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EXCITATION OF A SUBMARINE HULL BY PROPELLER FORCES

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Abstract

This paper begins with a review of the unclassified literature relating to the radiated sound-field of a submarine. Only sound generation induced by the propeller is considered. It is of interest to investigate excitation of the submarine hull due to fluctuating forces from the propeller that are transmitted to a submerged hull via both the external pressure field and the propeller shaft. This will assist in predicting the resulting underwater far-field pressure of the submarine due to direct sound radiation from the propeller and its hull vibration. The submarine hull is often modelled as a thin-walled cylinder with appropriate endcaps or, alternatively, as an ellipsoid. The hull deflection shapes excited by the vibratory shaft forces correspond primarily to the *accordion* modes. There is little unclassified work on the excitation of a submarine hull by the radiated sound field of the propeller. Some work has been done in order to determine the exciting forces acting on the hull without considering its vibrational response and fluid interaction. However, most of this work does not take into account the compressibility of the fluid. Subsequent research requires the development of meaningful models for the investigation of the combined effects and interactions of the exciting forces taking account of fluid compressibility and fluid-structure interaction. For this work, numerical methods such as finite element and boundary element methods (FEM, BEM), computational fluid dynamics (CFD) and their combination are of relevance. A fully coupled FEM/BEM model has been developed to investigate the the excitation of a submarine hull through the fluid and the shaft taking fluid compressibility into account. For a realistic hull, the induced vibration due to fluid forces was found to be between 10% and 50% of the vibration due to the shaft force.

1. INTRODUCTION

The sound signature of a submarine is a combination of broadband noise and tonal noise, where broadband noise is mainly due to flow and cavitation and tonal noise is due primarily to internal machinery and the propulsion system [1]. An operational submarine is subject to detection by passive sonar, thus it is desired to minimise the radiated sound. Submarines are usually operating at large depths where cavitation does not occur because of the high water pressure. Flow noise is moderated

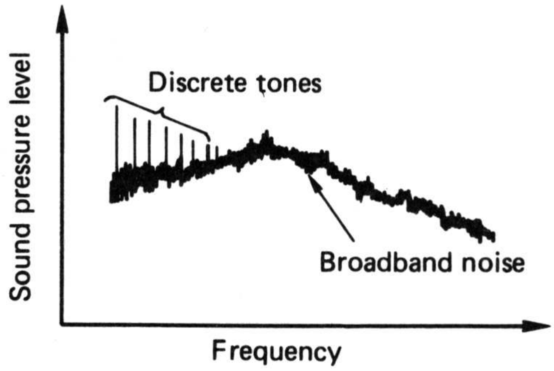

Figure 1. Non-cavitating noise of a marine propeller [3].

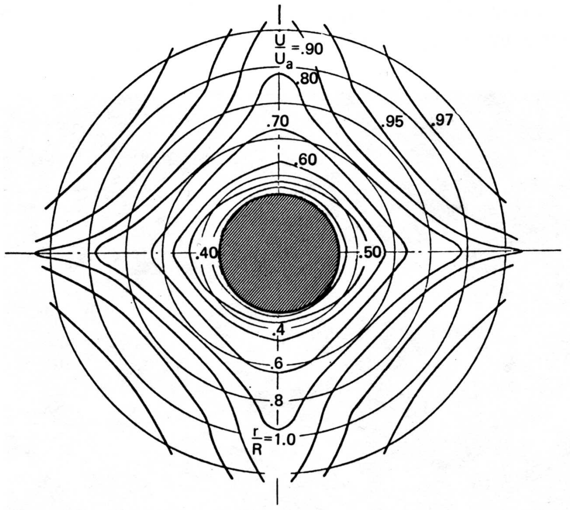

Figure 2. Wake of a torpedo [5].

by travelling at low speeds. In this case, the sound signature of a submarine is dominated by tonals associated with hull vibrations and propeller noise, where the hull vibrations are primarily caused by the propeller. The propeller forces are mainly due to unsteady blade loading of the propeller, operating in a spatially non-uniform wake. The mechanical excitation of a submarine from its propeller-shafting system can be reduced by a resonance changer, which acts as a hydraulic dynamic vibration absorber. This approach, however, may increase the sound power directly radiated by the propeller [2]. Figure 1 shows the non-cavitating noise spectrum of a marine propeller [3]. It is a combination of broadband noise and discrete tones, where the latter occur at the blade passing frequency (bpf) and its multiples. Usually, the highest tonal sound levels are emitted at bpf and its low multiples. The tonal levels due to propulsion tend to rise markedly with speed, while the associated frequencies are proportional to propeller shaft speed.

In general, tonal rotor noise is due to (i) steady, rotating sources and (ii) unsteady fixed sources [4]. The steady rotating sources result in Gutin noise and thickness noise, where dipole Gutin noise is associated with blade loading, and monopole thickness noise with displacement of fluid by the blades. Gutin noise in the far field is thought to be negligible, when the tip speed has a Mach number $M \ll 1$ [5]. It should be noted that both thickness noise and Gutin noise do not involve fluctuating shaft forces. Close to the propeller, the effects of fluid compressibility are small and the pressure field is hydrodynamic in nature rather than acoustic [6]. Blade loading was considered frequently by some authors as an important factor regarding the excitation of nearby boundaries [7, 8]. Thickness effects are almost always considered in conjunction with ship hull excitation. Early work finds comparable magnitudes of fluctuating pressure caused by the thickness and loading effects in the near-field of the propeller [9], but the associated pressure field decays rapidly with distance so that the associated disturbing forces tend to be small.

The influence of the unsteady sources on the fluctuating pressure-field of the propeller is usually much larger than that of the steady rotating sources [5]. The unsteady forces occur primarily due to the fact that the propeller of a marine vessel operates in a spatially non-uniform wake, where the incident angle and velocity of the fluid depend on the radial and angular position on the propeller plane. A typical wake of a torpedo with four fins is shown in Figure 2, where U is the undisturbed velocity of the fluid and U_a is the wake velocity. The wake is shown for radii extending to 1.2 times the propeller tip radius. Rotation in the wake causes a fluctuating load on the blades, resulting in fluctuating shaft forces and dipole sound radiation at the blade passing frequency (bpf) and its harmonics. Turbulence in the wake introduces random flow velocities that lead to additional broadband noise.

Seol *et al.* [10] investigated the pressure field of a realistic ducted/unducted propeller subject to a non-uniform inflow. The velocity potential in the wake and on the blade surfaces is obtained first by using the panel method based on Green's third identity. The pressure on the surfaces is then obtained by applying Bernoulli's equation. The resulting pressure field is computed by solving the Ffowcs Williams-Hawkings equation, where the source terms represent monopoles and dipoles. Here the monopoles were associated with thickness noise and the dipoles with noise due to unsteady blade loading. The results predict that the noise due to unsteady blade loading is about ten times greater in magnitude than the noise due to blade thickness for the unducted propeller.

Tsakonas and Breslin [11] developed formulae to determine the force due to the axial component of the fluctuating pressure field acting on an ellipsoid of revolution. They concluded that the force depends strongly on the axial propeller clearance and the slenderness of the ellipsoid of revolution. Furthermore, it was predicted that the force exerted on the tail fins is only a small percentage of the force acting on the hull. Chertok [12] computed the ratio of steady and vibratory forces on a submarine hull transmitted via the fluid and the propeller shaft from the propeller. The hull had a conical tail. The fluid was assumed to be incompressible. He predicted that the vibratory force component transmitted through the fluid is only 6-8% of that transmitted through the shaft.

Rath Spivack *et al.* [4] used a boundary element method to compute forces on rigid submerged and floating ellipsoidals of revolution. Rotating volumes and forces represent thickness of the propeller and steady blade loading, respectively. Dipole sources fixed at the propeller hub represent net fluctuating tailshaft forces. The hulls were modelled as rigid surfaces. It was concluded that traditional hydrodynamic models tend to underestimate the force acting on the hull surface. Kinns *et al.* [6] computed the hull force due to axial and vertical dipoles at the propeller hub of a submarine, where the submarine is represented by a rigid cylinder. It was shown that the hull forces due to transmission via the fluid can be of the same order of magnitude as the forces transmitted through the propeller shaft, in the frequency range containing higher multiples of bpf. This is about ten times larger than proposed in [12], where infinite speed of sound was assumed.

A submarine hull can be simplified as a ring-stiffened cylinder [1]. The bulkheads are represented by circular plates. In the simplest case, the ends of the cylinder are modelled as flat plates. More realistic end closures are a hemisphere at the bow and a truncated cone at the aft. However, even more parameters then have to be considered in modelling of the structural behaviour of a submarine hull. Dylejko [2] modelled the ballast tank and casing as lumped masses at the cylinder ends. The surrounding water was treated as an added mass at the cylinder, acting in the radial direction on the shell, whereas the on-board machinery was also considered as an added mass, but acting only in the axial direction. Figure 3 shows the simplified physical model of a submarine hull [13].

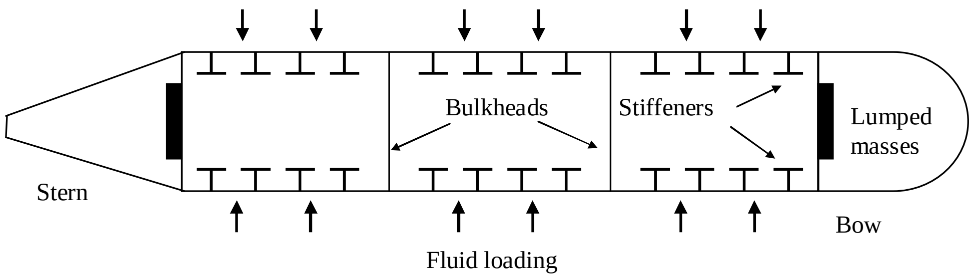

Figure 3. Simplified model of a submarine [13].

The modes of a closed finite cylinder, which are most important for sound radiation, are the *accordion* or *breathing* modes [5, 12]. In this case the movement of the cylinder is axisymmetric and the structural behaviour is similar to that of a solid rod. Those modes involve strong motion of the cylinder ends, but also motion of the cylinder surface normal to the fluid. In submarines, the axial shaft forces can be attenuated by a resonance changer (RC), acting as a dynamic vibration absorber. The RC consists of a piston, an oil reservoir and a pipe connecting those two elements. Dylejko *et al.* [15] used the transmission matrix method to model the propeller shafting system. The RC parameters were optimised for minimisation of the force and power transmission through the propeller shafting system.

Of interest is the relative magnitude of the hull response to excitation through to the fluid relative to response due to excitation by the shaft, with and without an RC. A simple BEM/FEM model was developed to compute the response of a submarine hull to an axial dipole at the propeller location, which represents the effects of unsteady fluid-loading of the propeller combined with a fluctuating shaft force at the propeller hub. The excitation of the hull without a resonance changer is presented.

2. A SIMPLE MODEL OF A SUBMARINE HULL

A fully coupled FEM/BEM model using ANSYS/Sysnoise and mode superposition [16] was set up for the presented results, where the submarine was modelled initially as a finite cylinder with flat, rigid endplates. As a variation, the aft endplate was replaced by a rigid truncated cone. The submarine also featured internal ring-stiffeners of rectangular cross-section and two evenly spaced bulkheads. The ballast tank and casing were represented by a lumped mass of 200 tonnes at each end of the submarine. The on-board machinery was considered as a distributed mass which has been added to the cylinder shell. For the FE model, linear axisymmetric shell elements were used for the hull and stiffeners, and mass elements for the lumped masses. An undamped modal analysis was then conducted to determine the structural behaviour of the hull. The direct BEM using Galerkin collocation was applied in the frequency domain for solving the Helmholtz equation. It was solved for the surface pressure on the hull and the displacement vector in modal coordinates, where mode shapes up to a frequency to $150\,\mathrm{Hz}$ were considered. A modal damping of 0.02 was applied for all mode shapes.

The submarine hull was excited by either a shaft force applied to the endplate or an axial dipole at a specified distance from the aft end of the hull. This distance was one cylinder radius for the flat endplate. In the case of the truncated cone, the dipole was positioned either one cylinder radius ($3.25\,\mathrm{m}$) or $1\,\mathrm{m}$ from the aft end of the cone. This shows the effect of propeller hub location on the forces transmitted to the hull through the fluid. Model data is given in Table 1. The dipole field is related to the shaft force by [5]

$$p = \frac{i\omega F_0}{4\pi r c_0}\, e^{i(\omega t - kr)} \left(1 - \frac{i}{kr}\right) \cos\theta, \tag{1}$$

where p is the acoustic pressure, ω is the circular frequency, F_0 is the shaft force, k is the acoustic wavenumber, r is the distance from the source, c_0 is the speed of sound and θ is the angle of the field point vector with respect to the source point to the cylinder axis.

Table 1. Model data.

Parameter	Value	Unit
Cylinder length	45.0	m
Cylinder radius	3.25	m
Shell thickness	0.04	m
Stiffener cross-sectional area	0.012	m^2
Stiffener height	0.15	m
Stiffener width	0.08	m
Stiffener spacing	0.5	m
Young's modulus of structure	210	GPa
Poisson's ratio of structure	0.3	

Parameter	Value	Unit
Density of structure	7,800	$\mathrm{kg/m^3}$
Structural loss factor	0.02	
Lumped mass	200	t
Distributed mass	1,000	$\mathrm{kg/m^2}$
Cone height	4	m
Cone smaller radius	1	m
Density of fluid	1,000	$\mathrm{kg/m^3}$
Speed of sound	1,500	$\mathrm{m/s}$
Exciting force	1	N

2.1 Flat endplates

The drive point response functions at the aft end due to the shaft force and the dipole at $3.25\,\mathrm{m}$ are shown in Figure 4 for the cylinder with the flat aft endplate. There is a general trend for the difference between the displacement due to shaft and dipole forces to reduce with frequency. The first three resonances occur at $20.1\,\mathrm{Hz}$, $41.6\,\mathrm{Hz}$ and $65.6\,\mathrm{Hz}$. The displacement due to the dipole is about 12% of that due to the shaft force at the first resonance frequency, but increases at the higher resonance frequencies. The hull deformations are depicted in Figure 5.

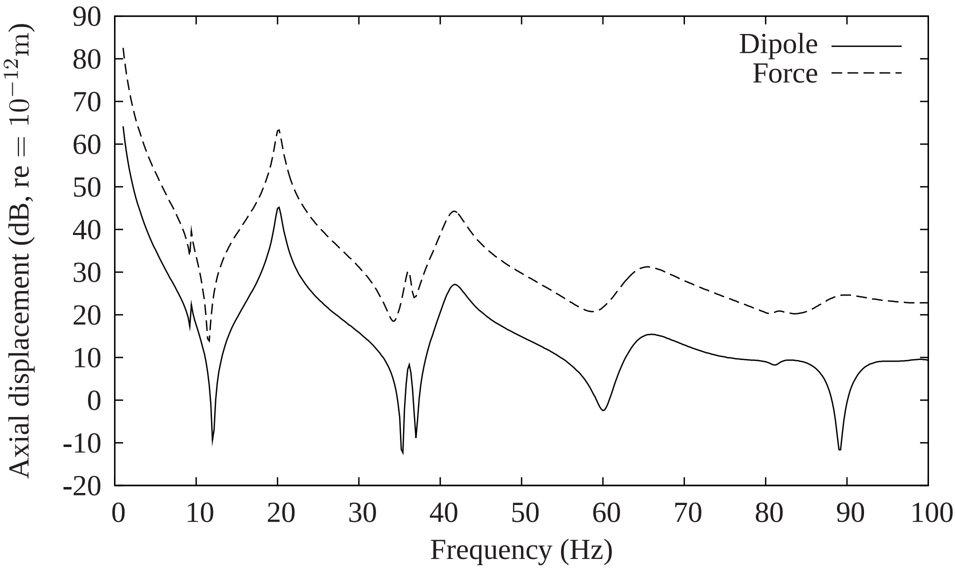

Figure 4. Axial displacement at the aft of the submarine with flat endplates due to a dipole and a force.

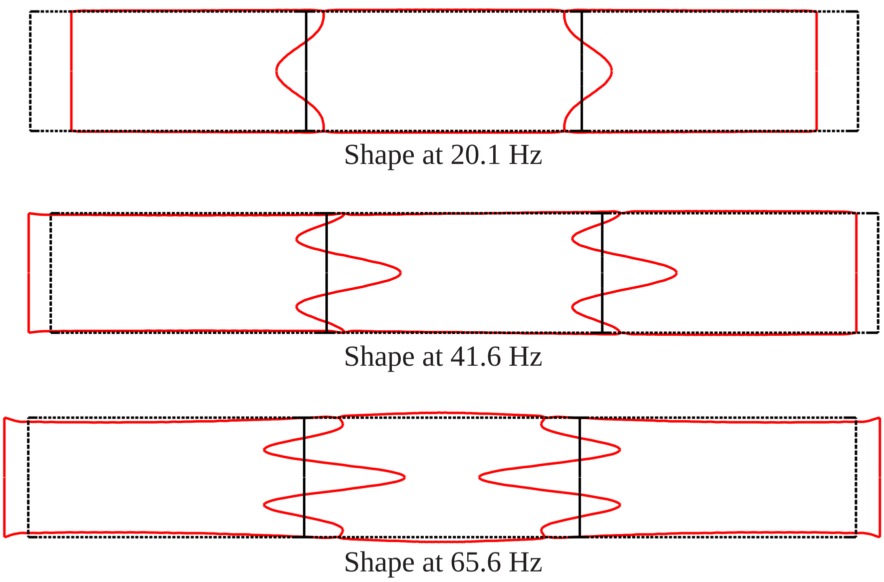

Figure 5. Hull shapes at the resonance frequencies.

Figure 6 shows the pressure amplitude along the cylinder surface of the submarine hull at the first axial resonance frequency under an axial dipole excitation at the aft end. Also shown on the same figure is the surface pressure for a rigid cylinder of the same dimensions as the submarine model under the same acoustic dipole excitation. It can be concluded that the excitation of the shell occurs mainly at the aft of the submarine for the frequency, since the pressure on a rigid cylinder decreases rapidly with distance from the source. In the case of an excited structure, there is also a significant pressure at the bow, involving increased overall sound radiation.

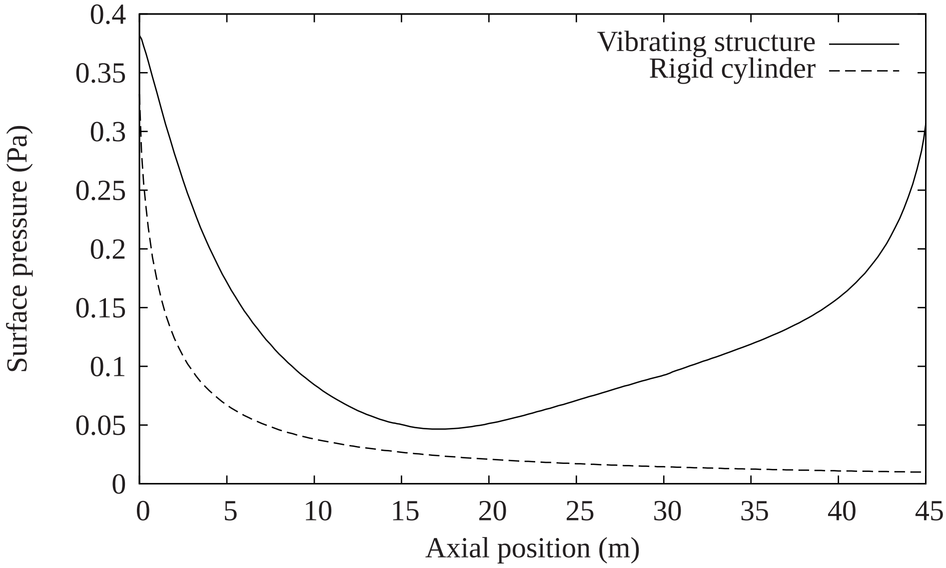

Figure 6. Pressure amplitude on the shell surface at 20.1 Hz due to a dipole on a rigid cylinder and an excited submarine hull.

2.2 Truncated cone at aft end

A truncated cone was used instead of a flat endplate to model the submarine more accurately. The $3.25\,\mathrm{m}$ distance between the dipole and the aft end of the cone is considerably larger than the distance of the propeller hub from the end of the cone in a real submarine, so the effects of fluid forces on the hull will tend to be reduced relative to the effect of shaft forces.

Figure 7 shows a comparison of the displacements for the flat endplate and the truncated cone, due to axial dipole excitation at one cylinder radius from the aft end. It can be seen that at the first three resonance frequencies, the displacement with the conical end is greater than with the flat endplate. It is assumed that this is an effect of the increased area, where the surface normal vector has a component in the axial direction. Furthermore the frequencies are slightly decreased. This might occur due to the greater mass of water adjacent to the additional area.

The effect of moving the propeller hub closer to the end of the cone is shown in Figure 8. Here, the axial displacement due to the shaft force applied at the forward end of the truncated cone is compared with the axial displacement due to the fluid forces for dipole locations $3.25\,\mathrm{m}$ and $1\,\mathrm{m}$ from the aft end of the truncated cone. The dipole force causes significant vibration relative to the shaft force, for realistic separations of the propeller and the submarine hull structure.

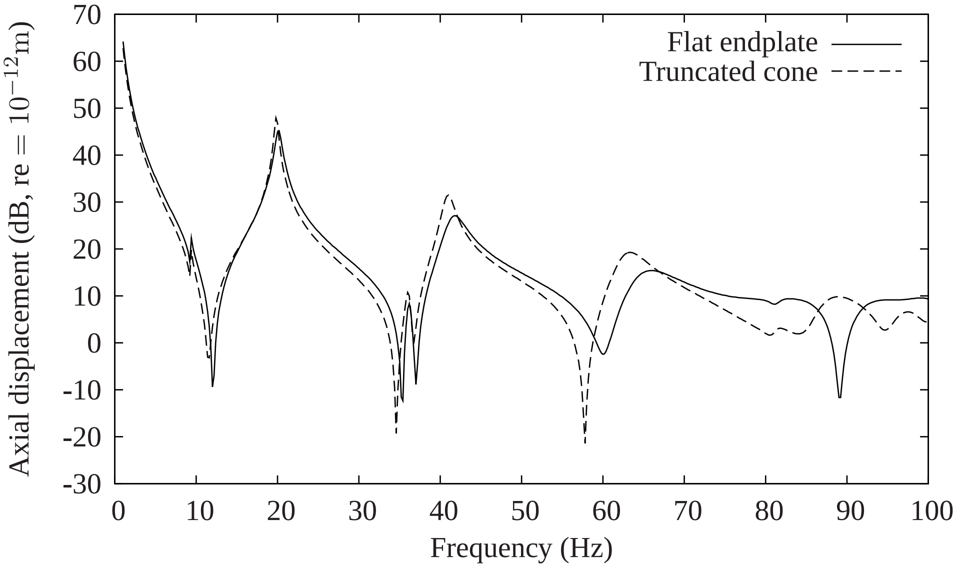

Figure 7. Axial displacement due to a dipole at $3.25\,\mathrm{m}$ distance from the aft of the submarine with different end closures.

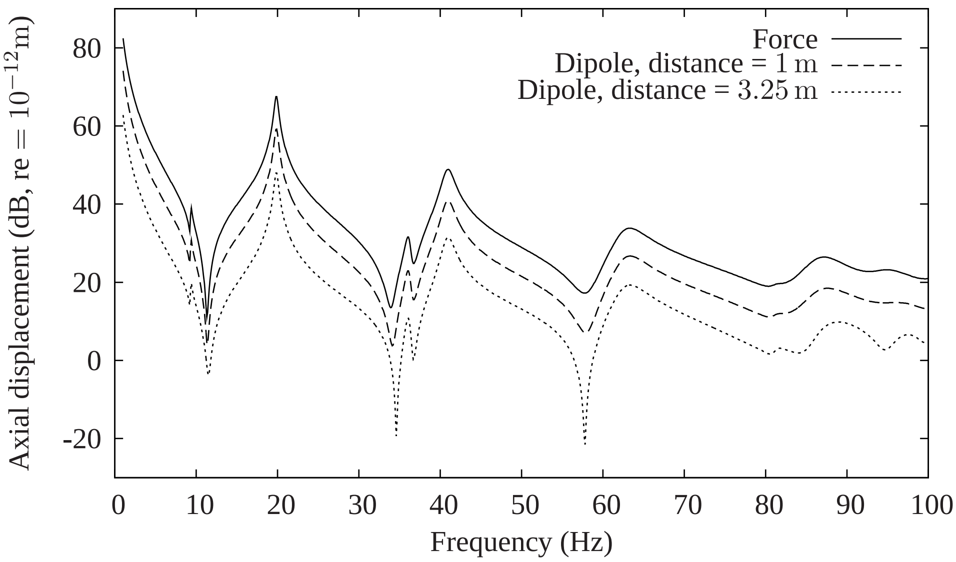

Figure 8. Axial displacement due to a dipole at the aft of the submarine with a truncated cone end under different excitations.

3. CONCLUSIONS

A review of the unclassified literature in respect of induced excitations of a submarine hull has been given. Some preliminary work is presented on excitation of a submarine hull via the shaft and through the fluid for a realistic hull shape. Earlier work dealing with acoustic excitation used a rigid representation of the submarine hull to compute the forces which act on the hull surface. A simple submarine model was therefore developed to estimate the order of magnitude of the acoustic excitation of the hull, where the induced hull vibration is taken into account. For the present low frequency model, vibration response due to excitation through the fluid is between 10% and 50% of the response due to excitation through the shaft. In a real submarine the shaft force may be attenuated by a resonance changer, exposing the effect of the fluid forces. It is assumed that the excitation of the submarine through the fluid and the consequent sound radiation from the hull is therefore of major importance. The development of a fully coupled submarine model will be investigated in future work to estimate the combined effect of fluid and shaft forces. The resonance changer will also be included to predict the contribution of the acoustic excitation, if the shaft forces are attenuated. Furthermore, it is of interest to investigate if the resonance changer has an influence on the dipole strength.

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