ACOUSTIC TIME-FREQUENCY PREDICTION SIMULATION METHOD IN SHIP STRUCTURE

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Acoustic prediction methods in ship structure mostly applies frequency domain mode superposition method at present, which can achieve structural vibration and noise prediction in specified frequencies. But this kind of method remains some disadvantages such as the lack of prediction accuracy and effectiveness. Against above problems exist in ship structure vibration and noise prediction domain, this paper is based on the wave method, introducing time domain wave analysis method into ship structure wide frequency linear-spectrum vibration and noise prediction process, trying to improve the problem of accuracy and effectiveness.
Keywords: time-frequency; simulation ship; wave method

1. INTRODUCTION

The presented acoustic prediction methods for ship (such as: Acoustic FEM, Acoustic FEM and BEM) are based on modal theory in frequency domain. These prediction methods possess the advantage of intuitive analysis. However these prediction methods still have series of shortcomings follow as:

Numerical acoustic prediction methods ignore many nonlinear influence ,due to basing on linear superposition principle.

Coupling influence of vibration in the low-frequency with wave-motion in the high-frequency is not taken into account.

Usually, scale of computing is large and frequency band of computing is narrow. Sound field distribution at one frequency point is merely obtained after one calculation. Therefore, many times calculation need carry out, if acoustic prediction of model in frequency band is accomplished. So, the efficient of solving is low. Furthermore, formant will be missed if the inappropriate calculate step is selected, which will bring obvious error for calculated results.

For the problems in the acoustic prediction, such as, the low calculation efficiency, the irrespective of nonlinear influence and coupling of vibration in the low-frequency with wave-motion in the high-frequency, the acoustic time-frequency complex prediction method based on wave theory is advanced. The method can solve the shortcomings of the irrespective of nonlinear influence and coupling of vibration in the low-frequency with wave-motion in the high-frequency.

The method avoids the phenomenon of leakage in ship structure acoustic forecast peak, with the purpose to provide the method about acoustic prediction and evaluation, and noise control for ship and ocean structure.

2. THEORY

2.1. The theory of time domain analysis for structural dynamic response

Time domain analysis for structural dynamic response is mainly applied to the analysis of transient structure, transient acoustic and nonlinear structure. The method obtains structural dynamic response in arbitrary time period, through carrying out numerical integration for the motion equation of the coupled system in time domain. In each time step Δt, dynamic response recognized as liner system is respectively calculated. Then, system parameters (displacement, stress, pressure, sound pressure) of structure are revised according to the results in the time step, which will be as the initial value in next time step. Therefore, dynamic response of nonlinear system is approximated to be series variable dynamic response of liner system.

To arbitrary multi-degree of freedom system, whether it is a linear system or nonlinear system, the motion equation can be written as

$$\{F^{\mathrm{I}}\}+\{F^{\mathrm{D}}\}+\{F^{\mathrm{S}}\}=\{P\} \tag{1}$$

Where {P} is the vector of exciting force. {F I} is the vector of inertial force. {F D} is the vector of damping force. {FS} is the vector of structural force resisting deformation.

Assuming the Eq.(1) state parameters of initial time t and ending time t+Δt at each integration step have known.

$$\{F_{\mathrm{t}}^{\mathrm{I}}\}+\{F_{\mathrm{t}}^{\mathrm{D}}\}+\{F_{\mathrm{t}}^{\mathrm{S}}\}=\{P_{\mathrm{t}}\} \tag{2}$$

$$\{F_{\mathrm{t}}^{\mathrm{I}}+\Delta F_{\mathrm{t}}^{\mathrm{I}}\}+\{F_{\mathrm{t}}^{\mathrm{D}}+\Delta F_{\mathrm{t}}^{\mathrm{D}}\}+\{F_{\mathrm{t}}^{\mathrm{S}}+\Delta F_{\mathrm{t}}^{\mathrm{S}}\}=\{P_{\mathrm{t+\Delta t}}\} \tag{3}$$

$$\Delta F_{\mathrm{t}}^{\mathrm{I}}=F_{\mathrm{t+\Delta t}}^{\mathrm{I}}-F_{\mathrm{t}}^{\mathrm{I}}=\left[M_T\right]\{\Delta\ddot{x}\} \tag{4}$$

$$\Delta F_{\mathrm{t}}^{\mathrm{D}}=F_{\mathrm{t+\Delta t}}^{\mathrm{D}}-F_{\mathrm{t}}^{\mathrm{D}}=\left[C_T\right]\{\Delta\dot{x}\} \tag{5}$$

$$\Delta F_t^S=F_{\mathrm{t+\Delta t}}^{\mathrm{S}}-F_{\mathrm{t}}^{\mathrm{S}}=\left[K_T\right]\{\Delta x\} \tag{6}$$

$$\Delta P_t=P_{t+\Delta t}-P_t \tag{7}$$

The motion equation at each integration step is written as equation in terms of increment

$$[M_T]\{\Delta\ddot{x}\}+[C_T]\{\Delta\dot{x}\}+[K_T]\{\Delta x\}=\{\ \Delta P_t\ \} \tag{8}$$

Where, [MT], [CT],[KT] are respectively mass matrix, damping matrix and stiffness matrix of increment equation.

$$k_{ij}(t)=\left(\frac{\partial F_i^S(x)}{\partial x_j}\right)_t;\quad m_{ij}(t)=\left(\frac{\partial F_i^I(\ddot{x})}{\partial \ddot{x}_j}\right)_t;\quad c_{ij}(t)=\left(\frac{\partial F_i^D(\dot{x})}{\partial \dot{x}_j}\right)_t \tag{9}$$

The coefficient can be described in terms of matrix as

$$[M_T]=\left(\frac{\partial\{F^I\}}{\partial\{\ddot{x}\}}\right)_t;\quad [K_T]=\left(\frac{\partial\{F^S\}}{\partial\{x\}}\right)_t;\quad [C_T]=\left(\frac{\partial\{F^D\}}{\partial\{\dot{x}\}}\right)_t \tag{10}$$

In the dynamics, inertial force is the linear function of acceleration. Mass matrix is constant coefficient matrix. Damping force of structure is converted into constant coefficient damping matrix by equivalent linear method. Usually, stiffness matrix is variable coefficient matrix.

If inertial force is the linear function of acceleration and nonlinear force {F} depends on $\{\dot{x}\}$ and {x}, the motion equation can be simplified as

$$[M]\{\ddot{x}\}+\{F(x,\dot{x})\}=\{P\} \tag{11}$$

Increment equation is

$$[M]\{\Delta\ddot{x}\}+[C_T]\{\Delta\dot{x}\}+[K_T]\{\Delta x\}=\{\Delta P_t\} \tag{12}$$

Tangent damping matrix [CT] and tangent stiffness matrix [KT] are defined to be

$$[C_T]=\left(\frac{\partial\{F(x,\dot{x})\}}{\partial\{\dot{x}\}}\right)_t;\quad [K_T]=\left(\frac{\partial\{F(x,\dot{x})\}}{\partial\{x\}}\right)_t \tag{13}$$

Generally, when nonlinear force {F} merely depends on displacement {x}, differential equation describes as

$$[M]\{\ddot{x}\}+[C]\{\dot{x}\}+\{F(x)\}=\{P\} \tag{14}$$

Increment equation is

$$[M]\{\Delta\ddot{x}\}+[C]\{\Delta\dot{x}\}+[K_T]\{\Delta x\}=\{\Delta P_t\} \tag{15}$$

Where, the definition of tangent stiffness matrix is same as Eq.(13). The motion equation of structure at the time $t+\Delta t$ is

$$[M]\{\ddot{x}_{n+1}\}+[C]\{\dot{x}_{n+1}\}+[K_T]\{\Delta x_n\}=\{P_{n+1}\}-\{F(x_n)\} \tag{16}$$

The relationship of acceleration, velocity and displacement at xn and xn+1 is constructed through Eq.(16).

Increment equation can be solved by the method of FEM, Newmark-β, and Wilson-θ. For example of central difference method, displacement (velocity, acceleration) is expanded in terms of Taylor series as

$$x(t+\Delta t)=x(t)+\dot{x}(t)\Delta t+\frac{1}{2}\ddot{x}(t)\Delta t^2+\frac{1}{6}\dddot{x}(t)\Delta t^3+\cdots \tag{17}$$

Assuming

$$x(t+\Delta t)=x_{n+1}\,;\quad x(t)=x_{\mathrm{n}}\,;\quad x(t\text{-}\Delta t)=x_{\mathrm{n}-1} \tag{18}$$

The former differential equation is obtained by Eq.(18)

$$x_{\mathrm{n}+1}=x_n+\dot{x}_n\Delta t+\frac{1}{2}\ddot{x}_n\Delta t^2+\frac{1}{6}\dddot{x}_n\Delta t^3+\cdots \tag{19}$$

The after differential equation is written as

$$x_{\mathrm{n}-1}=x_n-\dot{x}_n\Delta t+\frac{1}{2}\ddot{x}_n\Delta t^2-\frac{1}{6}\dddot{x}_n\Delta t^3+\cdots \tag{20}$$

The below equations can be obtained through adding or misusing Eq.(19) and Eq.(20).

$$\dot{x}_n\Delta t=\frac{1}{2}(x_{\mathrm{n}+1}-x_{\mathrm{n}-1})+\mathrm{O}\left(\Delta t^3\right) \tag{21}$$

$$\ddot{x}_n\Delta t^2=(x_{\mathrm{n}+1}-2x_{\mathrm{n}}+x_{\mathrm{n}-1})+\mathrm{O}\left(\Delta t^4\right) \tag{22}$$

Velocity and acceleration at the time t can be approximately described by the displacements of three points (n-1, n, n+1)

$$\dot{x}_n=\frac{1}{2\Delta t}(x_{\mathrm{n}+1}-x_{\mathrm{n}-1}) \tag{23}$$

$$\ddot{x}_n=\frac{1}{\Delta t^2}(x_{\mathrm{n}+1}-2x_{\mathrm{n}}+x_{\mathrm{n}-1}) \tag{24}$$

Displacement, velocity and acceleration at the time t satisfy the differential equation follow as

$$m\{\ddot{x}_{\mathrm{n}}\}+c\{\dot{x}_{\mathrm{n}}\}+k\{x_{\mathrm{n}}\}=f_n \tag{25}$$

Substitution Eqs.(23-24) into Eq.(25), the equation is obtained

$$\hat{m}x_{\mathrm{n}+1}=\hat{f}_n \tag{26}$$

Where $\hat{m}=\left(\frac{m}{\Delta t^2}+\frac{c}{2\Delta t}\right)$; $\hat{f}_n=f_n-\left(k-\frac{2}{\Delta t^2}m\right)x_n-\left(\frac{1}{\Delta t^2}m-\frac{1}{2\Delta t}c\right)x_{n-1}$

For a single degree of freedom system, the displacement xn+1 at the time t+Δt can be gained through Eq.(26). For multi-degree of freedom system, after coefficient is replaced by matrix and displacement, velocity and acceleration is written as vector, the vibration of structure is also obtained by the same method.

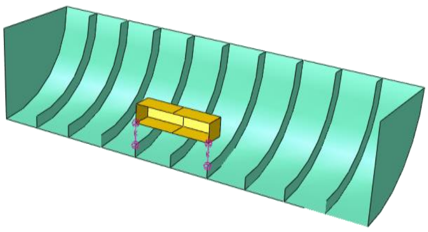
The method of time domain analysis is not only applied to dynamic analysis, but also widely used to transient radiated noise and the field of ship underwater explosion[1-3] . The basic principle is consistent with the above, which will not be repeated here.

3. THE VALIDATION OF METHOD

The paper will valid the effectiveness of the acoustic time-frequency complex prediction method of ship. Due to the acoustic time-frequency complex prediction method include acoustic time domain prediction and acoustic frequency domain prediction, and the correctness of acoustic frequency domain prediction method are accepted by ship industry, so, the paper just need to valid the the effectiveness of the acoustic time domain prediction method.

3.1. Introduction of validation model

Ship cabin model are choose to be validation model, which is shown in fig.1. The model parameter is that, ship structure is semi-cylindrical shell, its radius is R=1500mm , the shell thickness is t=10mm ,the bulkheads locate at the ends of ship cabin, ship cabin includes some ribs. Rib spacing is L=600mm , the rib parameter is t=10mm、h=200mm. The equipment parameter is 1200mm×600mm×300mm . The shell thickness of equipment is t=20mm. Two vibration isolator support the ends of equipment along the center line of ship cabin model. The stiffness of vibration isolator is k=4kN/m . The boundary condition at the ends of the ship cabin model is simply-simply. Equipment exciting load is $F = 1\times\sin(2\pi ft)$ N , which vertically acts on the the center point of equipment. The radius of fluid field is R=6m. The outer surface of fluid field is laid by infinite element mesh, which is shown in fig.2(a). The calculation frequency range is 20Hz~400Hz and the frequency spacing is Δf =5Hz, Damping is η =0.05. In order to facilitate the contrast, the acoustic time domain prediction model and the acoustic frequency domain prediction model are the same model. The ship cabin model consist of 4736 quadrilateral linear elements. The fluid field model consist of 45710 hexahedral elements and 3716 infinite elements. The models have 54612 elements in total.



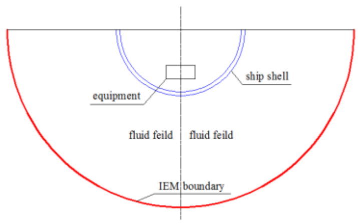


Fig. 1 Structure of radiated noise model of a ship cabin

To facilitate the contrast analysis, vibration and sound pressure observation points are set in the ship cabin model and fluid field model. Vibration observation points locate on the location of ribs along the ship cabin symmetric axis, which is seen in fig.2. Fluid field sound pressure observation points are laid in the interface of ship cabin and fluid field, and the fluid field below directly with radius R=6m . The locations of fluid field sound pressure observation points is shown in fig.2.

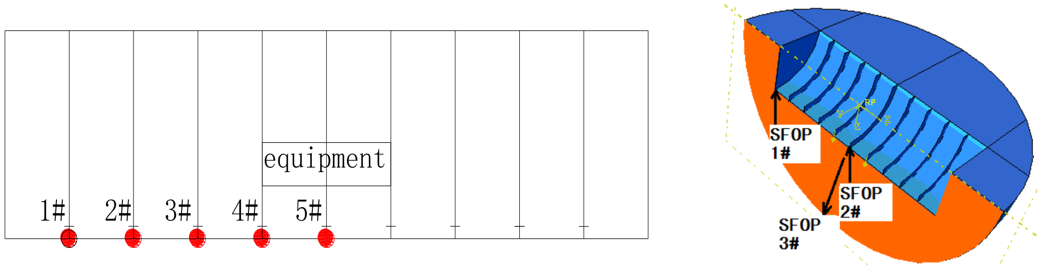


Fig.2 Sketch of vibratioin and sound pressure observation points of ship cabin

Due to the frequency exciting loads need to transfer into time exciting loads in acoustic time domain analysis, the paper transfer the frequency exciting loads into time exciting loads through the Eq.(29).

$$F(t) = \sum_{i=0}^{76} \sin(2\pi(20+5i)t) \tag{29}$$

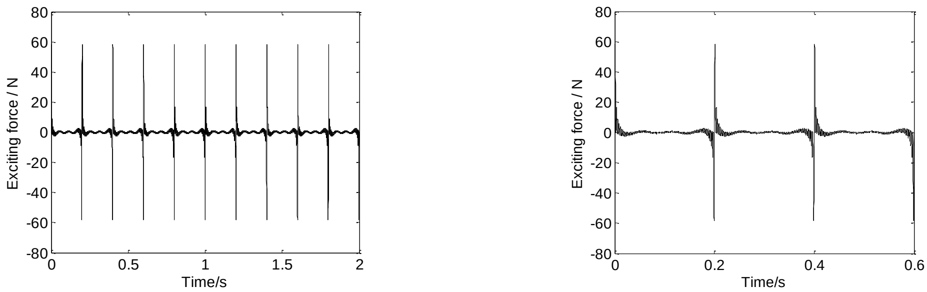


Fig.3 Load curve in time domain of ship structural borne noise prediction model

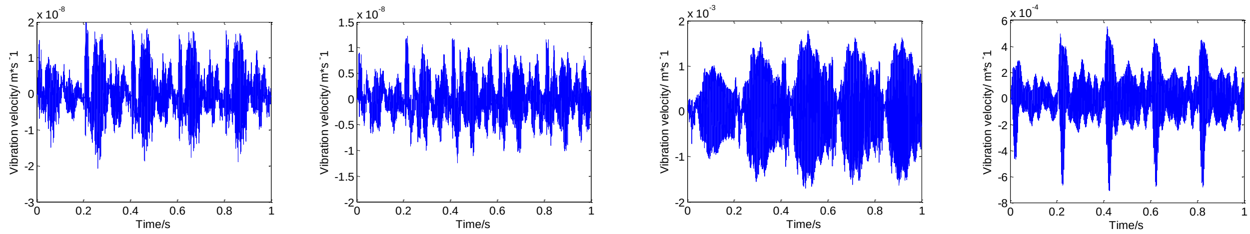
To take into account that upper limit frequency is 400Hz in this analysis, time exciting loads time spacing is ΔT=10-4s, and the time of ship cabin acoustic radiation underwater reaching steady state, so, the acting time of time domain exciting loads is taken Tt=1.5s, the curve of time domain exciting loads is shown in fig.3.

For ensuring the effectiveness of the results, the sampling time spacing of vibration and sound pressure observation points is ΔT=10-4s.

3.2. The analysis of results

3.2.1. The analysis of method effectiveness

The paper just need to valid the effectiveness of ship cabin acoustic time domain prediction method. The paper forecast the noise of the ship cabin by using the time domain prediction and frequency domain prediction method. Contrast the results from time domain prediction with frequency domain prediction method, the effectiveness of acoustic time domain prediction method is validated. The vibration and acoustic radiation time curve of typical observation points on ship cabin acoustic prediction model is shown in fig.4. The vibration and acoustic radiation contrast curve from time domain analysis and frequency domain analysis are examined in fig.5.



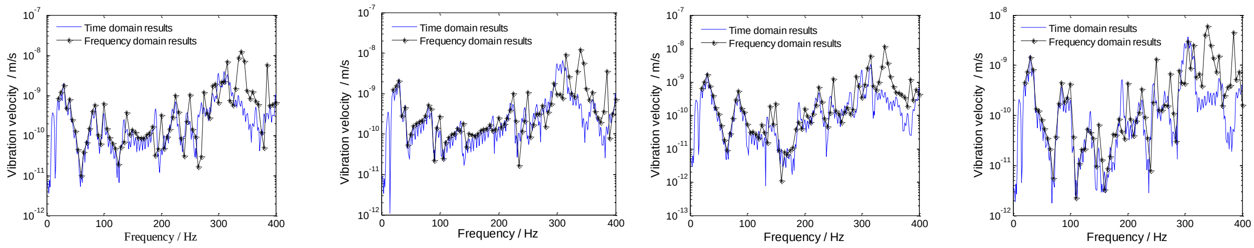
(a) Curve of vibration velocity of ship cabin at point 1#

(b) Curve of vibration velocity of ship cabin at point 3#

(c) Curve of sound pressure of fluid domain at point 1#

(d) Curve of sound pressure of fluid domain at point 3#

Fig.4 Vibration and sound radiation curve at typical observation points of ship cabin in time domain

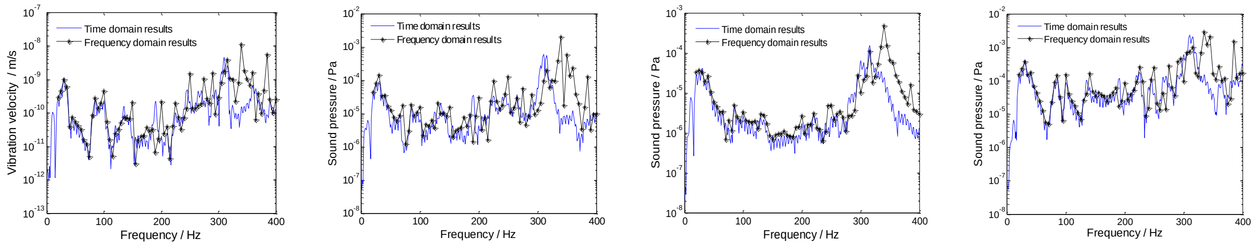


(a) Curve of vibration velocity of cabin at point 1#

(b) Curve of vibration velocity of cabin at point 2#

(c) Curve of vibration velocity of cabin at point 3#

(d) Curve of vibration velocity of cabin at point 4#



(e) Curve of vibration velocity of cabin at point 5#

(f) SPL comparison of fluid domain at point 1#

(g) SPL comparison of fluid domain at point 2#

(h) SPL comparison of fluid domain at point 3#

Fig.5 Vibration and sound radiation curve at typical observation points of ship cabin in time domain

It can be seen from the Fig.4 that when the calculation time is t $\geq$ 0.4s, the vibration and acoustic radiation of ship cabin model has reach the steady state. So, the vibration responding of model at steady state can be obtain by using the acoustic time domain prediction method.

In Fig.5, the results from acoustic time domain analysis agree well with the the results from acoustic frequency domain analysis in the low frequency. But the difference of results from the two methods become great with the increasing of frequency, especially in the frequency 300Hz~350Hz. Further more, comparison of the results curve from the acoustic time and frequency domain analysis at the same observation point, it can be found that the frequency composition from the acoustic time analysis are richer than from the acoustic frequency analysis. It is easy to stir structural vibration modal by the acoustic time analysis, which is closed to the real physical experiment. In one word, the acoustic time analysis is available for acoustic radiation prediction of ship, which results from acoustic time analysis is more closed to the test results.

5. CONCLUSION

The paper mainly research on the numerical analysis method about acoustic prediction that is output of truncated prediction model of ship structural borne noise. the acoustic time-frequency complex prediction method based on wave theory is advanced. After the effectiveness of this method is validated, the following conclusions are obtained.

(1) The results from time domain analysis perfectly match with results from frequency domain analysis at the low frequency. Comparison of the results curve from the acoustic time and frequency domain analysis at the same observation point, it can be found that the frequency composition from the acoustic time analysis are richer than from the acoustic frequency analysis, which is closed to the real physical experiment

(2) In a word, to acoustic prediction of ship model, the time domain prediction method not only take possession of perfect efficiency, but also consider the coupling of vibration in the low-frequency with wave-motion in the high-frequency. It truly reflects acoustic radiation underwater of ship and avoids the phenomenon of leakage in ship structure acoustic forecast peak.

(3) Firstly, in order to improve the solution efficiency and precision, frequency acoustic of ship and ocean structure is obtained by using time domain analysis method in single analysis, After the completion of prediction analysis of the ship structure frequency noise, some noise in concerned frequency points is predicted through the forecast method of frequency domain. This method can improve forecasting precision and efficiency, and rapidly discover the bright spot problems of ship structure acoustic radiation. Further more, the method can investigate the main transmission component and the main transmission way in the concerned frequency points, with the purpose to provide the method about acoustic prediction and evaluation, and noise control for ship and ocean structure.

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