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THE CONTROL OF PROPELLER-INDUCED VIBRATIONS IN SHIP TRANSMISSION SHAFTS

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Abstract

The fluctuating force generated by a ship's propeller leads to axial and radial vibrations which are subsequently transmitted to the hull. The simulation study presented here demonstrates how a tuneable vibration absorber utilizing a magnetic thrust bearing and inertial mass employed in parallel with a hydrodynamic thrust bearing can reduce propeller drive shaft axial vibrations. The sensitivity to controller errors and the influence of vibration absorber mass are examined. It is shown that a practical device using well proven magnetic actuation techniques could be developed.

The variable stiffness and damping characteristics of hydrodynamic thrust bearings mean that a fixed parameter closed loop controller will not perform well at all operating conditions. It is argued that the use of self tuning or model-reference type of closed-loop adaptive control is impractical. An open-loop speed-dependent control strategy is developed. This is inherently stable but may increase vibration amplitudes at frequencies other than the propeller passage frequency.

1. Introduction

One of the principal causes of vibration in a ship is the fluctuating axial force generated by the propeller or propulsor. Generally, this force comprises a steady load superimposed with a fluctuating component at the blade passage frequency [1]. This periodic force is a direct result of the non uniform wake of the hull and the clearance between the propulsor and the hull. Although the harmonic content of the thrust load may be reduced by good design it cannot be eliminated. As a result, alternative methods of vibration reduction have been investigated and some have been applied to sea going vessels.

Acoustic resonator devices and 'vibration absorber' mass/spring/damper systems are presently employed on sea going vessels. These passive devices which are tuned to a single frequency attenuate vibration at a specific operating condition [2] but are of little benefit at other operating conditions. This limitation in passive devices has led to work in the field of magnetic bearings [3, 4]. These devices, which are now available in both radial support and axial thrust form, may be controlled actively, responding rapidly to transducer and microprocessor signals.

Magnetic bearings have been applied both theoretically and experimentally to flexible rotors. Whilst most researchers have used collocation to effectively replace the passive squeeze film damper device with a system which can be tuned [4] others have applied more sophisticated techniques which aim to reduce shaft vibration throughout its length. It has been shown that the changing stiffness and damping coefficients of hydrodynamic support bearings can lead to instability in a closed loop multiple input system. As a result Burrows and Sahinkaya have used an open loop adaptive control strategy to minimize radial vibration in a flexible rotor [5].

The application of magnetic thrust bearings is in its infancy. Lewis and Allaire [6] have shown theoretically that a magnetic thrust bearing installed in parallel with the existing hydrodynamic thrust bearing on a ship can reduce vibration over certain frequency ranges. In this work a magnetic bearing acting directly between the thrust bearing housing and hull was employed in order to generate a negative stiffness which when combined with the thrust bearing stiffness reduced the transmission of force from the propeller to the hull. The consequences of changing stiffness and damping on system stability and vibration attenuation are noted as are the restrictions imposed by limited shaft motion and limited magnetic bearing negative stiffness.

This study considers the forces transmitted to the hull of a vessel in a typical uncontrolled thrust bearing/shaft/propulsor system. This simplified linearised axial vibration model is used for comparison purposes. Following this, a tuned vibration absorber utilising a magnetic thrust bearing is included in the theoretical model. The effects of absorber mass and thrust bearing damping on vibration attenuation and absorber mass amplitude are established at a number of operating conditions and a means of implementing the system is proposed.

2. Shaft Axial Vibration Model

A schematic diagram of the thrust bearing, shaft and propulsor system is shown in figure 1. A tilting pad

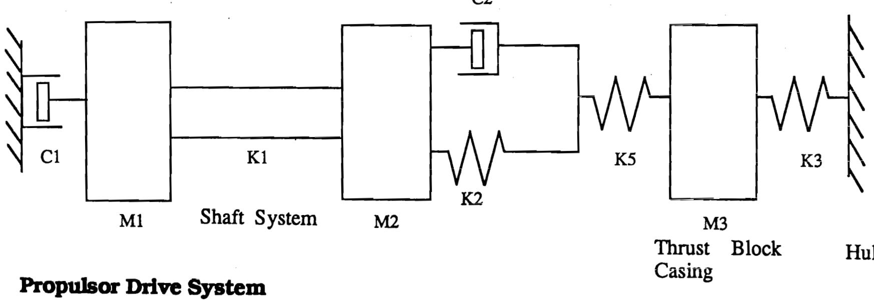

Figure 1. Propulsor Drive System

double acting thrust bearing carries both the primary and fluctuating loads and is mounted via a flexible element to the hull. The thrust bearing is modelled as a fixed mass with a linearised stiffness and damping coefficient. The stiffness was split into two components in series. A constant stiffness accounted for the thrust block structure and a variable stiffness accounted for the fluid element. Since the evaluation of the thrust bearing parameters was not the primary objective of this paper, and since both stiffness and damping are known to be complicated functions of speed, load and operating temperature the numerical values used in this work were obtained experimentally.

The hull of the vessel is assumed to be fixed since its mass is significantly larger than that of the shaft/propulsor system. An extensive finite element analysis of the shaft, main hull and local structure members as carried out by Fujii and Tanida [7] was not thought to be appropriate in this simplified model. The shaft itself is modelled as two fixed masses. At the thrust bearing end an element of the shaft mass is combined with the coupling whilst at the other end the remaining shaft mass is combined with that of the propulsor. Data concerning the shaft mass, stiffness and damping were typical for the given application.

A set of simultaneous differential equations describing the motion of each of the mass elements was developed and analysed in the frequency domain using fixed stiffness and damping parameters.

Three operating speeds and a range of hydrostatic loads were considered in this work (Table 1). The results of figures 2a and 2b demonstrate that the shaft mass/spring/damper combination operating at a rotational speed of 24 rev/min is below its lowest natural frequency throughout the range of operating conditions and the transmitted force is slightly higher than the fluctuating applied force between 100N and 150N. The lowest resonant frequency of the standard system (based on the highest speed and highest load operating condition) was 17Hz.

Speed (rev/min)	Steady Load (kN)	Film Stiffness K2 (MN/m)	Film Damping C2 (MNs/m)
10	13	15.3	10.6
18	42	64.0	21.6
24	75	130.8	34.0
10	1413	16200.0	5670.0
18	1442	12500.0	2260.0
24	1475	11300.0	1460.0

Table 1. Thrust bearing film stiffness and damping

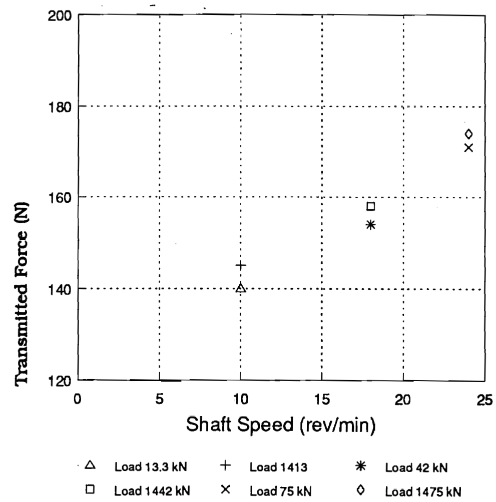

Figure 2a. Force Transmitted to Hull

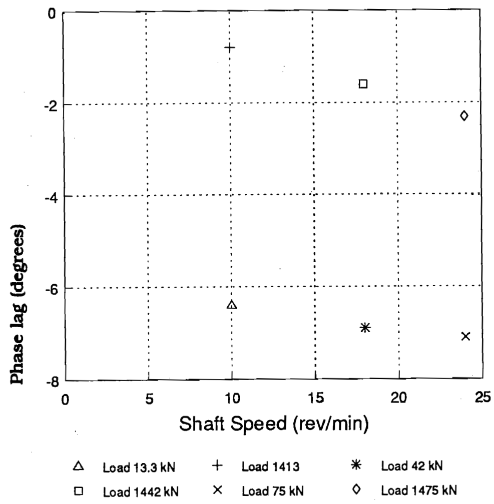

Figure 2b. Phase of Transmitted Force

3 The Active Vibration Absorber

If the vibration generated by the propeller and transmitted to the hull were always at one frequency this effect could be countered by attaching a vibration absorber [1] to the thrust block casing represented by mass M_3 in Fig.1. In variable speed machines the absorber can still be effective if damping is introduced in parallel with the absorber spring [2]. An alternative to this classical approach is to use an active vibration absorber instead of a passive mass, spring, damper system. With this approach the absorber can be tuned to be effective in a range of frequencies.

One possibility is to couple the thrust block casing mass M_3 to an auxiliary mass M_4 as shown in Fig.3. by using an electro-magnetic actuator. A magnetic actuator, or bearing in open-loop is essentially unstable because the force of attraction decreases with increasing relative displacement, but with position feedback the magnetic actuator is equivalent to a spring. By varying the feedback gain this is equivalent to having a spring of variable stiffness. This stiffness can be controlled to minimise the

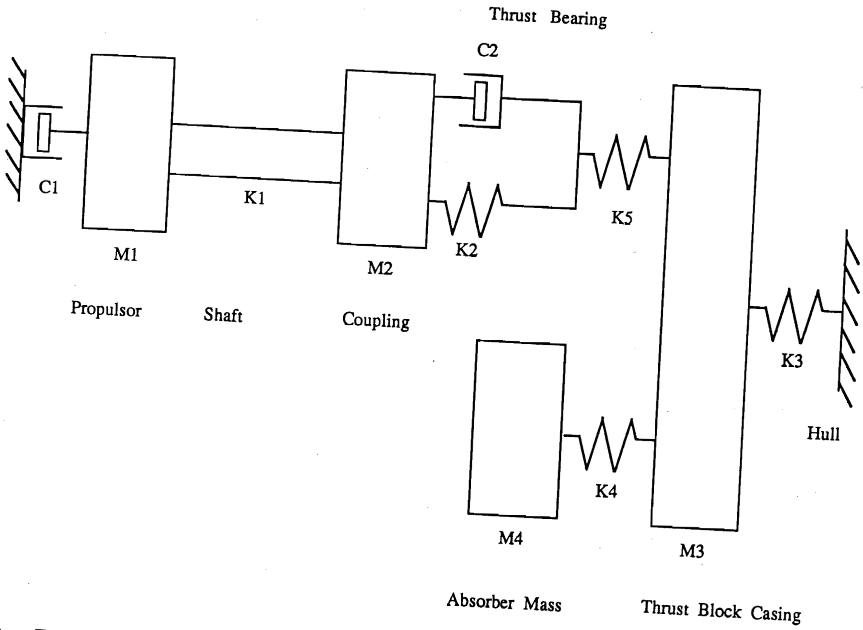

Figure 3. Propulsor Drive with Mass/Spring Vibration Absorber

vibration at a given (variable) operating speed of the propellor. If the actuator is also controlled by a signal proportional to the relative velocity of masses M_3 and M_4 the effect is <u>equivalent</u> to introducing a damping force between the masses. Thus by using velocity feedback around the magnetic actuator and varying the gain this is equivalent to introducing a variable dashpot between the masses.

If an active vibration absorber is used the gain values need to vary automatically as a function of motor speed in order to minimise the vibrations transmitted to the hull. If system parameters do not change with time and are a simple function of rotor speed the appropriate gain settings could be selected by a scheduling process. In practice it is necessary to develop a strategy to select the gain on-line to account for short-term and long-term variations in parameter values. This is discussed later.

Before parameter values were specified the magnitude of the control force required from a magnetic actuator was unknown. The specification of a propulsor variable force component of between 100N and 150N was such that this can easily be met using current technology. This figure was based on measurements taken from a typical large vessel.

<u>3.1 Undamped Active Vibration Absorber</u> In this study the methods for achieving on-line control of the magnetic actuator are not developed. The work is confined to determining the reduction in transmitted forces that can be achieved over a range of operating conditions. The emphasis is on examining the feasibility of achieving a reduction in transmitted force with an auxiliary mass value which would be acceptable in a sea going vessel. Thus the effect of different mass values is considered. In addition it is inevitable that there will be tuning errors in the magnetic actuator control system. Both of these variables are examined in figure 4. It is clear that the system is very sensitive to errors, a 1% error is sufficient to raise the transmitted force to its original level. However, the system is less sensitive if a large absorber mass is used. A 10 tonne reaction mass would reduce the transmitted force by 70% even if the tuning error was a great as ±0.5%. Although a mass of this size would be difficult to support on low friction bearings it should be appreciated that the volume of steel would be little more than $1m^3$.

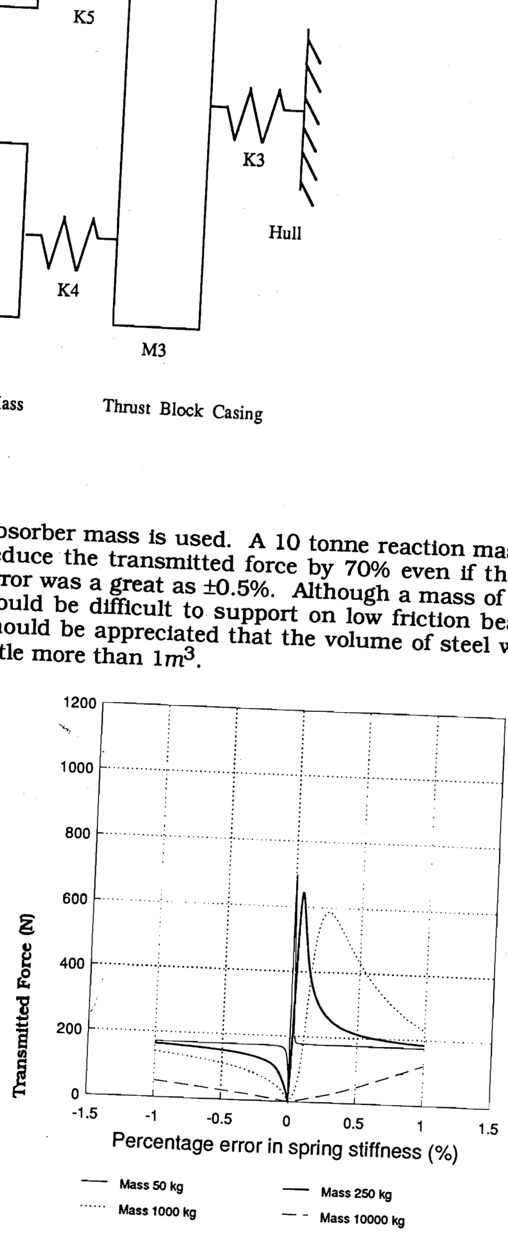

Figure 4. Influence of vibration absorber spring stiffness error on force transmitted to hull

The amplitude of the vibration absorber mass is an important aspect in the design of a magnetic device. The maximum allowable amplitude of vibration is typically less than $1mm$. The results of figure 5 show that a vibration absorber mass of over 70 Kg is necessary to reduce the amplitude to acceptable levels.

3.2 Damped Absorber (Magnetically tuned mass/spring/damper)

In order to reduce the system sensitivity to speed changes and errors in tuning a magnetic damper in parallel with the magnetic spring was added (figure 6). The damping was introduced into the equations of motion and the computer model modified accordingly.

The introduction of damping into the magnetic vibration absorber has two effects. First, it is no longer possible to cancel completely the force transmitted to the hull, even with an 'ideal' system [2]. The degree of force attenuation is dependent on the level of damping and the accuracy of tuning. Secondly, the vibration absorber sensitivity to speed changes is reduced. Although this is of benefit the maximum attenuation that may be achieved with a damped device is much reduced (figure 7).

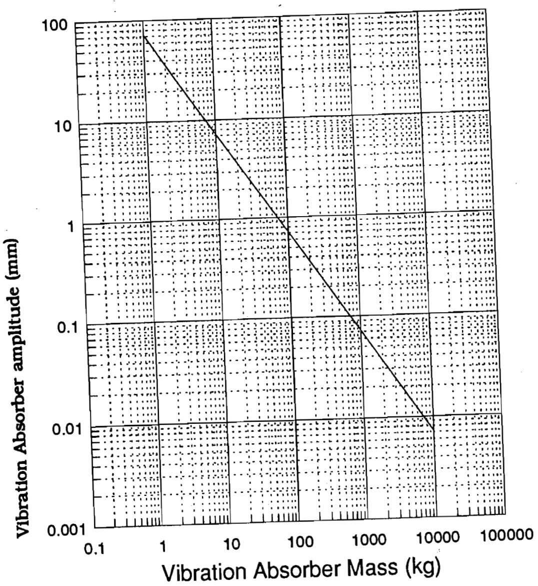

Figure 5. Vibration Absorber Mass and Amplitude

The problem of tuning errors and speed changes might be overcome by switching off the magnetic device if the exciting frequency fell below the tuned frequency. This would avoid the problem of force amplification but would be difficult to implement in a real system since the switch from force cancellation to force amplification is very sudden. In addition, the degree of force cancellation would vary from excellent to poor depending on if the system was operating above or below its tuned frequency.

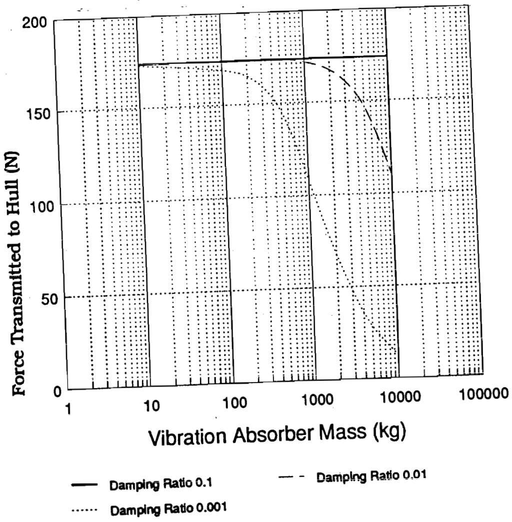

Figure 7. Influence of Mass and Damping on Force Transmitted to Hull

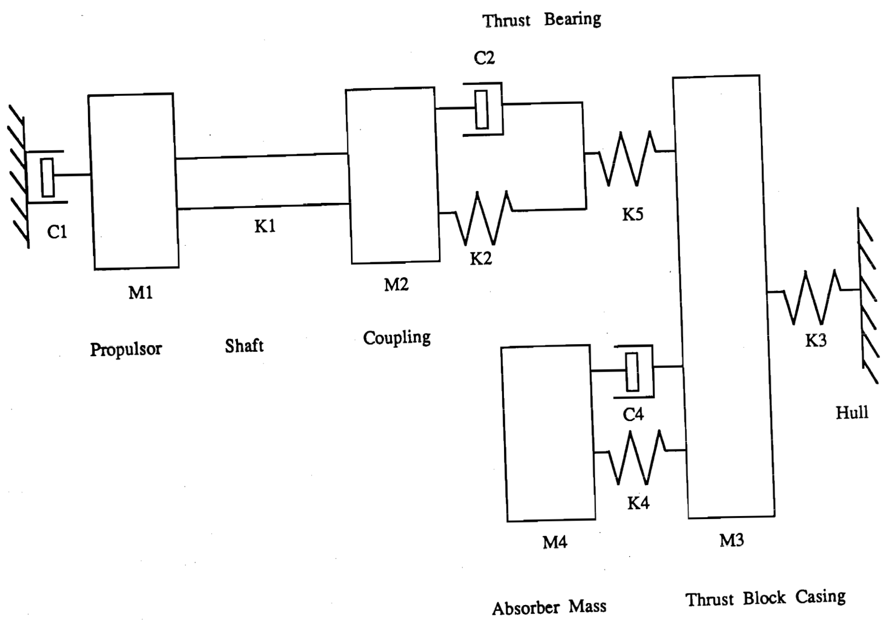

Figure 6. Propulsor Drive with Mass/Spring Damper Vibration Absorber

For example, a 1 tonne vibration absorber with active damping providing a damping ratio of 0.01 would raise the transmission of force to within 65% of the standard vessel.

Alternative mounting positions for an active device were modelled as above but as before the system was very sensitive to tuning errors.

4 The limitations of a magnetically tuned vibration absorber

The use of a magnetic actuator as a tunable mass/spring/damper vibration absorber clearly has limitations. In effect the tunable passive device is subject to the same problems as a simple mass/spring/damper. In this application, the sensitivity of the system to speed changes either requires unacceptably accurate tuning from the magnetic actuator or a large absorber reaction mass. Clearly, this approach is not acceptable and an adaptive vibration controller reacting to the system parameters is necessary. Once again, the magnetic actuator would generate a force which was reacted against a free mass. The size of the reacting mass required in an adaptive device would be no greater than that necessary in the device investigated above. A reaction mass of 250 Kg would have an amplitude of less than 1 mm if used to cancel a 7 Hz fluctuating force of between 100N and 150N at the propulsor. In this case the actuator force would no longer be a function of the mass position and velocity but would be dependent on transducer signals from elsewhere in the vessel.

5 Alternative control strategies

As noted earlier, the use of localised feedback of position and velocity which causes a magnetic actuator to behave like a controllable spring and dashpot unnecessarily restricts the capabilities of these devices. The primary reason for using collocation of the sensors and actuators [8] referred to by some authors, as decentralized control [9] is that it avoids the problem of instability associated with the use of state feedback to assign the eigenvalues in a multivariable system. This problem will become manifest if the simple model used in Fig.1 is made more representative of the physical system by modelling the shaft by a larger number of mass/spring elements, and by examining the effect of coupling between the radial and axial motion. Thus four state variables will not be sufficient to model the system. In order to achieve eigenvalue assignment to arbitrarily determined locations it is necessary to measure or estimate all of the state variables. This requirement, coupled with the possibility of the closed-loop system becoming unstable explains why other workers have adopted a very simple control philosophy. By using local feedback and causing the magnetic actuator to act like a spring and damper the possibility of instability is avoided. This is at the cost of an improved performance attainable by utilising the full capability of the magnetic actuator.

These problems can be overcome by using an open-loop adaptive control strategy developed by Burrows and Sahinkaya [5] for the control of the synchronous vibration of an unbalanced rotor. As well as avoiding problems associated with stability the technique in its developed form [10] allows the control action to minimise rotor vibrations without prior knowledge of system parameters. This is of prime importance in the application considered here where the bearing stiffness and damping properties vary with speed and temperature etc.

This approach employs the full capability of magnetic actuation compared with the more restricted approach adopted in many applications reported in the literature.

The open-loop adaptive algorithm uses measurements of rotor displacement distributed along the shaft. Any measurement at a critical shaft location can be weighted in the performance index which assesses the effectiveness of the control action. The object is to control the amplitude and phase of the force applied by the magnetic actuator to minimise the performance index or cost function which typically is the sum of squares of displacement. The results presented in [10] show how effective this strategy is. Sahinkaya and Burrows have shown elsewhere [11] how it can be combined with local feedback to achieve various design goals. In summary, if more complex control strategies are used it is clear that it will be possible to significantly reduce the force transmissibility compared with the simple tuned absorber discussed earlier.

It may be possible to extend this approach and control subharmonic and multiple harmonic vibration in addition to that at the blade passage frequency.

6 Conclusions

This study has shown that the use of electro-magnetics for vibration isolation in a ship at the blade passage is feasible using an absorber mass of acceptable magnitude.

In order to achieve acceptable performance over a range of operating conditions it is necessary to employ the full capability of magnetic actuators and not restrict their use to tunable spring and damper elements. This collocation of the sensors and actuators is not capable of achieving the required reduction in transmissibility.

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