A THEORETICAL SOLUTION OF A WATER- SPHERICAL SHELL-DAMPING LAYER INTERACTION SYSTEM FOR UNDERWATER NOISE REDUCTIONS

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Summary Based on a fluid-structure interaction analysis (FSIA), the theoretical solution is developed for an integrated spherical shell-damping layer-water interaction system subjected to an internal sinusoidal pressure excitation. The effect of damping layer on reducing underwater noises is investigated. The pressure reduction factor is defined to measure the efficiency of noise reduction by the damping layer. Some guidelines to design effective damping layers to reduce underwater noise prolusions are suggested.

As reported by BBC [1], the world's oceans are now filled with noises such that low-frequency ambient marine noise levels have risen in the northern hemisphere by two orders of magnitude over the last 60 years and whales (Fig.1 a) and other marine mammals are killed because of these noises. It has been concluded that main sources of undersea noise are from man-made marine structures searching for oil and gas as well as powerful sonar systems. Large vessels are typically loud vessels and the increase in traffic has fundamentally changed the noise profile of the world's oceans. The Whale and Dolphin Conservation Society in the UK is launching a campaign to deal with this pollution and an action plan is suggested to regulate underwater noise pollution by a global treaty to tackle the problem. Therefore, designers of marine structures also have to take the responsibilities in reducing underwater noise pollution during the design process of these structures. To address this important issue, knowledge and experiences on noise / vibration control [2]-[3] and FSIA [4] are necessary. One of the means often used to reduce noises is using damping materials to absorb energies associated with noises, see, for example [5]-[7]. However, the reported investigations rarely considered integrated FSIA. Due to fluid-structure interactions [4], the dynamic behavior of the integrated system is very different from the one obtained with no couplings. Therefore, it is more accurate to determine an effective means to reduce underwater noise pollution caused by marine structures using FSIA. This paper intends to address this problem through a theoretical investigation of a spherical shell-water interaction system excited by an internal dynamic pressure. The aim is to find out the efficiency of the damping materials on reducing underwater noise pollutions.

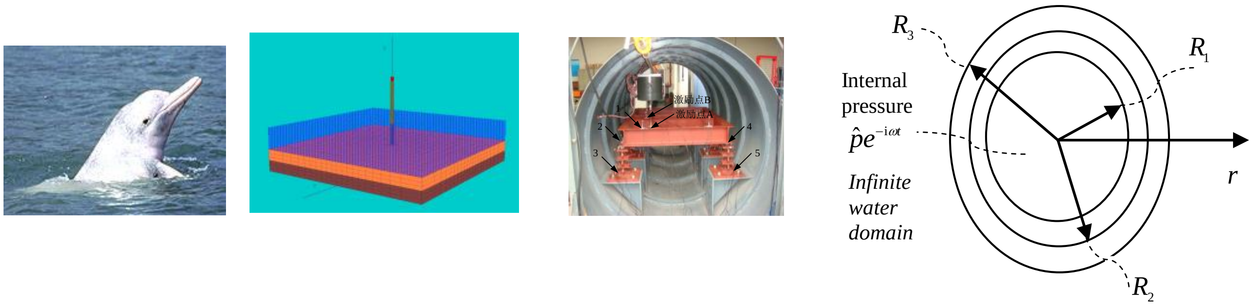

Fig. 1 (a) white whale, (b) FSI numerical model (UoS), (c) Vibration reduction test (SJTU), (d) The theoretical study model

Fig.1 (d) shows a spherical shell of wall thickness $h_1 = R_2 - R_1$, subject to an internal sinusoidal dynamic pressure $\hat{p}e^{-i\omega t}$ and covered by an outside damping layer of thickness $h_2 = R_3 - R_2$, is located in an infinite water domain. The outside shell consists of a uniformly distributed damping material. Since the system is centre-symmetrical, there are no shear deformations in either shells and therefore only bulk strains as well as corresponding pressure transmitting in the solids and fluid exists. The water is assumed compressible and its motion is described by a wave equation. Using a spherical coordinate system fixed at the centre of the shell, the governing equations describing the dynamics of the interaction system have been formulated following the laws in continuum mechanics. Using an index I to identify the inner shell (I =1, $R_1 < r < R_2$), outside damping shell (I =2, $R_2 < r < R_3$) and water domain (I = 3, $R_3 < r < \infty$) as well as considering the center-symmetry, we derive the following equations describing pressure wave propagations in the system

$$p_I = K_I \Theta_I, \qquad [d(r^2 dp_I / dr)/dr]/r^2 = d^2 p_I /(C_I^2 dt^2), \qquad C_I = \sqrt{K_I / \rho_I}. \tag{1}$$

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Here, dynamic pressure p_I is a function of coordinate r and time t, C_I represents a complex speed of sound in continuum I, K_I and Θ_I bulk modulus and deformation of materials, respectively. On the internal boundaries $r = R_1$, the interface $r = R_2$ between two shells and the wet boundary $r = R_3$, the dynamic pressures must satisfy the corresponding coupling conditions or Somerfield radiation condition [8]-[9]

$$p_1 = \hat{p}e^{-i\omega t}, \quad r = R_1; \quad p_2 = p_1, \quad r = R_2; \quad p_3 = p_2, \quad r = R_3; \quad r(dp_3/dr - i\omega p_3/C_3), \quad r \Rightarrow \infty. \tag{2}$$

Based on Bessel equation [10], we obtain the dynamic pressure caused by the internal excitation $\hat{p}e^{-i\omega t}$ of the system in the form

$$\tilde{p}_1(r,t) = \hat{p}R_1 r^{-1} e^{ik_1(r-R_1)} e^{-i\omega t}, \quad \tilde{p}_2(r,t) = \hat{p}R_1 r^{-1} e^{ik_1(R_2-R_1)} e^{ik_2(r-R_2)} e^{-i\omega t}, \quad \tilde{p}_3(r,t) = \hat{p}R_1 r^{-1} e^{ik_1(R_2-R_1)} e^{ik_2(R_3-R_2)} e^{ik(r-R_3)} e^{-i\omega t}. \tag{3}$$

This dynamic pressure is a complex pressure with a phase angle representing its time delay relative to the excitation pressure $\hat{p}e^{-i\omega t}$. The corresponding bulk strain in the system is now represented as

$$\tilde{\Theta}_1(r,t) = \hat{p}R_1 K_1^{-1} r^{-1} e^{ik_1(r-R_1)} e^{-i\omega t}, \quad \tilde{\Theta}_2(r,t) = \hat{p}R_1 K_2^{-1} r^{-1} e^{ik_1(R_2-R_1)} e^{ik_2(r-R_2)} e^{-i\omega t}, \quad \tilde{\Theta}_3(r,t) = \hat{p}R_1 K_3^{-1} r^{-1} e^{ik_1(R_2-R_1)} e^{ik_2(R_3-R_1)} e^{ik(r-R_3)} e^{-i\omega t}. \tag{4}$$

The boundary pressures represented by (3) are continuous but the bulk strains on the boundaries between different materials are dis-continuous due to jumps of bulk moduli and mass densities.

Relative pressure amplitude

Choosing the excitation pressure amplitude as our reference pressure, we define relative amplitude of sound pressure with its decibel expression in the form

$$p_I^R(r) = |\tilde{p}_I| / \hat{p}, \qquad p_I^{RD}(r) = 20\log p_I^R(r). \tag{5}$$

Pressure Reduction Factor

We assume the three complex bulk moduli and complex speeds C_I of sound in (1) as well as wave number k_I as follows

$$K_I = \hat{K}_I e^{-i2\theta_I}, \quad \varepsilon_I = \tan 2\theta_I, \quad C_I = \hat{C}_I e^{-i\theta_I}, \quad \hat{C}_I = \sqrt{\hat{K}_I / \rho_I}, \quad k_I = \hat{k}_I e^{i\theta_I}, \quad \hat{k}_I = \omega / \hat{C}_I, \tag{6}$$

in which θ_I and ε_I are phase angles and damping factors of continuum, respectively, and hats "^" indicate real numbers. As a result of equations (3)-(6), we obtain

$$p_1^R(r) = R_1 r^{-1} e^{-\omega \sin\theta_1 (r-R_1)/\hat{C}_1}, \quad p_2^R(r) = R_1 r^{-1} e^{-\omega[\sin\theta_1 h_1/\hat{C}_1 + \sin\theta_2 (r-R_2)/\hat{C}_2]}, \quad p_3^R(r) = R_1 r^{-1} e^{-\omega[\sin\theta_1 h_1/\hat{C}_1 + \sin\theta_2 h_2/\hat{C}_2 + \sin\theta_3 (r-R_3)/\hat{C}_3}, \quad p_I^{RD}(r) = 20\log(R_1/r) - (20/\log 10)\omega\alpha_I,$$
$$\alpha_1 = \sin\theta_1 (r-R_1)/\hat{C}_1, \quad \alpha_2 = \sin\theta_1 h_1/\hat{C}_1 + \sin\theta_2 (r-R_2)/\hat{C}_2, \quad \alpha_3 = \sin\theta_1 h_1/\hat{C}_1 + \sin\theta_2 h_2/\hat{C}_2 + \sin\theta_3 (r-R_3)/\hat{C}_3. \tag{7}$$

We define a *pressure reduction factor* by the damping layer

$$p_d^{RD}(r) = (20/\log 10)\omega \sin\theta_2 h_2/\hat{C}_2 = 8.96\omega \sin\theta_2 h_2/\hat{C}_2 \text{dB}. \tag{8}$$

This implies that the damping material layer outside the spherical shell reduces the noise pressure by p_d^{RD} dB. To increase the efficiency of the damping layer on underwater noise reduction, we may consider the following guidelines to design the damping layer covering vibrating structures. (i) Increasing the thickness h_2 of damping layer as much as you could; (ii) Choosing a damping material with high damping factor $\varepsilon_2 = \tan 2\theta_2$, so that $\sin\theta_2$ is larger; (iii) Materials with lower bulk modulus and larger mass density have a lower speed of sound $\hat{C}_2$ so that to increase the pressure reduction factor.

The theoretical solution of a water-spherical shell-damping layer interaction system subjected to an internal sinusoidal pressure is presented. Based on this solution, the effect of damping layer on reducing underwater noise pollution is investigated. The pressure reduction factor is defined to measure the efficiency of noise reduction by the damping layer. Investigation suggests that the efficiency of noise reduction can be improved by using thick damping layer constructed by materials with high damping factors, large mass density and small speed of sound, which could be the guidelines to design effective damping layers to reduce underwater noise caused by marine structures. The developed theoretical solution provides a useful reference for the related researches. It has been realised that marine structures are quite complex, for which theoretical solutions of FSIA could not be found [4]. The experiments (Fig.1 c) in SJTU and the developed numerical method [4] (Fig.1 b) with computer code [11] have provided available approaches to deal with underwater noise reduction problems, which has been carried on in UoS and SJTU initiated by RAE project.

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