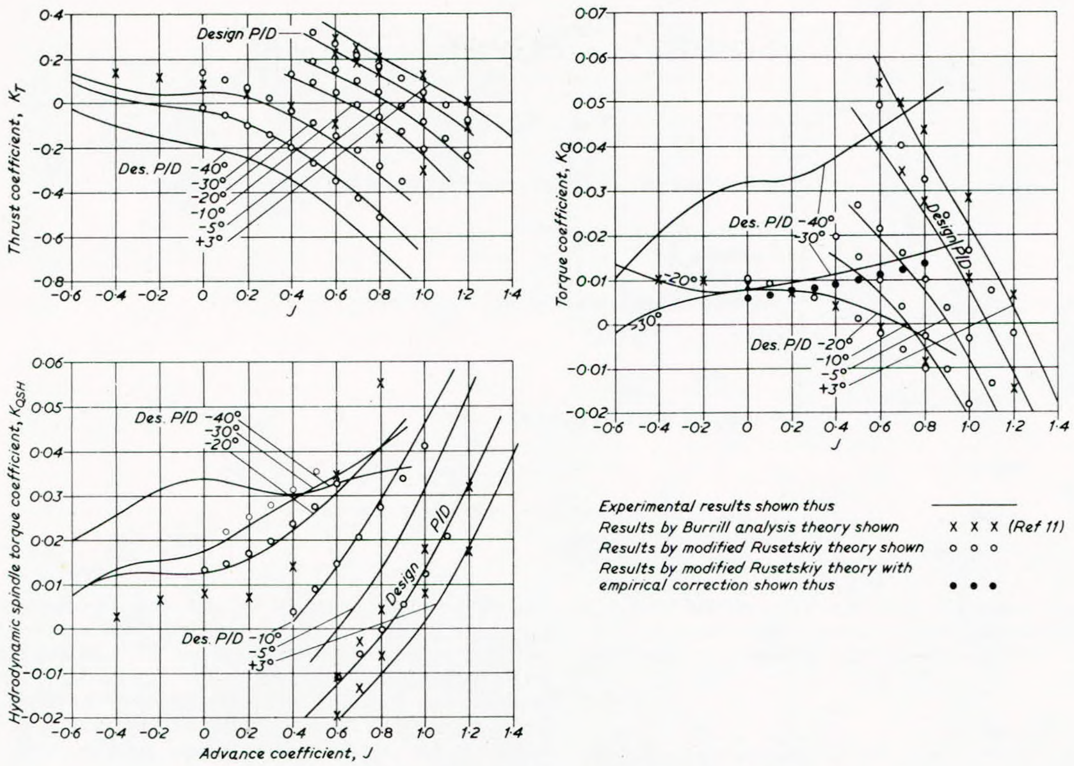


FIG. 12—*C.P. model propeller no. 120 — Comparison of experimental and theoretical results*

the propeller thrust and torque coefficients may then be established. Fig. 12 shows the comparison obtained between the results of this method and the cavitation tunnel results for model no. 120.

Having determined the thrust and torque characteristics for the propeller at a given pitch setting and advance coefficient it only remains, in order to complete the hydrodynamic loading definition, to evaluate the hydrodynamic spindle torque. This may be done either by a direct moment integration of the pressure distribution about the spindle axis, or by giving consideration to the position of the aerodynamic centre of the sections and the forces and moments acting about it. If the latter of the alternatives is adopted, then the elemental spindle torque on the section is given by,

$$Q'_{SH} = \frac{\pi^2 \rho x^2 n^2 D^2 c(1-a')^2}{2} \sec^2\beta_i \left[c\, C_{Mac} + (C_L \cos\beta_i - C_D \sin\beta_i)\kappa + (C_L \sin\beta_i + C_D \cos\beta_i)\eta\right]$$

Where κ and η are the respective moment arms of the force components about the spindle axis, and C_{Mac} is the pitching moment coefficient for the aerofoil in a viscous cascaded flow.

This equation can then be expressed in the convention non-dimensional form as follows

$$K_{QSH} = \frac{\int_{x_h}^{1\cdot 0} Q_{SH}\, dx}{\rho n^2 D^5}$$

In both foregoing equations it should be remembered that the minor effects due to the section defined in the three dimensional helical form rather than the two dimensional planar form have been neglected.

Fig. 12 demonstrates the correlation obtained using the foregoing method, from which it can be seen that generally satisfactory agreement between the calculated and experimental values exists for all but the more extreme pitch settings.

Transient Hydrodynamic Loadings

When the pitch setting of the blades of a controllable pitch propeller is changed whilst the propeller is rotating about the shaft axis an additional vortex system is induced on the blades, thereby introducing a further loading on the blades.

This additional loading has been considered analytically by examining the loadings on an aerofoil working in a combination of translational and rotational flow. From this study it has been shown that for all normal pitch change rates the transient hydrodynamic loadings are of a small order when compared to the quasi-static loading. Table III gives an appreciation of this finding by delineating the theoretical transient loading for an aerofoil typical of a 0·7R section of a propeller blade.

TABLE III—TYPICAL TRANSIENT HYDRODYNAMIC LOADING OF AN AEROFOIL

Angle of attack = 1·0 deg: y_c = 0·02: Inflow velocity = 30m/s

Pitch Change Rate (rad/s)	0	0·02	0·04	0·06	0·08
Percentage change in C_L	0	0·6	1·1	1·7	2·5

A conclusion similar to the one outlined above was reached by Rusetskiy who approached the problem from a consideration of Joukowski's propeller theory. From this analysis he showed that for all practical pitch change rates the velocities induced on the blade by the additional vortex system were negligible.

BLADE MECHANICAL CHARACTERISTICS AND LOADINGS

Section Characteristics

The blade section mechanical characteristics include properties such as the principal moments of inertia, positions of centroids and section areas, all of which may be calculated by employing numerical applications of the standard text-book techniques. Consequently, these methods of evaluation need not be discussed in any great detail here, however, the results of these calculations show some interesting features.

As with the aerodynamic characteristics of the blade sections there is a dependence between the magnitudes of these geometric quantities and the blade pitch setting. A typical example of these variations is shown in Table IV for a section at 0·7R of model no. 120 after the blade has been subjected to a change in pitch setting of −40 degrees from its design position. Upon examining the changes indicated by

TABLE IV — TYPICAL VARIATIONS IN THE GEOMETRIC CHARACTERISTICS FOR MODEL 120 AFTER BEING SUBJECTED TO A PITCH CHANGE OF −40 DEGREES

$r/R = 0{\cdot}700$

Quantity	Axis About Which Quantity is Measured	Percentage Change
Principal Moments of Inertia	Inclined at angle α to pitch line, passing through c.g.	+28
	Inclined at angle α to normal to pitch line, passing through c.g.	+11
Centre of Gravity	Pitch line	+24
	Normal to pitch line at t.e.	+1
Section Area	————	+5
Angle α	Between principal axis and axis parallel to pitch line through c.g.	+24

this table it is of interest to note that although the area of a section at any given radius is a variable depending upon the pitch angle, for any given blade the volume must naturally remain constant.

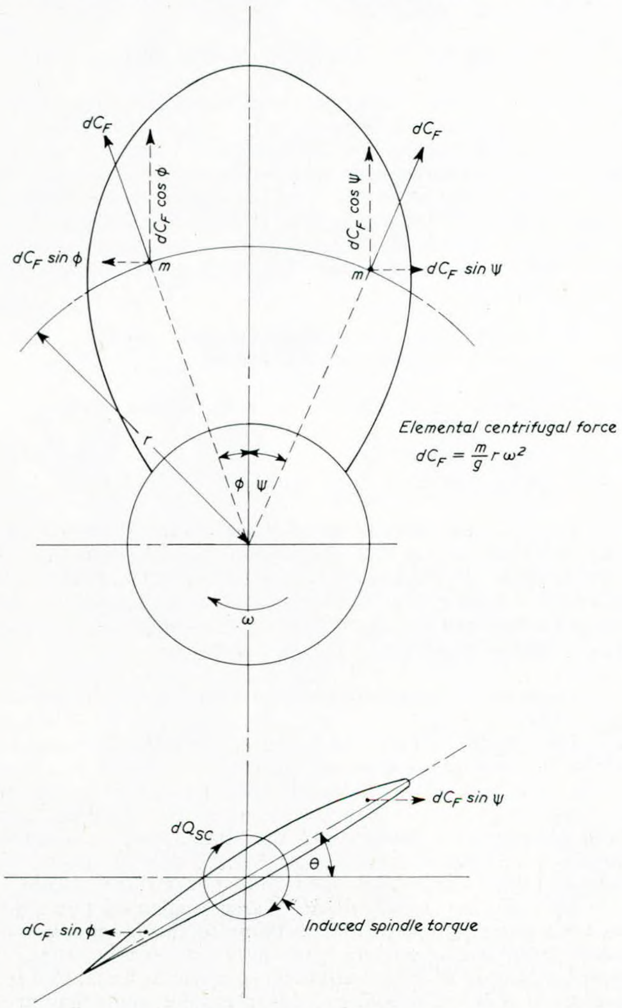
Inertial Spindle Torque

The blade inertial spindle torque arises from the moments produced by the two primary types of angular motion, namely, rotation of the blades about the shaft axis and of the blade about its spindle axis. The former of these moments is due to the centrifugal loading on the blade particles and consequently depends on both the blade pitch angle and the shaft revolutions. By contrast the latter moment evolves from the transient accelerations of the blade elements and is primarily a function of the angular motion of the blade about its spindle axis when taken in association with the propeller revolutions.

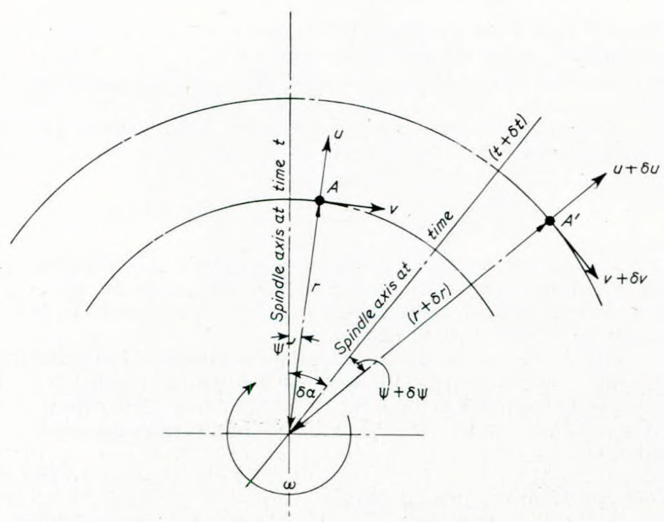
The centrifugal component of the spindle torque is perhaps the easiest to evaluate since it depends, in addition to the blade pitch angle and shaft speed, on the blade geometry.

Consequently, having derived the distorted section geometry at off-design pitch settings by the methods discussed the components of the centrifugal loading in a plane normal to the spindle axis can be readily calculated as described below. Fig. 13(a) shows a diagrammatic representation of the nature of this twisting moment.

Methods for evaluating the centrifugal effects are legion and from the available techniques one proposed by Boswell[12] has been selected, primarily due to its ease of application. If the section geometric characteristics are either known or readily available, then the method reduces to substituting these values into the following expression,



(a) CENTRIFUGAL FORCES



(b) TRANSIENT MOTION

FIG. 13—*Inertial forces acting on blade*

$$dQ_{SC} = \rho_m \omega^2 \left[\frac{(I_{max} - I_{min}) \sin 2\theta}{2} + A\bar{y}\bar{z}\right] dr$$

where, $\bar{y} = r.\sin\left[\frac{(y - c/2 + x_{cg})\cos\theta - y_{cg}.\sin\theta}{r}\right]$

and, $\bar{z} = [(y - c/2 + x_{cg})\sin\theta + y_{cg}.\cos\theta - Ra]$

Having determined the radial distribution of spindle torque from the above relationship the total value may then be obtained by numerical integration, this integral being non-dimensionalized to present the resultant torque in the more familiar twisting moment coefficient form. It should, however, be remembered that with this coefficient, depending as it does on the material density, in order to preserve the compatibility with the hydrodynamic component the density of water must be substituted into the non-dimensional expression,

$$K_{QSC} = \frac{Q_{SC}}{\rho n^2 D^5}$$

This then yields the following expression for the centrifugal spindle torque acting on the blade,

$$K_{QSC} = \frac{2\pi^2}{D^4}\frac{\rho_m}{\rho}\int_{x_h}^{1\cdot 0}\left[\frac{(I_{max} - I_{min}) \sin 2\theta}{2} + A\bar{y}\bar{z}\right] dx$$

The direction of this torque is such that it will always tend to rotate the blade towards a neutral pitch setting. Results obtained from this calculation procedure have been compared with the experimentally derived values as illustrated in Fig. 14. From these curves it becomes apparent that the correlation between experiment and theory is of a high order and consequently it may be concluded that this method is satisfactory for practical calculation purposes.

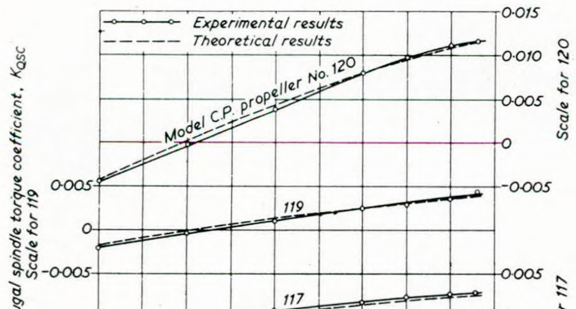


FIG. 14—*Centrifugal spindle torque correlation*

Transient Inertia Spindle Torque

The transient inertial spindle torque is that due to the rotation of the blade mass about the spindle axis when changing pitch as opposed to the purely centrifugal effects and is induced in the following way. Consider a particle A contained within the blade which at some time t is defined relative to the spindle axis by the polar co-ordinates (r, Ψ, z), Fig. 13(b). By the reasoning put forward in the section on c.p.p. blade geometry, after a small increment in time Δt the particle will then be defined by the new set of co-ordinates ($r + \delta r$, $\Psi + \delta\Psi$, $z + \delta z$) relative to the spindle axis, which itself will have been displaced by an angle $\delta\alpha$ from its original position at time t.

If the accelerations of the particle resulting from these displacements are then resolved in a plane normal to the spindle axis, it can be shown that providing the local Strouhal number is small then the transient effects are negligible. In this case the Strouhal number being defined by

$$S_T = f'(t)/g'(t)$$

Where the function $f'(t)$ defines the rate of pitch change and $g'(t)$ defines the angular shaft speed.

PRACTICAL ASPECTS OF C.P. PROPELLER DESIGN

Constraints on C.P.P. Blade Design

The design of a controllable pitch propeller blade is a result of the compromise between a series of conflicting requirements involving parameters such as hub diameter, interference limits, blade flange size, and the hydrodynamic requirements of the propeller.

When determining the size of a c.p.p. hub the following basic factors should be borne in mind:

a) the design and dimensions of the blade actuating mechanism and bolting arrangements must be of sufficient strength to absorb the loadings induced during the absorption of the maximum power of the main propulsion machinery;
b) the blade turning mechanism must be capable of changing the pitch at all operating and manoeuvring conditions, bearing in mind that it is during important emergency conditions that the spindle torque is likely to be the highest;
c) the blade palms must be of sufficient diameter to adequately accommodate the blade root sections and blade securing bolts.

It is economically important, both from the aspects of initial costs and long term efficiency, that the hub diameter is made as small as possible, but this aim is not always compatible with the various technical considerations outlined above.

The size of the blade flange is governed within small limits by the diameter of the hub for a given number of blades. Whilst it may be possible to design a suitable mechanism which will allow the adoption of a relatively small hub diameter, the resulting blade flange ports may not be of sufficient diameter to properly house an acceptable blade root section or to give an adequate blade bolt fixing arrangement. Consequently the designer may thus be forced to accept short, thick root sections to achieve sufficient strength, however, this may be undesirable from the cavitation and efficiency aspects of the propeller. When such cases arise it may be necessary to adopt a marked overhang of the sections beyond the blade palm in order to achieve an acceptable hydrodynamic section shape. If this procedure becomes impracticable then it may be necessary to increase the hub size. Therefore, when designing a hub the maximum possible blade flange diameter, consistent with hub strength, should be selected in order to give the greatest degree of freedom to the hydrodynamic design of the blades.

The blade spindle torque is to some extent related to the shaft torque, but not directly so, since it is dependent on both propeller diameter and the blade width distribution which in turn are dependent on the type of ship and its speed, thus also the cavitation number at which the propeller operates. From the spindle torque equations given previously it can be deduced that the spindle torque increases directly with blade width. In the case of high power and high rev/min applications, for example, warships, container ships and ferries, all of which could demand propellers of a high surface area, it is necessary to restrict the blade widths to give a B.A.R. of less than about 0·80.

This restriction must be imposed in order to allow the free passage of the blades when moving from ahead to astern pitch. In the case of a four bladed c.p. propeller the blade interference widths can be approximated by the following relationship.

Max. Width $= D\,[0{\cdot}771\,x + 0{\cdot}025\,(P/D - 1) + 0{\cdot}023]$

For blades where this restriction applies more than the usually acceptable cavitation is likely to take place, which while being undesirable from the normal hydrodynamic aspects may result in a reduction in the total spindle torque.

It is for the above reasons, namely the effects of blade shape, propeller diameter and hub size on the spindle torque, and the necessity to design a blade turning mechanism of sufficient strength while not being excessive on scantlings, that it has become important to develop a method of estimating more closely the hydrodynamic forces on the blades, not only in the normal ahead design condition but also in the extreme pitch settings and in the various manoeuvring conditions.

The Estimation of Blade Spindle Torque

To enable design studies to be carried out for new ships it is necessary to have the ability to estimate the loadings to which a given propeller will be subjected under manoeuvring conditions. For the cases where a controllable pitch propeller is projected this is especially true of the total non-frictional spindle torque, since this factor is a major determinant in selecting the appropriate hub size and hence the initial cost of the propeller.

Before being able to deduce the conditions which will maximize the spindle torque for a given propeller it is necessary to examine the way in which the loci of the more extreme types of manoeuvre, in terms of advance coefficient and pitch setting, move over the characteristic curves of the propeller, these characteristic curves being the normal thrust, torque and spindle torque coefficients plotted to a base of advance coefficient. One method of investigating these loci is by computer simulations where the ship and propeller are modelled by a system of differential equations which will be described later.

After having obtained the results of such studies for a variety of ships in their different manoeuvring modes it becomes apparent upon making due allowances for cavitation, that for a first approximation, the maximum non-frictional spindle torque at constant shaft speed will generally be less the quasi static spindle torque at zero pitch in association with a zero or slightly negative advance coefficient. For variable revolution designs the problem is rather more difficult since, at the early design stage, the precise nature of shaft speed and pitch changes are not usually known, consequently, these manoeuvres are generally accounted for by treating them initially as constant speed designs and then reducing the resulting spindle torque by a factor of 10 per cent to 25 per cent depending upon the class and type of ship.

Fig. 15 shows a typical estimation diagram for a four bladed propeller having a design pitch ratio of 1·0 and a normal variation in radial skew distribution for differing blade area ratios. Diagrams of this type for a series of design pitch ratios and skew distributions, together with similar data for the associated propeller thrusts and torques, provide a means of estimating the appropriate hub size for a given propeller. Incidentally, it is worth remembering when designing propeller hub mechanisms and also thrust bearings that the maximum thrusts attainable with a controllable pitch propeller under manoeuvring conditions are usually of the order of $-1{\cdot}5$ to $-2{\cdot}2$ times the design ahead thrust.

The Effects of Rake and Skewback on the Total Spindle Torque

The spindle torque may be minimized by consideration of the blade geometry with reference to the spindle axis. The position of centres of pressure and gravity for a given screw section may be moved in an axial direction by changes in rake angle and transversely by modifying the skewback of the blade, without drastically changing the hydrodynamic performance of the propeller.

The effects of these two geometric parameters on the value of the total spindle torque has been examined for the case of a four-bladed c.p. propeller of B.A.R. 0·665 at the design operating condition. Calculated values of spindle torque have been obtained for:

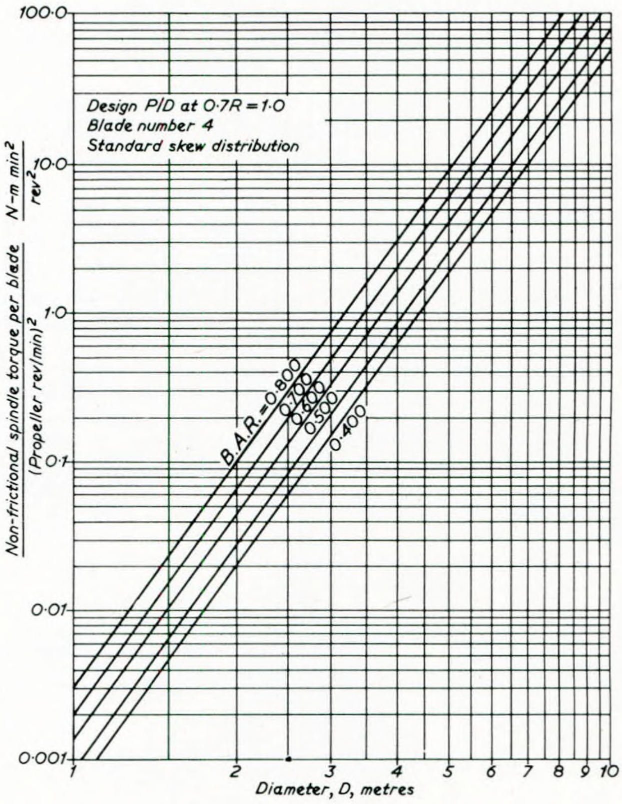


FIG. 15—*Estimated maximum non-frictional spindle torque for constant revolution applications*

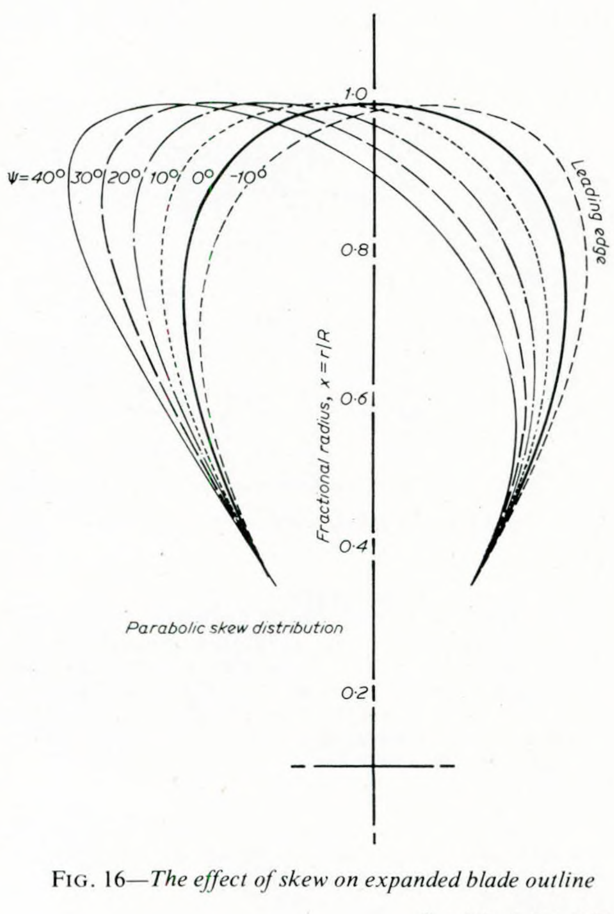


FIG. 16—*The effect of skew on expanded blade outline*

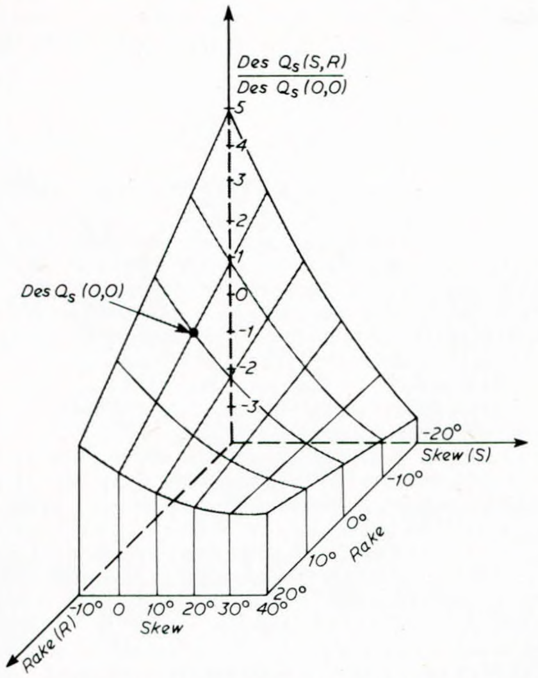


FIG. 17—*The effect of skew and rake on design spindle torque*

1) a parabolic skew distribution, where the angle of skew was varied from $-10°$ to $+10°$;
2) rake angles of $-20°$ to $+20°$.

The blade outlines derived by the varying skewback distributions are shown in Fig. 16 and the resulting total spindle torque values illustrated by the three dimensional diagram, given in Fig. 17.

From this figure it may be seen that the curves of spindle torque with respect to both skew and rake for the greater part of their range have negative gradients. Now the spindle torque including the effects of friction at the design pitch position should ideally be equal and opposite to the maximum spindle torque which is likely to be encountered during manoeuvres, thereby minimizing the overall maximum value of this moment. In terms of Fig. 17 this condition normally implies that the ratio of Design $Q_S(S,R)$ /Design $Q_S(O,O)$ should be greater than unity. Consequently by considering the following relationship which is approximately true for small values of skew *(S)* and rake *(R)*

$$\lambda = 1 + \frac{\partial\lambda}{\partial S}.S + \frac{\partial\lambda}{\partial R}.R$$

we can deduce that for an ideal optimization of the twisting moment in normal case of a propeller with slight positive skew, the possible introduction of forward rake would prove advantageous.

However, in practise the physical constraints of the wake induced vibration, apperture clearances, c.p.p. pump durability, hub mechanical details, etc, often prevents full advantage being taken of the ideal choice of skew and rake. Nevertheless, the concept of optimization within the practical limits of a given application should always be borne in mind when designing a c.p.p. system. If this is correctly done then it is usually desirable that the hub mechanism be capable of producing a greater turning effect from astern to ahead pitch then vice-versa.

The Effects of Ducts on Blade Spindle Torque

At this stage in the development of controllable pitch propeller off-design calculation methods it was considered worthwhile to conduct some preliminary investigations into the changes in blade spindle torque characteristics due to the presence of a duct. Consequently, a c.p. propeller was designed using the method described in Ref. [15] and then tested in the cavitation tunnel both with and without a duct at blade pitch settings of design, -20 degrees and -40 degrees.

For strict compatability between a ducted and a corresponding "open" propeller, there should be a difference in the radial pitch distribution, thereby enabling the available power to be absorbed at the same advance coefficient in both cases. However, in this instance to overcome this problem the comparison is made at differing advance and torque coefficients, such that the power absorbed in each case is the same. This is achieved by plotting values of K_Q for various advance coefficients so that the quotient K_Q/J^3 remains constant. The intercept of this curve with the experimental torque coefficient results gives the advance coefficient J at which equal power is absorbed.

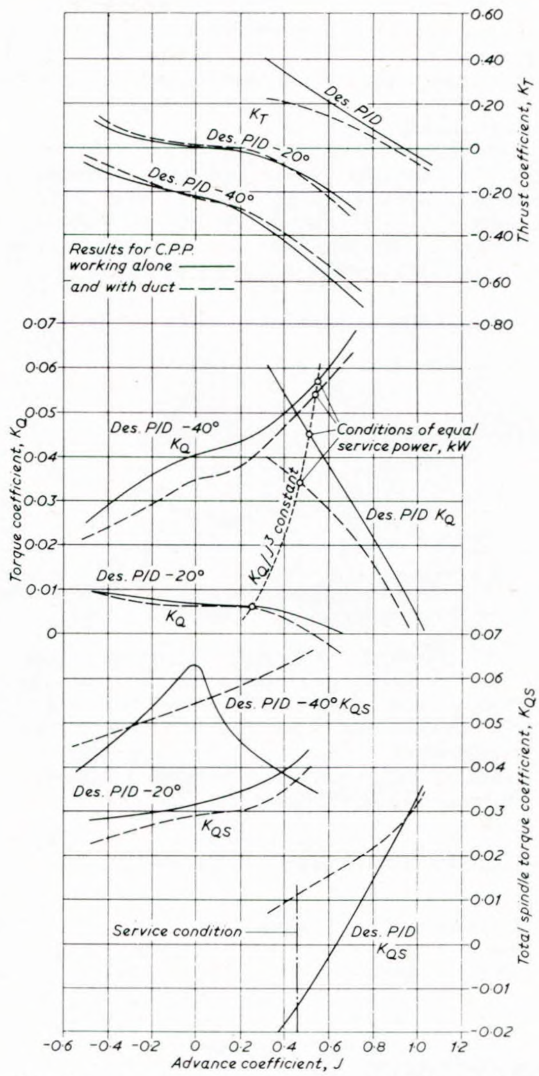


FIG. 18—*Comparative results for a c.p. propeller working alone and in a duct*

The results of these experiments are shown in Fig. 18, from which it can be seen that the presence of the duct tends to reduce the total spindle torque acting on the blades in the design pitch -20 degree case over the entire range of advance coefficient, this would also seem to be true of the design pitch condition for values of J less than the service condition. However, in the case of the -40 degree pitch position it would appear that the presence of a duct tends to increase the spindle torque for the majority of the range considered, except perhaps at the bollard pull condition where the "open" propeller results tend to peak. This latter effect is no doubt due to differences in loading assumptions between ducted and non-ducted propellers.

It is often considered that the most onerous spindle torque condition is to be found when the blades are in a moderate astern pitch setting in association with a low negative advance coefficient. Consequently, it would appear from the experimental results that the use of a duct may perhaps provide a slight reduction in the blade spindle torque.

Blade Loadings Encountered during Manoeuvres

When a ship is undergoing a manoeuvre involving an alteration to either the blade pitch or shaft speed the blade loadings can no longer be calculated from the conventional form of dynamic equilibrium analysis that is used for self propulsion studies.

In the case of manoeuvring there is an imbalance between the propeller thrust and ship resistance which produces the accelerating or braking force, consequently when trying to analyse this type of ship performance appeal must be made to Newton's laws of motion in their respective forms for application to the ship and propeller shaft.

i) For the motion of the ship,

$$(m + \Delta m)\frac{dV_S}{dt} = T(1 - t_d) - R_S \qquad (1)$$

ii) For the motion of the propeller shaft,

$$2\pi I \frac{dn}{dt} = Q_P + Q_E + Q_{ALT} + \sum_{1}^{K} Q_{LOSS} \qquad (2)$$

Where K = No. of stations along the shaft where losses can occur (e.g. bearings, gears, etc)

These equations form the basis of a mathematical model for the study of the motions of the ship and shafting, however, in order to complete the system definition, thereby making it also soluble, it is necessary to introduce some control functions for the rates of change of pitch and shaft speed. The control of pitch can be quite adequately represented by a function of the form

$$\frac{d\theta}{dt} = f \text{ (initial pitch setting, time, manoeuvre, etc)} \qquad (3)$$

However, the control of shaft speed is not quite so easy since to model the system precisely some account must be taken of the engine response. Normally, this control parameter is expressed in terms of the amount of energy supplied to the engine together with a mathematical idealization of the engine response. Consequently, this final relationship can take the following form,

$$\frac{dE}{dt} = g\text{(time, load on engine, engine response, speed, environment, etc)} \qquad (4)$$

Having obtained a set of equations of the type indicated above, one method of solution is to interpret them in a suitable numerical form for analysis on a digital computer and to solve the differential equations using the Runge-Kutta method. Experience with this method has shown that provided the problem is properly posed then this provides a fast and reliable method of solution for use on computers of quite limited core.

When solving equations of the form shown above it is necessary to employ relationships which describe input data. This is very often not at all easy to estimate, for example the entrained mass of the ship and the wake and thrust deduction factors. With regard to the entrained mass Δm of the ship, Rusetskiy [3] proposes the use of a curve which expresses the entrained mass as a function of the ratio of the water-line length to the cube-root of the volumetric displacement of the ship. Now if this ratio is represented as ζ then it is possible to approximate the data presented in Ref. [3] by the following relationship,

$$\mathrm{Ln}\left(\frac{\Delta m}{m}\right) = 0{\cdot}059\,(10 - \zeta)^{2{\cdot}06} - 3{\cdot}058$$

Little information is available on the transient effects of both the wake and thrust deduction factors during accelerating and reversing manoeuvres. Consequently, each problem has to be treated on its own merits with due consideration to any other ships that have undergone similar analysis. There is, however, evidence to suggest that in the case of the wake fraction a relationship of the form might not lead to serious errors being incurred.

$$w_t(V_S) = w_{to}\left(\frac{V_S}{V_{SO}} + 1\right)/2$$

Where w_{to} = design Taylor wake fraction at speed V_{so}.

When considering ship speeds in the astern direction then it would be reasonable to assume a zero or very small value for the Taylor wake term. For the case of the thrust deduction factor, it appears that this factor in the steady state would tend to be rather more constant than the wake fraction in the ahead direction. However, for the manoeuvring case it is suggested in Ref. [13] that an additional factor to account for the various pitch settings of the c.p.p. should be incorporated into the analysis procedure.

Despite the problems mentioned above there is much valuable information to be gained by the designer from estimating the loadings induced by manoeuvring operations, since it is at these off-design conditions that maximum forces and twisting moments are encountered. To illustrate this point Fig. 19 shows the estimated changes in the non-frictional spindle torque likely to be encountered during first a braking and then an accelerating manoeuvre with a 13 500 tonne displacement cargo ship. Similar diagrams can also be produced for the variations in thrust and shaft torque against pitch for different rates of pitch change. The relationship between shaft torque and time or pitch is of great importance in cases where the torque becomes negative, thereby implying that the propeller is transmitting energy from the water along the shaft to the gear box or prime mover. This effect most usually occurs in the faster ships when the pitch is suddenly reduced at high speed. From analysis of this condition it has been deduced that this effect can occur at propeller advance coefficients in excess of about 0·53, consequently this should be borne in mind when designing a ship having a design advance coefficient greater than about 0·5.

It is often necessary to determine the head reach of ship under emergency stopping conditions. This is easily done using the mathematical model outlined earlier since the head reach is defined as

$$H = \int_0^T V_S(t)\,dt$$

Where T is the time taken to stop the ship from the commencement of the manoeuvre.

Fig. 20 shows a comparison of a theoretical prediction obtained from this method with the measurements recorded from the sea trials of a warship.

Constant Revolutions Operation

The operation of a controllable pitch propeller at constant shaft speed undoubtedly provides an ideal situation for the generation of electrical power using the main propulsion engines. However, from the ship propulsion viewpoint there are several problems that need careful consideration.

If it is decided to operate a c.p. propeller at constant revolutions then a penalty must be paid in propulsive efficiency at a second or off-design operating condition. For example, consider the case of a twin screw ferry having a full power capability of 4474 kW (6000 shp) per shaft at 240 rev/min in association with a ship speed of 22 kn, and suppose a duality of operation was required in so far as full power was required during the day and half power at night. Then if the propeller was designed for a compromise between the two power conditions and the shaft speed held constant, an overall loss in efficiency at both operating conditions of about 4 or 5 per cent would result, when compared to the case where the revolutions were allowed to vary by say 20 per cent.

In addition to the efficiency problems discussed above, constant shaft speed operation produces more onerous cavitation conditions under which the propeller is required to work. At the full power condition the loading is such that a heavy

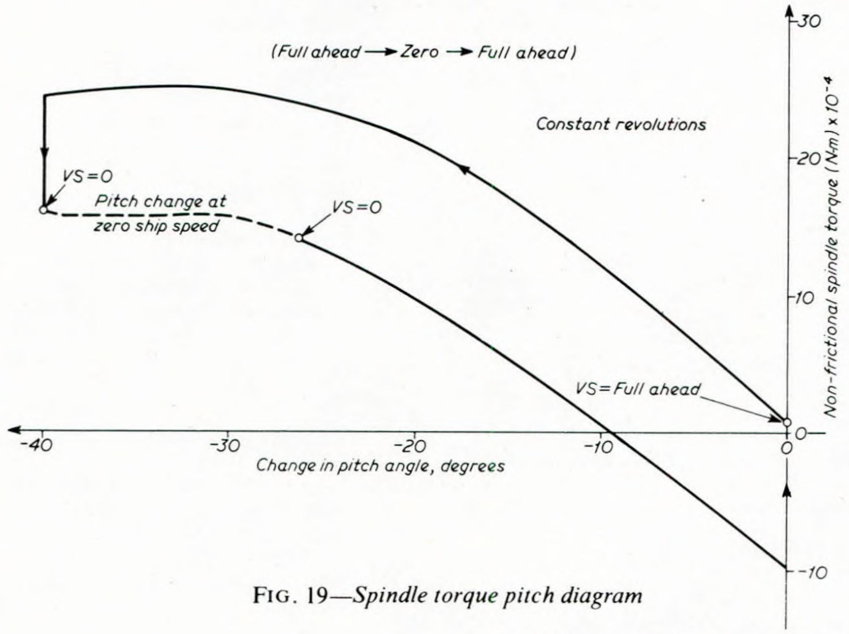


FIG. 19—*Spindle torque pitch diagram*

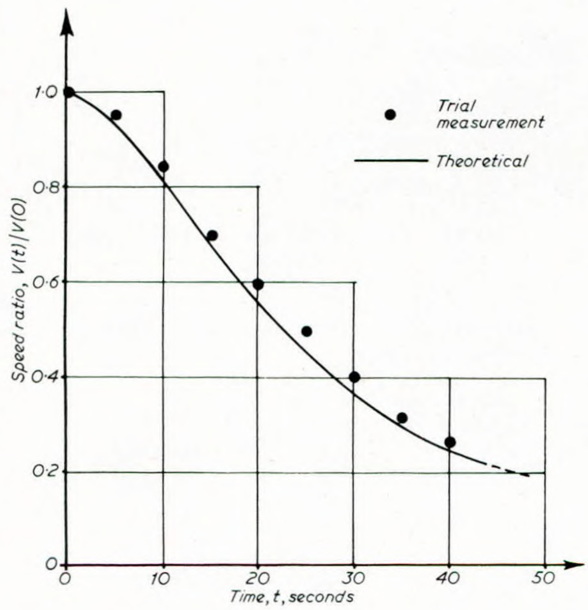


FIG. 20—*Ship-simulation correlation*

suction pressure is induced on the back of the blade sections, consequently there is a tendency toward back cavitation. If the blade is then moved to a finer pitch setting in order to accommodate a lower power condition, since the angles of attack of the section have become reduced in magnitude there is a bias toward face cavitation. It becomes apparent that in such cases as these the blade section design has to be based upon a careful compromise between these two conflicting conditions. Often this process is made more difficult by blade interference limits if the overall cavitation conditions are onerous or by a poor wake field ahead of the propeller.

Constant shaft speed systems also imply larger blade loadings when undergoing manoeuvring conditions, in particular the blade spindle torque may be increased by between 10 and 30 per cent, thereby necessitating the use of greater oil pressures. If these pressures become too high then a larger servo-motor would be required for the given application which would increase the initial cost of the controllable pitch propeller installation.

ACKNOWLEDGEMENTS

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NOMENCLATURE

Upper Case Letters

A	Section Area
B.A.R.	Blade Area Ratio
B_p	Power Coefficient
C_D	Drag Coefficient
C_L	Lift Coefficient
C_{Mac}	Moment Coefficient at Aerodynamic Centre
$C_{Mc/4}$	Moment Coefficient at Quarter Chord.
D	Propeller Diameter
H	Head Reach
I	Polar Moment of Inertia of Propeller Shaft System
I_{max}	Maximum Moment of Inertia of Blade Section about the Centroid
I_{min}	Minimum Moment of Inertia of Blade Section about the Centroid
J	Propeller Advance Coefficient = $Va/(n.D)$
K_T	Thrust Coefficient = $T/(\rho n^2 D^4)$
K_Q	Torque Coefficient = $Q_P/(\rho n^2 D^5)$
K_{QSC}	Centrifugal Spindle Torque Coefficient
K_{QSH}	Hydrodynamic Spindle Torque Coefficient
P	Mean Pitch
Q_{ALT}	Torque to Drive Shaft Driven Alternators
Q_E	Engine Torque
Q_P	Propeller Torque
Q_S	Non-frictional Spindle Torque
Q_{SC}	Centrifugal Spindle Torque
Q_{SH}	Hydrodynamic Spindle Torque
R	Radius of Propeller Tip
Ra	Axial Offset of Blade Section due to Rake
R_S	Ship Resistance
T	Propeller Thrust
Va	Speed of Advance
V_S	Ship Speed
W	Resultant Inflow velocity to propeller

Lower Case Letters

a	Axial Inflow Factor
a'	Tangential Inflow Factor
c	Section Chord Length
m	Ship Mass Displacement
Δm	Entrained Mass of Ship
n	Shaft Revolutions
Δp	Static Pressure
q	Dynamic Pressure
r	Section Radius
r_h	Hub Radius
t	Time
t_d	Thrust Deduction Factor
w_t	Mean Taylor Wake Fraction
w_t'	Local Taylor Wake Fraction
x	Non-dimensional Section Radius
x_c	Non-dimensional Chordal Station
x_{cg}	Distance of Section Centroid from Mid-Chord, parallel to the Nose-Tail Line
x_h	Non-dimensional Hub Radius
y	Distance of Spindle Axis from Blade Section Leading Edge
y_b	Non-dimensional Back Ordinate of Section
y_c	Non-dimensional Camber of Section
y_{cg}	Distance of Section Centroid from and perpendicular to the Nose-Tail Line
y_f	Non-dimensional Face Ordinate of Section
z	Blade Number

Greek Letters

α_o	Zero Lift Angle
β	Advance Angle
βi	Hydrodynamic Pitch Angle
Γ_B	Bound Circulation
Γ_F	Free Circulation
δ	Speed Coefficient
ξ	Water Line Length/(Volumetric Displacement)$^{1/3}$
θ	Blade Section Pitch Angle
θ_o	Section Effective Pitch Angle
λ	Ratio of Spindle Torque for blade of Arbitary Skew and Rake to Spindle Torque at Zero Skew and Rake
ρ	Density of Water
ρ_m	Density of Propeller Material
σ	Cavitation number = $\Delta p/q$
φ	Angular Position of Blade in Disc
ψ	Skew Angle and Angular Displacement
ω	Rotation Speed

Discussion

MR. J. N. EDGAR, M.I.Mar.E., said he hoped to be forgiven if, in the interests of brevity, he limited his contribution to a few simple questions.

Firstly, he asked at what Reynold's number were the various tests in the cavitation tunnel conducted?

Secondly, if the authors were satisfied with the difference in K_T, of, in some cases, 10 per cent or more, between the Gutsche and Schroeder data and the other results shown in Fig. 3, could this possibly be because the models were in a different category from those tested by the authors, i.e., more akin to Gawn/Burrill KCA type?

Thirdly, would it be possible, in the interests of examining the behaviour of the c.p. propellor under some dynamic operational conditions, to simulate in the cavitation tunnel what would really happen in some of the failure modes?

In other words, what would happen to the propeller in the event of oil pressure failure, for whatever reason? Would it be practicable, for example, to modify the propeller model so that all four blades were constrained at some predetermined pitch setting and, having attained some steady state operating condition in the tunnel, to then release the constraints and allow the blades to take up some new attitude depending on the particular operating condition chosen, i.e. to simulate an oil pressure failure? This was something a lot of people were interested in. It was appreciated that it would probably not be possible to retrieve any measured data from the propeller, but even a demonstration of the behaviour might be useful. For example, some typical conditions that might be simulated in this manner could be:

i) what would happen at oil pressure failure when the propeller was operating at the normal free running ahead condition at design pitch?;

ii) what would happen at oil pressure failure when the propeller was operating at the normal free running astern, steady state condition?;

iii) with one propeller of a twin or triple screw ship windmilling with the ship moving ahead, what would happen to the pitch?;

iv) what would happen in the same instance with the propeller locked, i.e. zero rev/min, with the ship moving ahead?;

v) with propeller(s) at zero pitch and low speed of advance, what would happen on oil pressure failure?

If these things could be simulated or demonstrated in the tunnel, possibly without getting any measurements, it would be useful, just to see what would happen.

These matters were often of interest to ship operators. It could be argued that the information could be obtained in full scale, but even so, the facility to demonstrate such behaviour at model scale would be useful and might pave the way to an examination of the dynamic studies which were evidently much needed if only for the qualitative information that could be obtained.

Fourthly, he asked if the authors could elaborate on the ease with which the various hydrodynamic components of the equations in the section on calculation of blade loadings, could be arrived at. Should it be possible to use the equations in the form given to arrive at hydrodynamic spindle torque, etc, without recourse to diagrams such as shown in Fig. 15? He asked if the authors could also explain the penultimate paragraph of this section on the calculation of blade loadings.

Fifthly, he asked if it would be worth while in certain circumstances to introduce some form of air injection on to the propeller blades to alter the pressure distribution configurations which created high spindle torque.

Sixthly, he asked for ideas from the authors on how to standardize in some manner the method of producing and examining the geometric and hydrodynamic properties of the type of distorted section shown in the paper. The study of such sections could be useful in other types of propeller, i.e., the controllable pitch bow or transverse propulsion unit, where one started with a flat plane blade, because its performance had to be the same in both directions. It could be also useful in the design of propellers for such special ships as, for example, a polar ice-breaker, where there was perhaps the need for the same ahead and astern thrust. One could start with a propeller blade which was flat and turn it either into an astern position or an ahead position, and it would develop similar thrusts for a given torque.

Knowledge of the characteristics of blade sections with initial "distorted" type of "*S*" shaped camber lines might also prove useful in the design of fixed pitch propellers with a requirement for a proportionately high astern thrust; as for example in the case of ice-breakers or tugboats.

Finally, he asked if the skew referred to in Figs. 16 and 17 was defined as a shift of the generating line in the transverse plane, or did it also include rake by being a skew of the blade tip along a helical path?

MR. H. WOODS, M.I.Mar.E., said that on sea trials there always seemed to be an air of relief that at full power the fixed pitch propeller was running two to three revolutions fast, as was intended for a clean ship. This allowed a margin for increased hull resistance after a time at sea when the revolutions and thrust would come to the design values. However, this margin did not always occur, and when it did not, it could prove expensive.

There would seem to be some advantage in using a controllable pitch propeller in that some compensation in pitch could be made to allow for errors in design (propeller or hull) and compensation for increased ship resistance with time where the propeller was operating at off-design conditions.

He asked what information ought to be sought by the shipowner if one assumed that a c.p. propeller was to be employed and one was offered by a manufacturer. He intended to show that in some cases the initial assessment of c.p. propellers by an owner was simple and meaningful, and also to show how the information could be easily provided.

If the vessel in question was a normal cargo ship or tanker one would expect the normal continuous full power to be used for the majority of the time, and it was in this area where inefficiencies really counted most. Assuming the engine speed must stay constant for electric power generation and a specific propeller was being offered then the following parameters were constant: diameter, revolutions, delivered power and torque.

To show how the final information was arrived at, one could start with a Troost diagram, admittedly for a fixed pitch propeller. This was shown in Fig. 21.

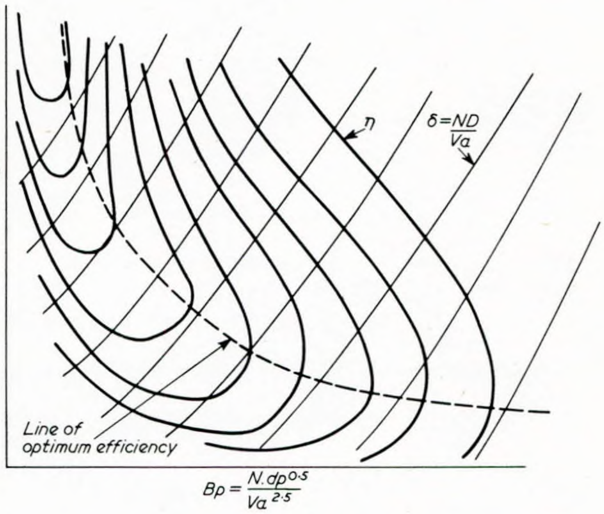

FIG. 21—*Typical Troost diagram*

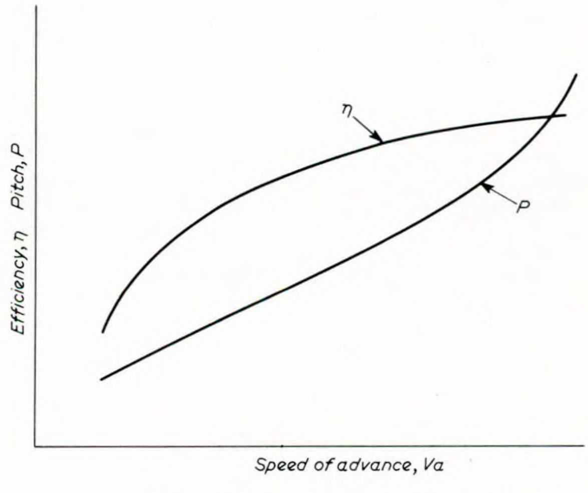

FIG. 22—*Graphs of pitch and efficiency against speed of advance extracted from a typical Troost diagram along the line of optimum efficiency with revolutions* (N), *diameter* (D), *and delivered power* (dp) *all constant*

Factors involved were pitch (P), diameter (D), efficiency (η), delivered power (dp), revolutions (N) and speed of advance (Va). As D, dp and N were constant one could plot pitch (P) and efficiency (η) against the speed of advance (Va) along the line of optimum efficiency. The resultant graph would probably be as in Fig. 22.

However:

$$\text{the efficiency, } \eta = \frac{T \times Va}{Q \times 2\pi \times N}$$

where Q = shaft torque
and T = propeller thrust

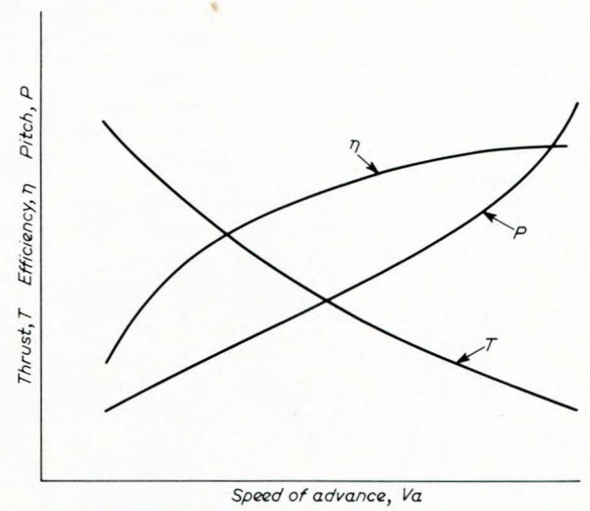

FIG. 23—*Thrust line derived from Fig. 22 as thrust was proportional to the efficiency divided by the speed of advance*

But torque (Q) was constant as were the revolutions (N) and 2π

$$\text{So } T = \frac{\eta}{Va} \times (QN2\pi)$$

$$\text{or } T = \frac{\eta}{Va} \times \text{a constant}$$

Thrust T could be derived from Fig. 22 and could be shown in Fig. 23.

Mr. Woods said the basis of the graphs was from a Troost fixed pitch propeller diagram. This meant that any change of pitch necessitated the selection of a new propeller, which effectively meant that a change in the pitch requirements was obtained by a change in the pitch angle all the way down the blade. However, for a controllable pitch propeller, when the pitch was changed, all sections turned through the same angle so that the pitch face was no longer a true helical surface to the detriment of efficiency. He submitted that for a controllable pitch propeller, the resulting efficiency and thrust lines would be as shown on Fig. 24.

If one decreased the pitch from the design pitch P_1 to P_2 the speed of advance would drop from Va_1 to Va_2 and the thrust would increase as there would be constant power input.

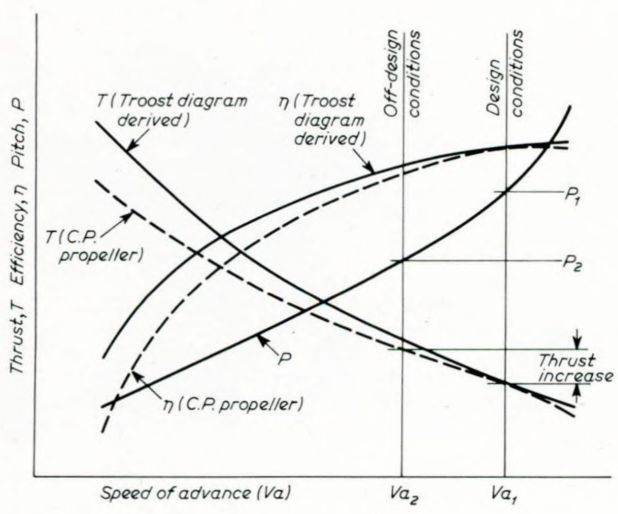

FIG. 24—*Trend of c.p. propeller efficiency and thrust compared with Troost diagram derived efficiency and thrust*

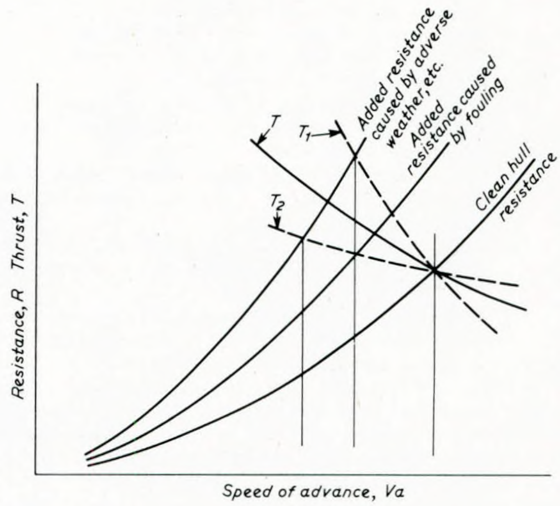

FIG. 25—*Showing that the slope of the thrust/speed of advance line indicated the suitability of c.p. propellers for off-design conditions*

If hull resistance lines were superimposed on the thrust line one could see the effect of hull fouling, bad weather or error in hull design on the speed of the ship. Shown in Fig. 25.

As fouling increased from the clean hull condition the speed of advance would fall and pitch and thrust would be changed to compensate. The lines T, T_1 and T_2 could be representative on thrust of three controllable pitch propellers offered to an owner and it would be seen that a ship with a line T_1 would fare a great deal better than one which had a line T_2. The thrust lines were dependent upon design of the c.p. propeller.

He said the shape, or slope, of the thrust line of a propeller should be asked for by an owner who was attempting to assess the features of controllable pitch propellers offered. The information could be easily provided and for the conditions assumed the assessment by the thrust line method was simple and meaningful. He asked if the authors would care to comment.

MR. P. F. C. HORNE, M.I.Mar.E., asked for the authors' comments on the stressing which resulted from operating in off-design conditions both from the point of view of ship fouling and also from operating in a wake field. It would seem that one could, from the information obtained from the tunnel tests, derive some figures for the bending on the blade roots which could be used to derive loadings on the blade fixings, both the fixing ring (or retaining ring) and also the blade bolts. He thought it would be of considerable advantage if one could get some realistic information of these loadings, which could be correlated with known wake forms of ship designs. He asked if the authors could produce any figures on this.

MR. W. MCCLIMONT, F.I.Mar.E., (the Chairman), said it was interesting to see the data the authors had presented in Fig. 6, although it was a little difficult by just looking at this to find a very easy correlation between the torque coefficient and the spindle torque. Many years ago he had been concerned with problems arising from the transfer of torsional characteristics of the shafting by the propeller to the thrust. He wondered if the authors had had the opportunity of studying in a free water condition what happened if one put vibratory torque into the system rather than a steady torque, and determined, by measurements, just what the resulting spindle torque was. In other words, to what extent did the vibratory torque input to the propeller blade reflect itself in the spindle torques? Further, if one superimposed a wake condition on the foregoing, just what could one build up in spindle torque?

It would also be of great interest to know whether one could, in this way, develop a limiting criterion for torsional vibration in the shafting system.

Correspondence

MR. T. P. O'BRIEN, F.I.Mar.E., said in his written contribution that the design of marine screw propellers was a complicated subject, the design of modern controllable pitch screws was even more complicated and the data given in the paper provided a useful addition to the somewhat scarce amount of information available to the designer.

However, at the outset it was relevant to enquire what were the alternatives to the controllable pitch propeller. In its existing form it was a complex mechanism which had a number of blades that could each be rotated about its axis thus altering the operating condition by changing the pitch. A simpler propulsion device consisted of a fixed pitch screw driven *via* a two-speed gearbox, and here the operating conditions were altered by changing the gear speed and thus operating at a different rate of rotation. By comparison with a conventional fixed pitch screw a controllable pitch screw generally had wider blade tips, thicker root sections and large boss diameter ratios, and the combined effects of those changes in geometrical features on screw performance could be as much as a loss of 4 per cent in efficiency.

However, although the adoption of a two speed gearbox, instead of a single speed gearbox, for driving a fixed pitch screw would also result in adverse performance due to the additional gearing, here the loss in efficiency would be not more than about 2 per cent. A number of small tugs fitted with two speed gearboxes have been found to give satisfactory performance under both towing and free running conditions, and it would be interesting to have the opinions of the authors on the use of two speed gearboxes *in lieu* of controllable pitch screws for moderate size vessels.

In Table I, where some of the geometrical features of the model screws were given, model screw numbers 117 and 118 appeared to be very similar and both the pitch ratio and blade area ratio were very close. Were there wide differences in the other geometrical characteristics of these screws which warranted them both to be tested?

DR. E. J. GLOVER, wrote that in the sections on blade geometry and propeller hydrodynamics, considerable emphasis was placed on the problems of determining the shape and characteristics of "distorted" aerofoil profiles derived from the intersection of cylindrical planes with the blade at various pitch angles. The concept of the use of blade elements was common to most propeller design and analysis methods and there was an abundance of data available for the more typical sections associated with the propeller at design pitch.

For the sections corresponding to extreme pitches there was little experimental data and the analytical methods might not be sufficiently accurate. In fact it was not the geometry of the blade which changed but its position in relation to a fixed set of co-ordinated axes. Would it not be possible to move away from dependence on the blade element theory to the consideration of the entire blade as a lifting body?

The geometry of the blade in a form suitable for this type of approach had already been established (Fig. 8) and would remain fixed, the variable becoming the direction of the onset flow. At a particular pitch angle the problem would involve the determination of the surface distribution of source and vortex strength to represent the blade and its loading. This type of approach was developed by Hess and Smith for the case of the aircraft wing and had successfully been applied by Luu to the design and analysis of propellers at design pitch. The numerical work involved was considerable but within the capacity of large, high speed computing systems.

The expression for determining the maximum blade width for four bladed propellers was useful. To what extent was the expression a function of blade shape and to which blade shape was it applicable?

Could the widths for other blade numbers be derived by direct proportioning or were similar expressions available?

Authors' Reply

In reply to Mr. J. N. Edgar, the authors said that following the completion of the tests reported in the paper, experiments were conducted with c.p. propeller model no. 120 running at three different advance coefficients, i.e., at the ship service operating condition, the bollard pull and the astern running condition. In each case the advance speed and revolutions were adjusted to give a range of Reynold's number from 6×10^5 to 2×10^6. The results indicated that at low advance speeds there was some slight scale effect on the thrust and torque, so that in the range of advance coefficient $J = -0{\cdot}15$ to $+0{\cdot}15$, the K_T and K_Q results given in Figs. 4 and 12, were subject to slight error.

Despite the above mentioned scale effect, spindle torque was unaffected.

The comparison of the experimental results with those from other sources as given in Fig. 3 indicated certain differences in the value of the thrust coefficients K_T. At design pitch $-40°$ the resulting thrust, as measured by the tunnel shaft dynamometer, was considerably greater in a negative sense than that measured by the bladed dynamometer and also that given by the Gutsche-Schroeder series. This might be accounted for by the relatively poor response of the standard shaft dynamometer when measuring the fluctuating forces associated with eddy shedding flow conditions which existed over almost the complete range of the tests with the blades at design pitch $-40°$

Referring to the thrust coefficients at design pitch $-20°$, here both dynamometers yielded similar results but those from the Gutsche-Schroeder series showed considerable difference. A cross plot at $J = 0{\cdot}6$ indicated a slight unfairness in the Gutsche series results at this pitch but it should be pointed out that the results of this series had not been corrected for equal power aborption with that of the model propeller.

While the authors agreed that an examination of the behaviour of c.p. propellers in failure modes was important, with the present experimental instrumentation certain difficulties arose which prevented such tests being carried out. The blades were attached to the hub independently of each other and there were no means of releasing the pitch locking arrangements by remote control. However, the design of a hub assembly which allowed the pitch to be modified remotely while the model propeller was in a running condition had now been prepared. This dynamometer was intended for use when conducting experiments in transient working conditions so that examination of the failure modes mentioned by Mr. Edgar should be possible with this type of instrumentation.

Nevertheless, the authors had carried out certain full scale experiments into determining the behaviour of c.p.p. blades under a failure mode. Fig. 26 showed the results of one such test on a twin screw ferry where an oil failure was simulated on the port propeller and the blades allowed to take up an unrestrained position of equilibrium.

With regard to the question raised as to the relative ease with which the equations on page 170 could be used, it would be seen that these were basically the same as those derived by Burrill in his 1943 paper. However, the hydrodynamic parameters used in these equations were derived from the model outlined in the preceding paragraphs of the paper and based on Rusetskiy's approach, with certain modifications. The purpose of deriving the diagrams such as that shown in Fig. 15 was to provide a means of estimating the non-frictional spindle torque without having recourse to the more elaborate lifting line calculations until the design had reached a sufficiently advanced stage to warrant such an approach.

The minor effects referred to in the penultimate paragraph of the section on the blade loadings referred to the necessity of resolving the lift and drag force vectors about a three dimensional cartesian frame due to the blade section lying on a helical surface. For example, the lift force on a particular section would produce a component of force and moment about each of the cartesian axes upon which the

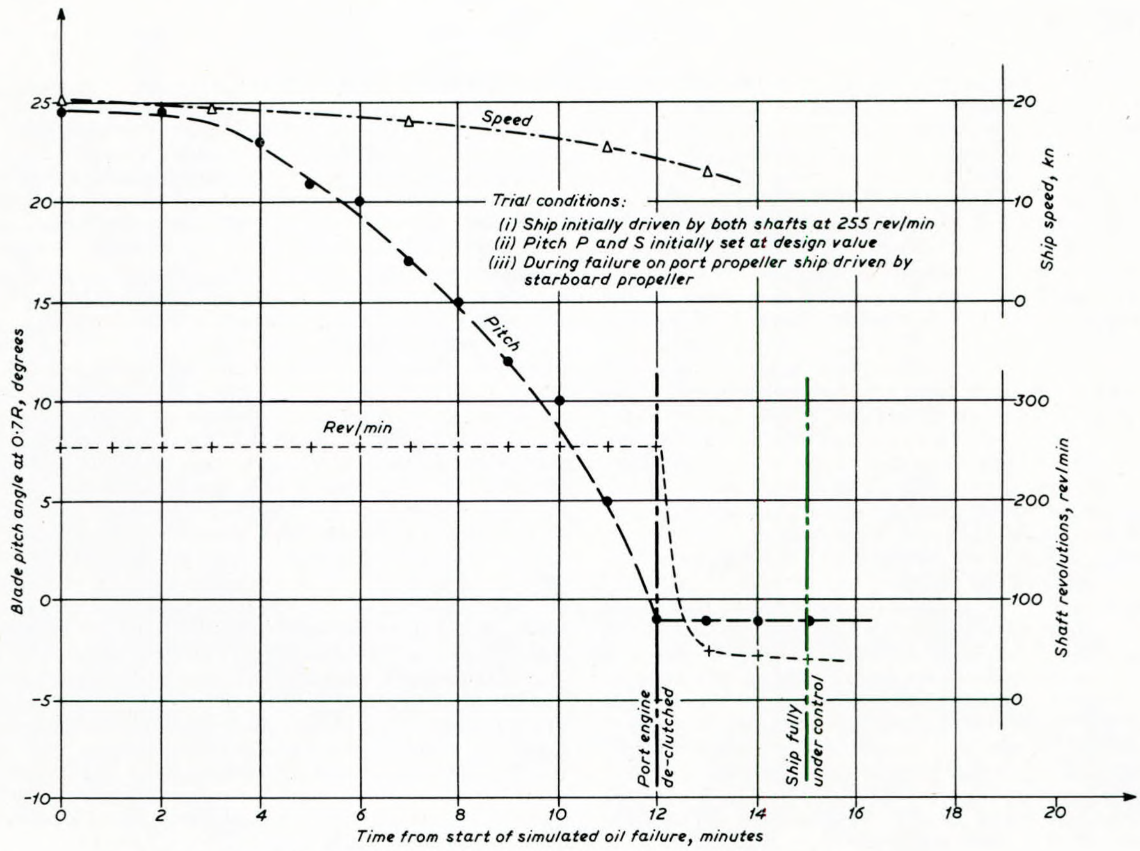

FIG. 26—*Example of full scale simulation of oil failure on a twin-screw ferry.*

blade was defined.

Experiments carried out by Van Gunsteren had shown that ventilating air through holes on the face of the blade could lower the actuating forces by more than 50 per cent. Whilst such reductions would appear very attractive, one should carefully balance the increased costs of providing the necessary air passageways through the hub to the blades, compressors and reliability, together with manufacturing costs against the cost of using a large hub size. Nevertheless, it remained true that the spindle torque could be reduced by using air ventilation and a paper by Van Gunsteren[16] provided a good introduction to the methods employed with this technique.

Rusetskiy[3] had produced a standard technique of deducing the deformation of the mean line of an aerofoil section at off-design pitch settings, however, his method did not include the effects of section thickness. In order to overcome this restriction the numerical analysis method of blade definition was developed, however, the analysis proposed by Rusetskiy was found to give a good first approximation to the zero lift angle.

The skew mentioned in Figs. 16 and 17 referred purely to the offset of the sections from the spindle axis and did not include any rake effects.

Mr. H. Woods raised a number of interesting points concerning power absorption and efficiency of c.p. propellers operating in off-design conditions.

On his initial point regarding revolutions margins, it was now accepted practice to design fixed pitch propellers to allow for fall-off in performance after the ship had entered service. Most manufacturers of diesel machinery took a realistic view of this inevitable situation, and many made firm recommendations as to the amount of initial margin to be allowed and the degree of overspeeding that was possible during the early service life of the ship. By proper discussion and understanding between shipbuilder, engine builder, shipowner and propeller designer at the design stage, it was usually possible to reach an acceptable solution. It was certainly not necessary to use a c.p. propeller to solve this problem alone.

However, Mr. Woods went on to the case of maintaining constant shaft rev/min to allow for electric power generation by shaft driven alternators. Since no such equipment seemed to exist which would tolerate more than very small variations in rev/min, then a c.p. propeller becomes necessary for this purpose.

With a c.p. propeller it was possible by adjusting the pitch, to maintain rev/min under varying conditions of load, whether these be caused by changes in weather conditions, degree of fouling or changes in draught. The propeller could only be designed to give optimum efficiency for one set of conditions, and the diameter, pitch ratio and radial pitch distribution were chosen for the conditions most likely to apply for the greater part of the intended service. In moving away from the designed conditions by changing pitch, not only was the radial pitch distribution changed, but also, and more importantly, the diameter and mean pitch ratio were no longer optimum for the new loading.

Mr. Woods suggested that the change in radial pitch distribution would lead to a rather dramatic loss in efficiency, but did not quantify this loss nor described how this had been estimated. It was possible, however, that particularly for the minor changes likely during normal operation, the drop in efficiency on this account alone could be small. Reference to the paper by Burrill and Young[17] would show the effect on propeller efficiency of widely differing radial pitch distributions. More important, of course, was the effect of pitch changes on the cavitation performance.

With regard to estimates of the characteristics of c.p. propellers at the design stage, it was agreed that these could be readily made available, and, as in the case of fixed pitch propellers mentioned earlier, it was important that all parties concerned should discuss the likely performance at an early stage of the design project.

The authors replied to Mr. P. F. C. Horne that during the cavitation tunnel tests conducted in a uniform stream with each of the c.p. propellers listed in Table I, the blade dynamometer measured the thrust, torque, centre of thrust, centre of torque force and the spindle torque. The values of centres of thrust and torque had been excluded from the

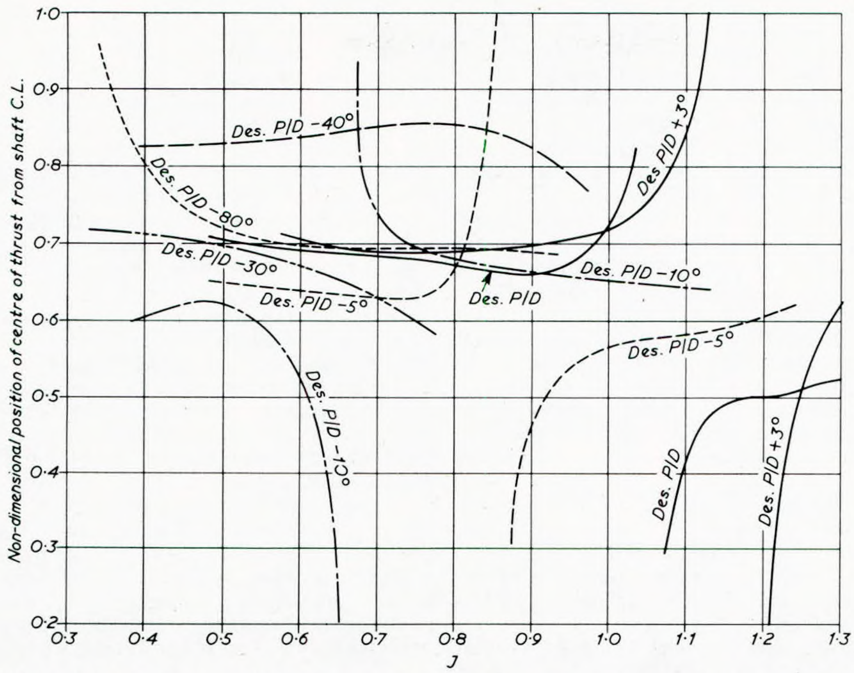

FIG. 27—*C.P. model propeller no. 120 — Curves of centre of thrust to a base J — Forward advance only*

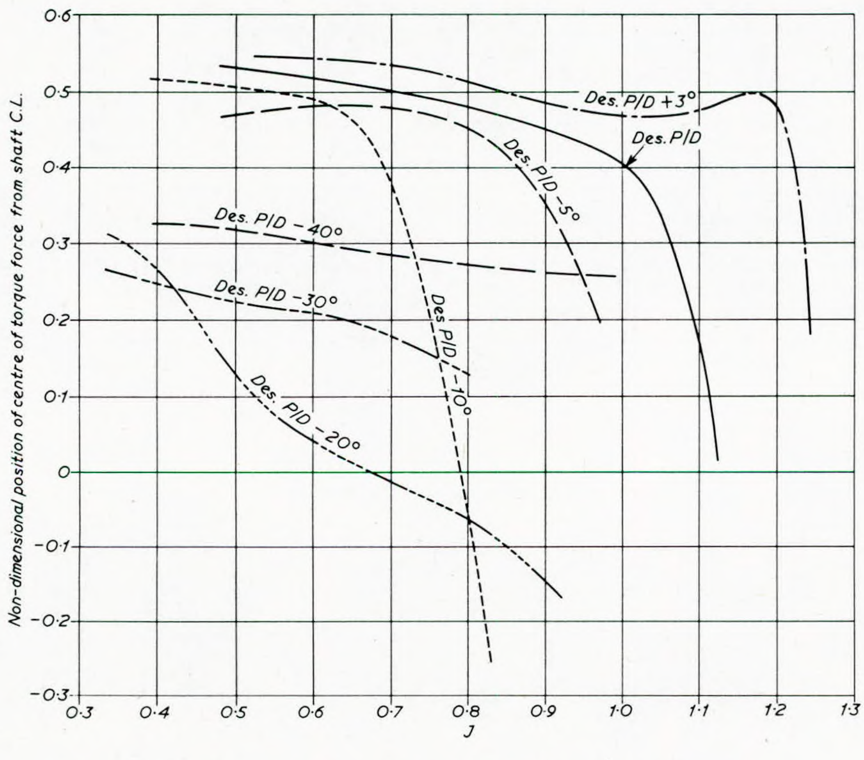

FIG. 28—*C.P. model propeller no. 120 — Curves of centre of torque force to a base of J CPT/R — Forward advance only.*

paper simply to reduce the overall cost of printing. The results obtained for c.p. model propeller no. 120 in design pitch were given as a typical example in Figs. 27 and 28.

Although the centres of thrust and torque were not measured during the variable stream tests conducted with c.p. model propeller no. 120, the authors could see no reason why in subsequent tests this information should not be obtainable.

So far the work undertaken had been restricted to the examination of the hydrodynamic and centrifugal forces applied to the propeller blade, correlation of bending moments with values determined by calculation or from full scale experiments had not yet been attempted. The authors, however, by assuming the root section profile of model 118 approximated to an elipse estimated that the shear stresses induced by the maximum spindle torque condition was of the order of 20 per cent of the normal tensile stresses in the blade.

With regard to the problem of assessing practical limits for the stressing of propellers operating in a variable wake stream the authors would refer Mr. Horne to a paper by Webb, Eames and Tuffrey[18].

With regard to the Chairman's remarks on the effects of propeller vibratory torque on the spindle torque, the authors had at the present time little information. Certainly in the case of the tests they had carried out to date this effect had not been examined because the University of Newcastle Cavitation Tunnel had a model propeller shaft drive of the variable speed hydraulic motor type which was not readily adaptable to the input of a vibratory torque.

It was, however, likely that the spindle torque would have a non-steady component dependent upon the torsional vibration characteristics of the system and the introduction of a variable wake field would no doubt augment these effects. In order to study these interesting effects one would need either a very advanced theoretical model which could cater for some of the more complicated flow regimes set up at off-design conditions or a cavitation tunnel capable of variable torque input, neither of which were available at the time this work was undertaken. Indeed a study of this type was also outside the initial scope of the research project as originally envisaged.

The authors would, however, agree that a line of research of the type Mr. McClimont suggested would prove a valuable addition to the knowledge of c.p. propeller practice.

Mr. T. P. O'Brien sought a simple alternative to the controllable pitch propeller and cited the two speed gearbox driving a fixed pitch propeller.

As he pointed out, this combination had been used in tugboats and trawlers and was believed to have given satisfactory service. Of course, these were ships which were called upon to operate under two sets of clearly defined conditions — towing (or trawling) and free running — when the application of a two speed gearbox and the choice of appropriate gear reduction ratios could readily be decided.

One was unlikely these days to find a shipowner adopting a controllable pitch propeller without good reasons, and they were usually for ease of manoeuvrability, including rapid reversing, or the application of gas turbine machinery which up to the present time had no simple reversing stage. All of these required infinite pitch control, for which the two speed gearbox would not provide a solution.

Regarding the similarity between models 117 and 118 it should be explained that the models were all replicas of actual propellers in service and the similarity in pitch ratio and blade area ratio was purely coincidental.

In reply to Dr. E. Glover, the authors said that when this particular programme of research was initiated the theoretical method of Boswell was known to give reasonably accurate results for propellers operating at the optimum design conditions. It was a natural progressive step to extend this existing procedure for the calculation of the blade loading at off-design conditions. At the same time the approach by Rusetskiy offered satisfactory results over a wider range of operation than that given by Boswell.

The sophisticated approach of Hess and Smith as suggested by Dr. Glover was not then available. Application of this method would obviate the need for empirical correction factors to the off-design section profile data, in particular that required for the correction of single aerofoil in two dimensional flow to that of the cascade flow in the propeller. The method, however, was relatively untried and required modification for the viscous effects of the boundary layer.

The authors understood that Dr. Glover was actively involved in the development of this approach and looked forward to its possible application in the field of c.p. propeller design and analysis.

The expressions for maximum blade width were based on the maximum chord at a given radius for the zero pitch condition and a correction was included for the change in radius by increasing the pitch to its design value. Consequently, the functions were largely independent of blade outline provided it formed a continuous and smooth curve. Similar expressions for other blade numbers were given below.

Three Blades

Max. Width $= D\,[1{\cdot}01\,x + 0{\cdot}05\,(P/D - 1) + 0{\cdot}055]$

Five Blades

Max. Width $= D\,[0{\cdot}623\,x + 0{\cdot}0125\,(P/D - 1) + 0{\cdot}010]$

ADDITIONAL REFERENCES

16) Van Gunsteren, L. A. 1968. "Reduction of Blade Spindle Torque by Ventilation." *I.S. Prog., Vol. 15.*
17) Burrill, L. C. 1953, "The Effect of Radial Pitch Variations on the Performance of a Marine Propeller." *Trans.R.I.N.A.*, Vol. 95, p. 437.
18) Webb, A. W. O., Eames, C. F. W., and Tuffrey, A. 1975. "Factors Affecting Design Stresses in Marine Propellers". Paper No. 9. *S.N.A.M.E.* Propellers 75 Conference.