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Paper for written discussion

ON DAMPING OF TORSIONAL VIBRATION IN A PROPULSION SYSTEM HAVING A FLUID DRIVE

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SUMMARY

The magnitude of serious torsional vibration in a ship diesel propulsion system is limited by damping. It is shown that the effects of the damping can be radically different, depending on whether or not that damping couples the principal modes. It is common, perhaps usual, to ignore coupling but it may not be safe to do so - particularly if a fluid coupling is used in the drive.

INTRODUCTION

It is well known that resonant torsional vibration in a ship's propulsion system is limited by damping. Accordingly precautions are taken at the design stage to ensure that either resonance will not occur under operating conditions or, if it does, there is adequate damping to render it harmless.

Generally speaking there are two approaches available for making the necessary checks.

(a) "Critical speeds" and principal modes may be calculated and overall modal damping factors applied to the resonant conditions at such of those speeds as are relevant. This is the "classical" concept of modal analysis and it is quite sufficient to meet the practical needs for most systems. Indeed one variant of it is the only technique discussed by den Hartog (1956)(3), perhaps the best known specialist in this field, while another forms the basis of the Guidance Notes on the subject in Lloyd's Register of Shipping's Rules and Regulations.

(b) Alternatively an attempt may be made to "distribute" the damping as it is thought to occur. Unfortunately the distribution of the damping is notoriously difficult to specify with any confidence, as den Hartog (1956)(3) points out. With a given distributed damping, the analyst may proceed to estimate steady oscillation of any desired order.

The main purpose of this paper is to demonstrate, by reference to a particular diesel propulsion system which actually failed in service, that when a fluid drive is used there is really no question of choice. The second of the two approaches must be used since the first may lead to optimistic false conclusions. A second purpose is to present the theoretical argument in a concise way so as clearly to show the relationship between the two approaches; for the writers are unable to cite any paper or book in which it is shown how to proceed from a knowledge of the principal modes and natural frequencies of a multi-cylinder engine system to the calculation of responses at any prescribed speed of given orders.

THE PROPULSION SYSTEM

It will be helpful to illustrate the theory by means of an actual system. Fig. 1 shows a marine propulsion system which embodies a fluid coupling (or "fluid flywheel"). The failure of coupling bolts suggests that excessive torque fluctuations occurred in the stub shaft connected to the engine side of the coupling when the engine ran at its normal operating speed.

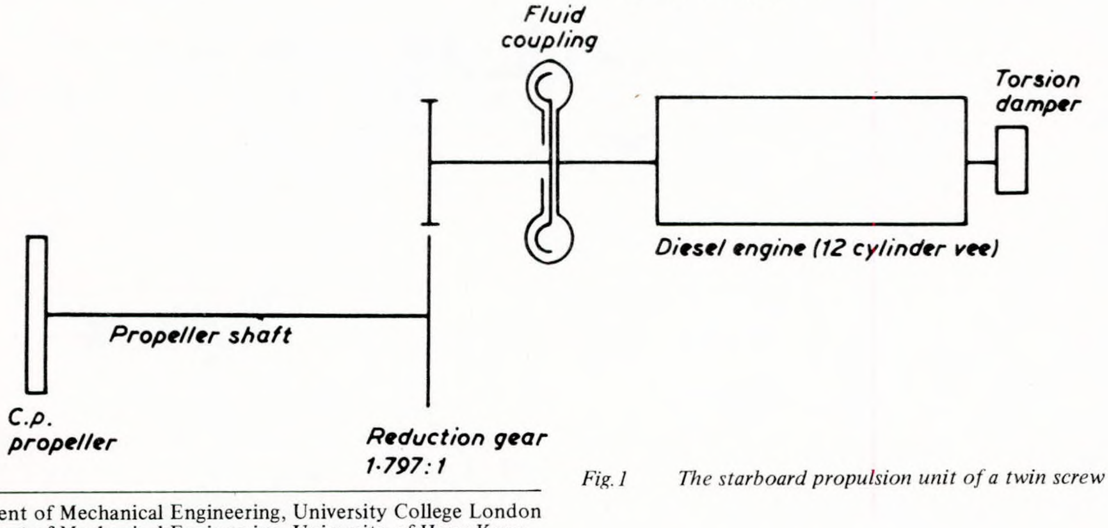

Fig. 1 *The starboard propulsion unit of a twin screw ship*

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A number of authorities suggest that torsional analysis may legitimately be performed for that part of the system on one side of the coupling, as if the system were severed at the fluid in the coupling (e.g. see Tuplin 1934[5], den Hartog 1956[3] and Nestorides 1958[4]). The theory presented in Appendix A is offered as general support for this conclusion.

According to Tuplin (1934)[5]:

(a) very little vibration is transmitted across a fluid coupling:
(b) the amplitudes of relative motion across the coupling vary roughly as the inverse of the frequency;
(c) for most purposes the system can be said to act as if it is completely severed at the coupling;
(d) to investigate torsional vibration of the system on one side of the coupling the other side may be treated as if it has a constant velocity:
(e) under the assumption (d), the side of interest may be assumed to be subject to damping that is proportional to the vibration velocity.

These observations are entirely compatible with those of den Hartog (1956)[3] and Nestorides (1958)[4]. In fact, den Hartog derives a constant of proportionality for the relationship referred to in (e).

The assumption (c) will be made here and, as suggested in Appendix A, the natural frequencies and principal modes will be sought for the free-free system comprising discs A to K inclusive along with their connecting massless shafts. The system under discussion is therefore as represented in Fig.2, the relevant mechanical constants being those given in Table 1.

Rotations of the discs are represented by the matrix

$$\underset{\sim}{x}=\{x_1, x_2, \ldots\ldots, x_{11}\}$$

and we are interested in the torque amplitude $I_1\,\omega^2 X_1$ where I_1 ($=1293$ kg m^2) is the polar moment of inertia of the input side of the coupling, ω is the frequency of vibration in rad/s and X_1 is the amplitude of the oscillatory component of x_1.

Further details of the system and its operating conditions are given in Table II.

TABLE II - DETAILS OF OPERATING CONDITIONS

Engine continuous service rating : 3.486 MW (brake) at 440 rev/min
Maximum continuous rating : 4.101 MW (brake) at 465 rev/min
Propeller service speed : 237 rev/min
Propeller maximum speed : 250 rev/min
Engine idling speed : 250 rev/min
Indicated mean effective pressure : 1.489 MN/m^2
Vee-angle of engine : 45°
No. cylinders : 12
Firing order : 1,3,5,6,4,2
Crankshaft arrangement : see Fig. 3
Firing interval between banks for corresponding cylinders : 45° of crankshaft rotation
$m_{recip.}$ equivalent reciprocating mass/cylinder : 228 kg
A, cylinder area : 0.126 m^2
ℓ/r : 4.13
stroke : 460mm

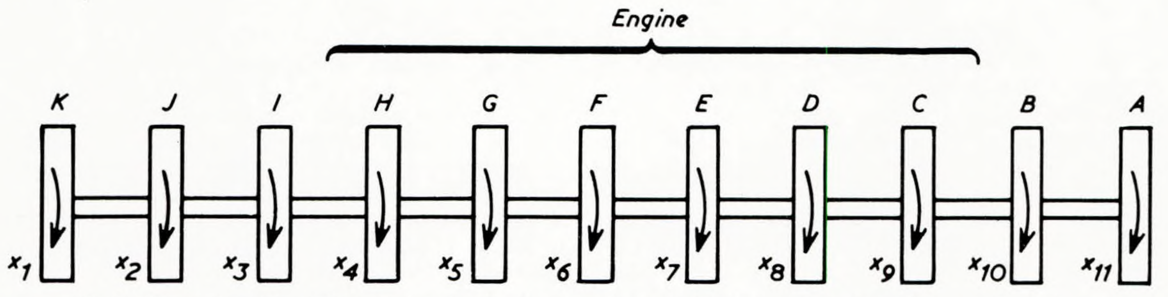

Fig.2 The engine side of the complete torsional system shown in Fig.1. (For the values of the mechanical parameters see Table 1.)

TABLE 1 - DATA FOR THE TORSIONAL SYSTEM SHOWN IN FIG.2.

Station	Item	Coordinate	Moment of inertia kg.m^2	Damping constant Nms	Stiffness Nm/rad
A	Damper outer	x_{11}	64	9,355	1.55×10^6
B	Damper inner	x_{10}	18		101×10^6
C	Cyl. 1	x_9	95	778	77×10^6
D	Cyl. 2	x_8	95	778	77×10^6
E	Cyl. 3	x_7	95	778	77×10^6
F	Cyl. 4	x_6	95	778	77×10^6
G	Cyl. 5	x_5	95	778	77×10^6
H	Cyl. 6	x_4	95	778	98×10^6
I	Cam gear train	x_3	91		222×10^6
J	Flanges	x_2	26		48×10^6
K	Fluid coupling input	x_1	1293	1,480	

STEADY FORCED TORSIONAL VIBRATION

It is convenient, first, to adapt certain general results of linear vibration theory to torsional oscillation of the system under discussion. An equation may be written down for the torsional motion. It is

$$\underset{\sim}{A}\ddot{\underset{\sim}{x}} + \underset{\sim}{B}\dot{\underset{\sim}{x}} + \underset{\sim}{C}\underset{\sim}{x} = \underset{\sim}{\Phi}_n e^{in\Omega t}, \qquad (1)$$

where $\underset{\sim}{A}$ $\underset{\sim}{B}$, $\underset{\sim}{C}$ are respectively the inertia, damping and stiffness matrices, Ω is the engine speed in rad/s, n is the order of vibration and $\underset{\sim}{\Phi}_n$ represents the matrix of generalised excitation amplitudes at the coordinates $\underset{\sim}{x}$. For a 2-stroke engine, n=1,2,3,4... and for a 4-stroke engine n=½,1,1½,2....The steady forced response is evidently

$$\underset{\sim}{x} = (\underset{\sim}{C} - n^2\Omega^2\underset{\sim}{A} + in\Omega\underset{\sim}{B})^{-1}\,\underset{\sim}{\Phi}_n e^{in\Omega t} \qquad (2)$$

The column vector $\underset{\sim}{\Phi}_n$ may be expressed as

$$\underset{\sim}{\Phi}_n = M_n\underset{\sim}{\phi}_n,$$

where M_n is the amplitude of excitation at a single cylinder or row of cylinders and $\underset{\sim}{\phi}_n$ is a complex vector representing the phase differences between excitations at the various cylinders. The quantity M_n contains a Vee-factor to account for the two cylinders of a common row if required and, for n=1,2 and 3, it must contain a contribution from reciprocating inertia. For the engine under discussion, whose crankshaft arrangement is that of Fig.3,

$$\phi_3 = \begin{bmatrix} 0 \\ 0 \\ 0 \\ 1 \\ -\tfrac{1}{2}-i\sqrt{3}/2 \\ -\tfrac{1}{2}+i\sqrt{3}/2 \\ -\tfrac{1}{2}+i\sqrt{3}/2 \\ -\tfrac{1}{2}-i\sqrt{3}/2 \\ 1 \\ 0 \\ 0 \end{bmatrix}$$

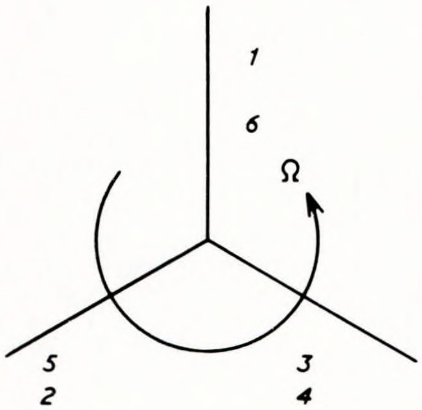

Fig.3. The throws of the crankshaft. The numbers relate to the cylinders (number 1 being the furthest forward in the ship) and the firing order is 1,3,5,6,4,2.

The system of Fig.2 has 11 natural frequencies $\omega_0, \omega_0, \ldots, \omega_{10}$. The corresponding principal modes are $X^{(0)}, X^{(1)}, \ldots X^{(10)}$, the first being the "rigid body" mode for which $\omega_0 = 0$. Thus a square modal matrix X of order 11 may be defined whose columns are the principal modes. (The elements of the first column will all be the same number.) Having found X we can change over from the physical coordinates x to principal coordinates

$$p = \{p_0, p_1, \ldots\ldots, p_{10}\}$$

in the conventional manner, since

$$x = Xp \tag{3}$$

Let

$$Z = C - n^2\Omega^2 A + in\Omega B$$

so that the receptance in equation (2) is

$$\alpha = Z^{-1},$$

and so

$$x = \alpha \Phi_n e^{in\Omega t}. \tag{4}$$

According to elementary matrix theory, if

$$X^T \alpha^{-1} X = X^T Z X$$

then

$$X^{-1} \alpha (X^T)^{-1} = (X^T Z X)^{-1}$$

and so

$$\alpha = X(X^T Z X)^{-1} X^T \tag{5}$$

where the superscript T means "transposed".

The bracketed quantity in equation (5) is the square matrix

$$N - n^2\Omega^2 L + in\Omega M$$

where L, M and N are respectively the inertia, damping and stiffness matrices corresponding to the principal coordinates. Thus equations (2) and (3) show that

$$p = (N - n^2\Omega^2 L + in\Omega M)^{-1} X^T \phi_n M_n e^{in\Omega t}. \tag{6}$$

The theory that we have adapted to torsional vibration is, of course, very well known and easily accessible in textbooks: see for example Bishop, Gladwell and Michaelson (1965)[1]. In particular, it will be recognised that the matrices L and N are both diagonal and, in our example, of order 11.

RESONANT TORSIONAL VIBRATION IN PRINCIPAL MODES

It is common to assume that, not only L and N are diagonal, but also M. This assumption is implicit in the first of the two approaches mentioned in the Introduction. When this is the case the responses at the principal coordinates are uncoupled. Resonance occurs in the r^{th} mode if

$$n\Omega = \omega_r, \tag{7}$$

where ω_r is the r^{th} natural frequency (r=1,2,....10). The resonant condition may be assumed to dominate the response when it occurs and so it is examined for each of the critical speeds.

Suppose that resonance occurs of the n^{th} order in the r^{th} mode, so that only the response p_r is of major interest. Equation (6) reduces to

$$p_r = (c_r - n^2\Omega^2 a_r + in\Omega b_r)^{-1} X^{(r)T} \phi_n M_n e^{in\Omega t} \tag{8}$$

since

$$L = \text{diag}\ \{a_0, a_1, \ldots\ldots\ldots, a_{10}\}$$

$$N = \text{diag}\ \{0, c_1, c_2, \ldots\ldots, c_{10}\}$$

and, here,

$$M = \text{diag}\ \{0, b_1, b_2, \ldots\ldots, b_{10}\}$$

with

$$\omega_r^2 a_r = c_r.$$

At resonance, then,

$$p_r = \frac{X^{(r)T} \phi_n M_n e^{in\Omega t}}{i\omega_r b_r}. \tag{9}$$

The corresponding amplitude at any coordinate of interest (x_j say) is thus

$$X_j = \frac{X_j^{(r)} \underset{\sim}{X}^{(r)T} \underset{\sim}{\phi}_n M_n}{\omega_r b_r} \tag{10}$$

whence the resonant behaviour can be investigated and potentially excessive responses can be identified.

It is of interest to compare this result with the Guidance Notes issued by Lloyd's Register of Shipping. The Notes require that the modes be scaled in such a way that, at engine cylinder 1 with rotation x_a, $X_a^{(r)} = 1$, for r =0,1,2 ...10. Now

$$\frac{X_j^{(r)} \underset{\sim}{X}^{(r)T} \underset{\sim}{\phi}_n M_n}{c_r}$$

is the appropriate static response at x_j when there is no dynamic magnification. Hence the dynamic multiplier or Q-factor is

$$Q_r = \frac{c_r}{\omega_r b_r} \quad \text{or} \quad \frac{a_r \omega_r^2}{\omega_r b_r}$$

and

$$X_j = Q_r \frac{X_j^{(r)} \underset{\sim}{X}^{(r)T} \underset{\sim}{\phi}_n M_n}{a_r \omega_r^2}$$

This is the result that would be found by following the Lloyd's Register Notes and, further, the Notes contain empirical rules for arriving at the value of Q_r. Unfortunately, it appears that those rules cannot reasonably be applied directly to the system of Fig.1 as they do not cater for a fluid coupling.

The lowest three non-zero natural frequencies of the system are

$$\omega_1 = 144.5 \text{ rad/s or } 1380 \text{ c/m},$$

$$\omega_2 = 217.5 \text{ rad/s or } 2077 \text{ c/m},$$

$$\omega_3 = 506.2 \text{ rad/s or } 4834 \text{ c/m}.$$

If, then, we were interested in dangerous vibration over a speed range of, say, 300 rev/min - 465 rev/min, we might reasonably wish to investigate the criticals listed in Table III. Of these one would surmise that the worst cases might be those of 346 and 460 rev/min, for the vector sums $\underset{\sim}{X}^{(r)T} \underset{\sim}{\phi}_n$ in an Argand diagram are then additive.

TABLE III - POSSIBLE CRITICAL SPEEDS

Ω	307	319	345	346	378	394	403	415	460	462
r	1	2	1	2	2	1	3	2	1	2
n	4	6½	4	6	5½	3½	12	5	3	4½

Without being able to do more than guess the Q-factors we should perhaps expect a response with perceptible peaks at the speeds listed in Table III. As a check the 3rd order resonance in mode 1 was computed assuming a Q-factor of 20. The amplitude of torque fluctuations in the stub shaft was found to be about 52 KNm. Even with this light damping there is no prediction of anything approaching torque reversal, since the mean torque was around 76 KNm.

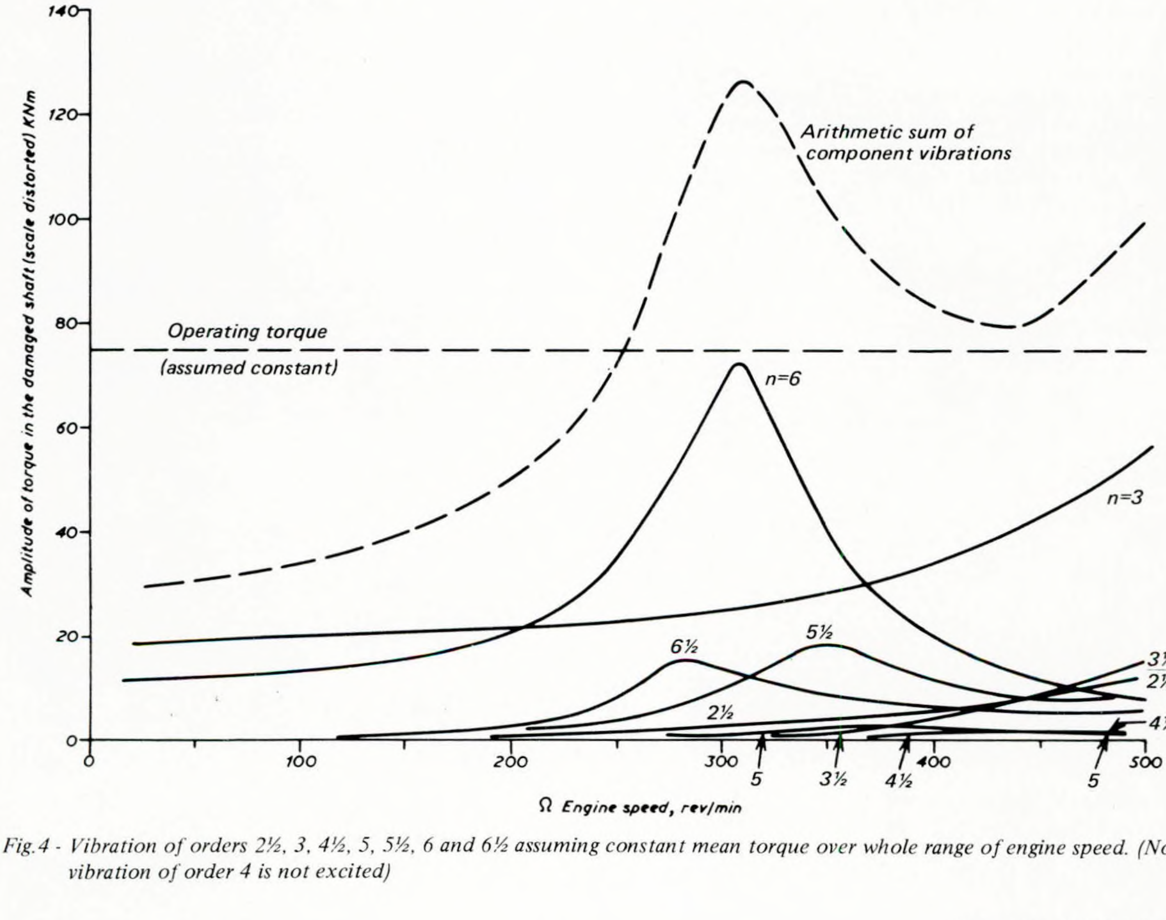

Fig.4 - Vibration of orders 2½, 3, 4½, 5, 5½, 6 and 6½ assuming constant mean torque over whole range of engine speed. (Note vibration of order 4 is not excited)

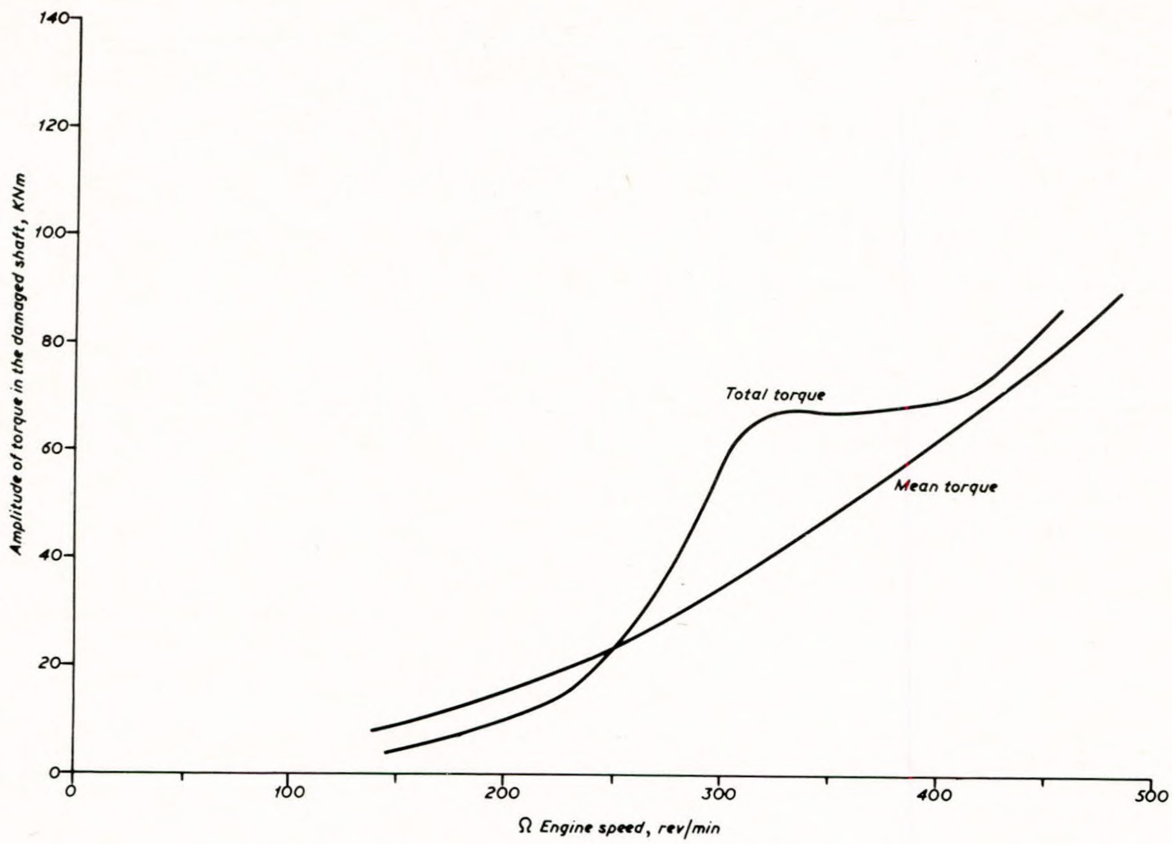

Fig.5 - Approximate curves of mean torque and fluctuating torque

MEASURED VALUES

When the torque fluctuation in the stub shaft connected to the input side of the fluid coupling was measured it was found that

(i) torque reversal occurred over a substantial range of engine speed (300-430 rev/min);

(ii) there was a peak of 6th order vibration at about 324 rev/min.

Neither feature could possibly be said to confirm the predictions that might be made on the basis of Table III.

COUPLING OF THE PRINCIPAL MODES

If the analysis is performed by means of equation (2) the result is quite different and gives predictions that are much closer to the measured results. The data in Table 1 permit the matrices $\underset{\sim}{A}$, $\underset{\sim}{B}$ and $\underset{\sim}{C}$ to be formed and we can proceed directly to use equation (2) to find $\underset{\sim}{x}$, and hence X_j.

Alternatively one can use the principal coordinates. From $\underset{\sim}{A}$ and $\underset{\sim}{C}$, the principal modes may be found and hence $\underset{\sim}{X}$ formed. Then the matrices

$$\underset{\sim}{L} = \underset{\sim}{X}^T \underset{\sim}{A}\underset{\sim}{X}, \qquad \underset{\sim}{M} = \underset{\sim}{X}^T \underset{\sim}{B}\underset{\sim}{X}, \qquad \underset{\sim}{N} = \underset{\sim}{X}^T \underset{\sim}{C}\underset{\sim}{X}$$

can be computed. But when this was done, $\underset{\sim}{M}$ turned out to be very far from diagonal because of the presence of the fluid coupling at one end of the system and so damping couples the modes closely. As a check, equations (3) and (6) can now be used to recalculate $\underset{\sim}{x}$ and, hence, X_j and the torque amplitude. As one would expect the results were found to be identical to those found by the direct route. The results found by these two methods in which coupling is admitted, are shown in Fig.4.

In arriving at Fig.4, an admittedly gross assumption has been made, namely that the mean torque at 440 rev/min is the same over the whole speed range. It is pointed out by den Hartog[3] on page 200 of his book, however, that a fluctuating disturbance retains approximately the same percentage of the mean torque over virtually the whole of its speed range. If, then, the mean torque curve of Fig.4 is given the more realistic form shown in Fig. 5, the disturbance becomes that shown in Fig. 5. In all essential details this agrees with observations made on the system in service.

CONCLUSIONS

It is not safe to investigate the torsional vibration characteristics of a propulsion system which embodies a fluid coupling, using the well-established procedures involving consideration of "critical speeds". Being placed, in effect, at one end of a separately disturbed vibrating system and being relatively heavily damped, the coupling closely couples the principal modes. It is advisable then to return to first principles when estimating responses at different orders. If, however, the data needed for analysis in terms of critical speeds are available, they can be employed in an extraction of the required data.

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APPENDIX
RESONANCE OF A SYSTEM CONTAINING A FLUID COUPLING

It is shown by den Hartog (1956)[3] that the fluid coupling acts as a simple viscous damper. Let the damping constant be b. We can now identify three subsystems:

B; the torsional system extending from the propeller up to and including the output side of the coupling,
C; the massless damper across the coupling,
D; the remaining torsional system extending from and including the input side of the coupling.

These subsystems are simply linked at rotations q_1 and q_2 as indicated in Fig.6.

Compatability of the subsystems requires that

$$\Psi_{1b} = \Psi_{1c} = \Psi_1, \text{ say},$$
$$\Psi_{2c} = \Psi_{2d} = \Psi_2, \text{ say},$$
$$\Psi_{4c} = \Psi, \text{ say}.$$

Further, equilibrium requires that

$$\Phi_{1b} + \Phi_{1c} = 0,$$
$$\Phi_{1c} + \Phi_{2c} = 0,$$
$$\Phi_{2c} + \Phi_{2d} = 0,$$
$$\Phi_{3d} = \Phi, \text{ say}.$$

If all these equations are suitably combined it is found that

$$\Psi = \alpha_{43}\,\Phi = \left\{ \delta_{43} - \frac{\delta_{42}\delta_{23}}{\beta_{11} + \delta_{22} + \frac{1}{i\omega b}} \right\} \Phi$$

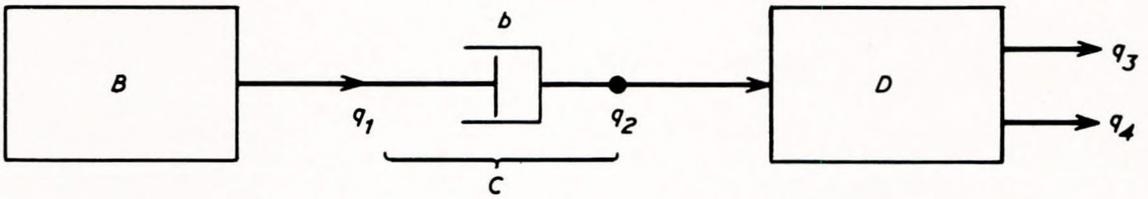

Fig 6 - The torsional system represented as three sub-systems. B is the driven side and D is the driving side. C is the massless damper which represents the effect of the fluid coupling.

Since we are concerned with the possibility of resonant conditions within subsystem D, let D be excited at q_3 and consider a general coordinate q_4, also in D. What are the conditions under which D resonates?

Using the notation of Bishop and Johnson (1960)[2], we have

$$\Psi_{1b} = \beta_{11}\Phi_{1b},$$
$$i\omega b(\Psi_{1c} - \Psi_{2c}) = \Phi_{1c},$$
$$\Psi_{2c} = \delta_{22}\Phi_{2d} + \delta_{23}\Phi_{3d},$$
$$\Psi_{4c} = \delta_{42}\Phi_{2d} + \delta_{43}\Phi_{3d},$$

The quantities β are receptances of B while the δ are receptances of D. The Ψ are amplitudes of the generalised coordinates q and the Φ are the amplitudes of the corresponding generalised forces.

where α_{43} is the cross receptance between q_4 and q_3 of the composite system. Suppose that B and D are devoid of damping so that all the receptances β and δ are real functions of ω. The quantity

$$\beta_{11} + \delta_{22} + \frac{1}{i\omega b}$$

cannot be set equal to zero as is usual with simple systems because to do so would require both the real and imaginary parts to vanish separately - which is impossible.

The conclusion is that the response at q_4 will become infinite only when δ_{43} becomes infinite. This, however, is the condition that is fulfilled when the *unattached* subsystem D is excited at resonance.

Discussion

DR I J BICKLEY (Mirrlees Blackstone (Stockport) Ltd) commented on the paper in two respects; a) the original coupling bolt failure, and b) the adequacy of current torsional vibration calculation methods to predict the torsional vibration behaviour of propulsion systems with a fluid drive.

With regard to a) it should firstly be appreciated that this system had no flywheel and that fluid coupling input had been used to serve the purpose of a flywheel. It was to be expected, therefore, that the vibratory torque levels in the stub shaft for this system would be similar to those expected in conventional arrangements utilizing a flywheel and with the bolting system designed accordingly.

It had been stated that the torque fluctuation in the stub shaft was such that torque reversal over a substantial speed range occurred; but this was quite usual for medium speed diesel engines with 12 cylinders or less. It was not customary to calculate the vibratory torque conditions for a flywheel/crankshaft bolting arrangement because experience had shown that it was not necessary with a properly designed bolting arrangement (using fitted bolts, or similar). In the instance described it would appear that a totally inadequate bolting arrangement had been used.

With regard to b) the purport of the paper seemed to be that, with fluid drive systems, the classical normal mode calculation methods no longer sufficed to ensure that systems would perform satisfactorily in service. This had been appreciated for a long time now and references to the necessity of forced frequency calculation methods had appeared in the Lloyd's Register of Shipping. Rules since 1969 (currently Part 5 Chapter 8 Section 7 Item 2). Nearly all engine builders, coupling, damper and gearbox manufacturers had their own forced vibration calculation programs, as had the classification societies and the CAD Centre's TORVAP A.

All these programs would solve the authors' equation (2) and so would be capable of producing the results shown in the authors' Figs 4 and 5.

Dr Bickley's company, for example, had used forced vibration methods of calculation for the past ten years and now all systems were investigated using these methods to determine stresses and vibratory torques throughout the system under differing conditions of engine balance.

It should also be pointed out that whereas the authors had quoted a torque in the subshaft of 52 kNm at 460 rev/min for the 3rd order one node using classical calculation methods; and stated that this gave no prediction of anything approaching torque reversal, their own Fig 4 showed a vibratory torque in the stubshaft of about 44 kNm which was less than the classical methods. Were the vibratory torque for the orders shown in Fig 4 to be calculated (at 460 rev/min) using classical methods, and added arithmetically, the total would be about 87 kNm, i.e. higher than the 83 kNm shown in Fig 4.

This implied that no calculations, rather than the wrong type of calculations, had been carried out, and in no way substantiated the claim that classical methods had led to optimistic false conclusions.

By using the data supplied by the authors, and assuming a minimum shaft diameter of 285mm and resultant harmonic components of tangential effort, or Tm values, from Mirrlees Blackstone's K major engine frequency and forced vibration calculations, results comparable to the authors' Fig 4 and 5 had been obtained.

The natural frequencies were:

1st	=	1382	vpm
2nd	=	2079	vpm
3rd	=	4840	vpm

which differed about 0.1% from the authors' figures. The forced vibration results were shown as Figs A and B. These compared with the authors' Figs 4 and 5, respectively.

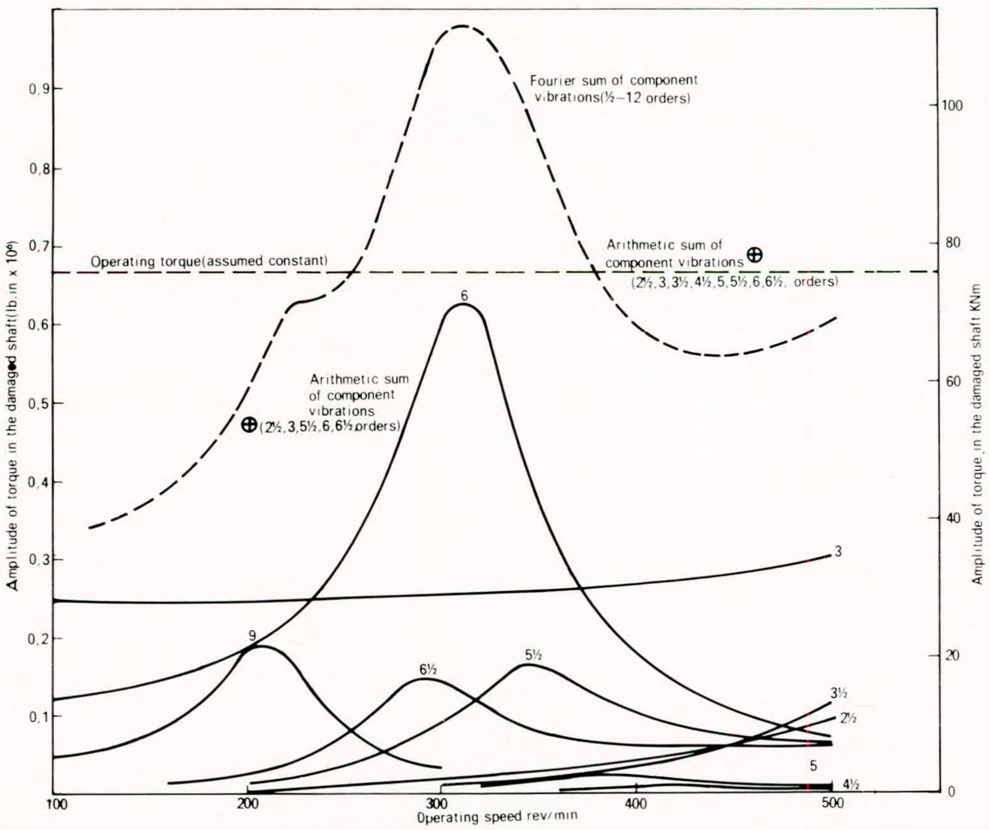

FIG A *Vibration of orders 2½, 3, 4½, 5, 5½, 6 and 6½ assuming constant mean torque over the whole range of engine speed.*

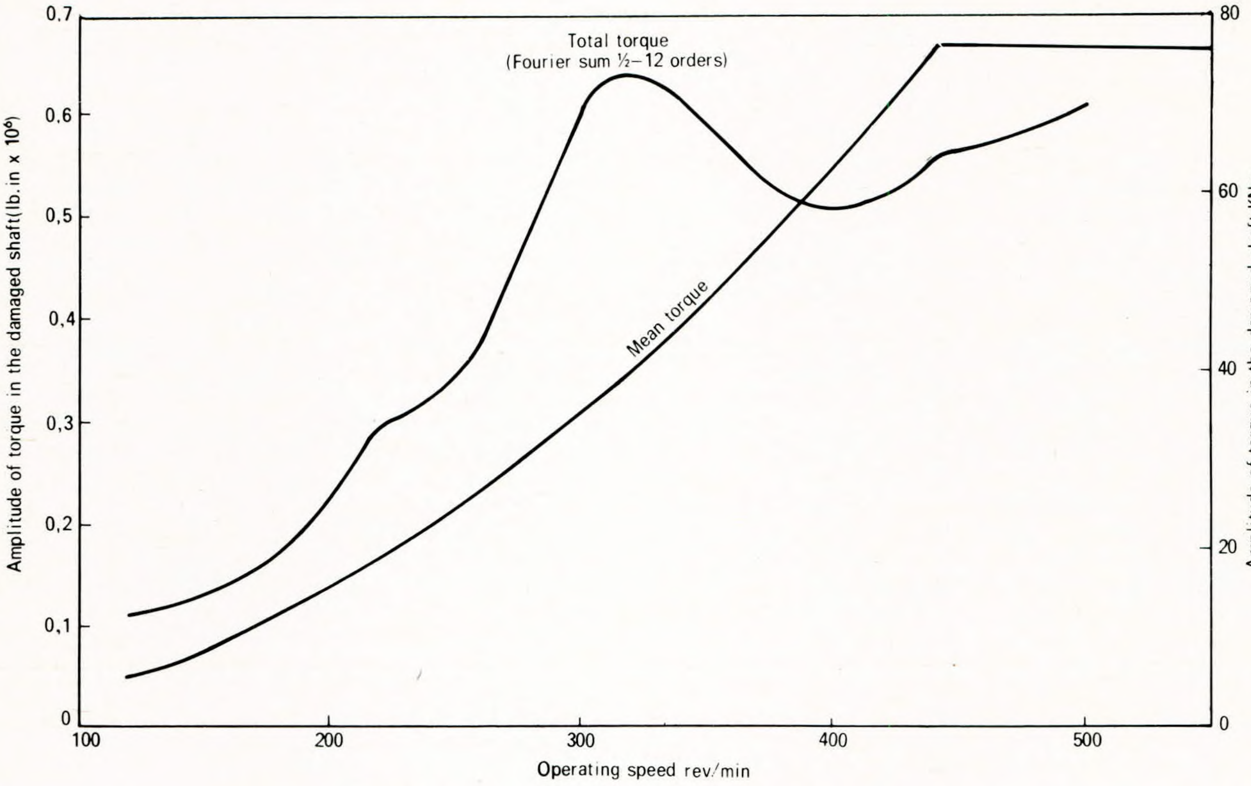

FIG B *Curves of mean torque and fluctuating torque.*

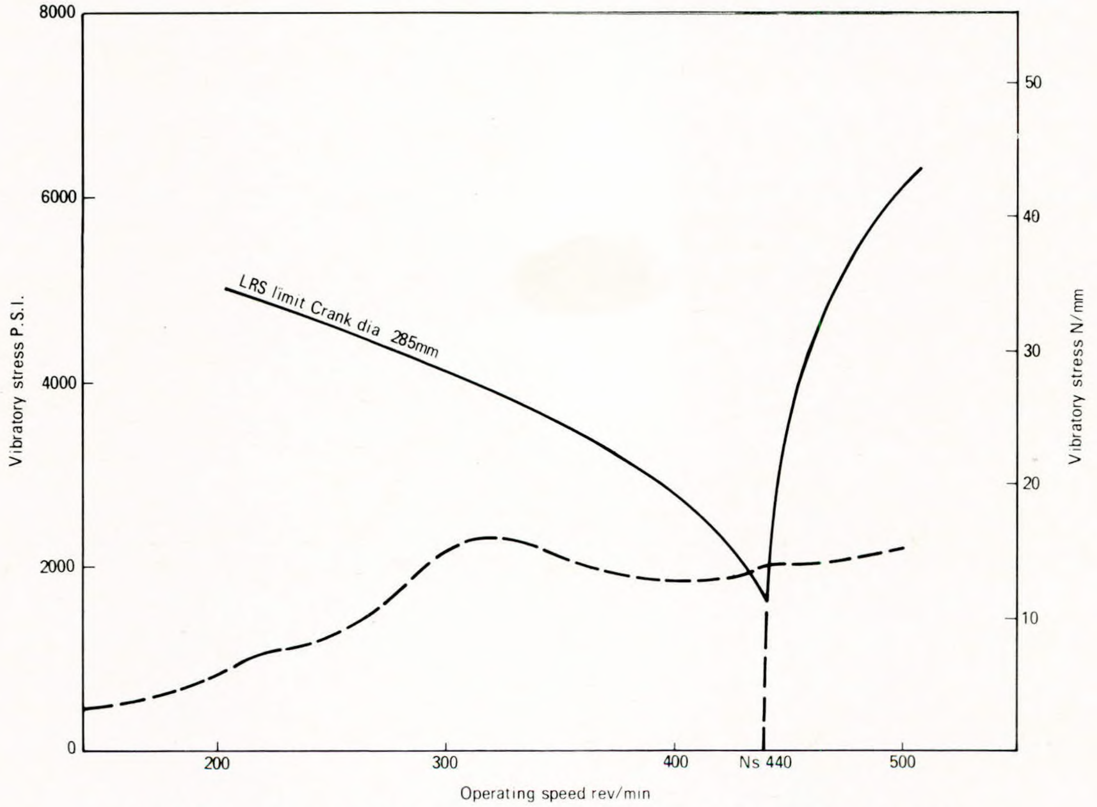

FIG C *Fourier summated torsional stress (½ –12 orders) in damaged shaft assuming propeller power law and shaft minimum dia. of 285mm.*

Comparing the forced vibration results showed generally similar vibratory torque levels, especially the 6th peak which was just over 70 kNm in both Figs 4 and A and about 70 kNm (total stress) in both Figs 5 and B. The differences were that the Mirrlees Blackstone results were lower than the authors' above the 6th order peak, and higher below the 6th order peak. The results were based on 24 harmonic orders (½ - 12) and the summated levels of vibratory torque included the effect of phase angles.

Using only the orders plotted in Fig 4, an arithmetic addition of vibratory torques calculated by the company's forced vibration program gave 78 kNm at 460 rev/min and 54 kNm at 200 rev/min (Fig 4 gave figures of 83 kNm and 50 kNm, respectively).

The authors' simpler approach to torque summations, together with slightly different Tm values, particularly the 3rd, probably explained the slightly different results.

Fig C gave the total stress in the stub shaft and compared it with the Lloyd's Register of Shipping limit for crankshafts (based on 285mm dia). The stress was modest and gave no cause for concern.

To sum up then, current methods of torsional vibration calculations predicted the authors' results. The bolts probably failed due to faulty design, not due to misapplication of torsional vibration calculations and the fact that the authors "are unable to cite any paper or book in which it is shown how to proceed from a knowledge of the principal modes and natural frequencies of a multi-cylinder engine system to the calculation or responses at any prescribed speed of given orders", was because, when using forced vibration calculation methods, it was not necessary to do so.

D McKINLAY (Lloyd's Register of Shipping) said that the authors had presented an interesting and provocative paper but two of their conclusions appeared to be questionable.

They had claimed that their calculations and measurements showed that excessive vibratory torque existed in the stubshaft and that was the reason for failure of coupling bolts. Subsequently they felt that the methods of calculation described in the Rules of Lloyd's Register of Shipping were inadequate.

The torque fluctuations in the stubshaft calculated by the authors and said to be substantiated by measurement were not unusual in practice. He recommended that they look further for the reason for the failure of the coupling bolts.

The authors had stated that the Rules of Lloyd's Register could not reasonably be applied directly to the system of Fig 1 as they did not cater for a fluid coupling. The authors should refer to Part 5 Chapt 8, section 7.2 of the Rules which referred to vibration of systems requiring complex and detailed analytical treatment. Experience had shown that for forced damped vibrations, tabulation methods were preferred since these automatically took care of all modes of vibration, dealt with excitation and damping in detail, and gave results close to measured values. Further, vectorial synthesis of the component harmonic order of vibration was necessary; not the arithmetic analysis used in the paper. Indeed, calculations and measurements from a system apparently identical to that chosen by the authors showed reasonable agreement.

Lloyd's Register of Shipping had introduced torsional vibration considerations into classification requirements in 1944 and since that time had examined the characteristics of all systems in all ships classed by the Society. These examinations were not confined to calculation but had been confirmed by many shipboard measurements and it could be claimed that the Rules were a reflection of experience gained within the Industry.

It was of interest to study the authors' example further. Perhaps they would care to comment on why they had chosen to omit the gearing and propeller from the calculations. Consideration of the complete system indicated that the fluid coupling and the propeller did not significantly contribute to the damping of the crankshaft. Effectively the input member of the fluid coupling became the undamped flywheel of a simple engine system.

Table A1 showed the modal mass (L) modal stiffness (N) and critical damping for the first four modes, the figures being derived by matrix methods as adopted by the authors and based on the mass-elastic system given in the paper.

Inspection of these normal modes revealed substantial damper motion; the fundamental being its natural frequency (Table A2).

Taking into account the damping values detailed in the paper the first four damped natural frequencies, together with the damping ratios (C/C_c) and Q-factors ($\frac{1}{2}C/C_c$) appeared as shown in Table A3.

The fundamental mode was shown to be extremely heavily damped; a not unexpected result, but one which contrasted markedly with the authors' assumption of a Q-factor of 20.

TABLE AI – NATURAL FREQUENCY ANALYSIS

MODE No	FREQUENCY Hz	MODAL MASS Kgm^2	MODAL STIFFNESS MNm/rad	CRITICAL DAMPING kNmS
1	23.00	75.47	1.57	21.8
2	34.61	537.47	25.42	233.8
3	80.56	422.28	108.21	427.5
4	131.12	406.03	275.59	669.0

TABLE A2 – NORMAL USE

Mode No.		1	2	3	4
Damper Outer	X11	1.00	1.00	.10	.03
Damper Inner	X10	.14	-.95	-.97	-.96
Cyl. 1	X9	.12	-.97	-.94	-.86
Cyl. 2	X8	.10	-.95	-.61	-.01
Cyl. 3	X7	.08	-.86	-.08	.85
Cyl. 4	X6	.05	-.73	.47	1.00
Cyl. 5	X5	.03	-.55	.87	.31
Cyl. 6	X4	-.01	-.34	1.00	-.64
Cam gear train	X3	-.02	-.17	.85	-.97
Flanges	X2	-.03	-.08	.70	-.84
Fluid Coupling Input	X1	-.08	.03	-.12	.05

TABLE A3 – DAMPED NATURAL FREQUENCY ANALYSIS – DAMPER IN OPERATION

MODE No.	FREQUENCY Hz	C/C_c	Q
1	23.76	.443	1.1
2	31.06	.098	5
3	79.98	.029	17
4	130.81	.017	29

TABLE A4 -- DAMPED NATURAL FREQUENCY ANALYSIS – DAMPER NOT WORKING

MODE No.	FREQUENCY Hz	C/C_c	Q
1	23.00	.002	300
2	34.60	.012	40
3	80.56	.006	84
4	131.12	.003	144

The results of the calculation based on the assumption that the damper was not working were given in Table A4. It could be seen that the damping from the cylinders and fluid coupling was so weak that the modes were to all intents and purposes uncoupled and the frequencies corresponded to the undamped values. This confirmed that the damper at the forward end of the engine had the major effect.

Mr KcKinley concluded by saying that this contribution had been produced in collaboration with other members of the department of Lloyd's which dealt with vibration analysis. They hoped that the authors would correct the error in Table III in the final text.

MR R S KEYS, MA (Department of Marine Engineering, Southampton College of Higher Education) wrote that the authors had given consideration to an engine closely coupled to a fluid coupling in which the damping effects might be considered to be purely viscous. They had shown that the effects of the coupling of principal modes resulting from this damping rendered the more simplistic approach adopted in Lloyd's Register of Shipping's Rules unacceptably inaccurate for this type of system. However, it seemed strange that in the Appendix they had justified the total isolation of the engine at the fluid coupling with the assumption that the driven system, B, contained negligible damping when in fact it included the propeller which was subject to considerable turbulent damping (proportional to the square of the vibrational velocity).

The authors had given no details of system B. If this involved a long propeller shaft of low flexibility then it could be shown that the effects of the damping at the propeller on the remainder of the system were negligible; if however the shaft was short and stiff the effects of the propeller damping might be considerable, thus calling into question the validity of the statement in the Appendix that β_{11} is real.

With the rigidly coupled type of propulsion system a similar theoretical analysis could be performed but involving turbulent damping at the propeller. In this case not only did the matrix $\underset{\sim}{M}$ become non-diagonal but also the matrix $\underset{\sim}{L}$ which would now contain the turbulent damping terms. This would seem to indicate that coupling of principal modes should be expected in marine propulsion systems. However, as already stated, the presence of a long flexible propeller shaft could be shown to reduce the non-diagonal elements to very low values, thus implying almost negligible mode coupling. Under those circumstances the conditions for resonance would reduce to those which had been so widely used in the past.

With the increasing use of "all-aft" machinery configurations, the propeller-shaft stiffness was considerably increased and one would expect this to lead to significant mode coupling in the traditional propulsion systems. It would seem reasonable, therefore, to question the reliance which could be placed on Lloyd's Rules in the case of any 'all-aft' configuration, bearing in mind also the greater occurrence of tailshaft failures by crack propagation from the keyway in short-shaft propulsion systems. These failures had, in many cases, been simply attributed to "the effects of the keyway as a stress raiser" whereas it might have been more correct to attribute them to the higher stresses, resulting from mode coupling, using up the factor of safety inherent in the present Rules.

The implication of the authors' work was surely that the time had now come to take a new look at the Rules in the light of modern advances with a view to being able to predict more accurately the behaviour of non-traditional systems for which the established empirical criteria no longer applied.

MR F E B WEBB, C Eng, MIMechE (A.P.E.-Allen Limited) expressed surprise that the authors, using traditional methods (set out for example in the Rules of Lloyd's Register of Shipping*) had been unable to predict that the two-noded 6th order critical at 346 rev/min would induce a vibratory torque in shaft number 10 greater than the mean torque at the same speed. Using the data given he had been able to arrive at the following results.

The mean torque at 346 rev/min according to the propeller law, would be 47 kNm.

The harmonic component for the 6th order was 0.06 N/mm^2 and the "static response" was ± 0.0004 radians.

Lloyd's Register Rules did not give a formula for the dynamic magnifier for a tuned damper, but Mr Webb understood that $MD = \dfrac{4\Sigma(J\Delta^2)}{J_R\,(\Delta_{D1} - \Delta_{D2})^2}$ gave accurate results (using the notation of Lloyd's Rules) and in this particular case gave a figure of about 8.5. Engine damping by comparison was small, and no allowance need be made for the fluid coupling.

The vibratory torque in shaft number 10 was ± 64 kNm, i.e. 1.36 times the mean torque at 346 rev/min.

The disturbing feature of this case was that the coupling bolts failed. The vibratory torques were not excessive. It would appear that the fluid coupling had not been designed to cope with the torque variations produced by a diesel engine.

MR K-H KLUGE (Blohm & Voss, Hamburg) commented that his company, as manufacturers under license of SEMT Pielstick medium-fast engines of Types PC2, 3 and 4 normally drew up calculations for the whole system, from the diesel engine up to, and including the propeller. This meant that for the excitation from the engine and from the propeller the whole system was examined (as installed), and calculation of the natural frequencies of the two systems were dealt with separately, up to the fluid coupling in both cases.

However, the system specified in the paper only contained data for the section from the diesel engine to the fluid coupling.

Blohm & Voss had carried out a comparison calculation for this section, using their torsion computer programs; the values did not differ greatly from the ones presented in the paper.

It might be of interest to note one or two features of those calculations:

a) They were made individually for each system section and for any specified speed (in this case 440 rev/min).
b) Calculations were made for torques (and also the stress, if the corresponding diameter was known). for the deflection; for the geometric sum up to order 12; for the largest individual deflection with its order; and for the torques of the four highest individual orders.
c) It was also possible to determine the moments for four important system points over a required speed range. This provided a rapid survey with very little paperwork.
d) In the calculations the system had been assumed to have the fluid coupling attached. The masses of the parts, from the transmission to the propeller, had been assumed to be treated in the coupling estimated. The results were almost identical with those of the previous

* Rules and Regulations for the Classification of Ships 1978 Lloyd's Register of Shipping, Part 5, Chapter 8, Sections 7-12 inclusive.

calculations. The natural frequencies remained, of course, unchanged.

As the authors of the paper had correctly assumed, a fluid coupling separated a torsional vibration system. Theories with which Blohm & Voss were acquainted and the calculation in their computer programs were based on this assumption. In the case of two ship's systems, however, experience had led them to use a fluid coupling in an auxiliary drive on the diesel engine.

Contrary to expectations (and also contrary to the preliminary calculations) almost the same values were measured on the auxiliary drive as at the free engine end (Measurements had been made with an electric vibration pick-up). There was, therefore, no separation of the systems by the fluid coupling.

If the authors had had any similar experiences he would be pleased to hear of them.

PROFESSOR D E NEWLAND, Ma, ScD, C Eng, FIMechE (Professor of Engineering, University of Cambridge) was surprised that the torsional vibration of systems with a fluid coupling had not been studied in detail before. The authors had made a valuable contribution by drawing attention to this omission and by describing their calculations for a particular case. Prof. Newland had studied the paper with great interest and would like to comment on a number of points arising from the results and conclusions.

In order to explain the background to these comments, it was helpful to refer first to a primitive system consisting of two rigid flywheels joined by a massless elastic shaft, (Figure 7). This system has two degrees-of-freedom and two natural frequencies, $\omega=0$ rad/s (corresponding to steady rotation with no torsional vibration) and $\omega=\sqrt{k\frac{I_1+I_2}{I_1 I_2}}$ (for the I-node mode of torsional vibration).

The system's characteristic equation was

$$s^2\{I_1 I_2 s^2 + k(I_1 + I_2)\} = 0 \qquad (1)$$

and, in formal terminology, the eigenvalues of the system were

$$\begin{aligned} s_1 &= 0\\ s_2 &= 0\\ s_{3,4} &= 0 \pm i\sqrt{k\frac{(I_1+I_2)}{I_1 I_2}} \end{aligned} \qquad (2)$$

where the real part of these expressions indicated (zero) damping and the imaginary part gave the angular frequency of the characteristic oscillations (i.e. the natural frequency). If coordinates ϕ_1 and ϕ_2 gave the angular displacement of the two wheels (Figure 7), then, for example, the transient solution of the equations of torsional vibration had the form

$$\phi_1 = A + Bt + C\cos\sqrt{k\left(\frac{I_1+I_2}{I_1 I_2}\right)}\,t + D\sin\sqrt{k\left(\frac{I_1+I_2}{I_1 I_2}\right)}\,t \qquad (3)$$

where A, B, C and D were constants determined by the initial conditions. The repeated root $s_1 = 0$, $s_2 = 0$ gave rise to the Bt term, corresponding to steady rotation at constant angular speed.

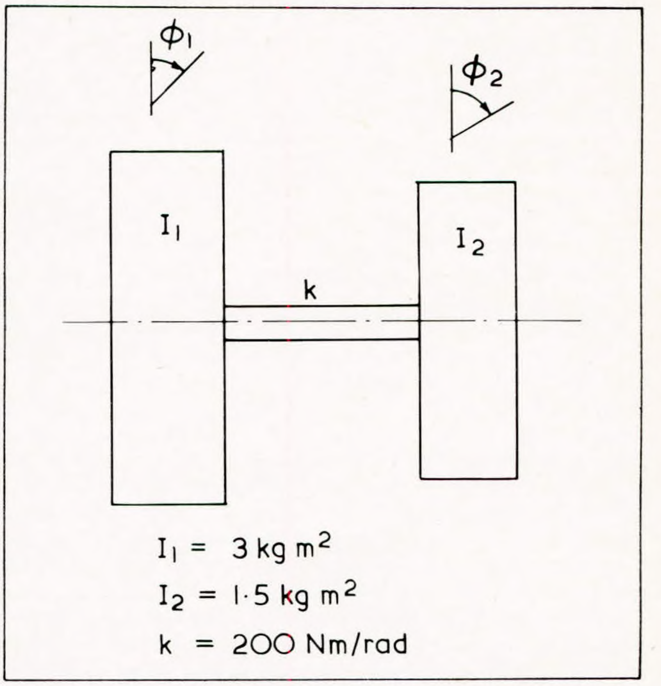

FIG 7 *Uncoupled sub-system with two degrees-of-freedom.*

For the parameter values shown in Figure 7, the numerical eigenvalues were:

$$\begin{aligned} s_1 &= 0\\ s_2 &= 0\\ s_{3,4} &= 0 \pm i\ 14.14 \qquad (s^{-1}) \end{aligned} \qquad (4)$$

indicating that the natural frequency of undamped torional vibration was 14.14 rad/s = 135 cycle/min.

Next, consider the system shown in Figure 8: In this case the wheel of inertia I_2 was subjected to rotary viscous damping; the wheel might be close to a parallel wall from which it was separated by a layer of oil. The coefficient of viscous damping was λ. The eigenvalues of this system were the solutions of the characteristic equation

$$s\{I_1 I_2 s^3 + \lambda I_1 s^2 + k(I_1 + I_2)s + k\lambda\} = 0 \qquad (5)$$

and, for the same parameter values of Figure 7 plus the damping coefficient λ given in Figure 8, the numerical eigenvalues were

$$\begin{aligned} s_1 &= 0\\ s_2 &= -4.42 \qquad (s^{-1})\\ s_{3,4} &= -3.42 \pm i\ 12.57 \qquad (s^{-1}) \end{aligned} \qquad (6)$$

The corresponding transient solution had the form

$$\phi_1 = A + Be^{-4.42t} + e^{-3.42t}(C\cos 12.57t + D\sin 12.57t) \qquad (7)$$

The natural frequency was, for these parameter values, 12.57 rad/s = 120 cycle/min, and the damping ratio approximately $\frac{3.42}{12.57} = 0.27$

Thirdly, consider the system shown in Figure 9. In this case a third wheel of inertia I_3 was added to replace the rigid wall in Figure 8, the system remaining otherwise unchanged. The new characteristic equation was then

$$s^3\{I_1 I_2 I_3 s^3 + \lambda I_1(I_2+I_3)s^2 + kI_3(I_1+I_2)s + k\lambda(I_1+I_2+I_3)\} = 0 \qquad (8)$$

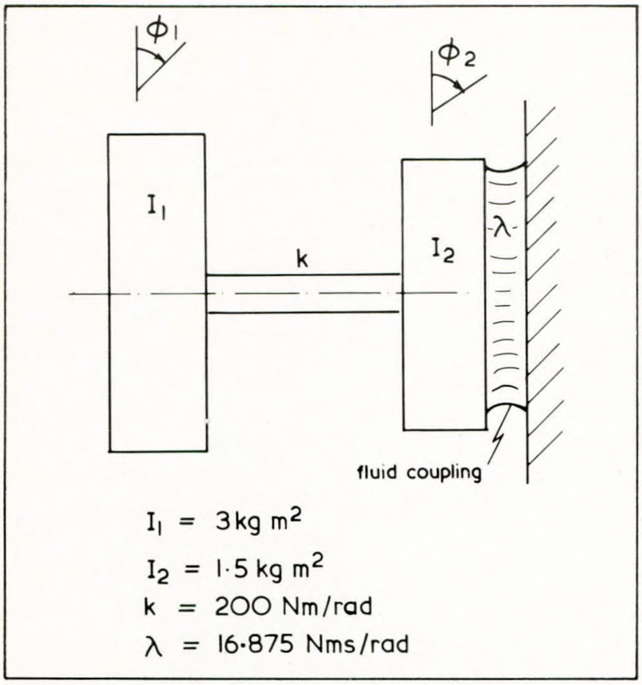

FIG 8 *Sub-system coupled to a rigid wall by a viscous restraint.*

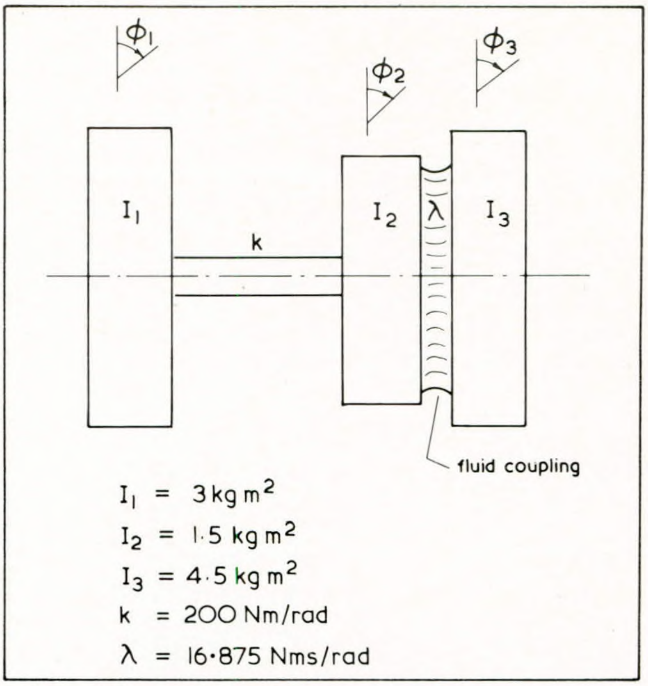

FIG 9 *Three degrees-of-freedom system incorporating a viscous coupling element.*

and, if I_3 had the value shown in Fig 9 (comparable to I_1 and I_2), all other parameter values remaining unchanged, the eigenvalues were:

$$\begin{aligned} s_1 &= 0 \\ s_2 &= 0 \\ s_3 &= 0 \\ s_4 &= -10\,(s^{-1}) \\ s_{5,6} &= -2.5 \pm i\,12.0\,(s^{-1}). \end{aligned} \tag{9}$$

The corresponding transient solution was

$$\phi_1 = A + Bt + Ct^2 + De^{-10t} + e^{-2.5t}(E\cos 12t + F\sin 12t) \tag{10}$$

(in which the constant C was always zero since constant angular acceleration was not possible in the absence of a constant applied torque). The natural frequency was now 12.0 rad/s = 115 cycle/min and the damping ratio approximately $\frac{2.5}{12}$ = 0.21.

Under conditions of forced vibration, maximum amplitudes of vibration would occur when the frequency of excitation was near to the natural frequency, which, for the simple systems described above, had the values tabulated in Table IV.

TABLE IV CALCULATED NATURAL FREQUENCIES AND DAMPING RATIOS FOR THE SYSTEMS SHOWN IN FIGURES 7, 8 AND 9

	NATURAL FREQUENCY	DAMPING RATIO
Uncoupled system (Fig. 7)	135 cycle/min	0
Viscous coupling to a fixed wall (Fig. 8)	120 cycle/min	0.27
Viscous coupling to another wheel (Fig. 9)	115 cycle/min	0.21

Turning now to the published paper, the authors had given numerical data on the natural frequencies of their uncoupled system, the first two of which were 1,380 cycle/min (for the I-node mode) and 2,077 cycle/min (for the II-node mode). However they had not given the corresponding eigenvalues of the coupled system, when viscous restraint of the form shown in Figure 8 had been introduced. It would be most valuable to have this extra data in order to assess how the resonant frequencies had been changed by the introduction of viscous damping at the coupling.

Inspection of the authors' Figure 4 suggested that, since there was a 6th order resonance at 310 rev/min approximately, there must be a torsional critical frequency of about 1,860 cycle/min. The smaller resonance peaks for the 5½ and 6½ order torques occurred at speeds in agreement with this figure. It appeared, therefore, that the natural frequency of the II-node mode had been moved from 2,077 cycle/min to 1,860 cycle/min by the introduction of damping at the coupling.

The position of the I-node mode was not clear from Figure 4. There was no visible resonant peak between 400 and 500 rev/min where 3rd order excitation of a frequency near the I-node resonant frequency would be expected to show a peak. Above 500 rev/min the amplitude curves were evidently moving up towards another peak, but this was probably the result of lower order resonance of the II-node mode (e.g. 3rd order resonance at 620 rev/min).

If, as Prof. Newland believed, a 3rd order resonance of the I-node mode was not visible in the authors' Figure 4, this would explain why they had found that, in the uncoupled case, an assumed Q-factor of 20 gave only relatively small torque amplitudes in the I-node mode. In order to complete the comparison between the uncoupled and the coupled cases, it would be most helpful if the authors could provide an estimated amplitude level for 6th order excitation of the II-node mode in the uncoupled case (using an appropriate assumed Q-factor for this mode).

It seemed to him that, although the presence of a fluid drive altered the natural frequencies of torsional vibration of the system concerned, it might still be possible to use well-established approximate calculation procedures to determine vibration amplitudes, provided that the critical speeds (and mode shapes) were correctly determined. This meant that the eigenvalues (and eigenvectors) of the dynamic system must be extracted by a suitable numerical computer procedure, but there was no real difficulty here, once the problem had been appreciated and the coupled equations of motion established.

Lastly, it was interesting to see that, in the primitive example considered above, the natural frequency of the full system in Figure 9 was significantly different from that of the coupled sub-system shown in Figure 8. This suggested that it might not always be possible to rely on calculations for the propulsion system on one side only of a fluid coupling, and that the adjacent sub-system on the other side of the coupling might have a significant effect on the natural frequencies of the first sub-system. The authors' views on this point would be extremely helpful. In this connection, it seemed that the theoretical analysis in the authors' Appendix was inconclusive in the sense that, although the term

$$\beta_{11} + \delta_{22} + \frac{1}{i\omega b}$$

could not be zero (just as the authors had said), nevertheless it could become very small. Hence the response at coordinate q_4 could become very large (although not infinite), even though δ_{43} was not infinite. Furthermore, even when δ_{43} was infinite, it did not follow that α_{43} was infinite. The reason was that, when δ_{43} approached infinity, then, in general, so would δ_{42}, δ_{23} and δ_{22} all approach infinity. Under this condition, (provided that β_{11} was not infinite) the authors' equation gave;

$$\alpha_{43} \simeq \left\{ \delta_{43} - \frac{\delta_{42}\,\delta_{23}}{\delta_{22}} \right\} \tag{11}$$

which might not be infinite, even though all the δ's were infinite. The truth of this statement could be seen by considering the condition for resonance of the sub-system D when it was rigidly restrained at coordinate 2. Its natural frequencies would then generally differ from those when it was not restrained at 2. When restrained at coordinate 2, $\Psi_2 = 0$, and so, in the authors' notation,

$$\delta_{22}\,\Phi_2 + \delta_{23}\,\Phi_3 = 0 \tag{12}$$

The amplitude at coordinate 4 was given by

$$\Psi_4 = \delta_{43}\,\Phi_3 + \delta_{42}\,\Phi_2 = \Phi_3 \left\{ \delta_{43} + \delta_{42}\,\frac{\Phi_2}{\Phi_3} \right\} \tag{13}$$

which, from equation (12), could be written in the form

$$\Psi_4 = \Phi_3 \left\{ \delta_{43} - \frac{\delta_{42}\,\delta_{23}}{\delta_{22}} \right\} \tag{14}$$

Since, in general resonance no longer occurred at the frequencies for which $\delta_{43} = \infty$ (the condition for resonance in the unrestrained case), it was clear that

$$\left\{ \delta_{43} - \frac{\delta_{42}\,\delta_{23}}{\delta_{22}} \right\}$$

would not necessarily be infinite at the same frequencies as those which made δ_{43} infinite. In short, the coupled system might have resonant frequencies (defined as the frequency for a resonant peak) different from the uncoupled sub-system's resonant frequencies. The simple example quoted above in which the resonant frequency had changed from 135 cycle/min (for the unattached sub-system, Figure 7) to 115 cycle/min (for the coupled system, Figure 9) appeared to support this conclusion.

The Professor looked forward with pleasure to the further comments and conclusions of Professor Bishop and his co-authors, who were to be congratulated on their interesting and thought-provoking paper.

Authors' Reply

PROFESSOR R E D BISHOP, replying for the authors, said that before turning to the individual contributions they would like to thank all contributors for their trouble. Some of the comments were quite vigorous and provocative, but all were stimulating. It was encouraging to find so much interest in what some might think of as a pretty drab and esoteric subject.

Turning to what one or two of the contributors had written, the authors felt a little like the Englishman abroad who, when he finds himself not understood, merely shouts a little louder. In the paper they had referred to a problem of torsional vibration when there was a fluid coupling in the vibrating system and had pointed out that:

1) Judging from the limited information on the subject there was confusion on how to tackle that problem.
2) It could be tackled on the basis of an assumption which, the authors suggested, was self-evidently empirical and could not possibly be regarded as exact.
3) On the basis of this assumption the problem could be resolved using a commonly quoted technique or "closed form analysis" in view of its mathematical objectives. On the other hand, direct application of "critical speed analysis" did not lead to correct predictions although it was often recommended in the literature.
4) The solution found with the critical speed approach could in fact be reconciled with predictions made on the basis of the closed form analysis and, as the authors could not find such reconciliation in the literature, they had showed the truth of this.

This had all been clearly set out in the paper, but one or two contributors seemed to have missed the point.

In the terms of the above four aims, Mr McKinlay and Dr Bickley had held that Point 1 was false because there was *no* confusion Point 2 they had not referred to explicitly. Point 3, they had felt, was common knowledge. And point 4 was "not necessary" according to Dr Bickley!

These two contributions came so close to each other that it would be convenient to take them together. Dr Bickley had first referred to the coupling bolt failure, contending that "it would appear that a totally inadequate bolting arrangement had been used". Assuming acceptance of his assertion that the "fluid coupling input had been used to serve the purpose of a flywheel", it seemed strange that "it was not customary to calculate the vibratory torque conditions for a flywheel/crankshaft bolting arrangement . . ." Why not? (Indeed, judging from para. 6 of Dr Bickley's own contribution, it was not clear that he really meant what he said.)

The connection in question had 12 fitted bolts and the authors knew of no reason for questioning its design or construction. On the contrary they had suggested that the experience Dr Bickley relied on should be supplemented with something less vulnerable (i.e. some simple calculations); as, if calculation was "not customary", it was not clear how he would judge what connection would be adequate. And if, as here, the connection was to withstand substantial torque reversal over the whole range of operating speed, it seemed that the customary attitude left much to be desired.

Dr Bickley, in his second point, had taken the authors to task for suggesting that there was either anything novel in their analysis, or that Classification Society rules were in any way deficient. His contribution gave the impression that what they had said was widely known. Before writing the paper, the authors had checked the Lloyd's Register's *Rules and Regulations for the Construction and Classification of Steel Ships 1976,* Chapter R(E) and with great interest they had now checked the *Rules*, Part 5, Chapter 8, Section 7, Item 2. But they had only found two general "catch-all" paragraphs which made neither mention of fluid couplings nor how they should be treated.

The authors had said in the paper that there were two approaches to forced torsional vibration calculations, one of which was satisfactory in this context, and one which was not. Dr Bickley's reference inferred that the satisfactory technique was known to Lloyd's Register and that Lloyd's Register preferred it to be used "where more complex vibratory conditions arose" (whatever that meant). Even this had not been made clear in the reference used by the authors.

Dr Bickley had admitted that the authors had got their sums more or less right, but felt that anyone not only could, but would, have found similar results. In this latter contention he did not appear to be supported by any of the facts known to the authors (and incidentally quite a lot had been written about this particular propulsion system). Unless the users' manual of TORVAP A provided this information – which seemed highly unlikely – he had provided no references that really supported him and he was plainly out of step with others who had discussed this paper.

Unless he could provide a clear, unambiguous and satisfactory statement of how a fluid coupling was covered by authoritative design rules, the authors hoped he would forgive them if they registered a degree of scepticism. What he had said could too easily have been said with the help of hindsight.

Turning to Mr McKinlay: first of all, the authors wished to reject his view on the actual failure they had referred to for precisely the reasons they had stated. How one could seek to discredit the design of a coupling, or its manufacture and installation, while apparently ignoring a considerable torque reversal over the whole working range of operating speed, was not clear. If this really was thought to be good practice in marine engineering, the quicker such thinking was tightened up the better.

No one questioned the standing of Lloyd's Register's Rules. Nor had the authors suggested that they were like a Fair Isle jumper, i.e. comforting, complicated and beautiful – but made of pure wool. What they had done was drop a gentle hint that the jumper might not adequately cover all the relevant parts.

Now, since attention had been focussed on this detail, they were bound to say that the one thing absent from this discussion had been a clear and unequivocal statement of how Lloyd's Register had treated fluid couplings up till now, with suitable references and dates. If such a statement could be made and the techniques shown to be equivalent to, or better than, what the authors had suggested, they would gladly withdraw and ask to be forgiven for hinting that all might not be well.

It was understandable that the technical points the authors had mentioned seemed to raise a question regarding Lloyd's Register's Rules. But it was far from understandable that they had provoked little more than assertions that the rules were sacred.

Like Dr Bickley, Mr McKinlay had asked a number of detailed questions. The authors felt that they were of somewhat secondary importance and did little more than distract attention from the main matters in debate and they would therefore ignore them. To take a specific point, common to both contributions, did the vectorial addition of responses (as opposed to arithmetic summation) matter much when, at the same time, one was concerned with guessed values of Q factors?

Mr Keys' doubts appeared to centre on Item 2 i.e., on the hypothesis on which the problem could be tackled. Like Professor Newland, he might be assured that the Appendix did not contain a proof of its validity; in the authors' words "it is offered in general support" of the suggestion.

Mr Keys was perfectly justified in raising the wider issues and, although those were not the authors' immediate concern, they were most interested by them. He had perceptively noted that the sea itself complied with the hypothesis (or something very like it) even when the propulsion system was rigidly coupled. The modes were therefore coupled, but the coupling was very weak (and so did not matter) when the propeller shaft was long. And it was this, apparently, which saved Lloyd's Register's rules from failure in all cases (whether or not there was a fluid coupling in the drive).

Professor Bishop added that the authors hoped Mr Keys would follow the matter up; for they agreed with his final observations. Could he deduce a general theorem about the effect of propeller shaft stiffness on modal-coupling? Could he suitably quantify "coupling" in this context? There did seem to be a worthwhile research project in what Mr Keys had pointed out and the authors strongly urged that his comments be taken very seriously.

Mr Webb's contribution was exceedingly difficult to discuss because it contained a number of statements whose basis was not at all clear to the authors. His assertion that "no allowance need be made for the fluid coupling" presumably referred to damping, and one wondered why he thought the coupling was fitted. If he had employed traditional methods (for example as in Lloyd's Rules) the authors were not surprised that he had reached results remote from theirs, from those of other contributors to the discussion and, as far as they could remember, from any that had been found by a number of authorities on the system in question.

The communication from Herr Kluge was very interesting (and should, incidentally, serve to convince Mr Webb that all was not well with traditional methods). That company appeared to have some excellent software and its technique for investigating torsional vibration appeared to have produced the same result as shown in the paper.

The authors' main concern had been to examine a working hypothesis. Inasmuch as the calculations agreed with measured data, the hypothesis had been upheld. But there remained the question of whether or not the hypothesis would *always* produce the correct result. Blohm & Voss had apparently run across a case in which it did not and the authors offered the following comments:

a) The hypothesis could not possibly be exactly right, as they had already pointed out.
b) They would expect the anomalous behaviour of the auxiliary drive they had mentioned to be dependent – and perhaps heavily so – on frequency of excitation.
c) It would be of great interest to examine the system in question in the light of the type of reasoning suggested by Professor Newland.
d) It should be possible to examine any particular system

adequately, when necessary, by calculating both sides and allowing for a quadrature linking at the coupling. The authors should perhaps have done this themselves, as other contributors to the discussion had implied, but it could only be at the expense of lengthening the paper unduly.

e) They had no experience of a similar case.

This had not been the only contribution to the discussion which had persuaded the authors that there was a serious matter here, and one needing more attention.

Professor Newland's comments were both welcome and characteristically elegant. Although the results the authors had obtained did suggest that the hypothesis used was adequate, it was self-evident that it could not be exactly true. In fact Professor Newland himself had demonstrated this quite clearly and his observations had led in the end to questions about its validity. The authors agreed that there did remain a good deal that could and should be sorted out.

Professor Newland had asked a number of specific numerical questions and, short of making the computations again from scratch, the authors were unable to supply the crisp answers that he would expect and they would have liked to give – at least without a considerable delay. The reason was that the author (P K Y Tam) who had made the detailed calculations now resided in the Far East, but if the program came to light and an opportunity to examine these matters should present itself, they would communicate with Professor Newland privately.

He had stated that "it might not always be possible to rely on calculations for the propulsion system on one side only of a fluid coupling and that the adjacent sub-system on the other side of the coupling might have a significant effect on the natural frequencies of the first sub-system". As the last portion of his contribution was devoted to a discussion of this point, the authors offered a number of comments on what he had said.

a) One was not really concerned with natural frequencies but with resonance frequencies since, strictly, the argument disappeared if damping was ignored. To put the matter another way, natural frequency, as Professor Newland had used the term, was the imaginary part of a root and was unlikely to be a helpful parameter to contemplate.

b) This was a matter which seemed well worthy of study but the authors could not accept that the facts would be laid bare without due regard to phase differences: amplitude and phase in steady harmonic vibration seemed to be crucial, and not just amplitude in free vibration.

c) The argument about the possible smallness of

$$\beta_{11} + \delta_{22} + \frac{1}{i\omega b}$$

seemed to be quite sound; but the phase angle should be noted.

d) The second point centred upon Professor Newland's equation (14), could be argued more simply since that equation was obtained from the authors' result

$$\alpha_{43} = \delta_{43} - \frac{\delta_{42}\delta_{23}}{\beta_{11} + \delta_{22} + \dfrac{1}{i\omega b}}$$

by merely noting that it was obtained when

$$\beta_{11} \to 0 \text{ and } b \to \infty .$$

This seemed to be a rather more illuminating approach since it underlined the nature of the necessary limiting process. What its practical significance was the authors were not sure.

Two approaches to more general theory suggested themselves. One was through particular idealized systems (which need not, of course, be specified numerically). This line of attack had been used by Professor Newland to considerable effect and it could obviously be made to shed much light on the function of fluid couplings in torsional systems. The other was through suitable extensions of receptance theory. Either way it seemed to the authors that questions of phase were of the essence and it was in this direction that the authors recommended that Professor Newland's contribution should be generalized.

To sum up then, it seemed that some tidying up of current design rules was needed for systems embodying fluid couplings. At the very least a more incisive formulation of those rules seemed desirable. Although the working hypothesis that the authors had quoted from Tuplin's book[5] gave excellent results in the system examined, there was practical evidence (from Herr Kluge and Professor Newland) that it might not always do so.

The existence of difficult special cases was by no means precluded by theory (as had been agreed by Mr Keys and Professor Newland). And the very reasons why the fluid coupling raised such issues had wider implications where classification society rules were concerned (as Mr Keys had pointed out).