Fluid-structure coupling by the entrained fluid in submerged concentric double-shell vibration

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Vibrational characteristics of a point-driven "double shell" (two concentric submerged cylindrical shells coupled by the entrained fluid) are investigated. Strong- and weak-coupling theories are proposed to model the interaction of how the inner and outer shells are coupled by the entrained fluid. Each theory is based on Flügge's infinite-shell equations, the Helmholtz wave equation, and boundary conditions at the fluid-structure interfaces which depend on the coupling situation. Strong coupling forms a standing wavefield in the entrained fluid; weak coupling, a free wavefield. Experiments are carried out where generalized near-field acoustical holography (GENAH) is employed to provide the experimental vibration characteristics in wavenumber-frequency space of the finite double-shell. It is confirmed theoretically and experimentally that the outer shell of the double shell exhibits two separate dispersion curves: A higher-frequency curve bears in-phase vibrations with respect to the inner shell and suggests a forced response of the inner shell; the other lower-frequency curve, out-of-phase vibrations, and an induced response of the outer shell. Fluid-structure coupling condition definitely affects the lower-frequency curve, but barely the higher-frequency one. Comparison between theoretical and experimental results demonstrates the validity of the strong-coupling model.

Keywords: Underwater structural vibration, Fluid-structure interaction, Double shell, Entrained fluid, K-space analysis, Propagator, Dispersion curve, Generalized near-field acoustical holography

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1. INTRODUCTION

Sound radiation of very low frequencies from ships and submersibles is presumably caused by the overall vibration of their whole or central body. Manifest evidences of such vibration or radiation, however, have not yet obtained experimentally or theoretically. This is because real ships take shapes and structures of large scale and high complexity. It should be thus recognized that present available theories^{1,2)} applicable to the research of ship vibration and radiation are far from reality, particularly in low frequency region. Also, it will be profitless so far to resort to the computer simulation (based on

the finite element and/or boundary element method) without promoting proper theoretical or experimental investigations because there still remains many problems in modeling itself.³⁾ For example, we do not know exactly how to model the ribbed structures for ship-hull reinforcement, the discontinuity on hull plates, and the fluid-structure interaction through the entrained fluid in double-hull configuration. The research of these major contributing factors will thus take priority over that of the whole in order to construct ship structural acoustics.

As an example of such factor research, this paper is then focusing on the vibroacoustical role of the fluid entrained between two concentric submerged

cylindrical shells (the shells having a configuration like this will be referred to as a *double shell* below). Double-hull structure of submersibles is here simplified as a cylindrical double shell without ribs. Such a simplification will make it possible to create fundamental theory on the interaction among the inner shell, the outer shell, the entrained fluid, and the surrounding fluid.

From the viewpoint of vibroacoustical differences between single and double shells immersed in fluid, the coupled vibration of a double shell was already studied theoretically and experimentally.⁴⁻⁷⁾ Then, only brief outline of Flügge's shell equations and their K-space (*i.e.*, spatial wavenumber/frequency-space) representation will be given in Section 3 after the description of our point-driven double-shell model in Section 2. Since acoustic pressure may propagate from the inner shell to the outer shell through the annular fluid space between them, the Helmholtz wave equation is applicable to this field (Section 4). The acoustic pressure in this internal field exerts fluid-loading force to the enclosing shells, whose strength depends on the coupling between the fluid field and shell structure. Theories on strong and weak couplings will be described in Section 5, from which 6×6 matrix equation with fluid-structure coupling elements will be derived in Section 6 and numerically calculated in Section 7. Experimental results obtained from the GENAH (GEneralized Near-field Acoustical Holography) measurement^{8,9)} on a double-shell model immersed in water pool will be explained in Section 8. Theoretical and experimental results, which are expressed as dispersion curve given by K-space analysis, will be compared with each other to examine which coupling is predominant (Section 9). In conclusion, vibrational behaviors due to the fluid-structure coupling will be summarized.

2. MODEL DESCRIPTION

The coupled vibration of a theoretical model consisting of two concentric circular cylindrical shells (a double shell, see Fig. 1) is considered in this paper. The inner shell contains air, the outer shell is surrounded by fluid, and the annular space between them is occupied by fluid. This annular fluid is referred to as the *entrained fluid*. The inner and outer shells are vibroacoustically coupled by this entrained fluid. The objective of this paper is to consider what kind of fluid-structure interaction is

Fig. 1 Theoretical model of the double shell. The shell is infinite and driven by a point force F_1 at $r=R_1$, $\theta=0$, and $z=z_0$. The inner shell contains air, the outer shell is surrounded by fluid, and the annular space between them is occupied by fluid.

occurring among them.

Theoretical double-shell model of Fig. 1 is infinite and driven radially with a point force F_1 located at $r=R_1$, $\theta=0$, and $z=z_0$ in the cylindrical coordinate system (r, θ, z) . The force is harmonic and has the time dependence of $\exp(-i\omega t)$, where ω is the angular frequency, t the time, and i the square root of -1 . Shell thickness and mean radius are denoted by h and R , respectively. Subscript i is added to those symbols to discriminate the inner shell ($i=1$) from the outer one ($i=2$). Each shell is made of the same material and we denote its density, Young's modulus, and Poisson's ratio by ρ , E , and ν , respectively.

Experimental model corresponding to the theoretical one shown in Fig. 1 is also constructed (see Fig. 2). Its dimensions and material constants are shown in Table 1. The inner shell is closed by disk endcaps. These endcaps are supported by three rods which go through the holes arranged on the disk endcap. Three small holes are perforated in each ring endcap (not shown in Fig. 2) to conduct water into the annular space between the inner and outer shells. Rubber packings are used to obtain waterproofness of the inner shell and to give the outer shell the boundary condition similar to that of the inner shell. An O-ring is, moreover, used for a lid that screws on a supporting rod to assure waterproofness. The boundary condition at the shell end may be considered as *simply-supported*. A small and lightweight piezoelectric shaker is attached on the inner shell. The position of this shaker is

Fig. 2 Experimental model of the double shell. The shell is finite and simply-supported by the endcaps with rubber packings. A small shaker is provided as a harmonic point-force driver.

Table 1 Numerical values of the experimental double-shell model.

Inner Shell		stainless steel
Density	ρ (kg/m ³)	7.93×10^3
Young's modulus	E (N/m)	1.95×10^{11}
Poisson's ratio	ν	0.28
Loss factor	η	0.001
Mean radius	R_1 (m)	0.081
Thickness	h_1 (m)	0.003
Length	l (m)	0.800
Outer Shell		stainless steel
Mean radius	R_2 (m)	0.107
Thickness	h_2 (m)	0.003
Length	l (m)	0.800
Fluid		water
Density	ρ_0 (kg/m ³)	1.0×10^3
Sound speed	c_0 (m/s)	1.4883×10^3
Driver		harmonic point-force
Position	z_0 (m)	0.1 (from the center)

0.10 m up from the shell center. The shaker can include a force gauge to measure the force exerted on the shell.

Finite-shell vibration shows the spatial periodicity distributed along the shell axis and has discrete axial wavenumber spectrum; infinite-shell vibration shows the spatial nonperiodicity and has continuous axial wavenumber spectrum. The simply-supported boundary condition for the finite shell is expressed as

$$k_z l = m\pi, \quad (1)$$

where k_z is the axial wavenumber, m the axial mode number (the number of half-wavelength along the shell axis, $m=1, 2, 3, \dots$), and l the shell length. These k_z and m are discrete for finite shells, while they should be treated as continuous for infinite shells. However, if we consider an infinite shell with an axially periodic distribution of vibration restricted to a finite part of shell which satisfies Eq. (1), it may correspond to the finite-shell vibration.¹⁰⁾ Therefore, the comparison between theory on an infinite shell and experiment on a finite shell is safely allowed.

3. INFINITE-SHELL EQUATIONS AND THEIR K-SPACE REPRESENTATION

Brief outline of infinite-shell equations and their K-space representation is described in this section. See Ref. 4) for more details.

The equation of motion of the shell ($i=1, 2$) shown in Fig. 1 can be written as¹¹⁾

$$\begin{pmatrix} l_{i1} & l_{i2} & l_{i3} \\ l_{i2} & l_{i4} & l_{i5} \\ l_{i3} & l_{i5} & l_{i6} \end{pmatrix} \begin{pmatrix} u_i \\ v_i \\ w_i \end{pmatrix} = \frac{R_i^2(1-\nu^2)}{Eh_i} \begin{pmatrix} 0 \\ 0 \\ -q_i + F_i \delta(z-z_0) \delta(\theta)/R_i \end{pmatrix}, \quad (2)$$

where the axial, circumferential, and radial displacements of the shell are given by u_i , v_i , w_i , respectively. The quantity q_i is the acoustic pressure loading on the cylinder. Since the driving force is applied to the inner shell only, F_2 equals zero. According to our convention, F_1 is defined as positive outward, while q_i is defined as positive inward. The δ is the Dirac delta function. Subscript i of the matrix elements l_{ij} ($j=1, 2, 3, 4, 5, 6$) takes 1 for the inner shell and 2 for the outer one. These l_{ij} are differential operators with values specified by the particular shell theory used. For Flügge's theory,¹¹⁾ for example,

$$l_{i6} = 1 + \left(\frac{h_i^2}{12R_i^2} \right) \left[1 + R_i^4 \left(\frac{\partial^4}{\partial z^4} \right) + 2R_i^2 \left(\frac{\partial^4}{\partial z^2 \partial \theta^2} \right) + \left(\frac{\partial^4}{\partial \theta^4} \right) + 2 \left(\frac{\partial^2}{\partial \theta^2} \right) \right] + \left(\frac{R_i^2}{c_p^2} \right) \left(\frac{\partial^2}{\partial t^2} \right),$$

where the quantity $c_p = [\sqrt{E/\rho(1-\nu^2)}]$ for $i=1$ and 2 is the compressional wave speed in both inner and outer shells.

Equation (2) is transformed to algebraic equations by application of the two-dimensional, spatial Fourier transform:

$$\begin{pmatrix} L_{i1} & L_{i2} & L_{i3} \\ L_{i2} & L_{i4} & L_{i5} \\ L_{i3} & L_{i5} & L_{i6} \end{pmatrix} \begin{pmatrix} U_{in} \\ V_{in} \\ W_{in} \end{pmatrix} = \frac{R_i^2(1-\nu^2)}{Eh_i} \begin{pmatrix} 0 \\ 0 \\ -Q_{in} + \frac{F_i \exp(-ik_z z_0)}{2\pi R_i} \end{pmatrix}, \quad (3)$$

where L_{ij} , Q_{in} , U_{in} , V_{in} , and W_{in} represent the transforms of l_{ij} , q_i , u_i , v_i , and w_i , respectively. For example, W_{in} is defined as

$$W_{in}(k_z, \omega) = \frac{1}{2\pi} \int_0^{2\pi} d\theta e^{-in\theta} \int_{-\infty}^{\infty} dz e^{-ik_z z} w_i(\theta, z, \omega), \quad (4)$$

where n represents the circumferential wavenumber. Corresponding to the above l_{i6} , we get the following:

$$L_{i6} = 1 - \Omega_i^2 + \left[\left(\frac{1}{12} \right) \left(\frac{h_i}{R_i} \right)^2 \right] \cdot \{ 1 - 2n^2 + [(k_z R_i)^2 + n^2]^2 \},$$

where Ω_i is the nondimensional frequency defined as

$$\Omega_i = \frac{R_i}{c_p} \omega. \quad (5)$$

The expression of the inverse transform of Eq. (4) gives a decomposition of the radial surface displacement in terms of helical waves, i.e., $W_{in}(k_z, \omega) \exp[i(k_z z + n\theta - \omega t)]$. Since these components make up the helical-wave or K-space spectrum, we will refer to Eq. (3) as the K-space representation of the shell equations. Also, we will refer to the complex amplitude $W_{in}(k_z, \omega)$ as wavenumber-frequency (or k - ω) representation of shell displacement $w_i(z, \theta, \omega)$ when we consider the K-space spectrum as a function of k_z and ω by fixing n . Theory and experiment described below are based on this k - ω representation. Major advantages of this representation are that i) the algebraic form of Eq. (3) is hand-

ed more easily than the operator form of Eq. (2) and ii) advanced signal processings are easily available in k - ω space.

4. ACOUSTIC PROPAGATION IN FLUID FIELD

Since the acoustic wave propagates in fluid fields, we may apply the Helmholtz equation to them:

$$\nabla^2 p_j + k_0^2 p_j = 0, \quad (6)$$

where $p_j(r, \theta, z, \omega)$ expresses the acoustic pressure. The subscript j denotes the annular entrained fluid ($j=1$) and the surrounding free fluid ($j=2$). The wavenumber k_0 is defined as ω/c_0 using the sound speed c_0 in the fluid (we are considering the same fluid in two fields). The Laplacian ∇^2 has the form $(1/r)(\partial/\partial r)(r\partial/\partial r) + (1/r^2)(\partial^2/\partial \theta^2) + (\partial^2/\partial z^2)$ in cylindrical coordinates.

If we apply the spatial Fourier transform of Eq. (4) to Eq. (6), we have the following Bessel's equation:

$$\left(\xi \frac{d}{d\xi} \right)^2 P_{jn} + \left(\xi \frac{d}{d\xi} \right) P_{jn} + (\xi^2 - n^2) P_{jn} = 0, \quad (7)$$

where $P_{jn}(r, k_z, \omega)$ corresponds to k - ω representation of p_j , and $\xi = r\sqrt{k_0^2 - k_z^2}$. This Eq. (7) gives the general solution

$$P_{jn}(\xi) = a_j A_{jn}(\xi) + b_j B_{jn}(\xi), \quad (8)$$

where

$$A_{jn}(\xi) = \begin{cases} J_n(k_r r) & c > c_0 \\ I_n(k_r' r) & c \leq c_0, \end{cases} \quad (9)$$

$$B_{jn}(\xi) = \begin{cases} Y_n(k_r r) & c > c_0 \\ K_n(k_r' r) & c \leq c_0, \end{cases} \quad (10)$$

$$k_r = \sqrt{k_0^2 - k_z^2} = k_z \sqrt{(c/c_0)^2 - 1}, \quad (11)$$

$$k_r' = \sqrt{k_z^2 - k_0^2} = k_z \sqrt{1 - (c/c_0)^2}, \quad (12)$$

$$k_z = \frac{\omega}{c}, \quad (13)$$

$$k_0 = \frac{\omega}{c_0}. \quad (14)$$

The quantity c in Eqs. (9) to (13) means the phase velocity of the helical wave propagating in the axial direction. Functions J_n and Y_n , which are the Bessel's functions of the first and second kind respectively, are used in the supersonic region $c > c_0$, while I_n and K_n , which are the modified Bessel's functions, are used in the subsonic region $c \leq c_0$. The quantities k_r and k_r' are the radial wavenumber in

these regions, respectively. The quantities k_0 and k_z are the acoustic wavenumber of fluid and the axial wavenumber of shell, respectively.

Coefficients a_j and b_j of Eq. (8) are determined by the boundary conditions at the fluid-structure interfaces and the radiation condition for the free field. Since there are four unknown coefficients a_1 , a_2 , b_1 , and b_2 , we have to prepare four equations describing boundary and radiation conditions to determine them. It will be carried out in the next section.

5. FLUID-STRUCTURE COUPLING

There seems to be two kinds of possibility on the vibroacoustical function of the entrained fluid.

Strong Coupling: The forced vibration of the inner shell propagates to the outer shell through the entrained fluid. A part of the incidental energy of this vibratory wave drives the outer shell, and the rest of it is reflected back to the inner shell. The induced vibration of the outer shell radiates sound into the free field; the reflected wave affects the original forced vibration of the inner shell in a degree. In other words, we may say that a standing wavefield is formed in the entrained fluid. Strongly coupled vibration of the double shell will thus take place.

Weak Coupling: However, if the reflection at the outer shell is weak because of small difference in mechanical impedance between the entrained fluid and outer shell, or if the driving force applied to the inner shell is so sturdy that the reflected wave gives only negligible contribution to the inner shell, the coupling will become weak. The outer shell is still driven by the acoustic wave from the inner shell and radiates sound toward the free field, while the inner shell may behave like an independent vibrator because it will be unaffected by the presence of the entrained fluid and outer shell. In other words, we may say that a free wavefield is formed in the entrained fluid. Weakly coupled vibration of the double shell will thus take place.

Much simpler examples illustrating the above-mentioned concepts of strong and weak couplings are found in Ref. 12), where idealized models of i) sound transmission through a double-leaf partition and ii) cavity wavefield between a vibrating surface and its cladding enclosure to reduce sound radiation are considered. For more complete treatment, the theories described below should be applied to them.

5.1 K-space Fluid Loadings due to Strong Coupling

The boundary conditions are given by the continuity of radial velocity at the fluid-shell interfaces:

$$\begin{aligned}\dot{w}_1 &= \left(\frac{1}{i\omega\rho_0} \right) \left(\frac{\partial p_1}{\partial r} \right)_{r=R_1}, \\ \dot{w}_2 &= \left(\frac{1}{i\omega\rho_0} \right) \left(\frac{\partial p_1}{\partial r} \right)_{r=R_2} = \left(\frac{1}{i\omega\rho_0} \right) \left(\frac{\partial p_2}{\partial r} \right)_{r=R_2}\end{aligned}\quad (15)$$

where the dot denotes the time derivative. In K-space, by applying Eqs. (4) and (8) to Eq. (15), we obtain respectively

$$\begin{aligned}a_1 A_{1n}'(R_1) + b_1 B_{1n}'(R_1) &= \omega^2 \rho_0 W_{1n}, \\ a_1 A_{1n}'(R_2) + b_1 B_{1n}'(R_2) &= \omega^2 \rho_0 W_{2n}, \\ a_2 A_{2n}'(R_2) + b_2 B_{2n}'(R_2) &= \omega^2 \rho_0 W_{2n},\end{aligned}\quad (16)$$

where the prime denotes differentiation with respect to r . The radiation condition requires that the transformed acoustic pressure $P_{2n}(r)$ in the free field ($j=2$) must have the form of outgoing wave, i.e., the Hankel function $H_n^{(1)}(r)$ ($c > c_0$) or the modified Bessel function $K_n(r)$ ($c \leq c_0$). This means that

$$b_2 = ia_2 \quad (c > c_0), \quad a_2 = 0 \quad (c \leq c_0). \quad (17)$$

The above four equations of Eqs. (16) and (17) determine a_1 , a_2 , b_1 , and b_2 as functions of W_{1n} and W_{2n} . Finally, we can obtain the following solution of the transformed acoustic pressure:

$$\begin{aligned}P_{1n}(r) &= \begin{cases} \frac{(\omega^2 \rho_0 / k_r)}{J_n'(k_r R_1) Y_n'(k_r R_2) - J_n'(k_r R_2) Y_n'(k_r R_1)} \\ \cdot \{ [Y_n'(k_r R_2) J_n(k_r r) - J_n'(k_r R_2) Y_n(k_r r)] W_{1n} \\ + [-Y_n'(k_r R_1) J_n(k_r r) + J_n'(k_r R_1) Y_n(k_r r)] W_{2n} \}, & (c > c_0) \\ \frac{(\omega^2 \rho_0 / k_r')}{I'(k_r' R_1) K'(k_r' R_2) - I'(k_r' R_2) K'(k_r' R_1)} \\ \cdot \{ [K_n'(k_r' R_2) I_n(k_r' r) - I_n'(k_r' R_2) K_n(k_r' r)] W_{1n} \\ + [-K_n'(k_r' R_1) I_n(k_r' r) + I_n'(k_r' R_1) K_n(k_r' r)] W_{2n} \}, & (c \leq c_0) \end{cases} \quad (18) \\ P_{2n}(r) &= \begin{cases} \frac{\omega^2 \rho_0}{k_r} \frac{H_n^{(1)}(k_r r)}{H_n^{(1)}(k_r R_2)} W_{2n}, & (c > c_0) \\ \frac{\omega^2 \rho_0}{k_r'} \frac{K_n(k_r' r)}{K_n(k_r' R_2)} W_{2n}, & (c \leq c_0) \end{cases} \quad (19)\end{aligned}$$

where the prime defines differentiation with respect to the argument ($k_r r$ or $k_r' r$) of the Bessel functions.

Fluid (or acoustic) loadings involved in Eq. (2) are given by

$$q_1 = (p_1)_{r=R_1}, \quad q_2 = (p_2 - p_1)_{r=R_2}. \quad (20)$$

By inserting the solution given as Eqs. (18) and (19) into the K-space representation of Eq. (20), we can obtain the corresponding K-space fluid loadings as follows:

$$Q_{1n}(k_z, \omega) = [G_{12}(R_1)W_{1n}(k_z, \omega) + G_{21}(R_1)W_{2n}(k_z, \omega)], \quad (21)$$

$$\begin{aligned} Q_{2n}(k_z, \omega) = & -[G_{12}(R_2)W_{1n}(k_z, \omega) \\ & + G_{21}(R_2)W_{2n}(k_z, \omega) \\ & + G_{22}(R_2)W_{2n}(k_z, \omega)], \end{aligned} \quad (22)$$

where

$$G_{12}(r) = \begin{cases} \left(\frac{\omega^2 \rho_0}{k_r} \right) \cdot \frac{J_n(k_r r) Y_n'(k_r R_2) - Y_n(k_r r) J_n'(k_r R_2)}{J_n'(k_r R_1) Y_n'(k_r R_2) - Y_n'(k_r R_1) J_n'(k_r R_2)}, & (c > c_0) \\ \left(\frac{\omega^2 \rho_0}{k_r'} \right) \cdot \frac{I_n(k_r' r) K_n'(k_r' R_2) - K_n(k_r' r) I_n'(k_r' R_2)}{I_n'(k_r' R_1) K_n'(k_r' R_2) - K_n'(k_r' R_1) I_n'(k_r' R_2)}, & (c \leq c_0) \end{cases} \quad (23)$$

$$G_{21}(r) = \begin{cases} \left(\frac{\omega^2 \rho_0}{k_r} \right) \cdot \frac{J_n'(k_r R_1) Y_n(k_r r) - Y_n'(k_r R_1) J_n(k_r r)}{J_n'(k_r R_1) Y_n'(k_r R_2) - Y_n'(k_r R_1) J_n'(k_r R_2)}, & (c > c_0) \\ \left(\frac{\omega^2 \rho_0}{k_r'} \right) \cdot \frac{I_n'(k_r' R_1) K_n(k_r' r) - K_n'(k_r' R_1) I_n(k_r' r)}{I_n'(k_r' R_1) K_n'(k_r' R_2) - K_n'(k_r' R_1) I_n'(k_r' R_2)}, & (c \leq c_0) \end{cases} \quad (24)$$

$$G_{22}(r) = \begin{cases} \left(\frac{\omega^2 \rho_0}{k_r} \right) \frac{H_n^{(1)}(k_r r)}{H_n^{(1)}(k_r R_2)}, & (c > c_0) \\ \left(\frac{\omega^2 \rho_0}{k_r'} \right) \frac{K_n(k_r' r)}{K_n(k_r' R_2)}, & (c \leq c_0) \end{cases} \quad (25)$$

The quantities $G_{12}(r)$, $G_{21}(r)$, and $G_{22}(r)$ defined by

Fig. 3 Fluid-structure coupling in point-driven, double-shell vibration on the basis of K-space representation. (a) strong coupling, (b) weak coupling.

Eqs. (23), (24), and (25) respectively, may be called *propagators*: $G_{12}(r)$ describes the propagation from the inner shell to the outer shell; $G_{21}(r)$ the propagation from the outer shell to the inner shell; and $G_{22}(r)$ the radiation from the outer shell to the surrounding free field. Figure 3(a) illustrates the strong fluid-structure coupling defined by these propagators.

5.2 K-space Fluid Loadings due to Weak Coupling

Weakly coupled vibration will happen if a free wavefield instead of a standing wavefield may be formed in the entrained fluid. Schematically, this is done by ignoring the propagator $G_{21}(r)$ in Fig. 3(a). Mathematically, this means that the radiation condition is satisfied for the entrained fluid as well as for the surrounding fluid:

$$b_1 = ia_1 \quad (c > c_0), \quad a_1 = 0 \quad (c \leq c_0). \quad (26)$$

Also, the second equation of Eq. (16) should be eliminated because the motion of outer shell w_2 does not contribute to the acoustic pressure p_1 in the entrained fluid if the free wavefield is assumed there. Hence, we get two boundary conditions [the first and third equations of Eq. (16)] and two radiation conditions [Eqs. (17) and (26)] to determine unknown coefficients a_1 , a_2 , b_1 , and b_2 . From these conditions we obtain the following expression of the transformed pressure in K space instead of Eq. (18):

$$P_{1n}(r) = \begin{cases} \frac{\omega^2 \rho_0}{k_r} \frac{H_n^{(1)}(k_r r)}{H_n^{(1)'}(k_r R_1)} W_{1n}, & (c > c_0) \\ \frac{\omega^2 \rho_0}{k_r} \frac{K_n(k_r' r)}{K_n'(k_r' R_1)} W_{1n}, & (c \leq c_0) \end{cases} \quad (27)$$

The transformed pressure $P_{2n}(r)$ in the surrounding free wavefield is given by the previous Eq. (19).

The corresponding K-space fluid loadings are obtained by inserting Eqs. (19) and (27) into Eq. (20):

$$Q_{1n}(k_z, \omega) = G_{11}(R_1) W_{1n}(k_z, \omega) \quad (28)$$

$$Q_{2n}(k_z, \omega) = -G_{11}(R_2) W_{1n}(k_z, \omega) + G_{22}(R_2) W_{2n}(k_z, \omega) \quad (29)$$

where the propagator

$$G_{11}(r) = \begin{cases} \left(\frac{\omega^2 \rho_0}{k_r} \right) \frac{H_n^{(1)}(k_r r)}{H_n^{(1)'}(k_r R_1)}, & (c > c_0) \\ \left(\frac{\omega^2 \rho_0}{k_r'} \right) \frac{K_n(k_r' r)}{K_n'(k_r' R_1)}, & (c \leq c_0) \end{cases} \quad (30)$$

The propagator $G_{22}(r)$ was already given by Eq. (25). Figure 3(b) illustrates the weak coupling defined by these two propagators.

6. 6×6 MATRIX EQUATION WITH FLUID-STRUCTURE COUPLING ELEMENTS

6.1 Strongly-Coupled Vibration of the Double Shell

Substituting Q_{in} ($i=1, 2$) given by Eqs. (21) and (22) for those in Eq. (3), we have the following 6×6 matrix equation of double-shell vibration:

$$\begin{pmatrix} L_{11} & L_{12} & L_{13} & 0 & 0 & 0 \\ L_{12} & L_{14} & L_{15} & 0 & 0 & 0 \\ L_{13} & L_{15} & L_{16} + L_{16F} & 0 & 0 & L_{17F} \\ 0 & 0 & 0 & L_{21} & L_{22} & L_{23} \\ 0 & 0 & 0 & L_{22} & L_{24} & L_{25} \\ 0 & 0 & L_{27F} & L_{23} & L_{25} & L_{26} + L_{26F} \end{pmatrix} \begin{pmatrix} U_{1n} \\ V_{1n} \\ W_{1n} \\ U_{2n} \\ V_{2n} \\ W_{2n} \end{pmatrix} = \begin{pmatrix} 0 \\ 0 \\ F_1' \\ 0 \\ 0 \\ 0 \end{pmatrix}, \quad (31)$$

where

$$L_{16F} = D_1 G_{12}(R_1), \quad (32)$$

$$L_{17F} = D_1 G_{21}(R_1), \quad (33)$$

$$L_{26F} = D_2 [G_{22}(R_2) - G_{21}(R_2)], \quad (34)$$

$$L_{27F} = -D_2 G_{12}(R_2), \quad (35)$$

$$F_1' = D_1 F_1 \exp(-ik_z z_0)/(2\pi R_1), \quad (36)$$

$$D_i = (1 - \nu^2) R_i^2 / (E h_i) \quad (i=1, 2), \quad (37)$$

$$\alpha_i = \rho_0 R_i / \rho h_i \quad (i=1, 2). \quad (38)$$

Equations (5), (23), (24), and (25) are employed to get the explicit, nondimensionalized forms of Eqs. (32) to (35). The matrix elements L_{16F} , L_{17F} , L_{26F} , and L_{27F} indicate strong fluid-structure coupling in our immersed double shell. Particularly, L_{17F} and L_{27F} couple the inner and outer shells to each other via the entrained fluid.

The radial displacements W_{1n} and W_{2n} in K space are expressed as follows by solving Eq. (31):

$$W_{1n}(k_z, \omega) = |L'|/|L|, \quad (39)$$

$$W_{2n}(k_z, \omega) = |L''|/|L|, \quad (40)$$

where the matrix L denotes 6×6 matrix in the left side of Eq. (31) and its determinant is given by

$$|L| = |L_1| |L_2| - L_{17F} L_{27F} \begin{vmatrix} L_{11} & L_{12} \\ L_{12} & L_{14} \end{vmatrix} \cdot \begin{vmatrix} L_{21} & L_{22} \\ L_{22} & L_{24} \end{vmatrix}. \quad (41)$$

In Eq. (41) L_1 is 3×3 matrix of the immersed inner shell alone and defined by nine elements of row 1 to 3 and line 1 to 3 in L . Similarly, L_2 is that of the immersed outer shell alone and defined by nine elements of row 4 to 6 and line 4 to 6 in L . Determinants $|L'|$ and $|L''|$ in Eqs. (39) and (40) are given by

$$|L'| = F_1' |L_2| \cdot \begin{vmatrix} L_{11} & L_{12} \\ L_{12} & L_{14} \end{vmatrix}, \quad (42)$$

$$|L''| = -F_1' L_{27F} \begin{vmatrix} L_{11} & L_{12} \\ L_{12} & L_{14} \end{vmatrix} \cdot \begin{vmatrix} L_{21} & L_{22} \\ L_{22} & L_{24} \end{vmatrix}, \quad (43)$$

respectively.

6.2 Weakly-Coupled Vibration of the Double Shell

We can obtain similar expressions for the weak coupling by using Eqs. (28) and (29) instead of Eqs. (21) and (22). We then have to employ the following forms of coupling elements in Eq. (31):

$$L_{16F} = D_1 G_{11}(R_1), \quad (44)$$

$$L_{17F} = 0, \quad (45)$$

$$L_{26F} = D_2 G_{22}(R_2), \quad (46)$$

$$L_{27F} = -D_2 G_{11}(R_2). \quad (47)$$

The radial displacements W_{1n} and W_{2n} are respectively given by Eqs. (39) and (40) in which $|L| = |L_1| \cdot |L_2|$ is used instead of Eq. (41):

$$W_{1n}(k_z, \omega) = F_1' \begin{vmatrix} L_{11} & L_{12} \\ L_{12} & L_{14} \end{vmatrix} / |L_1|, \quad (48)$$

$$W_{2n}(k_z, \omega) = -F_1' L_{27F} \begin{vmatrix} L_{11} & L_{12} \\ L_{12} & L_{14} \end{vmatrix} \cdot \begin{vmatrix} L_{21} & L_{22} \\ L_{22} & L_{24} \end{vmatrix} / |L_1| \cdot |L_2|. \quad (49)$$

The form of Eq. (48) indicates that the inner shell shows the forced response independent of the existence of the entrained fluid and outer shell. In other word, Eq. (48) corresponds to the forced response of an immersed single shell (*i.e.*, the inner shell alone of a double shell) driven by a point force.

6.3 General Characteristics of the Solution

From Eqs. (39) and (40) [also from Eqs. (48) and (49)] we get the following relation common to both strong and weak couplings:

$$W_{2n}(k_z, \omega) = -[L_{27F} W_{1n}(k_z, \omega)] \cdot \frac{\begin{vmatrix} L_{21} & L_{22} \\ L_{22} & L_{24} \end{vmatrix}}{|L_2|}. \quad (50)$$

Comparing Eq. (50) with Eq. (48), we know that the outer shell is driven by the apparent force

$$F_2' = -L_{27F} W_{1n}(k_z, \omega).$$

Therefore, as general characteristics of the double-shell vibration, we may state the followings:

- 1) The vibrational characteristics of the inner shell $W_{1n}(k_z, \omega)$ should appear on the outer shell of the double shell through the fluid-structure coupling element L_{27F} .
- 2) The vibrational characteristics of the outer shell itself should appear through the matrix L_2 .

Next let us consider the phase of vibration. The determinant $(L_{21}L_{24} - L_{22}^2)$ in Eq. (50) takes a real value. Another determinant $|L_2|$ becomes real if $c \leq c_0$ but complex if $c > c_0$ because of the propagator $G_{22}(R_2)$ of Eq. (25). The coupling element L_{27F} of Eq. (35) for strong coupling and of Eq. (47) for weak coupling takes a positive real value if $c \leq c_0$ is satisfied.¹³⁾ Hence, we may add the following

statements on the phase of vibration:

- 3) In-phase coupled vibration occurs if the subsonic condition ($c \leq c_0$) is satisfied and $(L_{21}L_{24} - L_{22}^2)/|L_2|$ takes a negative value.
- 4) Out-of-phase coupled vibration occurs if the subsonic condition is satisfied and $(L_{21}L_{24} - L_{22}^2)/|L_2|$ takes a positive value.
- 5) Neither in-phase nor out-of-phase vibration occurs if the supersonic condition ($c > c_0$) is satisfied.

7. NUMERICAL CALCULATION OF DISPERSION CURVE

Before calculating the K-space radial displacement $W_{1n}(k_z, \omega)$ [defined as Eq. (39) or (48)] or $W_{2n}(k_z, \omega)$ [defined as Eq. (40) or (49)] numerically, structure-borne energy dissipation should be introduced into the above theory by treating Young's modulus as a complex quantity, namely

$$E \rightarrow E(1 - i\eta),$$

where η denotes the hysteresis loss factor. When using this complex Young's modulus, we make the following replacements in the equations described in the previous sections:

$$c_p^2 \rightarrow c_p^2(1 - i\eta) \quad \text{and} \quad \Omega_i^2 \rightarrow \frac{\Omega_i^2}{1 - i\eta}.$$

Introducing the structural loss, we can certainly avoid the infinity because the denominators of Eqs. (39), (40), (48), and (49) will not become null.

Now let us compare the vibration due to the strong coupling as shown in Fig. 3(a) with that due to the weak coupling as shown in Fig. 3(b). We will concentrate on the outer shell because the vibrational characteristics of the outer shell are much more important than those of the inner shell from the viewpoint of underwater radiation, and because the vibrational characteristics of the inner shell should appear on the outer shell as shown by Eq. (50). The logarithmic level $20 \log |W_{2n}(k_z, \omega)|$ is then drawn in Fig. 4(a) for the strong coupling of Eq. (40) and in Fig. 4(b) for the weak coupling of Eq. (49), respectively. The circumferential wavenumber n is fixed to 3, 4, and 5 for the left, middle, and right frames, respectively. The abscissa is the axial wavenumber m which is treated as a continuous quantity according to Eq. (1). The ordinate is the frequency f of vibration in Hz. The curve shown in Fig. 4 is referred to as the *dispersion curve* since the

Fig. 4 Theoretical dispersion curve of the absolute radial displacement of the outer shell as a function of the axial mode number m and frequency f . (a) strong coupling, (b) weak coupling.

value of ω/k_z , which is given by a point (m, f) on a curve, defines the local phase velocity c of the helical wave in Eq. (13). For numerical calculation, parameter values listed in Table 1 were employed, and an appropriate value was input into the driving force F_1 in Eq. (36). The range of calculation was $20 \leq f \leq 4,000$ Hz (in 20-Hz steps) and $0.00025 \leq m \leq 12.9$ (in 0.10 steps). The digital data obtained by this calculation is converted to the logarithmic one in which the largest value corresponds to 100 dB. Such resulting data is displayed using color intensity to represent the logarithmic level: 32 shades of green for 0 to 50 dB and another 32 shades of red for 50 to 100 dB. As the displacement amplitude approaches maximum, the color approaches white from red.

From Fig. 4 we may state the followings:

- 1) Both strong and weak couplings give two dispersion curves.
- 2) These two curves are well separated from each other in Fig. 4(a), while the lower curve ap-

proaches to the upper one and intersects with it for $n=3$ and 4 in Fig. 4(b).

- 3) The upper curve which gives higher frequencies for the same m [or gives higher cutoff frequency at $m=0$ for $n=3$ and 4 in Fig. 4(b)] looks almost identical with each other in Figs. 4(a) and 4(b).
- 4) The above 3) means that the upper dispersion curve suggests the forced response of the inner shell driven by a point force.
- 5) The lower dispersion curve then indicates the induced response of the outer shell and reflects the difference in the coupling by the entrained fluid.
- 6) Strong coupling moves the lower dispersion curve downwards more than weak coupling does.

Numerical calculation on the phase of vibration adds the following statements [see Ref. 4] for detailed numerical results]:

- 7) In-phase vibration happens on the upper dis-

Fig. 5 Schematic of GENAH signal processing. An ensemble of acoustic pressure $p(t)$ at the grid points (z, θ) on the measurement surface (cf. frame I) reveals characteristic mode patterns $w(z, \theta)$ of the vibrating surface for a given frequency range (cf. frame VIII). See text items 1) to 8) which correspond to frames I to VIII, respectively.

Fig. 6 Comparison of double-shell's dispersion curves. (a) strong-coupling theory, (b) GENAH experiment.

persion curve.

- 8) Out-of-phase vibration happens on the lower dispersion curve.

These results indicate that items 3) and 4) stated in Section 6.3 are realized in the numerical calculation on our experimental model. Exactly speaking, if the structural loss is introduced, these items are not correct because two determinants and L_{27F} in Eq. (50) take complex values. However, as long as the loss factor is as small as 0.001, it brings only negligible effect to the phase relation.

8. BROADBAND GENAH EXPERIMENT AND DISCUSSION

8.1 Outline of Generalized Near-field Acoustical Holography

The experiment was carried out in the U.S. Naval Research Laboratory's AARF (advanced acoustic research facility) pool. An automated hydrophone positioner has been developed for the experimental implementation of GENAH at this facility. The procedure of the broadband GENAH experiment^{8,9)} is outlined below and shown in each frame (from I to VIII) of Fig. 5, where the case of single shell is illustrated for simplicity:

- 1) One hydrophone receives the radiated acoustic pressure due to the pulsed excitation with a chirp signal (a 10-ms pulse covering the frequency range of 100 Hz to 6 kHz drove the shaker at a repetition rate of 1.0 Hz in our experiment). Analog pressure signal is sampled in very short (a few 10 μ s) intervals. Sampling inception time is precisely controlled by a computer, which works as the reference source for holographic interference. 32 responses of the excitation are added to enhance the SN ratio. The same things are repeated on the preset measurement surface concentric to the vibrating cylindrical shell. This measurement surface was at a distance of $\Delta = 20$ mm from the shell surface ($r = R_2$), twice as long as the shell, and divided into 64×64 grid points in our experiment.
- 2) Sequential data ensemble of acoustic pressure $p(t)$ at the grid points (z, θ) is converted to an ensemble of acoustic pressure distribution $p(z, \theta)$ over the measurement surface at each sampling time t . Such resulting data set is called *acoustic pressure hologram*.
- 3) Temporal Fourier transform is applied to

this hologram.

- 4) Moreover, the two-dimensional, spatial Fourier transform is applied and we obtain another hologram with respect to wavenumber-frequency space. It is called *K-space representation of the hologram*.
- 5) This pressure hologram is back propagated from the measurement surface onto the vibrating surface, and transformed to K-space vibrational velocity hologram by multiplying velocity propagator.
- 6) Appropriate window in wavenumber space is applied to this velocity hologram.
- 7) K-space dispersion curve is given by plotting $\dot{W}_n(k_z, \omega)$ as a function of k_z and ω instead of k_z and n .
- 8) Characteristic mode patterns $\dot{w}(z, \theta)$ of the vibrating surface for a given frequency range are reconstructed by applying the two-dimensional, spatial inverse Fourier transform. The surface velocity exhibits (4, 4) mode of 1,411 Hz in Fig. 5, where red and green colors correspond to 0 to 10 and 0 to -10 respectively, and white color to 10 or -10 in arbitrary unit.

8.2 Comparison between Theory and Experiment

Vibrational modes of the radial surface velocity obtained by the above GENAH experiment are shown in Fig. 5 of Ref. 4), Fig. 1 of Ref. 7), and Fig. 3 of Ref. 14). These figures indicate that the simply-supported boundary condition of Eq. (1) may be realized because the surface velocity approaches null near the shell ends. For clear comparison of theory with experiment, it is, however, much more persuasive to discuss on the basis of the dispersion curve rather than the vibrational mode. See Figs. 6 and 7 of Ref. 4) or Fig. 2 of Ref. 7) for detailed information of the experimental dispersion curves corresponding to the theoretical ones shown in Fig. 4(a) of this paper.

Figure 6 is prepared to examine which model of strong and weak couplings shows better agreement with the experimental results. Figure 6(a) shows $20 \log |\dot{W}_{2n}(k_z, \omega)|$ in which Eq. (40) of strong coupling was recalculated in the range of $200 \leq f \leq 5,200$ Hz (in 20-Hz steps) and $0.00025 \leq m \leq 19.9$ (in 0.157 steps). Figure 6(b) shows $20 \log |\dot{W}_{2n}(k_z, \omega)|$ of the experimentally obtained radial velocity of the outer shell of the double shell. Time-series data of

the measured acoustic pressure consists of 1,024 points resulting in 512 frequency bins which start from 198 Hz in 11.4-Hz steps. On the other hand, 64 points were measured along the axial length which is twice as long as the shell, and then the scanning step of 25 mm determines the maximum value of m as 32. The resolution of Fig. 6(b) is, however, relatively low because the plotting was done in one steps with respect to m . Also, the maximum value of color intensity corresponds to 60 dB because dynamic levels of experimental data were not so high. Moreover, in Fig. 6(b) you may notice many streaks which extend from the high-level, red points to either side of them laterally. This effect is recognized as an oscillation of a sinc function $\sin(x)/x$ centered at a point on a dispersion curve. This sinc function is due to the finite length of the shell, and derived by changing the infinite integral on z in Eq. (4) into the finite integral bounded from $-l/2$ to $l/2$. See Ref. 15) for more detailed discussion on the effects of shell length and signal processing.

Comparing Fig. 6(a) with Fig. 6(b), we notice that the shape of theoretical dispersion curves based on the strong-coupling model is very similar to that of experimental ones. This general agreement between the theory and experiment proves that the strong coupling controls the coupled vibration of our experimental double shell.

9. CONCLUSIONS

Two kinds of theory on the vibration of a point-driven "double shell" were proposed from the viewpoint of fluid-structure coupling. One of them is based on the strong coupling which forms a standing wavefield in the entrained fluid; The other is based on the weak coupling which forms a free wavefield in it. In each theory, 6×6 matrix equation with fluid-structure coupling elements was finally derived from Flügge's infinite-shell theory, Helmholtz' fluid-field equation, and boundary conditions given at the fluid-shell interfaces. The solution of this matrix equation for the coupled vibration, that is, the radial surface displacements of the inner and outer shells of a double shell were represented in wavenumber-frequency ($k-\omega$) space.

Numerical calculation on the experimental model of a double shell revealed the following vibrational behaviors of the double shell:

- 1) Both strong and weak couplings in the double

shell bring two separate dispersion curves.

- 2) One dispersion curve defines higher-frequency, in-phase vibration.
- 3) The other curve defines lower-frequency, out-of-phase vibration.
- 4) The higher-frequency dispersion curve, which indicates the forced response of the inner shell, is almost the same whether the fluid-structure coupling is strong or weak.
- 5) The lower-frequency dispersion curve, which indicates the induced response of the outer shell, depends on the coupling strength: Strong coupling moves this curve downwards (to lower-frequency direction) more than weak coupling does.

The existence of two separate dispersion curves was confirmed by the GENAH experiment on the outer shell of our double-shell model. Experimental dispersion curves were very similar to theoretical ones based on the strong coupling. This agreement demonstrates the validity of strong-coupling theory on the double-shell vibration. This means that the reflection of acoustic wave takes place at the interface between the shell of stainless steel and the entrained water in our experimental model. We may hence comment that the outer shell of double-shell structure is not acoustically transparent but can *shield* sound radiation from the inner shell to some degree.

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REFERENCES

- 1) M. C. Junger and D. Feit, *Sound, Structures, and Their Interaction*, 2nd Ed. (MIT Press, Cambridge, Mass., 1986).
- 2) F. Fahy, *Sound and Structural Vibration* (Academic Press, London, 1985).
- 3) S. Yoshikawa, T. Abe, H. Fujita, and K. Koyano, "Study on vibration and sound radiation of concentric double cylinder," Proc. Spring Meet. Acoust. Soc. Jpn. 3-8-7, 775-776 (1987) (in Japanese).
- 4) S. Yoshikawa, E. G. Williams, and K. B. Washburn, "Vibration of two concentric submerged cylindrical shells coupled by the entrained fluid," NRL Rep. (1991) (unpublished).
- 5) S. Yoshikawa, E. G. Williams, and K. B. Washburn, "Theory and experiment of two concentric submerged cylindrical shells coupled by the entrained fluid," J. Acoust. Soc. Am. **89**, 6SA8, 1955 (1991).
- 6) S. Yoshikawa, "Coupled vibration of two concentric submerged cylindrical shells as a model of submersible's hull structure: Theory," Proc. Spring Meet. Acoust. Soc. Jpn. 2-8-9, 1035-1036 (1992) (in Japanese).
- 7) S. Yoshikawa, E. G. Williams, and K. B. Washburn, "Coupled vibration of two concentric submerged cylindrical shells as a model of submersible's hull structure: Experiment," Proc. Spring Meet. Acoust. Soc. Jpn. 2-8-10, 1037-1038 (1992) (in Japanese).
- 8) E. G. Williams, H. D. Dardy, and K. B. Washburn, "Generalized nearfield acoustical holography for cylindrical geometry: Theory and experiment," J. Acoust. Soc. Am. **81**, 389-407 (1987).
- 9) E. G. Williams, B. H. Houston, and J. A. Bucaro, "Broadband nearfield acoustical holography for vibrating cylinders," J. Acoust. Soc. Am. **86**, 674-679 (1989).
- 10) Ref. 1) pp. 151-152.
- 11) A. W. Leissa, "Vibration of shells," NASA Rep. SP-288, Chap. 2 (1973).
- 12) Ref. 2) pp. 166-172, 186-191, 259-268.
- 13) According to the *Handbook of Mathematical Functions with Formulas, Graphs, and Mathematical Tables*, M. Abramowitz and I. A. Stegun, Eds. (National Bureau of Standards, Washington, D.C., 1970), the numerator $I_n(b)K_n'(b) - K_n(b)I_n'(b)$ of Eq. (35), where $b = k_r R_2$, equals $-1/b$ (see formulas 9.6.15 and 9.6.26). On the other hand, the denominator $I_n'(a)K_n'(b) - K_n'(a)I_n'(b) < 0$, where $a = k_r R_1$, because $I_n'(b) > I_n'(a) > 0$ and $K_n'(a) < K_n'(b) < 0$ (see Fig. 9.7, formulas 9.6.7, 9.6.9, 9.7.3 and 9.7.4.)
- 14) S. Yoshikawa, "Joint research on the "coupled vibration of two concentric submerged cylindrical shells" conducted at the U. S. Naval Research Laboratory and brief introduction of the laboratory," J. Mar. Acoust. Soc. Jpn. **19**, 123-131 (1992) (in Japanese).
- 15) E. G. Williams, B. H. Houston, and J. A. Bucaro, "Experimental investigation of the wave propagation on a point-driven, submerged capped cylinder using K-space analysis," J. Acoust. Soc. Am. **87**, 513-522 (1990).

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