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OPTIMUM QUASI-ADAPTIVE RESONANCE CHANGER FOR SUBMERGED VESSEL SIGNATURE REDUCTION

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Abstract

Reduction of radiated noise from maritime vessels is an important issue. A significant source of structure-borne sound corresponds to excitation of the vessel due to vibration transmission through the propeller-shafting system. This vibration transmission can be reduced using a resonance changer. A resonance changer introduces virtual elastic, damping and inertial influences by hydraulic means and thereby acts as a hydraulic dynamic vibration absorber. This paper theoretically examines the use of a quasi-adaptive and hybrid resonance changer system in the propeller-shafting system of a submarine to directly reduce the far-field radiated noise. Results show that a quasi-adaptive system exhibits similar performance to multiple passive resonance changers within the active frequency bandwidths. The significance of this result is that the quasi-adaptive resonance changer system uses only a single piston whereas multiple passive resonance changers each consist of their own piston. The hybrid system consists of an additional passive resonance changer to improve the broadband response.

1. INTRODUCTION

It is crucial that submarines run as quietly as possible to reduce the likelihood of detection. A fully submerged submarine is observable to other vessels through its radiated noise signature. The longitudinal vibration transmission through the propeller-shafting system represents a dominant form of excitation of the hull which is under consideration in this work. Excitation occurs at the propeller when it rotates through a non-uniform wake. The resulting disturbance occurs at the blade pass frequency and is transmitted through the propeller-shafting system to the hull which in turn radiates noise. Literature on the acoustic responses of submarines is not generally available due to its sensitive nature. However, simplifications have been made to allow a submarine to be modelled as combinations of general shapes which have been studied extensively in the public domain [1-3]. Axial transmission through marine propulsion systems has been examined by various researchers [4-6] with the aim of reducing the vibration transmission to the hull. Goodwin [5] examined the reduction of excessive vibration through the propeller-shafting system by using an existing hydraulic device called the “Michell Thrustmeter” or as it was later named, a resonance changer (RC). Originally this device was used to measure the thrust at the thrust bearing and is normally located in series between the thrust bearing and supporting foundation. The RC introduces virtual elastic, damping and inertial influences by hydraulic means, thereby acting as a dynamic vibration absorber (DVA). Passive, active and semi-active or adaptive DVAs have been used in many applications [7-12]. A literature survey was presented by Sun *et al*. [12] on work concerning all of the various control strategies used with DVAs. Dylejko *et al*. [13] investigated the tuning of a passive RC to minimise the force transmission and time averaged power transmission to a submarine. The hull was modelled as a simplified 1-D rod with its mass density adjusted to maintain neutral buoyancy. Although passive DVAs can perform well over a wide frequency range, DVAs tuned to control the response at a single frequency can become mistuned with time. Tunable or adaptive DVAs (ADVAs) can address this problem by modifying their dynamic properties [14]. A preliminary study on minimisation of the vibration transmission through the propeller-shafting system in a submarine using a tunable RC was presented by Dylejko *et al*. [15].

2. QUASI-ADAPTIVE RESONANCE CHANGER

A simplified model of the proposed quasi-adaptive (QA) RC is shown in Figure 1, representing a variant of the passive RC model presented by Goodwin [5]. The RC consists of a piston of cross sectional area A_0, multiple oil reservoirs of volume $V_{\mathrm{N_{QA}}}$ and a pipe connecting these two elements of length $L_{\mathrm{N_{QA}}}$ and cross sectional area $A_{\mathrm{N_{QA}}}$ where $\mathrm{N_{QA}}$ is the number of independent reservoirs. The RC virtual stiffness (k_r), mass (m_r) and damping (c_r) are associated with the force required to compress the oil in the reservoir, overcome the inertia of the oil in the pipe and to overcome the viscous resistance of the oil in the pipe, respectively [5]. Automatic shut off valves would be connected to a controller using a signal from a tachometer to ensure that only one reservoir is in operation at a time. The switching between reservoirs would occur at a predetermined shaft rotational speed measured by a tachometer which is correlated to the excitation frequency of the propeller.

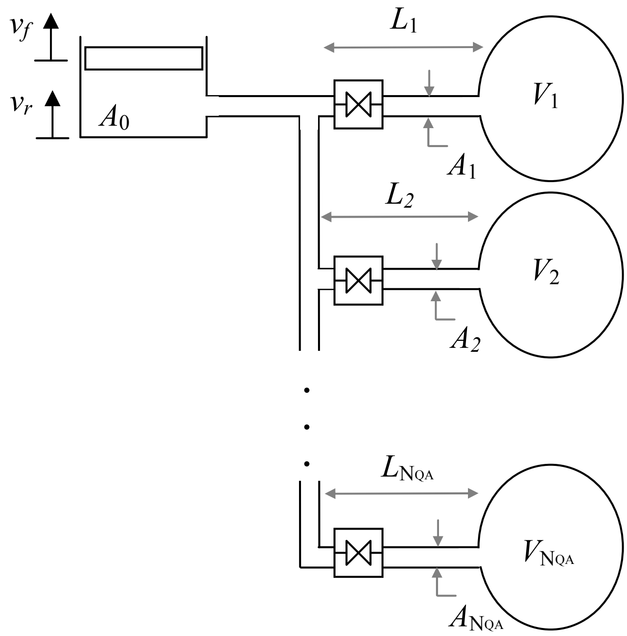

Figure 1. Simplified model of the quasi-adaptive RC system.

3. STRUCTURAL AND ACOUSTIC RESPONSES OF THE HULL

The transmission matrix method has been used to model the vibration transmission through the propeller-shafting system. The propeller-shafting model developed in [13] has been used. A transmission matrix schematic of the propeller-shaft model is given in Figure 2.

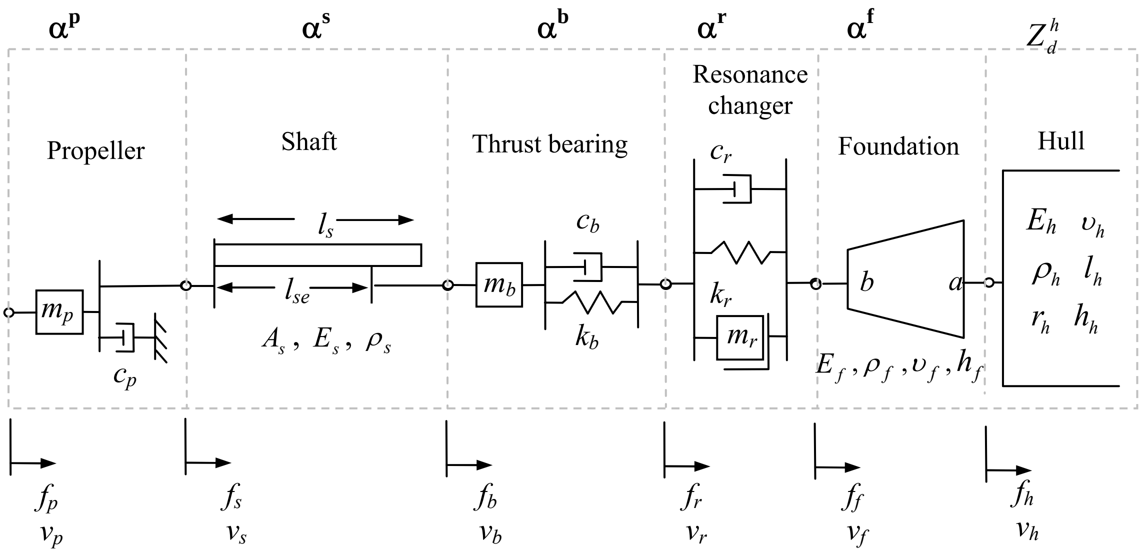

Figure 2. Transmission matrix model of the propeller-shafting system connected to the submarine hull.

Only the axisymmetric response of the cylindrical hull has been considered in this work since it is the most efficient radiator. The low frequency response of a rib stiffened, fluid loaded submarine hull including ballast tanks, bulkheads and a distributed mass loading due to onboard equipment can be approximated using the appropriate boundary conditions and by including the complicating effects in the Donnell-Mushtari equations of motion for a cylindrical shell [1, 16, 17]. The drive point impedance can be formulated from the calculated axial response at $x = 0$.

Under axial excitation, it has been assumed that the far-field radiated pressure field is dominated by the response of the end casings of the cylinder. This assumption allows the analysis of the sound radiation from the submarine to be obtained by examining the three radiating surfaces individually [3]. By assuming that the interactions between the radiating surfaces are negligible, an approximate closed form solution may be obtained. Figure 3 shows the geometric relationship between the three radiating surfaces. The centre of the cylinder has been used as reference to the field-point which is the location where the sound pressure $p(r,\theta)$ is obtained. In the far-field, the stationary phase approximation may be used for the radiation from the cylindrical shell [1]. The noise components associated with the ends have been approximated by treating the ends as rigid disks enclosed in baffles radiating with half the source strength. As a result, the far-field pressure resulting from the axisymmetric response of a submerged vessel can be approximated by:

$$p(r,\theta) = p_{e,1}(r,\theta) + p_c(r,\theta) + p_{e,2}(r,\theta) \tag{1}$$

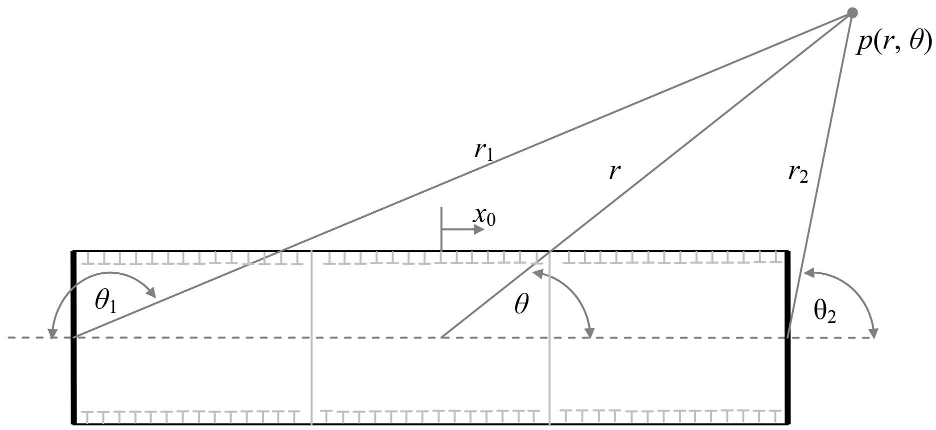

Figure 3. Geometry defining the relationships between the field point and radiating surfaces.

$$p_{e,1}(r,\theta) = -\omega^2 \rho_f \, \pi r_h^2 \frac{e^{ik_f r_1}}{4\pi r_1} \frac{2J_1(k_f r_h \sin\theta_1)}{k_f r_h \sin\theta_1} u_1 \tag{2}$$

$$p_c(r,\theta) = \frac{j\rho_f \omega^2 e^{jk_f r} \tilde{w}(k_f \cos\theta)}{\pi k_f r(\sin\theta) H_1[(k_f^2 - \gamma^2)^{1/2} r_h]} \tag{3}$$

$$p_{e,2}(r,\theta) = \omega^2 \rho_f \, \pi r_h^2 \frac{e^{ik_f r_2}}{4\pi r_2} \frac{2J_1(k_f r_h \sin\theta_2)}{k_f r_h \sin\theta_2} u_2 \tag{4}$$

$\gamma = k_f \cos\theta$. ρ_f, k_f, J_1 and H_1 are the fluid density, fluid wavenumber, Bessel function of the first kind and Hankel function of the first kind. u_1 and u_2 are respectively the axial displacements at each end of the cylinder. w is the radial displacement of the cylindrical shell and $\tilde{w}$ is the spatial transform of the radial displacement.

Since the wavenumbers are not influenced by the axial impedance attached to the hull in this analysis, the far-field radiated sound pressure is only a function of wave amplitude. The wave amplitude is dependent on the force delivered to the hull. This relationship allows a transfer function description of the maximum far-field radiated sound pressure. The resulting acoustic transfer function can be defined as:

$$H_a(r) = \frac{\max\limits_{0\le\theta_i\le 2\pi} p(r,\theta_i)}{f_h} \tag{5}$$

where θ_i represents a discrete angle and f_h is the force at the hull. The maximum far-field radiated sound pressure can be calculated by multiplying the force transmission to the hull by the acoustic transfer function [13]. The frequency squared weighted maximum far-field radiated sound pressure as a function of the RC parameters, field point radius and frequency can be expressed as:

$$P_{a,W}(\mathbf{x},\mathbf{x}_{\mathrm{N_{QA}}},\omega_i,r)=\left(\frac{\omega_i}{\Delta\omega}\right)^2 H_a(\omega_i,r)f_h(\mathbf{x},\mathbf{x}_{\mathrm{N_{QA}}},\omega_i) \tag{6}$$

$\mathbf{x}$ is a vector containing the virtual mass, stiffness and damping parameters associated with the N number of passive RCs and is given by:

$$\mathbf{x}=\left\{ k_{r,1} \quad m_{r,1} \quad c_{r,1} \quad k_{r,2} \quad m_{r,2} \quad c_{r,2} \quad \dots \quad k_{r,\mathrm{N}} \quad m_{r,\mathrm{N}} \quad c_{r,\mathrm{N}} \right\}^T \tag{7}$$

$\mathbf{x}_{\mathrm{N_{QA}}}$ is a vector containing the virtual mass, stiffness and damping parameters associated with the $\mathrm{N_{QA}}^{\mathrm{th}}$ parameter set of the quasi-adaptive RC and is given by:

$$\mathbf{x}_{\mathrm{N_{QA}}}=\left\{ k_{r,\mathrm{N_{QA}}} \quad m_{r,\mathrm{N_{QA}}} \quad c_{r,\mathrm{N_{QA}}} \right\}^T \tag{8}$$

The frequency weighting has been included in Eq. (6) since the excitation force at the propeller is approximately proportional to the propeller's rotational speed squared [4].

4. OPTIMISATION

The aim of the quasi-adaptive system is to minimise the weighted maximum far-field radiated sound pressure by using optimal RC reservoirs with independent parameter sets over specified frequency bands. The problem of finding the optimal RC parameter sets is similar to optimisation of a passive RC system [13]. The difference is that the frequency range is split into bandwidths with each frequency bandwidth corresponding to a different quasi-adaptive RC parameter set. The switching frequencies ω_s are also incorporated in the fitness criteria such that the optimum frequency bandwidths and switching frequencies are obtained. Hybrid control corresponds to the use of a quasi-adaptive RC in series with a passive RC. The aim of the passive RC element is to improve the response of the quasi-adaptive system further by including an additional passive RC piston and reservoir which is tuned considering the response over the entire frequency range. The fitness criteria to be minimised is given by:

$$\begin{aligned} &J(\mathbf{x},\mathbf{x}_1,\omega_{s,1},\mathbf{x}_2,\omega_{s,2}\dots\mathbf{x}_{\mathrm{N_{QA}}\text{-}1},\omega_{\mathrm{s},(\mathrm{N_{QA}}\text{-}1)}) \\ &=20\log_{10}\left\{\frac{\max\limits_{\omega_l\le\omega_i\le\omega_u}\left|\underset{\omega_l\le\omega_i\le\omega_{s,1}}{P_{a,W}(\mathbf{x},\mathbf{x}_1,\omega_i)},\underset{\omega_{s,1}\le\omega_i\le\omega_{s,2}}{P_{a,W}(\mathbf{x},\mathbf{x}_2,\omega_i)},\dots\underset{\omega_{s,(\mathrm{N_{QA}}-1)}\le\omega_i\le\omega_u}{P_{a,W}(\mathbf{x},\mathbf{x}_{\mathrm{N_{QA}}},\omega_i)}\right|}{\max\limits_{\omega_l\le\omega_i\le\omega_u}\left|P_{a,W}(\mathbf{x}=0,\mathbf{x}_1=0,\mathbf{x}_2=0\dots\mathbf{x}_{\mathrm{N_{QA}}}=0,\omega_i)\right|}+\varphi\right\} \end{aligned} \tag{9}$$

where

$$\varphi = \left| P_{a,W}(\mathbf{x},\mathbf{x}_1,\omega_{s,1}) - P_{a,W}(\mathbf{x},\mathbf{x}_2,\omega_{s,1}) \right| + \ldots \\ \left| P_{a,W}(\mathbf{x},\mathbf{x}_{\mathrm{N_{QA}}-1},\omega_{s,(\mathrm{N_{QA}}-1)}) - P_{a,W}(\mathbf{x},\mathbf{x}_{\mathrm{N_{QA}}},\omega_{s,(\mathrm{N_{QA}}-1)}) \right| \quad (10)$$

φ is an additional penalty term used to reduce the ambiguity of the fitness criteria. This penalty term ensures that cost function converges toward a solution when the responses are close to equal at the sqitching frequencies. The optimal design of the QA and the hybrid RC systems are achieved by minimisation of the fitness criteria defined in Eq. (9). The optimal system results in N optimal passive RCs, $\mathrm{N_{QA}}$ optimal parameter sets for the QA RC and $\mathrm{N_{QA}} - 1$ optimal switching frequencies. It should be kept in mind that only one independent parameter set is in use at any one time. The frequency range included in Eq. (9) is bound by lower, ω_l, and upper, ω_u, limits. Lower $\mathbf{x_l}$ and upper $\mathbf{x_u}$ limits are also enforced on the RC parameters, that is, $\mathbf{x_l} \le \mathbf{x} \le \mathbf{x_u}$. In order to provide a fair penalty distribution, the constraints have been normalised by using the normalisation defined in [13]. It should be noted that the weighted function described in Eq. (6) does not represent a physical quantity. It is however suitable for measuring the reduction in the far-field radiated sound pressure.

5. RESULTS

The following section presents the results of minimising Eq. (9) with a QA RC and hybrid RC configuration. Results are presented for a QA RC configuration with three independent parameter sets while the hybrid passive QA RC system was made up of a QA RC with two independent parameter sets and a single passive RC. A genetic and general non-linear constrained algorithm was used to find the optimal RC parameters [13]. The physical parameters for the propeller-shafting system and cylindrical hull have been taken from [13] and [17], respectively. The optimum RC parameter sets, RC natural frequencies, RC damping ratio and switching frequencies for the optimal QA and hybrid passive QA RC systems are presented in Tables 1 and 2, respectively. The maximum weighted far-field radiated sound pressure versus frequency for the case without the RC and using each of the three independent RC parameter sets for the three stage QA RC system is given in Figure 4. The switching frequencies are shown using dotted vertical lines at 33.4 and 62.4 Hz. A significant reduction in the maximum far-field radiated sound pressure level is observed for the in-band response. The in-band response made up of the active frequency bands is defined as the response using the QA RC parameter set which is active at each particular frequency. The broadband response of the QA RC system is less desirable, with some out-of-band frequency regions exhibiting a larger response than the system without the RC. This makes the use of a QA RC system more desirable for systems that are excited by predictable pure tones. An advantage of the QA RC system over multiple passive RCs is that only a single RC piston is needed. The additional passive RC is introduced in the hybrid control system to improve the poor broadband response. Figure 5 presents the weighted far-field radiated sound pressure against frequency using the hybrid QA-passive system. The switching frequency at 23.7 Hz (shown with a vertical dotted line) is lower than the switching frequencies of the three stage QA RC system. The in-band response of the hybrid system which consists of two RC pistons and three independent RC parameter sets exhibits similar reductions in the peak response levels to the three stage QA RC system. The hybrid system however does not suffer from the large out-of-band response of the QA RC.

Table 1. Optimum RC parameter sets for quasi-adaptive control.

RC virtual parameter	RC_1	RC_2	RC_3
Stiffness k_r (MN/m)	67.31	275.6	304.1
Mass m_r (tonnes)	5.435	2.695	1.000
Damping c_r (tonnes/s)	611.6	41.22	5.046
Natural frequency f_n (Hz)	17.7	50.9	87.8
Damping ratio ζ	0.506	0.024	0.005
Switching frequency f_s (Hz)	-	33.4	62.4

Table 2. Optimum RC parameter sets for hybrid RC system.

RC virtual parameter	Passive RC	Quasi-adaptive RC	
		RC_1	RC_2
Stiffness k_r (MN/m)	146.8	358.1	401.4
Mass m_r (tonnes)	1.022	19.45	1.158
Damping c_r (tonnes/s)	48.44	231.6	95.50
Natural frequency f_n (Hz)	60.3	21.6	93.7
Damping ratio ζ	0.063	0.044	0.070
Switching frequency f_s (Hz)	N/A	-	23.7

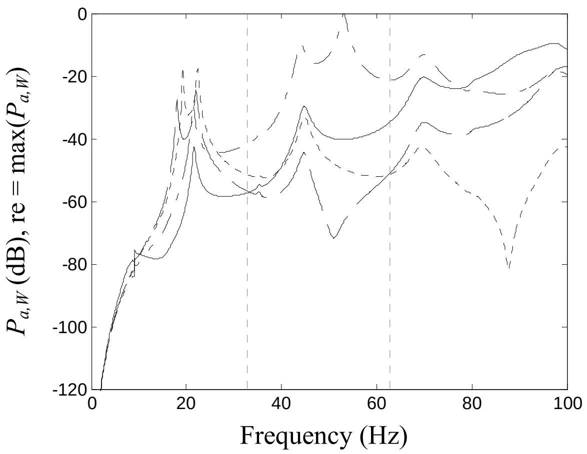

Figure 4. Weighted far-field radiated sound pressure: without an RC (— - -), with the first QA RC parameter set in operation (——), with the second QA RC parameter set in operation (—— -) and with the third QA RC parameter set in operation (- - - - -).

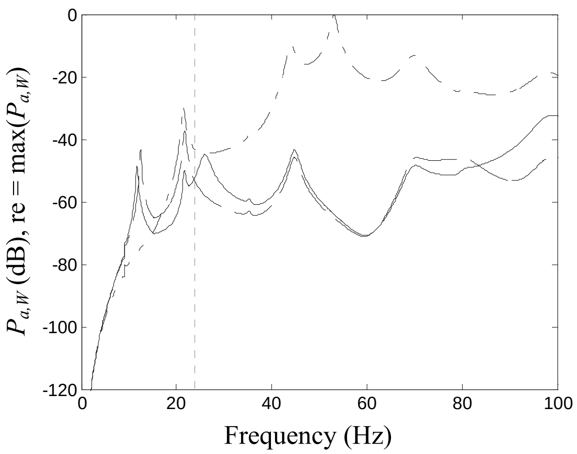

Figure 5. Weighted far-field radiated sound pressure: without an RC (— - -), the hybrid system with the first QA RC parameter set in operation (——) and the hybrid system with the second QA RC parameter set in operation (—— -).

6. CONCLUSIONS

A preliminary study into the reduction of the far-field radiated sound levels of a submerged vessel using a QA RC and hybrid passive QA RC has been undertaken. The vibration transmission through the propeller-shafting system connected to a cylindrical shell model of the hull has been modelled using the transmission matrix approach. The structural and acoustic responses of the cylindrical hull included the effects of the surrounding fluid, rib stiffeners, ballast tanks, distributed mass loading of the onboard equipment and the bulkheads. The far-field radiated sound pressure resulting from the low frequency excitation of the hull from the propeller has been approximated and formulated as a function of the RC parameters. The optimum RC parameters have been obtained by minimising the radiated sound pressure using a genetic and general non-linear constrained algorithm. The QA RC system was shown to provide good reductions in specific frequency bands, but suffered from a poor broadband response. The hybrid passive QA RC system overcomes this limitation by providing excellent reductions over the entire working frequency range.

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