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Analysis on spectra of hydroacoustic field in sonar cavity of the sandwich elastic wall structure

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Abstract. In this paper, the characteristics of the mechanical self - noise in sonar array cavity are studied by using the elastic flatbed - filled rectangular cavity parameterization model. Firstly, the analytic derivation of the vibration differential equation of the single layer, sandwich elastic wall plate structure and internal fluid coupling is carried out, and the modal method is used to solve it. Finally, the spectral characteristics of the acoustic field of rectangular cavity of different elastic wallboard materials are simulated and analyzed, which provides a theoretical reference for the prediction and control of sonar mechanical self-noise. In this paper, the sandwich board as control inside the dome background noise of a potential means were discussed, the dome background noise of qualitative prediction analysis and control has important theoretical significance.

1. Introduction

The mechanical self - noise system in sonar array is transmitted to the sonar position by the vibration of the ship's vibration source device, it is particularly important for the low-speed submarine underwater navigation. According to some test and analysis results, the typical mechanical noise sources such as main motor slot frequency and high order harmonic, podium shell around the low-frequency pattern, they transmit distance and stability, which is an important factor in the low frequency detection ability of sonar[1].

Dowell[2] has made theoretical analysis on the model of the coupling system of the plate and closed space, and the general form of the vibration coupling problem of the acoustic field and solid structure is given by the green function method[3]. Pan[4] shows that the response of the coupled system can be regarded as the result of the coupling of plate mode and cavity mode. Because the sandwich panel has the advantages of high strength, good sound insulation performance, strong design sex. As early as 1959, was proposed to improve the sound insulation performance of plate[5], used in the field of aerospace and Marine industry more and more widely. Zhenyu Feng[6] studied free vibration problems of several main viscoelastic model sheet. Cupial[7] calculated the free vibration problem of core material based on the constant complex model of constraint damping plate, studied the free vibration characteristic of the sandwich board. Narayanan[8] calculated the sound field of noise through the sandwich plate into the cavity.

2. Theoretical model

2.1. The closed-cavity indoor underwater acoustic field with a fluid-elastic structure coupled boundary

Figure 1 sandwich board - rectangular acoustic field coupling model, the upper is subjected to external excitation pe has elastic and viscoelastic - elastic three layers structure of sandwich board D_B, the rest of the wall are rigid (D_S). Underwater acoustic field of water filling airtight chamber interior ministry satisfies wave equation.

$$\nabla^2 p(\sigma,t) - \frac{1}{c_0^2}\frac{\partial^2 p(\sigma,t)}{\partial t^2} = 0 \tag{1}$$

In this equation, p is the sound pressure, and c_0 is the sound velocity.

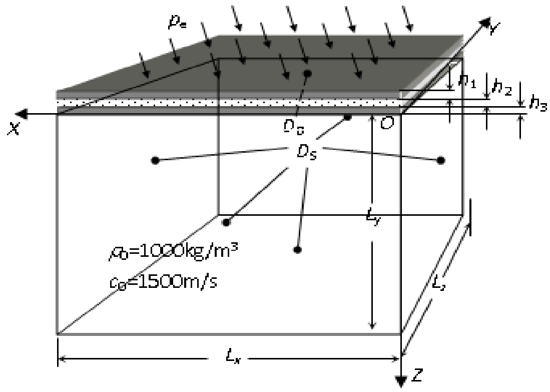

Figure 1. Coupling model of sandwich panel - rectangular underwater acoustic field

On the elastic sandwich panel($\sigma \in D_B$), Eq. (1) has boundary conditions

$$\partial p(\sigma,t)/\partial n = -\rho_0 a_n(\sigma,t) \tag{2}$$

In the formula, n is the outer normal direction of the boundary wall plate, ρ_0 is the medium (water)density, and an is the normal acceleration of the vibration of the sandwich plate.

On the rigid wall boundary($\sigma \in D_S$), Eq. (1) has boundary conditions

$$\partial p(\sigma,t)/\partial n = 0 \tag{3}$$

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

$$p = \rho_0 c_0^2 V \sum_r \frac{F_r^{A}(\sigma) e^{j\omega_r^{A} t}}{\left(\omega_r^{A}\right)^2 M_r^{A}} \tag{4}$$

For rectangular chamber

$$\omega_r^{A} = c_0 \sqrt{\left(\frac{m_r^{A}\pi}{L_x}\right)^2 + \left(\frac{n_r^{A}\pi}{L_y}\right)^2 + \left(\frac{l_r^{A}\pi}{L_z}\right)^2} \tag{5}$$

$$F_r^{A}(\sigma) = \cos\left(\frac{m_r^{A}\pi x}{L_x}\right)\cos\left(\frac{n_r^{A}\pi y}{L_y}\right)\cos\left(\frac{l_r^{A}\pi z}{L_z}\right) \tag{6}$$

$$\rho_0 c_0^2 \left(\omega_r^{A}\right)^2 M_r^{A} = \iiint_V \left[F_r^{A}(\sigma)\right]^2 \mathrm{d}\sigma \tag{7}$$

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

$$\ddot{P}_r + \left(\omega_r^{A}\right)^2 P_r = -\frac{A_r}{V} \quad (r = 0,1,2,\ldots) \tag{8}$$

2.2. Vibration response of sandwich elastic wall plate

The motion differential equation of viscoelastic sandwich board under transverse excitation is established under the following assumptions by Mead: (1) In the sandwich panel, there is no significant positive strain in the direction perpendicular to the plate, so the three layers have uniform lateral vibration displacement. (2) The bending stress within the sandwich layer is negligible compared with the normal stress in the panel. (3) The shear strain of the sandwich layer in the direction of the panel is ignored. (4) The contact surface of the panel and sandwich layer has the continuity of displacement, without the continuity of stress.

$$D_{\mathrm{t}}\nabla^4 w-\frac{G_{\mathrm{c}}(h_1+h_2)}{h_2}\left[(h_1+h_2)\nabla^2 w-2\left(\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}\right)\right]+(2\rho_{\mathrm{p}}h_1+\rho_{\mathrm{c}}h_2)\frac{\partial^2 w}{\partial t^2}=q(x,y,t) \tag{9}$$

$$\frac{E}{1-\mu^2}\frac{\partial^2 u}{\partial x^2}+\frac{E}{2(1+\mu)}\frac{\partial^2 u}{\partial y^2}-\frac{2G_{\mathrm{c}}}{h_1h_2}u+\frac{E}{2(1-\mu)}\frac{\partial^2 v}{\partial x\partial y}+\frac{G_{\mathrm{c}}(h_1+h_2)}{h_1h_2}\frac{\partial w}{\partial x}=0 \tag{10}$$

$$\frac{E}{1-\mu^2}\frac{\partial^2 v}{\partial y^2}+\frac{E}{2(1+\mu)}\frac{\partial^2 v}{\partial x^2}-\frac{2G_{\mathrm{c}}}{h_1h_2}v+\frac{E}{2(1-\mu)}\frac{\partial^2 u}{\partial x\partial y}+\frac{G_{\mathrm{c}}(h_1+h_2)}{h_1h_2}\frac{\partial w}{\partial y}=0 \tag{11}$$

The differential equations of lateral displacement w can be obtained by algebraic manipulation of equation (9) ~ (11)

$$D_{\mathrm{t}}\nabla^6 w-D_{\mathrm{t}}g(1+Y)\nabla^4 w+(\rho h)\partial^2(\nabla^2 w)/\partial t^2-g(\rho h)\partial^2 w/\partial t^2=(1-g)q(x,y,t) \tag{12}$$

Applying the simple damping mode, the solution of equation (12) has the superposition of vibration mode

$$w(x,y,t)=\sum_j W_j^{\mathrm{B}}(x,y)B_j(t) \tag{13}$$

$$\nabla^6 W-g(1+Y)\nabla^4 W-\Omega^2\nabla^2 W+g\Omega^2 W=0 \tag{14}$$

In order to simplify the calculation, consider the situation of the boundary conditions of the four sides

$$W_j^{\mathrm{B}}(x,y)=\cos\left[m_j^{\mathrm{B}}\pi\left(\frac{x}{L_x}-\frac{1}{2}\right)\right]\sin\left(\frac{n_j^{\mathrm{B}}\pi y}{L_y}\right) \tag{15}$$

$$\omega_j^{\mathrm{B}}=\overline{\omega}_j^{\mathrm{B}}\sqrt{1+\mathrm{j}\eta_j^{\mathrm{B}}}=\pi^2\gamma_j^{\mathrm{B}}\sqrt{\frac{D_{\mathrm{t}}}{(\rho h)}\left[1+\frac{gYL_x^2}{\gamma_j^{\mathrm{B}}+gL_x^2}\right]} \tag{16}$$

The modal coordinate equation can be obtained

$$M_j^{\mathrm{B}}\left[\ddot{B}_j+(\omega_j^{\mathrm{B}})^2 B\right]=Q_j \quad (j=1,2,3,\ldots) \tag{17}$$

2.3. Coupling vibration response of underwater acoustic field - sandwich plate

The underwater acoustic field, On the rigid wall D_{S}, a_n=0, On the fluid-structure coupling boundary, $a_n=\partial^2 w/\partial t^2$.

$$A_r(t)=\iint_{D_{\mathrm{B}}} F_r^{\mathrm{A}}\Big|_{z=0}\sum_j W_j^{\mathrm{B}}\ddot{B}_j\mathrm{d}A=\sum_j L_{rj}\ddot{B}_j \tag{18}$$

The sandwich board, making the generalized force $Q_j=Q_j^{\mathrm{A}}+Q_j^{B}$, Q_j^{A} is the generalized force corresponding to the internal fluid excitation of the cavity. Q_j^{B} is the generalized force corresponding to the external excitation p_{e} in the direction of the direction of the panel.

$$Q_j^{\mathrm{A}}=\iint_{D_{\mathrm{B}}} p\Big|_{z=0}W_j^{\mathrm{B}}\mathrm{d}A=\iint_{D_{\mathrm{B}}} V\sum_r\frac{F_r^{\mathrm{A}}\Big|_{z=0}P_r}{(\omega_r^{\mathrm{A}})^2 M_r^{\mathrm{A}}}W_j^{\mathrm{B}}\mathrm{d}A=V\sum_r\frac{L_{rj}P_r}{(\omega_r^{\mathrm{A}})^2 M_r^{\mathrm{A}}} \tag{19}$$

The fluid - structure coupled vibration equation

$$V\left[\ddot{P}_r+\left(\omega_r^{\mathrm{A}}\right)^2 P_r\right]=-\sum_j L_{rj}\ddot{B}_j\ ,\ M_j^{\mathrm{B}}\left[\ddot{B}_j+\left(\omega_j^{\mathrm{B}}\right)^2 B\right]=\sum_r \frac{VL_{rj}P_r}{\left(\omega_r^{\mathrm{A}}\right)^2 M_r^{\mathrm{A}}}-Q_j^{\mathrm{B}} \tag{20}$$

Taking the first R+1 order of hydroacoustic field mode, the first N order of the sandwich plate mode. $\boldsymbol{P}=[P_0, P_1, P_2, \ldots, P_R]^{\mathrm{T}}$, $\boldsymbol{B}=[B_1, B_2, \ldots, B_N]^{\mathrm{T}}$, matrix equation form of equation (20)

$$V\left(\ddot{\boldsymbol{P}}+\boldsymbol{\Omega}_{\mathrm{A}}^2\boldsymbol{P}\right)=-\boldsymbol{L}\ddot{\boldsymbol{B}} \tag{21}$$

$$\boldsymbol{M}_{\mathrm{B}}\left(\ddot{\boldsymbol{B}}+\boldsymbol{\Omega}_{\mathrm{B}}^2\boldsymbol{B}\right)=V\boldsymbol{L}^{\mathrm{T}}\boldsymbol{\Omega}_{\mathrm{A}}^{-2}\boldsymbol{M}_{\mathrm{A}}^{-1}\boldsymbol{P}+\boldsymbol{Q}_{\mathrm{B}} \tag{22}$$

Making $\boldsymbol{Q}_{\mathrm{B}}$=0, and assuming that it has vibration solution

$$\boldsymbol{P}=\frac{\boldsymbol{\Omega}_{\mathrm{A}}\boldsymbol{M}_{\mathrm{A}}^{1/2}}{\sqrt{V}}\boldsymbol{\chi}_{\mathrm{A}}\mathrm{e}^{\mathrm{j}\omega t}\ ,\ \boldsymbol{B}=\sqrt{V}\left(\boldsymbol{\Omega}_{\mathrm{B}}\boldsymbol{M}_{\mathrm{B}}^{1/2}\right)^{-1}\boldsymbol{\chi}_{\mathrm{B}}\mathrm{e}^{\mathrm{j}\omega t} \tag{23}$$

Substituting the upper equation into the equation (21) and (22), the eigenvalue problem

$$\boldsymbol{A\chi}=\begin{bmatrix}\boldsymbol{A}_{11} & \boldsymbol{A}_{12}\\ \boldsymbol{A}_{21} & \boldsymbol{A}_{22}\end{bmatrix}\begin{bmatrix}\boldsymbol{\chi}_{\mathrm{A}}\\ \boldsymbol{\chi}_{\mathrm{B}}\end{bmatrix}=\omega^2\begin{bmatrix}\boldsymbol{\chi}_{\mathrm{A}}\\ \boldsymbol{\chi}_{\mathrm{B}}\end{bmatrix}=\omega^2\boldsymbol{\chi} \tag{24}$$

$\boldsymbol{A}_{11}=\boldsymbol{\Omega}_{\mathrm{A}}^{2}+(\boldsymbol{\Omega}_{\mathrm{A}}\boldsymbol{M}_{\mathrm{A}}^{1/2})^{-1}\boldsymbol{L}\boldsymbol{M}_{\mathrm{B}}^{-1}\boldsymbol{L}^{\mathrm{T}}(\boldsymbol{\Omega}_{\mathrm{A}}\boldsymbol{M}_{\mathrm{A}}^{1/2})^{-1}$, $\boldsymbol{A}_{12}=\boldsymbol{A}_{21}^{\mathrm{T}}=-(\boldsymbol{\Omega}_{\mathrm{A}}\boldsymbol{M}_{\mathrm{A}}^{1/2})^{-1}\boldsymbol{L}(\boldsymbol{\Omega}_{\mathrm{B}}\boldsymbol{M}_{\mathrm{B}}^{1/2})^{-1}$, $\boldsymbol{A}_{22}=\boldsymbol{\Omega}_{\mathrm{B}}^{2}$

The coupling mode function of cavity and sandwich plate is respectively

$$F_k^{\mathrm{C}}=\sqrt{V}\boldsymbol{F}_{\mathrm{A}}^{\mathrm{T}}\left(\boldsymbol{\Omega}_{\mathrm{A}}\boldsymbol{M}_{\mathrm{A}}^{1/2}\right)^{-1}\boldsymbol{\chi}_{\mathrm{A}}^{(k)}=\boldsymbol{F}_{\mathrm{A}}^{\mathrm{T}}\boldsymbol{\kappa}_{\mathrm{A}}\boldsymbol{\chi}_{\mathrm{A}}^{(k)} \tag{25}$$

$$W_k^{\mathrm{C}}=\sqrt{V}\boldsymbol{W}_{\mathrm{B}}^{\mathrm{T}}\left(\boldsymbol{\Omega}_{\mathrm{B}}\boldsymbol{M}_{\mathrm{B}}^{1/2}\right)^{-1}\boldsymbol{\chi}_{\mathrm{B}}^{(k)}=\boldsymbol{W}_{\mathrm{B}}^{\mathrm{T}}\boldsymbol{\kappa}_{\mathrm{B}}\boldsymbol{\chi}_{\mathrm{B}}^{(k)} \tag{26}$$

Fluid-structure coupling vibration response (initial conditions of 0)

$$p(\sigma,t)=-\boldsymbol{F}_{\mathrm{A}}^{\mathrm{T}}\boldsymbol{\kappa}_{\mathrm{A}}\boldsymbol{X}_{\mathrm{A}}\boldsymbol{\alpha}(t) \tag{27}$$

$$w(x,y,t)=\boldsymbol{W}_{\mathrm{B}}^{\mathrm{T}}\boldsymbol{\kappa}_{\mathrm{B}}\boldsymbol{X}_{\mathrm{B}}\boldsymbol{\alpha}(t) \tag{28}$$

3. Numerical simulation and analysis

3.1. Modal frequency and influencing factors of fluid-solid coupled vibration

Table 1 is the modal frequency of the rectangular plates with four edges and the rectangular plates obtained by applying formula (16), and compared with the single layer board. Sandwich board of the upper and lower panels are 45 # steel, sandwich layer is viscoelastic damping materials (taking its density ratio 0.2 with panel material, the damping loss factor β=0.3, the dimensionless shear modulus g=1+jβ).

Table 1. The modal frequency of the rectangular plate with four edges and the comparison with the single layer plate

	Single sheet elastic sheet L_x=0.4m,L_y=0.6m,h=0.005m		Sandwich board 1 L_x=0.4m,L_y=0.6m h_1=0.0015m,$_2$=0.002m		Sandwich board 2 L_x=0.4m,L_y=0.6m,h_1=h_2=0.005m	
j	$(m_j^{(1)}, n_j^{(1)})$	$f_j^{(1)}$(Hz)	$(m_j^{(1)}, n_j^{(1)})$	$f_j^{(1)}$(Hz)	$(m_j^{(1)}, n_j^{(1)})$	$f_j^{(1)}$(Hz)
1	1,1	107.963	1,1	43.9552	1,1	138.018
2	1,2	207.622	1,2	73.5652	1,2	236.412
3	2,1	332.195	2,1	109.445	2,1	356.831
4	1,3	373.719	1,3	121.303	1,3	396.749
5	2,2	431.854	2,2	137.859	2,2	452.538

3.2. The spectral characteristics of sound pressure of underwater acoustic field

Assuming that the sandwich panel is used to load the uniform-changing mechanical load $p_E(\sigma, t)=P_E e^{j\omega t}, P_E=1$.Taking the excitation frequency of continuous change and the acoustic pressure of the cavity is drawn, water sound pressure spectrum are shown in figure 2 ~ 7.

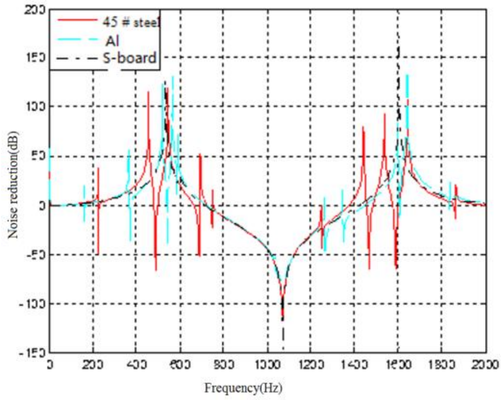

Figure 2. Comparison of internal water pressure in rectangular cavity of elastic wall of different materials

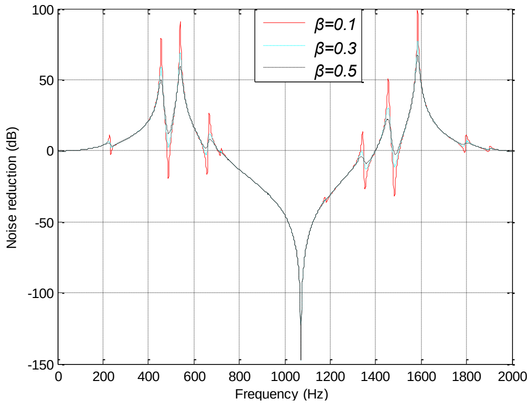

Figure 3. The influence of sandwich damping on the acoustic pressure of the internal water

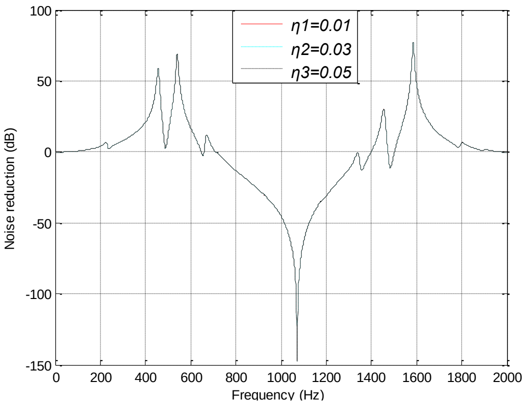

Figure 4. The effect of panel damping on noise reduction in cavity is studied

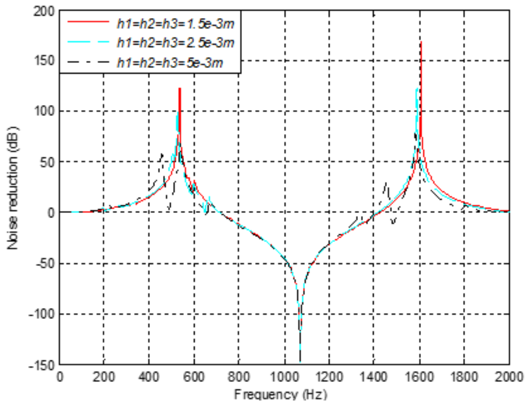

Figure 5. The influence of different plate thickness on noise reduction of cavity

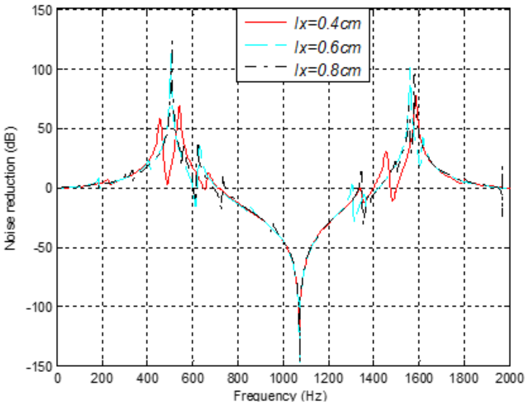

Figure 6. The influence of different *Lx* sides on the acoustic field of the cavity

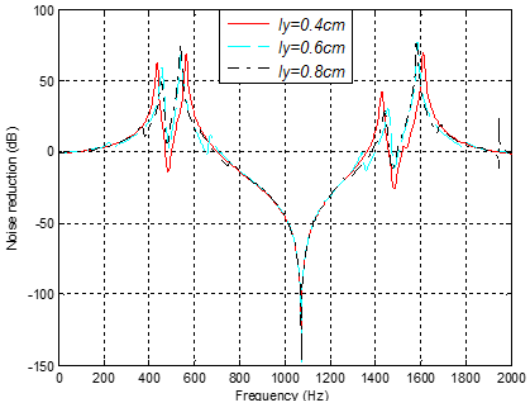

Figure 7. The influence of different *Ly* sides on the acoustic field of the cavity

Figure 2 compares the acoustic pressure spectrum of the rectangular cavity of three different elastic wall plates, sandwich plate, 45 # steel and aluminum alloy. The structure of sandwich plate greatly reduces the number of resonant peaks.In addition, the rectangular cavity of aluminum alloy wall plates has lower resonance frequency than 45 # steel wall rectangular cavity. Figure 3 and 4 show that the inner damping of the sandwich and panel is used only to weaken the resonance peak. Figure 5 shows that the thickness of sandwich plate is not affected by the thickness of the laminates. In detail, the resonant peaks of the thicker plates are more known, but the peaks of each resonance are reduced. Figure 6 changes the length of the sandwich plate when L_y=0.6m, L_z=0.7m. Figure 7 When L_x=0.4m, L_z=0.7m, the length of the sandwich panel is changed. The size change of sandwich plate will change the mode of vibration mode, and the coupling resonant frequency will change. The overall change trend is that the size of the plate decreases and the resonant frequency increases, which is conducive to the control of low frequency noise. In addition, it is worth pointing out that the number of resonant peaks that are stimulated in all square plates is significantly less than that of rectangular plates.

4. Conclusion

This paper presents a rectangular cavity structure composed of a sandwich panel by establishing a plate-filled rectangular cavity fluid-solid coupled vibration model and solving the cavity of the cavity. The coupling vibration response of the underwater acoustic field is derived. The results of MATLAB simulation show that the clamping plate of the same thickness is better than that of the rectangular laminates with the same thickness, and the noising effect of the rectangular sandwich panel of the square is better than that of the rectangle. When the plate is rectangular, as the size of the plate increases, its noise reduction effect is better. The damping of the sandwich plate and the loss factor of the sandwich layer also have an effect on the noise reduction of the plate. The damping and increasing loss factors can reduce the mutation of the internal noise in the vicinity of the resonance point.

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