Reduction of the acoustic signature of a submerged vessel

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ABSTRACT

This paper examines the reduction of the low frequency acoustic signature of a submarine by optimal passive tuning of a resonance changer. The propeller-shafting system is modelled with a combination of lumped parameter and continuous parameter systems utilising the transmission matrix method. The submarine hull is 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

It is desirable to minimise the vibro-acoustic responses of maritime vessels to improve passenger comfort, minimise crew fatigue, and in the case of naval vessels, to avoid detection. Oscillations generated by onboard equipment such as the diesel engines and the propulsion system results in significant vibration that propagates through the supporting structure to the hull, where it is radiated as structure borne noise. The vibration transmission through the propeller-shafting system of a submarine represents a critical component that must be addressed in order to reduce the low frequency acoustic signature. The propeller-shafting system can be simplified into the key features shown in Fig. 1. The excitation of the propeller-shafting system is primarily influenced by axial excitation of the propeller due to the non-uniform wake velocity caused by asymmetry in the hull or protrusions of control surfaces. These perturbations are the result of small variations in thrust at the propeller when the 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). The disturbances at the propeller result in vibration transmission through the propeller-shafting system and subsequent axial excitation of the hull.

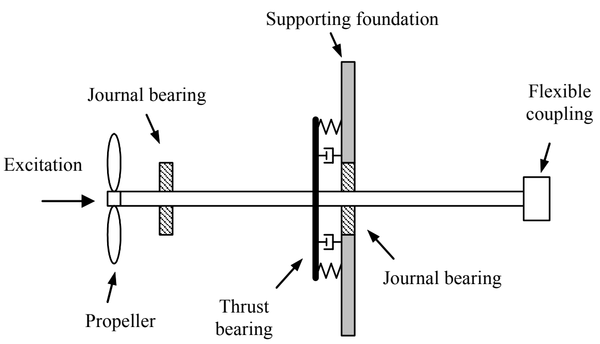

Figure 1. Simple schematic of the propeller-shafting system (Pan *et al.* 2002).

Development of propeller-shafting models for maritime vessels has been undertaken by numerous researchers (Goodwin 1960; Lewis *et al.* 1989; Pan *et al.* 2002). 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 (1960) examined the reduction of excessive vibration through the propeller-shafting system by using an existing hydraulic device called the "Michel Thrustmeter". This device was located in series between the thrust bearing and supporting foundation, and was used to measure the thrust which is transmitted to the vessel from the propeller-shafting system. Goodwin adapted and redesigned the Michel Thrustmeter to reduce the vibration transmission through the propeller-shafting system. For this new application, the device was known as a resonance changer (RC). The RC introduces virtual elastic, damping and inertial influences by hydraulic means, thereby acting as a dynamic vibration absorber.

In this paper, the transmission matrix method is used to model the dynamic response of the propeller-shafting system in a submarine. A submarine subject to excitation from its propeller/propulsion system is idealised as a finite cylinder under axial excitation. In this case, it is assumed that only the breathing mode corresponding to the zeroth circumferential mode is excited, which gives rise to an axisymmetric case. The submerged vessel is modelled as a ring-stiffened cylindrical shell with finite end closures, and separated by bulkheads into a number of compartments. The effect of various influencing factors corresponding to the ring stiffeners, boundary end conditions, bulkheads, and fluid-loading effects on the low frequency vibrational modes are included in the modelling. A cost function associated with the acoustic signiture of a submarine as a function of the RC parameters is developed. This cost function is minimised using a genetic algorithm within realistic constraints, resulting in optimal values for the virtual RC parameters.

TRANSMISSION MATRIX DESCRIPTION OF THE PROPELLER SHAFTING SYSTEM

Propeller-Shafting System

Due to the symmetry of the geometry and loading of the propeller-shafting system in a submarine, the transmission matrix method has been chosen to characterise the dynamic response (Bishop and Johnson 1960; Rubin 1966; Snowdon 1971). A transmission matrix description of the propeller-shafting system is shown in Figure 2. The mechanical components have been broken down into subsystems to enable a modular description of the complete system's dynamic response. The proposed dynamic model assumes the propeller and the entrained water around the propeller can be represented as a lumped mass of mass m_p with viscous damping c_p.

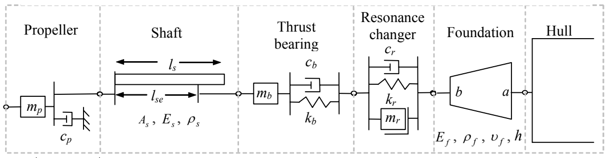

Figure 2. Modular representation of propeller-shafting model connected to the submarine hull.

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. The resonance changer exhibits inertial, elastic and damping properties, represented by m_r, k_r and c_r respectively. The thrust block is coupled to the hull via a truncated conical shell foundation.

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 forces at the previous locations are given by f_p, f_s, f_b, f_r, f_f and f_h.

The forward transmission parameters (also called the four-pole parameters) of the propeller (ignoring the damping due to the surrounding fluid) are given as (Snowdon 1971):

$$\boldsymbol{\alpha}^{\mathbf{p}} = \begin{bmatrix} 1 & j\omega m_p \\ 0 & 1 \end{bmatrix} \tag{1}$$

The shaft parameters were obtained by manipulating the receptance matrix for a free-free rod undergoing longitudinal vibration (Bishop and Johnson 1960), 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.

$$\boldsymbol{\alpha}^{\mathbf{s}} = \begin{bmatrix} \cos k_s l_{se} & j\dfrac{A_s E_s k_s \sin k_s l_s}{\omega \cos k_s (l_s - l_{se})} \\ j\omega \dfrac{\cos k_s (l_s - l_{se}) - \cos k_s l_s \cos k_s l_{se}}{A_s E_s k_s \sin k_s l_s} & \dfrac{\cos k_s l_s}{\cos k_s (l_s - l_{se})} \end{bmatrix} \tag{2}$$

The forward transmission parameters of the thrust bearing and RC are respectively expressed as:

$$\boldsymbol{\alpha}^{\mathbf{b}} = \begin{bmatrix} 1 - \dfrac{\omega^2 m_b}{k_b + j\omega c_b} & j\omega m_b \\ \dfrac{j\omega}{k_b + j\omega c_b} & 1 \end{bmatrix} \tag{3}$$

$$\boldsymbol{\alpha}^{\mathbf{r}} = \begin{bmatrix} 1 & 0 \\ \dfrac{j\omega}{k_r + j\omega c_r - \omega^2 m_r} & 1 \end{bmatrix}. \tag{4}$$

To model the foundation of the propeller-shafting system in a submarine, a simplified model of a truncated conical shell has been used. It is assumed that the axisymmetric response of the foundation in the low frequency range can be approximated using membrane theory (Hu and Kana 1968). The physical values used for the conical foundation are given in Table 1, where a and b are the radii of the major and minor base of the conical shell respectively, as shown in Figure 2. E_f is the Young's modulus of the conical shell, ρ_f is the density, υ_f is Poisson's ratio, and h is the shell thickness. ϕ is the semivertex angle of the conical shell.The four-pole parameters were obtained by numerical integration of the second order equations of motion, and are given by equation (5) and Figure 3. The inverse in equation (5) is applied since the geometry of the conical shell used by Hu and Kana (1968) was inverted compared to the geometry of the foundation shown in Figure 2.

Table 1. Foundation parameters.

Foundation parameter	Value
a (mm)	1250
b (mm)	520
ρ_f (kg/m^3)	7700
E_f (GPa)	200
υ_f	0.3
h (mm)	10
ϕ (deg)	15

$$\boldsymbol{\alpha}^{\mathbf{f}} = \begin{bmatrix} \alpha_{11}^f & \alpha_{12}^f \\ \alpha_{21}^f & \alpha_{22}^f \end{bmatrix}^{-1} \tag{5}$$

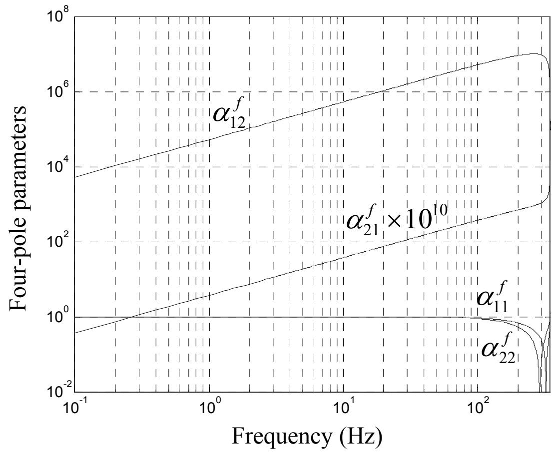

Figure 3. Four pole parameters of the foundation $\boldsymbol{\alpha}^{\mathbf{f}}$.

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. For N number of resonance changers in series, the combined response becomes:

$$\boldsymbol{\beta}^{\mathbf{ps}} = \boldsymbol{\alpha}^{\mathbf{p}} \boldsymbol{\alpha}^{\mathbf{s}} \boldsymbol{\alpha}^{\mathbf{b}} \prod_{q=1}^{N} \boldsymbol{\alpha}^{\mathbf{r}q} \, \boldsymbol{\alpha}^{\mathbf{f}} \,. \tag{6}$$

Submarine Hull

The hull is modelled as a ring-stiffened fluid-loaded cylindrical shell with finite end closures, and separated by bulkheads into a number of compartments. For the low frequency analysis of axisymmetric vibration of the stiffened shell, several assumptions are used. Firstly, at low frequencies, the effect of the stiffeners can be smeared onto the cylindrical shell as an increase in bending stiffness, such that the hull plate is considered as orthotropic (Hoppmann 1958; Leissa 1993). Secondly, the bulkheads that separate the pressure hull into compartments are modelled as circular plates in bending motion, and result in an increase in stiffness around the region of the structural junctions. Thirdly, at low frequencies, where the structural wavenumber is greater than the fluid wavenumber, heavy fluid loading effects can be modelled as an increase in inertia (Junger and Feit 1985). With these assumptions, the modified equations of motion based on the Donnell-Mushtari theory, for the axisymmetric response of a ring stiffened cylindrical shell under fluid loading, are given by (Leissa 1993; Junger and Feit 1985):

$$\frac{\partial^2 u}{\partial x^2} - \frac{\gamma}{c_L^2}\frac{\partial^2 u}{\partial t^2} + \frac{v}{a}\frac{\partial w}{\partial x} = 0 \tag{7}$$

$$\frac{v}{a}\frac{\partial u}{\partial x} + \frac{\tau}{a^2} w + \beta^2 a^2 \frac{\partial^4 w}{\partial x^4} + \frac{\mu}{c_L^2}\frac{\partial^2 w}{\partial t^2} = 0 \tag{8}$$

$$\gamma = 1 + \frac{A_r}{bh} + \frac{m_{eq}}{\rho h} \tag{9}$$

$$\tau = 1 + \frac{A_r (1 - v^2)}{bh} \tag{10}$$

$$\mu = 1 + \frac{A_r}{bh} + \frac{m_f}{\rho h} \tag{11}$$

where u, w are the axial and radial displacements of the cylinder respectively, as shown in Figure 4. a is the shell mean radius, h is the shell thickness, and $\beta^2 = h^2/12a^2$ is the thickness parameter. $c_L = \sqrt{E/\rho(1-v^2)}$ is the longitudinal wave speed, where E, ρ and v are the Young's modulus, density and Poisson's ratio of the hull, 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 to maintain a condition of neutral buoyancy. The fluid loading parameter m_f can be derived using a standing wave configuration of an infinite cylinder (Junger and Feit 1985).

Substituting the following general solutions for the axial and radial displacements

$$u(x,t) = U e^{-jkx + j\omega t} \tag{12}$$

$$w(x,t) = W e^{-jkx + j\omega t} \tag{13}$$

into the equations of motion results in two linear equations in terms of the displacement amplitudes U and W, which are arranged in matrix form as:

$$\begin{bmatrix} \gamma\Omega^2 - (ka)^2 & -jvka \\ -jvka & \beta^2 (ka)^4 + \tau - \mu\Omega^2 \end{bmatrix} \begin{bmatrix} U \\ W \end{bmatrix} = \begin{bmatrix} 0 \\ 0 \end{bmatrix} \tag{14}$$

where k is the axial wavenumber and $\Omega = \omega a / c_L$ is the non-dimensional frequency. For a non-trivial solution, the determinant of the coefficient matrix must be zero. The expanded determinant results in a third order dispersion equation in terms of k^2.

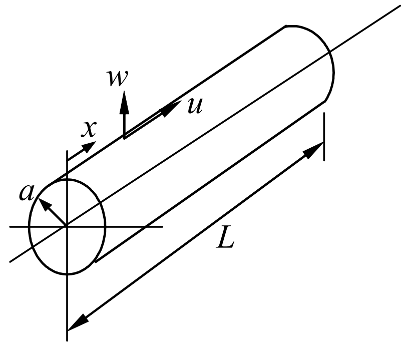

Figure 4. Coordinate system of a cylindrical shell.

In the absence of torsional motion, the three axial wavenumbers correspond to propagating extensional and flexural waves (two real wavenumbers), and near-field evanescent bending waves (one imaginary wavenumber), and where each wavenumber represents wave motion in the positive and negative directions. The characteristic equation of the cylindrical shell is given by

$$\beta^2 (ka)^6 - \gamma\Omega^2\beta^2 (ka)^4 + (\tau - \mu\Omega^2 - v^2)(ka)^2 - \gamma\tau\Omega^2 + \gamma\mu\Omega^4 = 0 \tag{15}$$

Since the characteristic equation given by equation (15) is a function of m_f, a numerical solution is required to determine the axial wavenumbers. For each axial wavenumber k_i ($i = 1, 2, \ldots 6$), the axial to radial amplitude ratio C_i can be obtained:

$$C_i = \frac{U_i}{W_i} = \frac{-jvk_i a}{(k_i a)^2 - \gamma\Omega^2}, \qquad i = 1, 2, \ldots 6 \tag{16}$$

For harmonic motion, the complete solution of the cylindrical shell becomes:

$$u(x) = \sum_{i=1}^{6} C_i W_i e^{-jk_i x} \tag{17}$$

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

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

$$Z_d = \frac{1}{j\omega u(x)}\,. \tag{19}$$

Force transmissibility through the propeller-shafting system

The magnitude of the force at the hull resulting from a unit load at the propeller ($f_p = 1$ N) is defined by (Snowdon 1971):

$$f_h = \left| \beta_{11}^{ps} + \frac{\beta_{12}^{ps}}{Z_d} \right|^{-1} . \tag{20}$$

Z_d is the driving point impedance of the cylindrical hull. 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 equation (6).

RADIATED SOUND PRESSURE

The approach to the solution of the acoustic pressure field generated by a finite cylindrical shell is based on the Helmholtz integral equation. In the analysis in the preceding section, the cylinder is subjected to an axial excitation at one end, and the dynamic response of the cylinder has been determined from the solution of the boundary conditions. The motions of the cylinder that contribute to the radiated sound pressure thus consist of (i) the rigid body motion of the end plates in the axial direction, and (ii) radial motion of the cylindrical surface. The Helmholtz integral equation for the pressure field due to a bounded radiating surface is given by (Junger and Feit 1985):

$$p(\mathbf{R}) = -\int_S \left(p_S \frac{\partial g(|\mathbf{R} - \mathbf{R_0}|)}{\partial \xi} + \rho_{fl} \ddot{z} g(|\mathbf{R} - \mathbf{R_0}|) \right) dS(\mathbf{R_0}) \tag{21}$$

$$g(|\mathbf{R} - \mathbf{R}_0|) = -\frac{e^{-jk_f |\mathbf{R} - \mathbf{R}_0|}}{4\pi |\mathbf{R} - \mathbf{R}_0|} \tag{22}$$

where $g(|\mathbf{R} - \mathbf{R}_0|)$ is the free space Green's function, $\mathbf{R}_0$ is the source point, $\mathbf{R}$ is the field point, p_S is the surface pressure, ρ_{fl} is the fluid density, S is the radiating surface, $\ddot{z}$ is the normal acceleration of the radiating surface, and ξ is the coordinate in the direction of the surface normal. For the finite cylinder, the total surface area S consists of three components as shown in Figure 5, corresponding to the end plate at $x = 0$ (S_1), the cylindrical shell (S_2), and the end plate at $x = L$ (S_3). 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 (Perreira and Dawe 1984). Expressions for the radiated pressure from the three surface areas can then be obtained (Tso *et al.* 2005).

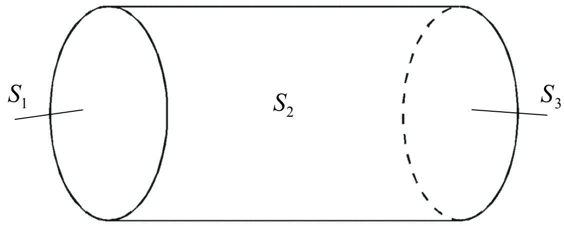

Figure 5. Surface areas of the end plates and cylindrical shell.

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, $p_{h,\max}(\omega)$.

OPTIMISATION OF THE RESONANCE CHANGER PARAMETERS

Development of Fitness Criteria

The force which acts on the propeller in a marine vessel has been shown to be approximately proportional to the propeller rotational speed squared (Goodwin 1960; Pan *et al.* 2002). 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:

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

It should be noted that this relationship does not represent a physical quantity due to the lack of information regarding realistic forcing magnitudes. It is, however, suitable for measuring the reduction in force transmissibility.

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

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

The frequency range included in equation (24) is bound by lower (ω_l) and upper (ω_u) limits. $\mathbf{x}$ is a vector containing the virtual mass, stiffness and damping parameters associated with the resonance changer. For *N* RCs in series, $\mathbf{x}$ is given by:

$$\mathbf{x} = \{k_{r1},\ m_{r1},\ c_{r1},\ \ldots\ k_{rN},\ m_{rN},\ c_{rN}\}^T \tag{25}$$

Genetic Algorithm Based Optimisation

Only the low frequency range (< 100 Hz) is of interest due to the excitation of the propeller occurring at the blade pass frequency. Lower ($\mathbf{x_l}$) and upper ($\mathbf{x_u}$) limits are also enforced on the RC parameters, that is, $\mathbf{x_l} \le \mathbf{x} \le \mathbf{x_u}$. There are numerous optimisation techniques available to solve the constrained fitness criteria defined in the previous section. Many of these techniques are robust but sometimes fail to find the global optimum. Complex fitness functions such as the one defined in equation (24) may contain several local optima. Since a mathematical condition defining the best solution for this case does not exist, finding this global optimum is usually more computationally involved. One method that has been frequently used by researchers to solve various non-linear optimisation problems is the genetic algorithm or GA (Goldberg 1989). GAs are an artificial application of Darwin's notion of natural selection and evolution, and have been proven to provide robust and accurate solutions to these problems. Although constraints cannot be applied directly within the GA, equality constraints may be subsumed into a system model and violated inequality constraints can be penalised such that the desired conditions are met.

RESULTS

The values of the propeller-shafting system used in the modelling are given in Table 2. The limits imposed on the RC parameters within the optimisation process are presented in Table 3. 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.

Figure 6 shows the frequency response of the maximum far-field radiated sound pressure at a distance of 1000 m from the hull resulting from a unit axial force input. A frequency band up to 100 Hz has been considered in the analysis. The peaks in the acoustic frequency response correspond to the axial resonances of the cylindrical hull. Figure 6 shows a reduction of the peaks at the first and third axial modes. This is due to the fact that the radiated pressure due to the radial motion of the cylinder is out of phase with that of the axial motion of the end plates. This is attributed to the fact that when the end plates are vibrating out of phase with each other and stretch out to generate a positive pressure field, the Poisson effect causes a reduction in the cylinder diameter, thereby reducing the radiated pressure (Tso *et al.* 2005).

The optimal RC parameters were obtained by minimising the cost function defined in equation (24). Figure 7 shows the weighted force transmissibility through the propeller-shafting system (including the hull impedance), without a RC, using one RC, and using two RCs in series. The optimal values using one or two resonance changers are given in Tables 4 and 5, respectively. The RC is comparable to a dynamic vibration absorber; it introduces an additional resonance which causes a shift in the original resonances whilst reducing the overall response. The optimal values of the RC parameter sets are within the lower and upper limits given in Table 3. Figure 8 presents the maximum weighted far-field radiated sound pressure from the cylindrical hull without a RC, using one RC, and using two RCs. It is evident that introduction of a resonance changer to the propeller-shafting system has caused a significant reduction in both the force transmissibility and radiated acoustic signature. In the case of using one RC, two peaks in the radiated sound pressure (Figure 8) have been equated (ie. have the same amplitude), which presents an optimal condition. Similarly, four peaks have been equated using two RCs. The introduction of a second RC has also caused a further reduction in the far-field radiated sound pressure compared with using a single RC. Comparing Figures 7 and 8 shows that minimisation of the maximum far-field radiated sound pressure does not correlate to a minimisation of the maximum weighted force transmissibility. This highlights the need to directly consider minimsing the acoustic response rather than reducing the vibration transmission to the hull.

Table 2. 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 3. 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

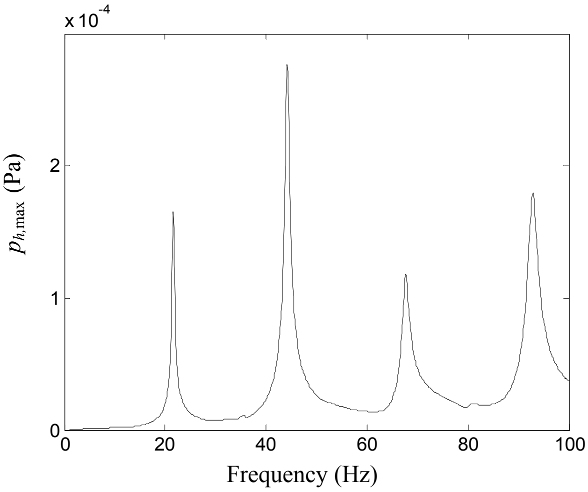

Figure 6. Cylinder acoustic frequency response function.

Table 4. Optimal parameters using one RC.

RC parameter	Optimal value
k_r (MN/m)	206.1
m_r (tonnes)	1
c_r (tonnes/s)	69.91

Table 5. Optimal parameters using two RCs in series.

RC parameter	Optimal value RC 1	Optimal value RC 2
k_r (MN/m)	881.6	106.0
m_r (tonnes)	25.81	1
c_r (tonnes/s)	74.79	46.54

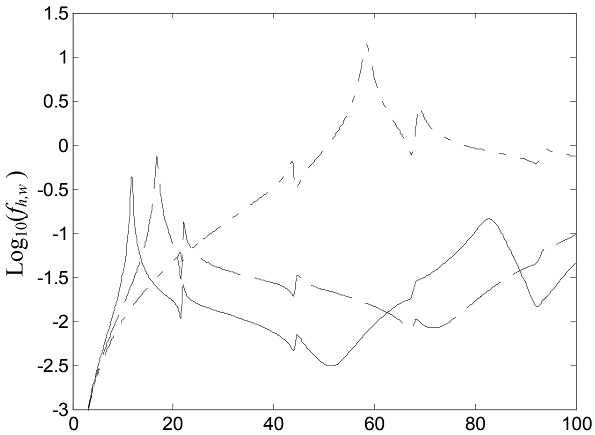

Figure 7. Force transmissibility (– · – no RC, ········ with 1 optimal RC, —— with 2 optimal RCs).

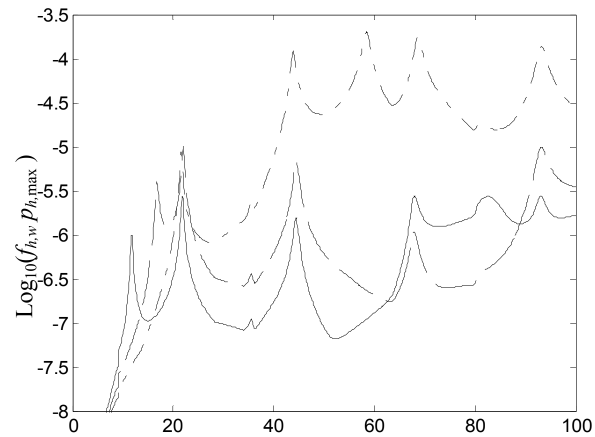

Figure 8. Radiated sound pressure (– · – no RC, ········ with 1 optimal RC, —— with 2 optimal RCs).

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 using the transmission matrix approach. The submarine hull was modelled as a ring stiffened finite cylindrical shell submerged in a fluid and undergoing axial excitation from the propeller-shafting system. An acoustic frequency response function has been developed. Optimal resonance changer parameters were obtained within realistic limits by minimising the far-field radiated sound pressure. The effect of using one or two resonance changers in series was examined. Significant reductions in the acoustic signature were observed, with a greater performance achieved using two resonance changers.

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