PRELIMINARY MODELLING OF A MARINE PROPULSION SYSTEM

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Abstract

In order to transmit thrust effectively a marine propulsion system is made up of mechanically stiff components. Consequently, unsteady propeller loads are also effectively transmitted to the hull structure. At low speeds, these result in tonal oscillations at the blade-passing frequency and its harmonics. At high speeds, broadband vibration is generated by cavitation and turbulence ingestion. The relative magnitudes of the axial, tangential and radial unsteady forces acting on the propeller are dependent on the non-uniform hull wake and propeller design. Much of the previous research has focussed on studying the effect of these components independently. This paper theoretically investigates the three-dimensional vibrational behaviour of the propulsion system to gain insight into the relative importance of the different excitation mechanisms and transmission paths. The propeller blade dynamics are included by considering an equivalent beam. This work provides the foundations for further efforts, which will investigate the vibration attenuation through the propulsion system of marine vessels.

1. Introduction

Marine vessels can suffer from large levels of vibration and radiated noise that negatively influence passenger comfort, crew fatigue and marine wildlife. The propeller is an important source of both broadband and tonal noise. The tonal components, which are at the blade-passing frequency (BPF) and its harmonics, are associated with the propeller rotating through a non-uniform wake generated by the hull. Turbulence ingestion and cavitation generate the broadband noise and vibration which is most problematic at higher speeds [1].

Although mathematical modelling of vibration transmission through marine propulsion systems has been undertaken in the past [2-8], most publicly available studies have focused on reducing the axial component of vibration. Dylejko et al. for example, [3-10] developed a transmission matrix model to investigate the attenuation of vibration transmission through the propulsion system using a resonance changer. Although axial vibration transmission is important, transverse vibration is also of concern [1,12] and should be included in a full analysis. The propeller is also often simplified by considering an equivalent rigid mass. This simplification ignores the propeller flexure and resulting blade resonances [1, 11-13]. Other complicating factors which tend to be neglected include nonlinearity and coupling between the different degrees-of-freedom. It should also be acknowledged that the lack of publically available experimental data makes validating these theoretical models difficult. The force produced by a marine propeller is not purely axial, but contains components in other directions due to drag and uneven loading. In this study, the immittance method is used to predict the three-dimensional vibration of a candidate marine propulsion system, which consists of a propeller, shaft, journal bearing and thrust bearing. The influence of the propeller compliance is evaluated by considering both a flexible and rigid propeller and the results discussed. The flexible propeller blade is modelled as a cantilever beam coupled to the end of the propulsion shaft.

2. Mathematical Model

The mathematical model is developed using the immittance method that allows for separating a complex system, such as the propulsion system, into smaller modular systems that are easier to define. A diagram of the system is shown in Figure 1(a). The system is made up of the propeller, two shaft elements, a journal bearing and a thrust bearing. A schematic of the simplified system used in previous studies for axial transmission (see Dylejko [3]) is shown in Figure 1(b). The proposed model is represented by the schematic in Figure 1(c).

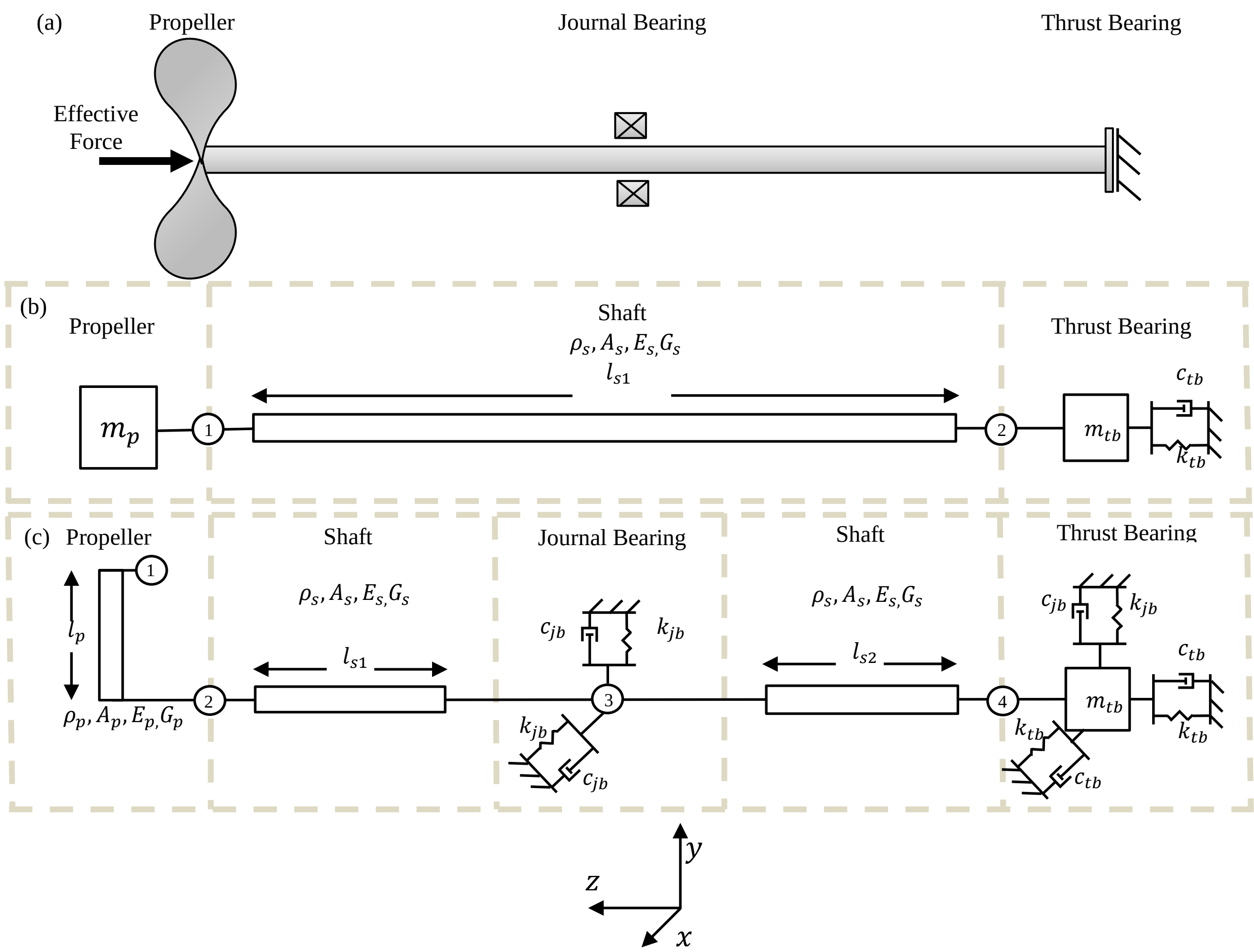

Figure 1. Top: (a) Diagram of system. Middle: (b) Schematic of propeller shafting system of previous studies. Bottom: (c) Schematic of propeller shafting system in this study and its coordinate system

The global coordinate system is defined with x in the lateral direction, y in the vertical direction and the z acting in the axial direction of the shaft. The translational velocities, in the direction designated by the subscript, are defined as $\dot{u}_x, \dot{u}_y, \dot{u}_z$ and the rotational velocities are described by

$\dot{\theta}_x, \dot{\theta}_y$ and $\dot{\theta}_z$. The same coordinate system is used for the individual elements. The following simplifications and assumptions are made:

1. Euler–Bernoulli beam theory is used to model beam elements.
2. The profile of the propeller can be represented by a rectangular cross section.
3. Linear stiffness and damping coefficients are used to represent the bearings.
4. The global coordinate system rotates with the shaft.
5. The hull and foundation have a sufficiently high impedance such that they can be treated as rigid.
6. The rotational resistance in the bearings is ignored.

This study examines the low frequency behaviour of the propulsion system where the axial wavelength is much larger than the radius of the shaft. For this reason, the authors believe that Euler-Bernoulli beam theory provides a reasonable approximation. It is acknowledged, however, that at higher frequencies, shear and rotary effects are important and must be considered. Although the vibratory response of a propeller blade will be different to that of a uniform beam, this simplification allows for an easy assessment of the importance of the propeller flexibility on the propeller/shafting interaction. Also, it is known that the bearing oil film stiffness is highly non-linear under realistic propeller loads, linear dynamic characteristics are a reasonable approximation for a specific operating condition. The authors believe that the simplifications made in this work are justified in the context that this investigation is primarily concerned with broadly evaluating the importance of propeller compliance on vibration transmission through the propulsion system.

The subscripts p, s, jb and tb denote the propeller, shaft, journal bearing and thrust bearing respectively. l is the length, ρ is the density, A is the cross sectional area, E is the Young's Modulus, G is the bulk modulus, m is the mass of the element, c is the damping and k is the stiffness of the element respectively. Locations 1, 2, 3 and 4 are the terminals for each element. The parameters used in this study are shown in Table 1. Although some parameters are taken from Dylejko's study [3], other parameters are assumed. The system impedance matrix $\mathbf{Z_{sys}}$ can be assembled by using continuity and equilibrium conditions across the terminals between the elements. The system impedance matrix is given by

$$\mathbf{Z_{sys}} = \begin{bmatrix} Z_{p11} & Z_{p12} & 0 & 0 \\ Z_{p21} & Z_{p22} + Z_{s22} & Z_{s23} & 0 \\ 0 & Z_{s32} & Z_{s33} + Z_{jb33} + Z_{s33} & Z_{s34} \\ 0 & 0 & Z_{s43} & Z_{s44} + Z_{tb} \end{bmatrix} \tag{1}$$

The subscripts 11, 12, 21 and 22 represent either, the drive point impedance, or, the transfer impedance. The individual elements that make up the $\mathbf{Z_{sys}}$ matrix are six-by-six matrices that represent each degree of freedom at terminals 1 to 4 and are discussed in detail later. The state and force vectors for each terminal are

$$\begin{aligned} \mathbf{v_i}^T &= [\dot{u}_x \quad \dot{u}_y \quad \dot{u}_z \quad \dot{\theta}_x \quad \dot{\theta}_y \quad \dot{\theta}_z] \\ \mathbf{F_i^T} &= [F_x \quad F_y \quad F_z \quad H_x \quad H_y \quad H_z] \end{aligned} \tag{2}$$

where F and H are the forces and moments in the directions corresponding to the subscript, and i denotes the terminal number. The relationship between the force and the velocity using the impedance of the system is given by

$$\mathbf{F} = \mathbf{Z_{sys}} \mathbf{v} \tag{3}$$

where **F** is the force vector containing the forces at terminals 1, 2, 3 and 4, **v** is the velocity vector containing the velocities at terminals 1, 2, 3 and 4.

Table 1. Table of model constants

Parameter	Value	Source	Parameter	Value	Source
m_p	1755 kg	[2]	A_s	0.0707 m^2	This study
I_{pxx}	1258 kg·m^2	This study	I_{szz}	0.000795 m^4	This study
I_{pyy}	1258 kg·m^2	This study	$I_{sxx} = I_{syy}$	0.000398 m^4	This study
I_{pzz}	2450 kg·m^2	This study	$l_{s1} = l_{s2}$	5.25 m	[2]
l_p	2 m	This study	c_j	300 tonnes/s	[2]
A_p	0.1125 m^2	This study	k_j	200 kN/m	[2]
I_{pzz}(Vertical Blade)	0.0013 m^4	This study	m_t	200 kg	[2]
I_{pxx} (Vertical Blade)	0.000843 m^4	This study	c_t	300 tonnes/s	[2]
I_{pyy}(Vertical Blade)	0.0022 m^4	This study	k_t	20000 MN/m	[2]
$E_s = E_p$	200 GPa	[2]	I_{txx}	13 kg·m^2	This study
$G_s = G_p$	79.3 GPa	[2]	I_{tyy}	13 kg·m^2	This study
$\rho_s = \rho_p$	7800 kg/m^3	[2]	I_{tzz}	25 kg·m^2	This study

The propeller is modelled as either a lumped mass or as a continuous beam. The beam model is described by the same receptance element used by Bishop and Johnson [14], which is based on Euler–Bernoulli beam theory where shear deformation and rotary effects are ignored. This element is a continuous 3D beam that considers transverse, longitudinal and torsional motion. It should be noted that the propeller blade is modelled as a continuous beam of rectangular cross section that is unchanged along the length of the beam. The aerofoil cross section, the varying chord length and the swept characteristics of the propeller are not considered for the sake of simplicity. Also, as this study is a preliminary investigation, a single propeller blade is modelled to ascertain its influence on the vibration of the system. The impedance matrix can be determined using the following relationship, noting that the inverse is a matrix inverse

$$\mathbf{Z_p} = \frac{\boldsymbol{\alpha}_\mathbf{p}^{-1}}{j\omega} \tag{4}$$

where $\boldsymbol{\alpha}_\mathbf{p}$ is the receptance matrix of the Euler–Bernoulli beam [14]. This element provides six degrees of freedom at either end of the beam. The lumped mass propeller is modelled using the following impedance matrix

$$\mathbf{Z_{p\,lumped}} = j\omega \cdot \mathrm{diag}\left[m_p, m_p, m_p, I_{pxx}, I_{pyy}, I_{pzz}\right]. \tag{5}$$

A beam is used for the shaft. To conserve the element's local coordinate system with the global coordinate system, the beam element is rotated such that the axial length of the element aligns with the z direction. Two shaft elements are used to represent the shaft before and after the journal bearing, this is shown in Figure 1.

The journal and thrust bearings are critical elements in marine propulsion systems. For this investigation, it is assumed that both bearings are linear and may be characterised by damping and stiffness coefficients c_{jb} and k_{jb} for the journal bearing and c_{tb} and k_{tb} for the thrust bearing respectively. The journal and thrust bearings are approximated as lumped parameter systems [15], subsequently, the impedance matrices can be written as

$$\mathbf{Z_{jb}} = \text{diag}\left[\frac{k_{jb}}{j\omega} + c_{jb}, \frac{k_{jb}}{j\omega} + c_{jb}, 0,0,0,0\right] \tag{6}$$

$$\mathbf{Z_{tb}} = \text{diag}\left[j\omega m_{tb} + \frac{k_{jb}}{j\omega} + c_{jb}, j\omega m_{tb} + \frac{k_{jb}}{j\omega} + c_{jb}, j\omega m_{tb} + \frac{k_{tb}}{j\omega} + c_{tb}, j\omega I_{tbxx}, j\omega I_{tbyy}, j\omega I_{tbzz}\right] \tag{7}$$

where m_{tb} is the mass of the thrust bearing and I_{tb} is the mass moments of inertia of the bearing in the corresponding directions. The supporting structure for the bearings is assumed rigid due to the relatively high impedance of the foundation and hull structure. It should be noted that both bearings are modelled as rigidly terminated elements and hence only require a total of six degrees of freedom rather than twelve. The inclusion of the mass and inertia for the thrust bearing is to account for the collar that is attached to the shaft that supports the load in the z direction. There is no such mass or inertia concerning the journal bearing as no collar is attached to the shaft.

3. Results and Discussion

Figure 2 shows the axial force transmission through the propeller-shafting system for a unit propeller hub axial force and rigid propeller. Although not shown, the predicted value is identical to that predicted using the simplified model presented by Dylejko [3].

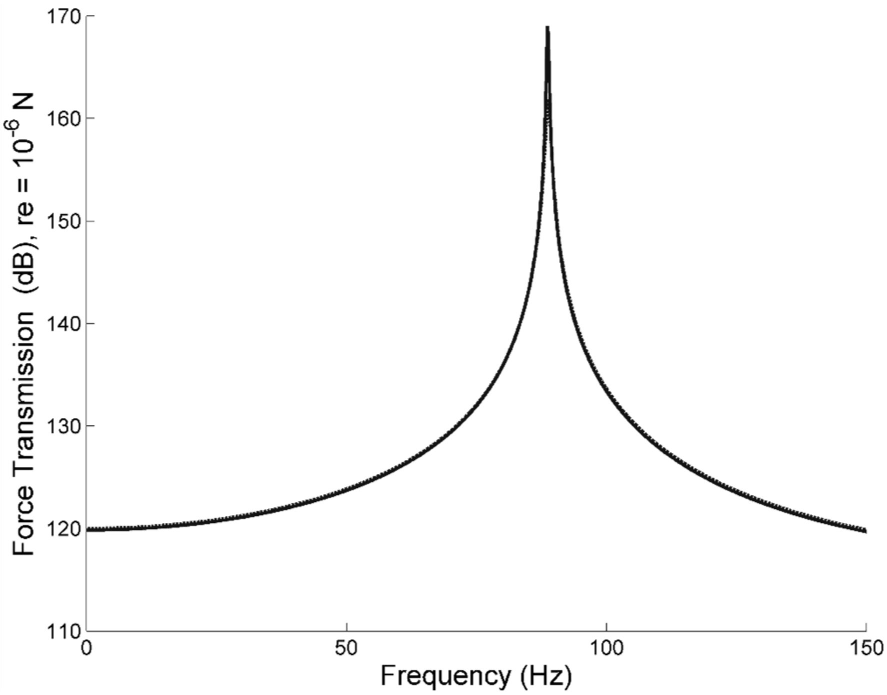

Figure 2. Axial force transmission with a rigid propeller

The next set of results is produced with the flexible propeller model with a unit excitation force/moment applied to all degrees-of-freedom at the propeller hub. Figure 3 (top and middle) shows the force transmission to the journal bearing and the thrust bearing, respectively, in the y direction for a frequency range of 0 to 150 Hz. The bottom plot shows the force transmission to the thrust bearing in the z direction. The difference in transmitted force predicted with the rigid propeller model and with the flexible propeller model demonstrates the potential significance of neglecting to take into account the propeller flexibility. The propeller dynamics influence the reaction forces in both axial and transverse directions. This is not surprising given that the transverse and rotational motion of the propeller directly coupled to the axial and transverse motion of the shaft. It can be seen that the rigid propeller model significantly under-predicts the transmitted force at the thrust bearing. The rigid body propeller model, however, predicts with reasonable accuracy, the journal bearing transmissibility. This implies that the propeller resonances have the least impact on the journal bearing reaction forces when compared with the thrust bearing reaction forces, for this particular system.

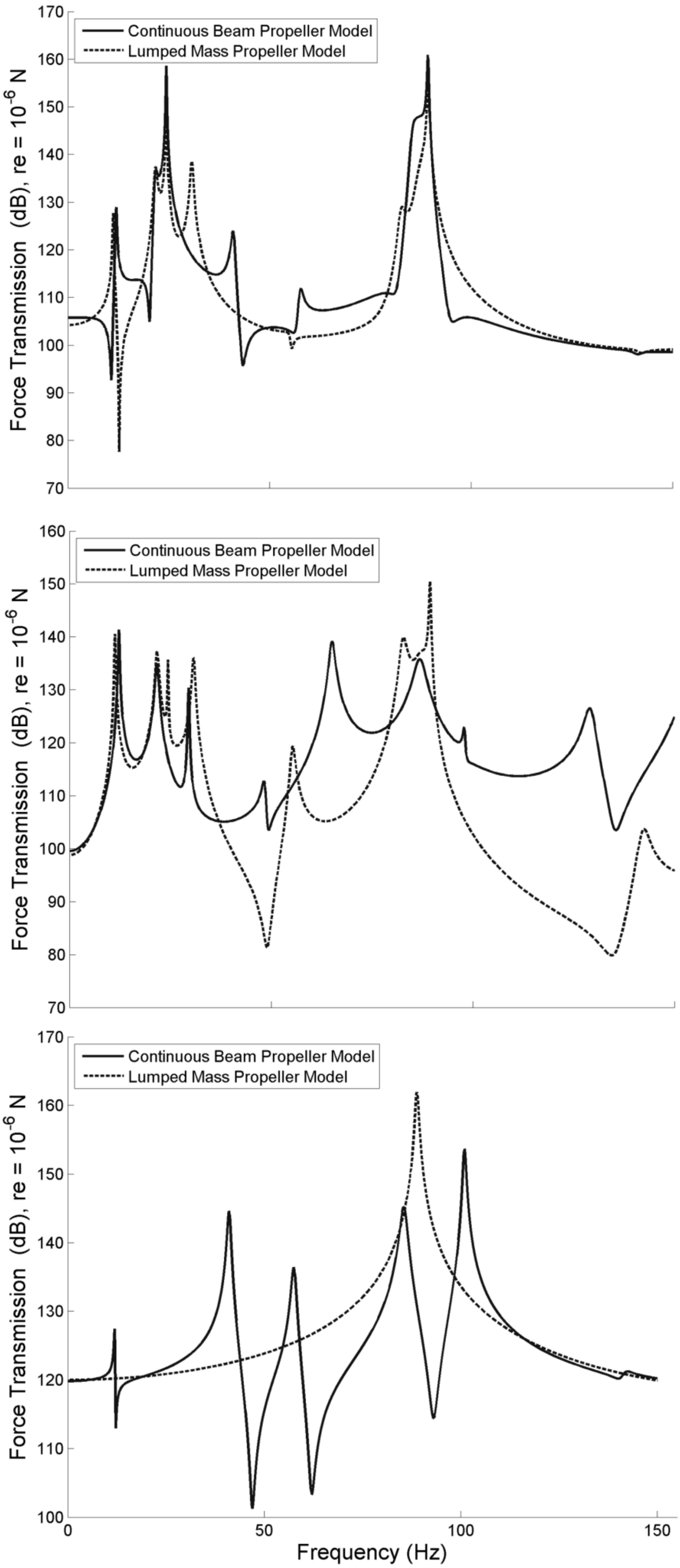

Figure 3. Force transmission through the journal bearing (top), thrust bearing (middle) in the y directions and force transmission through the thrust bearing in the *z* direction (bottom)

To gain a better understanding of the motion of the propulsion system with a flexible propeller blade, the mode shapes of the system are shown in Figure 4. Due to the lack of terminal points, the resolution is poor, but useful information can still be obtained from the figure. The amplitudes of the mode shapes are normalized to the tip of the propeller blade so that the relative magnitudes of the terminal displacements with propeller motion can be easily identified. Modes 1-3 demonstrate significant propeller tip motion indicating that these resonances are associated with the propeller. The fourth mode at 85 Hz is consistent with a rigid body propeller mode as there is very little difference between the motion of the propeller tip and hub. This frequency also matches the first resonance frequency of the lumped mass model. The subsequent modes show that the propeller tip and the hub are out of phase further suggesting that these resonance frequencies are related to the flexibility of the propeller blade.

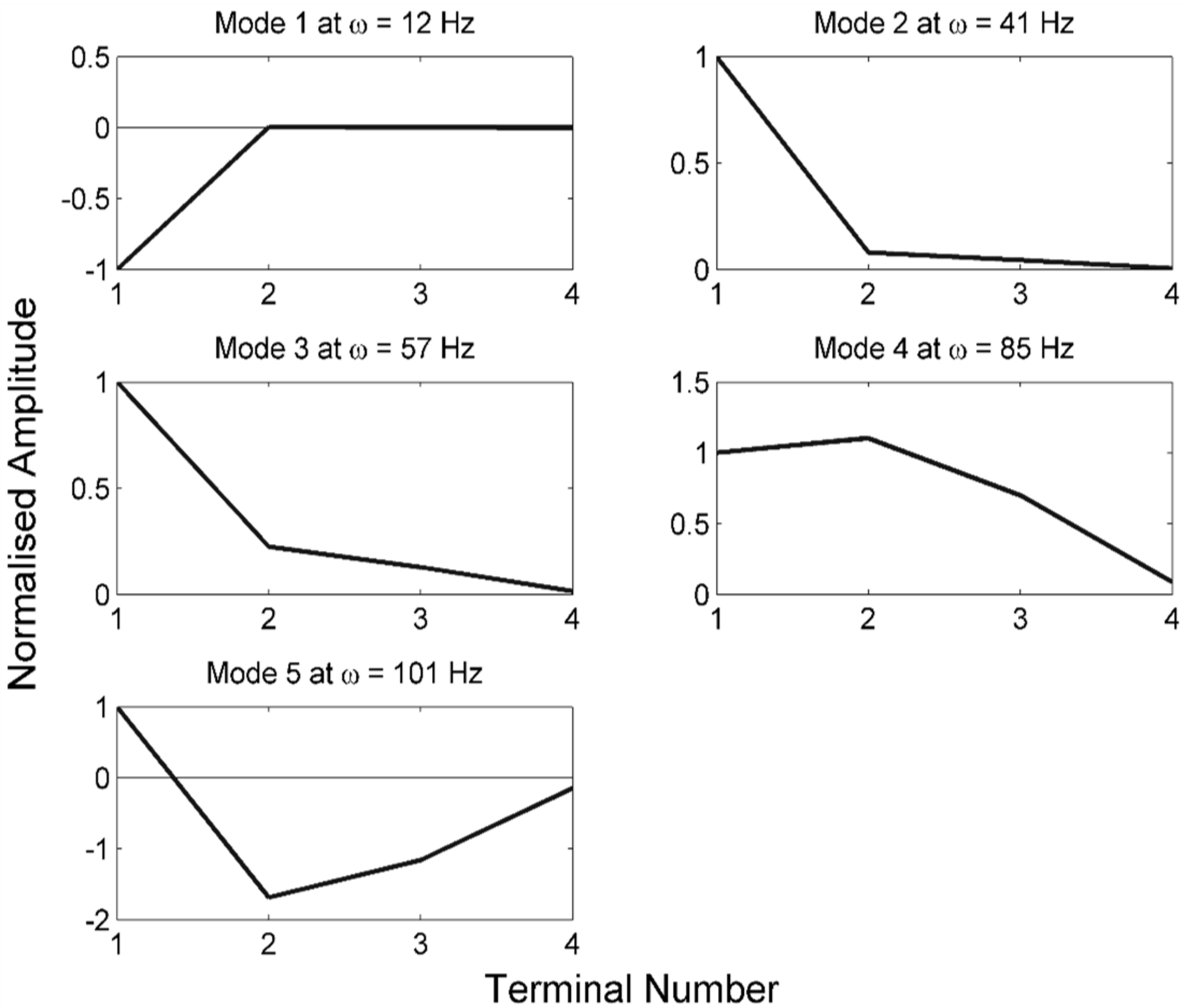

Figure 4. Axial vibration mode shapes

4. Conclusions

This paper examined a simplified three-dimensional model of a marine propulsion system using immittance methods. Propeller flexibility was included by modelling a propeller blade as a uniform cantilever beam. Although it was acknowledged that this is an over-simplification, the results still demonstrate that the compliance of the propeller can significantly contribute to the transmitted force through the propulsion system. This work provides the foundations for further efforts which will investigate the vibration attenuation through the propulsion system.

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