ASEAN Engineering Journal

Full Paper

A REVIEW OF THE DYNAMIC ANALYSIS OF AXIAL VIBRATIONS IN MARINE PROPULSION SHAFTING SYSTEM DUE TO PROPELLER EXCITATION

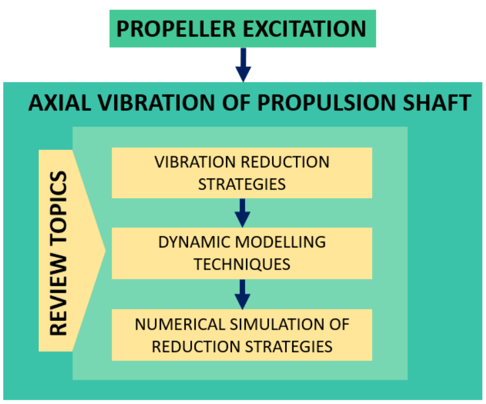
Awang Azhad Awang Amirudin, Khairul Izzati Kamarumtham, Aditya Agung Haripriyono, Azrul Aminur Rahman Yunus, Nur Amirah Syahmina Suheimy, Yaseen Adnan Ahmed*

School of Mechanical Engineering, Faculty of Engineering, University of Technology Malaysia, Johor Bahru, Malaysia,

Article history
Received
04 March 2021
Received in revised form
03 May 2021
Accepted
01 January 2022
Published online
31 May 2022

*Corresponding author
adnanahmed@utm.edu.my

Graphical abstract



Abstract

Over the past decades, the attenuation of axial vibration due to propeller excitation has long been a complex problem due to the coupling dynamics of the propeller-shafting system. As axial vibration is often the cause of fatigue damage to the propulsion shaft components as well as the root cause to acoustical radiation along a hull of a ship, there has been continuous interest in understanding the complex dynamic characteristics of the coupled propeller-shaft system and the methods to suppress the axial vibration in the system. Therefore, several studies have been conducted to solve this vibration problem on the longitudinal axis of the marine propulsion shafting system. This paper aims to provide the theoretical foundations of this problem by reviewing the modelling techniques of this coupled dynamic problem and cover the vibration reduction strategies that are proposed by the cited studies.

Keywords: Axial vibration, Coupled propeller-shaft system, Marine propulsion shafting system, Vibration reduction strategies

1.0 INTRODUCTION

According to [1], the vibration experienced on-board ships can be divided into two categories: 1) Local vibration and 2) Synchronous vibration. When the vibration only occurs in isolated parts or certain fittings, it is classified as a local vibration. These vibrations can cause much discomfort to the crew-members but unlikely to cause deformation on the ship structures [2]. The local vibration can be treated by calibrating the resonating structural parts or utilising vibration absorption and damping devices.

On the other hand, as reported in [1], synchronous vibration can cause the whole hull girder to vibrate, in which the motion of the hull can be observed manifesting along the length of the ship, reaching an amplitude of up to 1 inch at the bow and stern. In addition to the discomfort of the crew members, Ran et al. [2] added that synchronous vibration can cause deformation on the ship structures. Synchronous vibration is primarily due to the continuous transmission of excitation forces from the main engine and propeller [3]. Additionally, it was noted in [4] that the rotation of the propeller in an unsteady, random and fluctuating wake field results in the blade passing frequency (BPF) and the corresponding harmonics and the random broadband spectrum. In turn, the propeller force will excite the hull, propeller and shaft resonances which will produce broadband noises.

According to [3], propeller-induced vibrations are typically due to the following exciting forces: 1) surface forces, and 2) shaft forces. The presence of a propeller in the non-uniform wake of a hull, the hydrodynamic pressures levied by a propeller due to cavitation, or the dynamic characteristic of a propeller itself will cause the shaft forces to fluctuate. Ultimately, as stated in [5], the fluctuations of the shaft forces will cause axial vibration to occur on the coupled propeller-shaft system. Additionally, [6] reported that the vibration energy that is transmitted along the axial direction is dominant. The aim of this work is to review the working principles of several vibration suppression technologies and the dynamic responses of the propeller-shaft system with vibration isolators. This review encompasses four principal sections. The first section will review the analysis methods used by the cited studies. The second section reviews the literature behind the different vibration suppression strategies and devices. The third section will introduce the equivalent models of the propeller and propeller shaft system. Finally, the last section will discuss the numerical analysis of the dynamic response in the longitudinal axis attributed to the propulsion shaft systems with different vibration attenuation devices described in third section.

2.0 METHODOLOGY

Analysis Methods

The analysis and design of the propulsion shaft system subjected to fluctuating loads and vibrations require reliable and efficient analysis tools to predict the dynamic responses of the system. In the case of this propeller-shaft coupling problem, several works have opted to use the Transfer Matrix Method such as in [4] and [7], while in [8], the equivalent model of the propeller-shaft system was developed using the Euler-Bernoulli beam theory. Conversely, according to [9], the Euler-Bernoulli beam is not accurate enough compared to the Timoshenko beam. In [8], [10], the FRF sub-structuring method was used to obtain the propulsion shaft response due to fluctuating propeller forces.

Several studies employed commercial software such as ANSYS and ABAQUS to develop the dynamic models of the propeller [4], [8], [11]. In [4] and [8], it was noted that the mode shapes of the propeller could be solved using the coupled Finite Element Method (FEM) or Boundary Element Method (BEM) using the software. Additionally, Huang et al. [8] also used ANSYS to validate their proposed equivalent model of the propeller-shaft system. Similarly, Liu et al. [10] used the modal analysis module in ANSYS to determine the natural frequencies of their Semi-Active Dynamic Vibration Absorber (SDVA) and its frequency shift characteristics. Owing to the complexity of the propulsion system, these models can only analyse the vibrations of the propeller-shafting system in a two-dimensional space. To tackle this, several studies have opted to conduct experimentation to analyse the dynamic responses of the propulsion shafting when introduced with static or fluctuating thrust [10], [12], [13].

3.0 LITERATURE REVIEW

Force and Noise Transmission

The transmission path of propeller thrust in the propeller-shafting system can be described as follows: propeller → shafting → thrust bearing → foundation → hull. While the thrust bearing transmits propeller thrust to the hull, it also provides the transmission channel of axial vibration from propulsion shafting to different hull regions, resulting in subsequent reactions of the hull. Thus, it was stated in [14] that the axial vibration of the propeller-shafting system is one of the secondary excitation sources of hull vibration relative to the primary excitation sources, such as propeller, diesel generator, and auxiliary machinery. Moreover, the acoustic radiation of hull at low frequencies is associated with axial vibration transmission through the propeller-shafting [13], [15]. Hence, the minimisation of axial vibration intensity of propulsion shafting and oscillatory thrust load transmitted to the hull had received much research attention [16].

Effect of Propeller Forces on Local and Machinery Vibrations

According to [17] the vertical beam-like modes of vibration of the hull girders of modern ships may become serious in two respects: 1) They can be excited to excessive levels by resonances with the dominant low-frequency excitations of slow-running diesel main engines, and 2) Vertical vibration of the hull girder in response to propeller excitation is a direct exciter of objectionable fore-and-aft superstructure vibration.

The propeller is generally incapable of exciting the hull girder modes themselves to dangerous levels. It is mainly because the higher hull girder modes with natural frequencies that fall in the frequency range where the propeller has significantly low excitability. However, the low-level vertical hull girder vibration that does occur either directly from the propeller or indirectly via the main shafting thrust bearing serves as the base excitation for excessive vibration of superstructures and other attached subsystems which are in resonance with the propeller exciting frequencies.

Propeller Excitation Reduction Strategies

Non-Contact Underwater Vibration Measurement Method

In [18], [19], it was found that the laser Doppler vibrometer (LDV) showed good potential as a non-contact measurement method and can perform remote measurements of structural vibration. Meanwhile, in [20], two vibration measurement techniques were proposed in the Lagrange coordinate system using the TLDV and the Euler coordinate system which uses the stationary LDV. Both systems work by impinging a sensing laser beam onto the blade remotely. According to [20], the LDV detects the shift in the Doppler frequency applied to the laser beam as the target object vibrates. This frequency shift is the basis of the velocity or displacement of the object. For blade tracking, the LMS is actuated using a rapid galvanomotor which synchronises with the rotation of the propeller based on the triggers from the MCU-XP device. The vibration signals captured are sent to a DAQ PC after passing through the in-line bandpass (BP) filter. Figure 1 illustrates the signal flow of the entire system.

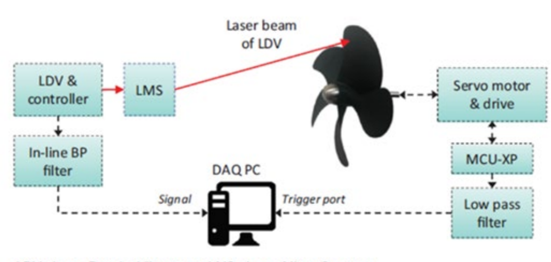


Figure 1. Schematic for the laser Doppler vibrometer based blade dynamic response measurement system [20]

Vibration Isolator Types and Theoretical Analysis

In general, there are two types of vibration control devices: 1) Passive control devices, and 2) Active control devices. Passive control devices require no additional energy to operate, whereas, active control devices require additional energy such as electric or magnetic energy to operate. Additionally, the propeller-induced vibrations may be realised after the completion of a ship, and the initiatives that can be taken to suppress the vibration is very limited after this point [25]. Saydam [25] also added that wake modification devices, namely duct, and vortex generator, can be used as retrofit solutions to suppress the vibration. This section will focus on the discussion of these mechanisms.

i) Passive Control Devices

Periodic Isolator

A lot of work has shown promising results for the use of periodic structures as vibration-reducing strategies [21]–[23]. Periodic structures are structures that are constructed by a series of identical substructures, joined together in identical ways to form the complete system [24]. According to [22], [23], periodic structures are widely known for their dynamic behaviours, in which vibration can only pass through the structures in some specific frequency band, called the "pass band", and alternatively, the frequency band in which vibration could not pass is called the "stop band". In [21], periodically layered isolator models were developed to assess the capacity of periodic structures in reducing propeller-induced axial vibrations of an underwater vehicle.

According to [21], overall, periodic isolators have superior performance in suppressing vibration compared to the homogeneous isolator. It was found that the amplitudes of the peaks due to the resonance of the periodic isolators are smaller but bigger compared to ideal isolators [21]. Nevertheless, overall, periodic isolators perform better than ideal and homogeneous isolators. Yet, improvement could be made to enhance the performance of the periodic isolators in the low-frequency pass band. Thus, this study proposed an integrated isolation device, which consist of a periodic isolator and a dynamic vibration absorber to solve the issue. Song et al. also [21] found that the performance of the integrated isolation devices are superior compared to the basic periodic and homogeneous isolators, particularly at low frequencies. Better performance can also be obtained when the absorber is fixed into the periodic isolator as opposed to the shaft, as the light and soft structure of the isolator will suppress the vibration even further [20].

Ducts

A lot of work has been done to study the efficacy of the enclosure of a propeller within a duct in improving the propulsion performance and the cavitation conditions of the propeller [26]–[29]. Çelik [28] claimed that a wake equalising duct (WED) is one of the most frequently utilised energy saving devices to enhance the propulsion performance of a ship, and suppressing the propeller-induced vibrations and viscous resistance forces. A numerical study conducted in [28] for calculating the effect of a WED on the propulsion performance of a chemical tanker has shown that propulsive efficiency is improved by 9.7 per cent when a well-designed WED is used. However, Martinas [29] found that WED has negligible effects in reducing the negative effects of cavitation, even after dimensional and form optimisation of the device.

Alternatively, [26], [27] studied the effects of decelerating duct and accelerating duct, respectively, on the characteristics of the flow around a marine propeller. These studies concluded that an accelerating duct is commonly adopted when a better propulsion performance is desired, whereas, when suppression of the cavitation and retardation of its adverse effects namely noise and vibrations are the matter of concerns, a decelerating duct should be adopted.

Vortex Generator

Numerical studies in [25], [30] found that vortex generators can distinctly suppress the propeller-induced pressure fluctuations. The Reynold's number is the primary parameter in determining the optimal angle of attack of the vortex generator [30]. In Huang et al. [31] an experiment was conducted to study the influences of vortex generators on propeller cavitation and hull pressure fluctuations, and have found that the vortex generators can cause a drop in propeller-induced pressure fluctuations by preventing boundary layer separation and reducing the turbulence of the wake field. Numerical and experimental studies that were carried out in [25] have also shown that addition of 3 well-designed vortex generators on each side of the tanker has reduced the propeller-induced longitudinal and vertical vibrations by 50 per cent.

Composite Propulsion Systems

Many studies have been conducted to assess the capacity of composite material in improving vessel performance, particularly for military vessels as stealth performance for such vessels is indispensable [32]–[34]. However, the number of published papers encompassing this subject is rather limited due to confidentiality issues [32].

In [33], a large-diameter composite propulsion shaft was developed to fulfil navy-standard performance requirements. This study claimed that advanced composite material can improve the marine propulsion shafting system in terms of vibration damping. Mouritz et al. [32] reported that a propulsion shaft made from composite material namely carbon fibre/epoxy and glass fibre/epoxy could be 25-80 per cent lighter than a steel shaft of an identical dimension. Furthermore, Mouritz et al. also wrote that the minimisation in the transmission of noise from machinery and propellers can be expected from the utilisation of a composite shaft due to its intrinsic damping properties. Additionally, propellers made from composite materials reduced the magnitude of the resonance vibrations in the engine and propeller shafting by approximately 25 per cent, thereby reducing hull vibration and noise [35].

Resonant Changer

Since the pioneering work conducted by Goodwin in 1960, studies undertaken to improve Goodwin's innovation by optimising several of the device parameters have been steadily growing in number [7], [36], [37]. The performance of the resonance changer developed by Goodwin had exceeded expectations by eliminating the resonance conditions of a ship that previously had a resonance problem [36]. In [36], a genetic and a general nonlinear constrained algorithm was adopted to determine the optimal values for the virtual resonant changer parameters. Alternatively, in [37], a numerical study was conducted to identify the global optimum virtual stiffness, damping, and mass parameter of the resonance changer of a submarine. This study implemented the method of moving asymptotes to optimise the resonant changer parameters. On the other hand, Zechao et al. [7], adopted the design method of the parameter correction of the curve area to improve the longitudinal vibration suppression effect of the resonant changer on the marine propulsion shafting.

ii) Active Vibration Control Devices

Pneumatic Servo Active Control Technology

Baz et al. [5] developed and tested a theoretical model and a prototype of an active vibration control system based on a self-contained pneumatic servo-control system that is powered by high-pressure air. The theoretical model was developed to predict the performance of the pneumatic servo active control system when subjected to step and sinusoidal thrust forces of different magnitudes and frequencies. In this study, a discrepancy was observed between the theoretical and experimental data which was attributed to the exclusion of bearing and seal friction forces in the theoretical model.

Nevertheless, the result was sufficiently consistent. The developed active control system can effectively suppress the axial vibrations of propeller shafts [5]. The system is found to minimise the magnitude of propeller-induced axial vibrations by approximately 11dB for excitation frequencies up to 10 Hz. However, the performance of the system is found to rapidly deteriorate with increasing frequency beyond this range. From this study, it was proposed that a hydraulic, rather than pneumatic controller can maintain the favourable performance from the system in a wider range of frequencies.

Piezoelectric-Integrated Active Control Technology

Several piezoelectric-integrated active vibration controller models have been developed by a number of researchers [38]–[41]. Kwak et al. [39] developed a piezoelectric-integrated active vibration controller by considering the modified higher harmonic control (HHC), in which, according [42], [43], HHC is, in fact, a band rejection filter. A theoretical study of the controller's vibration suppression capacity revealed that the proposed controller has the capacity to achieve the expected performance without any instability issues. Subsequently, an experimental study was conducted to validate the theoretical study by submerging a harmonically disturbed ring-stiffened cylindrical shell into a tank of water. The findings from the water-tank experiment suggest that the controller proposed by Kwak et al. [39] can suppress the subjected harmonic disturbance possessing either resonant or non-resonant frequency. Similarly, Loghmani et al. [40] found that increasing the number of piezoelectric sensors and actuators can improve the performance of a piezoelectric-integrated active controller.

Alternatively, Lu et al. [41] have designed and theoretically analysed piezoelectric-based propeller blades for a novel unmanned underwater vehicle (UUV), and conducted an FEM simulation analysis to verify the results from the theoretical analysis. From the analyses in [41], it was discovered that the amplitudes of propeller blades could be suppressed by more than 70 per cent by applying a piezoelectric coating.

Semi-Active Dynamic Vibration Absorbers Based on Magnetorheological Elastomers (MRE)

In [44], [45], it was found that dynamic vibration absorbers (DVA) is an effective method to reduce shafting's axial vibration. MREs can be used as variable stiffness elements to make the device more robust and lighter According to [46], MREs are composites in which highly elastic polymer matrices are filled with magnetic particles. The unique characteristic of MRE is that its shear storage modulus can be controlled by the external magnetic field rapidly, continuously, and reversibly. In [46], an adaptive tunable vibration absorber using MREs was developed. Similarly, in [47] and [48] a torsional adaptive tunable vibration absorber using an MRE for vibration reduction of a powertrain test rig was developed. The experimental results show that the ADVA can work between 10.75 to 16.5 Hz (53% relative change), and the numerical simulations show that the steady-state vibration of the powertrain can be significantly reduced.

As reported in [10], despite being vastly applied in many domains, the applications of MRE in axial vibration absorption of the propeller-shafting system are still rare. Nevertheless, the findings in [10] and [44], demonstrated a sufficient overview of the capacity of an MRE-SDVA in minimising propeller-induced axial vibrations. Yang et al. [44] designed a single MRE-SDVA for ship shafting axial vibration control and conducted the theoretical analysis to identify the shift-frequency property of the system. In a comparative study in [44] using FEM structural models of the DVA-fitted ship shafting system, it was found that the proposed MRE-SDVAs have a superior capacity in reducing propeller-excited axial vibration over the ship-shafting with a passive DVAs and without DVA. Liu et al. [10] found that their proposed MRE-SDVA model can reduce vibration in the medium and high-frequency range. However, the proposed model was unable to reduce vibration in a low-frequency range, and Liu et al. claimed that the untoward outcome was due to the high damping of the MRE-SDVA model.

4.0 MODELLING

Dynamic Model of Propeller

Several studies have been conducted to study the dynamics of a coupled propeller-shaft system [4], [8]. The propeller is not an entirely rigid component, which would under-predict the transmission of force to the thrust bearing [4], [8]. Therefore, both works deliberated that the propeller should be modelled elastically to simulate its dynamics more accurately. The propeller's first several mode shapes are assumed as the deformation of the blades as the hub has a very high stiffness and any deformation is negligible [4]. In this work, the propeller is represented as an m_2-spring-k_2 system (Figure 2). The coupling factor for the model in [4] is denoted as α_1 described by Eq. (1).

$$\alpha_1 = \frac{IM_p\phi}{\phi^T M_p \phi} \tag{1}$$

Where φ is the propeller's mode vector in the axial direction, I is the unitary matrix, and M_p is the mass matrix of the propeller. In [8] and [47], the blades of the propeller are represented as cantilever beams, which can be expressed analytically as continuous beams based on the Euler-Bernoulli theory, Figure 2 illustrates the reduced propeller model proposed in [4] and [8]. In [8], the elastic section of the blade (the section which flexes the most and accounts for the anti-resonance of the blades) is modelled as m_2 and a spring with a constant of k_2, representing the stiffness of the edge of the blade. Meanwhile, the forces that act on the propeller described in this work are the static thrust force, F_0, and the fluctuating force, F.

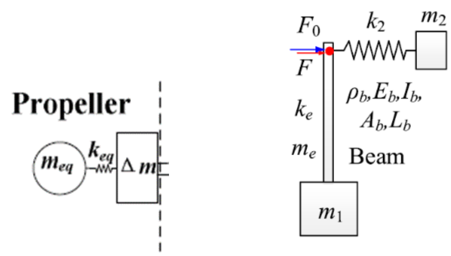


Figure 2. (Left) Dynamic model of a propeller [4], (Right) Dynamic model of a propeller [8].

Dynamic Model of Propeller-Shaft System

The propellers can be represented by a lumped mass attached to a continuous elastic shafting model [4], [8], [47]. In contrast, Huang et al. proposed a shaft model comprising of several lumped masses (Figure 3) [48].

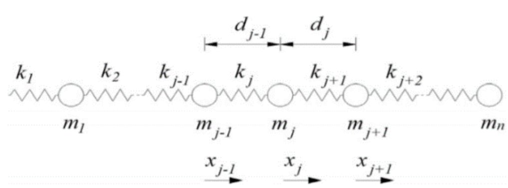


Figure 3. Schematic of the lumped mass shaft model [48]

In [4], the propulsion shaft is isolated from the engine side (Figure 4). The Transfer Matrix Method is utilised to solve the responses between the different subsystems (i.e., propeller, vibration isolator, thrust bearing, shaft). The responses between the components of the subsystems, in turn, are described by transfer matrices which would reflect the interaction between said components in terms of axial displacement and force. In [4], the transfer matrix of the propeller, T_p is given by Eq. (2).

$$T_p = T_{\Delta m} \cdot T_{k_{eq}} \cdot T_{m_{eq}} \tag{2}$$

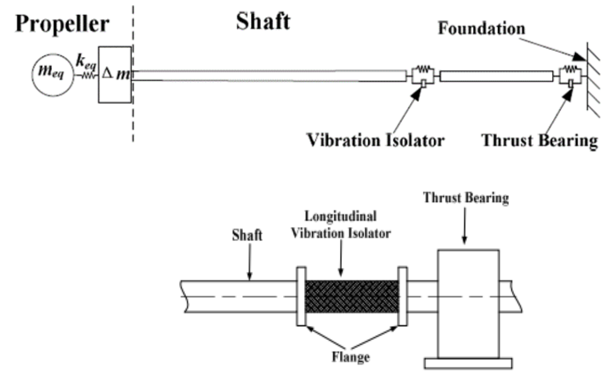


Figure 4: (Top) Equivalent model of a propeller shaft system, (Bottom) Longitudinal vibration isolator [4]

Li et al. presented a transfer matrix to represent a bar in the longitudinal vibration in Eq. (3) [4].

$$T_{bar} = \begin{bmatrix} \cos \beta_s L_s & -\dfrac{\sin \beta_s L_s}{\beta_s E_s A_s} \\ \beta_s E_s A_s \sin \beta_s L_s & \cos \beta_s L_s \end{bmatrix} \tag{3}$$

Where L_s is the length of the shaft, A_s is the cross-sectional area of the shaft, E_s is Young's modulus of the shaft and $\beta_s = \frac{\omega}{c_s}$ is the longitudinal wave number. In turn, the whole propulsion shaft system can be expressed in terms of subsystems as described in Eq. (4) and Eq. (5).

$$T_{sys\,(with\,isolator)} = T_{th} \cdot T_{s2} \cdot T_i \cdot T_{s1} \cdot T_p \tag{4}$$

$$T_{sys\,(no\,isolator)} = T_{th} \cdot T_s \cdot T_p \tag{5}$$

Where T_i is the transfer matrix for the isolator. In Eq. (6), the shaft is represented T_{s1} and T_{s2} to indicate the two sections of the shaft separated by the isolator. Huang et al. have presented a different dynamic model of the propeller shaft, as illustrated in Figure 5, and the equation of motion of the shaft rod is described by Eq. (6) and Eq. (7) [8].

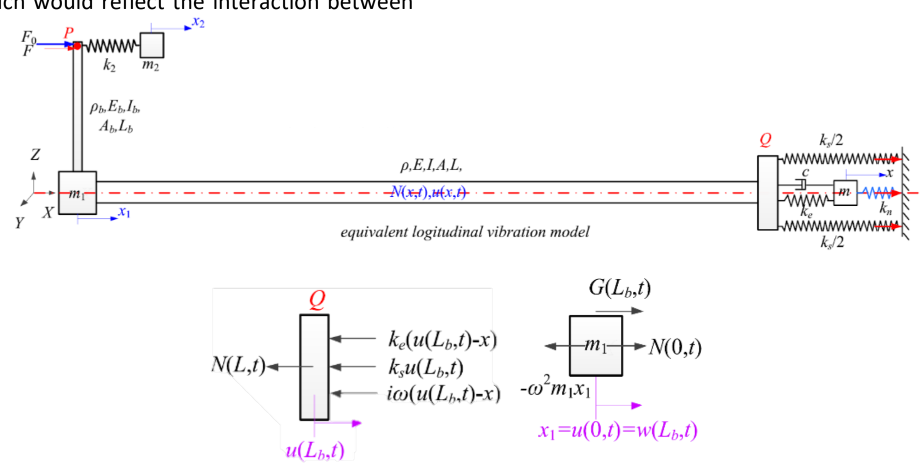


Figure 5. Equivalent model of a propeller-shaft-DVA system and shaft end nodes [8]

$$N(x,t) = EA\frac{\partial u(x,t)}{\partial x} \tag{6}$$

$$= iEAk\left(u_1 e^{ikx} + u_2 e^{-ikx}\right)e^{i\omega t}, k = \sqrt{\frac{E}{\rho}} \tag{7}$$

Where $u(x,t)$ is the displacement in the longitudinal direction, $N(x,t)$ is the force in the longitudinal direction, k is the wave number of the longitudinal waves, and u_i $(i = 1,2)$ are the coefficients that need to be determined.

In contrast to Li et al.'s vibration isolator in [4], Huang et al. [8] proposed the Dynamic Vibration Absorber (DVA) as a mechanism for axial vibration attenuation. Therefore, the equation of motion of the propeller-shaft-bearing system accounting for the thrust bearing and DVA is given by Eq. (8).

$$m\ddot{x} + c\left(\dot{x} - u(L,t)\right) + k_e\left(x - u(L,t)\right) + f_n(x) = 0 \tag{8}$$

Where m is the mass of the embedded DVA, k_e is its stiffness, $c = 2\zeta\sqrt{mk_e}$ is the damping coefficient, $f_n(x) = -k_n x$ is the force due to the negative stiffness, x_1, x_2 and x is the displacement of the mass m_1, m_2 and m, and $k_e\left(x - u(L,t)\right) + f_n(x)$ represents the force transmitted by the thrust bearing. The force transmissibility *TF* is defined by Eq. (9)[4]. Where f_{th} represents the reaction force at the thrust bearing, f_p represents the axial force at the propeller. Whereas in [8], the force transmitted by the thrust bearing to the foundation is given by Eq. (10).

$$TF = \left|\frac{f_{th}}{f_p}\right| = \left|\boldsymbol{T}_{sys}(2,1)\left(\frac{u_p}{f_p}\right) + \boldsymbol{T}_{sys}(2,2)\right| \tag{9}$$

$$k_e\left(x - u(L,t)\right) + f_n(x) \tag{10}$$

5.0 NUMERICAL ANALYSIS AND DISCUSSION

Influence of Propeller Elasticity

For the numerical analysis of the system, a three bladed propeller was considered in [4]. To account for the effect of the surrounding water, Li et al. [4] included an added mass effect of 1.1~1.3 times the density of the propeller. Meanwhile, in [8], the added mass coefficient was set to 1.3 of the mass of the propeller. Li et al., [4], the hysteric damping was accounted for in the propeller and shaft system by assigning a complex Young's Modulus E(1-jη) with a loss factor η of 0.02. In [44], the structural model of the shafting with SDVA was modelled in ANSYS using the Finite Element Method. In [4], the transfer matrix dynamic model is validated using FEM, while in [8], the dynamic model was validated using ANSYS.

Rigid propellers models do not accurately reflect the dynamic responses of the propellers [4]. Li et al. [4] investigated the effect of the propeller's natural frequency on the vibration transmission in the propeller-shaft system by simulating the reaction force transmissibility for three cases of propeller elasticity. The equivalent stiffnesses of each case were determined in a way so that the first natural frequency of the propeller is 78Hz, 50Hz and 38Hz denoted as Cases B, C and D, respectively. Meanwhile, the natural frequency of the shaft is defined as 55 Hz. As expected, this analysis shows that the propeller with the closest natural frequency (i.e. 50 Hz) produced the largest axial force transmission to the shaft as illustrated in Figure 6.

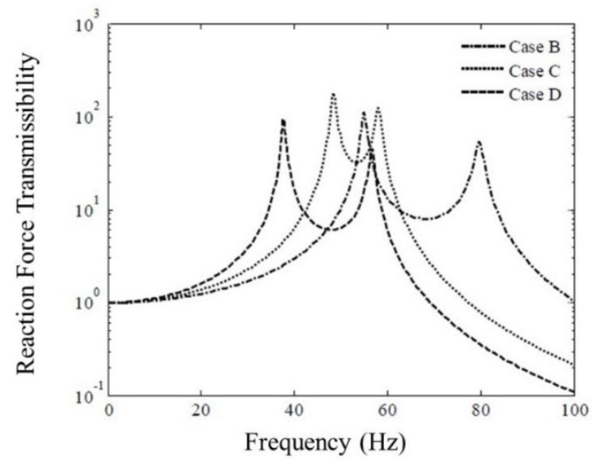


Figure 6. Influence of the natural frequency of the propeller on the reaction force transmissibility [4]

Longitudinal Vibration Isolator Performances

To investigate the relationship between the stiffness of the isolator and force transmissibility, [4] examined the dynamic responses (i.e., amplitude and resonance frequency) of the shaft with vibration isolators with three different stiffnesses. In this study, the amplitude and resonance frequency of the shaft at the first peak increases with the stiffness of the vibration isolator. However, all three cases (with isolators) showed little difference in the dynamic responses at the second peak. This is most apparent for the second mode or peak as illustrated in Figure 7. In [44], the same pattern is observed with the passive DVA, in which the highest vibration suppression efficiency occurs at the natural frequency of the shaft but two resonant frequency peaks were introduced at 82.5 and 100 Hz. In comparison, an SDVA can attenuate the vibration across the entire excitation frequency bandwidth as shown in Figure 7.

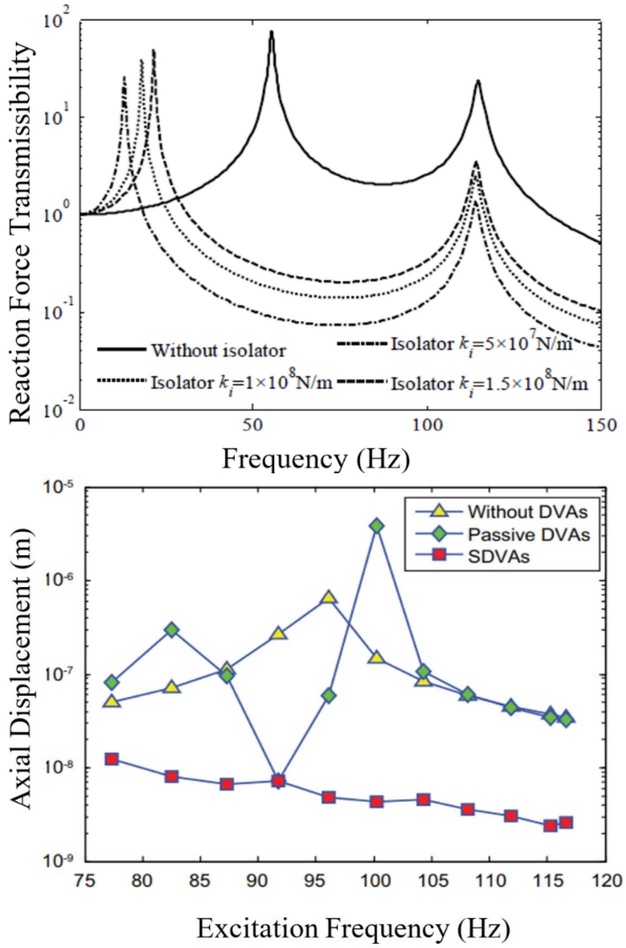


Figure 7. (Top) Influence of the isolator on the reaction force transmissibility [4]. (Bottom) Axial vibration amplitudes of shaft without isolators, with PDVA and with SDVA [44]

In Huang et al., a comparative study was conducted between DVAs with no negative stiffness and DVAs with maximum negative stiffness and a mass ratio of the absorber to modal mass of only µ_r=0.003 in a static propeller thrust setting [8]. It was observed that DVAs with negative stiffnesses exhibit the suppression in longitudinal force transmission but can suffer extreme static deformation without the proper optimisation of the mass and damping ratio (0.003 and 0.69 respectively). From this study, the effect of the mass ratio on the vibration attenuation was concluded to be insignificant. This study then investigates the relationship between the stiffness ratio, p_r, of the negative stiffness and the modal stiffness of the DVA. A larger p_r denotes a larger negative stiffness. In this parametric study, it was observed that a higher p_r will result in greater vibration attenuation, as shown in Figure 8. Additionally, Huang et al. have noted that the damping coefficient was a key factor in controlling the force transmission [8].

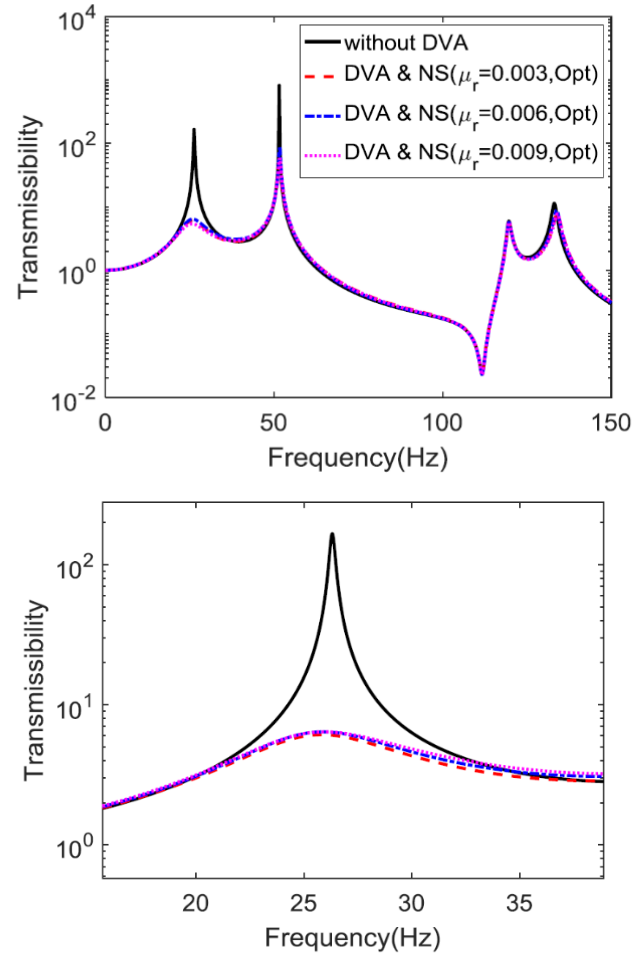


Figure 8. Comparison of longitudinal force transmissibility by the thrust bearing for different p_r (µr= 0.003) [8].

6.0 CONCLUSION

This paper presents a review of the vibration suppression devices and the dynamic analysis of a coupled propeller shaft system subjected to axial vibration due to propeller excitation. There are numerous implications to unsuppressed axial vibration along the propulsion shaft, namely the fatigue of the thrust bearing and the transmission of reaction force due to the propeller excitation to the hull of the vessel and thrust bearing. The latter is known to induce fatigue damage in equipment. From the standpoint of the vibration attenuation, there are numerous methods of vibration suppression that range from passive to active controllers. Furthermore, through various modelling techniques (i.e. TMM, FEM, and BEM), it can be observed that to produce a near accurate reduced model of a propeller, the flexural characteristics of the propeller should be accounted as this will affect the mode shapes of the propeller at different frequencies. Finally, this work investigates the effectiveness of several in-line vibration isolators in suppressing axial vibration along the propeller shaft. Overall, this work can instigate further studies to propose cost-effective vibration suppression methods such as using DVAs with negative stiffness.

Acknowledgement

We would like to thank Dr Henry Kang for his guidance while writing this paper.

References

[1] H. Ucar, 2011. "Ship Hull Girder Vibration," *Journal of Naval Science and Engineerin* 7(1): 1–21,

[2] Ran, J. Pan, P. J. O. Shea, and C. K. Mechefske, 2009. "A study of vibration and vibration control of ship structures," *Marine Structures.* 22(4): 730–743,

[3] Y. Okumoto, Y. Takeda, M. Mano, and T. Okada, 2009. *Design of Ship Hull Structures, A Practical Guide for Engineers*. Springer,

[4] C. Li, X. Huang, and H. Hua, 2016 "Dynamic modeling and analysis of axial vibration of a coupled propeller and shaft system," *Journal of Mechanical Scence and. Technology*, 30(7): 2953–2960, doi: 10.1007/s12206-016-0604-9.

[5] A. Baz, J. Gilheany, and P. Steimel. 1990, "Active vibration control of propeller shafts," *Journal of Sound and Vibtration*, 136(3): 361–372. doi: 10.1016/0022-460X(90)90450.

[6] Y. Chen, L. Wang, and H. X. Hua, 2017, "Longitudinal vibration of marine propeller-shafting system induced by inflow turbulence," *Journal of Fluids and Structures*, vol. 68, pp. 264–278, doi: 10.1016/j.jfluidstructs.2016.11.002.

[7] H. Zechao, H. Lin, X. Wei, L. Zhengmin, and Z. Xingqian, 2019. "Optimization design of resonance changer for marine propulsion shafting in longitudinal vibration," *Chinese Journal of Ship Research*, 14(1): 78–85

[8] X. Huang, Z. Su, and H. Hua, 2018, "Application of a dynamic vibration absorber with negative stiffness for control of a marine shafting system," *Ocean Engineering*, 155: 131–143, doi: 10.1016/j.oceaneng.2018.02.047.

[9] W. Kang, Z. Zhang, K. Zhou, and Y. Chen, 2019, "The random vibration and force transmission characteristics of the elastic propeller-shafting system induced by inflow turbulence," *Ocean Engineering* 188: 06317, doi: 10.1016/j.oceaneng.20 19.106317.

[10] G. Liu, K. Lu, D. Zou, Z. Xie, and Z. Rao, 2017. "Development of a semi-active dynamic vibration absorber for longitudinal vibration of propulsion shaft system based on magnetorheological elastomer," *Smart Materials and Structures*, 26(7): 075009, 1–16,

[11] G. Zhang, Y. Zhao, T. Li, and X. Zhu, 2014. "Propeller excitation of longitudinal vibration characteristics of marine propulsion shafting system," *Shock and Vibration*, 1–19

[12] J. Pan, N. Farag, T. Lin, and R. Juniper, 2002. "Propeller induced structural vibration through the thrust bearing," in *Procedding of Acoustics 2002-Innovation in Acoustics and Vibration Annual Conference of the Australian Acoustical Society* .390–399

[13] Z. W. Huang and Y. H. Tan, 2019. "Experimental Investigation on Active Longitudinal Vibration Suppression of the Thrust Bearing," *Journal of Physics. Conference Series*, 1314: 12070

[14] G. B. Zhang and Y. Zhao, 2012. "Analysis of vibration and acoustic radiation of submarine hull induced by longitudinal vibration of propulsion shafting," Noise and Vibration Control, 32(3): 155–158

[15] [15] Y. S. Wei and Y. S. Wang, 2013. "Unsteady hydrodynamics of blade forces and acoustic response of a model scaled submarine excited by propeller's thrust and side-forces," *Journal of Sound and Vibration*, 332: 2038–2056

[16] Y. Zhao, G. B. Zhang, and L. W. Li, 2011. "Review of advances on longitudinal vibration of ship propulsion shafting and its control technology," *Shipbuilding in China*. 52(4): 259–269

[17] Ship Vibration, "Guidance Notes On," Eagle.org." [Online]. Available:https://ww2.eagle.org/content/dam/eagle/rules-and-guides /current/conventional_ocean_service/147_shipvibration/ship-vibration-gn-feb18.pdf (accessed April 03, 2021).

[18] P. Castellini, 2002. "Vibration measurements by tracking laser Doppler vibrometer on automotive components," *Shock Vibration.* 9(1–2): 67–89

[19] C. T. Truong, D. H. Kang, J. R. Lee, and F. CR, 2015 "Comparative study of laser Doppler vibrometer and capacitive air -coupled transducer for ultrasonic propagation imager and the new development of an efficient ultrasonic wavenumber imaging algorithm," *Strain*, 51(4): 332–342, , doi: 10.1111/str.12144.

[20] S. H. Abbas, J. K. Jang, D. H. Kim, and J. R. Lee, 2020, "Underwater vibration analysis method for rotating propeller blades using laser Doppler vibrometer," *Optics and Lasers in Engineering.* 132: 106133, doi: 10.1016/j.optlaseng.2020.106133.

[21] Y. Song, J. Wen, D. Yu, Y. Liu, and X. Wen, 2014, "Reduction of vibration and noise radiation of an underwater vehicle due to propeller forces using periodically layered isolators," *Journal of Sound and Vibration*, 333(14): 3031–3043, doi: https://doi.org/ 10.1016/j.jsv.2014.02.002.

[22] Y. Song, J. Wen, D. Yu, and X. Wen, 2015, "Suppression of vibration and noise radiation in a flexible floating raft system using periodic structures," *JVC/Journal of Vibration Control*, 21(2): 217–228, doi: 10.1177/1077546313488156.

[23] L. J. Wu and H. W. Song, 2019. "Band gap analysis of periodic structures based on cell experimental frequency response functions (FRFs," *Acta Mechanica Sinica. Xuebao*, 35(1): 156–173,

[24] S. S. Mester and H. Benaroya, 1995, "Periodic and near-periodic structures," *Shock Vibration*, 2(1): 69–95, doi: 10.3233/SAV-1995-2107.

[25] A. Z. Saydam, S. Gokcay, and M. Insel, 2018. "CFD based vortex generator design and full-scale testing for wake non-uniformity reduction," *Ocean Engineering,*, 153: 282–296,

[26] R. Bontempo, M. Cardone, and M. Manna, 2016 "Performance analysis of ducted marine propellers. Part I-Decelerating duct," *Applied Ocean Research*, 58: 322–330, , doi: 10.1016/j.apor.2015.10.005.

[27] R. Bontempo and M. Manna, 2016, "Performance analysis of ducted marine propellers. Part II – Accelerating duct," *Applied Ocean Research,* 75: 153–164, doi: 10.1016/j.apor.2018.03.005.

[28] F. Çelik, 2007, "A numerical study for effectiveness of a wake equalizing duct," *Ocean Engineering,* 34(16): 2138–2145 doi: 10.1016/j.oceaneng.2007.04.006.

[29] G. Martinas, 2015, "Cavitation of a Propeller and Influence of an Wake Equalizing Duct," *TransNav, Intertanional Journal on Marine Navigation and Safety of Sea Transportation*, 9(2): 235–241, doi: 10.12716/1001.09.02.11.

[30] H.-J. Heinke and K. Hellwig-Rieck. 2011., "Investigation of scale effects on ships with a wake equalising duct or with vortex generator fins,"

[31] [H. Huang, Y. Long, and B. Ji, 2020, "Experimental investigation of vortex generator influences on propeller cavitation and hull pressure fluctuations," *Journal of Hydrodynamic*, 32(1): 82–92, doi: 10.1007/s42241-020-0005-5.

[32] A. P. Mouritz, E. Gellert, P. Burchill, and K. Challis, 2001. "Review of advanced composite structures for naval ships and submarines," *Composite Structure*, 53: 21–41,

[33] G. F. Wilhelmi, W. M. Appleman, and F. T. C. Loo, 1986. "Composite Shafting for Naval Propulsion Systems," Wiley Online Library, 129–136

[34] D. Xu, B. Han, W. He, and Z. Cheng, 2016. "Review of advances on longitudinal vibration of submarine propulsion shafting & its vibration reduction technology," *Vibroengineering Procedia*, 10: 52–57

[35] Y. K. Ashkenazi, I. B. Gol'fman, L. P. Rezhkov, and N. P. Sidorov, 1974.Glass-fiber-reinforced plastic parts in ship machinery. Leningard: Sudostroyeniye-Publishing House

[36] P. G. Dylejko, N. J. Kessissoglou, and C. J. Norwood, 2007. "Optimisation of a resonance changer to minimise the vibration transmission in marine vessels," *Journal of Sound and Vibration*. 300(1–2): 101–116

[37] S. Merz, N. Kessissoglou, R. Kinns, and S. Marburg, 2009. "Optimisation of a submarine's resonance changer using the method of moving asymptotes," Proceedings of ACOUSTICS, 1–7

[38] S. Herold and D. Mayer, 2016, "Adaptive Piezoelectric Absorber for Active Vibration Control," actuators, 5(7): 1–13, doi: 10.3390/act5010007.

[39] M. K. Kwak and D. Yang, 2013. "Active vibration control of a ring-stiffened cylindrical shell in contact with unbounded external fluid and subjected to harmonic disturbance by piezoelectric sensor and actuator," *Journal of Sound and Vibration*, 1–23

[40] A. Loghmani, M. Danesh, and M. Keshmiri, 2015. "Theoretical and Experimental Study of Active Vibration Control of a Cylindrical Shell Using Piezoelectric Disks," *Journal of Low Frequency Noise, Vibration and Active Control*, 34(3): 269–287

[41] D. Lu, C. Li, J. Wang, and J. Fang, 2020. "Coupled vibration of piezoelectric- based propeller blades for a novel unmanned

underwater vehicle," Advances in Mechanical Engineering, 12(4): 1–14

[42] L. A. Sievers and A. H. Flotow, 1992. "Comparison and Extensions of Control Methods for Narrow-Band Disturbance Rejection," *IEEE Transactions on Signal Processing*, 40(10): 282–296,

[43] M. Lovera, P. Colaneri, and R. Celi, 2003. "Periodic analysis of Higher Harmonic Control techniques for helicopter vibration attenuation," in *Proceedings of the American Control Conference*, 999–1004

[44] Z. Yang, C. Qin, Z. Rao, N. Ta, and X. Gong, 2014. "Design and analyses of axial semi-active dynamic vibration absorbers based on magnetorheological elastomers," J*ournal of Intelligent Material Systems and Structures*, 25(17): 2199–2207

[45] M. H. Holdhusen, 2005. The state-switched absorber used for vibration control of continuous systems. Atlanta, GA: Georgia Institute of Technology

[46] H. Deng and X. Gong, 2007. "Adaptive tuned vibration absorber based on magnetorheological elastomer," *Journal of Intelligent Material Systems and Structures*, 18: 1205–1210

[47] F. Chen, Y. Chen, and H. Hua, 2018, "Vibration analysis of a submarine elastic propeller-shaft-hull system using FRF-based substructuring method," *Journal of Sound and Vibration,* 68: 264–278, [Online]. Available: https://doi.org/10.1016/j.jsv.2018.11.053.

[48] Q. Huang, X. Yan, Y. Wang, C. Zhang, and Z. Wang, 2017, "Numerical modeling and experimental analysis on coupled torsional-longitudinal vibrations of a ship's propeller shaft," Ocean Engineering, 136: 272–282 [Online] doi: 10.1016/j.oc eaneng.2017.03.017.