Hydroacoustic field in sonar cavity under mechanical excitations

Weike Wang[1], Rui HUO [2,3], Zhidong Wang [2,3]

[1]Key Laboratory of High Efficiency and Clean Mechanical Manufacture (Shandong University) Ministry of Education, Ji'nan 250061, China

[2]School of Mechanical Engineering, Shandong University, Ji'nan250061, China

[3]Key Laboratory of High Efficiency and Clean Mechanical Manufacture (Shandong University) Ministry of Education, Ji'nan 250061, China

[1]449200779@qq.com, [2]huorui@sdu.edu.cn, [3]670245537@qq.com

Keywords: Sonar cavity, mechanical self-noise, fluid-structure interaction, hydroacoustic field.

Abstract. In view of the destructive influence of mechanical self-noise on sonar's detection capability at lower frequencies, a parametric model composed of panels and a water filled rectangular cavity is established for investigation of characteristics of hydroacoustic field caused by mechanical excitation loaded on sonar platform area. The analytical derivation and numerical simulation analysis of differential vibration equations provide theoretical reference for prediction and control of mechanical self-noise.

Introduction

The self-noise in sonar refers to the sonar sound interference at work,according to its source is divided into hydrodynamic noise , machinery noise, propeller noise and the ocean ambient noise. At present, many the self-noise in sonar problem discussions are in the majority with hydrodynamic noise, however, the mechanical noise problem also can't be ignored and particularly important for the low-speed submarine underwater navigation.Musha et al. Literature[1] proposed a method to determine the self-noise of sonar caused by turbulence by using a simplified acoustic vibration transfer function. Literature[2,3,4]based on the rectangular cavity model used the statistical energy method and the modal method to calculate and analyze the hydrodynamic component of sonar self-noise; In the literature[5], the low frequency (10Hz~500Hz) self-noise of the platform area is predicted by using the finite element modeling of sonar platform area structure and empirical radiation coefficient.

In this paper, we first establish a parametric model of the noise analysis of rectangular cavity for sonar array mechanical cavity. Qualitative theoretical analysis is made to the modal frequency and influencing factors of fluid-structure interaction vibration and the influencing effect of acoustic window, providing a theory basis to the denoising of sonar array.

Rectangular sonar array cavity model

Model parameter definition
Fig. 1 rectangle filling water chamber on the panel is subject to external excitation pe elastic plate used to simulate sonar platform area and the rest of the panels are considered as rigid, the shadow part of the graph to simulates the acoustic window. Establishing in the Cartesian coordinate system OXYZ.

Fig. 1: An rectangular enclosure model for Sonar cavity

The establishment of fluid solid coupling equation
Underwater acoustic field of water filling airtight chamber interior ministry satisfies wave equation
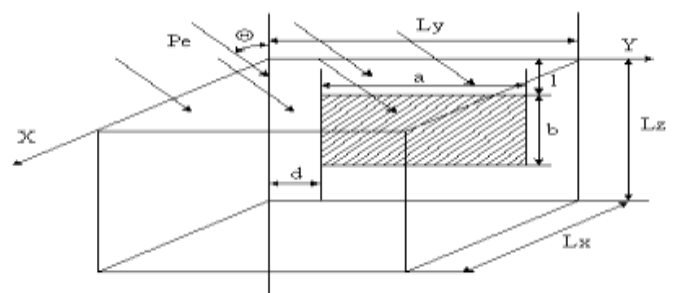
$$\nabla^2 p(\sigma, t) - (1/c_0^2)\partial^2 p(\sigma, t)/\partial^2 t = 0. \tag{1}$$

In this equation, p is the sound pressure, and c0 is the sound velocity.

If all the boundary are rigid, Eq.(1) has the general solution of the following form

$$p = \rho_0 c_0^2 V \sum_r \frac{F_r(\sigma) e^{j\omega_r^A t}}{\left(\omega_r^A\right)^2 M_r^A}. \tag{2}$$

In this equation, V is chamber volume, ω_r^A, F_r^A and M_r^A are the each order natural frequencies, the natural vibration mode function and the modal mass of of the cavity interior underwater acoustic field in the rigid boundary conditions respectively.

When there is a non rigid boundary, referencing Gauss formula can be obtained[6]

$$\ddot{P}_r + \left(\omega_r^A\right)^2 P_r = -A_r/V \qquad (r = 0,1,2,...). \tag{3}$$

In this equation,

$$P_r(t) = \frac{1}{V}\iiint_V \frac{p(\sigma,t)F_r(\sigma)}{\rho_0 c_0^2} dv. \tag{4}$$

$$A_r(t) = \oiint_D F_r(\sigma)a_n(\sigma,t)dA. \tag{5}$$

Consider the following that description of fluid solid coupling vibration with non rigid boundary. The modal equation obtained is

$$M_{Cj}^{(\iota)}\left\{\ddot{q}_j^{(\iota)} + C^{(\iota)}\dot{q}_j^{(\iota)} + \left[\omega_{Cj}^{(\iota)}\right]^2 q_j^{(\iota)}\right\} = Q_{Cj}^{(\iota)} + Q_{Ej}^{(\iota)}. \tag{6}$$

$\omega_{Cj}^{(\iota)}$ and $M_{Cj}^{(\iota)}$ are respectively the natural frequency and modal mass of boundary structure $D_C^{(\iota)}$. $C^{(\iota)} = Z_O/m_C^{(\iota)}$, $Q_{Cj}^{(\iota)}$ and $Q_{Ej}^{(\iota)}$ are respectively j order modal generalized incentives.

In the Eq. (5), let $a_n(\sigma, t) = \partial^2 w^{(\iota)}(\sigma, t)/\partial t^2$, and substituting into Eq. (3), At the same time to express $p_C^{(\iota)}$ of Eq. (6), finishing available

$$\begin{cases} \ddot{P}_r + \left(\omega_r^A\right)^2 P_r = -\dfrac{1}{V}\sum_\iota\sum_j L_{rj}^{(\iota)}\ddot{q}_j^{(\iota)} & r = 0,1,2,... \\ M_{Cj}^{(\iota)}\left\{\ddot{q}_j^{(\iota)} + C^{(\iota)}\dot{q}_j^{(\iota)} + \left[\omega_{Cj}^{(\iota)}\right]^2 q_j^{(\iota)}\right\} & j = 1,2,... \\ \qquad = V\sum_r \dfrac{L_{rj}^{(\iota)}P_r}{\left(\omega_r^A\right)^2 M_r^A} + Q_{Ej}^{(\iota)} & \iota = 1,2 \end{cases}. \tag{7}$$

In this equation, $L_{rj}^{(\iota)} = \iint_{D_C^{(\iota)}} F_r(\sigma)\Psi_j^{(\iota)}(\sigma)dA$. \tag{8}

Eq. (7) which describes the water filled closed chamber in fluid and elastic cavity wall occurred flow solid coupling vibration differential equation (Group).

Numerical simulation and analysis

Modal frequency and influencing factors of fluid solid coupling vibration

Table 1 compares the acoustic modal frequency of rigid boundary rectangular cavity $f_i^A = \omega_i^A/2\pi$, the modal frequency of elastic platform area wall board $D_C^{(1)}$ $f_{Ci}^{(1)} = \omega_{Ci}^{(1)}/2\pi$ and the modal frequency f_i of fluid-solid coupling vibration in two parts. $L_X = 0.4$m, $L_Y = 0.6$m, $L_Z = 0.7$m, $h^{(1)} = 5\times10^{-3}$m, the material of elastic wall panel $D_C^{(1)}$ is the No. 45 steel.

Visible from Table 1, The first two order natural frequency (non-zero) of cavity fluid and elastic wall cavity coupling vibration is close to the natural frequency of elastic chamber wall. However, with the increase of the modal order, the modal frequency of the coupled vibration tends to be lower than that of the cavity wall structure of the corresponding order. Further analysis also shows that strengthening elastic wall rigidity can make high order of fluid-solid coupling vibration modal frequency increased more significantly, but it has little effect on the first few order of fluid-solid

coupling modal frequency (Fig. 2); The change of fluid acoustic mode frequency in the cavity has little effect on the fluid-solid coupling modal frequency (Fig. 3).

Order	f_i^A(Hz)	$f_{Ci}^{(1)}$(Hz)	f_i (Hz)
0	0		4.63×10^{-5}
1	1071.43	107.963	109.2842
2	1250	207.622	205.3723
3	1646.35	332.195	247.3392
4	1875	373.719	293.087
5	2142.86	431.854	460.0024
6	2159.53	597.951	467.6415
7	2253.47	606.256	490.819
8	2480.79	705.915	578.766
9	2495.21	805.573	643.0509
10	2500	830.488	

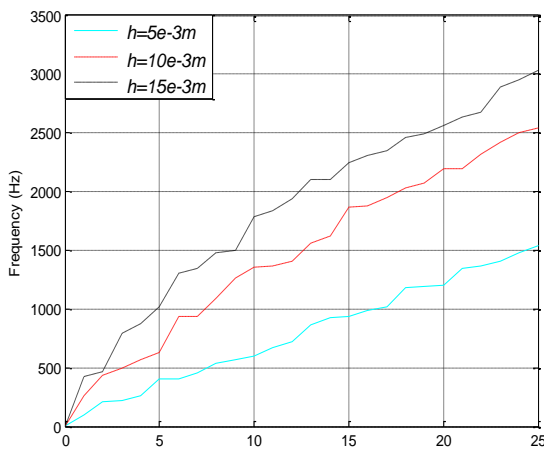Table 1: various modal frequencies of the rectangular cavity model

Fig. 2: Influence of $D_C^{(1)}$'s thicknesson coupling modal frequencies
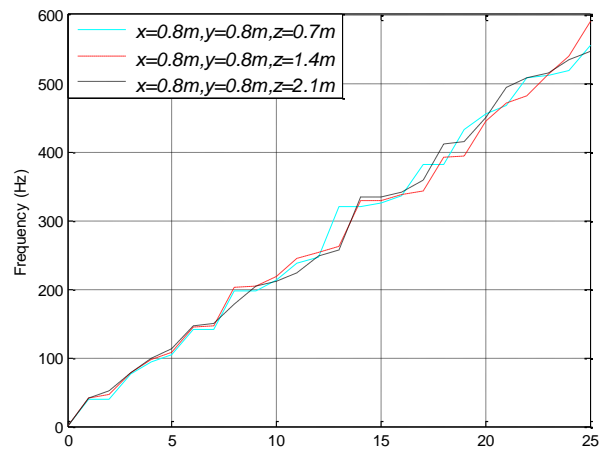

Fig. 3: Influence of cavity hight L_Z on coupling modal frequencies

The influence effect of acoustic window

The influence of acoustic window has not been considered in calculation in a lot of literatures, the acoustic window in the actual system usually adopts glass steel or titanium alloy material, and making it have approximately the same acoustic impedance ratio with the outside sea water, so acoustic window can also be approximately regarded as absorbing boundary.

Fig. 4 is introduced through the different acoustic pressure spectrum of cavity acoustic window boundary, it is visible that the acoustic window boundary has little influence on the vibration modal, but reduces the acoustic pressure peak of resonance frequency. In addition, the hydroacoustic pressure at each point in parallel with the sound source in the plane is roughly the same. And the hydroacoustic pressures at different depth in the lower frequency are roughly equal, but in the medium-high frequency domain, different sound source distances make larger difference with spectrum of hydroacoustic pressure.

Fig. 5 shows that the different sizes of the acoustic window border are not affected by the coupling mode frequency of the cavity. The large acoustic window can reduce the sound pressure resonance peak, but also reduces the noise attenuation in the anti resonance frequency band.
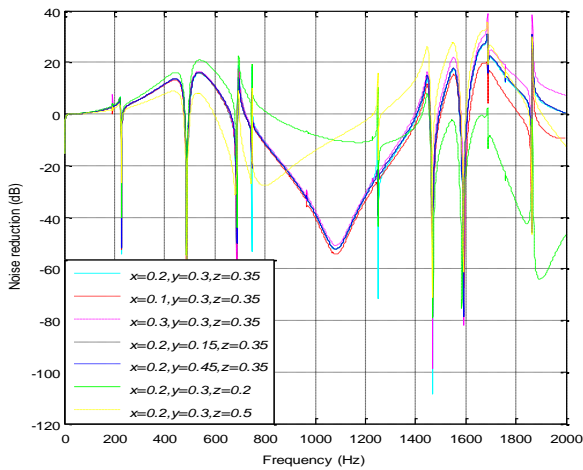
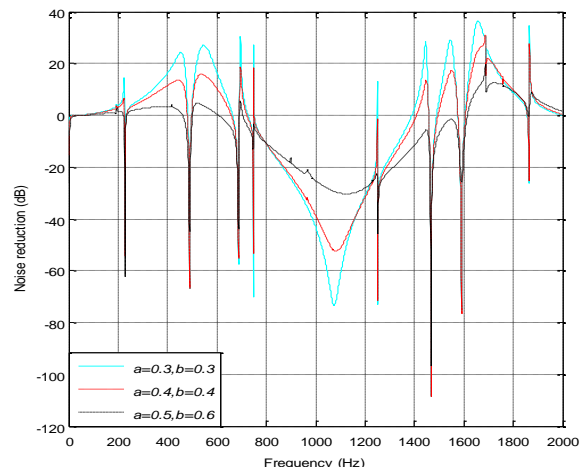Fig.4: Hydroacoustic pressure under consideration of sonar window

Fig. 5: Hydroacoustic pressurewith different sonar window size

Conclusion

The model of fluid-solid coupling vibration of plate and rectangular cavity with filling water was established and the underwater acoustic field was solved in this paper. The mechanical self-noise in sonar array cavity was qualitatively analyzed from theory.

The low order natural frequency of coupling vibration between cavity fluid and elastic chamber walls is more related to the structure modal frequency of elastic chamber wall than acoustic modal frequency of cavity fluid.

Acoustic window had a main influence on water acoustic field in the cavity that it could reduce the peak values of sound pressure at the resonant frequency and noise attenuation at the anti-resonant frequencies.

From the view of optimal design, the structure shape and size of cavity had great effect on its internal acoustic characteristics of underwater acoustic field, the influencing rule was very complex. With the increase of rigidity of the elastic wall plate, the number of the excited modal could be reduced; but it had little effect on the basic features of frequency distribution of underwater acoustic field.the Text

References

[1] Musha T, Kikuchi T. Numerical calculation for determining sonar self noise sources due to structural vibration[J]. Applied Acoustics. 1999, 58(1): 19-32.

[2] Mengsa Yu, Dongsheng Li. Calculation of hydrodynamic component of sonar self-noise by SEA method[J]. Journal of ship mechanics, 2004, 8(1): 99~105.

[3] Mengsa Yu, Zhu Zhengdao. Calculation of hydrodynamic component of sonar self-noise by integro-SEA method[J]. Journal of ship mechanics, 2007, 11(2): 273~283.

[4] Xiaobin Liu, Shijin LÜ, Mengsa Yu. Prediction and test of cavity's hydrodynamic self noise induced by turbulent boundary layer[J]. Acta Acustica, 2015, 40(6): 845~849.

[5] Hualin Guo, Dongping Luo, Meixia Chen, etc. Prediction method for low frequency of self-noise in submarine's forebody sonar platform area[J]. Ship science and technology. 2006, 27(4): 74~77.

[6] Dowell E H, Corman G F, Smith D A. Acoustoelasticity: general theory, acoustic natural modes and forced response to sinusoidal excitation, including comparisons with experiment[J]. Journal of sound and vibration, 1977, 52(4): 519~542.