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Structural and Acoustic Responses of a Submarine due to Propeller Forces Transmitted to the Hull via the Shaft and Fluid

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ABSTRACT

The structural and acoustic responses of a submarine hull to fluctuating propeller forces are investigated in the low frequency range. The fluctuating forces are caused by operation of the propeller in a non-uniform wake, and are transmitted to the submarine hull through the shaft and also via the fluid. Numerical models have been developed to simulate the strongly coupled fluid-structure interaction of the submerged vessel in the frequency domain. The hull is modelled using the finite element method (FEM), which allows for structural complexities such as the ring-stiffeners, bulkheads, end closures and the propeller/shafting system. A simple passive vibration attenuation system known as a resonance changer (RC) is included in the model of the propeller/shafting system. The fluid is modelled using the boundary element method (BEM) in which the radiation damping and added mass effect of the water are taken into account. It is shown that the performance of the RC is influenced greatly by the effects of forces transmitted to the hull via the fluid. Propeller vibration is taken into account in determination of the fluid and structural excitation of the hull. The effect of changing the propeller mass is observed.

INTRODUCTION

The reduction of noise emitted by submarines has long been a key topic in naval research, because submarines can be detected by passive sonar over large distances (Ross 1987). Noise emitted from sources internal to the hull and from the propeller can be distinguished. Propeller noise is due to cavitation, flow noise, the rotation of blade thickness and thrust, blade vibration and velocity variations in the wake (Carlton 1994, Rath Spivack 2004). Since submarines usually operate at large depths, cavitation is suppressed by the high water pressure. Propeller blades are shaped to reduce net fluctuating forces due to imperfections in the incident wake field.

Flow noise is moderated by travelling at low speeds. The overall sound signature is a combination of tonals and broadband random noise. Tonal components are more distinctive than random noise, but the combination of tonal components at different frequencies with broadband noise provides important information about the identity and speed of the submarine. At higher speeds in particular, the propeller is usually the most significant source of tonal and broadband noise.

The operation of the propeller in a non-uniform wake, as shown in Fig. 1, is the most important reason for the generation of tonal noise. The non-uniformity of the wake is due to asymmetry in the hull or protrusions of control surfaces, as shown in Fig. 2. As the propeller blades rotate through areas of different water velocity, fluctuations in thrust are generated at the blade-passing frequency (*bpf* = number of blades multiplied by the propeller rotational speed) and its multiples. Fluctuating forces of similar order also arise in the vertical and transverse directions. The observed tonal components arise primarily from this spatial variation in the wake field, combined with the usually smaller effects of the rotation of blade thrust and blade thickness. These tonal components are complemented by random components due to turbulence in the wake flow at entry to the propeller and also turbulence generated by flow over the blades. The variation in thrust causes structural excitation of the hull through the propeller/shafting system, resulting in vibration of the hull and the propeller. The low frequency vibrational modes of the hull and propeller-shafting system can result in a high level of radiated noise (Dylejko 2008). In addition, the same hydrodynamic mechanism results in dipole sound radiation from the propeller, where the dipoles are related to structural forces in strength and direction. The dipole sound radiation of the propeller also contributes to excitation of the hull, via the combination of the hydrodynamic near field and the acoustic far field. The pressure field in the immediate vicinity of the propeller is highly complex (Kehr et al., 2004). Simplification to a single dipole with both near and far field characteristics allows the nature of hull excitation via the fluid to be explored, without specifying the detail of the propeller itself.

As a simplification, the propeller and shaft can be envisaged as a spring-mass system having one natural frequency, where the propeller is the mass and the shaft is the spring. In reality, the complete propeller/shafting system is more complicated, as other components such as the thrust bearing and foundation contribute to its overall dynamic behaviour. Since the shaft passes through the tail cone, it is comparatively long and flexible. This leads to a low fundamental resonant frequency of the propeller/shafting system, involving significant axial vibration at this frequency. Consequently, the propeller/shafting system can cause significant axial vibration of the pressure hull and sound radiation, even at a frequency that does not match one of the natural frequencies of the hull (Dylejko 2008).

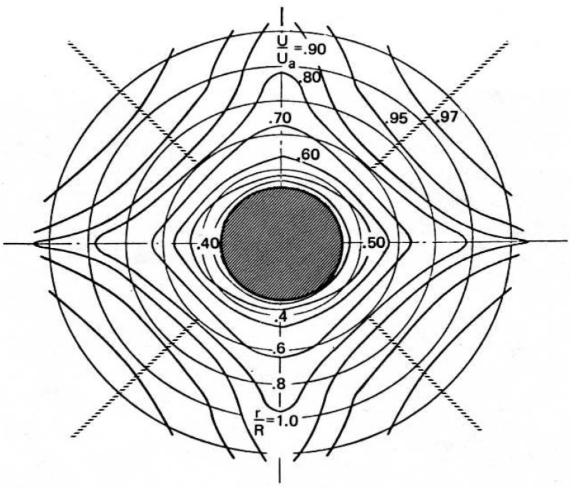

Figure 1. Wake of a torpedo (Ross, 1987)

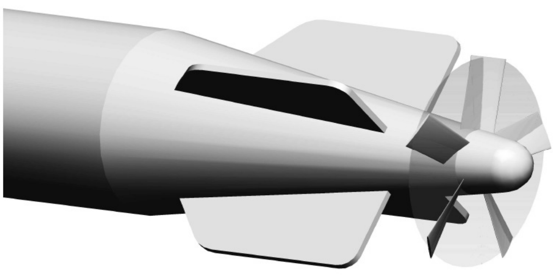

Figure 2. Stern of a torpedo

At low frequencies, the pressure hull can be represented by a thin-walled cylinder reinforced by ring-stiffeners and bulkheads to withstand hydrostatic pressure at large depths. The pressure hull end closures are typically flat or hemi-spherical. Changes in hydrodynamic pressure do not cause the overall dynamic behaviour to change markedly at operational depths (Norwood 1995). The foundation of the propeller/shafting system is mainly supported at the stern side pressure hull end plate. For this reason, vibration of the hull can be excited through the propeller/shafting system that is correlated to the bending and accordion modes of a free thin-walled cylindrical shell. Vibration correlated to the accordion modes is known to be an effective sound radiator, whilst vibration correlated to the bending modes is usually less effective.

To attenuate axial vibration of the propeller/shafting system and thereby reduce the transmission of axial forces from the propeller to the hull, a device known as a resonance changer (RC) can be implemented in the propeller/shafting system. The RC, initially derived from a thrust-meter, is a hydraulic vibration absorber that can be represented by a virtual spring-mass-damper system (Goodwin 1960). It detunes the natural frequencies of the propeller/shafting system and introduces additional damping.

Early models to find optimal RC parameters treated the submarine hull as a rigid termination (Goodwin 1960) or one-dimensional rod model (Dylejko 2007). In recent work, a simplified physical model of a submarine was included to consider the hull impedance, hull resonances and radiated sound due to shaft excitation (Dylejko 2008). However, the excitation of the submarine hull by the dipole field of the propeller had still to be taken into account. It was assumed previously that the dipole excitation is negligible, as earlier research predicted a contribution that is only 6-8% of the contribution of the structural force, where the Laplace equation was used to model the fluid (Chertok 1965). Recent work using the Helmholtz equation has shown that the contribution of fluid forces to the overall excitation is between 10% and 50% of the response due to excitation through a rigid shaft (Kinns et al., 2007; Merz et al., 2007). This is because the transition from the hydrodynamic near field to the acoustic far field occurs close to the propeller at frequencies of practical interest. Therefore, the fluid forces have been included in the most recent work concerning resonance changer performance (Merz et al., 2008). Another problem arising from the use of a resonance changer to minimise excitation of a submarine hull is that the axial movement of the propeller can be increased (Dylejko, 2008). This causes additional sound radiation from the propeller and additional excitation of the hull via the fluid.

The dynamic behaviour of the free-flooded tail cone that supports the aftmost propeller bearing is of particular interest. Pan et al. (2008) used a simplified analytical representation of this cone to show that it could have a major effect on sound radiation due to an axial force applied at the propeller thrust bearing, depending on assumed boundary conditions for the cone. The dipole forces were not considered but the excitation of the tail cone by the pressure field near the propeller further increases the potential significance of tail cone characteristics. For this reason, the dynamic behaviour of the cone has been explored in this work, using various assumptions concerning cone stiffness and internal water loading. Numerical models have been developed to simulate the strongly coupled fluid-structure interaction of a submerged vessel at low frequencies. The hull is considered to be under both structural and acoustic excitation by the propeller.

The effect of the tail cone on the hull excitation and dynamic response is presented. The performance of a RC implemented in a propeller/shafting system to attenuate axial forces from the propeller to the hull, in the presence of dipole excitation of the hull surfaces, is investigated. Furthermore, the magnitude of additional sound radiation from the propeller due to its increased axial vibration induced by the RC is examined. For some of the models developed in this work, the finite element method (FEM) (Zienkiewicz and Taylor, 2005; Bathe, 1985) was used to model the structure and fluid. For the remaining models, FEM was used to model the structure and the direct boundary element method (DBEM) (Brebbia and Ciskowski, 1991) was used to model the fluid domain. The FEM software package ANSYS 11 and the BEM/FEM software package Sysnoise 5.6 were chosen for the analysis, which was conducted in the frequency domain.

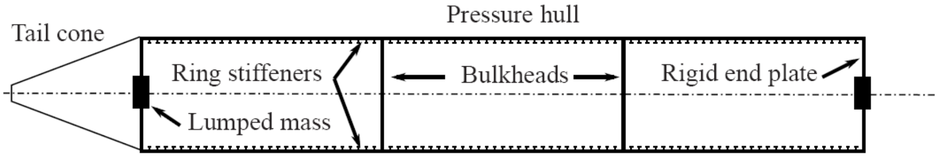

Figure 3. Simplified physical model of the submarine

DYNAMIC MODEL OF THE SUBMARINES

Only axial excitation of the submarine hull has been considered in this work as the accordion modes are particularly effective sound radiators. An axisymmetric model was used. The model consists of two parts, namely, the propeller/shafting system and the submarine hull. A real submarine has the pressure hull as its main structure with external attachments, such as buoyancy tanks that are of relatively light construction. The pressure hull, depicted in Fig. 3, has been shown to control hull dynamic properties at low frequencies (Tso and Jenkins, 2003). The pressure hull was modelled as a thin-walled cylinder with evenly spaced ring-stiffeners of rectangular cross-section. In addition, two evenly spaced bulkheads were included in the model as circular plates. The end plates of the pressure hull have been treated as rigid, as they are relatively stiff. The on-board machinery and remaining internal structure were considered as a distributed mass of the cylindrical shell. The distributed mass was chosen in such a way that neutral buoyancy of the submarine is guaranteed (Dylejko 2008). Lumped masses were added to both end plates to represent the water in ballast tanks and free-flooded structures. The elongated tail cone was considered explicitly since the dipole excitation of the hull originates at the propeller hub.

The basic elements of the propeller/shafting system relevant for axisymmetric analysis are the propeller, shaft, thrust bearing, resonance changer (RC) and foundation, as shown in Fig. 4. A modular approach for the propeller/shafting system is shown in Fig. 5, where the propeller force and velocity amplitude are given by f_p and v_p, respectively. The hull drive point force and velocity are denoted by f_h and v_h. The propeller is represented by a lumped mass m_p that also includes the added mass effect of the water. The propeller dimensions for calculating the propeller mass and the added mass of water effect are chosen for the principal model by assuming that the propeller volume is 1/1000 of the volume displaced by the pressure hull. The propeller diameter is assumed to be half the pressure hull diameter. Later, the propeller mass, but not the diameter, is varied. The propeller shaft was modelled as a simple rod with an effective length l_{se} and overall length l_s, where the overhang was represented by another lumped mass. The shaft properties are also defined by its cross-sectional area A_s, Young's modulus E_s and density ρ_s. The thrust bearing was assumed to act as a spring-mass-damper system with mass m_b, damping coefficient c_b and spring constant k_b. For the present model, the thrust bearing is attached to a single RC that has been reduced to a spring-mass-damper system according to Goodwin (1960). The RC incorporates a hydraulic cylinder that is connected to a reservoir via a pipe. Virtual mass, damping and stiffness are calculated using Goodwin (1960)

$$m_r = \frac{\rho_r A_0 L_1}{A_1}\ ,\ c_r = 8\,\pi\mu L \frac{A_0}{A_1}\ ,\ k_r = \frac{A_0 B}{V} \qquad (1)$$

where ρ_r is the density of the hydraulic medium, μ is the dynamic viscosity and B is the bulk modulus of the oil in the RC. V is the volume of the reservoir, A_1 is the cross-sectional area of the pipe, L is the pipe length and A_0 is the cross-sectional area of the cylinder. In a real submarine, the foundation of the propeller/shafting system is a complex shell-like structure, but here it is represented as a truncated cone for the axisymmetric model with end radii a and b. The Young's modulus, density, Poisson's ratio and thickness of the foundation are given by E_f, ρ_f, v_f and h_f, respectively. The parameters used in the model are given in Tables 1 and 2.

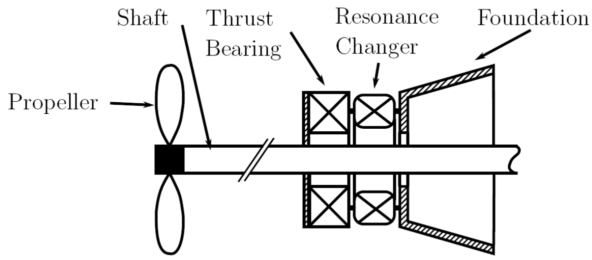

Figure 4. Propeller/shafting system

FORMULATION OF THE PROBLEM

The specified problem is a classic fluid/structure interaction problem with strong coupling due to the similar densities of fluid and structure. It also includes sound radiation and scattering due to structural vibration and acoustic sources. Furthermore, it can be characterised by sound radiation into an unbounded domain which requires the Sommerfeld radiation

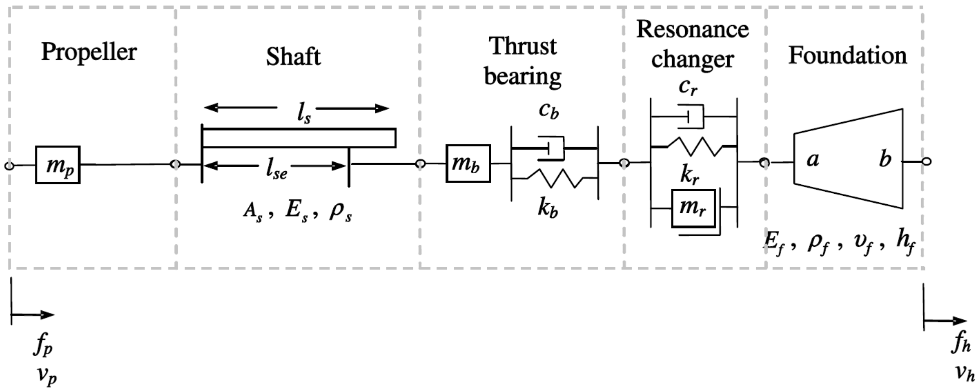

Figure 5. Simplified physical model of the propeller/shafting system

condition to be satisfied. Vibration is assumed to be harmonic and linear. Spectral analysis allows broadband random excitation to be taken into account, so the analysis can be conducted in the frequency domain and all displacements, pressures and velocities are represented by their complex amplitude. The time harmonic dependence has been omitted, if not stated otherwise. The problem consists of the structural part and the acoustic part. For the structural part, the finite element method (FEM) was utilised. For the acoustic part, both FEM and the direct boundary element method (DBEM) were used. This hybrid modelling was chosen because the software packages do not allow direct coupling and the FE capabilities of the Sysnoise program are limited. For each case, the field variables are described for each element and have to be coupled at the structure/fluid interface. This can be done using the linearised momentum equation, where the surface normal velocity of the structure can be related to the surface pressure gradient by:

$$\frac{\partial p}{\partial n} = -\ j\omega\rho v_n \tag{2}$$

where p is the acoustic pressure, $\mathbf{n}$ is the surface normal vector, ρ is the density of the fluid and v_n is the velocity component normal to the surface. The discrete systems for both the structural and acoustic parts must be expressed in terms of the acoustic pressure and structural displacement, respectively.

A complete description of the FE and BE models is given in Merz et al. (2008). The structure that interacts with the fluid is represented by a thin-walled axisymmetric shell of finite elements based on Kirchhoff-Love theory, where transverse shear deformation is not taken into account. The direct boundary element method (DBEM) is based on the Kirchhoff-Helmholtz equation that relates the field pressure to the surface pressure of an oscillating structure by Kirchhoff (1883). In order to couple the DBEM acoustic problem to the FEM structural problem, modal superposition is used (Seybert 1993). This means that the displacements of the structure are expressed as a superposition of modal contributions

SOUND RADIATION BY THE PROPELLER

The sound field radiated by the propeller is due to (i) the hydrodynamic mechanism that arises through the propeller operating in a non-uniform wake and (ii) the axial vibration of the propeller associated with the dynamic characteristics of the shafting system. The sound radiation originates from the propeller blades as multiple dipoles. The dipoles can be simplified to a single dipole located at the propeller hub, because the wavelength is large relative to the propeller diameter and the propeller is small relative to the submarine.

A derivation of the dipole field pressure due to a fluctuating force is provided by Ross (1987). The directivity pattern of the dipole is governed by $\cos\theta$, where θ is the angle between the field point vector with respect to the source and the force direction. The amplitude is directly proportional to the structural force. The radial variation of the amplitude follows $1/r^2$ in the near field and $1/r$ in the far field, where r is the distance of the field point from the source. The transition is a function of the wavelength λ and occurs at $\lambda/2\pi$. A polar diagram of a dipole is given in Fig. 6. For (i), the pressure field due to the force at the propeller hub is that due to an axial dipole. For (ii), the propeller was simplified as a rigid circular disc. The pressure field reduces to that of an axial dipole if $ka < 0.5$, where a is the disc radius. The added mass of water due to fluid loading of the propeller is then defined by the radius a (Merz et al., 2008), while the radiation damping due to fluid loading is negligible for the model presented here. The two dipoles do not usually have the same magnitudes and phase.

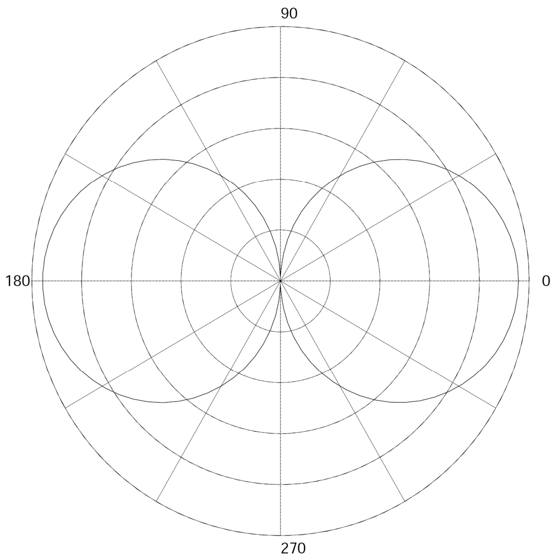

Figure 6. Dipole directivity pattern

RESULTS

Due to limitations of the available software, several restrictions regarding the modelling of the submarine apply. Development of a fully coupled model was not feasible using the available combination of ANSYS and Sysnoise, where combined structural and acoustic excitation, internal fluid loading of the free flooded tail cone and the propeller/shafting system are considered simultaneously. Therefore a semi-coupled, hybrid model has been developed, where the force transmissibility and the propeller vibration are computed using a FE/FE model. The data is then used for a FE/BE model, where both structural and acoustic excitation are present. An error is introduced because the FE/FE model does not allow consideration of acoustic excitation of the hull. However, it will be shown that the influence of the drive point impedance of the submarine on the force transmissibility of the propeller/shafting system is small, so this error is negligible. It is investigated whether the free flooded tail cone can be simplified to a rigid cone without significantly changing the dynamic and acoustic behaviour of the submarine (Pan et al., 2008). The model parameters are given in Tables 1 and 2 for the propeller/shafting system and the submarine hull, respectively.

PROPELLER/SHAFTING SYSTEM

The dynamic response of the propeller/shafting system has been investigated with emphasis on the force transmissibility and the propeller vibration, with and without an RC. The RC parameters have been optimised for a submarine model of similar dimensions, where the cost function is the maximum weighted force transmissibility. For the optimisation of the RC parameters, dipole excitation of the hull was neglected. The drive point impedance has been examined in order to establish the accuracy of the hybrid model. The force transmissibility of the structurally excited propeller/shafting system with a rigid termination is compared to the force transmissibility of the propeller/shafting system that is coupled to the simplified physical model of the fluid loaded submarine hull in Fig. 7. The local peaks at 21 Hz and 43 Hz are the first and second axial hull resonant frequencies. The RC reduces the maximum force transmissibility and detunes the funda-

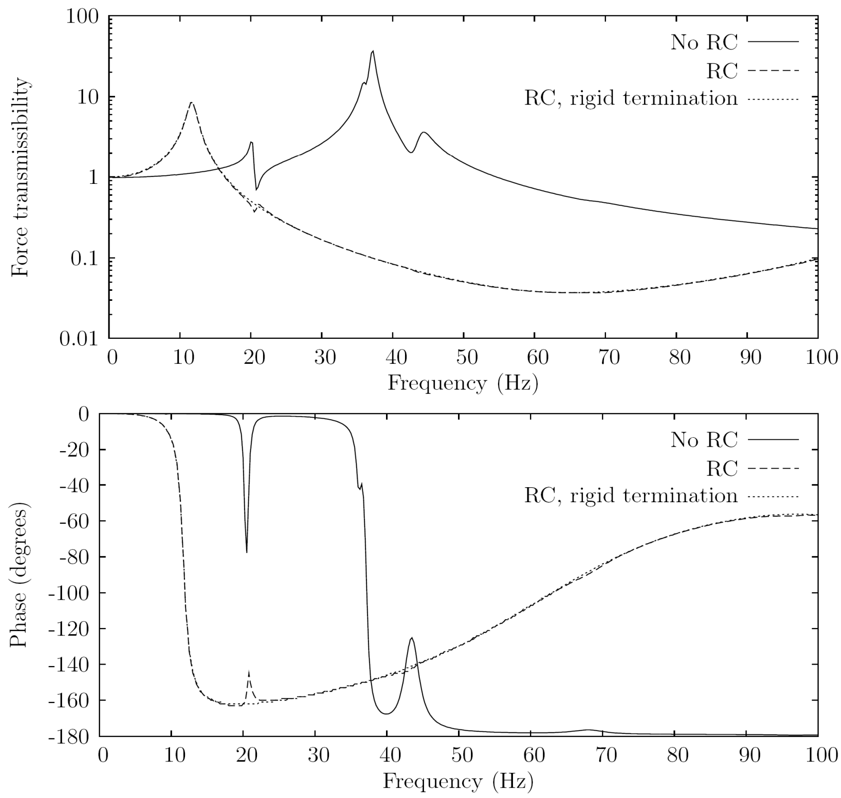

Figure 7. Force transmissibility of the propeller/shafting system

mental resonant frequency of the propeller/shafting from 37 Hz to 12 Hz.

The RC also leads to a significant decrease of the force transmissibility above the fundamental resonant frequency. It is evident that the -180^o phase shift due to the resonant frequency of the propeller/shafting system is not sustained in the case of the model with the RC, but reduces gradually to a -58^o phase shift. This is likely to play an important role for the combined dipole and structural excitation, as cancellation or reinforcement effects may occur, depending on their relative phase.The difference between the model with a rigid termination and the model with the simplified physical model of the submarine are the local peaks at the hull resonant frequencies. The effect of the drive point impedance on the force transmission is very small for the first resonant frequency and negligible for the second and higher hull resonant frequencies. Clearly, the natural resonances of the submarine hull at 21 Hz and 43 Hz have much less influence on the model with the RC, showing that the propeller/shafting system with the RC is then weakly coupled to the submarine hull.

INFLUENCE OF THE TAIL CONE PROPERTIES

It was not possible to model the tail cone internal water using Sysnoise, which was needed to model dipole excitation, nor was it possible to compute the radiated sound field using ANSYS. Hence, it was necessary to develop a simplified physical model of the tail cone and to investigate its accuracy. A first approximation to the tail cone is a rigid cone with no internal water, where the mass effect of the water is considered as a lumped mass at the pressure hull end plate in order to maintain the global dynamic behaviour of the pressure hull. However, the stiffness of the tail cone has an influence on the sound radiation as well as the sensitivity of the structure to excitation from a nearby sound source. Therefore, additional models were developed that represent a link between the model with internal fluid loading and the model with a rigid tail cone. The range of examined models includes:

(a) a flexible tail cone with internal water;
(b) a flexible tail cone with the same structural parameters as (a) but with a lumped mass representation of the internal water, where the lumped mass is attached to the rear end plate;
(c) a flexible tail cone with the lumped mass representation of the water as used for (b), but with an increased Young's modulus;
(d) a rigid tail cone with the lumped mass representation of the internal water as used for (b) and (c).

Figure 8 shows the point mobility of the pressure hull end plate, where the results have been obtained from the FE/FE model. The lumped mass used to simulate the effect of the cone internal water was chosen such that the simplified model yields the best match with the model that includes internal water. A lumped mass that is about half the mass of the internal water was found to be the best approximation. In the case of the flexible cone without internal water, the stiffness was increased through the Young's modulus. The global behaviour of the structure is seen to be only weakly dependent on assumptions about the tail cone properties. The first four hull axial resonances can be identified at 21, 43, 68 and 96 Hz.

Figure 9 shows the transfer mobility of the pressure hull end plate for a structural force applied at the end of the tail cone. All models are good approximations to the model with the flexible cone and internal water for frequencies below about 50 Hz. However, for higher frequencies, the results for the model of the flexible tail cone with the lumped mass approximation show a significant deviation from the results for the model with the internal water, because the first axial

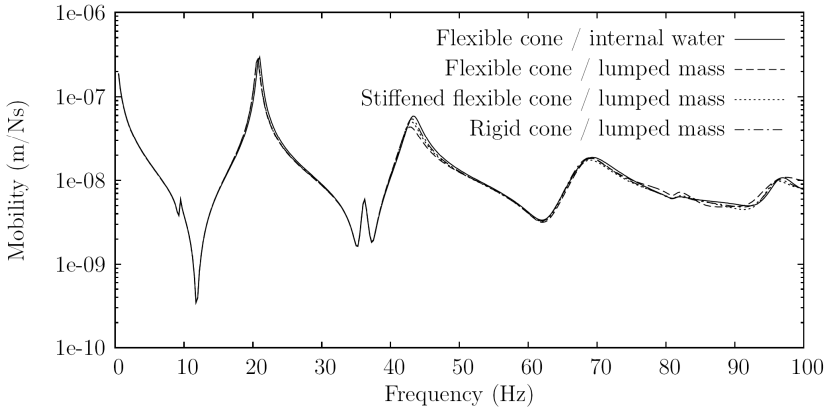

Figure 8. Point mobility of the rear end plate using different representations of the free-flooded tail cone

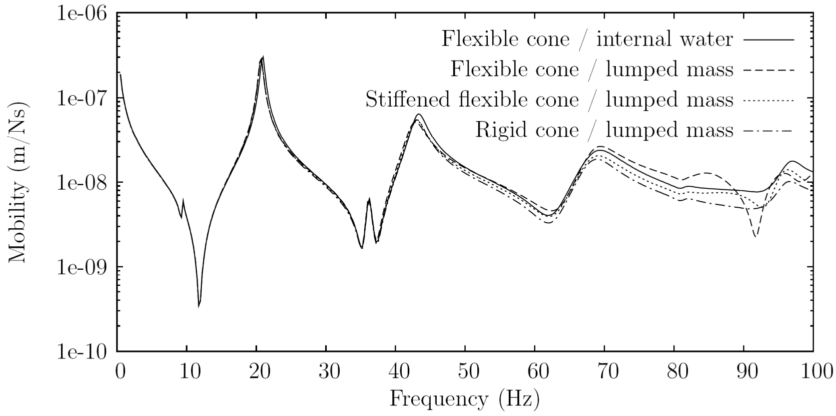

Figure 9. Mobility of the rear end plate due to excitation of the tail cone end, using different representations of the free-flooded tail cone

resonance of the cone then lies within the investigated frequency range at about 92 Hz.

RESPONSES OF THE COUPLED MODEL UNDER STRUCTURAL AND ACOUSTIC EXCITATION

The structural and acoustic responses of the submarine hull have been investigated, where the tail cone was modelled as a rigid structure. Both structural excitation through the propeller/shafting system and acoustic excitation of the submarine hull have been considered. The acoustic excitation is due to the dipole pressure field caused by operation of the propeller in the non-uniform wake as well as propeller vibration. The acoustic response in the far field is a combination of sound radiated from the submarine hull due to structural and dipole excitation and sound radiated directly from the propeller.

To assess stealth, the overall radiated sound power has been considered. The sound power was computed by integration of the intensity over a sphere of 1000 m radius with the centre between the end plates of the submarine for models with a rigid cone. Two cases have been examined. For each case, models with and without an RC have been investigated. For the first case, only structural excitation of the submarine hull through the propeller/shafting system is taken into account. For the second case, structural excitation via the propeller shaft and excitation by both the dipole field due to operation

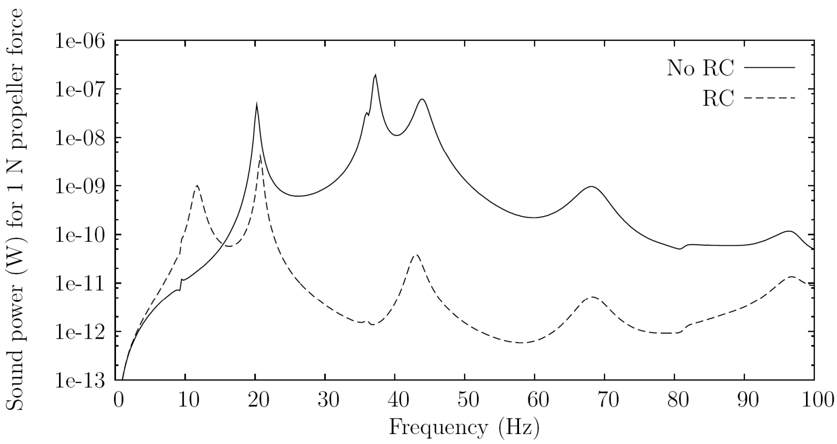

Figure 10. Radiated sound power due to structural excitation

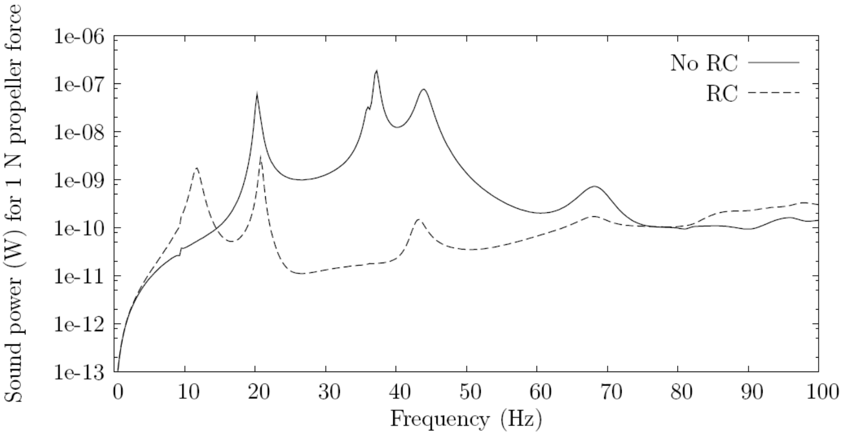

Figure 11. Radiated sound power, where structural and acoustic excitation of the hull are considered. The dipole due to propeller vibration is included.

of the propeller in a non-uniform wake and the dipole field caused by propeller vibration, have been taken into account.

Figure 10 presents the results for the first case and shows that implementation of an RC in the propeller/shafting system leads to a significant reduction of overall radiated sound power over a wide frequency range. It can be seen that the peak at 37 Hz due to the fundamental resonance of the propeller/shafting system shifts to 12 Hz, leading to an increase in radiated sound power at lower frequencies.

Figure 11 shows the radiated sound power for the second case. Implementation of an RC does not lead to a decrease of sound radiation at higher frequencies due to dominance of dipole excitation. In addition, the dipole due to propeller vibration has a smaller magnitude with the RC implemented at higher frequencies, so that a cancellation effect of the two dipoles is more significant for the model without an RC. The dipole field due to propeller vibration causes an additional increase in radiated sound power at the fundamental resonance of the propeller/shafting system, when an RC is implemented.

THE EFFECT OF PROPELLER MASS ON SOUND RADIATION

The propeller mass has a significant influence on the sound radiation. In order to demonstrate this, the propeller diameter has been fixed, so that the fluid loading remains the same. The propeller mass has then been varied. The change in mass influences the dynamic behaviour of the propeller/shafting system, so that the frequency of the principal shafting axial resonance is reduced when the mass is increased and vice versa. The fluid loading is present throughout, so the overall mass change is less than that of the propeller alone. The second effect is to change the balance between the two dipole fields, because the phase and amplitude of the dipole due to vibration changes relative to the dipole that is associated with hydrodynamic forces.

Figures 12 shows the effects on radiated sound power of increasing the propeller mass from 10 to 20 tonnes. The changes are complicated, because the dynamic properties of the propulsion system, as well as the effects of the dipoles to hydrodynamic loading and propeller vibration, are modified. The effect of propeller blade flexibility has not been taken into account in the present model; it will further complicate the nature of the pressure field that excites the hull.

The results show why careful examination of parameters is necessary, if an RC system is to have a beneficial effect at important frequencies. In particular, consideration must be given to the effect on components at higher multiples of *bpf*, which are likely to differ markedly from those at *bpf* itself.

CONCLUSIONS

A numerical model has been developed to investigate the acoustic and structural responses of a submarine hull due to axial excitation from propeller forces. The propeller forces arise from a non-uniform wake and are transmitted to the hull via the fluid and the propeller/shafting system. The dynamic behaviour of the propeller/shafting system causes propeller

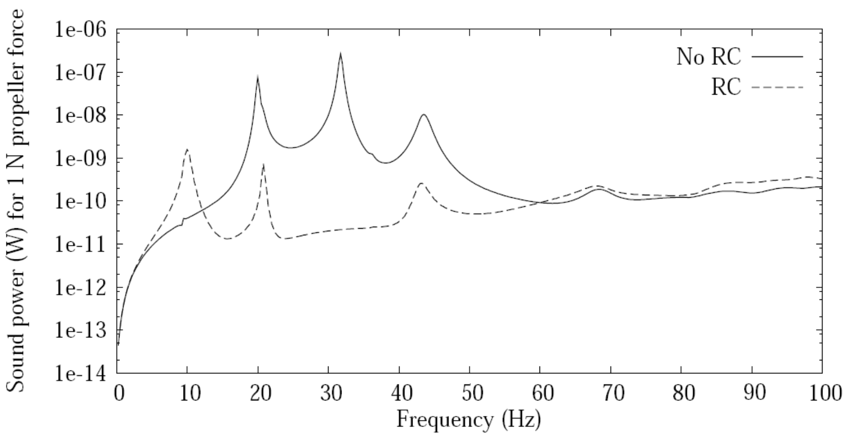

Figure 12. Effect of the RC with propeller mass = 20 tonnes

vibration that generates additional sound radiation from the propeller. The tail cone of the submarine was represented as a rigid structure connected to the pressure hull. This was shown to be a reasonable approximation to a flexible free flooded cone at low frequencies, but the effects of dipole forces near the propeller on overall sound radiation may be underestimated at higher frequencies. The effectiveness of a vibration attenuation system known as a resonance changer in the propeller/shafting system has been examined.

It was found that the performance of the RC is affected significantly by the influence of the dipole fields due to both hydrodynamic forces, which are always present, and propeller vibration. The sound radiation at the resonant frequency of the propeller/shafting system tends to be increased when an RC is used. The two dipole fields may reinforce or partially cancel each other, depending on frequency and parameter selection. It is shown how the radiation, with and without the RC, is influenced by the propeller mass. In general, the RC can provide a significant reduction in overall radiated sound power at low frequencies, including propeller blade passing frequency, but will have a much more limited effect at higher frequencies where the dipole fields tend to be the dominant cause of hull vibration excitation.

Table 1. Propeller/shafting system data

Parameter	Value
Propeller structural mass (kg)	10,000
Propeller added mass of water (kg)	11,443
Shaft Young's modulus (GPa)	200
Shaft Poisson's ratio	0.3
Shaft density (kg/m^3)	7,800
Shaft length (m)	10.5
Shaft cross-sect. area (m^2)	0.071
Effective shaft length (m)	9
Bearing mass (kg)	200
Bearing stiffness (MN/m)	20,000
Bearing damping (kg/s)	300,000
Resonance changer mass (kg)	1,000
Resonance changer stiffness (MN/m)	169
Resonance changer damping (kg/s)	287,000
Foundation major radius (m)	1.25
Foundation minor radius (m)	0.52
Foundation half angle (deg)	15
Foundation thickness (mm)	10
Foundation Young's modulus (GPa)	200
Foundation density (kg/m^3)	7,800

Table 2. Hull data

Parameter	Value
Cylinder length (m)	45
Cylinder radius (m)	3.25
Shell thickness (m)	0.04
Stiffener cross-sect. area (m2)	0.012
Stiffener spacing (m)	0.5
Young's modulus (GPa)	210
Poisson's ratio	0.3
Density (kg/m3)	7,800
Structural loss factor	0.02
Added mass (kg/m2)	796
Stern lumped mass (kg)	188,000
Bow lumped mass (kg)	200,000
Cone half angle (deg)	24
Cone length (m)	9.079
Cone smaller radius (m)	0.3
Density of fluid (kg/m3)	1,000
Speed of sound (m/s)	1,500

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