The Design of a Resonance Changer to Overcome Excessive Axial Vibration of Propeller Shafting

COMMANDER A. J. H. GOODWIN, O.B.E., R.N.(ret.) (Member)*

The Admiralty has been approached by various shipping interests to make known to them any further experience they may have in the field of axial vibration in the propeller shafting, subsequent to that given in Mr. C. P. Rigby's paper[1] to the Institute of Marine Engineers in 1948.

The author, who was associated with the problem when serving in the Admiralty from 1949/53 and since then as a member of the Yarrow-Admiralty Research Department, was therefore asked to produce this paper which deals mainly with the performance and design of resonance changers, which were first adopted by the Admiralty as a solution for troublesome axial vibration problems in H.M.S. *Eagle* in 1950.

The resonance changer is a tuned damper of a novel type using hydraulic means for mass and stiffness as well as for damping, and has the advantage of being readily and cheaply fitted to existing systems. Use of this device obviates the necessity for adopting unorthodox positioning of thrust blocks, or departing from the most suitable number of propeller blades.

This paper gives a condensed version of Rigby's original paper, most of which remains valid, followed by an account of the trials which indicated that the use of multibladed propellers did not always constitute a satisfactory cure for excessive vibration. A mathematical explanation is given of the action of the resonance changer, leading up to suggestions for optimizing the design, and the results of trials with the resonance changer in use are given in support of the theory. A recommended design procedure for the shafting system in new construction is included.

INTRODUCTION

The possibility of the main propeller shafting of ships suffering excessive axial vibration has only been recognized comparatively recently. This vibration can be envisaged in its simplest form as that of a mass, the propeller, on the end of a spring, the propeller shaft, which is rigidly attached at the other end. If a disturbing force of a frequency equal to the natural frequency of this system is applied at the propeller, resonance occurs with the possibility of large amplification of the vibrating forces and displacements. Such a disturbing force may be present due to the changes in thrust that occur as each propeller blade passes through different wake conditions.

The system is in fact more complicated than the simple form described, since the thrust block is not completely rigid, and the mass of the gear wheel and associated parts affects the motion and these together result in the natural frequency being somewhat lowered. They also result in the introduction of a second and higher natural frequency, mainly dependent on the relationship between thrust block stiffness and gear wheel mass, but fortunately this second critical speed is usually well above the running speed.

There are several factors that make naval vessels more prone to suffer from axial vibration of the shafting than merchant ships. The higher shaft speeds used, and the relatively long hollow shafting with its lessened axial stiffness, combine to make the critical speed more likely to fall within the running range; the more common use of three or four shafts increases the likelihood of aggravating the problem by causing one propeller to run in another's slipstream.

It is not unnatural therefore that this problem first demanded attention in the Royal Navy. H.M.S. *Warspite* ran into serious trouble on her full power turning trials on completion of re-engining just prior to the 1939/45 War. A means of overcoming the trouble was devised although the problem was not fully understood, and during the war all large multishaft ships had to reduce power on their outer shafts on the outside of a turn when high speeds and rudder angles were used.

In 1943 Mr. C. P. Rigby, then serving in the Engineer-in-Chief's Department in the Admiralty, had his attention drawn to the problem during discussions with other departments on an apparent "critical speed" in H.M.S. *Furious*. His investigations and recommendations led to a series of trials in H.M. ships and the introduction of four-bladed or five-bladed propellers to overcome the vibration in certain ships, and he gave a paper to the Institute of Marine Engineers in 1948[1] giving a full account of his findings. Some of the outstanding points in his paper are summarized below.

SUMMARY OF RIGBY'S PAPER

Rigby reached the conclusion that resonance between propeller impulses and the axial natural frequency of the shaft system was responsible for building up large amplitudes of vibration. It was also concluded that the propeller thrust variation existed on a straight course due to the blades passing through the comparatively dead water near the hull, and was much accentuated during turns if the blades also encoun-

* Yarrow-Admiralty Research Department.

tered, at the opposite end of a diameter, the fast running water in the slipstream from a wing screw.

He reasoned that it should be possible to reduce the vibration to acceptable limits if this resonance could be avoided, and two possible methods presented themselves:

(a) Increasing the number of propeller blades and running through the critical at a lower power where it might be harmless. This method was the simplest for ships in service, provided that such a propeller did not suffer from excessive cavitation.

(b) Moving the thrust block further aft, which had the advantage of avoiding any loss of efficiency which might arise by increasing the number of blades, but had the disadvantages of not being readily applicable to existing ships and of complicating the lubricating oil system.

It was obviously desirable that further experimental evidence should be collected, and therefore a series of trials was carried out in a number of vessels covering twin, triple and quadruple-screw systems with a view to obtaining sufficient fundamental data to allow reliable prediction of the performance of any arrangement of shafting and number of propeller blades.

The data required consisted principally of the following: flexibility of thrust block and seat, weight of entrained water to be added to the propeller, propeller thrust variation fraction, referred damping factor, and effect of turns.

In analysing the results the critical speeds had to be calculated and the method used by Rigby was to convert the shaft system into an equivalent mass-elastic system and evaluate the natural frequency by the well known Holzer tabular method[2].

Using the observed critical speed and having made an estimate of the spring constant between the thrust collar and the bottom of the ship or "earth", based on the results of a static deflexion test on the thrust block of a ship then under construction, Rigby evaluated, from trials in this ship, an expression relating the entrained water to the developed blade surface. Using this in examining other trial results, he evaluated thrust block flexibilities varying from 0·036-in. to 0·057-in. forward deflexion under full power thrust. From these he concluded that, in estimating the critical speed for a new ship with thrust block and seat of conventional proportions, the use of a value of 0·045-in. forward deflexion under full power thrust would give satisfactory results.

On the assumptions that the mean thrust varies as the square of the speed of revolution, that for a given propeller and hull the thrust variation is a constant fraction of the mean thrust, and that all the damping present in the system is at the propeller, analysis of the magnitudes of the amplitude of vibration allowed him to estimate values for the propeller thrust variation and for the damping.

The results of the trials carried out in H.M.S. *Formidable* with three-bladed propellers and in H.M.S. *Illustrious* with five-bladed propellers suggested that the damping factor increased with the number of blades.

The effect of turns was studied by trials at various conditions of speed and rudder angle and from the results Rigby suggested that, in estimating amplitudes for new construction, a "turn factor" of 5 be used for the inner shaft on outside of turn for a quadruple-screw ship with A-brackets and a factor of 4 for the centre shaft of a triple. The latter value did not mean that the triple suffered less during turns, merely that it was worse on straight course. The effect of turning is less in twin-screw ships, the turn factor amounting to about 2 and affecting principally the shaft on the inside of the turn.

The work which he had carried out gave data on and methods of predicting the amplitude of vibration, but it was still necessary to decide what was considered acceptable. From Naval experience he concluded that the amplitude should not be allowed to exceed ±0·010in. on straight course and ±0·025in. when turning, with avoidance of thrust reversal as the overriding consideration. In triple and quadruple-screw ships, with a turn factor of 4 or 5, the amplitude on the turn was the criterion and, to keep the limit of ±0·025in., the amplitude on the straight should be kept down to ±0·005in.

Having derived fundamental data which could be used in new construction, Rigby also commented on the desirability of having a short method of estimating the critical speed. Where the thrust block is in the conventional position, by treating the system in its simplest form, i.e. as a weight (the propeller and entrained water) attached to a heavy bar (the shaft between propeller and thrust collar, disregarding couplings, gunmetal liners, etc.), and ignoring the flexibility of the thrust block, the "rigid natural frequency" can be determined by the method given by Timoshenko[2]. Rigby modified this method to give a more suitable equation related to Fig. 16 of his paper. When the "rigid natural frequency" has been calculated it must be corrected to allow for the thrust block flexibility. This is done by multiplying it by a "flexibility factor" obtainable from Fig. 17 of his paper, which gives a plot of flexibility factor against shaft length based on the trials results.

(N.B. Rorke has subsequently suggested that it would be better to plot it against the ratio, shaft sectional area/shaft length.)

Theoretical studies were also carried out, as reported in the appendices of his paper, to assess the effect of altering the thrust block position and number of blades.

In the well informed discussions and correspondence that followed Rigby's paper several interesting suggestions were made and, amongst them, some for devices for reducing axial vibrations. The late Mr. H. G. Yates, in commenting on Rigby's remarks regarding Dr. G. H. Forsyth's proposal, in a paper on gearing, to introduce flexibility into the shafting, so that vibration could be reduced by the classical method of running at a frequency considerably higher than the natural frequency of the system, pointed out the practical difficulties of this method. He proposed, instead, the fitting to the propeller shaft as far aft as possible of a form of hydraulic damper. Captain (then Lieut. Cdr.) G. F. A. Trewby proposed the fitting of a tuned mechanical damper to the propeller shafting.

Other points arose and some of these now seem particularly pertinent. Rear Admiral (then Captain) J. G. C. Given doubted the validity of using displacement amplitude as a design criterion. Commander L. Baker referred to a magnification of amplitude during a turn of a value probably well exceeding 6·4 in a ship in which he had served. Mr. S. Archer questioned the assumption that substantially all the damping was contributed by the propeller, since theory suggested smaller damping for the propellers alone than was experienced in practice. Dr. T. W. F. Brown saw no reason to depart from integral thrust block design for gear cases on the score of axial vibration. Mr. F. McAlister drew attention to the loss in efficiency that might be involved in increasing the number of propeller blades. Captain Trewby suggested a means for including the shafting in the Holzer Table in one step rather than by a large number of sections.

EXPERIMENTAL RESULTS, 1949-1959

During the period that has elapsed since Rigby's paper, axial vibration trials have been carried out on a number of Naval vessels. Experience gained in H.M. Ships *Vanguard, Eagle, Savage, Triumph, Ark Royal, Bulwark,* and *Whitby* is recounted hereinafter. In the first of these, it transpired that the fitting of multibladed propellers was not always such a successful cure for axial vibration troubles as the promising results attained in H.M.S. *Illustrious* and reported by Rigby had given reason to hope. The improved instrumentation available has confirmed that the possibility of thrust reversal is a better criterion for unacceptable vibration than the magnitude of the amplitude at the critical speed, but has shown that the magnification due to turning in a quadruple-screw ship may be as great as 10 rather than the value of 5 given by Rigby. Furthermore, experience in some twin-screw ships has shown that thrust vibration short of thrust reversal may loosen thrust block securities and lead to unacceptable vibrations. As a result, it has been necessary to devise other means of reducing excessive axial vibration where the critical speed occurs within or nearly within the running range. A description of these means and methods for their design are included in this paper.

TRIALS ON H.M.S. *Vanguard*

Trials carried out with the original three-bladed propellers showed a critical frequency that caused excessive vibration at 208 r.p.m. Calculations carried out by Rigby indicated that substitution of five-bladed propellers on the inner propeller shafts would constitute a cure, and that the critical speed to be expected at 125 r.p.m. would not lead to excessive vibration or thrust reversal when turning.

New five-bladed propellers were manufactured and fitted, and trials were arranged to take place in the autumn of 1949 shortly after the author had been appointed to the Admiralty in the Gearing, Propellers and Shafting Section. The results of these trials were both disappointing and perplexing. A severe vibration occurred at 152 r.p.m., accompanied by thrust reversal and large amplitude vibration when the vessel was turning.

In view of the previous calculations, there was considerable doubt as to whether this was an axial vibration, but a rather crude experiment showed that it was undoubtedly so. This consisted of putting marking on the shafting near to the stern tube and using a scriber on it when the severe vibration was taking place. Five clear sinusoidal oscillations around the periphery of the shafting were regularly produced with an axial amplitude of about an eighth of an inch. As a temporary expedient the ship was warned against steaming steadily within 5 r.p.m. on either side of this critical speed and particularly against turns at this speed, but it was clear that something further was required for a permanent cure.

ORIGIN OF RESONANCE CHANGER

H.M.S. *Eagle* was due to carry out sea trials in 1950 and arrangements had been made to provide both three- and four-bladed propellers for the inner shafts. Calculations had indicated that there was a strong possibility of the critical speed occurring at about full power with three-bladed propellers, and the four-bladed propellers had been ordered to allow for this contingency on the assumption that the increase in the number of blades would prevent serious axial vibration. It was thought however that the three-bladed propellers would have a higher propulsive efficiency than the four-bladed, and it was hoped that the trials might show that the critical speed was sufficiently above the full power revolutions to permit their use. In the light of the trials results from *Vanguard* it seemed possible that neither the three- nor the four-bladed propeller would be satisfactory from the point of view of axial vibration, and it became urgently necessary to find some other solution to the problem.

In 1944, when the investigations into axial vibration were in their early stages, Michell Bearings Ltd., the well known thrust block designers, reported that the chief engineer of a merchant ship had informed them that he had observed that there seemed to be a noticeable reduction in general vibration when the Michell thrustmeter fitted in his ship was in use. They suggested that in fact the thrustmeter might be acting as a damper for axial vibration. In pursuit of this suggestion the Admiralty arranged comparative trials, with and without thrustmeters in use, in a ship subject to axial vibration, but found no measurable difference, a result, incidentally, which has been confirmed in the trials reported in this paper. It was however thought that an increased damping effect might be produced by the addition of an oil reservoir, but this was not taken further because it was considered that this could only be effective if there were increased amplitude of vibration of the thrust collar, the very thing it was the purpose to minimize.

The Admiralty decided on a different approach for a possible solution of the problem during the impending trials in *Eagle*. This was to provide additional flexibility at high powers so that, under these conditions, the system would be running supercritically and hence be subjected to attenuated thrust and amplitude variation. At lower powers, where the greater flexibility would cause increased amplitudes and, at some speed, a critical frequency, it was the intention to cut out the device providing the additional flexibility.

The means for accommodating this device, which was named a resonance changer, lay readily to hand in the Michell thrustmeter, which provides an oil filled space between the thrust collar and the thrust block, with the position of the thrust collar maintained substantially constant relative to the thrust block by the action of the "Limit Valve" incorporated in the thrustmeter, thus overcoming one of the main difficulties associated with increased flexibility, *viz.* that of preventing bodily movement of the shafting when the thrust changes. The additional flexibility was provided by the bulk compressibility of oil contained in an oil bottle which was connected by a pipe or pipes to the cylinders of the thrustmeter, and for effective operation these connecting pipes clearly had to be adequate in size to permit free passage of the oil between the cylinders and the bottle.

For each of the inner shafts in H.M.S. *Eagle* it was decided that a bottle of 9-cu. ft. capacity would be more than adequate, giving a calculated stiffness of 466 tons/in. compared with the shaft stiffness of 825 tons/in. and the thrust block stiffness of about 3,000 tons/in. The avoidance of trapped air in the bottle was very important, since this would have increased the flexibility excessively, and combined vent and filling valves were accordingly fitted on the tops of the bottles, which were mounted in a vertical position. It was considered that a bore of 1·125in. for each of the two 14-ft. long connecting pipes would be adequate, but it was

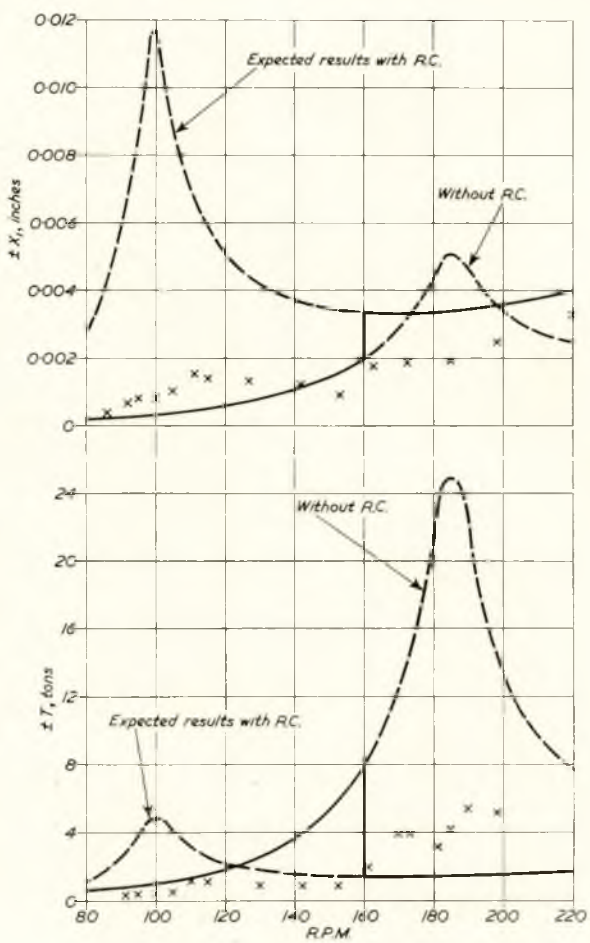


FIG. 1—*H.M.S. Eagle: 9-cu. ft. resonance changer*

clearly necessary to open out the interconnecting passages between the cylinders in the thrustmeter to prevent undue restriction of flow, and a substantial increase in their size was arranged. It was realized that there might be an optimum amount of damping and a skirted valve was fitted at the inlet to the bottle to provide means for varying this. During sea trials an optimum condition seemed to be established with this valve three turns open.

Another modification made to *Eagle's* thrust blocks was to increase the axial clearance from 0·015in. to 0·060in. to provide sufficient room for the thrust collar to vibrate relative to the thrust block proper without bottoming. Michell Bearings Ltd., at Admiralty request, carried out a trial on their "Limit Valve" and demonstrated that it could follow faithfully the expected vibratory movements of the thrust collar at the frequencies involved.

While the resonance changer was still experimental, the hand pump on the Michell thrustmeter had to be used for maintaining the cylinders full of oil. On a straight course this was relatively simple since the plunger rings almost completely eliminated leakage and, when vibration occurred, the "Limit Valve" let out surplus oil until the vibration could continue just without lifting the valve. Only on increases of power, and hence thrust, was it necessary to top up by the hand pump. During turns, however, the thrust on the inner propeller on outside of turn reduced considerably when operating in the slipstream of the outer propeller, so that the relief of pressure in the bottle permitted expansion of the oil and escape through the "Limit Valve". As the full thrust came on again it was necessary to supply the deficit rapidly by the hand pump, which entailed considerable manual labour during elaborate turning trials.

On successful conclusion of the first set of trials, small pumps driven by electric motors were fitted to give a continuous supply of oil to make up such deficits, and a more than adequate supply of 10 cu. in/minute was arranged, the duty being based on the rate of thrust build-up on coming out of a turn.

The results that it was expected to obtain with the four-bladed propeller are shown in Fig. 1 and it was intended that the resonance changer should be in use only above 160 r.p.m. so that the resulting thrust and displacement amplitudes would be as shown by the full lines in this figure.

The first sea trials were carried out in *Eagle* in 1950 with a three-bladed propeller fitted to the starboard inner shaft and a four-bladed one to the port inner shaft. The actual results of these trials considerably exceeded expectations, for not only did the resonance changer eliminate the serious vibration, with thrust reversal during turns, that occurred within the running range with both the three- and four-bladed propellers, but there was no sign of the expected newly introduced critical speed below 160 r.p.m. It was therefore clear that the theory on which the forecasted performance had been based must have been oversimplified and warranted investigation to discover why this critical frequency at lower speeds had not been produced.

At that time, the Admiralty had placed a contract with Alexander Stephens and Sons Ltd. for an investigation into propeller shaft stressing, which was being carried out by Mr. J. Rorke, then of that firm. He was therefore given the task of investigating the performance of the resonance changers. His first discovery was that the oil in the connecting pipes was introducing a very considerable inertia effect, since its velocity was some 133 times the axial velocity of the thrust collar, and this, combined with its hydraulic jack effect, led to its being an effective mass $(133)^2$ times its actual mass. It was at first thought that this effective mass should be added to the gear wheel mass, but the resulting calculations showed that this also did not explain the measured results.

Mr. Rorke continued his investigation and was associated with the subsequent Admiralty trials on axial vibration that were carried out in 1954 and 1955, and it was only after the completion of these that he and the author found the mathematical treatment, given below, that explains the results. It is still more recently that the suggested method, given in Appendix 3, for optimizing the design of resonance changers has been developed as a result of the author's association with other work on vibration problems at Yarrow and Co. Ltd. It is interesting to note that the resonance changer provides the flexibility suggested by Dr. Forsyth, the hydraulic damping suggested by Mr. Yates, is tuned as suggested by Captain Trewby and incorporates the Michell thrustmeter.

ACTION OF THE RESONANCE CHANGER

The resonance changer is illustrated diagrammatically in Fig. 2.

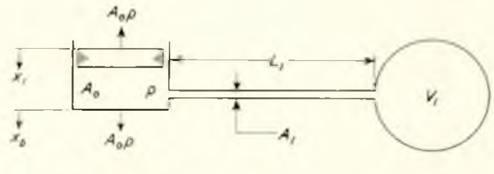


FIG. 2—*Diagrammatic representation of resonance changer*

Assuming that the oil bottle is rigid and that all compression of the oil takes place in the bottle, then the force applied along the pipe by the thrustmeter cylinder pressure must be balanced by the sum of:

(i) The force required to overcome the inertia of the oil in the pipe;

(ii) The force required to overcome the viscous resistance of the oil in the pipe; and

(iii) The force required to compress the oil in the bottle by introducing into it the volume displaced from the thrustmeter cylinders.

On the further assumptions that laminar flow exists and that the hydraulic equivalent length of the system equals the actual length of the pipe, a mathematical analysis can be made and the forces in the pipe equated as follows:

$$*A_1p = \rho_1 A_1 L_1 \frac{(\ddot{x}_1 - \ddot{x}_b)A_0}{A_1} + 8\pi\mu_1 L_1 \frac{(\dot{x}_1 - \dot{x}_b)A_0}{A_1} + A_1 B_1 \frac{(x_1 - x_b)A_0}{V_1} \qquad (1)$$

This equation can be converted to give equivalent forces in the thrustmeter cylinders by multiplying each term by A_0/A_1 thus giving:

$$A_0p = \frac{\rho_1 A_0^2 L_1}{A_1}(\ddot{x}_1 - \ddot{x}_b) + 8\pi\mu_1 L_1 \frac{A_0^2}{A_1^2}(\dot{x}_1 - \dot{x}_b) + \frac{A_0^2 B_1}{V_1}(x_1 - x_b) \qquad (2)$$

where B_1 is the bulk modulus of the oil.

The symbolism can now be simplified by making the following substitutions:

$$\text{Virtual mass, } m_1 = \frac{\rho_1 A_0^2 L_1}{A_1} \qquad (3)$$

$$\text{Virtual damping, } c_1 = 8\pi\mu_1 L_1 \frac{A_0^2}{A_1^2} \qquad (4)$$

$$\text{Virtual stiffness, } k_1 = \frac{A_0^2 B_1}{V_1} \qquad (5)$$

Equation (2) can then be rewritten as:

$$A_0p = m_1(\ddot{x}_1 - \ddot{x}_b) + c_1(\dot{x}_1 - \dot{x}_b) + k_1(x_1 - x_b) \qquad (6)$$

where A_0p is the force transmitted to the external circuit.

ANALYSIS OF SHAFTING SYSTEM AND RESONANCE CHANGER

The marine propeller shaft system with a resonance

* A list of symbols appears in Appendix 1.

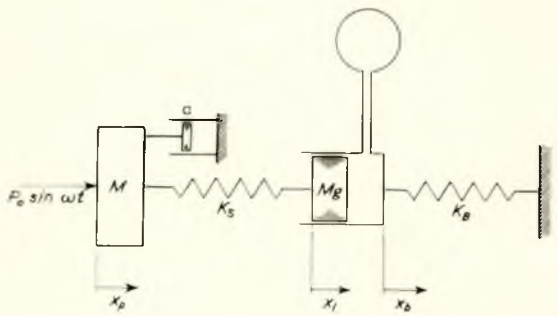


FIG. 3—*Diagrammatic representation of shafting system and resonance changer with equivalent shaft mass concentrated at ends*

changer fitted may be simplified for the purpose of this analysis to give the equivalent system shown in Fig. 3, assuming that allowance for the mass of the shaft can be made in M and M_g.

Appendix 2 gives an indication of the proportion of the shaft mass that should be included in M and M_g for this purpose.

The equations of forces in the system are:

$$P_o \sin \omega t = M\ddot{x}_p + a\dot{x}_p + K_S(x_p - x_1) \quad (7)$$

$$K_S(x_p - x_1) - M_g\ddot{x}_1 = m_1(\ddot{x}_1 - \ddot{x}_b) + c_1(\dot{x}_1 - \dot{x}_b) + k_1(x_1 - x_b) \quad (8)$$

$$m_1(\ddot{x}_1 - \ddot{x}_b) + c_1(\dot{x}_1 - \dot{x}_b) + k_1(x_1 - x_b) = K_B x_b \quad (9)$$

Solutions of these equations are:

$$\frac{X_p}{P_o} = \sqrt{\frac{[K_B(K_S - M_g\omega^2) + (K_B + K_S - M_g\omega^2)(-m_1\omega^2 + k_1)]^2 + [c_1\omega(K_B + K_S - M_g\omega^2)]^2}{\left\{\begin{matrix} -M\omega^2 K_B(K_S - M_g\omega^2) \\ -M\omega^2(K_B + K_S - M_g\omega^2)(-m_1\omega^2 + k_1) \\ -ac_1\omega^2(K_B + K_S - M_g\omega^2) - K_B K_S M_g\omega^2 \\ +K_S(K_B - M_g\omega^2)(-m_1\omega^2 + k_1) \end{matrix}\right\}^2 + \left\{\begin{matrix} a\omega[K_B(K_S - M_g\omega^2) \\ +(K_B + K_S - M_g\omega^2)(-m_1\omega^2 + k_1)] \\ +c_1\omega[K_S(K_B - M_g\omega^2) \\ -M\omega^2(K_B + K_S - M_g\omega^2)] \end{matrix}\right\}^2}} \quad (10)$$

$$\frac{T}{P_o} = K_B K_S \sqrt{\frac{(-m_1\omega^2 + k_1)^2 + (c_1\omega)^2}{\text{Same denominator as equation (10)}}} \quad (11)$$

$$\frac{X_1}{P_o} = K_S \sqrt{\frac{[K_B + (-m_1\omega^2 + k_1)]^2 + (c_1\omega)^2}{\text{Same denominator as equation (10)}}} \quad (12)$$

When the resonance changer is not fitted, these equations reduce to:

$$\frac{X_p}{P_o} = \frac{K_B + K_S - M_g\omega^2}{\sqrt{[-M\omega^2(K_B + K_S - M_g\omega^2) + K_S(K_B - M_g\omega^2)]^2 + [a\omega(K_B + K_S - M_g\omega^2)]^2}} \quad (13)$$

$$\frac{T}{P_o} = \frac{K_B K_S}{\text{Same denominator as equation (13)}} \quad (14)$$

$$\frac{X_1}{P_o} = \frac{K_S}{\text{Same denominator as equation (13)}} \quad (15)$$

OPTIMIZED DESIGN FOR RESONANCE CHANGER

The formulæ given in equations (10), (11) and (12) are virtually unmanageable for the purpose of analysing optimum conditions and it has been assumed for simplicity that both the propeller damping (a) and the gearing mass (M_g) can be neglected when assessing the resonance changer performance.

With these conditions it has been found that tuning the natural frequency of the resonance changer to that of the system without the resonance changer gives best overall results, and this can be done by making:

$$\frac{k_1}{m_1} = \frac{K_o}{M} \quad (16)$$

where $K_o = \dfrac{K_B K_S}{K_B + K_S}$

If $\beta_1 = \sqrt{\dfrac{m_1\omega^2}{k_1}}$ and $\beta_o = \sqrt{\dfrac{M\omega^2}{K_o}}$, then:

$$\beta_1 = \beta_o = \frac{N}{N_c} \quad (17)$$

On the further assumptions that the mean thrust is proportional to the square of the shaft speed and that the thrust variation at the propeller (P_o) is a constant fraction ${}_TK_P$ of the mean thrust at any shaft speed N, then:

$$P_o = {}_TK_P(F.P.M.T.)\left(\frac{N}{N_{FP}}\right)^2$$

$$\text{and } \beta_1^2 \frac{T}{P_o} = \frac{T}{{}_TK_P(F.P.M.T.)\left(\dfrac{N_c}{N_{FP}}\right)^2} \quad (18)$$

The denominator of the second term is constant for any given system and the expression therefore gives an absolute value for the thrust variation at the thrust block (T) at any value of β_1 (or N/N_c) and therefore at any value of N.

Similarly, it can be shown that:

$$\beta_1^2 K_o \frac{X_1}{P_o} = \frac{X_1}{{}_TK_P\left(\dfrac{F.P.M.T.}{K_o}\right)\left(\dfrac{N_c}{N_{FP}}\right)^2} \quad (19)$$

Again, the denominator of the second term is constant and the expression gives an absolute value for the amplitude of vibration at the thrust collar (X_1).

On the basis of the above conditions, equations (11) and (12), on dividing their numerators and denominators by $k_1(K_B + K_S)$ and putting $q_1 = K_o/k_1$ can be rewritten as:

$$\frac{T}{{}_TK_P(F.P.M.T.)\left(\dfrac{N_c}{N_{FP}}\right)^2} = \beta_1^2 \sqrt{\frac{(1 - \beta_1^2)^2 + \dfrac{c_1^2}{m_1 k_1}\beta_1^2}{[(1 - \beta_1^2)^2 - q_1\beta_1^2]^2 + \dfrac{c_1^2}{m_1 k_1}\beta_1^2(1 - \beta_1^2)^2}} \quad (20)$$

$$\frac{X_1}{{}_TK_P\left(\dfrac{F.P.M.T.}{K_o}\right)\left(\dfrac{N_c}{N_{FP}}\right)^2} = \frac{\beta_1^2 K_S}{K_B + K_S} \sqrt{\frac{\left[1 - \beta_1^2 + q_1\left(1 + \dfrac{K_B}{K_S}\right)\right]^2 + \dfrac{c_1^2}{m_1 k_1}\beta_1^2}{[(1 - \beta_1^2)^2 - q_1\beta_1^2]^2 + \dfrac{c_1^2}{m_1 k_1}\beta_1^2(1 - \beta_1^2)^2}} \quad (21)$$

Thrust Variations with Optimum Damping

From equation (20) it will be seen that for any given value of q_1 a family of curves can be produced for a range of c_1^2/m_1k_1 values between zero and infinity. Fig. 4 illustrates

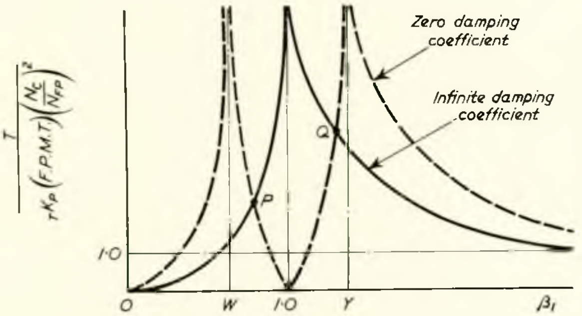


FIG. 4—*Thrust variation response curves*

this. All the curves of this family will pass through points P and Q but the amplification in the resonant regions will depend on the c_1^2/m_1k_1 value.

The desired value of c_1^2/m_1k_1 is that which will allow passage through the resonant regions with the least possible maximum value of $\frac{T}{\text{Constant}}$.

It is shown in Appendix 3 that the value of T at Q must be greater than that at P, hence the least possible value of the maximum thrust variation when passing through the resonant regions is achieved if the c_1^2/m_1k_1 value be so chosen that the curve for $\frac{T}{\text{Constant}}$ has a maximum value at Q. It is shown in Appendix 3, by differentiating equation (20) with respect to β_1 and equating the differential to zero at this value, i.e. when $\beta_1 = \beta_1\,(Q)$, that:

$$\text{Optimum } \frac{c_1^2}{m_1k_1} = \frac{q_1}{4}\left[6 + q_1 + \sqrt{8q_1 + q_1^2}\right] \qquad (22)$$

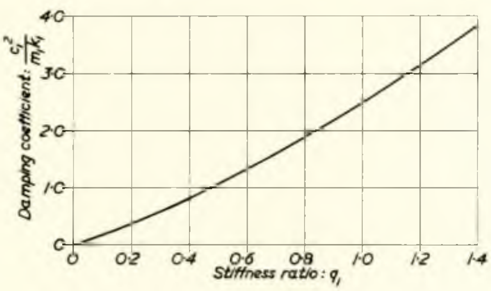


FIG. 5—*Optimum value of resonance changer damping coefficient*

Fig. 5, which is based on equation (22), shows the optimum value of the damping coefficient for various values of q_1. Fig. 6, which is based on equations (20) and (22), shows

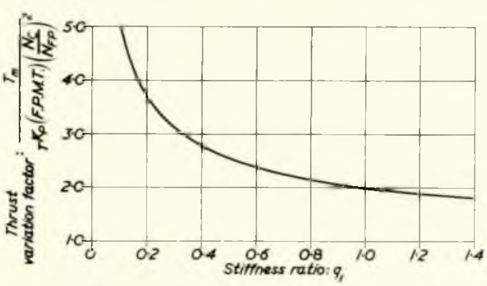


FIG. 6—*Maximum value of thrust variation occurring under optimum damping*

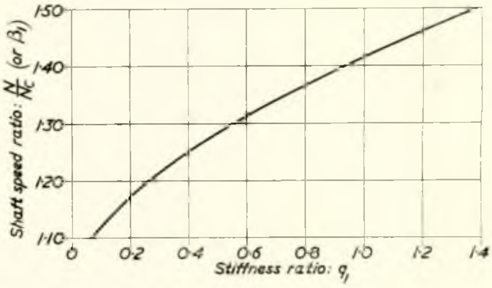


FIG. 7—*Shaft speed ratio at which maximum value of thrust variation occurs*

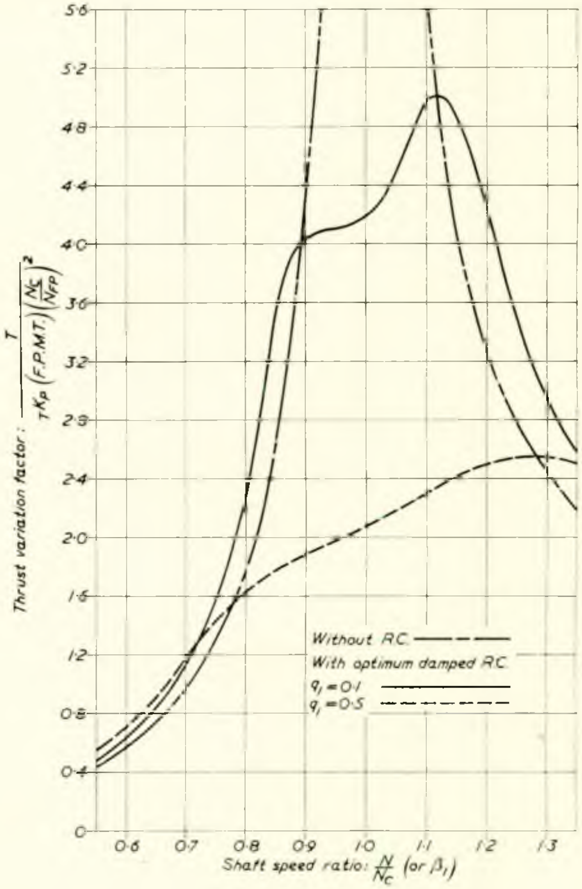


FIG. 8—*Response curves of thrust variation*

the maximum values of the thrust variation at the thrust block for various values of q_1 with optimum damping, and Fig. 7 shows the values of N/N_c at which these maxima occur for various values of q_1. Fig. 8 shows the values of $\frac{T}{\text{Constant}}$ throughout the important part of the running range, using resonance changers giving values for q_1 of 0·1 and 0·5.

Possibility of Thrust Reversal Occurring with Optimum Damping.

Fig. 6 gives values of $\frac{T_m}{{}_TK_P\,(F.P.M.T.)\left(\frac{N_c}{N_{FP}}\right)^2}$ for different values of q_1. Fig. 7 gives, for different q_1 values, the values of N/N_c at which T_m occurs. Hence, it is readily shown that, for a particular value of q_1, the ratio of the maximum thrust variation to the corresponding mean thrust is:

$$\frac{T_m}{(M.T.)_m} = {}_TK_P \frac{(\text{Value from Fig. 6})}{(\text{Value from Fig. 7})^2}$$

For example:

$$\frac{T_m}{(M.T.)_m} = \begin{cases} 1{\cdot}00\ {}_TK_P & \text{for } q_1 = 1{\cdot}0 \\ 1{\cdot}56\ {}_TK_P & \text{for } q_1 = 0{\cdot}5 \\ 2{\cdot}37\ {}_TK_P & \text{for } q_1 = 0{\cdot}25 \end{cases}$$

The ratio of the thrust variation to the mean thrust rises to a value greater than the above at a speed less than that at which T_m occurs and, in fact, becomes a maximum at the point P in Fig. 9 with optimum damping as stated in Appendix 3.

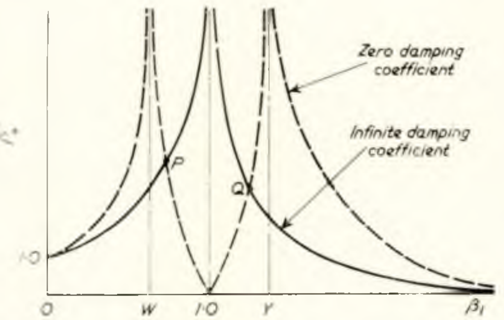


FIG. 9—*T/P_o response curves*

Since $P_o = {}_TK_P\ (M.T.)$, reference to Appendix 3 will readily show that:

$$\frac{T}{(M.T.)}\ (\text{max.}) = {}_TK_P \frac{4}{\sqrt{8q_1 + q_1^2} - q_1} \qquad (23)$$

Hence, for example:

$$\frac{T}{(M.T.)}\ (\text{max.}) = \begin{cases} 2{\cdot}00\ {}_TK_P & \text{for } q_1 = 1{\cdot}0 \\ 2{\cdot}56\ {}_TK_P & \text{for } q_1 = 0{\cdot}5 \\ 3{\cdot}37\ {}_TK_P & \text{for } q_1 = 0{\cdot}25 \end{cases}$$

Therefore, if the values of ${}_TK_P$ and the turning magnification factor be known, a suitable value for q_1 to prevent thrust reversal when a tuned resonance changer has been fitted, can be estimated.

Thrust Collar Movement with Optimum Damping

It is shown in Appendix 3 that, for practical values of K_B, K_S and q_1, under zero and infinite damping conditions the response curves derived from equation (21) will be of the form indicated in Fig. 10, where $\beta_1(Z)$ is greater than $\beta_1(Y)$ and that the value of:

$$\frac{X_1}{{}_TK_P \left(\frac{F.P.M.T.}{K_o}\right)\left(\frac{N_c}{N_{FP}}\right)^2} = \begin{cases} > 1{\cdot}0 \text{ at } \beta_1(A) \\ \ 1{\cdot}0 \text{ at } \beta_1(B) \\ < 1{\cdot}0 \text{ at } \beta_1(C) \end{cases}$$

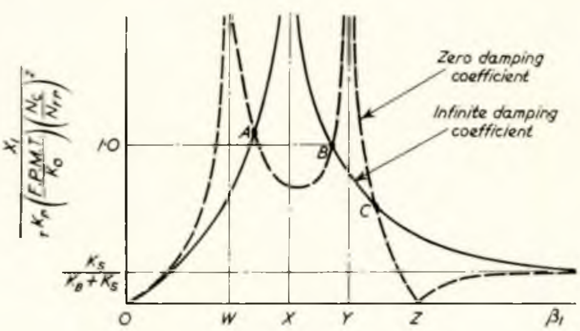


FIG. 10—*Thrust collar movement response curves*

In analysing the thrust amplification, the value of the damping coefficient which gave the optimum response with respect to thrust was derived. Although this value of damping does not quite optimize the displacement response, that it very nearly does is shown in Fig. 11. On these curves the point A is that shown in Fig. 10 above and for each combination of values of K_B, K_S and q_1 represents an amplitude

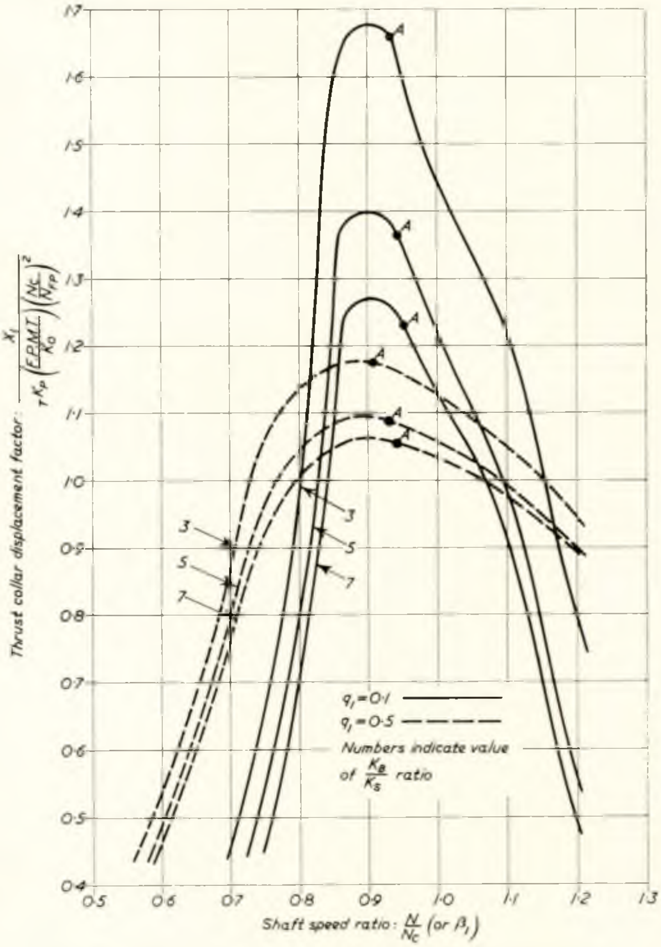


FIG. 11—*Response curves of thrust collar displacement*

which the displacement must reach at this frequency whatever the damping chosen for the resonance changer. Since in no case does the maximum value of the curve greatly exceed the value at A, the damping coefficient chosen on the basis of thrust amplification gives a very satisfactory control of the thrust collar displacement amplification.

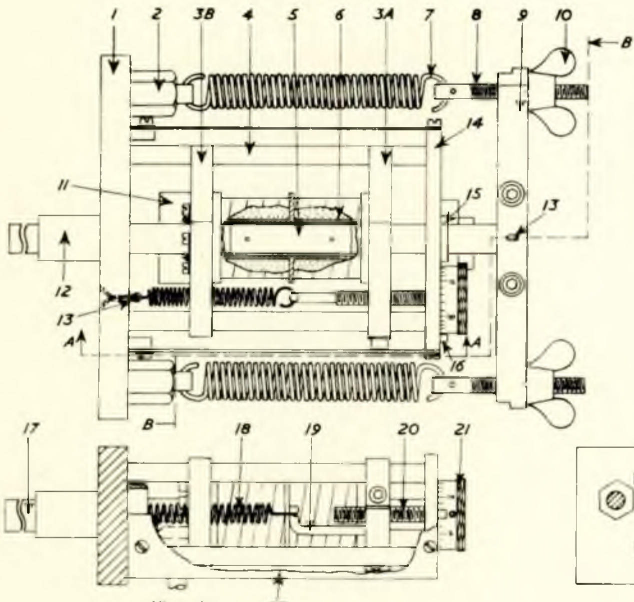
VECTORIAL REPRESENTATION

It is sometimes easier to obtain a mental picture of the physical effects from a vectorial representation than from formulæ, and Fig. 12 has been constructed to show the phase relationship between, and the relative magnitudes of, the propeller, thrust collar and thrust block movements, and the applied and resultant thrust variations over the speed range with zero, optimum and infinite damping in the resonance changer.

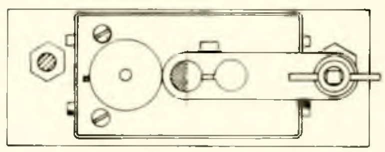
INSTRUMENTATION

It is not intended in this paper to go into the details of the instrumentation used, since this has been reported fully elsewhere by the Admiralty Engineering Laboratory, West Drayton. It is thought desirable, however, to give a broad indication of the means used for the measurements recorded.

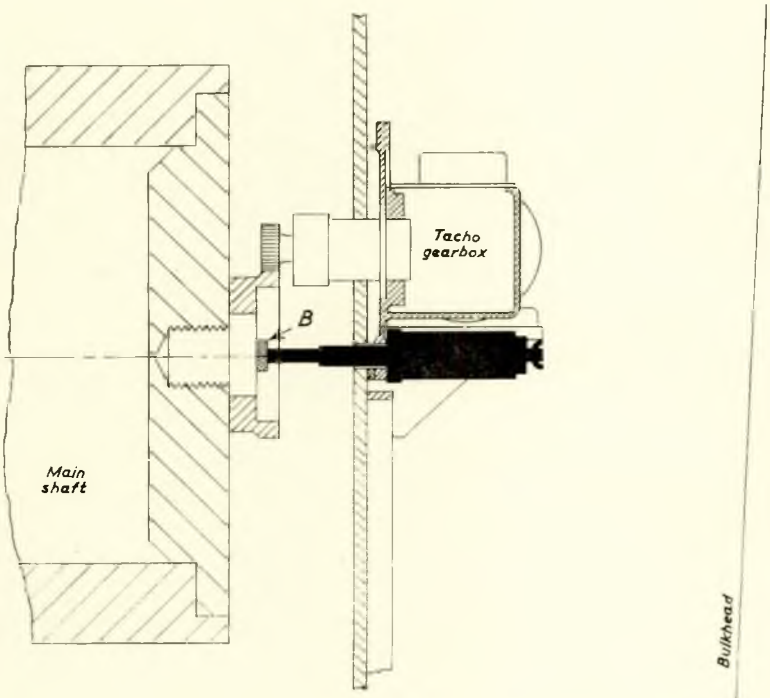
A multichannel pen recorder, constructed at A.E.L., was used for the simultaneous recording of the axial vibrations of



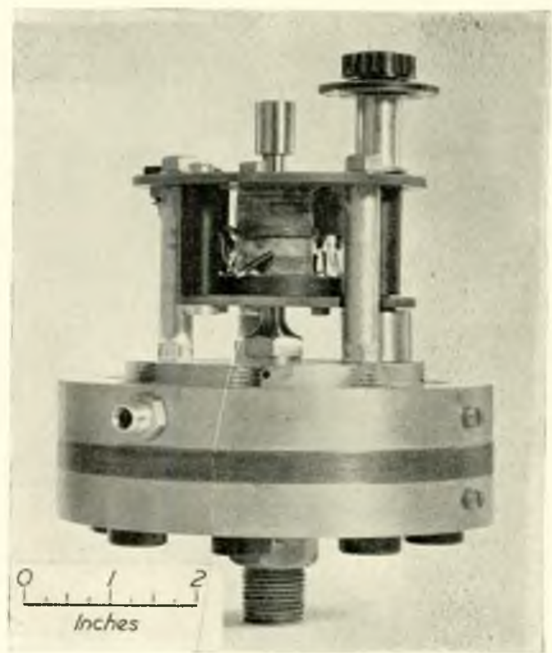
1) Baseplate; 2) Attachment nut; 3) Bobbin carrier; 4) Guide rod; 5) Slug; 6) Coil bobbin; 7) Rod spring; 8) Spring screw; 9) Spring arm; 10) Wing nut; 11) Cable clamp; 12) Baseplate bush; 13) 3/32-in. split pin; 14) End plate; 15) End plate bush; 16) Pointer; 17) Push rod; 18) Coil spring; 19) Spring hook; 20) Adjusting screw; 21) Adjusting knob; 22) Cover



AXIAL VIBRATION PICK-UP UNIT

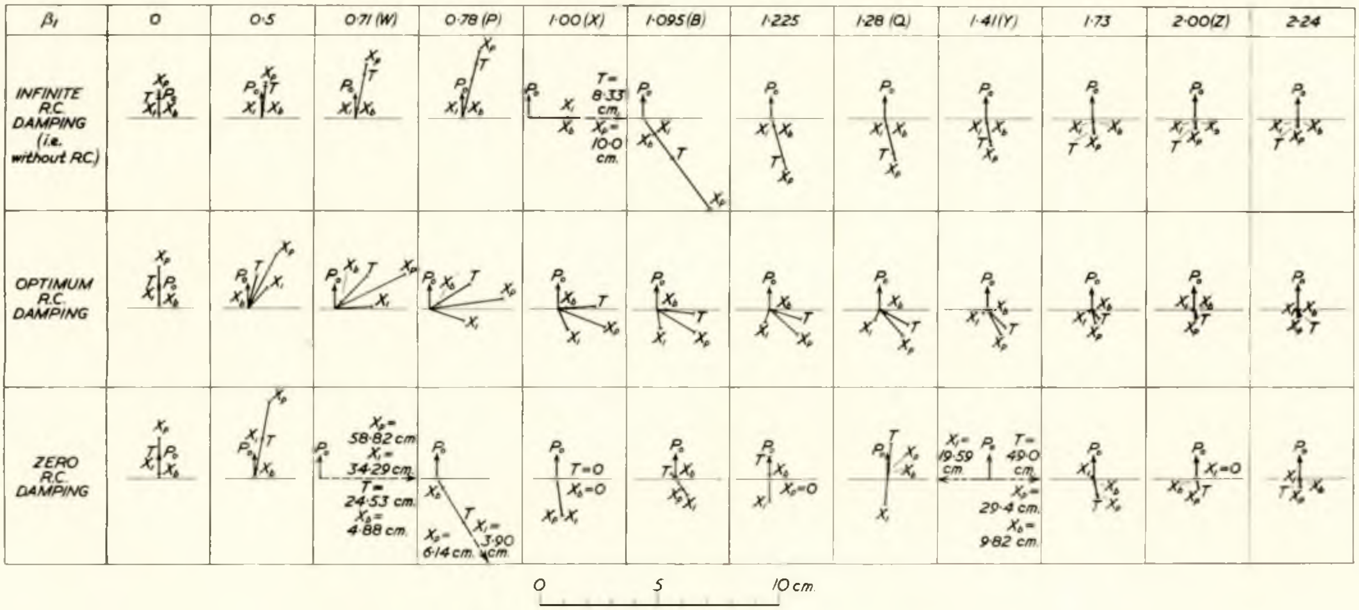


VIEW SHOWING POSITION OF AXIAL VIBRATION PICK-UP



VIEW OF PRESSURE PICK-UP WITH COVER REMOVED

FIG. 13



Basis: $q_1 = 0·5$ $\qquad \frac{K_B}{K_S} = 5 \qquad \beta_1 = \beta_0$

$\frac{a^2}{MK_0} = 0·015$ $\qquad$ e.g. $a = 1·0$ ton sec/in.

$M = 0·1$ ton sec²/in.

$K_B = 4{,}000$ ton/in.

Notes:

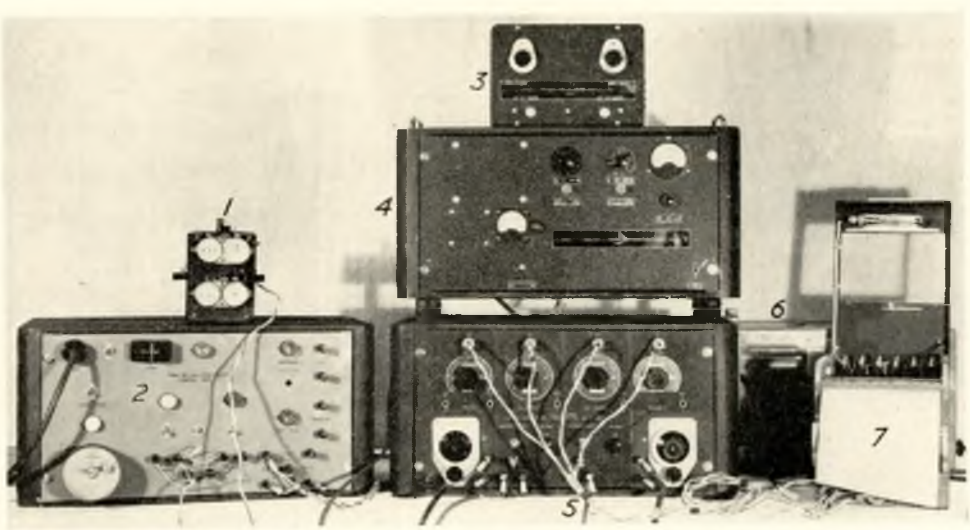
1) Letters given beside β_1 values correspond with those in Figs. 4, 9 and 10.

2) The vectors are drawn with a constant length for P_0. It should be noted that P_0 actually varies as β_1^2, so that to compare the relative values of each of the vectors at different frequencies they should be multipled by the appropriate β_1^2 value:

1 cm. of P_0 or T $= P_0$ tons

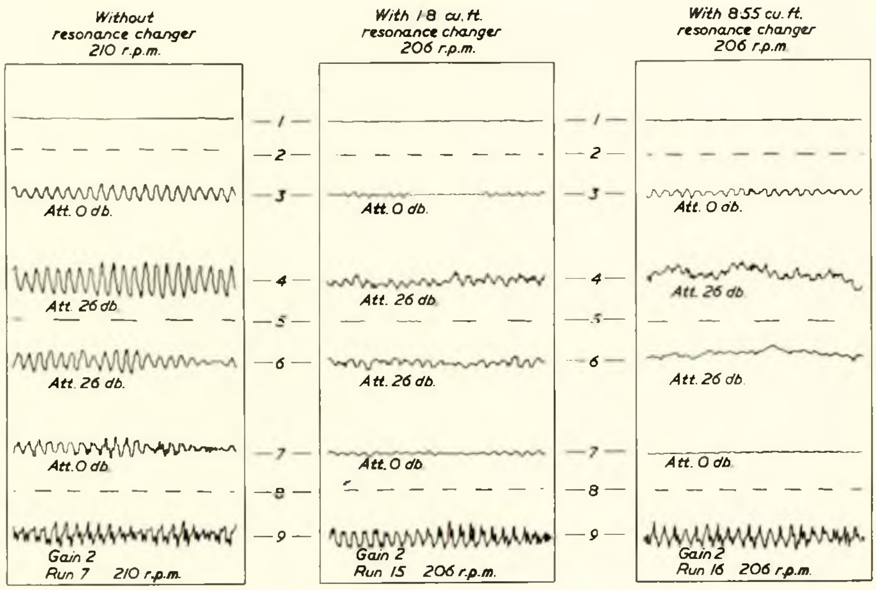
1 cm. of X_p, X_1 or X_b $= \frac{P_0}{K_S}$ in.

FIG. 12—*Vectorial representation of resonance changer operation*



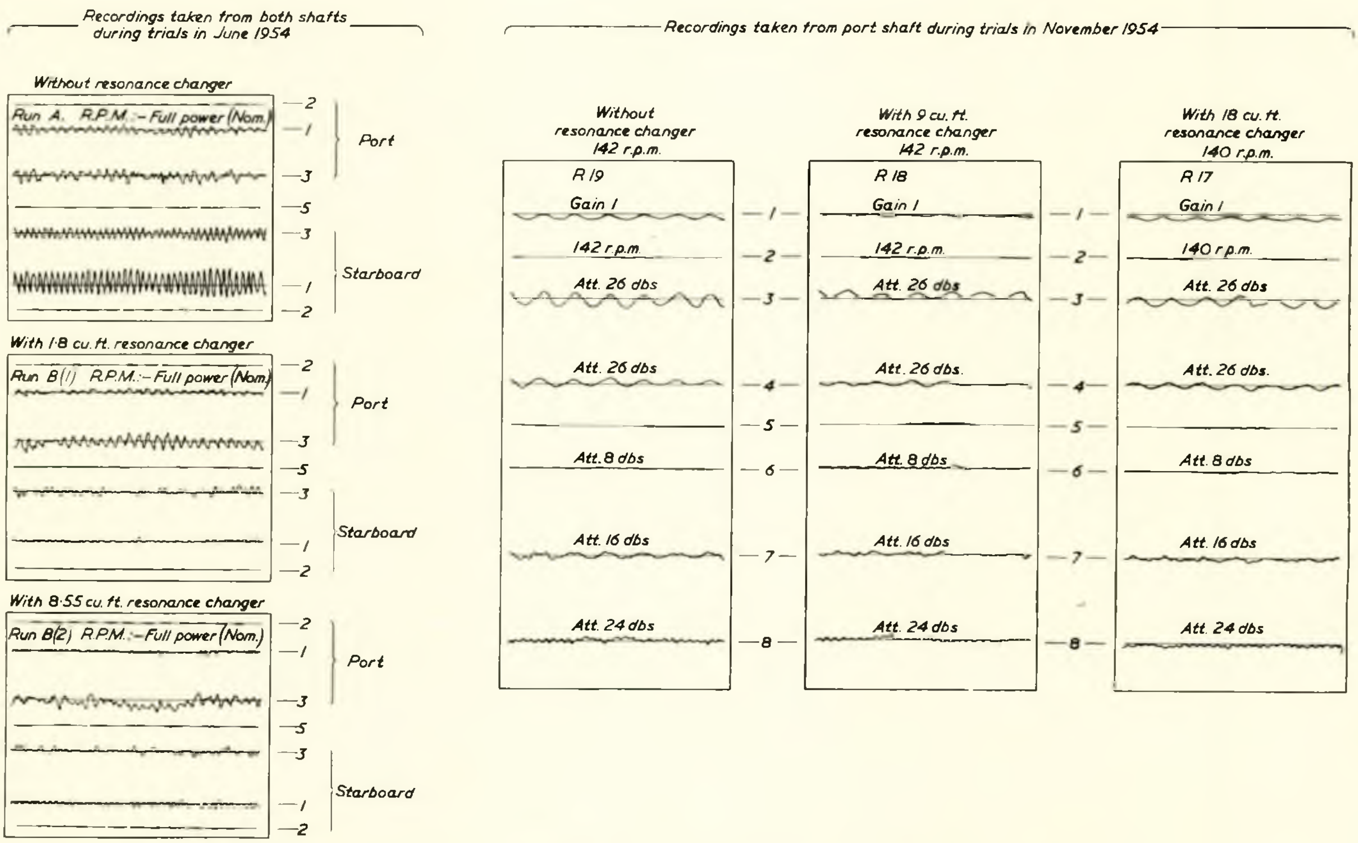
1) Revolution counters for inner shafts; 2) Timing and revolution indicator panel for counters and pen recorder; 3) Two-channel bridge for pick-ups on outer shaft thrust blocks; 4) 1,100-c.p.s. oscillator, channel pen amplifier and power pack; 5) Four-channel bridge and pen amplifier panel for inner shaft thrust and axial pick-ups; 6) Amplifier power pack; 7) Six-pen recorder

FIG. 14—*Instrument panel*



1) Port outer shaft thrust fluctuations; 2) Port inner shaft revolutions; 3) Port inner shaft thrust fluctuations; 4) Port inner shaft axial vibrations; 5) Timing mark (half second); 6) Starboard inner shaft axial vibrations; 7) Starboard inner shaft thrust fluctuations; 8) Starboard inner shaft revolutions; 9) Starboard outer shaft thrust fluctuations
Note: Resonance changers were not fitted to the outer shaft systems

FIG. 15—*H.M.S. Ark Royal: specimen pen recordings from straight course trials*



1) Thrust fluctuations at thrust block; 2) Shaft revolutions; 3) Axial vibrations of main gear wheel shaft; 4) Axial vibrations of shafting at loose coupling; 5) Timing mark (half second); 6) Axial vibrations of l.p. turbine (aft); 7) Axial vibrations of thrust block (casing joint); 8) Axial vibrations of gearcase (forward main bearing end cover)

FIG. 16—*H.M.S. Bulwark: specimen pen recordings from straight course trials*

the shafts and the thrust fluctuations on the thrust blocks. The pen recorder is fitted with magnets, moving coil units and pens supplied by Kelvin and Hughes (Industrial) Ltd. Each channel consists of a bridge balance unit, an amplifier, a demodulating circuit and a pen amplifier.

The method of recording the axial vibrations was based on the variation of inductance in a coil resulting from the longitudinal displacement of an iron core within the coil. The two matched coils (6) were mounted in line as shown in Fig. 13. The iron core (5) was mounted on an aluminium rod (17) and free to move longitudinally within the coils. The two coils formed the adjacent arms of a 1,100-cycle/second a.c. bridge. The other two coils had an adjustable iron core and were wired up in series with a variable apex resistance, thus enabling the whole bridge to be set off balance by any required amount. This control enables the apparatus to be initially set so that the maximum anticipated deflexion of the push rod will not drive the bridge through the balance point.

Over a limited distance, determined experimentally, the displacement of the core bears a linear relationship to the out of balance signal from the bridge. The units are mounted on the main gear wheel casings as shown in Fig. 13. By means of the spring loading device (7) the aluminium rod was held constantly in contact with the end of the main gear wheel shaft.

The pressure pick-ups used for recording the thrust fluctuations on the thrust blocks were developed and supplied by Vickers-Armstrongs Ltd. They consist of a pressure sensitive element embodying a diaphragm whose movement caused the displacement of a mumetal core in two coils energized by a 1,100-cycle/second carrier frequency. The deflexion of the diaphragm is proportional to the pressure difference across it and is linear over a range of 0—1,500 lb./sq. in. The pick-up unit is also shown in Fig. 13.

To obtain the revolutions of the shafts an inductive type pick-up was fitted. The inductance change in the coil operated relay contacts once per shaft revolution. By means of a transformer circuit the contacts were arranged to make revolution marks on the pen recorder.

By means of a clock timing unit a signal, once every half second, was fed to a central marking pen on the pen recorder.

A typical arrangement of instrument panels and pen recorder is shown in Fig. 14 and specimens of the type of records produced are shown in Figs. 15 and 16.

SUMMARY OF RESULTS

Table I gives tabulated data for H.M. Ships *Eagle, Ark Royal, Bulwark* and *Whitby*. *Vanguard* has been included in an endeavour to explain the critical speed that occurred at 152 r.p.m. instead of the expected 125 r.p.m.

Thrust Block Stiffness (K_B)

The thrust block stiffness values quoted for *Eagle, Ark Royal* and *Bulwark* are those obtained from the simultaneously measured values of X_1 and T without the resonance changer in use, whereas those for *Whitby* and *Vanguard* are estimates that are consistent with the measured results. In *Eagle* alone, there was a variation in stiffness, over the range of speeds tested, from 3,000 to 4,900 tons/in. In *Ark Royal* and *Bulwark* the stiffness remained constant over the whole speed range measured.

Shaft Stiffness (K_S)

The calculated shaft stiffnesses, between the thrust collar and propeller, are quoted. There seems little chance of these being significantly in error.

Mass (M)

The values of M included here are those that best fit the measured results for critical frequency. On the assumption that the mass of entrained water is 60 per cent of the propeller mass, values are given for the proportions of the shaft mass that have to be taken to suit the values of M derived from the records. *Vanguard* seems unduly low.

Mass (M_g)

The M_g values quoted are made up of the masses of the main gear wheel, associated pinions, thrust collar and shafting forward thereof, and half the mass of the thrust block casing. The calculations are not very sensitive to the value of M_g. The large value quoted for the three-bladed case of *Vanguard* includes the mass of the turbines and condensers.

Damping Factor at Propeller (a) *and Thrust Variation Fraction* (${}_TK_P$)

Although for the practical purposes of analysing the maximum amplitudes of thrust and displacements at the critical frequency, it is only necessary to determine the ratio ${}_TK_P/a$, values for each have to be established for analysis of measured results well away from resonance. The values of ${}_TK_P$ obtained are as follows:

3 blades	0·024 (*Ark Royal*)	0·020 (*Vanguard*)
4 blades	0·032 (*Eagle*)	0·036 (*Bulwark*)
5 blades	0·020 (*Vanguard*)	0·035 (*Whitby*—port)
		0·066 (*Whitby*—starboard)

(N.B. The values underlined are less well established than the others. *Bulwark* propeller is behind bossing; all others are behind A-brackets.)

In all cases, except *Whitby*, a good fit of the curves to the measured values is obtained by taking $a = 1{\cdot}0$ ton sec./in. The stern gland arrangements in *Whitby* differ from those in the other ships.

The damping inherent in the system is made up of propeller damping, shaft bracket, stern tube and bearing friction, shaft hysteresis, coupling friction and foundation hysteresis. It is convenient in the calculations, however, to refer all the damping to the propeller, although perhaps only half of the total is directly associated with the propeller. Kane and McGoldrick[3] give a formula for propeller damping which can be rewritten in the form:

$$a_p = \frac{N\,P\,D^2}{1{,}610{,}000} \cdot \frac{dC_t}{dS} \text{ ton sec./in.}$$

where C_t = thrust coefficient
S = real slip
N = r.p.m.
D = propeller diameter, ft.
P = propeller pitch, ft.

The required value for dC_t/dS is obtainable from Fig. 7 of their paper.

Values of this expression have been worked out for the various propellers referred to in this paper, but in every case the values of a_p so found is less than 0·5. On the basis of the trial results it seems that the effect of the other variables is too great to warrant an accurate approach and it is suggested that an assumption that the damping referred to the propeller (a) is approximately equal to 1·0 is adequate.

Ratio of Resonance Thrust Variation to the Mean Thrust at that Speed

Percentage straight course values are included for this, both measured and as calculated from the approximate formula:

$$\frac{T_c}{(M.T.)_c} \simeq \frac{{}_TK_P}{a} \cdot \frac{60K_S}{2\pi B_p N_c} \qquad (24)$$

The values calculated from equation (24) are listed below and these give reasonable agreement with the measured values.

	Twin-screw, per cent	*Quadruple-screw, per cent*
3 blades		31 (*Ark Royal*)
		23 (*Vanguard*)
4 blades	36 (*Bulwark*)	34 (*Eagle*)
5 blades	10 (*Whitby*—port)	19 (*Vanguard*)
	19 (*Whitby*—starboard)	

(N.B. The values underlined are less well established than the others.)

TABLE I.—SUMMARY OF TRIAL RESULTS

1	Ship	*Eagle*		*Ark Royal*				*Bulwark*				*Whitby*				*Vanguard*			
2	Number of shafts	4		4				2				2				4			
3	Number of blades on propeller	4		3				4				5				3		5	
4	Shaft on which trials were carried out	Both inners		Both inners				Port				Port		Starboard		St'b'd inner		St'b'd inner	
5	Block stiffness (K_B), tons/in.	3,000-4,900		2,900				2,500				2,000		2,000		2,000		4,500	
6	Shaft stiffness (K_S), tons/in.	825		825				567				856		856		750		750	
7	Mass (M), tons sec^2/in.	0·116		0·150				0·139				0·04		0·04		0·1		0·1	
8	Mass (M_P), tons sec^2/in.	0·066		0·079				0·069				0·025		0·025		0·065		0·065	
9	Mass of shaft (M_S), tons sec^2/in.	0·106		0·106				0·150				0·036		0·036		0·145		0·145	
10	Proportion of shaft mass to suit M	0·472		0·670				0·467				0·417		0·417		0·241		0·241	
11	Mass (M_g), tons sec^2/in.	0·092		0·092				0·090				0·01		0·01		0·24		0·072	
12	Critical speed (N_c), r.p.m.	185		206				136				231		231		208		152	
13	Damping at propeller (a), tons sec/in.	1·0		1·0				1·0				2·5		2·5		1·0		1·0	
14	Thrust variation factor (${}_TK_P$)	0·032		0·024				0·036				0·035		0·066		0·02		0·02	
15	$\frac{{}_TK_P}{a}$, in./ton sec.	0·032		0·024				0·036				0·014		0·026		0·02		0·02	
16	Measured ratio $\frac{T}{(M.T.)_c}$, per cent	32		27				32				—		—		—		18	
17	Approximate ratio $\frac{T_c}{(M.T.)_c}$, per cent (Based on equation (24))	34		31				36				10		19		23		19	
18	Dynamic magnification at resonance	10·1		11·1				9·0				2·1		2·1		13·2		9·0	
19	R.C. bottle volume (V_1), cu. ft.	9·0		1·8		8·55		9·0		18·0		0·082		0·082		—		—	
20	R.C. stiffness (k_1), tons/in.	466		2,320		490		466		233		1,800		1,800		—		—	
21	Stiffness ratio (q_1)	1·39		0·28		1·31		0·99		1·98		0·33		0·33		—		—	
22	R.C. mass (m_1), tons sec^2/in.	0·224		0·819		0·851		0·45		0·45		0·24		0·24		—		—	
23	R.C. tuning ratio: $\frac{\text{R.C. critical speed}}{\text{Original critical speed}} = \sqrt{\frac{k_1}{m_1}\frac{M}{K_0}}$	0·59		0·82		0·37		0·57		0·40		0·71		0·71		—		—	
24	R.C. damping (c_1), tons sec/in.	5	15	10	20	10	20	10	20	10	20	30	40	30	40	—	—	—	—
25	R.C. damping coefficient $\left(\frac{c_1^2}{m_1 k_1}\right)$	0·24	2·16	0·05	0·21	0·24	0·96	0·48	1·91	0·95	3·82	2·08	3·70	2·08	3·70	—	—	—	—

In the above formula N_c varies very approximately as $\sqrt{\frac{K_S}{M}}$ so that the ratio $\frac{T_c}{(M.T.)_c}$ varies as $\frac{{}_TK_P}{a} \cdot \frac{\sqrt{K_S M}}{B_p}$ very approximately and should decrease for:

(i) A multibladed propeller.
(ii) An axially soft propeller shaft.
(iii) A low propeller equivalent mass.

However, on the basis of a tenfold magnification factor due to turning, appropriate to inner propellers of quadruple-screw ships, thrust reversal would occur in all these cases, whereas a twofold magnification factor, appropriate to twin-screw ships, would not cause thrust reversal in any, but might still cause an unacceptable vibration.

It will be observed that the five-bladed propellers appear to have somewhat lower values for this ratio than the three- or four-bladed propellers.

Resonance Changer Constants (k_1, m_1 and c_1)

Calculated values are given for k_1 and m_1. In each case the two values for c_1 which have been chosen give calculated values which seem to straddle the measured results for thrust and displacement. Assuming oil at 80 deg. F. and equivalent length of pipe equalling double the actual length, values of c_1 have been calculated and these compare with the chosen two as follows:

	Calculated	*Chosen*
Eagle	9 ton sec./in.	5 and 15 ton sec./in.
Ark Royal	16 ton sec./in.	10 and 20 ton sec./in.
Bulwark	13 ton sec./in.	10 and 20 ton sec./in.
Whitby	50 ton sec./in.	30 and 40 ton sec./in.

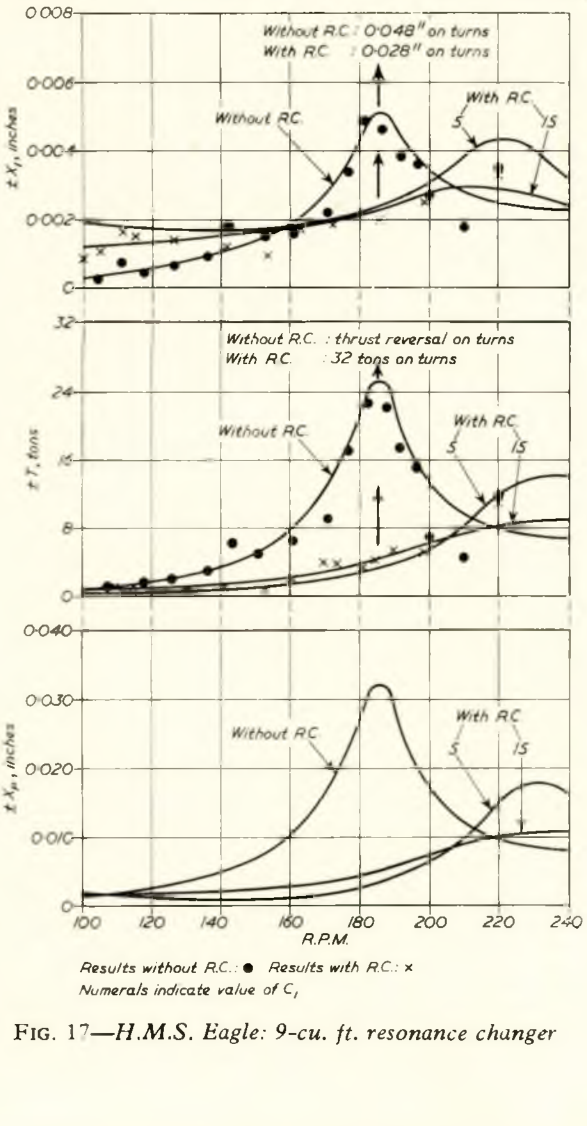


FIG. 17—*H.M.S. Eagle: 9-cu. ft. resonance changer*

Tuning Ratio of Resonance Changer

The tuning ratio of the resonance changer is the ratio of its critical speed to that of the original system and thus values of it can obviously be calculated from the expression

$$\sqrt{\frac{k_1}{m_1} \cdot \frac{M}{K_o}}.$$

Stiffness Ratio (q_1) and Damping Coefficient (c_1^2/m_1k_1)

Values for q_1 and c_1^2/m_1k_1 are included in the table for comparison with the recommended values given in the paper.

DISCUSSION OF RESULTS

H.M.S. Eagle

Fig. 17 shows calculated values for X_1, T and X_p, together with experimental points for X_1, the measured amplitude of displacement at the forward end of the gear wheel, and for T, the measured amplitude of thrust variation at the thrust block. As in all the trials carried out, there is some scatter of points and this is believed to be real and caused by the disturbing thrust at the propeller varying somewhat due to the effect of waves and small movements of the rudder.

The tuning ratio of the resonance changer of 0·59 permits larger values of thrust and displacement above the critical speed than would occur if the now recommended value of 1·0 were to be used. The higher value of damping gives less tendency to peak at speeds above the original critical speed than the lower value. The former gives a value for c_1^2/m_1k_1

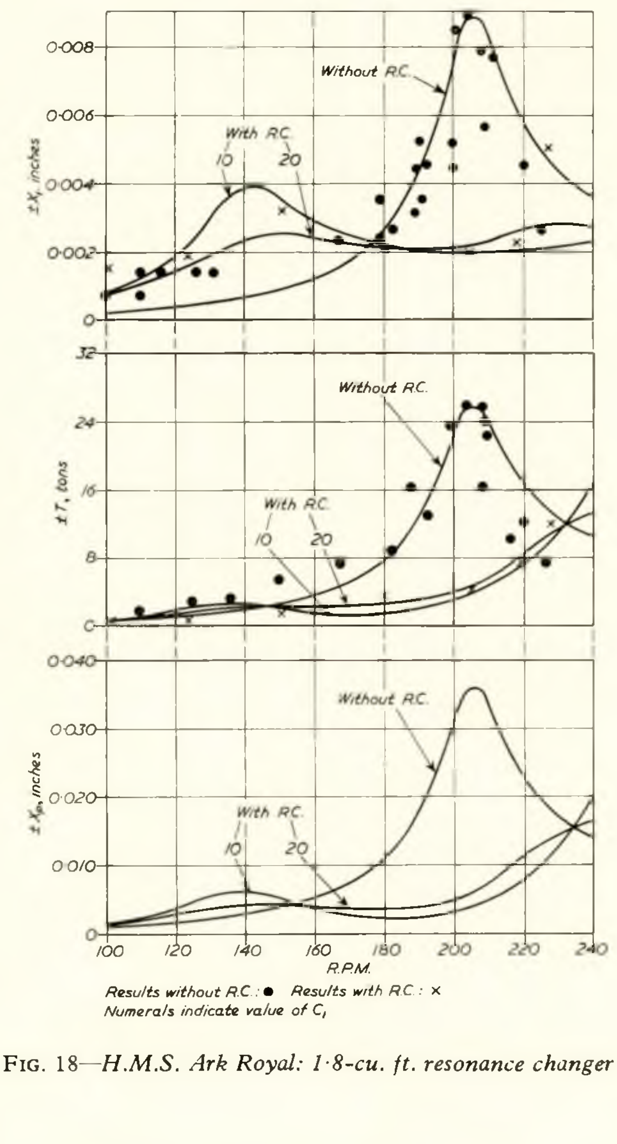


FIG. 18—*H.M.S. Ark Royal: 1·8-cu. ft. resonance changer*

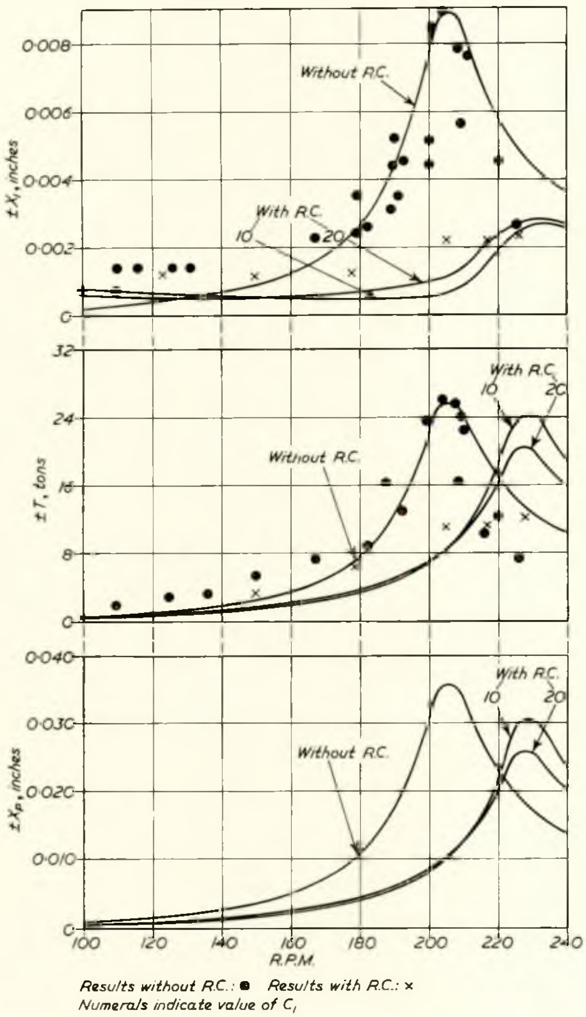


FIG. 19—*H.M.S. Ark Royal 8·55-cu. ft. resonance changer*

of 2·16 and, from Fig. 5, it can be seen that this is more appropriate to the q_1 value of 1·39 than the value for the lower damping.

The effect of turns was measured during the trials in *Eagle* and the following results were obtained at 185 r.p.m.

Straight course	*Without* resonance changer	*With* resonance changer
Alternating thrust at thrust block	± 25 tons	± 4 tons
Amplitude at gear wheel shaft	±0·005in.	±0·0024in.
Turning maxima		
Alternating thrust at thrust block	>±121 tons (thrust reversal)	±32 tons
Amplitude at gear wheel shaft	±0·048in.	±0·028in.

On this basis it was considered that allowance of up to 10:1 should be made for a magnification on the turn. This value was confirmed in *Ark Royal* trials.

H.M.S Ark Royal

Figs. 18 and 19 show calculated values for X_1, T and X_p, together with measured values where taken, both with and without resonance changers having bottle sizes 1·8 cu. ft. and 8·55 cu. ft. respectively. The corresponding q_1 values are 0·28 and 1·31. The tuning ratios are 0·82 and 0·37 respectively.

Comparison of the results with the two bottle sizes shows that the potential advantage of the larger bottle has been thrown away by the use of too low a value of the tuning ratio, which brings the intersections with the curves without the resonance changer at too low speeds and therefore at too high values of thrust and displacement. The results with the smaller bottle suffer to a lesser extent in the same manner.

The higher of the values of damping for each case gives the coefficient c_1^2/m_1k_1 values of 0·21 and 0·96 and these can be seen to be well below the optimum values for the respective q_1 values. The curves illustrate the peaks resulting from such underdamping.

H.M.S. Bulwark

Fig. 20 shows calculated values for X_1, T and X_p together with measured values, both with and without the resonance changer with a 9-cu. ft. bottle. Fig. 21 shows similar results with an 18-cu.ft. bottle capacity. The corresponding q_1 values are about 1·0 and 2·0 and the tuning ratios are 0·57 and 0·40 respectively.

With both these resonance changers, the low value of tuning ratio has resulted in higher peaks above the original critical speed than would have been the case with a tuning ratio of 1·0, and the potential advantage of the 18-cu. ft. bottle has been lost through having a very low value of this ratio.

The higher of the damping values shown in each case gives c_1^2/m_1k_1 values of 1·91 and 3·82 respectively and these approach the values now recommended for the q_1 values that apply. The curves in Figs. 20 and 21 indicate that they are nearly optimized.

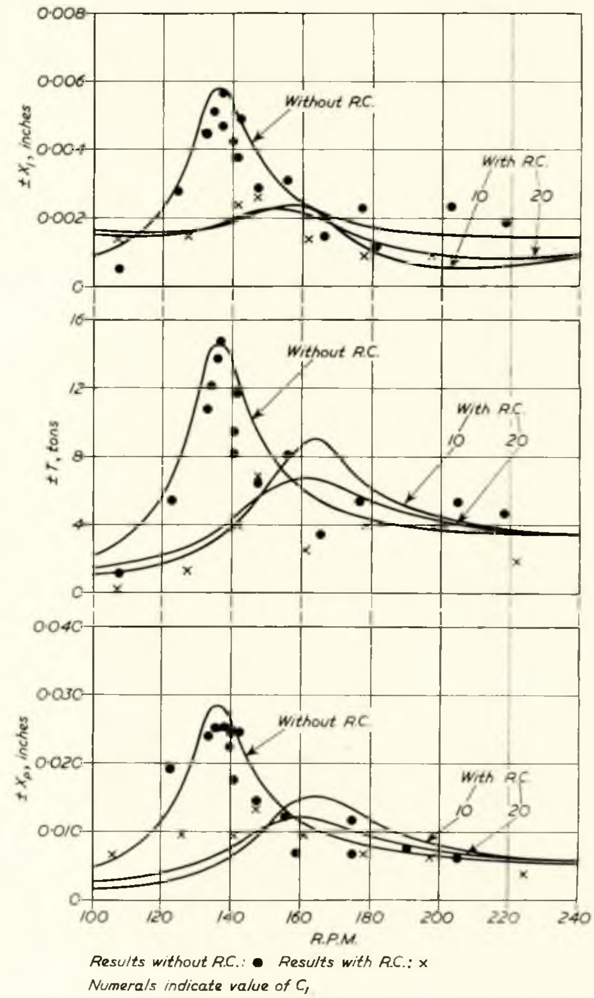


FIG. 20—*H.M.S. Bulwark: 9-cu. ft. resonance changer*

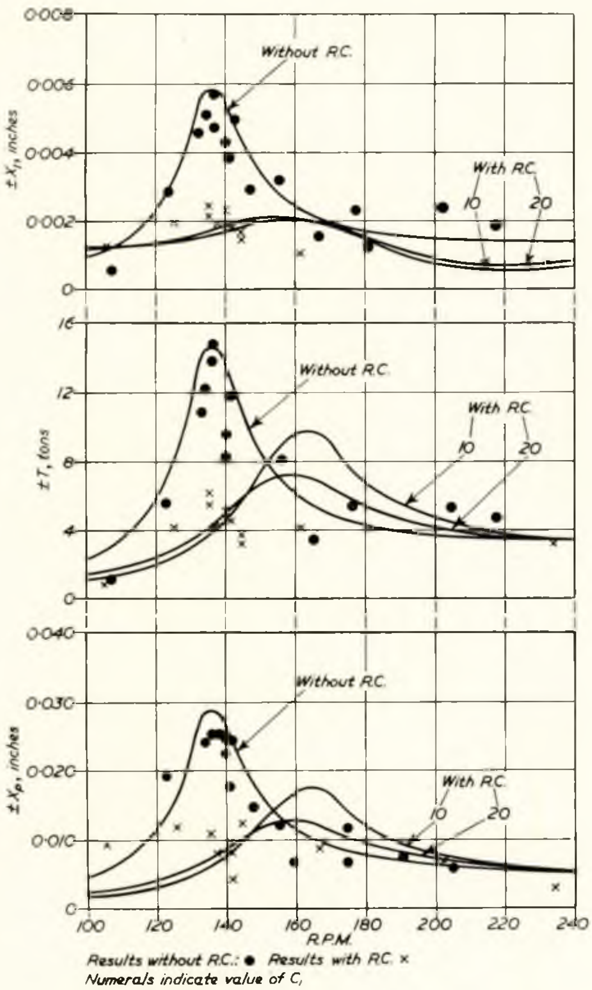


FIG. 21—*H.M.S. Bulwark: 18-cu. ft. resonance changer*

It should be noted that the plotted values for displacement at the propeller were taken as far aft inboard on the propeller shafts as possible and should, therefore, be slightly lower than the theoretical values.

H.M.S. Whitby

Figs. 22 and 23 show calculated values for X_1, T and X_p for the port and starboard shafts respectively, both with and without the resonance changers, both of which have a 0·082-cu. ft. bottle giving a q_1 value of 0·33. Measured values were obtained only for thrust variations and there is, therefore, no check on the value assumed for the thrust block stiffness K_B. Contrary to expectation the critical speed does not seem to occur within the running range, and this makes analysis of the results somewhat uncertain.

The values measured on the starboard shaft are pronouncedly greater than those on the port shaft, and it seems more likely that this is attributable to differences in the propeller, causing differences in $_TK_P$, than to differences in damping. The unduly high value of the latter that has had to be assumed to conform to the measured points may be due to the use of oil lubricated stern gland bearings with radial seals, whereas the stern gland bearings in all the other ships were water lubricated with circumferential packing.

There is some indication of a minor peak at about 186 r.p.m. with the resonance changer not in use. The only explanation suggested for this is that a larger number of records were taken at this speed, since a critical had been expected here. There is a natural tendency to measure the largest amplitudes appearing on any records, and the longer the period during which the records are taken the greater the chance of above average amplitude, due to variable waves, use of rudder, etc.

The c_1 values of 30 and 40 that best fitted the recorded measurements are less than the value of c_1, calculated on a basis of the equivalent length of pipe being double the actual length. It seems probable that this is due to resistance within the thrust block in this ship being a smaller proportion of the large resistance in the pipe itself.

The resonance changers are relatively ineffective in this case due to the excessive damping, as illustrated by the values of c_1^2/m_1k_1 being 2·1 and 3·7 instead of 0·66, which is the optimum corresponding to the q_1 value of 0·33. The resonance changers are also undertuned.

H.M.S. Vanguard

The information for H.M.S. *Vanguard* is rather sparse. At the time of the *Vanguard* trials instruments were not well developed and all that can be established is that, when originally fitted with three-bladed propellers, there was a serious critical speed at 208 r.p.m. giving rise to an X_1 amplitude of 0·011in. on a straight course. When fitted with five-bladed propellers, there was a critical speed at about 152 r.p.m. and the vibration was so severe that thrust reversal occurred when turning and thus caused large amplitude shuttling of the pinions and in the turbine flexible coupling. An axial amplitude of about an eighth of an inch was rather crudely measured on the aftermost inboard portion of the shafting.

Only one way has been found to reconcile these results.

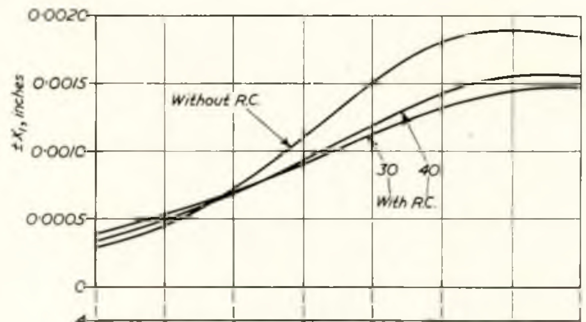


FIG. 22—*H.M.S. Whitby (port shaft): 0·082-cu. ft. resonance changer*

This is to assume that looseness in the thrust block attachment was taken up at the same time as the five-bladed propellers were fitted, leading to an increase of thrust block stiffness from 2,000 to 4,500 tons/in. and that when the critical speed of 208 r.p.m. was occurring with three-bladed propellers the flexible couplings were failing to slide, and thus causing h.p. and l.p. turbines and condensers to take part in the vibrations.

The first phenomenon, that of thrust block attachments working loose, has occurred in light fleet carriers, and the second phenomenon of there being two types of vibration, one with the flexible couplings slipping and one with them gripping, has been clearly detected in trials carried out in H.M.S. *Savage,* and to a lesser extent during trials in H.M.S. *Ark Royal.*

Based on these assumptions, the values shown in Table I are estimated and the curves shown in Fig. 24 derived. The dotted lines shown with the curves for thrust variations represent:

a) Mean thrust divided by five.

b) Mean thrust divided by ten.

For thrust reversal to occur, the peak of thrust variation amplitude must exceed the dotted line (*a*) if the turn magnification factor is 5 or the dotted line (*b*) if it is 10. From these, it appears that the five-bladed propeller might have been satisfactory, if the factor of 5 had been true, but that as it was in fact about 10 thrust reversal was bound to occur.

H.M.S. Savage

A number of trials with different resonance changer arrangements were carried out in H.M.S. *Savage* during 1952/1954 to find the effect of the different variables. This ship was chosen because she was being used as a trial ship for different propellers, and was therefore well instrumented and trials on the resonance changers could easily be fitted into the programme. Unfortunately the axial critical speed fell just outside the running range with all the propellers used, so that the results were not suitable for accurate analysis, and are not therefore included in this paper. The records included vibration readings on the turbines and gear case and were valuable in demonstrating the occurrence of two types of vibration, one with the flexible couplings slipping and the other with them gripping. It is not clear what factor governed whether or not they would slip, and change from one type of vibration to the other seemed to take place at random.

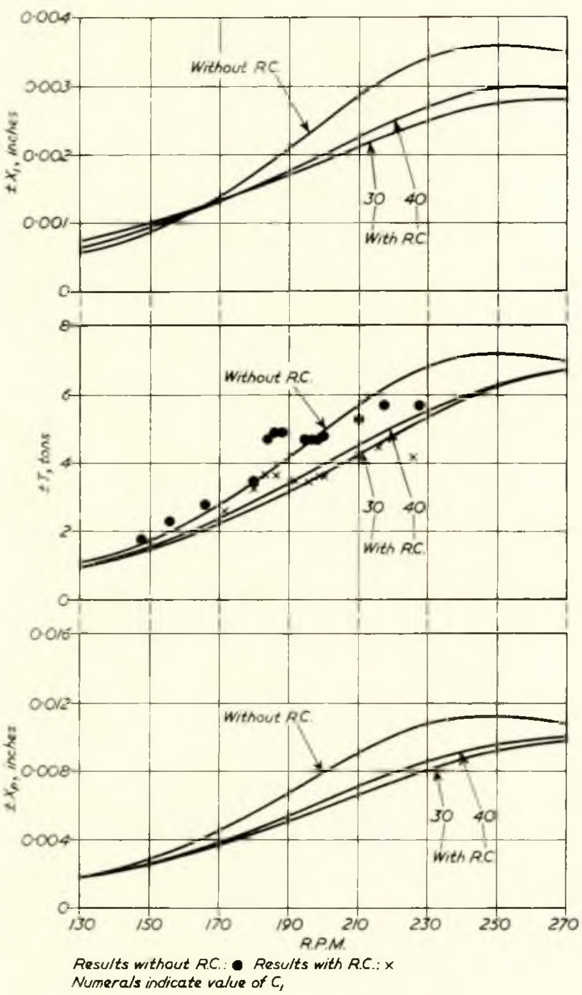


FIG. 23—*H.M.S. Whitby (starboard shaft): 0·082-cu. ft. resonance changer*

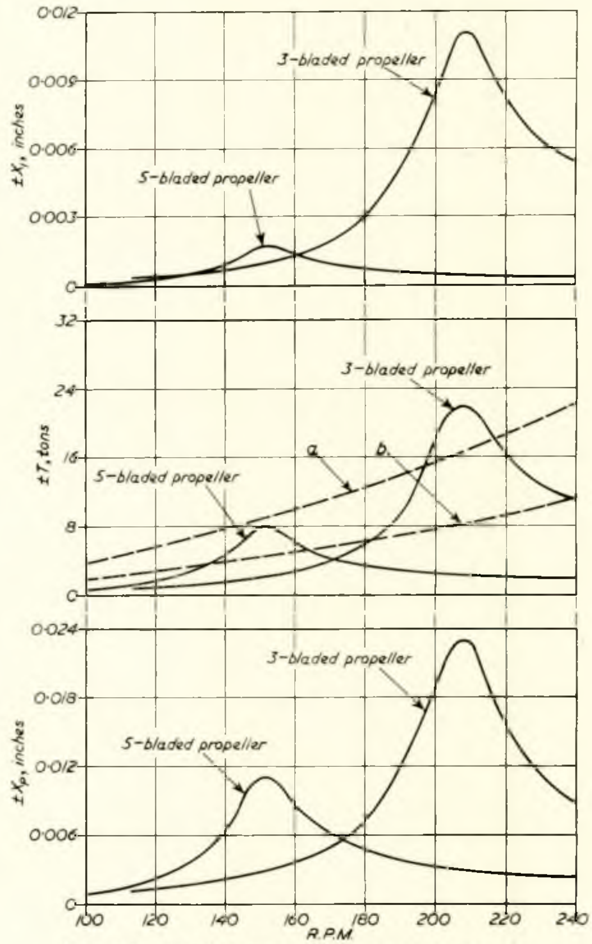


FIG. 24—*H.M.S. Vanguard: expected results*

Simultaneous readings of thrust and gear wheel shaft displacement, without the resonance changer in use, gave a value for K_B of 2,140 tons/in.

H.M.S. Triumph

Loosening of thrust block securities due to heavy axial vibration in service occurred on both shafts in H.M.S. *Triumph,* each with three-bladed propellers.

On the port shaft, where the critical speed had previously been above full power revolutions, it had reduced to 186 r.p.m., indicating a reduction in thrust block stiffness from 2,200 to 900 tons/in., resulting in a measured displacement amplitude of 0·017in. at the gear wheel on a straight course.

On the starboard shaft the critical speed was reduced from 180 to 162 r.p.m., indicating a reduction of thrust block stiffness from 1,800 to 1,080 tons/in., resulting in a measured vibration of 0·021in. on a straight course.

RECOMMENDATIONS FOR NEW CONSTRUCTION

In any new design of the propeller shafting system for a ship, it is desirable to investigate the possibility of trouble from axial vibration. However, since a resonance changer can be depended upon to overcome any excessive vibration, there is no need for the design to depart from the number of propeller blades, and the thrust block position, most suitable from other considerations.

In calculating the critical speeds of the system, the Holzer tabular method used by Rigby is satisfactory, or the expression given in equation (35) in Appendix 2 can be used. That is:

$$\tan Z_c = \frac{\frac{K_B}{K_S} - \left(\frac{M_P}{M_S} + \frac{M_F}{M_S}\right)Z_c^2}{Z_c\left[1 + \frac{M_P}{M_S}\left(\frac{K_B}{K_S} - \frac{M_F}{M_S}Z_c^2\right)\right]}$$

Hence, on solving this equation, the critical speeds may be derived since $Z_c = \sqrt{\frac{M_S\omega_c^2}{K_S}}$ and $N_c = \frac{60\omega_c}{2\pi B_p}$.

The solution in the first quadrant gives the first critical speed, and the solution in the second quadrant gives the second critical speed.

As calculations by even the most meticulous methods are based on assumptions, there is a degree of uncertainty in the critical speed estimated. Errors of 10 per cent or more can easily occur, so that unless the value so calculated is 40 per cent above the maximum running speed the possibility of troublesome axial vibration must be faced. The value of 40 per cent allows not only for errors in the estimate, but also for increases of shaft r.p.m. when turning and the desirability of keeping away from the build-up of vibration as the critical frequency is approached.

Approximate methods can be used to show that the first critical is determined mainly by K_S and M_P and the second by K_B and M_F, and that, as suggested by Rigby, provided the thrust block stiffness is sufficiently high to prevent any possibility of the second critical speed lying within the running range, it is not worth devoting a great effort to increasing the value of K_B.

K_S can be taken as $\left[160\cdot5 \times 10^3 \times \frac{A_S\ (\text{sq.ft.})}{L_S\ (\text{ft.})}\right]$ tons/in., and to obtain K_B, the thrust block stiffness between the thrust collar and the seating, the following have been suggested:

Rigby: Full power mean thrust (tons) ÷ 0·035 to 0·057 (in.)	
Kane and McGoldrick:	2,050 to 5,750 tons/in.
This Paper:	2,000 to 4,900 tons/in.

Any of these give satisfactory answers when guided by experience.

Ships with Three or More Shafts

Unless the propellers are abreast, which is most unlikely, there is a strong possibility of the after propellers having to work partly in the slipstream of the forward propellers when turning, with a resultant magnification of the disturbing thrust variation of up to 10 rather than the earlier quoted factor of 5, and furthermore the thrust on the after propeller tends to be reduced when it is acting in slipstream. When these values are applied to the straight course resonant thrust variations of from 19 per cent to 36 per cent of the mean thrust, it confirms experience that there is serious danger of thrust reversal and heavy vibration in the inner shafts when turning unless the critical speed is well outside the running range.

For inner shafts it therefore seems imperative to make provision for resonance changers, and design guidance is given in Appendix 4.

As shown in equation (23), the possibility of thrust reversal occurring when a tuned resonance changer has been fitted can be estimated from a knowledge of ${}_TK_P$ and q_1.

The following values have been quoted for ${}_TK_P$:

Rigby		
3 blades	0·03 (behind A-brackets) to 0·05 (centre shaft)	
4 blades	0·023 (behind A-brackets) to 0·038 (centre shaft)	
5 blades	0·018 (behind A-brackets) to 0·03 (centre shaft)	
Kane and McGoldrick		
3, 4 or 5 blades	0·02 to 0·05 (behind A-brackets)	
3 blades	0·08 to 0·12 (single-screw)	
4 or 5 blades	0·03 to 0·08 (single-screw)	
This Paper		
3 blades	0·024 (*Ark Royal*)	behind A-brackets
	0·020 (*Vanguard*)	behind A-brackets
4 blades	0·032 (*Eagle*)	behind A-brackets
	0·036 (*Bulwark*)	behind bossing
5 blades	0·020 (*Vanguard*)	behind A-brackets
	<u>0·035 to 0·066 (*Whitby*)</u>	behind A-brackets

(N.B. The values underlined are less well established than the others.)

Assuming a turn magnification of up to 10 and a value for ${}_TK_P$ of 0·05 as the largest ever likely to be applicable with this magnification, it is clear that q_1 must be chosen sufficiently large to prevent $\frac{T}{(M.T.)}$ (max.) exceeding $2\cdot0\,{}_TK_P$. It was shown earlier that a q_1 value of 1·0 suffices for this, even neglecting the beneficial effect of propeller damping. It is recommended that this value for q_1 be used for resonance changers on the inner shafts of triple and quadruple-screw ships, unless it is known that conditions are more favourable than those assumed. The modern trend towards higher thrust-meter pressures, and hence smaller values of A_o, should prevent excessive bottle sizes.

From Fig. 7 it can be seen that a value of $q_1 = 1\cdot0$ gives a value for N/N_c of 1·41, and, therefore, with a tuning ratio of 1·0, the calculated value of T_m will occur at less than full speed if N_c is less than 71 per cent of N_{FP}.

If N_c is greater than 71 per cent of N_{FP}, the calculated value of T_m will not occur unless the shaft overspeeds. Shaft speeds up to 120 per cent of the straight course full speed have been measured in the inner shafts of quadruple-screw ships when turning, so that the maximum value could occur if N_c is 85 per cent of N_{FP}. Where N_c is a higher percentage than this some advantage may be gained by adopting a tuning ratio of less than 1·0, since this brings the point P in Fig. 4 and A in Fig. 10 to a lower value, and the higher values at the points Q and B respectively will be outside the running range. The optimum damping for this condition can be calculated in a similar manner to that shown in Appendix 3 for a tuning ratio of 1·0.

The conditions for the outer shafts are very similar to those for two-shafted ships and the recommendations given below for the latter apply also to outer shafts of three- or four-shafted ships.

Ships with Two Shafts

Since the propellers in twin-screw ships are not called upon to work in the slipstream of other propellers, the disturbing thrust variation is not increased when turning to anything like the extent to which it is in multiscrew ships, and further measurements have confirmed the turn factor of 2 given by Rigby. While manœuvring, however, the propellers may be in disturbed water and subjected to larger disturbing thrust variations similar to the increased torque variations which magnify torsional oscillations under these conditions, but the axial critical speed, unlike the torsional, is usually outside the manœuvring range.

There seems, therefore, little danger in twin-screw ships of thrust reversal and the resultant heavy vibration, but there have been cases where the thrust variation at resonance has been sufficiently severe to loosen the securing arrangements for the thrust block and thus lead to excessive vibration. In one reported case, where the original critical speed was just above full power, this loosening effect reduced the thrust block stiffness sufficiently to bring the critical speed down into the running range.

If the critical speed is likely to lie within the running range, or just outside it, the decision whether or not to accept the resultant vibration must depend on judgement. Where there are reasons for wishing to minimize vibration, or in cases of doubt, it is suggested that provision be made in the design stages for the fitting of resonance changers and that sea trials be carried out to determine whether or not it is necessary to use them. The thrust block should be provided with a thrustmeter, which should have adequate internal passages and axial clearance, and suitable connecting pipes and oil bottles should be fitted. For trial purposes the hand pump or a temporary power driven pump could be used, pending decision on whether the resonance changer is essential.

In choosing a q_1 value for the resonance changer, a compromise has to be reached between unduly bulky oil bottles and inadequate effect, and the following may be helpful in reaching a decision:

$q_1 = 0{\cdot}25$ reduces the maximum thrust variation to about 30 per cent of the value at resonance without a resonance changer.

$q_1 = 0{\cdot}5$ reduces the maximum thrust variation to about 25 per cent of the value at resonance without a resonance changer.

$q_1 = 1{\cdot}0$ reduces the maximum thrust variation to about 20 per cent of the value at resonance without a resonance changer.

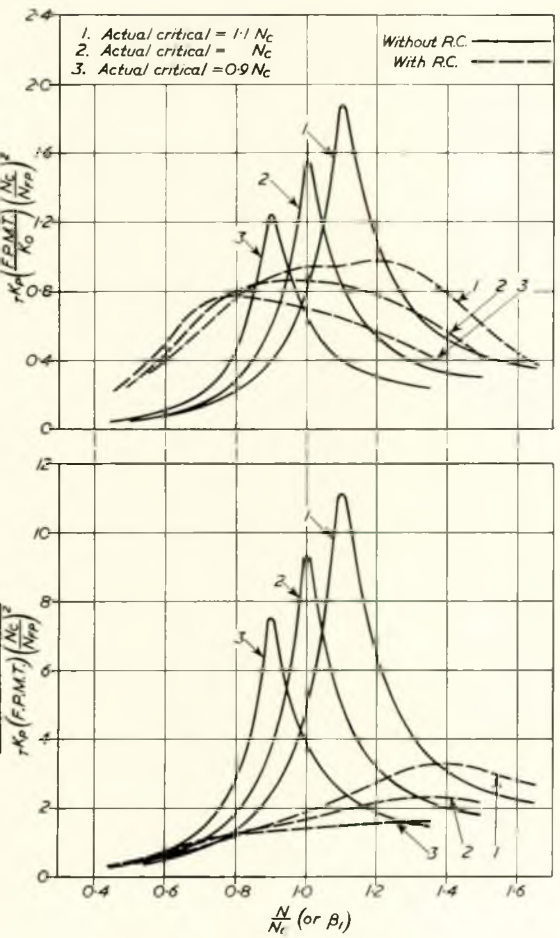
Where economy of size is of great importance, a value of q_1 of 0·25 should suffice, but in practice this may need to be increased, since it may not be feasible to design connecting pipes with sufficiently low damping and adequate effective mass without the volume in the pipes being too large in relation to the bottle size for the system to work as intended.

Viscosity Control

The design problem can be eased by reducing the viscosity of the oil in the pipes, and where there is a temperature controlled hot oil supply to the thrust block, this may be used to jacket the connecting pipes to keep down the viscosity. Where wide variations in engine room temperatures are expected, such a means for controlling the temperature, and hence the viscosity, of the oil in the pipes might in any case be desirable, since a change in oil temperature from 80 deg. F. to 120 deg. F. at the thrustmeter pressures reduces the viscosity to a quarter of its original value. Alternatively, a viscostatic oil could be used, which would ensure satisfactory damping values, but would introduce the need for carrying a special oil on board and for ensuring that it was compatible with the main lubricating oil, since some slight leakage must be expected from the thrustmeter cylinders into the thrust block proper.

Tolerance to Errors in Design Data

In view of the inability to obtain accurate estimates of



Basis: $q_1 = 0{\cdot}5 \quad \frac{K_B}{K_S} = 5 \quad \frac{a^2}{MK_o} = 0{\cdot}0115$

$\frac{c_1^2}{m_1k_1} = 1{\cdot}07$ (optimum)

FIG. 25—*Effects of error in calculated critical frequency*

Basis: $q_1 = 0{\cdot}5 \quad \frac{K_B}{K_S} = 5 \quad \frac{a^2}{MK_o} = 0{\cdot}0115$

$\beta_1 = \beta_o$

FIG. 26—*Effects of errors in M_g and in c_1^2/m_1k_1*

some of the factors affecting axial vibration, it is comforting to realize that the performance of the resonance changer is not seriously prejudiced by quite wide departures from the assumed condition.

Fig. 25 shows the effects of errors of ± 10 per cent in the calculated critical frequency of the system. Fig. 26 shows respectively the effects of:

(i) Change in the effective mass at the thrust collar (M_g) from zero to 100 per cent of the effective mass at the propeller (M).

(ii) Halving and doubling the intended damping in the resonance changer.

It will be seen from these figures that, provided reasonable care is given to the practical details, it is almost impossible to design a bad resonance changer, which accounts for the immediate success of the resonance changers first fitted in H.M.S. *Eagle.*

ACKNOWLEDGEMENTS

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Lastly, the author's thanks are due to his colleagues at the Yarrow-Admiralty Research Department, and particularly Mr. M. M. Johnston, in all the detailed calculations necessary in the preparation of this paper.

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Appendix 1

List of symbols

A_0	Total cross sectional area of thrustmeter cylinders.
A_1	Cross sectional area pipe bore.
A_S	Cross sectional area of shafting (ignoring liners and couplings).
B_1	Bulk modulus of oil in R.C. system.
B_p	Number of blades on propeller.
$F.P.M.T.$	Full power mean thrust.
K_0	Axial stiffness of original system (i.e. shafting plus block) $= \dfrac{K_B K_S}{K_B + K_S}$
K_B	Axial stiffness of thrust block (including foundation).
K_S	Axial stiffness of line of shafting.
${}_TK_P$	Thrust variation at propeller as a fraction of mean thrust.
L_1	Length of pipe.
L_E	Hydraulic equivalent length of R.C. system.
L_S	Length of shafting between the thrust collar and the propeller.
M_P	Mass of propeller plus cone, nut, tail end of shaft and allowance for entrained water (i.e. 60 per cent of propeller mass).
M_F	Mass of main gear wheel, associated pinions, thrust collar and shafting forward thereof, and half the mass of the thrust block casing.
M_S	Mass of shafting between the thrust collar and the propeller.
M	Effective mass at propeller $= M_P + R_1 M_S$
M_g	Effective mass at thrust collar $= M_F + R_2 M_S$
$M.T.$	Mean thrust at any particular shaft speed.
$(M.T.)_c$	Mean thrust at N_c.
$(M.T.)_m$	Mean thrust corresponding with T_m.
N	Any particular shaft speed.
N_c	Critical shaft speed of original system.
N_{FP}	Shaft speed at full power.
P_0	Amplitude of thrust variation at propeller at any speed.
R_1	Proportion of M_S to be added to M_P in equivalent system.
R_2	Proportion of M_S to be added to M_F in equivalent system.
T	Amplitude of thrust variation at thrust collar at any speed.
T_c	Value of T (with R.C. out) occurring at N_c.
T_m	Maximum value of T occurring when R.C. is in use.
V_1	Capacity of oil bottle.
X_0	Displacement of propeller, with R.C. out, under a static force equal to P_0 $= \dfrac{P_0}{K_0}$
X_1	Amplitude of x_1.
X_B	Displacement of block face, or thrust collar with R.C. out, under a static force equal to P_0 $= \dfrac{P_0}{K_B}$
X_b	Amplitude of x_b.
X_p	Amplitude of x_p.
X_r	Amplitude of x_r.
Z	$\sqrt{\dfrac{M_S \omega^2}{K_S}}$
Z_c	$\sqrt{\dfrac{M_S \omega_c^2}{K_S}}$
a	Damping factor of the original system referred to the propeller.
c_1	Virtual damping factor of the R.C. system.
d_1	Bore of pipe.
d_m	Minimum permissible bore of pipe (if viscosity is 100 centipoise and L_E/L_1 is unity).
j	Operator $\sqrt{-1}$.
k_1	Virtual stiffness of the R.C. system.
m_1	Virtual mass of the R.C. system.
m_s	Mass per unit length of shafting.

p	Instantaneous pressure in thrustmeter cylinders.
q_1	Stiffness ratio $= \dfrac{K_o}{k_1}$
r	Distance of any element of shafting from the propeller.
t	Time.
X_1	Instantaneous displacement of thrust collar face (or thrustmeter pistons assuming a rigid connexion between pistons and collar face) from its steady state position.
X_b	Instantaneous displacement of thrust block face from its steady state position.
X_p	Instantaneous displacement of propeller from its steady state position.
X_r	Instantaneous displacement of any element of shafting, distant r from the propeller, from its steady state position.
β_o	Forcing frequency ratio for the original system (neglecting M_g) $= \sqrt{\dfrac{M\omega^2}{K_o}}$
β_1	Forcing frequency ratio for the R.C. system $= \sqrt{\dfrac{m_1\omega^2}{k_1}}$
μ_1	Viscosity of oil in the R.C. system.
μ_d	Viscosity on which the design curves are based, i.e. 100 centipoise.
ρ	Density of oil in the R.C. system.
ω	Frequency of the applied periodic force, i.e. of P_o.
ω_c	Critical frequency of the original shaft system.
$\dfrac{c_1{}^2}{m_1k_1}$	Damping coefficient of the R.C. system.
$\dfrac{a^2}{MK_o}$	Damping coefficient of the original shaft system (neglecting M_g).
$\sin \omega t$	Periodic function of the applied force, i.e. of P_o.

Appendix 2

EQUIVALENT MASS OF SHAFT AND CRITICAL FREQUENCIES

If the marine propeller shaft is taken as a bar with a uniformly distributed mass, the shafting system may be considered to be as in Fig. 27, assuming that the resonance changer is not fitted and ignoring any damping.

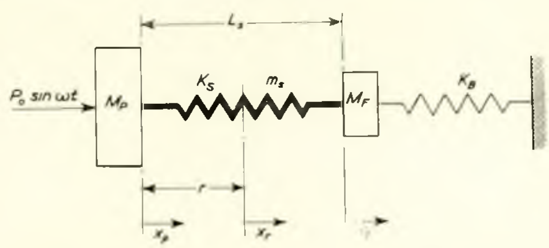


FIG. 27—*Diagrammatic representation of shafting system without resonance changer and ignoring damping*

Den Hartog[4] has shown that the axial vibration of such a shaft can be represented by the Wave Equation and, therefore, at r:

$$m_S\frac{\partial^2 x_r}{\partial t^2} = A_SE\frac{\partial^2 x_r}{\partial r^2} \qquad (25)$$

Assumption that the axial vibration of the shaft is simply harmonic at a frequency ω, so that $x_r = X_r \sin \omega t$, reduces equation (25) to:

$$\frac{d^2X_r}{dr^2} + \frac{m_s\omega^2}{A_SE}X_r = 0 \qquad (26)$$

This is a simple second-order differential equation for which the general solution can be written as:

$$X_r = C_1\cos r\sqrt{\frac{m_s\omega^2}{A_SE}} - C_2\sin r\sqrt{\frac{m_s\omega^2}{A_SE}} \qquad (27)$$

$\sqrt{\dfrac{m_s\omega^2}{A_SE}}$ can obviously be rewritten as $\dfrac{1}{L_S}\sqrt{\dfrac{M_S\omega^2}{K_S}}$

or for convenience as $\dfrac{Z}{L_S}$ where $Z = \sqrt{\dfrac{M_S\omega^2}{K_S}}$

Thus, equation (27) becomes:

$$X_r = C_1\cos\frac{rZ}{L_S} - C_2\sin\frac{rZ}{L_S} \qquad (28)$$

The end conditions to be satisfied are that $X_r = X_p$ when $r = 0$ and $X_r = X_1$ when $r = L_S$. Therefore:

$$C_1 = X_p \text{ and } C_2 = X_p\cot Z - X_1\operatorname{cosec} Z$$

Hence:

$$X_r = X_p\left[\cos\frac{rZ}{L_S} - \left(\cot Z - \frac{X_1}{X_p}\operatorname{cosec} Z\right)\sin\frac{rZ}{L_S}\right] \qquad (29)$$

The strain at any point $= \dfrac{\partial x_r}{\partial r}$ and the force exerted $= A_SE\dfrac{\partial x_r}{\partial r}$ or $K_SL_S\dfrac{\partial x_r}{\partial r}$ and from equation (29):

$$\frac{dX_r}{dr} = -X_p\frac{Z}{L_S}\left[\sin\frac{rZ}{L_S} + \left(\cot Z - \frac{X_1}{X_p}\operatorname{cosec} Z\right)\cos\frac{rZ}{L_S}\right] \qquad (30)$$

When the system shown in Fig. 27 is subjected to a periodic force $P_o\sin \omega t$, the equations of the forces in the system are:

At $r = 0$

$$P_o = M_P\omega^2X_p + K_SX_pZ\left(\cot Z - \frac{X_1}{X_p}\operatorname{cosec} Z\right) \qquad (31)$$

At $r = L_S$

$$K_SX_pZ\left[\sin Z + \left(\cot Z - \frac{X_1}{X_p}\operatorname{cosec} Z\right)\cos Z\right] = K_BX_1 - M_F\omega^2X_1 \qquad (32)$$

From equation (32):

$$\frac{X_1}{X_p} = \frac{Z\operatorname{cosec} Z}{\dfrac{K_B}{K_S} - \dfrac{M_F}{M_S}Z^2 + Z\cot Z} \qquad (33)$$

Natural Frequency

At the natural frequency, equation (31) is satisfied when

P_o, equals zero, therefore:

$$-M_P\omega_c^2X_p + K_SX_pZ_c\left(\cot Z_c - \frac{Z_c \operatorname{cosec}^2 Z_c}{\frac{K_B}{K_S} - \frac{M_F}{M_S}Z_c^2 + Z_c\cot Z_c}\right) = 0 \qquad (34)$$

This can be simplified to:

$$\tan Z_c = \frac{\frac{K_B}{K_S} - \left(\frac{M_P}{M_S} + \frac{M_F}{M_S}\right)Z_c^2}{Z_c\left[1 + \frac{M_P}{M_S}\left(\frac{K_B}{K_S} - \frac{M_F}{M_S}Z_c^2\right)\right]} \qquad (35)$$

This equation can be used to determine the first and second critical frequencies.

Equivalent System Forces

To simplify the equations using a resonance changer it is desirable to consider an equivalent system with the effect of the shaft masses represented by concentrated masses at the two ends of the shaft. If R_1M_S is added to M_P and R_2M_S is added to M_F and the shaft is considered massless, the equations of the forces in the system become:

$$P_o = -(M_P+R_1M_S)\,\omega^2X_p + K_S(X_p-X_1) \qquad (36)$$

$$K_S(X_p-X_1) = K_BX_1 - (M_F+R_2M_S)\,\omega^2X_1 \qquad (37)$$

If R_1 and R_2 have been selected to give the same values of X_1 and X_p in the equivalent system as with the massive shaft, then from equations (31), (36) and (37):

$$P_o = -(M_P+R_1M_S)\,\omega^2X_p + [K_B-(M_F+R_2M_S)\,\omega^2]\,X_1$$

$$= -M_P\omega^2X_p + K_SX_pZ\left(\cot Z - \frac{X_1}{X_p}\operatorname{cosec} Z\right) \qquad (38)$$

This resolves into:

$$R_1 + \left(\frac{X_1}{X_p}\right)R_2 = \frac{X_1}{X_p}\left[\frac{K_B}{K_SZ^2} - \frac{M_F}{M_S} + \frac{\operatorname{cosec} Z}{Z}\right] - \frac{\cot Z}{Z} \qquad (39)$$

For any value of Z, having evaluated X_1/X_p from equation (33), equation (39) gives a relationship for R_1 and R_2.

Equivalent System Energy

At the natural frequency the maximum kinetic energy of the equivalent system must equal the maximum kinetic energy of the massive shaft system, that is:

$$\frac{M_P+R_1M_S}{2}\,\omega_c^2X_p^2 + \frac{M_F+R_2M_S}{2}\,\omega_c^2X_1^2$$

$$= \frac{M_P}{2}\,\omega_c^2X_p^2 + \frac{M_F}{2}\,\omega_c^2X_1^2 + \int_0^{L_S}\frac{m_s\omega_c^2X_r^2}{2}\,dr \qquad (40)$$

This resolves into:

$$R_1 + \left(\frac{X_1}{X_p}\right)^2R_2 = \frac{1}{2}\left[1 + \left(\frac{M_P}{M_S}\right)^2Z_c^2\right] - \frac{M_P}{M_S}\sin^2 Z_c + \left[1 - \left(\frac{M_P}{M_S}\right)^2Z_c^2\right]\frac{\sin 2Z_c}{4Z_c} \qquad (41)$$

Equation (41) provides a second relationship for R_1 and R_2 at the natural frequency and combined with equation (39) gives values for R_1 and R_2.

The values established at the natural frequency for R_1 and R_2 are found to satisfy equation (39) over a fairly wide range of frequencies, thus giving some justification for the use of the simplified system using concentrated masses M and M_g in the analysis of the shafting system and resonance changer.

The introduction of the resonance changer does however have an effect on the ratio of K_B/K_S which influences the values of R_1 and R_2, as can be seen from Table II, calculated for $M_S=M_P=2M_F$, but it is not thought that this invalidates the theory on which the design of resonance changers is based.

TABLE II.—EFFECT OF VARIATIONS IN K_B/K_S AND FREQUENCY ON EQUIVALENT MASS OF SHAFT

Z	0·20		0·53		0·71		0·81		1·00	
$\frac{K_B}{K_S}$	R_1	R_2	R_1	R_2	R_1	R_2	R_1	R_2	R_1	R_2
1·0	0·47	0·56	0·48*	0·57*	0·49	0·57	0·49	0·58	0·50	0·59
3·0	0·42	0·82	0·43	0·83	0·43*	0·83*	0·43	0·83	0·43	0·84
10·0	0·38	1·79	0·39	1·81	0·39	1·81	0·39*	1·83*	0·39	1·84

* Indicates first critical frequency

N.B. In the calculations carried out for analysis of the systems in the various ships, equivalent mass was added at the propeller only, as indicated in Table I, since it was not at that time appreciated that the equivalence over a wider band of frequencies could be obtained by adding equivalent mass at both ends.

APPENDIX 3

DEVELOPMENT OF OPTIMUM DAMPING EQUATIONS

Thrust Variations at Thrust Collar

From equation (20), for any given value of q_1, a family of curves can be produced for a range of c_1^2/m_1k_1 values between zero and infinity. Fig. 4 illustrates this. All the curves of this family will pass through the points P and Q but their magnitude in the resonant regions will depend on their c_1^2/m_1k_1 value.

The desired value of c_1^2/m_1k_1 is that which will allow passage through the resonance regions with the least possible amplification. What the magnitudes are at the fixed points P

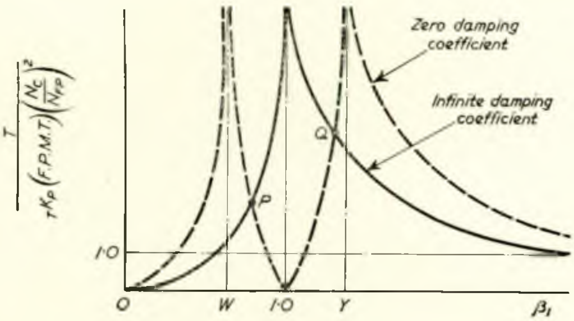


FIG. 4—*Thrust variation response curves*

and Q may be determined in the following manner.

When $\frac{c_1^2}{m_1 k_1}$ is zero, equation (20) reduces to:

$$\frac{T}{\text{Constant}} = \frac{\beta_1^2(1-\beta_1^2)}{(1-\beta_1^2)^2 - q_1\beta_1^2} \tag{42}$$

(N.B. The denominator of the first term being constant it is written as such for convenience.)

When $\frac{c_1^2}{m_1 k_1}$ is infinite, equation (20) reduces to:

$$\frac{T}{\text{Constant}} = \frac{\beta_1^2}{1-\beta_1^2} \tag{43}$$

At the points P and Q the magnitudes of both these curves are equal, and thus the second terms of equations (42) and (43) may be equated, but in doing so the sign of one must be reversed to allow for the phase relationship. Hence:

$$-\frac{\beta_1^2}{1-\beta_1^2} = \frac{\beta_1^2(1-\beta_1^2)}{(1-\beta_1^2)^2 - q_1\beta_1^2} \tag{44}$$

From which:

$$\beta_1(P) = \sqrt{\frac{4+q_1-\sqrt{8q_1+q_1^2}}{4}} \tag{45}$$

$$\beta_1(Q) = \sqrt{\frac{4+q_1+\sqrt{8q_1+q_1^2}}{4}} \tag{46}$$

Therefore from equation (43):

$$\frac{T}{\text{Constant}}(\text{at } P) = \frac{\beta_1^2(P)}{1-\beta_1^2(P)} \tag{47}$$

$$\frac{T}{\text{Constant}}(\text{at } Q) = -\frac{\beta_1^2(Q)}{1-\beta_1^2(Q)}, \tag{48}$$

the negative sign being again due to phase relationship.

It is therefore obvious, since $\beta_1(Q)$ is greater than $\beta_1(P)$, that the magnitude at Q is greater than at P.

It follows then that passage through the resonance regions with the least possible magnitude can be achieved if the c_1^2/m_1k_1 value is so chosen that the response curve has a maximum turning point at Q. That is, at Q, $\frac{d}{d\beta_1}\left(\frac{T}{\text{Constant}}\right)$ equals zero.

Now, from equation (18):

$$\frac{d}{d\beta_1}\left(\frac{T}{\text{Constant}}\right) = 2\beta_1\left(\frac{T}{P_o}\right) + \beta_1^2\frac{d}{d\beta_1}\left(\frac{T}{P_o}\right) \tag{49}$$

The relationship for T/P_o is contained within the square root bracket of equation (20) and it can be simplified by substituting in it:

$$A = (1-\beta_1^2)^2 + D\beta_1^2$$
$$B = (1-\beta_1^2)^2 - q_1\beta_1^2$$
$$C = D\beta_1^2(1-\beta_1^2)^2$$
$$D = \frac{c_1^2}{m_1k_1}$$

Equation (49) can then be rewritten as:

$$\frac{d}{d\beta_1}\left(\frac{T}{\text{Constant}}\right) = \frac{2\beta_1 A^{\frac{1}{2}}}{(B^2+C)^{\frac{1}{2}}} + \beta_1^2\frac{d}{d\beta_1}\left[A^{\frac{1}{2}}(B^2+C)^{-\frac{1}{2}}\right] \tag{50}$$

which can be rearranged to give:

$$\frac{d}{d\beta_1}\left(\frac{T}{\text{Constant}}\right) = \frac{2\beta_1^2}{A^{\frac{1}{2}}(B^2+C)^{\frac{3}{2}}}\left[(B^2+C)\left(\frac{A}{\beta_1} + \frac{1}{4}\frac{d}{d\beta_1}A\right) - \frac{A}{4}\left(2AB\frac{d}{d\beta_1}B + \frac{d}{d\beta_1}C\right)\right] \tag{51}$$

From equation (44) it will be seen that at Q:

$$B = -[1-\beta_1^2(Q)]^2 \tag{52}$$

As stated, at Q it is required that $\frac{d}{d\beta_1}\left(\frac{T}{\text{Constant}}\right)$ equals zero. That is, the differential equals zero when $\beta_1 = \beta_1(Q)$ and thus the abbreviated form of B given in equation (52) may be substituted in it for this particular case. It should, of course, be noted that this form of B is not applicable to $\frac{d}{d\beta_1}B$.

Therefore, from equation (51):

$$\frac{2\beta_1^2}{A^{\frac{1}{2}}(B^2+C)^{\frac{3}{2}}}\left(\left[(1-\beta_1^2)^4 + D\beta_1^2(1-\beta_1^2)^2\right]\left[\frac{(1-\beta_1^2)^2}{\beta_1} + D\beta_1 - \beta_1(1-\beta_1^2) + \frac{D\beta_1}{2}\right] - \left[\frac{(1-\beta_1^2)^2}{4} + \frac{D\beta_1^2}{4}\right]\left[8\beta_1(1-\beta_1^2)^3 + 4q_1\beta_1(1-\beta_1^2)^2 + 2D\beta_1(1-\beta_1^2)^2 - 4D\beta_1^3(1-\beta_1^2)\right]\right)$$
$$= 0, \text{ when } \beta_1 = \beta_1(Q) \tag{53}$$

Also, from equation (44) it will be seen that at Q:

$$q_1 = \frac{2[1-\beta_1^2(Q)]^2}{\beta_1^2(Q)} \tag{54}$$

This relationship may now be introduced into equation (53) and the expression regrouped to give:

$$\frac{2\beta_1^5(1-\beta_1^2)}{A^{\frac{1}{2}}(B^2+C)^{\frac{3}{2}}}\left[D - (1-\beta_1^2)^2\left(\frac{1+2\beta_1^2}{\beta_1^2}\right)\right]\left[D + (1-\beta_1^2)^2\left(\frac{1}{\beta_1^2}\right)\right] = 0, \text{ when } \beta_1 = \beta_1(Q) \tag{55}$$

and since $\beta_1(Q) \neq 0$ it follows from this equation that for optimum conditions:

$$D = [1-\beta_1^2(Q)]^2\left[\frac{1+2\beta_1^2(Q)}{\beta_1^2(Q)}\right] \tag{56}$$

$$\text{or } D = -[1-\beta_1^2(Q)]^2\left[\frac{1}{\beta_1^2(Q)}\right] \tag{57}$$

As c_1^2/m_1k_1 cannot be negative it follows from equation (56) that:

$$\text{Optimum } \frac{c_1^2}{m_1k_1} = [1-\beta_1^2(Q)]^2\left[\frac{1+2\beta_1^2(Q)}{\beta_1^2(Q)}\right] \tag{58}$$

Substitution in this equation of the relationship for $\beta_1(Q)$ given in equations (46) and (54) gives:

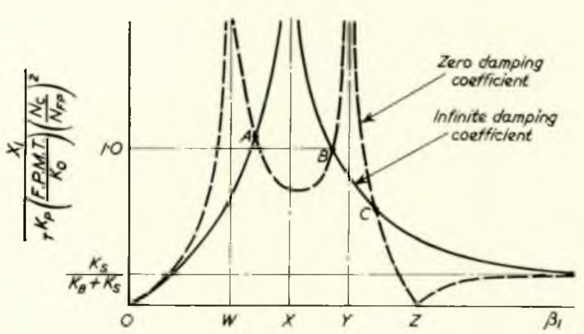


FIG. 10—*Thrust collar movement response curves*

$$\text{Optimum } \frac{c_1^2}{m_1 k_1} = \frac{q_1}{4}\left[6 + q_1 + \sqrt{8q_1 + q_1^2}\right] \qquad (59)$$

By similar methods it can be shown that this identical value for c_1^2/m_1k_1 gives a maximum value for T/P_o at $\beta_1 = \beta_1(P)$.

Thrust Collar Movement

It will be shown in the ensuing analysis that, for practical values of K_B, K_S and q_1 under zero and infinite damping conditions the response curves derived from equation (21) will be of the form indicated in Fig. 10.

When damping is zero, equation (21) becomes:

$$\frac{X_1}{\text{Constant}} = \frac{K_S\beta_1^2\left[1-\beta_1^2+q_1\left(1+\frac{K_B}{K_S}\right)\right]}{(K_B+K_S)[(1-\beta_1^2)^2-q_1\beta_1^2]} \qquad (60)$$

(N.B. The denominator of the first term being constant, it is written as such for convenience.)

This expression becomes zero when:

$$\beta_1^2(Z) = 1 + q_1\left(1+\frac{K_B}{K_S}\right) \qquad (61)$$

and infinite when:

$$\beta_1^2(W) = 1 + \frac{q_1}{2} - \frac{q_1}{2}\sqrt{1+\frac{4}{q_1}} \qquad (62)$$

and

$$\beta_1^2(Y) = 1 + \frac{q_1}{2} + \frac{q_1}{2}\sqrt{1+\frac{4}{q_1}} \qquad (63)$$

When the damping is infinite, equation (21) becomes:

$$\frac{X_1}{\text{Constant}} = \frac{K_S\beta_1^2}{(K_B+K_S)(1-\beta_1^2)} \qquad (64)$$

which becomes infinite when:

$$\beta_1^2(X) = 1 \qquad (65)$$

If $\beta_1(Z) > \beta_1(Y)$ it will be seen from equations (61) and (63) that:

$$1 + q_1\left(1+\frac{K_B}{K_S}\right) > 1 + \frac{q_1}{2} + \frac{q_1}{2}\sqrt{1+\frac{4}{q_1}} \qquad (66)$$

from which:

$$q_1 > \frac{1}{\frac{K_B}{K_S}\left(1+\frac{K_B}{K_S}\right)} \qquad (67)$$

In naval installations K_B/K_S is usually greater than 3, and for this value statement (67) is satisfied provided q_1 is greater than 0·083. For larger values of K_B/K_S, q_1 could be smaller and still satisfy statement (67). Experience has shown, however, that q_1 values smaller than 0·2 generally result in the value of T being unacceptably high in the resonance regions. Therefore it follows that, for practical values of K_B, K_S and q_1, $\beta_1(Z)$ is always greater than $\beta_1(Y)$ and therefore the magnitude of $\frac{X_1}{\text{Constant}}$ is greater at B than at C.

The values of β_1 corresponding to points A and C may be found from equations (60) and (64) by equating them in a manner which takes into account their phase relationship. Thus:

$$\frac{K_S\beta_1^2\left[1-\beta_1^2+q_1\left(1+\frac{K_B}{K_S}\right)\right]}{(K_B+K_S)[(1-\beta_1^2)^2-q_1\beta_1^2]} = -\frac{K_S\beta_1^2}{(K_B+K_S)(1-\beta_1^2)} \qquad (68)$$

which gives:

$$\beta_1^2(A) = \frac{4 + q_1\left(2+\frac{K_B}{K_S}\right) - \sqrt{8q_1 + q_1^2\left(2+\frac{K_B}{K_S}\right)^2}}{4} \qquad (69)$$

$$\beta_1^2(C) = \frac{4 + q_1\left(2+\frac{K_B}{K_S}\right) + \sqrt{8q_1 + q_1^2\left(2+\frac{K_B}{K_S}\right)^2}}{4} \qquad (70)$$

Substituting these values in equation (64) gives the appropriate magnitudes, which are:

$$\frac{X_1}{\text{Constant}}(\text{at } A) = \frac{K_S}{K_B+K_S}\left\{\frac{4 + q_1\left(2+\frac{K_B}{K_S}\right) - \sqrt{8q_1+q_1^2\left(2+\frac{K_B}{K_S}\right)^2}}{-q_1\left(2+\frac{K_B}{K_S}\right) + \sqrt{8q_1+q_1^2\left(2+\frac{K_B}{K_S}\right)^2}}\right\} \qquad (71)$$

$$\frac{X_1}{\text{Constant}}(\text{at } C) = \frac{K_S}{K_B+K_S}\left\{\frac{4 + q_1\left(2+\frac{K_B}{K_S}\right) + \sqrt{8q_1+q_1^2\left(2+\frac{K_B}{K_S}\right)^2}}{-q_1\left(2+\frac{K_B}{K_S}\right) - \sqrt{8q_1+q_1^2\left(2+\frac{K_B}{K_S}\right)^2}}\right\} \qquad (72)$$

From these two equations it can be shown that the magnitude at A is always greater than the magnitude at C.

The value of β_1 corresponding to point B can also be found by equating equations (60) and (64) with the appropriate phase relationship. That is:

$$\frac{K_S\beta_1^2\left[1-\beta_1^2+q_1\left(1+\frac{K_B}{K_S}\right)\right]}{(K_B+K_S)[(1-\beta_1^2)^2-q_1\beta_1^2]} = \frac{K_S\beta_1^2}{(K_B+K_S)(1-\beta_1^2)} \qquad (73)$$

from which:

$$\beta_1^2(B) = \frac{K_B+K_S}{K_B} \qquad (74)$$

Substituting this value in equation (64) shows the magnitude at B to be:

$$\frac{X_1}{\text{Constant}}(\text{at } B) = -1 \qquad (75)$$

The negative sign merely indicates the phase relationship.

If the magnitude at A is greater than that at B, then it will be seen from equations (71) and (75) that:

$$\frac{K_S}{K_B+K_S}\left\{\frac{4 + q_1\left(2+\frac{K_B}{K_S}\right) - \sqrt{8q_1+q_1^2\left(2+\frac{K_B}{K_S}\right)^2}}{-q_1\left(2+\frac{K_B}{K_S}\right) + \sqrt{8q_1+q_1^2\left(2+\frac{K_B}{K_S}\right)^2}}\right\} > +1 \qquad (76)$$

The positive sign is used because only the magnitude of the amplitude of X_1 is of concern.

Statement (76) can be transformed to state that:

$$16 + 8q_1\left(2+\frac{K_B}{K_S}\right)^2 + q_1^2\left(2+\frac{K_B}{K_S}\right)^4 > 8q_1\left(2+\frac{K_B}{K_S}\right)^2 + q_1^2\left(2+\frac{K_B}{K_S}\right)^4$$

which confirms that the magnitude is always greater at A than at B for a response curve of the form illustrated in Fig. 10. Therefore, for practical values of K_B, K_S and q_1 the magnitude at A is always greater than that at B.

In analysing the thrust amplification, the value of the damping coefficient which gave the optimum response was derived. Although this value of damping does not quite optimize the displacement response, that it very nearly does is shown in Fig. 11. On these curves the point A is that shown in Fig. 10 and for each combination of values of K_B, K_S and q_1 represents an amplitude which the displacement must reach at this frequency whatever the damping chosen for the resonance changer. Since in no case does the maximum value of the curve greatly exceed the value at A, the damping coefficient chosen on the basis of thrust amplification gives a very satisfactory control of the thrust collar displacement amplification.

APPENDIX 4

DESIGN OF RESONANCE CHANGER

Design Curves

The design of a suitably tuned resonance changer requires a practicable compromise for four factors, these being:

(i) Permissible thrust variation at thrust block.
(ii) Size of the oil bottle.
(iii) Length of pipe.
(iv) Bore of pipe.

Because of their mutual dependence upon the stiffness ratio q_1, it is possible, by making certain assumptions, to express these factors in a graphical form which can serve as the basis of the design, and the curves so derived are shown in Fig. 29. However, two of the assumptions are likely to be invalid in practice and these are:

(a) That the viscosity is 100 centipoise.
(b) That the hydraulic equivalent length equals the actual length.

Effect of Viscosity and Equivalent Length

Consider an installation which has vicsosity μ_1 and equivalent length L_E. Its virtual damping could be written from equation (4) as:

$$c_1 = \frac{32\mu_1}{d_1^2} \cdot \frac{L_E A_o^2}{A_1} \tag{77}$$

This equation can be converted to:

$$c_1 = \frac{32\mu_d}{d_m^2} \cdot \frac{L_1 A_c^2}{A_1} \cdot \frac{\mu_1}{\mu_d} \cdot \frac{L_E}{L_1} \cdot \left(\frac{d_m}{d_1}\right)^2 \tag{78}$$

The damping coefficient would then be given by:

$$\frac{c_1^2}{m_1 k_1} = \frac{1}{m_1 k_1}\left[\frac{32\mu_d}{d_m^2} \cdot \frac{L_1 A_o^2}{A_1}\right]^2 \left(\frac{\mu_1}{\mu_d} \cdot \frac{L_E}{L_1}\right)^2 \left(\frac{d_m}{d_1}\right)^4 \tag{79}$$

which can be rewritten as:

$$\frac{c_1^2}{m_1 k_1} = \left[\text{Optimum } \frac{c_1^2}{m_1 k_1}\right] \left(\frac{\mu_1}{\mu_d} \cdot \frac{L_E}{L_1}\right)^2 \left(\frac{d_m}{d_1}\right)^4 \tag{80}$$

If c_1^2/m_1k_1 is to be maintained equal to the optimum, then equation (80) reduces to:

$$1 = \left(\frac{\mu_1}{\mu_d} \cdot \frac{L_E}{L_1}\right)^2 \left(\frac{d_m}{d_1}\right)^4 \tag{81}$$

which gives the relationship that:

$$d_1 = d_m \sqrt{\frac{\mu_1}{\mu_d} \cdot \frac{L_E}{L_1}} \tag{82}$$

Thus, by assuming an L_E/L_1 ratio, and provided μ_1 is known, it is a simple matter to obtain a suitable value of d_1 from the value of d_m determined from Fig. 29.

Multipipe Resonance Changers

In some installations it may be desirable to connect the oil bottle to the thrust block by a number of pipes, as illustrated diagrammatically in Fig. 28.

Let there be n pipes connecting the bottle to the thrust block and let it be assumed that each pipe has its corresponding portion of the complete system, which has A_o and V_1 in total the same as for a single-pipe system. Hence:

$$A_{on} = \frac{A_o}{n} \tag{83}$$

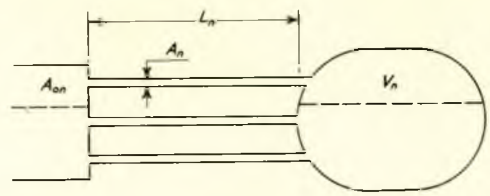


FIG. 28—*Diagrammatic representation of a multipipe resonance changer*

$$V_n = \frac{V_1}{n} \tag{84}$$

Considering one pipe of such a system, the equivalent version of equation (2) is:

$$A_{on}p = \frac{\rho_1 A_{on}^2 L_n}{A_n}(\ddot{x}_1 - \ddot{x}_b) + 8\pi\mu_1 L_n \frac{A_{on}^2}{A_n^2}(\dot{x}_1 - \dot{x}_b) + \frac{A_{on}^2 B_1}{V_n}(x_1 - x_b) \tag{85}$$

$$\text{Let } A_n = A_1 \tag{86}$$

Substitution of the relationships given in equations (83), (84) and (86) in equation (85) results in:

$$A_o p = \frac{\rho_1 A_o^2 L_n}{nA_1}(\ddot{x}_1 - \ddot{x}_b) + 8\pi\mu_1 L_n \frac{A_o^2}{nA_1^2}(\dot{x}_1 - \dot{x}_b) + \frac{A_o^2 B_1}{V_1}(x_1 - x_b) \tag{87}$$

This equation is obviously identical with equation (2) if:

$$L_n = nL_1 \tag{88}$$

Thus, the information obtained from the design curves (Fig. 29) can form the basis for the design of an n-pipe resonance changer if the bore and length of each pipe is made equal to the bore and n times the length of the pipe in the equivalent single-pipe resonance changer.

Design Rules

The procedure for using the design curves is as follows:

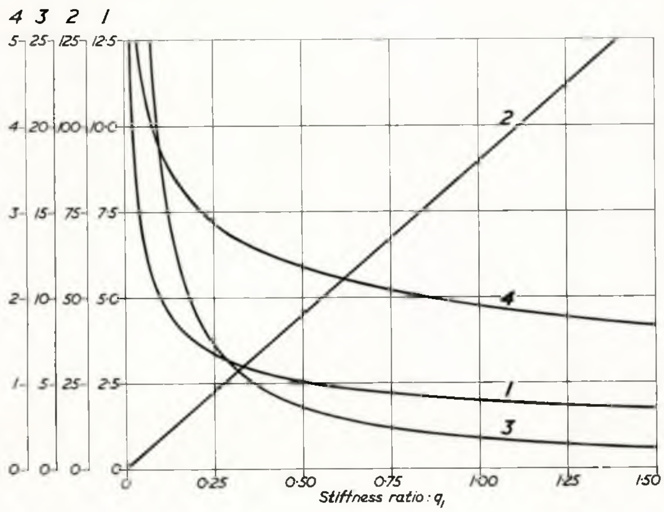
(a) From a knowledge of the shafting installation derive the constants associated with parameters (1) to (4).
(b) From curve (1) choose an approximate value of q_1 suitable for the requirements of the system.
(c) Select a suitable total oil bottle capacity from available oil bottles, and settle exact value for q_1.
(d) From the curves (1), (3) and (4) obtain the appropriate parameters and calculate the values of T_m, L_1/d_1^2 and d_m.
(e) Estimate a value for the L_E/L_1 ratio. Experience suggests 2·0 should be satisfactory.
(f) Knowing μ_1 and the value of d_m, calculate the value of d_1 and hence the value of L_1 from the L_1/d_1^2 ratio.
(g) Check that the volume of oil in the pipe(s) and thrustmeter passages and cylinders is sufficiently small in comparison with that in the oil bottle, to maintain the

validity of the calculations. 10 per cent of the bottle size is suggested as a maximum.

(h) Detail design the arrangement, taking particular care to ensure that no air can be trapped in the system. On completion, check that L_E is satisfactory and that laminar flow exists. Experience suggests this will almost always be so.

Specimen Design Calculations

Consider as an example the design of a resonance changer for a ship for which the undernoted is the required basic information:



Symbols:

A_o	Total area of thrustmeter cylinders, in.^{2}
B_p	Number of propeller blades
d_1	Bore of pipe, in.
d_m	Minimum bore of pipe, in.
F.P.M.T.	Mean thrust at full power, ton
K_o	Axial stiffness of shafting and thrust block, tons/in. $= \dfrac{K_B K_S}{K_B + K_S}$
L_1	Length of pipe, ft.
N_c	Critical shaft revolutions $= \dfrac{60\ \omega_c}{2\pi\ B_p}$ r.p.m.
N_{FP}	Full power shaft revolutions, r.p.m.
T_m	Maximum thrust variation at thrust block, ton
V_1	Capacity of oil bottle, in.^{3}
${}_TK_P$	Thrust variation at propeller as a fraction of mean thrust
ω_c	Critical frequency of shaft system (without resonance changer), rad/sec.

Parameters:

1) Maximum thrust variation at thrust block $\dfrac{T_m}{{}_TK_P\ (F.P.M.T.)\left(\dfrac{N_c}{N_{FP}}\right)^2}$

2) Capacity of oil bottle $\dfrac{V_1}{A_o{}^2/K_o}$

3) Pipe length/(bore)^2 ratio $\dfrac{L_1/d_1{}^2}{\dfrac{K_o \times 10^6}{A_o{}^2\ \omega_c{}^2}}$

4) Minimum bore of pipe $d_m\sqrt{\omega_c}$

Basis:

To derive these parameters it has been assumed that at operating conditions the oil has:

Bulk modulus	$2{\cdot}0 \times 10^5$ lb/in.^{2}
Specific gravity	0·864
Viscosity	100 centipoise

n-pipes:

The curves shown refer to a single-pipe resonance changer. Should it be desired to have n number of pipes, then T_m, V_1 and d_1 may be found direct from the curves, but the length of each pipe should be made $(n \times L_1)$ ft.

FIG. 29—*Design curves for resonance changers*

(a)	Total area of thrustmeter cylinders (A_o), sq. in	265
(b)	Full power mean thrust (*F.P.M.T.*), tons	145
(c)	Full power shaft revolutions (N_{FP}), r.p.m.	240
(d)	Length of intermediate shaft to aft face of thrust collar, ft.	224·5
(e)	Cross sectional area of intermediate shaft (excluding liners and couplings), sq. ft.	0·852
(f)	Length of propeller shaft to centre line of propeller, ft.	32·4
(g)	Cross sectional area of propeller shaft (excluding liners and couplings), sq. ft.	1·336
(h)	Weight of propeller, tons	14·7
(i)	Number of propeller blades (B_p)	4
(j)	Weight of propeller cone, nut and tail end of propeller shaft, tons	3·1
(k)	Weight of main gear wheel, associated pinions, thrust collar and shafting forward thereof, and half the weight of the thrust block casing, tons	33·2
(l)	Assumed value of thrust block stiffness (K_B), tons/in.	2,500
(m)	Assumed value of entrained water allowance, per cent of propeller weight	60
(n)	Assumed value of propeller thrust variation fraction (${}_TK_P$)	$\ngtr \pm 0{\cdot}040$
(o)	Assumed value of propeller damping factor (a), tons sec./in.	1·00

From the above data the following may be calculated:

Stiffness of intermediate shaft $= 160{\cdot}5 \times 10^3 \times \dfrac{0{\cdot}852}{224{\cdot}5}$
$= 610$ tons in.

Stiffness of propeller shaft $= 160{\cdot}5 \times 10^3 \times \dfrac{1{\cdot}336}{32{\cdot}4}$
$= 6{,}620$ tons/in.

Hence:

Stiffness of line of shafting (K_S) $= \dfrac{610 \times 6{,}620}{610 + 6{,}620}$
$= 560$ tons/in.

And therefore:

Stiffness of original system (K_o) $= \dfrac{560 \times 2{,}500}{560 + 2{,}500}$
$= 460$ tons/in.

Also:

$$\frac{K_B}{K_S} = \frac{2{,}500}{560} = 4{\cdot}46$$

If the density of steel is taken as 0·218 tons/cu. ft., then:

Weight of intermediate shaft $= 0{\cdot}218\ (224{\cdot}5 \times 0{\cdot}852)$
$= 41{\cdot}7$ tons

Weight of propeller shaft $= 0{\cdot}218\ (32{\cdot}4 \times 1{\cdot}336)$
$= 9{\cdot}4$ tons

Therefore:

Total weight of shafting $= 51{\cdot}1$ tons
$=$ say, 55 tons
(to allow for liners and couplings)

Also:

Total weight at propeller $= 14{\cdot}7 + 0{\cdot}6(14{\cdot}7) + 3{\cdot}1$
$= 26{\cdot}6$ tons

The mass ratios of the system therefore are:

$$\frac{M_P}{M_S} = \frac{26{\cdot}6}{55} = 0{\cdot}484$$

$$\frac{M_F}{M_S} = \frac{33{\cdot}2}{55} = 0{\cdot}604$$

The critical speeds may now be evaluated by using the relationship derived in equation (35) in Appendix 2, viz.:

$$\tan Z_c = \frac{\dfrac{K_B}{K_S} - \left(\dfrac{M_P}{M_S} + \dfrac{M_F}{M_S}\right) Z_c{}^2}{Z_c\left[1 + \dfrac{M_P}{M_S}\left(\dfrac{K_B}{K_S} - \dfrac{M_F}{M_S} Z_c{}^2\right)\right]}$$

Substitution of the above ratios in this gives:

$$\tan Z_c = \frac{4{\cdot}46 - 1{\cdot}088Z_c^2}{Z_c[3{\cdot}159 - 0{\cdot}292Z_c^2]}$$

Hence, by trial and error:

$$Z_c = \begin{cases} 0{\cdot}922 \text{ for 1st critical.} \\ 2{\cdot}512 \text{ for 2nd critical.} \end{cases}$$

By definition $Z_c = \sqrt{\frac{M_S\omega_c^2}{K_S}}$, therefore:

$$Z_c = \sqrt{\frac{55}{386{\cdot}4} \times \frac{\omega_c^2}{560}\left[\frac{\text{tons, sec}^2}{\text{inch}} \times \frac{\text{rad}^2\text{, inch}}{\text{sec}^2\text{, ton}}\right]}$$

$$= \frac{\omega_c}{63}$$

That is:

$$\omega_c = 63Z_c$$

Hence, from the derived values for Z_c:

$$\omega_c = \begin{cases} 58{\cdot}1 \text{ rad/sec. for 1st critical.} \\ 158{\cdot}3 \text{ rad/sec. for 2nd critical.} \end{cases}$$

Since $N_c = \frac{60\omega_c}{2\pi B_p}$ and B_p is 4:

$$N_c = \begin{cases} 138{\cdot}7 \text{ r.p.m. for 1st critical.} \\ 377{\cdot}9 \text{ r.p.m. for 2nd critical.} \end{cases}$$

The second critical speed is well outside the running range and can therefore be ignored.

The amplitude of the thrust variation to be expected at the first critical speed can be estimated from equation (24) which can be rewritten as:

$$T_c \simeq \frac{{}_1K_P}{a} \cdot \frac{K_S}{\omega_c}(F.P.M.T.)\left(\frac{N_c}{N_{FP}}\right)^2$$

Thus:

$$T_c \simeq \pm \frac{0{\cdot}040}{1{\cdot}0} \cdot \frac{560}{58{\cdot}1}(145)\left(\frac{138{\cdot}7}{240}\right)^2$$

$$\simeq \pm\ 18{\cdot}9 \text{ tons}$$

The design parameters of the resonance changer can now be evaluated from Fig. 29.

$$\text{Parameter (1)} = \frac{T_m}{\pm 0{\cdot}040\,(145)\left(\frac{138{\cdot}7}{240}\right)^2}$$

$$= \frac{T_m \text{ (tons)}}{\pm 1{\cdot}936}$$

For a ship of this size it would seem reasonable to try to restrict T_m to a about $\pm$ 5 tons. Therefore:

$$\text{Parameter (1)} = \frac{\pm 5{\cdot}0}{\pm 1{\cdot}936}$$

$$= 2{\cdot}583$$

From Fig. 29 the corresponding value of q_1 is 0·48 and at this value Parameter (2) has a value of 43·0. Therefore:

$$43{\cdot}0 = \frac{V_1 \text{ (cu. in.)}}{(265)^2/460}$$

From which:

$$V_1 = 6{,}565 \text{ cu. in.}$$

If the nearest available bottle size is 4·5 cu. ft., then:

$$V_1 = 4{\cdot}5 \text{ cu. ft.}$$

$$= 7{,}776 \text{ cu. in.}$$

And:

$$\text{Parameter (2)} = \frac{7{,}776}{(265)^2/460}$$

$$= 50{\cdot}9$$

Hence, from Fig. 29 the corresponding value of q_1 is now 0·57 and the appropriate parameter values are:

Parameter (1)

$$\frac{T_m \text{ (tons)}}{\pm 1{\cdot}936} = 2{\cdot}44$$

From which:

$$T_m = \pm\ 4{\cdot}72 \text{ tons}$$

Parameter (3)

$$\frac{L_1/d_1^2 \text{ (ft./sq. in.)}}{\frac{460 \times 10^6}{(265 \times 58{\cdot}1)^2}} = 3{\cdot}18$$

Giving:

$$L_1/d_1^2 = 6{\cdot}17 \text{ ft./sq. in.}$$

Parameter (4)

$$d_m \text{ (in.)} \sqrt{58{\cdot}1} = 2{\cdot}27$$

Which gives:

$$d_m = 0{\cdot}298 \text{ in.}$$

If the pressure and temperature conditions prevailing in the thrust block (say, 2,000lb./sq. in. and 80 deg. F.) result in the oil having a viscosity of 233 centipoise, and if an L_E/L_1 ratio of 2 is assumed, then the minimum required bore of pipe is:

$$d_1 = 0{\cdot}298\sqrt{\frac{233}{100} \times 2}$$

$$= 0{\cdot}643 \text{ in.}$$

Use of this diameter would make the pipe length:

$$L_1 = 6{\cdot}17\,(0{\cdot}643)^2$$

$$= 2{\cdot}55 \text{ ft.}$$

This length is obviously too short from the practical aspect. So it is necessary to depart from the minimum diameter, which would give optimum damping, and to use a larger diameter and introduce compensating damping by means of a damping valve.

It is suggested that the pipe be made 1·0-in. bore, which will make its length 6·17 ft. If this is still too short for adequate installing, then two pipes should be used rather than a further increase of bore.

If two pipes of 1·0-in. bore are used, then the length of each will be 12·34ft. (i.e. twice the length of the single pipe).

Even with the twin-pipe installation the volume of oil in the pipes will only be 233 cu. in. and this plus that in the thrustmeter passages is obviously well within the recommended limit of 10 per cent of the bottle volume. On completion of the detail design of the installation, a check can be made that L_E/L_1 is within the assumed value of 2.

Discussion

MR. C. P. RIGBY, B.Sc., in opening the discussion, said he was pleased to have the opportunity of congratulating the author on his success in compressing to readable dimensions and to a form that was of direct use to the marine engineer the history and mathematics of some fifteen years' work. He was very generous in the credit he handed out to others but was unduly modest about his own part in inventing and developing the resonance changer.

He (Mr. Rigby) had left the field in 1948, just in time to avoid the unsatisfactory results in *Vanguard* with five-bladed propellers, and the author found himself saddled with the task of finding a better solution, and finding it quickly, for the already well advanced *Eagle* and *Ark Royal*. How successful he was was evident from the results already seen.

Turning to the meat of the paper, which for the designer lay in the appendices and in particular Appendix 4, no doubt many would be wondering whether this naval device was equally applicable to merchant ships. Having just designed one for such an application he could give an assurance that, as far as mathematics were concerned, it was. There were some surprises in such a design which at first made him doubt his own arithmetic and the author's mathematics. The principal surprise was the relatively small oil bottle required for a chosen stiffness ratio in comparison with those used in *Eagle, Ark Royal* and *Bulwark*.

That was due to the recent increase in thrustmeter operating pressures, and to the much greater axial stiffness of the solid and slower running mercantile shafts. Thrust blocks were also likely to be stiffer and the upshot was that an oil bottle of 1 to 1½ cu. ft. capacity proved ample for even a large liner. In view of that there was no necessity to compromise on the size of the bottle, and he would advocate the choice of a stiffness ratio of 1 to 1·5 even in twin screw ships. The high value of axial stiffness in merchant shafting reduced the effectiveness of propeller damping and could give rise to remarkably high figures for alternating thrust. That was another reason to be generous with bottle size, that is, to use a high stiffness ratio.

While he agreed that a resonance changer tuned to the original natural frequency of the system was best when the original critical was well within the running range, he considered that tuning to a lower frequency, perhaps as low as half the original, was worth while when the critical was at or close above full speed. The advantages of that special tuning were explained on page 53, and were well illustrated by a comparison of the X_1 curves of Figs. 18 and 19.

There, incidentally, he had a point of criticism, which was that point B of Fig. 10, defined by equation (74), depended only on K_B and K_S for its location and could not be altered by the resonance changer design. Why therefore did the X_1 curves of Figs. 17 to 21, which appeared to be calculated, not pass through that fixed point irrespective of damping? In Fig. 18, for instance, the X_1 curves for 10 and 20 damping ought to cross the original at 233 r.p.m. and $\pm 0{\cdot}004$in. amplitude.

Existing resonance changers had shown remarkable tolerance to variations in damping and for that reason the author's suggestion for maintaining constant oil temperature, and hence viscosity, had not yet been adopted although obviously desirable.

Regarding venting of the system, it would clearly be desirable in new designs to fit the thrustmeter "Limit Valve" on the uppermost cylinder and keep an upward run of pipe from bottle to thrust block. Venting would then be automatic. An addition to the paper that one would like to see was a short treatise on adjustment of the damping valve to its best position with the minimum of sea trials and instrumentation.

In conclusion, he asssured potential users that designing a one-to-one tuned resonance changer with the aid of the paper was a fairly straightforward job without too much hard labour. There was, however, plenty of the latter if one set out to produce curves of its performance. Calculation of optimum damping for a tuning other than one-to-one was also a laborious exercise, and it was singularly fortunate that one-to-one tuning was best for most applications.

MR. W. MCCLIMONT, B.Sc. (Member), said that, as many members might be aware, the British Shipbuilding Research Association had been concerned for the past few years with the problem of axial vibration of propeller shafting systems in merchant ships. A number of instances had occurred where troublesome vibrations had been experienced and investigations had been made by members of the Association's staff. At the same time they had put in hand a comprehensive research into the factors which appeared to be significant in the problem; in much of that work there was collaboration with Pametrada. He had been closely associated with the study of the problem since B.S.R.A. became concerned with it, but the comments offered were his own personal views.

It was his view that the significance of axial vibration of shafting in merchant ships was increasing and might constitute a major problem in years to come. The transmission of higher powers per shaft meant an increase in the cyclic thrust variation applied to the thrust block and foundation. Not only was it his experience that stiffness of thrust blocks and seatings fell off at higher loading with present designs, but it was his opinion that such a falling off could not easily be avoided. At the same time, the speeds of revolution of high powered shafts had been moving upward and were likely to continue to do so. There was consequently the position where axial critical frequencies were falling and service revolutions rising and the position was being attained where they became coincident. The gap between those two speeds on the majority of ships had remained so great that it had been possible until now to ignore the whole subject, but there had been ominous signs that it was not any longer possible to do so. That, in his submission, meant that a position must be attained in which it was possible to estimate axial criticals before a ship was built with a reasonable degree of accuracy.

What should be the degree of accuracy? The author's 10 per cent was about right and Mr. McClimont would be happy to take that as a target. However, so far as merchant ships were concerned, he would modify the author's statement that the value of the critical calculated should be 40 per cent above the maximum running speed and put forward the figure of 25 per cent. That was 10 per cent for errors and 15 per cent for the range below an axial critical which, in his experience, was liable to give undesirable effects on a merchant ship. That last was a somewhat empirical rule based on subjective impressions rather than sound data; but until something better was established it had seemed to him to be wise to recommend that on a merchant ship the shaft speed should never be within ±15 per cent of an axial critical. He did not know about warships.

He welcomed the paper as an extremely valuable contribution to the literature on the subject of axial vibration and particularly because it was unique in dealing with the design of resonance changers. However, he was in complete disagreement with the author when he put the resonance changer forward in the opening paragraph of page 53 as a cure all and seemed to imply that it was only necessary to investigate design variations as a matter of academic interest. Mr. McClimont looked on the resonance changer as a palliative on a merchant ship, as something to be fitted to get one out of a hole if there had been a slip-up in calculations. He would not feel happy about a resonance changer operating permanently. He feared the author's attitude was rather like telling a man he need not worry about getting sick because there was a reliable prescription for his ailment. In his view, preventive medicine was much better.

The paper, of course, dealt with the axial vibration problem in warships and quite a lot of care was needed in relating the statements in it to merchant ships. That became apparent even in the introduction in the simplified form which was envisaged. On merchant ships, in general, the significant mass was not the propeller alone but included the shaft whose weight made a much greater contribution to the total effective mass than in the case of the small, hollow shaft of the warship. The spring was not so much the propeller shaft as the thrust block, again because of the different relative dimensions.

Mr. Rigby's paper proved of great value to all who had faced the problem, but one must stress again that it referred to warships and not merchant ships and there were dangers if it were followed slavishly and not intelligently. Mr. Rigby, who was obviously in a position to treat the data intelligently, had already spoken of getting some surprises when considering a merchant ship. It was to be hoped that he would not get too many more when the liner he had been studying went to sea. For instance, the stiffnesses of thrust blocks encountered in merchant ships were substantially different, and quite different overall stiffnesses from thrust collar to earth were applicable. That was a field in which B.S.R.A. and Pametrada were doing joint investigations, and in due course it would be possible to give a set of values for K_B applicable to merchant ships. They might well differ quite a bit from the three sets of figures given on page 53. Mr. Rigby and the present author were dealing, of course, with Admiralty type blocks and Kane and McGoldrick based their data on Kingsbury blocks. Calculating seating stiffnesses appeared at the moment to be an unrewarding process, though it might be possible to get somewhere by calculating the static strength of the structure in a fairly simple form and dividing the answer by three to obtain the dynamic stiffness. Probably most of that factor came from stress (and therefore strain) concentrations that pretty well defied calculations.

It would be interesting to know what the author's experience was with the change from riveted or bolted seating and adjacent ship structure construction to all-welded construction of those parts, both in respect to stiffness and to damping.

For merchant ship calculations Mr. McClimont was not so happy about equivalent systems which transferred the mass of the shaft to the ends for reasons he had already mentioned. That meant that he did not go a long way with the author in respect of Appendix 2. Another difference on merchant ships was in the significance of the weight of entrained water to be added to the propeller. Personally, he doubted whether it was worth bothering with, but if one wanted to feel that one was ½ per cent more accurate, there was no harm in adding 60 per cent. Again it was a matter of the relative weights of propeller and shafting.

Flexible couplings failing to slide was a most interesting subject. When trying to decide what was the criterion of acceptable axial vibration, one wondered whether it was the effect on the flexible couplings. Which situation was worse—if they slid or if they did not slide?

On merchant ships those concerned would feel very unhappy long before there was thrust reversal. Panel vibration of the double bottom, and anything from harmful shaking loose of boiler refractories to rattling of the Captain's toothbrush in its glass would ensure that something had to be done about it. Gearing could rather surprisingly stand most of what was liable to happen to it. Flexible couplings did however give him some misgivings. In that connexion the author's statement on page 49 that a two-fold magnification factor might cause an unacceptable vibration was interesting, because it seemed that it was at the point where the magnification factor rose to 2 that the majority of people started to draw attention to undesirable results of axial vibration. That again was a rather unsubstantiated subjective observation which he put forward in the absence of precise data.

The author had spoken about various conditions such as waves and small rudder movements causing variation of the disturbing thrust at the propeller. It would be of interest to know whether he had encountered a noticeable difference with a following sea compared with a head sea.

Mr. S. Archer, M.Sc. (Member) said that the paper dealt with a very specialized subject with which, fortunately, most marine engineers would probably never have to concern themselves. Nevertheless, a warning had been sounded by the author, and it would be foolish not to find out all that was possible beforehand rather than be overtaken by events.

Generally speaking, the lower shaft speeds and stiffer shafts in merchant ships, at least those propelled by geared steam turbines, were sufficient to place the fundamental axial criticals above the working speed range despite the predominance of four or five-bladed screws in merchant ships, other than multi-screw passenger ships. By and large, however, even in these latter ships with faster running shafts there had been very few cases of serious resonance difficulties, the exceptions being where the thrust stiffness had been unusually low, or had softened up.

Recently there had been a few cases of severe axial vibration in motor ships, in which conditions were such that little compression took place in the line shafting, the major flexibilities being in the crankshaft and thrust block and seating. It was therefore comforting to know that through the courtesy of the Admiralty and the outstanding work of the author and his associates it was now possible to design suitable devices which might prove to be the answer to this trouble also in motor ships. It was further reassuring to hear from the author that with reasonable care—and one presumed a modicum of brain power—it was almost impossible to go wrong in designing a suitable resonance changer!

The four curves taken in conjunction with the example in Fig. 29 were most valuable from the point of view of the engineer who had to design the resonance-changing device.

On the question of propeller damping, of course in Diesel engine torsional vibration calculations it was permissible, where a special damper is fitted, to omit engine damping because the torsional vibration damper usually absorbed an amount of energy so much larger. It would be of interest to know how much of the energy was actually dissipated by the resonance changer in proportion to the propeller damping, because it would seem that if it were very much larger, the question of propeller damping was then less important. In this connexion, was there any increased heating in the thrust block after fitting the resonance changers? Presumably the use of one of the modern silicones as the damping fluid had been considered but rejected on account of contamination difficulties.

With the very heavy thrust reversal in the naval ships in question, it would be interesting to know how the thrust pads stood up to the hammering which they received. Was there severe trouble in that respect, and were there any other deleterious effects of the axial vibration?

Mr. Archer quoted a case of a merchant ship propelled by an eight-cylinder Diesel engine (10,000 b.h.p. at 120 r.p.m.) installed amidships, in which severe axial vibration of the shafting developed due to the presence of a propeller-excited 4th order critical somewhat above the service speed, and the superintendent engineer was so alarmed by the thrust movement that he refused to accept the ship. The builders had to look into the possibilities of stiffening the thrust. The thrust block was separate from the engine bedplate, and after a number of trials and errors it was decided to bridge the gap between the thrust block and the bedplate by means of a very heavy fabricated structure. By so doing it was expected that the critical would be lifted still further above 120 r.p.m. and the trouble would be over, but the curious point was that the frequency rise achieved was very small, comparatively speaking, and it was concluded that the effect of the thrust seating itself more or less swamped the possibilities in that direction. The interesting feature was that the original range of thrust variation was 24 per cent; efforts were made successfully to reduce it to 15 per cent by rephasing the propeller, but when the thrust was stiffened it went to 17 per cent. This was measured by strain gauges. The reason the ship was accepted in the end was that the superintendent could not *see* the thrust block moving! In other words, psychology played a significant part! If something was not seen to be moving, it must be all right; but it had to be borne in mind that the thrust variation might still be just as great, or even greater!

As a matter of interest, on the *Empress of Britain* trials he took one or two clock gauge readings between thrust block and gear case corresponding to a 30 per cent change in r.p.m., and these gave a thrust stiffness of something like 15,000 tons/in. He did not know how that would compare with the stiffnesses which the author had been measuring, but one would imagine that it would be much higher than those quoted, and it might well be that there was some error from the fact that the measuring point might also have been moving under the influence of the thrust forces. (Mr. Archer showed on the blackboard how the thrust block was mounted on a heavy box girder linking the gear case seating with the after engine room bulkhead.)

Commander J. I. T. Green, O.B.E., R.N.(ret.) (Member) said that he was very interested in some of the general aspects of the paper, and it seemed to him that the solution to this very serious vibration problem deserved wide circulation.

It would have been helpful if there had been a sketch showing the general arrangement of the thrustmeter, the "Limit Valve" and the pressure vessel. It was not very clear whether the valve was a relief valve or a throttle valve, and to the student in such matters it seemed important to emphasize the difference between this arrangement and an ordinary hydraulic buffer.

Another omission was any reference to the pressure variations in the oil vessel. On the last page of the paper a figure of 2,000lb./sq. in. was mentioned. The working pressure was of interest in view of the possibility of variations in bulk modulus arising due to the air content of the oil. He imagined that with such heavy damping present the pressure fluctuations would be easy to record.

Concerning the general cause of the vibration, it seemed strange that this should suddenly arise at a most critical period just before the war. He well remembered the anxiety of the Engineer-in-Chief in those early days. The author had pointed out the rising trends in power and shaft revolutions, but this did not seem a sufficient explanation.

He asked in conclusion what had happened to the chief engineer who in 1944 reported a connexion between the use of his thrustmeter and the general vibration. It would be good to know that his awareness had been recognized.

Mr. P. W. Ayling, B.Sc., said that he felt somewhat at a disadvantage in that he had only received a copy of the paper a short time before the meeting and, secondly, because he was on the other side of the fence in the naval architectural field.

He congratulated the author on a valuable paper dealing with a subject which was very much in the minds of those interested in vibration in merchant ships. The naval architect who was giving consideration to the hull vibration characteristics of a new ship, or who was faced with the problem of reducing propeller excited hull vibration in an existing vessel, appreciated the importance of avoiding the onset of severe axial movements of the propeller-shafting-thrust block system.

Like his colleague Mr. McClimont, however, he did not think that the Admiralty's experience with the success of the resonance changer should be regarded by the merchant shipbuilding world as the final answer to the problem. A passenger liner or a cargo vessel steamed, for a large proportion of its life, at sensibly constant revolutions and one questioned the applicability of the resonance changer principle under such conditions. Could the author give any information concerning the reliability of the device over long periods?

If, on the basis of design calculations for a new ship, it appeared that axial vibration was likely to be of serious consequence, and for various reasons it was not desirable to alter the propeller characteristics or move the thrust blocks—although he had never really appreciated why the thrust block must remain in the conventional position—the permanent use of a resonance changer might be contemplated. Whilst he would be the first to praise the ingenuity of those responsible for finding so neat a solution to existing problems, it was his belief that a study should continue to be made of the many factors which influenced both the response and the excitation aspects with a view to rendering the use of a resonance changer unnecessary.

The naval architect had a vital part to play with regard to the consideration of the exciting forces. Not the least important factor was a knowledge of the full-scale wake conditions likely to be encountered in service, a field of research in which progress was long overdue. The choice of the best propeller for a given ship might then include consideration of calculated thrust and torque variations.

He would welcome further details of the troubles experienced on the *Vanguard*. A serious critical occurred at 208 r.p.m with three-bladed propellers. The substitution of a five-bladed propeller might have been expected to give rise to resonant vibration at about 125 r.p.m. In fact, during trials with the new propellers a critical was detected at 152 r.p.m. which, to judge from the paper, was more severe, and although it was not specifically stated, it was inferred that the vibration was accompanied by slipping in the flexible coupling. As suggested in Fig. 24, one would expect the magnitude of the thrust variations from a five-bladed propeller at 152 r.p.m. to be much less than those from a three-bladed propeller at 208 r.p.m. The observations made would not appear to substantiate that. In other words, whilst the author's explanation of the apparent change in the critical frequency might well be reasonable, the relative severities of the vibration with the two propellers was not readily understood. Perhaps the author would enlarge on the point.

Also, on the basis of those results, one might be tempted to infer that in a new design, the substitution of a five-bladed propeller for one having a smaller number of blades would be a retrograde step, a view to which he did not necessarily subscribe. As far as the *Vanguard* was concerned, perhaps it would be more correct to say that the conditions of the system under which the five-bladed propeller was to operate were not fully appreciated.

Mr. R. R. Ostler, D.S.C. (Member) said that Rear Admiral J. G. C. Given, C.B., C.B.E. (ret.) (Member) had asked him to express his regret at being unable to be present and to congratulate the author on his valuable and comprehensive paper on the subject, and would like to raise one or two points on the paper.

Ten years ago Kane and McGoldrick gave a comprehensive paper on the subject covering experiences in the United

States Navy and merchant fleet to that date. Was there any further information or experience from the United States Navy available for release since that date, and, if so, how did it compare with the experiences in the Royal Navy covered by the author?

On page 39, attention was drawn to the effect of the "Limit Valve" in reducing the bodily movements of the shafting under condition of changes of thrust, whilst near the beginning of page 40, reference was made to the increase in axial clearance provided in the thrust boxes of H.M.S. *Eagle* to provide sufficient room for the thrust collar to vibrate. That did not appear to be entirely in keeping with the previous statement, and perhaps some clarification on the point was desirable.

In connexion with the above it would, it was felt, be helpful if a drawing of the thrust block and resonance changer fitted in H.M.S. *Eagle,* with details of the pipes and valves, etc., could be provided as an addition to the paper. Reference was made to the random results regarding the axial freedom provided by flexible couplings and the effect of that on the characteristics of the system. That was an important matter which was also referred to in Kane and McGoldrick's paper, and the author's views or any other evidence regarding the relevant factors controlling whether or not the couplings gave axial freedom would be welcomed, also whether he had any suggestions as to what investigations might be carried out to throw more light on the matter.

It would seem desirable that in any marine turbine geared installation the whole of the rotating shaft system and its supports should be quantitatively analysed for its vibratory characteristics, both axial and torsional, and the effects of the possible cyclical disturbing forces. Would the author consider that such an analysis could be handled by a comprehensive method which would enable those characteristics to be readily and rapidly evolved by a modern computer, and so assist in standardizing the necessary design criteria to ensure trouble-free operation for such an installation?

Mr. R. J. B. Keig, M.A. (Member) congratulated the author on his profound and interesting paper, and said that as he was personally associated with Mr. Rigby during a number of the earlier trials, and had had a hand in the design of some of the instruments used throughout the trials, it was hoped that a few remarks concerning recent development in that field might be of interest. .

Prior to the trials in H.M.S. *Eagle,* axial vibration of the shafting was measured by means of a piezo-electric crystal accelerometer held in contact with the free end of the shaft. The output from that type of pick-up necessitated a double integration circuit and that, together with the microphony of the battery operated amplifiers and the temperature sensitivity of the crystal, led to difficulties. Also, it was not found practicable to obtain a continuous record of the vibrations. At that time, some thought was given to the design of a seismic mass type of velocity pick-up, but the low frequencies involved and the limited space available made that idea seem somewhat impracticable.

For the trials in H.M.S. *Eagle,* the instruments described in the paper were designed and developed. That equipment had given good service during the past ten years. However, with the advent of machinery having the thrust block integral with the gear case and in the case of designs where the free end of the shaft was not available for the fitting of the inductive pick-up shown in Fig. 13, it had been found necessary to develop an axial vibration pick-up suitable for use at any convenient point along the shaft. The first design consisted of a split collar fitted to the shaft and a pick-up for recording the axial movement of the collar relative to some fixed point on the ship's structure, but it was found extremely difficult to fit the collar on the shaft so that it ran true within the limits of accuracy desired.

That difficulty now appeared to have been overcome by an ingenious device suggested by one of his colleagues at the Admiralty Engineering Laboratory. It took the form of a split collar carrying a light flange. The flange was fitted to the shaft and positioned to provide a small axial clearance (about ⅛in.) between the flange and an insulated electrode of annular form, the latter being secured to a suitable fixed point on the ship's structure by means of a bracket. The axial vibrations of the shaft thus caused small changes in the air gap of the condenser formed by the flange and the fixed electrode. The resultant changes of capacity modulated an a.c. bridge circuit (10Kc carrier) and the signal was then amplified, demodulated and recorded in the usual manner. It was a great advantage of that device that only moderate care need be taken with the alignment of the flange on the shaft as "wobble" of the flange had only a secondary effect on the accuracy of the records, due to the averaging effect of the annular form of electrode. A successful sea trial had been carried out with a prototype and an improved form of the device was being developed.

Mr. H. Page said that, as a member of the author's staff at the Admiralty, it was his pleasure to be associated with the early development work in a small way. Mr. B. C. Northam of the Admiralty staff made the major contribution as he spent many long hours calculating critical speeds. Going back nine years, at that time those working in the field were much concerned with the problem of shafting axial vibration. It was a subject which had been bequeathed by Mr. Rigby when he moved from the Engineer-in-Chief's Department. Fortunately he had left behind his paper, which provided a valuable guide to the subject. At that time the main task was to follow the recommendations of that paper, but the *Vanguard's* trials posed problems. However, the author's conception of the use of the thrustmeter as a resonance changer completely altered the character of the work as a start was made in an entirely new field.

He recalled the conversation when the idea was first mooted by the author, and the latter's remarks did not do him full justice. There was available the knowledge from Mr. Rigby's paper, but the earlier history now given was not known at that juncture, and the idea came entirely as an original approach to the problem.

A remarkable feature was the short time which was required to convert the idea to a practicable proposal and arrange for trials in H.M.S. *Eagle.* Even more surprising were the excellent results which were achieved on trials of H.M.S. *Eagle.* One had to remember that in addition there was a large amount of other propeller work, and also gearing and shafting problems. The resonance changer was only one amongst a multitude of items.

The only unfortunate result from the H.M.S. *Eagle* trials was that the critical expected with the changer in use did not show up. He said "unfortunate", as most of the work that had been undertaken since had been virtually finding the answer to that problem. Consequently one was faced with a rather frightening formula such as equation (10) on page 41. That was a case where it was fortunate that theory had been made to fit practice. There was a compensating feature in that the design was now much better understood and the results were predictable.

The first efforts were rather hit and miss, and in addition many assumptions had to be made. It was very fortunate, therefore, that the resonance changer provided such a large degree of tolerance, otherwise the target might have been missed. Since its inception the resonance changer had virtually eliminated the troublesome and objectionable axial vibration, and the paper would no doubt become a standard reference on the subject.

Service experience with resonance changers over several years had been excellent. Minor pump failures and pipe fractures had been the only defects experienced. The pump defect had now been resolved. So that equipment, which involved little complication, could be fitted with confidence where there were difficulties with vibration.

Dr. J. F. Shannon said that the author was to be congratulated on carrying through the investigation to the stage where a simple and effective solution had been obtained, the results of which, in some cases, considerably exceeded expectations. The mathematical treatment of the behaviour of the resonance changer was convincing and the design of the apparatus had been made quite simple.

The behaviour of the resonance changer was similar to that of the damped vibration absorber.

Two essential features were

(1) correct tuning, and

(2) optimum damping.

With correct tuning, the heights of the two characteristic points *P* and *Q*, through which the response curve must pass, irrespective of damping, could be reduced to a minimum.

With the corresponding optimum damping the maximum amplitude of the response curve could be kept at a value no greater than the value at either *P* or *Q*. The optimum damping was shown by the author in equation (22) as a function of the stiffness ratio.

This applied irrespective of the external system, hence the tuning and optimum damping as deduced by the author when neglecting all extraneous damping and the gearing mass was quite justified.

The results shown in the paper indicated that correct values of tuning and damping had been chosen and that their choice was not a matter of precision. The effectiveness of the resonance changer was remarkable for such a small piece of simple equipment. As shown in equation (2), its power lay in the benefit gained by the terms being multiplied by the square of the ratio of piston area to pipe area, which, in the case quoted, was $(133)^2$. This permitted effective operation with a thrust collar movement of $\pm 0{\cdot}002$in.

This effect was very much greater than could be expected from the oil films in the thrust pads. Approximate estimation of the stiffness of the oil films in the thrust pads pointed to their stiffness being greater than the thrust block stiffness, hence their effect was negligible. While this appeared to be the case in thrust bearings, the flexibility and damping of the journal oil films were usually greater than that of the bearing housing of steam turbine sets, but it was clear that journal oil films alone could not perform as successfully in damping whirling vibration as could a suitable oil damped vibration absorber, or resonance changer, operated by journal movements.

The investigations reported in the paper were carried out on machinery with double helical gearing where there could be little or no coupling effect between torsional and axial movements.

A case experienced by the writer occurred in the H.M.S. *Bold Pathfinder,* a vessel with four propellers, the two inner shafts being driven by Diesel engines and the two outer shafts being driven by G.2 gas turbines through single-reduction, single-helical gearing. The main Michell thrust blocks of the gas turbine engines were incorporated in the gearbox.

No vibration was observed under normal operation. During the exceptional condition of the ship running with the port gas turbine only and at the full power of that engine, and with the rudders over to turn the ship to port, very severe vibration occurred.

Investigation showed that this was not due to axial resonance or combined axial and torsional resonance coupled through the single helical gear, but that it was due to the adverse conditions at the propeller, producing severe coupled vibration of the ship and thrust block. The question now arose, could a suitable resonance changer deal with such coupled vibrations? Would it also damp the vibrations usually associated with astern running?

It would seem possible that if there were coupled torsional and axial vibrations, as could occur with single helical gearing, the resonance changer, damping the axial vibration, would reduce the general amplitude of the combined vibration.

Correspondence

Mr. S. Archer, M. Sc. (Member), found that a closer study of Commander Goodwin's valuable paper had raised a number of questions, mostly concerning the worked example in Appendix 4, "Design of Resonance Changer".

Firstly, by comparison of principal data, the hypothetical vessel in Appendix 4 was not dissimilar to H.M.S. *Bulwark*. On this assumption and using Gawn's 1953 I.N.A. wide-bladed naval type propeller series* in conjunction with the following data from "Jane's Fighting Ships" for H.M.S. *Bulwark*, viz.:

Full load displacement 27,000 tons

L.B.P. 650 ft.

Beam 90 ft.

Draught 27 ft.

Two shafts giving, at 240 r.p.m., a total of 78,000 s.h.p. for a speed of 28 knots—

the dimensions of suitable four-bladed propellers could be approximately:

Diameter 13·9 ft.

Pitch 16·68 ft. ($P/D = 1{\cdot}2$)

Developed area 167 sq. ft. (D.A.R. $= 1{\cdot}1$)

(Gawn's data were for three-bladed screws but the differences were probably small for four-bladed screws of equivalent performance.)

If a_p was the propeller damping factor in axial vibration expressed as tons/in./sec., then using the curves of non-dimensional thrust coefficient, $K_T = \dfrac{T}{\rho n^2 D^4}$, against non-dimensional advance coefficient, $J = \dfrac{V_a}{nD}$, where

T = thrust in tons at 60 n revolutions per min.

ρ = mass density of sea water = 0·000074 tons sec²/in.

n = propeller revolutions per sec.

D = propeller diameter in ft.

V_a = propeller speed of advance in ft. per sec.

= $1{\cdot}69 V_s (1 - \omega_T)$

V_s = ship speed at full power in knots

ω_t = Taylor wake $= 1 - \dfrac{V_a}{V_s}$

it was possible to derive the theoretical axial vibration damping coefficient in the following convenient expressions:—

$$\text{or } a_p = \left(\frac{\partial T}{\partial J} \times \frac{\partial J}{\partial V_a}\right) n = \text{constant}$$

$$= \frac{1}{nD} \times \left(\frac{\partial T}{\partial J}\right) n = \text{constant}$$

$$= \frac{1}{nD} \times \rho n^2 D^4 \left(\frac{\partial K}{\partial J}\right) n = \text{constant}$$

$$a_p = \rho\, n\, D^3 f'(J)$$

where $f'(J)$ was the slope of the K_T curve of constant pitch ratio relative to the J abscissa axis and was, in general, always negative.

Now, assuming $T \propto n^2$, and $V_a \propto n$,

$$\therefore a_p = A_p \times \frac{n}{N}$$

where a_p = propeller axial vibration damping coefficient at n revolutions per sec.

A_p = corresponding value of a_p at full power revolutions, N.

Thus, for the example in Appendix 4 and for H.M.S. *Bulwark*,

* Gawn, R. W. L. 1953. "Effect of Pitch and Blade Width on Propeller Performance". Trans.I.N.A., Vol. 95, p. 157.

the following estimate of propeller damping coefficient was obtained using Gawn's Fig. 10, viz.:

$$\mathcal{J} = \frac{V_a}{nD}$$

$$= \frac{28\,(1-0{\cdot}05)^{1} \times 1{\cdot}69}{\frac{240}{60} \times 13{\cdot}9}\text{, if } \omega_t = 0{\cdot}05 \text{ (say)}$$

$$= 0{\cdot}81$$

$$A_p = \rho n D^3 f'(\mathcal{J})$$

$$= 0{\cdot}000074 \times \frac{240}{60} \times (13{\cdot}9)^3 \times 0{\cdot}63$$

$$= 0{\cdot}5 \text{ tons sec./in.}$$

$$\therefore a_p = 0{\cdot}5\,\frac{n}{N} \text{ tons sec./in.}$$

This compared with a value of 0·48 tons sec./in. by Kane and McGoldrick's formula (p. 47 of this paper).

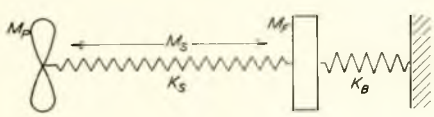
Taking a developed blade area of 167 sq. ft., the above value of A_p worked out at $\frac{0{\cdot}5}{167} = 0{\cdot}003$ tons sec./in./sq. ft. of developed blade area or 0·006 for a value of $A_p = 1{\cdot}0$, as assumed by the author throughout the paper. This compared with Rigby's value for overall damping (equivalent total at propeller) of 0·00739 tons sec./in./sq. ft. of developed blade area.

Having thus estimated the probable approximate dimensions and damping properties of the propellers for both the illustrative example (Appendix 4) and H.M.S. *Bulwark*, it was thought well to attempt an analysis of these examples with a view to assessing the relative amounts of damping dissipated at the various damping points and to compare the measured and calculated propeller amplitudes and frequencies for *Bulwark*.

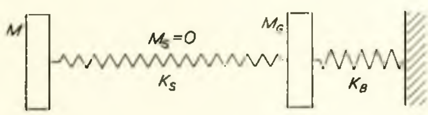
I. Illustrative Example (Appendix 4)

(a) *Shaft System Data* (no resonance changer):

Actual system
(heavy shaft)



Equivalent system
(Weightless shaft)



M_P = 26·6 tons
M_S = 55 tons
M_F = 33·2 tons
K_S = 560 tons/in.
K_B = 2,500 tons/in.

For one-node critical $Z_{c1} = 0{\cdot}922$ and $\omega_{c1} = 58{\cdot}1$ rad./sec.

$$\therefore \frac{X_1}{X_p} = \frac{Z_{c1}\operatorname{cosec} Z_{c1}}{\frac{K_B}{K_S} - \frac{M_F}{M_S} Z_{c1}^2 + Z_{c1}\cot Z_{c1}} = 0{\cdot}249$$

It was now possible to complete the one-node undamped frequency table giving a 4th order critical speed of 139 r.p.m. (see Table III). The two-node frequency would be well above the maximum service speed, i.e. at 378 r.p.m.

Using the energy method, it was interesting to examine the equilibrium amplitudes under the following conditions:

(1) *Propeller damping only* ($a_p = 1{\cdot}0$ as assumed by the author).

$$P_o = 0{\cdot}04 \times \left(\frac{138{\cdot}7}{240}\right)^2 \times 145 = \pm 1{\cdot}94 \text{ tons}$$

Energy input = $1{\cdot}94\,x_p$

Propeller damping energy = $1{\cdot}0 \times 58{\cdot}1 \times x_p^2$

Equating energies, $x_p = \pm\frac{1{\cdot}94}{58{\cdot}1} = \pm 0{\cdot}0334$ in.

Reference to Figs. 20 and 21 showed that this gave good agreement with the measured values for *Bulwark*, bearing in mind the 10 per cent lower excitation for that ship and allowing for the point of measurement not being quite at the propeller position.

(2) *True propeller damping* (based on above formula).

$$a_p = 0{\cdot}5 \times \frac{138{\cdot}7}{240} = 0{\cdot}289 \text{ tons sec./in.}$$

$$\therefore x_p = \pm 0{\cdot}0334 \times \frac{1}{0{\cdot}289} = \pm 0{\cdot}115 \text{ in.}$$

This was clearly much too high and additional damping must therefore be present in the system. Bearing in mind the severe damage to flexible coupling teeth, etc., frequently reported, and the probability of hydrodynamic damping in the thrust bearing oil films, etc., it seemed reasonable to account for the additional damping by an equivalent viscous damping source at the thrust collar having an amplitude of $\pm x_1$ in. and a coefficient a_G tons sec./in.

On this assumption, one obtained from *Bulwark* (see Table VI):

$$P_o = 0{\cdot}036 \times 140 \times \left(\frac{136}{240}\right)^2 = \pm 1{\cdot}62 \text{ tons}$$

$$\text{Propeller damping energy} = 0{\cdot}5 \times \frac{136}{240} \times 57\,x_p^2 = 16{\cdot}15 x_p^2$$

Additional damping energy

$$= a_G \times 57 \times (0{\cdot}252)^2 x_p^2 = 3{\cdot}62\,a_G\,x_p^2$$

Also x_p was measured at about $\pm$ 0·028 in. (say)

$$\therefore \frac{1{\cdot}62}{16{\cdot}15 + 3{\cdot}62\,a_G} = 0{\cdot}028$$

$$\therefore a_G = 11{\cdot}5 \text{ tons sec./in.}$$

Applying this extra damping to the example of Appendix 4, one got:

Excitation = $1{\cdot}94\,x_p$

$$\text{Propeller damping energy} = 0{\cdot}5 \times \frac{138{\cdot}7}{240} \times 58{\cdot}1 x_p^2 = 16{\cdot}8 x_p^2$$

Additional damping energy

$$= 11{\cdot}5 \times 58{\cdot}1 \times (0{\cdot}249)^2 x_p^2 = 41{\cdot}4\,x_p^2$$

$$\therefore x_p = \pm\frac{1{\cdot}94}{58{\cdot}2} = \pm 0{\cdot}0333 \text{ in.}$$

which also agrees with *Bulwark's* measured value.

(b) *With Resonance Changer Fitted*

R.C. data

A_o = 265 in.²
V_1 = 4·5 cu. ft. = 7,776 in.³
B_1 = 2 × 10⁵ lb./in.² = 89·3 tons/in.²
L_1 = 6·17 ft. = 74 in.

$$A_1 = \frac{\pi}{4} \times 1^2 = 0{\cdot}7854 \text{ in.}^2$$

$$\rho_1 = \frac{54}{386{\cdot}4 \times 12^3} = 81 \times 10^{-6} \text{ lb./in.}^{-4}\text{/sec.}^2$$

$$= 3{\cdot}61 \times 10^{-8} \text{ tons/in.}^4\text{/sec.}^2$$

$$\mu_1 = 233 \text{ centipoises} = \frac{233 \times 14{\cdot}53 \times 10^{-8}}{2{,}240}$$

$$= 1{\cdot}51 \times 10^{-8} \text{ tons/in.}^2\text{/sec.}$$

$$m_1 = \rho_1 \frac{A_o^2\,L_1}{A_1} = \frac{3{\cdot}61 \times 10^{-8} \times 265^2 \times 74}{0{\cdot}7854}$$

$$= 0{\cdot}2386 \text{ tons/in./sec.}^2$$

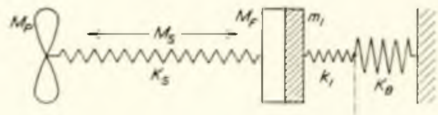
$$k_1 = \frac{B_1 A_o^2}{V_1} = \frac{89{\cdot}3 \times 265^2}{7{,}776} = 806 \text{ tons/in.}$$

$$c_1 = 8\pi\,\mu_1 L_1 \left(\frac{A_o}{A_1}\right)^2$$

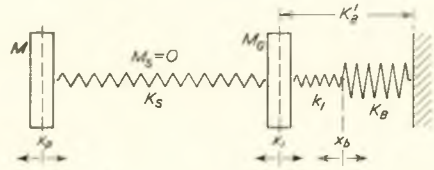
$$= 25{\cdot}1 \times 1{\cdot}51 \times 10^{-8} \times 74 \times \left(\frac{265}{0{\cdot}7854}\right)^2$$

$$= 3{\cdot}19 \text{ tons/in./sec.}$$

Actual system
(heavy shaft)



Equivalent system
(weightless shaft)



Using the data given in Appendix 4 and the method of substitution for the shaft mass as set out in Appendix 2, the equivalent weightless shaft systems for one and two-node modes, allowing for mass and stiffness of resonance changer, were as shown in the frequency tables (Tables IV and V, respectively).

From these the following amplitudes, x_p, were derived, using the energy equilibrium method as before, but including also the additional damping at the thrust collar ($a_G\,\omega_c x_1^2$ per cycle):

One-node mode, 4th order at 77·5 r.p.m., $\omega_{c1} = 32{\cdot}4$ rad./sec.

$$\text{Excitation energy} = 0{\cdot}04 \times \left(\frac{77{\cdot}5}{240}\right)^2 \times 145\,x_p = 0{\cdot}604\,x_p$$

$$\text{Propeller damping energy} = 0{\cdot}5 \times \frac{77{\cdot}5}{240} \times 32{\cdot}4\,x_p^2 = 5{\cdot}23\,x_p^2$$

$$\text{Additional damping energy} = 11{\cdot}5 \times 32{\cdot}4 \times 0{\cdot}75^2\,x_p^2 = 210\,x_p^2$$

$$\text{R.C. damping energy} = 3{\cdot}19 \times 32{\cdot}4 \times 0{\cdot}567^2\,x_p^2 = 33{\cdot}3\,x_p^2$$

$$\therefore\ 0{\cdot}604\,x_p = 248{\cdot}5\,x_p^2$$

$$\therefore\ x_p = \pm 0{\cdot}00243 \text{ in.}$$

TABLE III.—EXAMPLE (APPENDIX 4) WITHOUT RESONANCE CHANGER; ONE-NODE, 4TH ORDER AT 138·7 R.P.M.

Mass	Tons weight, W	Tons mass, (tons sec²/in.), $\frac{W}{g}$	Tons/in., $\frac{W}{g}\omega_c^2$	Amplitude, $\pm x$ (in.)	Inertia force, $\pm\frac{W}{g}\omega_c^2 x$(tons)	Σ Inertia forces, $\Sigma\pm\frac{W}{g}\omega_c^2 x$(tons)	Shaft stiffness, C (tons/in.)	Deflexion, δx (in.)	
M	48·1	0·1246	420·5	1·0000	420·5	420·5	560	0·751	$\omega_{c1}^2 = 3{,}376$
M_G	93·2	0·241	814	0·249	202·5	622·5	2,500	0·249	$\omega_{c1} = 58{\cdot}1$ rad./sec.
"Earth"	∞	∞	∞	0					F = 555 V.P.M.

TABLE IV.—EXAMPLE (APPENDIX 4) WITH RESONANCE CHANGER; ONE-NODE, 4TH ORDER AT 77·5 R.P.M.

M	51·5	0·1332	140	1·0000	140	140	560	0·25	$\omega_{c1}^2 = 1{,}050$
M_G	155	0·402	422	0·75	317	457	806	0·567	$\omega_{c1} = 32{\cdot}4$ rad./sec.
							2,500	0·183	F = 310 V.P.M.
"Earth"	∞	∞	∞	0					

TABLE V.—EXAMPLE (APPENDIX 4) WITH RESONANCE CHANGER; TWO-NODE, 4TH ORDER AT 208 R.P.M.

M	41·8	0·1081	822	1·0000	822	822	560	1·468	$\omega_c^2 = 7{,}600$
M_G	120	0·3105	2,360	−0·468	−1,108	−286	806	−0·354	$\omega_c = 87{\cdot}2$ rad./sec.
							2,500	−0·114	F_2 = 832 V.P.M.
"Earth"	∞	∞	∞	0					

TABLE VI.—H.M.S. *Bulwark*, WITHOUT RESONANCE CHANGER; ONE-NODE, 4TH ORDER AT 136 R.P.M.

M	50·5	0·1308	425	1·000	425	425	567	0·748	$\omega_{c1}^2 = 3{,}250$
M_G	96·6	0·250	814	0·252	205	630	2,500	0·252	$\omega_{c1} = 57$ rad./sec.
									F_1 = 544 V.P.M.
"Earth"	∞	∞	∞	0					

TABLE VII.—H.M.S. *Bulwark*, WITH 9-FT.³, RESONANCE CHANGER; ONE-NODE, 4TH ORDER AT 53·5 R.P.M.

M	54·0	0·140	70·8	1·000	70·8	70·8	567	0·125	$\omega_{c1}^2 = 506$
M_G	238·0	0·615	312	0·875	273·2	344	466	0·738	$\omega_{c1} = 22{\cdot}5$ rad./sec.
							2,500	0·137	F_1 = 214 V.P.M.
"Earth"	∞	∞	∞	0					

TABLE VIII.—H.M.S. *Bulwark*, WITH 9-FT.³ RESONANCE CHANGER; TWO-NODE, 4TH ORDER AT 188 R.P.M.

M	45·1	0·1166	724	1·000	724	724	567	1·277	$\omega_{c2}^2 = 6{,}210$
M_G	187	0·484	3,010	−0·277	−833	−109	466	−0·234	$\omega_{c2} = 78{\cdot}7$ rad./sec.
							2,500	−0·043	F = 752 V.P.M.
"Earth"	∞	∞	∞	0					

Two-node mode, 4th order at 208 r.p.m., $\omega_{c2} = 87{\cdot}2$ rad./sec.

$$\text{Excitation energy} = 0{\cdot}04 \times \left(\frac{208}{240}\right)^2 \times 145\, x_p = 4{\cdot}36\, x_p$$

$$\text{Propeller damping energy} = 0{\cdot}5 \times \frac{208}{240} \times 87{\cdot}2\, x_p^2 = 37{\cdot}8\, x_p^2$$

$$\text{Additional damping energy} = 11{\cdot}5 \times 87{\cdot}2 \times 0{\cdot}468^2\, x_p^2 = 220\, x_p^2$$

$$\text{R.C. damping energy} = 3{\cdot}19 \times 87{\cdot}2 \times 0{\cdot}355^2\, x_p^2 = 35\, x_p^2$$

$$\therefore 4{\cdot}36\, x_p = 292{\cdot}8\, x_p^2$$

$$\therefore x_p = \pm 0{\cdot}0149 \text{ in.}$$

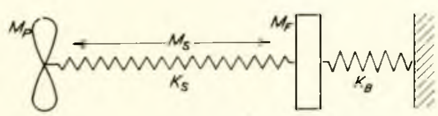
It would be observed that the R.C. damping energy in the two-node mode was actually slightly *less* than the propeller damping energy and both were swamped by the assumed additional damping at the thrust collar and gears. The propeller amplitude in the one-node mode was seen to be almost imperceptible and this doubtless explained why the author was unable to record this critical.

II. H.M.S. Bulwark

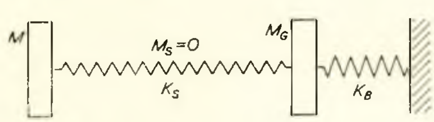
In this case, applying the data given in Table I of the paper and the methods of Appendix 2, similar calculations had been made for the 9-cu. ft. resonance changer, with results as follows:

(a) *Without Resonance Changer*
Shaft system data

Actual system
(heavy shaft)



Equivalent system
(weightless shaft)



$M_P = 26{\cdot}6$ tons, $M_S = 58$ tons, $M_F = 34{\cdot}8$ tons

$K_S = 567$ tons/in., $K_B = 2{,}500$ ton/sin.,

$$\frac{K_B}{K_S} = 4{\cdot}41, \quad \frac{M_P}{M_S} = 0{\cdot}458, \quad \frac{M_F}{M_S} = 0{\cdot}600$$

$$\tan Z_c = \frac{\dfrac{K_B}{K_S} - \left(\dfrac{M_P}{M_S} + \dfrac{M_F}{M_S}\right) Z_c^2}{Z_c\left[1 + \dfrac{M_P}{M_S}\left(\dfrac{K_B}{K_S} - \dfrac{M_F}{M_S} Z_c^2\right)\right]} = \frac{4{\cdot}41 - 1{\cdot}058\, Z_c^2}{Z_c\,[3{\cdot}02 - 0{\cdot}275\, Z_c^2]}$$

Try $Z_c = 0{\cdot}925$ radians $= 53$ degrees

$\therefore \tan Z_c = 1{\cdot}327$

$$\tan Z_c = \frac{4{\cdot}41 - 0{\cdot}98}{0{\cdot}935\,[3{\cdot}02 - 0{\cdot}235]} = \frac{3{\cdot}43}{2{\cdot}578} = 1{\cdot}330$$

$$\omega_c = 61{\cdot}6\, Z_c = 57 \text{ radians/sec. giving } N_c = \frac{57 \times 9{\cdot}55}{4} = 136 \text{ r.p.m.}$$

$$\frac{X_1}{X_p} = \frac{Z \operatorname{cosec} Z}{\dfrac{K_B}{K_S} - \dfrac{M_F}{M_S} Z^2 + Z \cot Z} = \frac{0{\cdot}925 \times 1{\cdot}252}{4{\cdot}41 - 0{\cdot}513 + 0{\cdot}697} = 0{\cdot}252$$

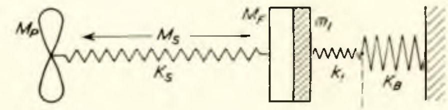
The frequency table for the condition "*Without R.C.*" was shown in Table VI and, as already noted, a value for x_p at the one-node critical occurring at 136 r.p.m. (measured) could be derived at $\pm 0{\cdot}028$ in., allowing true propeller damping and additional damping at gears, etc., only.

(b) *With Resonance Changer Fitted* (9 cu. ft.)
R.C. data (from Table I of the paper)

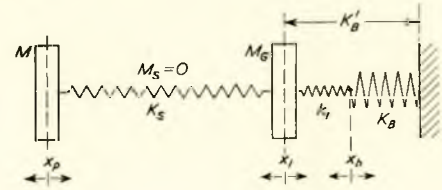
$m_1 = 0{\cdot}45$ tons sec.^{2}/in.
$k_1 = 466$ tons/in.
$c_1 = 13$ ton sec./in. (from page 49)

(i) *One-node mode*

Actual system
(heavy shaft)



Equivalent system
(weightless shaft)



$$\frac{K'_B}{K_S} = 0{\cdot}693, \quad \frac{M_p}{M_S} = 0{\cdot}458, \quad \frac{M'_F}{M_S} = 3{\cdot}600$$

$[M'_F = M_F + m_1]$

$$\therefore \tan Z_c = \frac{0{\cdot}693 - 4{\cdot}058\, Z_c^2}{Z_c\,[1{\cdot}318 - 1{\cdot}651\, Z_c^2]}$$

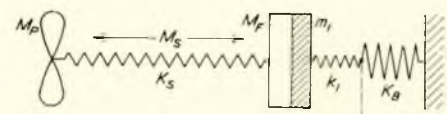
$\therefore Z_{c1} = 0{\cdot}365$, $\omega_{c1} = 61{\cdot}6\, Z_c = 22{\cdot}5$ radians/sec. giving
$N_{c1} = 53{\cdot}7$ r.p.m.

$$\frac{X_1}{X_p} = \frac{0{\cdot}365 \times 2{\cdot}801}{0{\cdot}693 - 0{\cdot}478 + 0{\cdot}954} = \frac{1{\cdot}022}{1{\cdot}169} = 0{\cdot}875$$

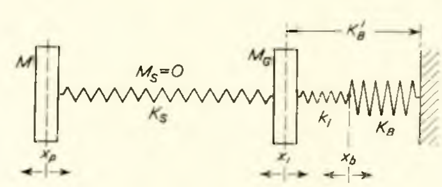
Table VII shows the frequency table for the one-node mode.

(ii) *Two-node mode*

Actual system
(heavy shaft)



Equivalent system
(weightless shaft)



$$\frac{K'_B}{K_S} = 0{\cdot}693 \quad \frac{M_p}{M_S} = 0{\cdot}458 \quad \frac{M'_F}{M} = 3{\cdot}600$$

$M'_F = M_F + m_1$

$$\therefore \tan Z_{c2} = \frac{0{\cdot}693 - (0{\cdot}458 + 3{\cdot}6)\, Z_{c2}^2}{Z_{c2}\,[1 + 0{\cdot}458\,(0{\cdot}693 - 3{\cdot}6\, Z_{c2}^2)]}$$

from which

$Z_{c2} = 1{\cdot}28$, $\omega_{c2} = 61{\cdot}6\, Z_{c2} = 78{\cdot}8$ radians/sec.,
giving

$$N_{c2} = \frac{752}{4} = 188 \text{ r.p.m.}$$

Also:

$$\frac{X_1}{X_p} = \frac{1{\cdot}28 \times 1{\cdot}0438}{0{\cdot}693 - 5{\cdot}9 + 0{\cdot}384} = -\frac{1{\cdot}337}{4{\cdot}823} = -0{\cdot}277$$

Table VIII showed the frequency table for the two-node mode.

The resonant amplitudes for both modes when fitted with the resonance changer could now be calculated as follows:

One-node mode, 4th order at 53·7 r.p.m., $\omega_{c1} = 22{\cdot}5$

Excitation energy $= 0{\cdot}036 \times \left(\frac{53{\cdot}7}{240}\right)^2 \times 140\, x_p$
$= 0{\cdot}253\, x_p$

Propeller damping energy $= 0{\cdot}5 \times \frac{53{\cdot}7}{240} \times 22{\cdot}5 \times x^2_p$
$= 2{\cdot}68\, x^2_p$

Additional damping energy $= 11{\cdot}5 \times 22{\cdot}5\, (0{\cdot}875)^2 \times x_p^2$
$= 198{\cdot}2\, x^2_p$

R.C. damping energy $= 13 \times 22{\cdot}5 \times (0{\cdot}738)^2 \times x_p^2$
$= 159{\cdot}2\, x_p^2$

$\therefore\ x_p = \pm \frac{0{\cdot}253}{360{\cdot}1} = \pm\, 0{\cdot}0007$ in. (undetectable)

Two-node mode, 4th order at 188 r.p.m., $\omega_{c2} = 78{\cdot}8$

Excitation energy $= 0{\cdot}036 \times \left(\frac{188}{240}\right)^2 \times 140\, x_p$
$= 3{\cdot}09\, x_p$

Propeller damping energy $= 0{\cdot}5 \times \frac{188}{240} \times 78{\cdot}8 \times x_p^2$
$= 30{\cdot}8\, x_p^2$

Additional damping energy $= 11{\cdot}5 \times 78{\cdot}8 \times (0{\cdot}277)^2\, x_p^2$
$= 69{\cdot}6\, x_p^2$

R.C. damping energy $= 13 \times 78{\cdot}8 \times (0{\cdot}234)^2\, x_p^2$
$= 56{\cdot}1\, x_p^2$

$\therefore\ x_p = \pm \frac{3{\cdot}09}{156{\cdot}5} = \pm\, 0{\cdot}0197$, say $\pm\, 0{\cdot}020$ in.

Again it was seen that the one-node critical would not be expected to appear and the two-node critical calculated by the author's method should resonate considerably higher in the speed range than it appeared to have been measured, i.e. at 188 r.p.m. instead of about 147 r.p.m. as shown in Fig. 20.

Whilst on the subject of natural frequency determinations, the author stated on page 53 (left hand column) that the solution of the equation for tan Z_c in terms of $f(Z_c)$ in the first quadrant gave the first critical speed and the solution in the second quadrant gave the second critical speed.

This was not, in general, correct, however, since it depended on the exact form of the expression of $f(Z)$, and as shown in the graph (Fig. 30) based on the author's illustrative example (Appendix 4), the first two roots of the frequency equation in that case fell in the first quadrant and the third root in the third quadrant. It was as well to avoid any misunderstanding on this point.

The calculated two-node propeller amplitude was some 50 per cent greater than the maximum value shown in Fig. 20 but possibly only about 33⅓ per cent greater if allowance were made for the point of measurement not being exactly at the propellers (see Fig. 31).

It was difficult to be precise as to the cause of the lower position of the two-node critical but since the frequency without R.C. required a thrust stiffness of 2,500 tons in. and presumably no appreciable axial vibration of turbine rotors was then reported, this seemed to be in line with experience in other vessels of comparable type. Thus, suspicion would tend to fall upon the resonance changer stiffness and possibly air might not have been completely excluded or even the volumetric strains in the steel bottle might not have been negligible.

In general, it would appear that propeller damping could not safely be dealt with in the manner proposed by the author, i.e. using a flat value of about twice the theoretical at full power and without due allowance for reduction proportional to r.p.m. Further, so long as any additional heavy source of damping was present in the system, the elaboration of theory to optimize

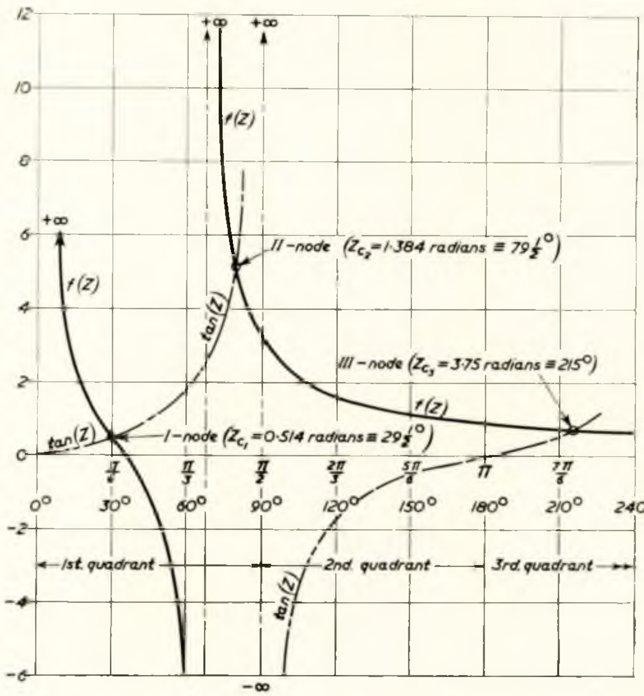


FIG. 30—*Based on illustrative example (Appendix 4)*

$$\tan Z_c = \frac{1{\cdot}087 - 2{\cdot}764 Z_c^2}{Z_c(1{\cdot}526 - 1{\cdot}103 Z_c^2)} = f(Z_c)$$

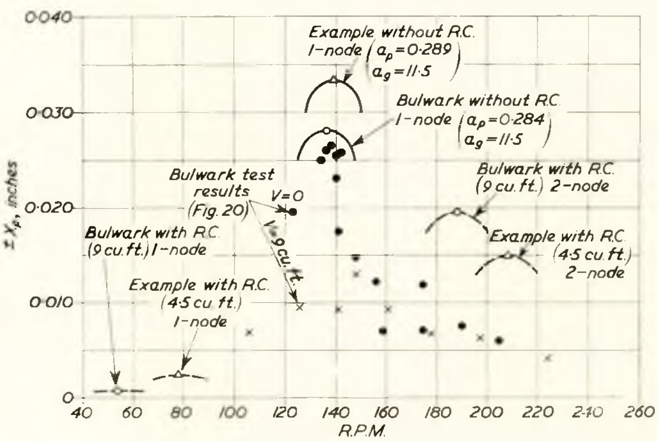


FIG. 31

R.C. damping was not justifiable and that might perhaps help to explain the author's statement that vibration amplitudes were relatively insensitive to R.C. design variations.

Calculations based on the example in Appendix 4 would suggest that the heating effect from a resonance changer was unlikely to be great, even in the connecting pipes, and this might be partly due to the laminar nature of the flow. Obviously, if turbulent flow could develop, such as in a throttle valve, the damping energy and hence heating would be expected to increase.

A further point was that in the two-node modes considered the relative amplitude $(x_1 - x_b)$ at the thrust block governing R.C. damping was usually smaller than that at the propeller.

In general, it appeared to the writer that the main function of the type of resonance changer described in the paper was indeed what its name inferred and its damping properties might well be of secondary importance.

The author's comments on the points raised above would be much appreciated.

The writer was indebted to his colleague, Mr. A. E. Toms, B.Sc., for his unstinted assistance in the preparation of this contribution to the discussion.

ING. ALBERTO GUGLIELMOTTI had read with great interest the text of the paper given by Commander Goodwin and he thought it might be useful to contribute some details of his company's experience on a similar question.

They had also recorded recently axial vibrations on the shafting of some merchant ships driven by Diesel engines; the cases investigated, however, were slightly different from those examined by the author, for the rigid connexion of the shafting to the engine crankshaft, which also underwent vibration, and the vibrations themselves, were excited not only by the propeller but also by the action of the piston on the crankshaft cranks.

These vibrations occurred quite unperceived on most of the propelling sets, in as much as the resonance of the main exciting actions dropped outside the normal operating range; in some plants, because of the special values of elasticity and of mass, there occurred instead a resonance with considerable amplitudes of vibration which it was considered advisable to eliminate.

The elastic lines of the vibration systems which were examined and the distribution of the forces of inertia were of the same type as those illustrated for instance in Fig. 32 for the first, second and third grade of vibration respectively; the vertical segments at the top of the figure indicated the several masses at which the actual system had been reduced; the horizontal segments represented the elasticity of the system, and the earth connexion the elasticity of the thrust bearing.

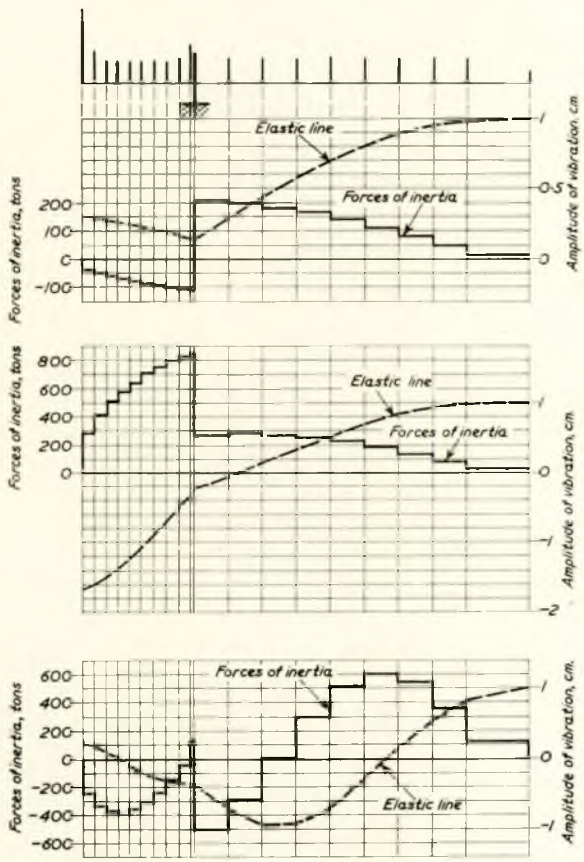


FIG. 32—*Fiat engine: elastic lines and diagrams of forces of inertia for the first three modes of vibration*

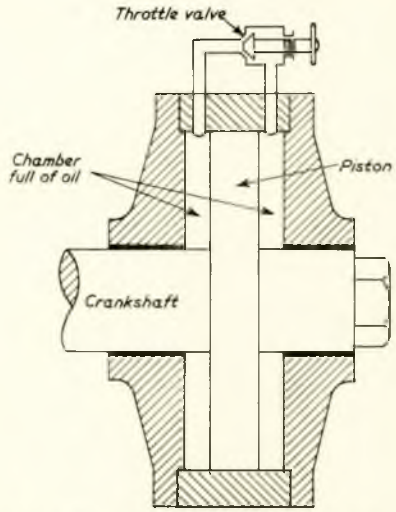


FIG. 33—*Axial vibration damper*

The problem of eliminating the vibrations was solved by fitting a damper at the free end of the crankshaft: this damper was shown schematically in Fig. 33; it consisted of a double acting piston sliding inside a cylinder full of oil which was forced to pass from one chamber into another through adjustable passages.

This system proved to be so efficient that the piston dimensions could be restricted, also on the more powerful plants, to 360-mm. bore; Fig. 34 gave an idea of the proportions of the damper and engine. Of course, in solving the problem they were greatly helped by the fact that the damper could be fitted at the free end of the crankshaft where rather

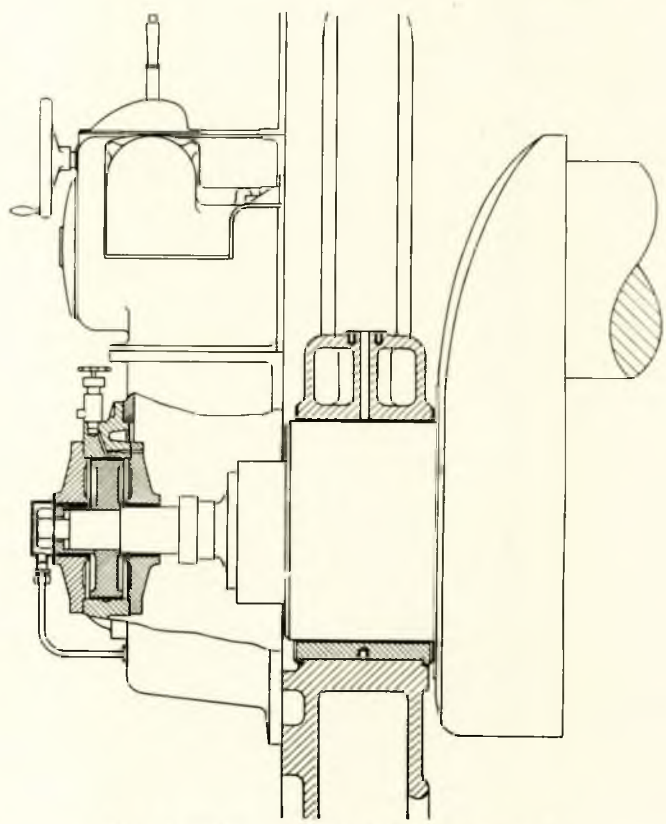


FIG. 34—*Frame of axial vibration damper*

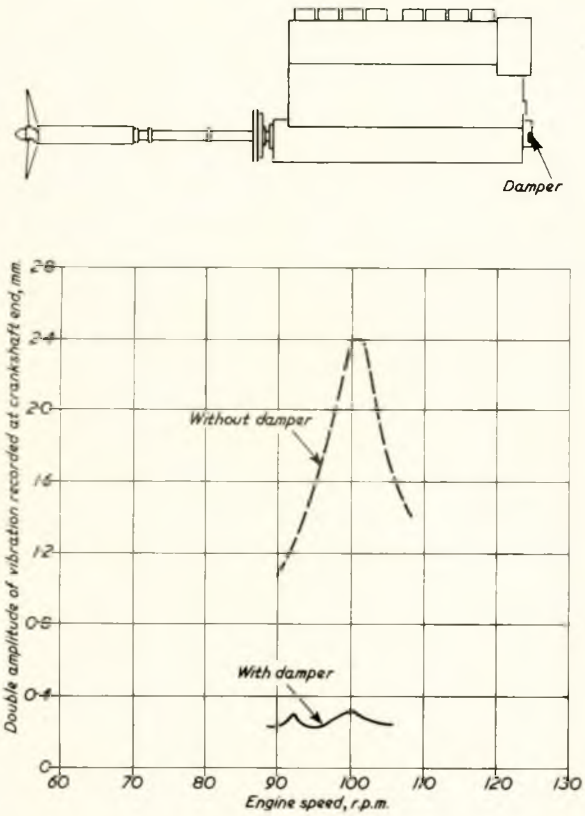


FIG. 35—*Recordings of axial vibrations taken from 758T engine in m.s.* Berkersheim

large amplitudes of vibration occurred, and in which position the damper proved to be most efficacious.

At the same time, a mathematical interpretation with quite good approximation was elaborated to analyse the experimental results and to determine the optimum characteristics of the damper for maximum efficiency.

The mathematical calculation was based on the simple hypothesis that the damper would apply forces of purely viscous type and of sufficiently small value to affect the amplitude of vibration, but not the frequency and shape of the elastic line of the vibrating system; it took into account of course the elasticity of the rigid supporting frame of the damper, which had a remarkable value with respect to the elasticity of the whole system.

This mathematical calculation, as well as the elaboration of the experimental results, was still being worked out; some interim results had already been published in Fiat Stabilimento Grandi Motori Technical Bulletin No. 2/1959; it was their intention to publish the full results of the experiments in an early number of the Bulletin.

As an example, Fig. 35 showed the amplitudes of vibration recorded before and after the installation of the damper in a propelling set with the following characteristics:

Engine	Borsig-Fiat two-stroke
Bore	750 mm.
Stroke	1,320 mm.
Number of cylinders	8
Weight of propeller	13,000 kg.
Length of shafting	14 m.

Fig. 36 showed similar results in a propelling set with the following characteristics:

Engine	Fiat two-stroke supercharged
Bore	680 mm.
Stroke	1,200 mm.
Weight of propeller	9,000 kg.
Length of shafting	48 m.

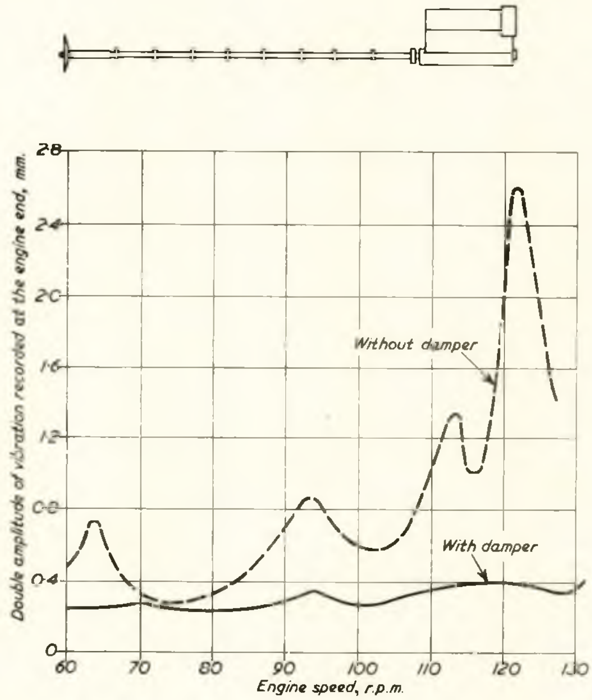


FIG. 36—*Recordings of axial vibrations taken from 688S engine in m.s.* Mocenigo

The calculation as well as the experiments proved that:

1) There existed an optimum value of the damping coefficient for which the amplitude of vibration was of minimum value.
2) For large variations of the damping coefficient around its optimum value, the amplitude of vibration varied very little.
3) It was advisable to realize a damping coefficient slightly lower than the optimum value, in order to obtain, with amplitudes of vibration slightly over the minimum and always of small entity, forces as small as possible acting on the damper and on the supporting frame.

Owing to the difficulty of foreseeing exactly, at the time of design, the value of the damping coefficient, which also depended very much on the oil temperature and viscosity, an adjustment was considered which would vary the damping itself and determine by experiment the best condition.

In this way it was realized that it was possible to obtain satisfactory results, like those of Figs. 33 and 34, with values of oil pressure of approximately $\pm$ 1·5 kg./cm.^{2}, low enough to allow the damper to be kept filled with oil from the engine lubricating oil system.

The two dampers of the propelling sets under consideration, like others of the same type, had been in satisfactory use for some years without ever requiring maintenance.

MR. J. H. MILTON (Member), bearing in mind recent investigations the Engineering Investigation Department of Lloyd's Register had effected into serious troubles with second-reduction gearing, wished to ask Commander Goodwin if he could give any information regarding the effect of these axial vibrations on gearing.

He noted that on page 38, limiting amplitudes of $\pm$10/1,000in. on a straight course and $\pm$0·025in. on turns are mentioned; also the amplitudes of $\pm$48/1,000in. on turns and 17/1,000 to 21/1,000in. on a straight course had been recorded in naval vessels. It would appear that under such conditions, dependent on whether the gear was apex trailing or leading, one helix of the second reduction must be momentarily taking all the torque plus some of the main thrust load. Whilst admitting that this question was rather outside the scope of this very informative paper, perhaps the author would be kind enough to comment on it.

Mr. J. Syme found that the problem of propeller excited longitudinal shafting vibration was almost particular to large high powered naval vessels. The large resonant straight course amplitudes and the very large turn resonant amplitudes encountered resulted in objectionally high vibration levels throughout the machinery spaces and in heavy wear, and also damage to flexible couplings and gearing.

The wartime expedient of imposing restrictions on high speed manœuvring could only be regarded as temporary. Locating the thrust block further aft or increasing the number of blades on the propeller had also to be regarded as temporary solutions; repositioning of the thrust block was hardly practical in existing vessels and it was apt to be neglected when removed from the watchkeeping machinery space; increasing the number of propeller blades resulted in a loss in efficiency.

Although it was agreed that in any new design the probability of axial resonance should be investigated, it must be stressed that the information at present available to the designer was approximate and semi-empirical, therefore calculated resonant frequencies, forces and displacements must only be regarded as approximate estimates. An error in the assumed thrust block stiffness could result in a 10 per cent error in frequency, the slip grip effect of flexible couplings with the additional masses of turbines, condenser gear cases, etc., could result in a further 10 per cent error in frequency; a 5 per cent frequency error could result from an incorrect assessment of entrained water. The total error in frequency could amount to 25 per cent. *Vanguard* three-bladed and five-bladed trials illustrated a 22 per cent change in measured frequencies; *Whitby* trials illustrated an error in excess of 21 per cent between measured and calculated frequencies. Similarly, errors of 50 per cent in thrust variation factor and 25 per cent system damping were not unreasonable and could result in design estimates with 75 per cent inaccuracy in resonant thrust forces and 150 per cent in displacement.

The development of a hydraulic resonance changer, which was a variation of the well established dynamic vibration absorber, offered a very attractive solution to the problem of axial vibration in its simplicity, effectiveness, low cost and ease of installation. One inherent advantage of the hydraulic resonance changer was that it was not susceptible to moderate discrepancies in estimating the original natural frequency of the system; again, it was not particularly sensitive to variation in oil viscosity, a major problem in most oil dampers.

Comparison of experimental results with theoretical curves illustrated that the resonance changer stiffness and inertia characteristics were approximately as theoretically formulated; however, uncertainty surrounded the damping term. A high percentage of the resonance changer damping must take place in the tortuous thrust cylinder interconnecting oil passages within the thrust block; also, depending on the actual dynamic conditions, laminar flow might break down in the interconnecting piping.

It would seem advisable to assess the actual damping within the thrust block and within the interconnecting piping separately, rather than to assume that the actual equivalent hydraulic length was twice the interconnecting piping and then formulate the resonance changer damping term without stressing its empirical nature. It would also seem at present advisable to compute damping to a value below optimum and to adjust to optimum during the sea trial of the first of a class.

Author's Reply

The author wished to thank those who had contributed to the discussion, which enhanced the value and completeness of his paper. Many relevant points had been raised and he would endeavour to answer those that he could.

He was interested in Mr. Rigby's remarks regarding the application of the resonance changer to merchant ships, and the factors leading to comparatively small bottle sizes. The stiffness ratios of up to 1·00 recommended in the paper were what he, the author, considered as being the minimum desirable and he therefore agreed with Mr. Rigby that higher values could justifiably be adopted.

The constancy of the point B in Fig. 10 arose from the simplified equation (21) which ignored both the propeller referred damping (a) and the effective mass at the thrust collar (M_g) in order to obtain the design basis. For such conditions Fig. 11 showed that the point B remained constant at unity value. The curves drawn in Figs. 17—24 were all calculated on a basis of the more complete system, using equations (10), (11) and (12); that is (a) and (M_g) were included, so that a true representation of the effect of the resonance changer could be obtained.

Optimizing the setting of the damping valve on trials should not present much difficulty if one concentrated on shaft speeds in the region of that corresponding to Q on Fig. 4.

Mr. McClimont had shown clearly some of the factors influencing the growing need to give consideration to axial vibration in merchant ships. The author was in sympathy with his suggestion that the value of the critical calculated in merchant ships need be only 25 per cent above the running speed since it was only in ships with three or more shafts that there was a tendency to overspeed on turns. However Mr. Syme had drawn attention to the large errors that might occur in practice, and there were still a number of uncertainties such as the slipping and gripping effect of the flexible couplings.

The author naturally did not agree with Mr. McClimont's opinion of the resonance changer as a mere palliative. Adopting

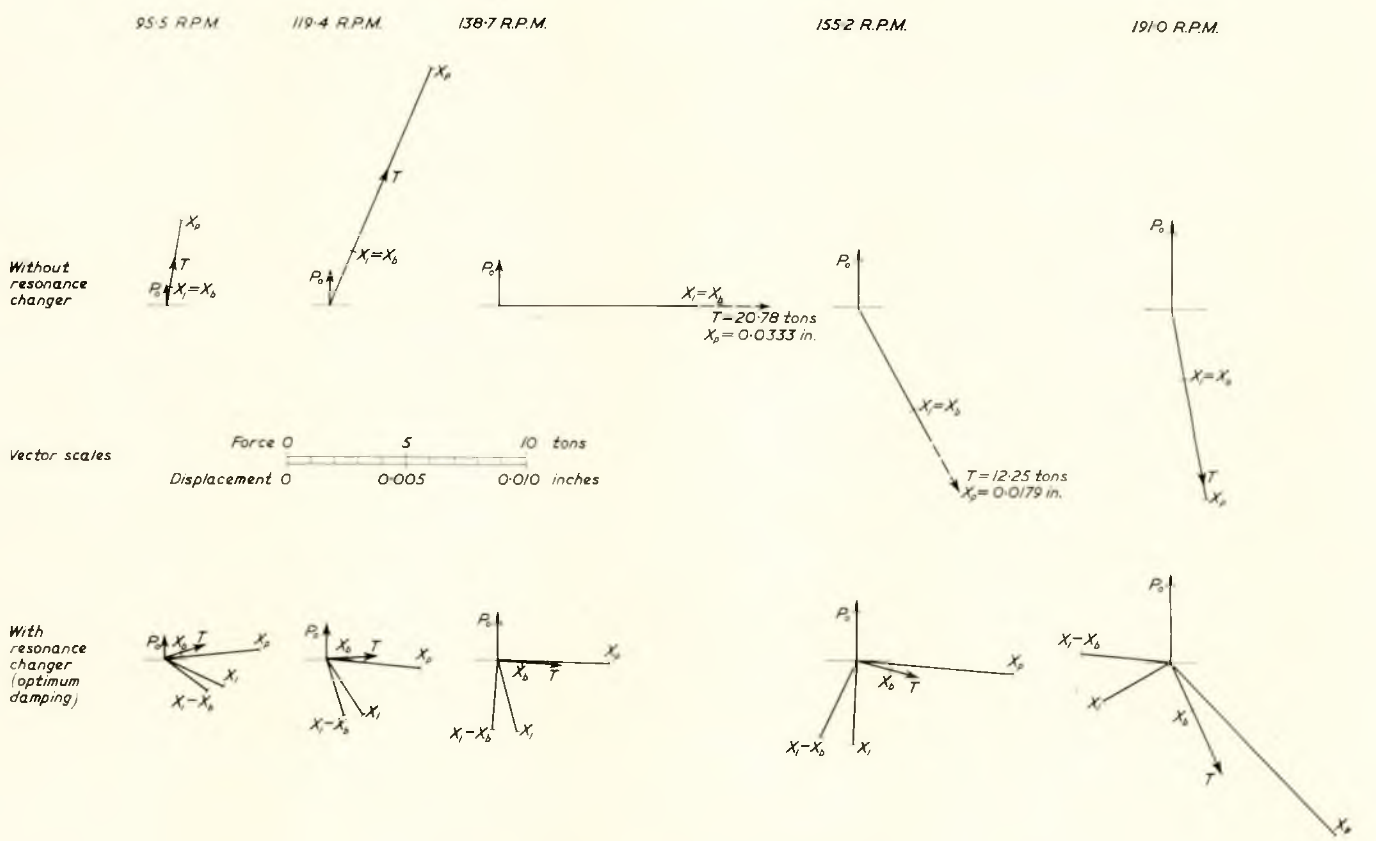


FIG. 37—*Vectorial relationships for the system described in Appendix 4*

his medical metaphor, it was better to provide oneself cheaply with a packet of anti-seasick pills than have to make expensive alterations in one's travelling arrangements whenever there was a hint of an adverse weather report.

The majority of thrust blocks referred to were on riveted seatings and there were not sufficient data on which to differentiate effects with welded seatings.

Trials carried out in a heavy sea had shown a marked increase in the magnitude of the cyclical thrust variation, but no systematic trials had been done to establish the variation in effect on different courses relative to the waves.

Mr. Archer had raised a number of interesting points and the first of these was the question of the proportions of energy dissipated in the referred propeller damping and in the resonance changer.

Fig. 37 had now been prepared showing the vectorial relationships for the system described in Appendix 4, both with and without the resonance changer. He was grateful to Mr. Archer for showing how equations (33) and (39) could be used to obtain R_1 and R_2, which was better than the combination of equations (39) and (41) if equivalence to obtain the precise critical frequency was the requirement.

Without Resonance Changer

$$\frac{K_B}{K_S} = 4\cdot46 \qquad \frac{M_F}{M_S} = 0\cdot604$$

$$Z_c = 0\cdot922 \qquad \frac{X_1}{X_p} = 0\cdot249$$

$$R_1 = 0\cdot39 \qquad R_2 = 1\cdot09$$

$$M = \frac{26\cdot6 + 0\cdot39\ (55)}{386\cdot4} = 0\cdot125 \text{ tons sec}^2\text{/in.}$$

$$M_g = \frac{33\cdot2 + 1\cdot09\ (55)}{386\cdot4} = 0\cdot241 \text{ tons sec}^2\text{/in.}$$

Vector diagrams constructed so that:

$$K_S X_p = (K_B + K_S - M_g\,\omega^2)\, X_1$$
$$P_o = (K_S - M\,\omega^2)\, X_p + ja\,\omega X_p - K_S X_1$$

With Resonance Changer

$$q_1 = 0\cdot57 \qquad \frac{c_1{}^2}{m_1 k_1} = 1\cdot25$$

$$k_1 = \frac{460}{0\cdot57} = 807 \text{ tons/in.}$$

$$m_1 = 0\cdot239 \text{ tons sec}^2\text{/in.}$$
$$c_1 = 15\cdot5 \text{ tons sec/in.}$$

(N.B. This value differed from Mr. Archer's, which did not allow for equivalent length and damping valve.)

Equivalent value of K_B was:

$$K_B{}^1 = K_B \sqrt{\frac{(k_1 - m_1\,\omega^2)^2 + (c_1\,\omega)^2}{(K_B + k_1 - m_1\,\omega^2)^2 + (c_1\,\omega)^2}}$$

and at the original critical frequency of 58·1 rad./sec. had the value:

$$K_B{}^1 = 847 \text{ tons/in.} \simeq 1\cdot5\ K_S$$

Taking $\frac{K_B{}^1}{K_S} = 1\cdot5$ gave:

$$R_1 = 0\cdot48 \qquad R_2 = 0\cdot63$$

$$M = \frac{26\cdot6 + 0\cdot48\ (55)}{386\cdot4} = 0\cdot137 \text{ tons sec.}^2\text{/in.}$$

$$M_g = \frac{33\cdot2 + 0\cdot63\ (55)}{386\cdot4} = 0\cdot176 \text{ tons sec.}^2\text{/in.}$$

Vector diagrams constructed so that:

$$K_B X_b = (k_1 - m_1\,\omega^2)\,(X_1 - X_b) + jc_1\,\omega\,(X_1 - X_b)$$
$$K_S X_p = K_B X_b + (K_S - M_g\,\omega^2)\, X_1$$
$$P_o = (K_S - M\,\omega^2)\, X_p + ja\,\omega\, X_p - K_S X_1$$

It was of interest to note in these vector diagrams that at 191·0 r.p.m. T was 5·37 tons, whereas in Appendix 4 (page 62) it was stated that T_m would be 4·72 tons. It had to be remembered, however, that the latter value was derived from the optimized design which ignored both a and M_g.

The effect of M_g on the response was illustrated in Fig. 26, and from this it could be seen that the actual value of T_m should be about 5·5 tons.

The energy per cycle put into the system by P_o was given by the expression $\pi P_o X_p$ multiplied by the sine of the phase angle between X_p and P_o. The energy dissipated per cycle in the damping referred to the propeller was $\pi a\,\omega X_p{}^2$, and that dissipated per cycle in the resonance changer was $\pi c_1\,\omega$ multiplied by the square of the vectorial difference between X_1 and X_b.

Taking P_o as 1·94 tons at $\omega = 58\cdot1$ rad./sec. the energies involved could be worked out as follows:

R.P.M.	ω, rad./sec.	No resonance changer: $\pi P_o X_p \sin\theta = \pi a\,\omega\, X_p{}^2$ ft.lb. per cycle	With optimum damping resonance changer: $\pi P_o X_p \sin\theta = \pi a\,\omega\, X_p{}^2 + \pi c_1\,\omega\,(X_1 - X_b)^2$ ft.lb. per cycle
95·5	40	0·32	2·17 = 0·38 + 1·79
119·4	50	3·50	3·30 = 0·45 + 2·85
138·7	58·1	37·88	5·28 = 0·73 + 4·55
155·2	65	12·16	9·14 = 1·72 + 7·42
191·0	80	3·12	14·88 = 4·69 + 10·19

The horse power dissipated could be found by multiplying these values by $2\cdot89 \times 10^{-4}\,\omega$.

With optimum damping the energy dissipated in the resonance changer was some six times that in the referred propeller damping in the critical range, but the major effect of the resonance changer was in preventing too much energy being put into the system by P_o.

In Mr. Archer's worked examples, the author believed that unjustifiable liberties had been taken with the Holzer method in that no allowance had been made for the effect of the assumed damping at M_g on the relative motion of X_1 and X_p. This did not arise if all the damping was assumed at M.

The equivalent system with the resonance changer fitted was not correct since the equivalent mass m_1 could not be taken to move with M_F. Its motion depended on the difference in motion between X_1 and X_b. The damping effect of the resonance changer should also have been assessed on the vectorial difference between X_1 and X_b and not on their arithmetical difference.

These were mistakes which the author had made early in the investigation.

However, Mr. Archer's main contention that not all of the damping of the original system was located at the propeller was undoubtedly correct. The author's reason for adopting the simplifying method of referring all the damping to the propeller was that from the results available a constant value of a applied at the propeller seemed to give the best reproductions of the trial results.

In addition to the propeller damping caused by the varying "advance" effect, it seemed to the author that there might be a viscous effect, which could be visualized if the propeller were considered to be moved axially when not rotating. Some of the factors contributing to the total damping might be identified as follows:

Propeller "advance" effect	a should vary as N
Propeller "viscous" effect	a should remain constant
Stern gland friction	a should vary as $1/N$
Viscous friction in bearings	a should remain constant
Flexible coupling when slipping	a should vary as N if torque varied as N^2
Flexible coupling when gripping	a should be zero

If it were a requirement to measure the various contributions accurately, special trials over a wide range of speeds and with arrangements to measure phase angles as well as amplitude at a number of points would be necessary.

Mr. Archer's statement that the second critical did not always occur in the second quadrant was of course correct. The author now thought that it should have been stated that:

The solution for the first critical frequency always lay in the first quadrant.

The solution for the second critical frequency lay in the second quadrant unless:

$$\frac{\frac{M_S}{M_P}+\frac{K_B}{K_S}}{\frac{M_F}{M_S}} < \frac{\pi^2}{4}$$ when it lay in the first quadrant

or $$\frac{\frac{K_B}{K_S}}{\frac{M_P}{M_S}+\frac{M_F}{M_S}} > \pi^2$$ when it lay in the third quadrant.

The author, however, did not agree with Mr. Archer's remarks on the position of the 2-node critical. Reference to the figures in his paper would show that peaks were determined by the amount of damping present in the resonance changer and might be anywhere between $\beta_1(X)$ and $\beta_1(Y)$. In fact, with optimum damping critical frequencies had little significance. It was only if damping in the resonance changer was ill-chosen that critical frequencies began to show up.

The author was very grateful to Mr. Milton for drawing attention to the interaction between the load on the gear wheel helices and the forces at the flexible coupling. At low speeds and displacement amplitudes it seemed likely that both the h.p. and l.p. turbines vibrated with the gearing, and there was no slipping at the flexible coupling. As the speed increased the torques at the flexible couplings tended to increase as N^2, and hence also the frictional axial forces which they were able to impose on the turbine rotors. As the first critical speed was approached, however, the displacement amplitude and hence the accelerations increased faster than N^2. At some point, and it might need the effect of a turn to produce it, the flexible coupling was no longer able to impose the required acceleration on the turbine and slipping would occur. It seemed probable that slipping would first occur in the l.p. flexible coupling, which meant that three different natural frequencies were involved:

(1) When neither flexible coupling was slipping.
(2) When the l.p. coupling was slipping but not the h.p.
(3) When both flexible couplings were slipping.

To add to the confusion the damping would be different in each of these cases. A further complication arose in that in some cases the gearing helices might not be able to pass on the loads required. Assuming double helical gearing with a helix angle of 30 degrees, if the axial load required for accelerating the turbines reached 50 per cent of the torque load, all of the load would be taken on each helix in turn, and any further increase in the maximum axial force required would lead to slipping up the helix until the backlash was absorbed. With the smaller helix angles customary in single helical gears, slipping in the gearing would occur at lower amplitudes and the pinions and turbines would be less likely to join in the vibration, and the flexible coupling less likely to be subjected to damaging slipping.

There was no positive answer to Mr. Milton's question as to whether axial vibration and the tendency to increase the load on the helices had led to trouble. In some cases where unexpected scuffing had been attributed to overtorque during manœuvring, it was possible that it had in fact been due to overloading caused by axial vibration.

Commander Green had requested further information about the "Limit Valve" and accordingly Fig. 38 had been prepared. The function of the "Limit Valve" was to limit the axial movement of the thrust collar, when the thrustmeter was in use, to avoid "nipping" it between ahead and astern pads. The valve was held in the piston by a screwed adjusting guide locked by a screwed pin, and was pressed against its seat by the spring. When the thrustmeter pump was operated, oil pumped into the cylinders caused the retaining ring to move aft until the pre-set travel was reached. The collar of the valve then contacted its guide and any further oil pumped was allowed to escape back to the reservoir tank in the control unit since the

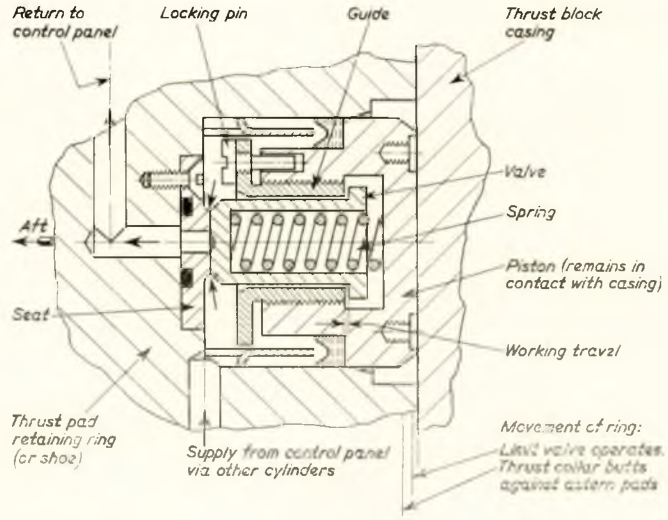


FIG. 38—*Diagram of "Limit valve" for an ahead thrustmeter*

ring had now carried the seat away from the valve. Thus the movement of the retaining ring was limited.

In the opinion of the author the sudden realization just before the war that the problem, subsequently found by Rigby to be caused by axial vibration, existed, arose from the dual facts that the troubles in *Warspite* had just been diagnosed as being caused by operation of the inner propellers in the slip stream of the outers, and that in peace time the only time full power was used tended to be during full power trials when a specially straight course was required in order to obtain good records. The author had himself been involved in bringing out the regulations for easing on turns, together with the provision of the necessary bells and lights, and agreed that things seemed critical at that time.

With regard to the chief engineer referred to, it was probably fortunate that the author only became aware of this report after he had arranged the fitting of a resonance changer in H.M.S. *Eagle*, since the suggestion that a thrustmeter by itself provided a cure for axial vibration was certainly untrue in naval vessels. In the light of trials carried out it could only be assumed that the chief engineer was extremely lucky in having the right amount of air in the system and appropriate damping to make any significant difference to the vibrations.

Mr. Ayling attributed some hull vibration to the axial vibration of the propeller, but the author had reluctantly to admit that he had not been able to detect any improvement in

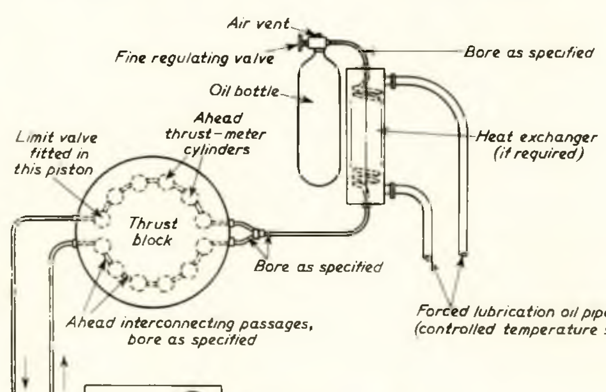


FIG. 39—*Diagrammatic arrangement of resonance changer system*

hull vibration when the axial vibration of the propeller shafting system was substantially reduced by resonance changers.

With regard to the *Vanguard* trials, Mr. Ayling was quite right and it was the author's fault if his wording had given the impression that the vibration with five-bladed propellers was worse than with three-bladed ones. He had only meant to imply that it was insufficiently better.

In answer to Admiral Given the author regretted that he had no recent information concerning United States Naval experience. He had, however, included at Fig. 39 a diagrammatic sketch of the arrangement of the resonance changer system.

Admiral Given questioned the necessity to increase the axial clearance in the thrust block since the vibrations were reduced, but this came about because the action of the resonance changer was to introduce relative movement between the thrust collar and the thrust block itself, i.e. the $X_1 - X_1$, term in the equation. In *Eagle,* excessive clearance might have been allowed to be on the safe side.

Admiral Given had also referred to the movements at the flexible coupling and the author hoped that his remarks on Mr. Milton's contribution covered these points.

The author was indeed working on the question of the use to which a computer could be put in solving the many vibratory problems associated with shafting systems and hoped shortly to be able to assess the benefits practically.

The new pick-up described by Mr. Keig sounded a useful addition to the instrumentation devised by the Admiralty Engineering Laboratory and might help to solve the difficulty of obtaining absolute rather than relative amplitudes of movement.

The author was grateful for Mr. Page's contribution, recalling the very early stages of the investigation when they had together approached, with considerable trepidation, the speeds at which they expected a severe critical vibration to show up and had been annoyed at its absence.

He was glad to learn that he had not bequeathed to Mr. Page any serious maintenance problems on resonance changers.

Dr. Shannon had summed up very concisely the important features of the resonance changer, and the author regretted that he could not throw any light on the interesting point raised as to the possible interaction between torsional and axial vibration, and the effect of a resonance changer thereon, although he had given considerable thought to the matter.

The author welcomed the very interesting contribution made by Ing. Alberto Guglielmotti. The latter was fortunate in having a free end to which to apply his damper and also a standing part of sufficient strength and rigidity to withstand the resulting forces. The device was in some respects analogous to the late Mr. Yates's proposal for the solution of the naval problem, which the author had not developed because of this difficulty with a starting point.

Mr. Syme's remarks on the errors that could occur had been referred to above, but Mr. Syme had also drawn attention to the difficulty in assessing the resonance changer damping correctly due to the tortuous passages within the thrust block. The author felt that this difficulty could be overcome by making the interconnecting passages large enough for the oil velocity in them to be small compared with that in the pipe leading to the oil bottle. He agreed that the diameter of the pipe leads should be chosen to give less than the optimum damping and it was for this reason that a minimum diameter was given in Fig. 29.

Summing up, the author felt that as a result of the discussion he would undertake any future calculation of critical frequency with even less confidence of being within 10 per cent of the true answer.

Mr. R. Cook, M.Sc. (Chairman of Council) and Mr. J. Calderwood, M.Sc. (Honorary Treasurer and Vice-President)

Mrs. W. L. S. Wallace, Sir William Wallace, C.B.E., LL.D. (President), Mrs. R. Cook, Mr. R. Cook, and Mr. W. L. S. Wallace

Annual Conversazione, 1959

INSTITUTE ACTIVITIES

Minutes of Proceedings of the Ordinary Meeting Held at The Memorial Building on Tuesday, 10th November 1959

An Ordinary Meeting was held by the Institute on Tuesday, 10th November 1959 at 5.30 p.m., when a paper entitled "The Design of a Resonance Changer to Overcome Axial Vibration of Propeller Shafting" by Commander A. J. H. Goodwin, O.B.E., R.N.(ret.) (Member), was presented and discussed. Mr. W. R. Harvey (Vice-Chairman of Council) was in the Chair and fifty-eight members and visitors were present. Nine speakers took part in the discussion that followed.

A vote of thanks to the author, proposed by the Chairman, was accorded by acclamation. The meeting ended at 7.30 p.m.

Minutes of Proceedings of the Meeting of the Joint Panel on Nuclear Marine Propulsion Held at The Memorial Building on Tuesday, 26th January 1960

A meeting of the Joint Panel on Nuclear Marine Propulsion was held at The Memorial Building on Tuesday, 26th January 1960 at 5.30 p.m., when a paper entitled "The Control and Instrumentation of a Marine Reactor" by R. Anscomb, O.B.E., and F. Hutber, was presented and discussed. Vice-Admiral Sir Frank Mason, K.C.B., (ret.) (Chairman of the Panel) was in the Chair, and 125 members of the constituent institutions and visitors were present. Twelve speakers took part in the discussion that followed.

A vote of thanks to the authors, proposed by the Chairman, was accorded by acclamation. The meeting ended at 7.55 p.m.

Annual Conversaziones

Two Conversaziones were held in 1959, on 4th and 18th December, at Grosvenor House, Park Lane, London, W.1. The President, Sir William Wallace, C.B.E., LL.D., and the Chairman of Council, Mr. R. Cook, M.Sc., and Mrs. Cook, received the 1,653 members and guests who attended these functions.

After dinner Sydney Jerome and his Ballroom Orchestra, and Miguelita and her Orchestre Tropicale, played for dancing, with intervals for the presentation of a cabaret and floor show. The following artists appeared: Leslie Roberts' Television Silhouettes, Johnny Laycock, Caruna and Dodo, The Champagne Young Ladies, The Schaller Brothers, Robb Murray, the Dazzle Young Ladies and the Great Alexis Troupe. On 18th December carols were sung after dinner and a Christmas cake was cut by the President.

Section Meetings

Kingston upon Hull

Ninth Annual Dinner

The Ninth Annual Dinner of the Kingston upon Hull Section was held at the Guildhall, Kingston upon Hull, on Friday, 6th November 1959. Once again a record number of 180 members and friends attended. The principal speaker was Sir William Wallace, C.B.E., LL.D., President of the Institute, who was supported by Mr. H. F. Hesketh (Chairman of the Section), Mr. F. C. M. Heath (Vice-President), Messrs. R. Rawlings and F. T. Green (Members of Committee), and Mr. S. W. Hobson, Chief Education Officer of Kingston upon Hull. The Lord Mayor of Kingston upon Hull was unfortunately absent due to serious illness and the Sheriff of Kingston upon Hull responded on his behalf.

As in the past, the occasion was extremely successful and was obviously enjoyed by the whole company.

Scottish

Joint Meetings

A joint meeting with the Greenock Association of Engineers and Shipbuilders was held at the Lorne Restaurant, Greenock, on Tuesday, 19th January 1960 at 7.30 p.m.

The President of the Association presided and after welcoming Mr. G. J. Thomas and Mr. A. W. Clark, Chairman and Secretary of the Scottish Section, intimated with regret that, due to illness, Mr. W. R. Harvey, the speaker for the evening, would be unable to present his paper, "The Care and Maintenance of High Pressure Watertube Boilers and Ancillary Equipment".

Mr. A. W. Clark therefore read the paper, and dealt with the discussion that followed.

Mr. J. F. Robb, in proposing a vote of thanks to Mr. Harvey, mentioned that all members and visitors present appreciated receiving copies of this interesting and instructive paper.

There was an attendance of fifty-five and the meeting terminated at 9.45 p.m.

A joint meeting with the Institution of Engineers and Shipbuilders in Scotland was held at 39 Elmbank Crescent, Glasgow, C.2, on Tuesday, 26th January 1960 at 6.30 p.m., with Mr. John Brown, B.Sc., President of the Institution, presiding.

There was an attendance of forty.

The Chairman introduced Mr. K. Maddocks, B.Sc.(Tech.), A.M.C.T., M.I.Mar.E., who then presented his paper entitled "Some Aspects of Marine Reactor Safety", which had been arranged under the auspices of the Joint Panel on Nuclear Marine Propulsion.

An interesting discussion followed, which was ably dealt with by the author.

A vote of thanks to Mr. Maddocks was aptly proposed by Mr. G. J. Thomas (Chairman of the Scottish Section) and carried with enthusiasm.

The meeting ended at 8.25 p.m.

West Midlands

At a meeting held at the Engineering Centre, Birmingham, on Thursday, 28th January 1960 at 7.0 p.m., Mr. J. A. Dorrat, A.H-W.C., presented a lecture on "Welding in Marine Engineering" to an audience of twenty-four members and guests. The Chair was taken by Mr. J. R. Cotterill, J.P. (Chairman of the Section).

Mr. Dorrat stressed the necessity for the drawing office to utilize the most economical welding technique when developing the design of a new structure. This point was brought out in the numerous slides which showed the fabrication of marine engine bedplates and A-frames. As a result of careful design, much of the welding was carried out in the downhand position and also the manipulation of the structures by means of overhead cranes was reduced to a minimum. The advantage of automatic welding in certain cases was discussed, and the lecture concluded with a description of the inspection and testing of welds by ultrasonic and magnetic crack detection.

A lively discussion followed in which the speaker answered extensively the nine questions put to him.

The Chairman thanked Mr. Dorrat for a very interesting lecture before closing the meeting at 9.0 p.m.

Student Section

A meeting of the Student Section was held at The Memorial Building, 76 Mark Lane, London, E.C.3, on Monday, 1st February 1960 at 6.30 p.m., when a lecture entitled "The Instrumentation of Marine Machinery" was given by Mr. F. P. Rout. Mr. P. J. Humphreys (Member) was in the Chair and there were twenty-four members and visitors present.

The lecture was followed by a lively discussion period and the vote of thanks to the lecturer proposed by the Chairman was carried by acclamation.

The meeting ended at 8.15 p.m.

Visit to the City of Hull

A party of students visited the Ellerman Lines s.s. *City of Hull* on the morning of Saturday, 13th February 1960. Navigational equipment in the ship was explained by the radio officer and the party was conducted round the engine room by the chief engineer, Mr. Smail.

The visit ended with refreshments in the lounge.

Election of Members

Elected on 8th February 1960

MEMBERS

Harold Charles Blamey
Allan Ray Caird
Oliver Cromwell
Leonard Norman Dine
William Douglas Forsyth
John Plewright Graham
Cecil Hunton
Gerard Adriaan Kemper
David McGrouther
Herbert Davison Metcalf
Hercules Sidney Quinnell
John Leslie Raddings
Guilhermino Ferreira Vila Real
Robert Pierre Tetreau
Frank Walker, Cdr., V.R.D., R.N.R.
Louis Offenberg Welch
Stewart Young

ASSOCIATE MEMBERS

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Alan Blythe
Alan Crow
Thomas Neil Foster
Walter Harry Rowland Girling
Douglas Haigh
Vincent Hanson
Edgar Alan Irving
Allan Hope Jolly
Kaushal Kishor
Keith Kling
Mark Anthony Lane
Russell Stephen Larkin
Eric James Lashmar
Henry Medforth
William Stephen Mockett
George Alfred Murray
John Michael Ramsden
Tarun Kanti Sen Gupta
Peter Shutt
Edward Anthony Smith
Henry Arthur Sparling, B.Sc.(Eng.) London
Gordon Robert Stephen
Owen Rankine Throssell
Leslie Williams
Thomas William Williamson

ASSOCIATES

Keith Bowen
Wai-Tim Law
Roderick Macmillan Murray
Richard Charles Soukup

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Michael Gordon Derham
Martin Herman Greeff
Robert George Herbert
Martin David Keith
Charles Frederick McNicoll
Alistair Peter Philp
Derek Sloan
Graham Robert Strachan, M.A.

STUDENTS

Norman Charles Graham Ballard
Andrew Douglas Bland
Keith Murray Bullimore
Robert George Finch
John Alexander Logan
Paul Richard Lowe
Anthony Ronald Lynton
Elfed Owen
Philip Godfrey Page
John Harwood Parry
William Scott Rogers
Derek Walter Russell
Trevor Wesley Watson
Michael Peter Wong Foo

PROBATIONER STUDENTS

Alan Vernon Albutt
Brian Desmond Armitage
Ian David Badenoch
Keith Dexter Ronald Baker
Charles Michael Beech
Michael Brewerton
Victor Carrell
C. M. Crawshaw
Robert Malcolm Denny
Simon Ashlin Ebsworth
Peter Greenwood
John Trevor Holmes
Michael John Holmes
Andrew Edward Hudson
Melvyn Francis Jones
David Barry Melhuish
Roger John Metcalf
John Frederick Morris
David Mullineux
David John Offord
John Parker
Godfrey David Rawes
Geoffrey Salmon
Nicholas Alan Valentine Stenning
Anthony Robert Turner
Arthur Frank York

TRANSFER FROM ASSOCIATE MEMBER TO MEMBER

James Archibald Campbell
James Edmund Gander
Malcolm Francis Heslop
Peter James Howard, B.Sc.(Eng.) London
Eric Thomas

TRANSFER FROM ASSOCIATE TO MEMBER

John Noel Abbott

TRANSFER FROM ASSOCIATE TO ASSOCIATE MEMBER

Alan Robert Berrett
David McCallum
Leslie Thomas Reed
William Whyte

TRANSFER FROM GRADUATE TO ASSOCIATE MEMBER

Sten Bengtson
Morris Carrick
Hugh Cain
Rajeswari Prasadarao Chitra
Zulfiqar Ali Choudry
Michael David Constable
Victor William Howard
Alexander Jenkins
Hari Kishen Kaul
Pothamsetti Prabuddha Kesava
Norman James Townsend
Christopher John Warren
Reginald Joachim Fernandes

TRANSFER FROM STUDENT TO ASSOCIATE MEMBER

Harold George Makin

TRANSFER FROM STUDENT TO GRADUATE

Norman Finlay
Colin Scorer

TRANSFER FROM PROBATIONER STUDENT TO GRADUATE

David Partrick

TRANSFER FROM PROBATIONER STUDENT TO STUDENT

Alan David Gambles
Steven George Hattrick
John Lilley
Sheffield George Preston
James David Wort
Daniel Greenhalgh Briggs

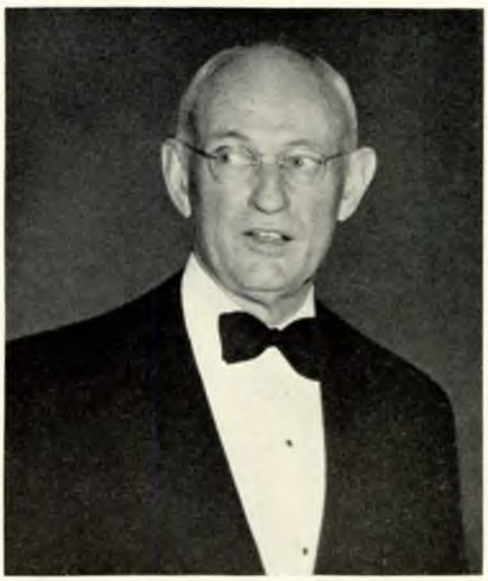
OBITUARY

Edward Lull Cochrane

Honorary Member

Vice Admiral E. L. Cochrane, K.B.E., D.S.M., D.Eng., LL.D., U.S.N.(ret.), chief of the Bureau of Ships during World War II and chief adviser to the president of Massachusetts Institute of Technology, died on 14th November 1959 while returning to his home in Cambridge, Massachusetts, following the 67th Annual Meeting of the Society of Naval Architects and Marine Engineers in New York.

Admiral Cochrane was born at Mare Island, California, on 18th March 1892, and following two years of study at the University of Pennsylvania entered the United States Naval Academy, graduating with distinction in the class of 1914. He pursued graduate studies at the Naval Academy and at Massachusetts Institute of Technology, receiving his master of science degree in naval construction in 1920.



During the period 1920/24 he was assigned to the Philadelphia Navy Yard, where he handled design, construction and repair posts. He was then transferred to the Bureau of Construction and Repair, where for the next five years he specialized in submarine and general ship design work. In 1929 he served as technical adviser to the United States Delegation at the International Conference of Safety on Life at Sea in London and upon his return he took charge of the design and construction of submarines at the Portsmouth Navy Yard. Then, in 1933, he spent two years at sea as constructor on the Scouting Force Staff, when he returned to the Bureau of Construction and Repair in charge of contract design. In 1940 he was appointed Naval Attaché at the American Embassy in London and returned from there to the Bureau of Ships.

In 1942 he was appointed Chief of the Bureau of Ships, with the rank of Rear Admiral, in which capacity he was in charge of the nation's large wartime naval construction programme, directing the construction efforts of more than 1,000,000 workers, who in five years raised the nation's naval strength from 400 to some 100,000 combatant vessels, including many unique and novel types. In 1945 he was advanced to the rank of Vice Admiral and in the following year was appointed Chief of Naval Material. Admiral Cochrane retired from active duty in 1947, and became head of the Department of Naval Architecture and Marine Engineering at the Massachusetts Institute of Technology.

He was recalled to Washington in 1950 to serve for three years as Maritime Administrator and Chairman of the Federal Maritime Board. It was during this period that the new Mariner class of cargo ships was developed and constructed. He also set up the National Shipping Authority to direct merchant ship operations. Then in 1952 he returned to the Massachusetts Institute of Technology as dean of engineering, and two years later became vice-president for industrial and governmental relations. Since 1957 he had held that title in an emeritus capacity and was acting as a special adviser to the president of the Institute at the time of his death.

Admiral Cochrane's distinguished professional career brought him many honours. He was awarded the Mexican Campaign Medal (1914), the Victory Medal (1919), the American Defense Medal (1942), the Asiatic Pacific Campaign Medal (1944), and the American Theater and Victory Medals (1945). In 1945 the Society of Naval Architects and Marine Engineers awarded him the David W. Taylor Medal for notable achievement in naval architecture. Also in that year he was made an Honorary Knight Commander, Military Division, of The Order of the British Empire. In 1946 the Navy bestowed upon him the Distinguished Service Medal.

Admiral Cochrane was elected an Honorary Member of the Institute of Marine Engineers in 1951 in recognition of his services to Britain during the 1939/45 war.

Walter Joseph Frier (Member 9304) died, aged sixty-one, on 10th December 1959.

He was apprenticed to John Readhead and Sons Ltd. from 1913/19 and then became a seagoing engineer, obtaining a First Class Board of Trade Certificate; from 1925/27 he sailed as chief engineer. He then joined the staff of the Mobil Oil Co. Ltd. (at that time the Vacuum Oil Co. Ltd.) as a marine representative, and in 1941 was appointed marine branch sales manager in Liverpool. In January 1947 he became branch sales manager at Newcastle upon Tyne, and in May 1952 manager of the Northern automotive division. This position he held until May 1959 when he went on leave of absence pending retirement.

Mr. Frier was elected to Membership of the Institute in 1941.

Arthur Hoare (Member 10035) died, aged seventy-one, while walking in Portsmouth on 2nd December 1959. He had been engineering manager for J. S. White and Co. Ltd. of Cowes in the Isle of Wight from 1926 until he retired in 1953.

He served an apprenticeship with Cammell Laird and Co. Ltd. in Liverpool and then joined the design staff of the Parsons Marine Steam Company at Wallsend on Tyne. Subsequently he spent four years as leading turbine draughtsman with the Palmers Shipbuilding Co. Ltd., at Jarrow; a year as chief draughtsman and a further year as works manager with Ramage and Ferguson Ltd. of Leith; and five years as engine works manager with the Caledon Co. Ltd. of Dundee. He then went to Cowes; for his services to marine engineering there, particularly during the second world war, he was awarded the O.B.E. in the 1951 New Year Honours.

Mr. Hoare was a Member of the Institutions of Mechanical Engineers and Naval Architects and of the Institute of Patentees. Since attaining in 1913 the award of a Whitworth Exhibition he had been a member of the Whitworth Society, of which he was president at the time of his death. He was elected to Membership of the Institute of Marine Engineers in 1944.

Mr. Hoare took a keen interest in local affairs and became president of the Newport Ratepayers' Association. He was elected to the town council for the Carisbrooke Ward in 1945 and served until 1957; he was chairman of the housing committee for five years and represented the council on the Cowes Harbour Commissioners and the Cowes Port Health Authority, the Carisbrooke Parish Charities, and I.W. Water Board, and the Council for the Preservation of Rural England.

Samuel Haslett Morrow (Member 13314) died on 2nd November 1959 at the age of sixty, only three weeks after his retirement from the Directorate Royal Engineer Equipment of the Ministry of Supply.

He was apprenticed to Harland and Wolff Ltd. in Belfast from 1915/22 and then served several companies as a seagoing engineer—the Peninsular and Oriental Steam Navigation Company, the Bank Lines Ltd., the Nelson Line Ltd. and the Royal Mail Lines Ltd.—until 1934, having obtained a First Class Board of Trade Steam Certificate with Motor Endorsement in the meantime. He was then appointed engineer in charge of Diesel engine tests with the Brush Electrical Engineering Co. Ltd. but after a year went to the Air Ministry as an engineering inspector in the aircraft inspection division; during the second world war he was British Air Commission Inspection representative in the U.S.A. A few months after the war ended he was made responsible by the Ministry of Supply for technical and inspection matters relating to the production of aluminium temporary houses at Blackburn Aircraft Ltd., Dumbarton; and in 1947 he was appointed engineering technical adviser in the internal combustion engine branch of the Directorate of Royal Engineer Equipment, Ministry of Supply.

Mr. Morrow was elected a Member of the Institute in 1951.

William Tilby (Member 7443) was apprenticed to Swan, Hunter and Wigham Richardson Ltd. at Neptune Yard, Newcastle on Tyne, from 1917/23. For the next year he served as junior guarantee engineer in the m.v. *Arnus,* which was owned and manned by Spaniards. He then went as fourth engineer to the m.v. *Lenfield,* owned by Messrs. E. J. Sutton and Company of Newcastle on Tyne, for some months. From 1925/32 he was employed by the Prince Line, chiefly as senior second engineer of the m.v. *Japanese Prince,* and he obtained a First Class Board of Trade Certificate. He joined the Institute as a Member in 1934.

From 1934/40 Mr. Tilby was an engineer in the m.v. *Oil Pioneer;* this ship was sunk off Narvik in June 1940 and all survivers were taken to Germany as prisoners. On his return from Germany in 1945 he was employed by John I. Jacobs and Co. Ltd. as chief engineer and served in many of their vessels until his death on 26th October 1958 at the age of fifty-six, whilst at home on leave.

Henry Neville Walker (Member 13729) was born on 11th November 1890 and served an apprenticeship with Richardsons Westgarth and Co. Ltd. at Hartlepool. He commenced his seagoing experience at fourth engineer of s.s. *Inkula* in November 1910 and made his way steadily through the grades until in November 1912 he became second engineer of s.s. *West Point.*

At the early age of twenty-three years Mr. Walker had obtained his First Class Board of Trade Certificate and he continued serving as second engineer until the outbreak of the first world war, when he joined the Royal Navy for the duration. He served from September 1914 to January 1919 and was decorated with the Distinguished Service Medal for his services in the raid on Zeebrugge in April 1918, during which action he was wounded.

From 1919/23 Mr. Walker was chief engineer of s.s. *Eskbridge.* He entered the service of J. and C. Harrison Ltd. in 1928, when these owners took over the National Steamship Co. Ltd. At that time, Mr. Walker was chief engineer of the *Fotinia.* In September 1930 he was appointed chief engineer of the new s.s. *Harpenden* and later, in May 1932, transferred to the *Harpalion.*

In March 1935 Mr. Walker was promoted superintendent engineer of J. and C. Harrison Ltd. and during the second world war his work was strenuous and exacting. He was promoted chief superintendent engineer for the Company in February 1952 and was responsible for the superintending of the building of the Company's new motor vessels.

His health deteriorated throughout the last two years and at the end of December 1959 he retired to become consultant engineer. It is to be deeply regretted that he passed away quite suddenly only a week later, on 7th January 1960, at the age of sixty-nine years.

Mr. Walker became a Member of the Institute in April 1952.

Albert Edward Wooley (Member 6306) died suddenly in hospital on 27th December 1959. He was born in Portsmouth and was apprenticed at the Portsmouth Royal Dockyard. In 1920 he was commissioned, with the rank of engineer lieutenant, in the Royal Indian Marine. In 1929 he came to Glasgow on a year's leave from India to take a Board of Trade course which qualified him as an engineer and ship surveyor. He served as fourth, then third, engineer and ship surveyor in Bombay from 1931/39, and on the outbreak of war (his rank then being Commander(E)) he was appointed to Calcutta as third engineer and ship surveyor. In 1943 he was transferred to Madras as principal engineer and ship surveyor, where he also represented Lloyd's Register of Shipping. He went to Calcutta in 1945 as principal engineer and ship surveyor to the Government of India, remaining there until an illness necessitated his return to England in March 1946 and as a result of which he retired from the Royal Indian Navy in July 1947.

Commander Wooley was elected a Member of the Institute in 1929. He was also a Member of the Institution of Naval Architects and a Fellow of the Royal Society of Arts.