Effects of mass distribution and buoyancy on the sound radiation of a fluid loaded cylinder

Hongjian Wu, Herwig Peters, Roger Kinns and Nicole Kessissoglou

School of Mechanical and Manufacturing, University of New South Wales, Sydney, Australia

ABSTRACT

Earlier studies have shown that the radiated sound power due to excitation of bending modes of a submerged cylindrical body depends strongly on the relative distributions of mass and buoyancy. A fluid-loaded cylindrical shell closed at each end by hemispherical shells and driven by transverse forces is examined. The effects of stepwise mass distribution in the axial direction of the cylindrical shell on the radiated sound power are observed. The influences of different buoyancy values and thickness of the shell are also investigated.

INTRODUCTION

Fluid loaded cylindrical shells can be found in a variety of engineering applications such as pressure vessels, autonomous underwater vehicles and submarine hulls. A submerged vessel experiences heavy fluid loading from the surrounding water. Heavy fluid loading alters the dynamic characteristics of the submerged structure (Junger, 1975). A range of analytical and numerical techniques have been used to predict the structural and acoustic responses of fluid loaded cylinders (Zhang et al., 2002; Merz et al., 2007; Tong et al., 2007). At low frequencies, the radiated sound field from a fluid loaded cylindrical shell is mainly dominated by the lowest order circumferential modes (Caresta and Kessissoglou, 2010).

Global modes of submerged structures are sensitive to internal mass distribution. The internal mass has been rigidly attached along the length of cylindrical hull (Crocker, 1998) or mounted flexibly on the cylindrical hull as dual spring-mass systems (Peters et al., 2012). The degree of internal mass isolation alters the global structural modes and consequent radiated sound power. The interaction between the structural waves in the internal structures and the cylindrical shell causes significant changes in sound scattering from the internally loaded cylindrical shell (Guo, 1992). The influence of added local mass on the sound radiation from a finite thin cylindrical shell was experimentally investigated by Ekimov and Lebedev (1996). They showed that a very small amount of added mass corresponding to only 1% of the total shell mass can increase the radiated sound power by up to 20 dB.

In an underwater vehicle, ballast tanks are used to alter the vessel buoyancy. When dived, neutral buoyancy is achieved by flooding tanks and spaces external to the pressure hull and by adusting the amount of water in internal ballast tanks according to depth. On the surface and in an emergency operation, positive buoyancy is achieved using air to eject water from the external ballast tanks. The buoyancy on the surface can vary between as little as 10% and as much as 20% depending on its size and shape (Burcher and Rydill, 1995). When the structure is submerged, the water in external ballast tanks and spaces can be regarded as part of the surrounding sea, so that the hull and its internal masses has positive buoyancy. In a simplified model, the adjustment of buoyancy is realised by subtracting or adding mass from the cylindrical shell.

In this paper, a fluid-loaded cylindrical shell closed at each end by hemispherical shells is examined. A fully coupled finite element/boundary element (FE/BE) model is developed using Nastran/Patran for the FE model, and SYSNOISE for the BE model. The excitation load is a transverse ring force applied at one end of the cylindrical shell. The effects of stepwise mass distribution in the axial direction of the cylindrical shell on the radiated sound power are observed. The influence of different buoyancy values and thickness of the shell are also investigated.

NUMERICAL MODEL

A fully coupled finite element/boundary element (FE/BE) model of the fluid-loaded cylindrical shell closed at each end by hemispherical shells is developed as follows.

Finite element analysis

The finite element model was developed in Patran/Nastran. A modal analysis was initially conducted to determine the free vibrational response. The following assumptions were used: (i) a linear analysis was conducted, (ii) the mass and stiffness properties of the structure were constant; (iii) damping was not considered. The equation of motion for a system in free vibration is given by (Bathe, 1982)

$$\mathbf{M}\ddot{\mathbf{q}} + \mathbf{K}\mathbf{q} = 0 \quad (1)$$

where $\mathbf{M}$ is the global mass matrix and $\mathbf{K}$ is the global stiffness matrix. $\ddot{\mathbf{q}}$ and $\mathbf{q}$ are the nodal acceleration and displacement vectors, respectively. Substituting a general solution for harmonic motion into the equation of motion results in a classic eigenvalue problem from which the natural frequencies and corresponding modes of vibration of the system can be obtained. In Fig. 1, a finite element mesh of the cylinder and hemispherical end caps was created in Patran/Nastran from which the structural uncoupled modes were obtained. Two-dimensional shell elements were used to model the cylinder and the hemispherical end caps. The thickness of shell was small compared with the dimensions of the structure. Quad8 shell elements were used to model the structure.

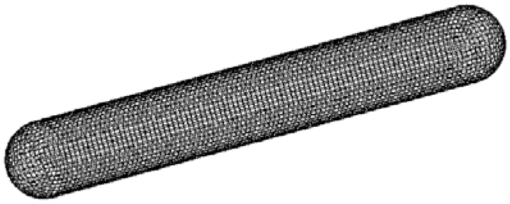

Figure 1. Finite element mesh of the cylindrical shell with hemispherical end caps

Boundary element analysis

A fully coupled FE/BE model of the boundary and the surrounding fluid was built using SYSNOISE. Only the exterior unbaffled surrounding fluid was considered. Using the acoustic BEM direct response approach, the solution for the displacement vector and surface pressure can be obtained by (Peters et al. 2012)

$$\begin{bmatrix} \mathbf{K}-\omega^2\mathbf{M} & -\mathbf{C}_{sf} \\ -i\omega\mathbf{G}\mathbf{C}_{fs} & \mathbf{H} \end{bmatrix} \begin{bmatrix} \mathbf{u}_s \\ \mathbf{p}_f \end{bmatrix} = \begin{bmatrix} \mathbf{f}_s \\ \mathbf{f}_f \end{bmatrix} \tag{2}$$

where $\mathbf{f}_s$ and $\mathbf{f}_f$ are respectively the nodal structural forces and forces due to fluid loading acting on the surface of the structure. $\mathbf{u}_s$ and $\mathbf{p}_f$ are nodal displacement and nodal pressure vectors, respectively. $\mathbf{H}$ and $\mathbf{G}$ are the frequency dependent boundary element influence matrices. $\mathbf{C}_{sf}$ and $\mathbf{C}_{fs}$ are coupling matrices due to the two coupling conditions corresponding to (i) equilibrium of the acoustic pressure and normal stress at the wet interface, and (ii) continuity of the fluid particles and structural nodal velocity normal to the wet surface, that is

$$\mathbf{f}_f = \mathbf{C}_{sf}\mathbf{p}_f \tag{3}$$

$$\dot{\mathbf{u}}_f = i\omega\mathbf{C}_{fs}\mathbf{u}_s \tag{4}$$

The radiated sound power is then computed using

$$P = \frac{1}{2}\Re\left\{\int_{\Lambda} p\,\mathrm{v}^*\,d\Lambda\right\} \tag{5}$$

where p is the acoustic pressure of the fluid, v is the fluid particle velocity on the wet surface of the structure, * denotes the conjugate complex, $\Re$ denotes the real part of, and the fluid-structure interface surface area is Λ.

RESULTS

The dimensions and material properties of the cylindrical hull as well as the fluid parameters are listed in Table 1. The hemispherical end closures are modelled using the same material properties as used for the cylindrical shell. The thickness of hemispherical shells is also the same as for the cylindrical shell. The total mass of displaced water is 1637 tonnes. The mass of the cylindrical hull and hemispherical end closures is approximately 307 tonnes for a thin shell and 614 tonnes for a thick shell. In this work, the buoyancy rate is adjusted by adding or subtracting mass from the structure.

A transverse ring force is applied at the junction between the cylindrical shell and a hemispherical cap at one end, as shown in Fig. 2. The magnitude of total force is 1N.

Table 1. Parameters of the fluid-loaded shell

Parameter	Value
Length of cylindrical shell	45 m
Overall length of shell	51.5 m
Diameter of cylindrical shell	6.5 m
Thickness of thin shell	0.04 m
Thickness of thick shell	0.08 m
Young's modulus	210 GPa
Poisson's ratio	0.3
Density of steel	7800 kg/m^3
Added density for a thick cylinder with neutral buoyancy	11660 kg/m^3
Added density for a thin cylinder with neutral buoyancy	31120 kg/m^3
Density of water	1000 kg/m^3
Speed of sound in water	1500 m/s

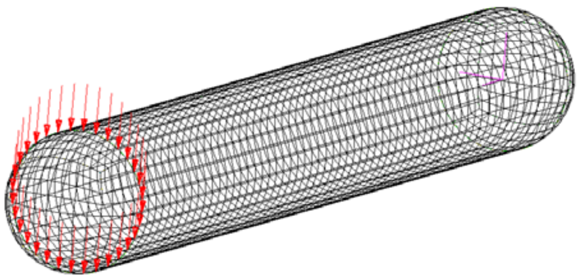

Figure 2. Transverse excitation at one end of the cylindrical shell with hemispherical end caps

Influence of buoyancy on the radiated sound

At low frequencies, the wavelength of the radiated sound is very large compared to the cylindrical hull dimensions. Hence, the resulting motion is considered as rigid body motion of a dipole nature (Ross, 1987). Physically, the acoustic dipole can be obtained by analysing the radiation from a fluid-loaded rigid sphere excited by a unit point force. The radiated sound power P_d is calculated by (Ross, 1987)

$$P_d = \frac{\omega^2 F_{rms}{}^2}{12\pi\rho_0 c_0^3} \tag{6}$$

where F_{rms} is the root mean square force amplitude, ω is the radian frequency, and ρ_0, c_0 are respectively the density of water and speed of sound in water. In what follows, the acoustic dipole is used as the reference source to calculate the normalized sound power.

In this work, neutral buoyancy (0% buoyancy) and 20% buoyancy are considered. The radiated sound power relative to a dipole for the two buoyancy cases is shown in Figs. 3 and 4 for thin and thick cylindrical shells, respectively, due to transverse force excitation as shown in Fig. 2. The resonant peaks correspond to successive circumferential bending modes of the cylindrical shell. At very low frequencies up to around 5 Hz, the normalized sound power levels for both thin and thick shells is very close to 0 dB which indicates that the fluid-loaded cylinder can be approximated as a dipole-like acoustic source. Increasing the buoyancy from 0% to 20% by reducing the mass of the cylindrical shell results in an increase in the resonant frequencies of the cylinder bending modes. This effect is particularly evident at frequencies below 50 Hz, where there is a shift to the right in the resonant frequencies from 0% to 20% buoyancy. At very low frequency below 5 Hz, the dB offset is noticeable when the buoyancy is 20%. This is attributed to the fact that only rigid body modes contribute to sound radiation.

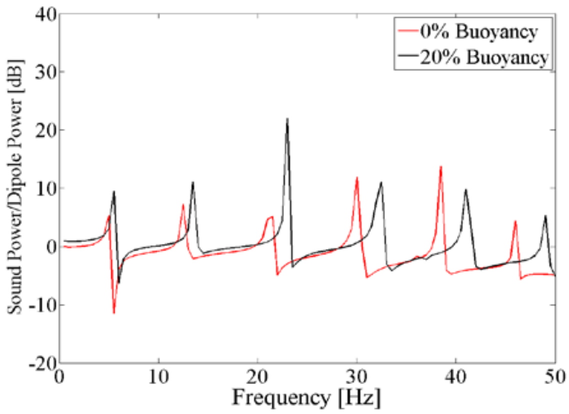

Figure 3. Sound power for transverse excitation of a fluid-loaded thin cylindrical shell with different buoyancies

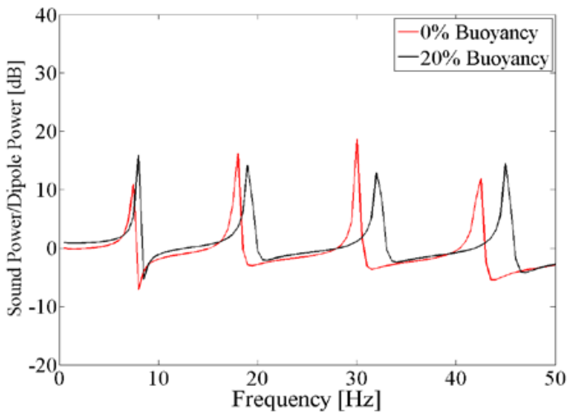

Figure 4. Sound power for transverse excitation of a fluid-loaded thick cylindrical shell with different buoyancies

Effects of stepwise mass distributions on the radiated sound

In order to investigate the effects of stepwise mass distributions on the radiated sound power, the cylindrical shell with hemispherical end closures is divided into three sections as shown in Fig. 5. The length of the central section is $0.5(L+D)$ and corresponds to half of the total shell length. The remaining two sections at each end of length $0.25(L+D)$ combined comprise the remaining half of the surface area of the total shell. In this case, the surface area of the central section equals the summation of the surface areas of two end sections. The overall mass of the shell is equal to the surrounding water in which the shell is neutrally buoyant. To implement the stepwise mass distribution, a decrease of 20% of mass in the central section is supplemented by an increase of 20% of mass in the two end sections.

In Table 2, α represents a percentage increase or decrease of the mass of a given section of the cylindrical shell. Three different cases of mass distribution along the axial length of the cylinder are compared to investigate the influence of the mass distribution on the radiated sound power. These are listed in Table 2 and correspond to (1) uniform mass distribution in which there is no increase or decrease in mass in any section of the cylindrical hull; (2) 20% increase of mass in the central section with a corresponding 20% decrease of mass in each end section; (3) 20% decrease of mass in the central section with a corresponding 20% increase of mass in each end section.

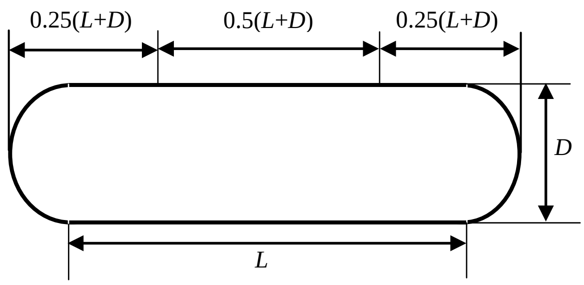

Figure 5. Cylindrical shell with hemispherical end closures divided into three sections for the stepwise mass distribution

Table 2. Stepwise mass distribution in the three sections of the cylindrical shell

Mass distribution	Left	Central	Right
Case 1	$\alpha_1 = 0$	$\alpha_2 = 0$	$\alpha_3 = 0$
Case 2	$\alpha_1 = -20\%$	$\alpha_2 = 20\%$	$\alpha_3 = -20\%$
Case 3	$\alpha_1 = 20\%$	$\alpha_2 = -20\%$	$\alpha_3 = 20\%$

In Figs. 6 and 7, the normalized sound power levels for uniform mass distribution (case 1 in Table 2) and for the two cases of stepwise mass distribution (cases 2 and 3 in Table 2) of a neutrally buoyant cylinder are compared, for thin and thick shells respectively. A structural damping ratio is introduced as 0.02 for both thin and thick cylinders. Both Figs. 6 and 7 show that at low frequencies, the radiated sound power levels are generally higher when mass is concentrated in the central section, particularly at off-peak frequencies. The results also show that the stepwise mass distribution has little effect on the locations of the resonant peaks at low frequencies which correspond to the bending modes of the cylinder. However, the stepwise mass distribution affects the normalised sound power levels. At the first two bending modes, the highest levels are observed when the mass is concentrated in the central section. By contrast, at higher bending modes, the highest levels are observed when the mass is allocated to the end sections.

If there is no structural damping, the only mechanism of energy loss is sound radiation from the structure. In the case of hull bending modes, radiation damping is weak, leading to peaks at resonance that are difficult to resolve. The effects of mass and buoyancy distributions on the radiation efficiency of bending modes become much clearer when structural damping is dominant. Then, the amplitudes of peaks in radiated sound power may be reduced markedly, even when structural damping is very light.

Effects of thickness on the radiated sound

In the design of a conventional underwater vehicle, the ratio between the thickness and radius of the cylindrical hull is approximately around 0.01. The free vibration of a cylindrical shell is sensitive to changes in thickness especially for the case of thinner shells. Hence it is important to study the effects of thickness on the radiated sound power. From Figs. 8 and 9, it is observed that the thickness of the cylindrical hull has significant effects on the radiated sound power for both neutral buoyancy and 20% buoyancy conditions. The natural frequencies of bending modes are increased by the increase of shell thickness especially for the low frequency range considered.

CONCLUSIONS

In this paper, a low frequency numerical model of a submerged hull was presented which can be used to analyze the effects of buoyancy rate and longitudinal mass distribution. The hull was modelled as a fluid-loaded cylindrical shell with hemispherical end closures. A parametric study examines the effects of varying mass distribution, buoyancy values and thickness of shell on the radiated sound power.

It is shown that the natural frequencies of the bending modes of the cylindrical shell are affected by its values of buoyancy and thicknesses for the frequency range considered. The variation of stepwise mass distribution in a neutrally buoyant hull has significant influence on the radiated sound power especially when the reduced mass of the central section is supplemented by added mass in the two end sections.

REFERENCES

Bathe, KE 1982, *Finite Element Procedures in Engineering Analysis*, Prentice Hall, Englewood Cliffs, New Jersey.

Burcher, R & Rydill LJ 1995, *Concepts in submarine design*, Cambridge University Press, London.

Caresta, M & Kessissoglou, NJ 2010, 'Acoustic signature of a submarine hull under harmonic excitation'. *Applied Acoustics*, vol. 71, no. 1, pp. 17-31.

Crocker, MJ 1998, *Handbook of Acoustics*, John Wiley and Sons, Inc., New York.

Ekimov, AE & Lebedev, AV 1996, 'Acoustic signature of a submarine hull under harmonic excitation', *Applied Acoustics*, vol. 48, no. 1, pp. 47-57.

Guo, YP 1992, 'Sound scattering from an internally loaded cylindrical shell', *Journal of the Acoustical Society of America*, vol. 91, no. 2, pp. 926-938.

Junger, MC 1975, 'Radiation and scattering by submerged elastic structures', *Journal of the Acoustical Society of America*, vol. 57, no. 6, pp. 1318-1326.

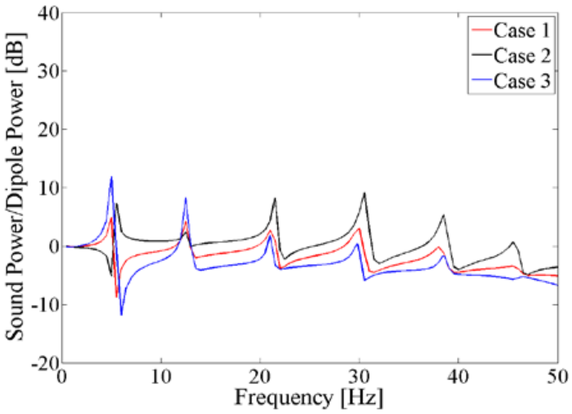

Figure 6. Sound power of a neutrally buoyant thin cylinder with different mass distribution (damping ratio=0.02)

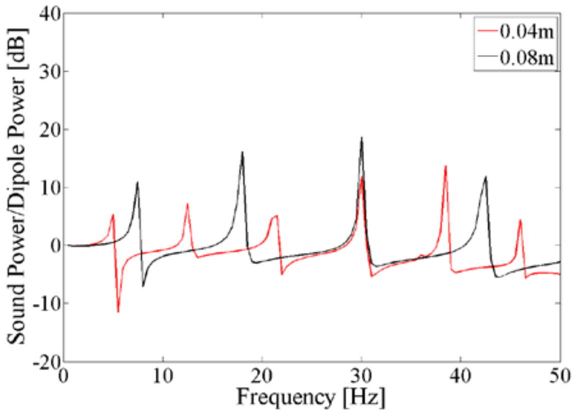

Figure 8. Sound power of a neutrally buoyant cylinder with different thickness

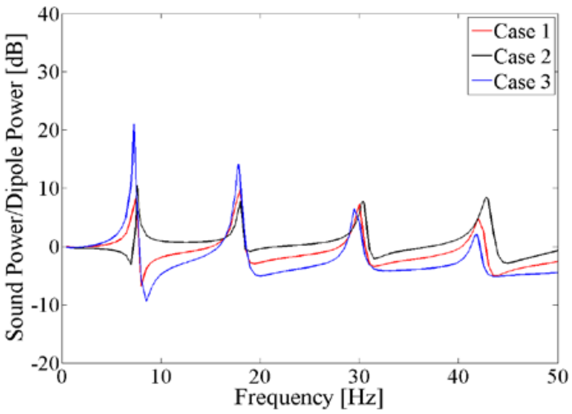

Figure 7. Sound power of a neutrally buoyant thick cylinder with different mass distribution (damping ratio=0.02)

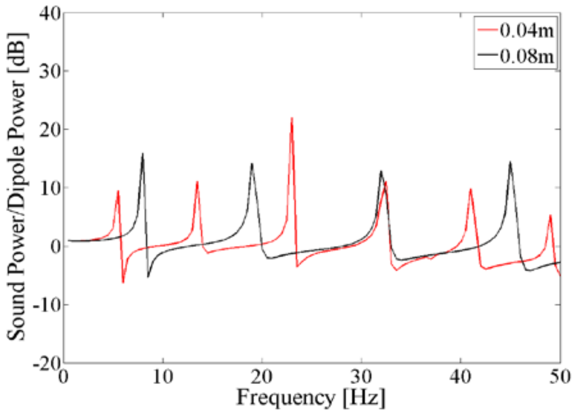

Figure 9. Sound power of a 20% buoyant cylinder with different thickness

Merz, S, Oberst, S, Dylejko, PG, Kessissoglou, NJ, Tso, YK & Marburg, S 2007, 'Development of coupled FE/BE models to investigate the structural and acoustic responses of a submerged vessel', *Journal of Computational Acoustics*, vol. 15, no. 3, pp. 23-47.

Peters, H, Kinns, R, Kessissoglou, NJ & Marburg, S 2012, 'Effect of internal mass isolation of the radiated sound power of a submerged hull', *Proceedings of Acoustics* 2012, Fremantle, Australia.

Ross, D 1987, *Mechanics of Underwater Noise*, Peninsula Publishing, Los Altos, CA.

Tong, Z, Zhang, Y, Zhang, Z & Hua, H 2007, 'Dynamic behavior and sound transmission analysis of a fluid–structure coupled system using the direct-BEM/FEM', *Journal of Sound and Vibration*, vol. 299, no. 3, pp. 645-655.

Zhang, XM 2002, 'Frequency analysis of submerged cylindrical shells with the wave propagation approach', *International Journal of Mechanical Sciences*, vol. 44, no. 7, pp. 1259-1273.