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REDUCTION OF THE RADIATED NOISE FROM A SUBMARINE UNDER EXCITATION FROM THE PROPELLER-SHAFTING SYSTEM USING A RESONANCE CHANGER

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Abstract

This paper examines the reduction of the low frequency acoustic signature of a submarine by using a resonance changer. The dynamic response of the propeller-shafting system has been modelled as a combination of lumped parameter and continuous parameter systems using the transmission matrix method. The submarine hull was modelled as a ring stiffened finite cylindrical shell submerged in a fluid undergoing axial excitation from the propeller-shafting system. The total sound pressure radiated into the far-field from the hull is obtained by using an approximate closed form solution to the Helmholtz integral equation. Optimal parameters for the resonance changer are obtained by minimising the maximum far-field radiated sound pressure using a genetic algorithm.

INTRODUCTION

The vibration transmission through the propeller-shafting system of a submarine represents a critical issue that must be addressed in order to reduce the low frequency acoustic signature of a submarine. Axial excitation of the propeller occurs at low frequencies due to the non-uniform wake velocity caused by asymmetry in the hull or protrusions of control surfaces. The oscillations which occur at the propeller are the result of small variations in thrust when the propeller blades rotate through the non-uniform wake. The frequency of these oscillations is at the blade passing frequency (rotational speed of the shaft multiplied by the number of blades on the propeller). Development of propeller-shafting models for maritime vessels has been undertaken by numerous researchers [1-3]. In most of these studies, the aim has been to reduce the axial vibration and its transmission into the hull. A detailed paper by Goodwin [2] examined the reduction of excessive vibration through the propeller-shafting system by using a hydraulic device called the "Michel Thrustmeter" or resonance changer (RC). This device is located in series between the thrust bearing and supporting foundation, and is used to measure the thrust which is transmitted to the vessel from the propeller-shafting system. The RC introduces virtual elastic, damping and inertial influences by hydraulic means, thereby acting as a dynamic vibration absorber. The simplified model of the RC introduced by

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Goodwin is shown in Fig. 1. The RC consists of a piston of cross sectional area A_0, an oil reservoir of volume V_1 and a pipe connecting these two elements of length L_1 and cross sectional area A_1. Reduction of the force transmissibility was achieved by firstly tuning the natural frequency of the resonance changer to that of the propeller-shafting system's natural frequency and then optimising the RC's damping rate.

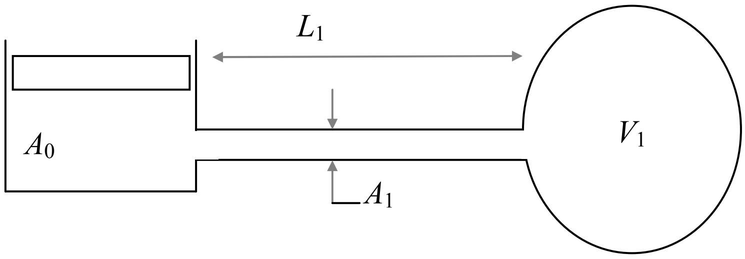

Figure 1. Simplified model of the resonance changer.

In this paper, the transmission matrix method, otherwise known as the four-pole parameter method, is used to model the dynamic response of the propeller-shafting system in a submarine. The submarine hull is modelled as a ring-stiffened finite cylindrical shell submerged in a fluid and undergoing axial excitation from the propeller-shafting system. Two cost functions associated with the vibration transmission to the hull over a low frequency range as a function of the RC parameters are developed. Both these costs functions are minimised using a genetic and general non-linear constrained algorithm within realistic constraints, resulting in optimal values for the virtual RC parameters.

TRANSMISSION MATRIX MODEL OF THE PROPELLER-SHAFTING SYSTEM

A transmission matrix schematic of the propeller-shafting system is given in Fig. 2. The proposed dynamic model assumes that the propeller and the entrained water around the propeller are represented as a lumped mass of mass m_p with viscous damping c_p. The propeller is attached to a continuous model of the shaft consisting of cross sectional area A_s, Young's modulus E_s and density ρ_s. Since the response at a point along the shaft corresponding to the location of the thrust bearing is desired, an effective length l_{se} is defined. The thrust bearing is represented by a linear stiffness k_b, damping coefficient c_b and mass m_b.

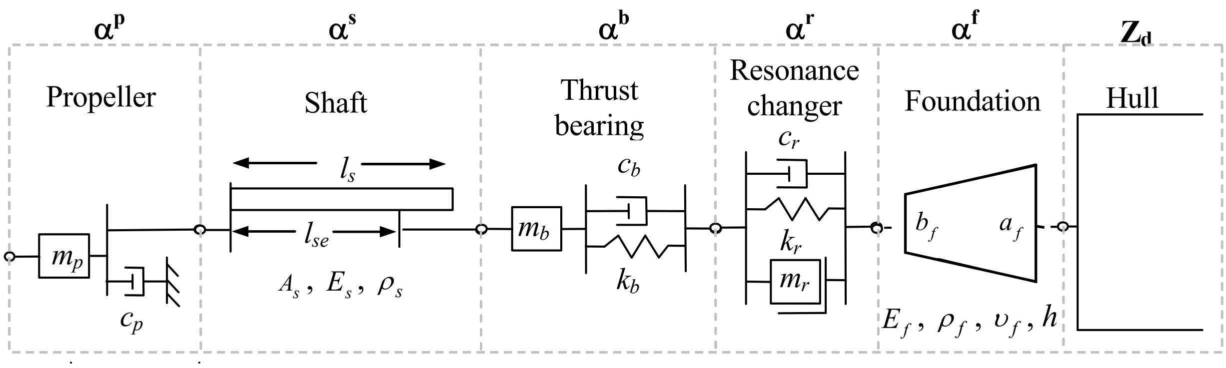

Figure 2. Transmission matrix model of the propeller shafting system connected to the submarine hull.

The RC exhibits inertial, elastic and damping properties, represented by m_r , k_r and c_r respectively. Referring to Fig. 1, these virtual mass, stiffness and damping parameters can be expressed by [2]:

$$m_r = \frac{\rho_1 A_0^{\,2} L_1}{A_1}, \qquad k_r = \frac{A_0^{\,2} B_1}{V_1}, \qquad c_r = 8\pi\mu_1 L_1 \frac{A_0^{\,2}}{A_1^{\,2}} \tag{1-3}$$

The velocities of the propeller, shaft, thrust bearing, resonance changer, foundation and hull are described by v_p, v_s, v_b, v_r, v_f and v_h, respectively, while the corresponding forces are given by f_p, f_s, f_b, f_r, f_f and f_h. The forward transmission parameters of the propeller (ignoring the damping due to the surrounding fluid) and shaft are respectively given as [4]:

$$\boldsymbol{\alpha}^{\mathbf{p}} = \begin{bmatrix} 1 & j\omega m_p \\ 0 & 1 \end{bmatrix}, \quad \boldsymbol{\alpha}^{\mathbf{s}} = \begin{bmatrix} \cos k_s l_{se} & j\dfrac{A_s E_s k_s \sin k_s l_s}{\omega \cos k_s (l_s - l_{se})} \\ j\omega \dfrac{\cos k_s (l_s - l_{se}) - \cos k_s l_s \cos k_s l_{se}}{A_s E_s k_s \sin k_s l_s} & \dfrac{\cos k_s l_s}{\cos k_s (l_s - l_{se})} \end{bmatrix} \tag{4,5}$$

The shaft parameters were obtained by manipulating the receptance matrix for a free-free rod undergoing longitudinal vibration [6], where $k_s = \omega / c_{Ls}$ is the longitudinal wavenumber, and $c_{Ls} = \sqrt{E_s/\rho_s}$ is the longitudinal wave speed of the shaft. The forward transmission parameters of the thrust bearing and RC are respectively expressed as:

$$\boldsymbol{\alpha}^{\mathbf{b}} = \begin{bmatrix} 1 - \dfrac{\omega^2 m_b}{k_b + j\omega c_b} & j\omega m_b \\ \dfrac{j\omega}{k_b + j\omega c_b} & 1 \end{bmatrix}, \quad \boldsymbol{\alpha}^{\mathbf{r}} = \begin{bmatrix} 1 & 0 \\ \dfrac{j\omega}{k_r + j\omega c_r - \omega^2 m_r} & 1 \end{bmatrix}. \tag{6,7}$$

To model the foundation of the propeller-shafting system in a submarine, a simplified model of a truncated conical shell was used. It is assumed that the axisymmetric response of the foundation in the low frequency range can be approximated using membrane theory [7]. The parameters for the conical foundation are shown in Fig. 2, where a_f and b_f are the radii of the major and minor base of the conical shell respectively. The four-pole parameters were obtained by numerical integration of the second order equations of motion given by Hu and Kana [7].

DYNAMIC RESPONSE OF THE SUBMARINE HULL

Under axial excitation of the submarine hull from the propeller-shafting system, it is assumed that only breathing motion of the cylinder is excited, which gives rise to an axisymmetric case. For low frequency analysis of the axisymmetric motion of the cylinder, modifications are made to the equations of motion, whereby an equivalent distributed mass of the shell is included to account for the mass of the internal structure and on-board equipment, orthotropic shell properties are used to account for the effects of the ring stiffeners, and an increase in inertia on the shell is included to account for the fluid loading effects. With these assumptions, the modified equations of motion based on the Donnell-Mushtari theory are given by [8,9]:

$$\frac{\partial^2 u}{\partial x^2} - \left(1 + \frac{A_r}{bh} + \frac{m_{eq}}{\rho_h h}\right)\frac{1}{c_L^2}\frac{\partial^2 u}{\partial t^2} + \frac{\upsilon_h}{R}\frac{\partial w}{\partial x} = 0 \tag{8}$$

$$\frac{\upsilon_h}{R}\frac{\partial u}{\partial x} + \left(1 + \frac{A_r(1-\upsilon_h^2)}{bh}\right)\frac{w}{R^2} + \beta^2 R^2 \frac{\partial^4 w}{\partial x^4} + \left(1 + \frac{A_r}{bh} + \frac{m_{fl}}{\rho_h h}\right)\frac{1}{c_L^2}\frac{\partial^2 w}{\partial t^2} = 0\,. \tag{9}$$

R is the shell mean radius, h is the shell thickness, and $\beta^2 = h^2/12R^2$ is the thickness parameter. $c_L = \sqrt{E_h/\rho_h(1-\upsilon_h^2)}$ is the longitudinal wave speed, where E_h, ρ_h and υ_h are the Young's modulus, density and Poisson's ratio respectively. A_r is the cross sectional area of the stiffeners and b is the stiffener spacing. m_{eq} represents the equivalent distributed mass of the internal structure and on-board equipment. The fluid loading parameter m_{fl} can be derived using a standing wave configuration of an infinite cylinder by [9]:

$$m_{fl} = \frac{\rho_{fl}\mathrm{H}_0^{(1)}[(k_{fl}^2 - k_1^2)^{1/2}R]}{(k_{fl}^2 - k_1^2)^{1/2}\mathrm{H}_1^{(1)}[(k_{fl}^2 - k_1^2)^{1/2}R]} \tag{10}$$

where $\mathrm{H}_n^{(1)}$ is the Hankel function of the first kind of order n. k_{fl} and ρ_{fl} are respectively the wavenumber and density of the fluid. k_1 is an axial wavenumber of the shell for travelling, non-evanescent waves. General solutions of the axial and radial displacements for harmonic motion are of the form:

$$u(x,t) = Ue^{jkx - j\omega t}\,, \qquad w(x,t) = We^{jkx - j\omega t} \tag{11,12}$$

Substitution of the general solutions into the equations of motion and taking the determinant of the coefficient matrix yields the following characteristic equation:

$$\beta^2(kR)^6 + \gamma\Omega^2\beta^2(kR)^4 + (\tau - \mu\Omega^2 - \upsilon_h^2)(kR)^2 + \gamma\tau\Omega^2 - \gamma\mu\Omega^4 = 0\,. \tag{13}$$

where $\Omega = \omega R/c_L$ is the non-dimensional frequency, and

$$\tau = 1 + \frac{A_r(1-\upsilon_h^2)}{bh}\,, \qquad \gamma = 1 + \frac{A_r}{bh} + \frac{m_{eq}}{\rho_h h}\,, \qquad \mu = 1 + \frac{A_r}{bh} + \frac{m_{fl}}{\rho_h h}\,. \tag{14-16}$$

The characteristic equation is a third order dispersion equation in terms of k^2. In the absence of torsional motion, the three axial wavenumbers for wave motion in the positive and negative directions correspond to a propagating wave (one real solution) and two attenuated standing waves (two solutions which are complex conjugated). The axial to radial amplitude ratio C_i can be obtained for each axial wavenumber k_i ($i \in \{1, 2, ..., 6\}$):

$$C_i = \frac{U_i}{W_i} = \frac{j\upsilon_h k_i R}{(k_i R)^2 - \gamma \Omega^2}, \qquad i \in \{1, 2, ..., 6\} \tag{17}$$

For harmonic motion (where the time dependent term $e^{-j\omega t}$ has been omitted in the proceeding equations), the complete solution of the cylindrical shell is given by the following expressions, where boundary conditions at the end closures are used to determine the axial and radial amplitudes:

$$u(x) = \sum_{i=1}^{6} C_i W_i e^{jk_i x}, \qquad w(x) = \sum_{i=1}^{6} W_i e^{jk_i x}. \tag{18,19}$$

The driving point impedance of the cylindrical shell is then given by:

$$Z_d = \frac{1}{j\omega u(x)}. \tag{20}$$

The approach to the analytical solution of the acoustic pressure field generated by a finite cylindrical shell is based on the Helmholtz integral equation. The motions of the cylinder that contribute to the radiated sound pressure consist of the rigid body motion of the end plates in the axial direction, and radial motion of the cylindrical surface. In the proceeding analysis, it is assumed that the interactions between the radiating surfaces may be ignored to enable an approximate closed form solution. For the finite cylinder, the total surface area consists of three components corresponding to the two end plates and the cylindrical shell. It is assumed that under the condition of an axial excitation, the radiating pressure field is due mainly to the axial movement at the ends of the cylinder. This allows the Helmholtz integral equation to be simplified by considering the three areas separately in the analysis [11]. Expressions for the radiated pressure from the three surface areas can then be obtained.

DEVELOPMENT OF COST FUNCTIONS

The combined response of the complete propeller-shafting system $\boldsymbol{\beta}^{\mathbf{ps}}$ is given by the matrix multiplication of the respective forward transmission matrix parameters of the subsystems:

$$\boldsymbol{\beta}^{\mathbf{ps}} = \boldsymbol{\alpha}^{\mathbf{p}}\boldsymbol{\alpha}^{\mathbf{s}}\boldsymbol{\alpha}^{\mathbf{b}}\,\boldsymbol{\alpha}^{\mathbf{r}}\,\boldsymbol{\alpha}^{\mathbf{f}}. \tag{21}$$

The magnitude of the force at the hull resulting from a unit load at the propeller ($f_p = 1\text{ N}$) is defined by [4]:

$$f_h = \left| \beta_{11}^{\mathrm{ps}} + \frac{\beta_{12}^{\mathrm{ps}}}{Z_d} \right|^{-1}. \tag{22}$$

β_{11}^{ps} and β_{12}^{ps} represent the first and second elements in the first row of the matrix $\boldsymbol{\beta}^{\mathbf{ps}}$, as given by Eq. (21). The maximum far-field radiated pressure at a given radius from the cylinder for a unit axial force as a function of frequency can be represented by an acoustic response function, $p_{h,\max}(\omega)$.

The force which acts on the propeller in a marine vessel has been shown to be approximately proportional to the propeller rotational speed squared [1,2]. This relationship can be accounted for in the cost function to be minimised, by weighting the force transmissibility through the propeller-shafting system by the square of the frequency ratio $\omega_i/\Delta\omega$, where ω_i is the discrete frequency in the frequency band of interest and $\Delta\omega$ is the frequency bandwidth used in the optimisation process. The weighted transmitted force at the i^{th} discrete frequency can be expressed as:

$$f_{h,W}(\omega_i) = \left(\frac{\omega_i}{\Delta\omega}\right)^2 f_h(\omega_i) \tag{23}$$

The first cost function to be minimised is the weighted force transmissibility through the propeller-shafting system, and is given by:

$$J_{force}(\mathbf{x}) = \log_{10}\left\{ \max_{\omega_l \le \omega_i \le \omega_u} f_{h,W}(\mathbf{x},\omega_i) \right\}. \tag{24}$$

The second cost function to be minimised is the maximum far-field radiated pressure scaled by the weighted force transmissibility through the propeller-shafting system:

$$J_{acoustic}(\mathbf{x}) = \log_{10}\left\{ \max_{\omega_l \le \omega_i \le \omega_u} f_{h,W}(\mathbf{x},\omega_i)\, p_{h,\max}(\omega_i) \right\} \tag{25}$$

$\mathbf{x}$ is a vector containing the virtual mass, stiffness and damping parameters associated with the RC, and is given by $\mathbf{x} = \{k_r \quad m_r \quad c_r\}^T$. Lower ($\mathbf{x_l}$) and upper ($\mathbf{x_u}$) limits are enforced on the RC parameters, that is, $\mathbf{x_l} \le \mathbf{x} \le \mathbf{x_u}$. The frequency range is also bound by lower (ω_l) and upper (ω_u) limits, that is $\omega_l \le \omega \le \omega_u$. Only the low frequency range (< 100 Hz) is of interest due to the excitation of the propeller occurring at the blade pass frequency. An optimisation scheme utilising a genetic and a general non-linear constrained algorithm has been used to minimise the cost functions defined in Eqs. (24) and (25). The genetic algorithm was used to approximately find the global optima, while the general non-linear constrained algorithm improved the accuracy of the approximate solution found by the GA.

RESULTS

The values of the propeller-shafting system used in the modelling are given in Table 1. The limits imposed on the RC parameters within the optimisation process are presented in Table 2. The submarine hull was modelled as a ring stiffened steel cylinder of 6.5 m diameter, 40 mm hull plate thickness, 45 m length, with two evenly spaced bulkheads. Internal structural damping was included in the analysis by using a structural loss factor of 0.02. The cylinder was submerged in water of density 1000 kg/m^3. A neutrally buoyant condition was maintained by applying an appropriate amount of distributed mass which represents the structural components and on board equipment.

Optimal values for the virtual mass, stiffness and damping RC parameters were obtained by optimising the cost functions given by Eqs. (24) and (25), corresponding to minimising the weighted force transmissibility through the propeller-shafting system (J_{force}), and minimising the maximum radiated pressure scaled by the weighted force transmissibility through the propeller-shafting system ($J_{acoustic}$). The optimal J_{force} and $J_{acoustic}$ RC parameter sets are given in Table 3. Figure 3 shows the far-field radiated pressure versus frequency in the absence of the RC, and using the optimal J_{force} and $J_{acoustic}$ RC parameter sets. The peaks at around 22, 46, and 73 Hz are due to excitation of hull axial resonances. The peak occurring at approximately 55 Hz corresponds to the fundamental propeller-shafting resonance. The small peaks at approximately 9, 37 and 81 Hz are caused by resonances of the bulkheads.

The introduction of the RC results in the elimination of the propeller-shafting resonance (at 55Hz). Since the RC is comparable to a dynamic vibration absorber, it introduces an additional resonance (at approximately 16 Hz), and causes a shift in the original resonances, as well as significantly lowering the overall response. Figure 3 shows that the introduction of an RC to the propeller-shafting system results in a significant reduction in the radiated acoustic signature. Directly minimising the acoustic response results in a better control performance than reducing the vibration transmission to the hull.

Table 1. Propeller-shafting system parameters.

Parameter	Value
m_p (tonnes)	10
E_s (GPa)	200
ρ_s (tonnes/m^3)	7.8
A_s (m^2)	0.707
L_s (m)	10.5
L_{se} (m)	9
m_b (tonnes)	0.2
k_b (MN/m)	20000
c_b (tonnes/s)	300

Table 2. Resonance changer limits.

RC parameter	Lower limit	Upper limit
k_r (MN/m)	15	1500
m_r (tonnes)	1	20
c_r (tonnes/s)	5	1100

Table 3. Optimal J_{force} and $J_{acoustic}$ RC parameter sets.

RC parameter	J_{force}	$J_{acoustic}$
k_r (MN/m)	169	206
m_r (tonnes)	1	1
c_r (tonnes/s)	287	70

CONCLUSIONS

The dynamic response of the propeller-shafting system in a submarine has been modelled as a combination of lumped parameter and continuous parameter systems. An acoustic frequency response function has been developed. Optimal resonance changer parameters have been obtained by minimising the frequency squared weighted maximum force transmissibility through the propeller-shafting system over a specified frequency range, and the maximum far-field radiated sound pressure. An optimisation scheme using a genetic and general non-linear constrained algorithm was applied to the cost functions. Realistic lower and upper bounds on the resonance changer parameters were applied as constraints within the optimisation process.

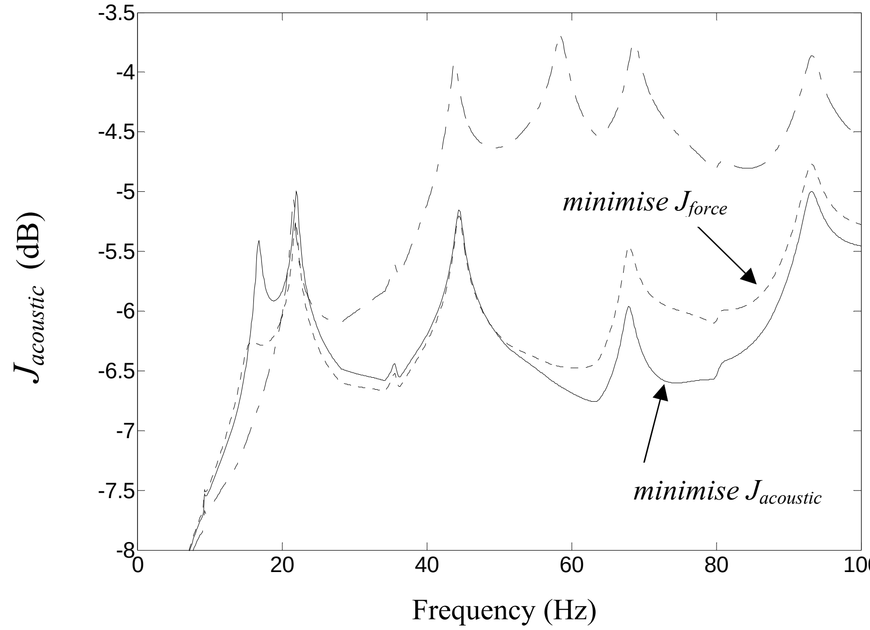

Figure 3. Radiated sound pressure (— – – no RC, - - - - minimise J_{force}, ——— minimise $J_{acoustic}$).

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