The acoustic time-frequency complex prediction method of ship and ocean structure

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ABSTRACT

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 combined time domain frequency spectrum analysis method with frequency domain analysis method. 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, such as the bright spot of ship structure acoustic problems. Then, in the numerical validation, research shows that the acoustic time-frequency complex prediction method not only can solve the above difficulties existing in acoustic forecast of ship and ocean structure, but also analyze noise in concerned frequency points in detail, to determine the main transmission component and the main transmission way in the concerned frequency points. At the same time, this method can improve forecasting precision and efficiency, avoiding 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.

Keywords: Ship and ocean, structural acoustics, acoustic radiation underwater, time-frequency complex prediction method  I-INCE Classification of Subjects Number(s): 76.9

1. INTRODUCTION

With the advancement of sonar technology and the stricter requirements of shipping channel underwater noise environments from people, the radiated noise of ship and ocean structure have been research hot spot around the world. Mechanical noise of ship, especially mechanical noise in mid-low frequency of ship which is main noise sources at ship quietly sailing, directly determines the noise level of ship. Mechanical noise of ship is important assessment indicator in acoustic designing of ship structure. Therefore, the works of researching on the acoustic prediction methods for ship and improving the prediction accuracy for mechanical noise of ship have important significant for ensuring the ship quietness underwater. Combined of VB language, Li Tianyun (1) developed a submarine sonar self-noise prediction platform based on SEA theory. The validity and reliability of prediction platform are verified after the contrast of results from calculating and test. Based on SEA method, mechanical noise model of AUV with power equipment is constructed by WANG Guo-zhi (2). Then, WANG defined the analysis parameters of SEA model for AUV, and presented the underwater vibration and radiated noise characteristics of AUV in wide frequency. B Laulaget (3, 4) studied the effects of acoustic cover layer on vibration and radiated noise characteristics of stiffened cylindrical shell based on SEA method. Sheng Meiping (5) discussed the characteristics of vibration and radiated noise for AUV, and comparing the results from prediction with the results from tests, on the basis of SEA theory. Furthermore, many exploring works that SEA is applied to acoustic prediction of ship are supply by researchers (6-8).

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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:

(1) Numerical acoustic prediction methods ignore many nonlinear influence, due to basing on linear superposition principle.
(2) Coupling influence of vibration in the low-frequency with wave-motion in the high-frequency is not taken into account.
(3) 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 \( \Delta t \), 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

\[
\{ F^1 \} + \{ F^D \} + \{ F^S \} = \{ P \} \tag{1}
\]

Where \( \{ P \} \) is the vector of exciting force. \( \{ F^1 \} \) is the vector of inertial force. \( \{ F^D \} \) is the vector of damping force. \( \{ F^S \} \) is the vector of structural force resisting deformation.

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

\[
\{ F^1_i \} + \{ F^D_i \} + \{ F^S_i \} = \{ P_i \} \tag{2}
\]

\[
\{ F^1_i + \Delta F^1_i \} + \{ F^D_i + \Delta F^D_i \} + \{ F^S_i + \Delta F^S_i \} = \{ P_{i+\Delta t} \} \tag{3}
\]

\[
\Delta F^1_i = F^1_{i+\Delta t} - F^1_i = [M_T] [\Delta \ddot{x}] \tag{4}
\]

\[
\Delta F^D_i = F^D_{i+\Delta t} - F^D_i = [C_T] [\Delta \dot{x}] \tag{5}
\]

\[
\Delta F^S_i = F^S_{i+\Delta t} - F^S_i = [K_T] [\Delta x] \tag{6}
\]

\[
\Delta P_i = P_{i+\Delta t} - P_i \tag{7}
\]

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

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

Where, \([M_T]\), \([C_T]\), \([K_T]\) are respectively mass matrix, damping matrix and stiffness matrix of increment equation.

\[
k_g(t) = \left( \frac{\partial F^o_i(x)}{\partial x} \right) ; \quad m_g(t) = \left( \frac{\partial F^i_i(x)}{\partial x} \right) ; \quad c_g(t) = \left( \frac{\partial F^D_i(x)}{\partial x} \right) \tag{9}
\]

The coefficient can be described in terms of matrix as

\[
[M_T] = \left[ \frac{\partial \{ F^1 \} }{\partial \{ x \} } \right] ; \quad [K_T] = \left[ \frac{\partial \{ F^S \} }{\partial \{ x \} } \right] ; \quad [C_T] = \left[ \frac{\partial \{ F^D \} }{\partial \{ x \} } \right] \tag{10}
\]

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 \( \{x\} \) and \( \{x\} \), the motion equation can be simplified as
\[
[M][\ddot{x}]+\{F(x,\dot{x})\}=\{P\}
\] (11)

Increment equation is
\[
[M]([\ddot{x}]_n+1)+[C][\Delta \dot{x}]+[K_r][\Delta x] = \{\Delta P\}
\] (12)

Tangent damping matrix \([C_r]\) and tangent stiffness matrix \([K_r]\) are defined to be
\[
[C_r]=\left(\frac{\partial \{F(x,\dot{x})\}}{\partial \{\dot{x}\}}\right); \quad [K_r]=\left(\frac{\partial \{F(x,\dot{x})\}}{\partial \{x\}}\right)
\] (13)

Generally, when nonlinear force \( \{F\} \) merely depends on displacement \( \{x\} \), differential equation describes as
\[
[M][\ddot{x}]+[C][\ddot{x}]+\{F(x)\}=\{P\}
\] (14)

Increment equation is
\[
[M][\ddot{x}]+[C][\ddot{x}]+\{F(x)\}=\{P\}
\] (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
\[
[M]\{\ddot{x}_{n+1}\}+[C]\{\Delta \dot{x}\}+[K_r]\{\Delta x\} = \{\Delta P\}
\] (16)

The relationship of acceleration, velocity and displacement at \( x_n \) and \( x_{n+1} \) 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
\[
x(t+\Delta t) = x(t) + \dot{x}(t)\Delta t + \frac{1}{2}\ddot{x}(t)\Delta t^2 + \frac{1}{6}\dddot{x}(t)\Delta t^3 + \cdots
\] (17)

Assuming
\[
x(t+\Delta t) = x_{n+1}; \quad x(t) = x_n; \quad x(t-\Delta t) = x_{n-1}
\] (18)

The former differential equation is obtained by Eq.(18)
\[
x_{n+1} = x_n + \dot{x}_n \Delta t + \frac{1}{2}\ddot{x}_n \Delta t^2 + \frac{1}{6}\dddot{x}_n \Delta t^3 + \cdots
\] (19)

The after differential equation is written as
\[
x_{n-1} = x_n - \dot{x}_n \Delta t + \frac{1}{2}\ddot{x}_n \Delta t^2 - \frac{1}{6}\dddot{x}_n \Delta t^3 + \cdots
\] (20)

The below equations can be obtained through adding or misusing Eq.(19) and Eq.(20).
\[
\dot{x}_n \Delta t = \frac{1}{2}(x_{n+1}-x_{n-1}) + O(\Delta t^3)
\] (21)
\[
\ddot{x}_n \Delta t^2 = (x_{n+1}-2x_n+x_{n-1}) + O(\Delta t^4)
\] (22)

Velocity and acceleration at the time \( t \) can be approximately described by the displacements of three points \((n-1, n, n+1)\)
\[
\dot{x}_n = \frac{1}{2\Delta t}(x_{n+1}-x_{n-1})
\] (23)
\[
\dddot{x}_n = \frac{1}{\Delta t^2}(x_{n+1}-2x_n+x_{n-1})
\] (24)

Displacement, velocity and acceleration at the time \( t \) satisfy the differential equation follow as
\[
m\{\dddot{x}_n\} + c\{\dot{x}_n\} + k\{x_n\} = f_n
\] (25)

Substitution Eqs.(23-24) into Eq.(25), the equation is obtained
\[
m\dot{x}_n + c\dot{x}_n + kx_n = \hat{f}_n
\] (26)
Where
\[
\dot{m} = \left( \frac{m}{\Delta t^2} + \frac{c}{2\Delta t} \right) \hat{f}_m - \hat{f}_m \left( k - \frac{2}{\Delta t^2}m \right) x_n - \left( \frac{1}{\Delta t^2}m - \frac{1}{2\Delta t}c \right) x_{n-1}.
\] (27)

For a single degree of freedom system, the displacement \( x_{n+1} \) at the time \( t^*+\Delta t \) 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 (9-11). The basic principle is consistent with the above, which will not be repeated here.

2.2 The principle of acoustic time-frequency complex prediction method

Known from chapter 2.1, due to the response of structure at next Increment time \( \Delta t \) is solved by iterative method in time domain analysis. Then, the tangent slope is using to correct mass matrix, damping matrix and stiffness matrix. Therefore, relative to frequency domain analysis, time domain analysis possess following advantages:

(a). The effects of material nonlinear, structural nonlinear and the coupling of vibration in the low-frequency with wave-motion in the high-frequency are taken into account.

(b). Because exciting force \( \{ P \} \) is the transient time domain signal, the time domain signal can consist of arbitrarily frequency. So, the response of structure in multiple frequency are obtained just at a single analysis. Solution efficiency is improved.

(c). Just like the real ship test data, the results from time domain analysis are time domain signal. To facilitate the contrast and validation with the results from the real ship test.

However, some disadvantages exist in the time domain analysis method.

(a). The rate of convergence in steady state response analysis is related to prediction frequency and damping. The higher prediction frequency and the greater damping, the greater the attenuation rate, the shorter time needed for steady-state vibration, the shorter time needed for steady state response analysis. And vice, the lower prediction frequency and the smaller damping, the longer time needed for steady-state vibration, the longer time needed for steady state response analysis, the slower rate of convergence.

(b). The analysis precision is restricted by element type and analysis time. Because structural steady state response in time domain is closely related to the analysis of time, in order to avoid "locking" phenomenon taking place in explicit analysis, low precised element type are usually applied to solve the structural steady state response, which influences the precision of explicit analysis.

To take the characteristic of time domain analysis and frequency domain analysis into consideration, if the advantages of time domain analysis and frequency domain analysis are combined, and the disadvantages of time domain analysis and frequency domain analysis are avoided, then, the disadvantages about the low calculation efficiency, the irrespective of nonlinear influence and coupling of vibration in the low-frequency with wave-motion in the high-frequency are overcome, the precision and efficiency of acoustic prediction of ship and ocean structure are greatly improved. On the basis, the paper advances the acoustic time-frequency complex prediction method of ship and ocean structure, which consist of time domain analysis and frequency domain analysis. Using the time domain analysis, frequency domain acoustic of ship and ocean structure are obtained in a signal analysis. After completing the frequency domain acoustic prediction of ship, the some concerned frequency points acoustic are need to predicted. Then, frequency domain analysis is applied to predict the some concerned frequency points acoustic for determining 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.

2.3 General steps of the acoustic time-frequency complex prediction method of ship

The acoustic time-frequency complex prediction method of ship include the time domain analysis and frequency domain analysis at the concerned frequency points. The steps of the method consist of determining equipment exciting loads, constructing prediction model, applying boundary condition, and post-processing the analysis results.

(1) Determining equipment exciting loads: if the time domain equipment exciting loads at the ship base are determined, the equipment exciting loads do not need to transfer in time domain analysis. If the frequency domain equipment exciting loads at the ship base are determined, the frequency domain equipment exciting loads need to transfer into time domain equipment exciting loads by FFT.

(2) Constructing prediction model: to the acoustic radiation prediction of ship underwater, because the ship is constructed by a large number of the frame structure of stiffened plates with periodicity, it will take a lot of manpower and material resources to analyze the full ship model, which will lead to that the prediction frequency domain and precision will greatly fall for the huge calculation scale. Even worse, calculation is breaking. Therefore, if the full ship model is replacing with partial ship cabin, and simplifying the acoustic
prediction model of ship, the prediction frequency domain and precision will greatly rise and the calculated cost will be reduced. 
(3) Applying boundary condition and controlling output: to consider that the acoustic steady state results rely on the calculation time and the sampling frequency of calculation results, we must control the calculation time and the sampling frequency except applying boundary conditions to the acoustic prediction model of ship. To the frequency acoustic prediction in concerned frequency points, the calculation results do not rely on the calculation time and the sampling frequency. Therefore, according to the calculation request of frequency domain analysis method, prediction frequency are determined, the corresponding boundary conditions are applied and output variable are available controlled.

3. THE VALIDATION OF METHOD

3.1 Introduction of Validation Model

Ship cabin model are choose to be validation model, which is shown in Figure.1. The model parameter is that, ship structure is semi-cylindrical shell, its radius is \( R = 1500 \text{mm} \), the shell thickness is \( t = 10 \text{mm} \), the bulkheads locate at the ends of ship cabin, ship cabin includes some ribs. Rib spacing is \( L = 600 \text{mm} \), the rib parameter is \( t = 10 \text{mm}, h = 200 \text{mm} \). The equipment parameter is \( 1200 \text{mm} \times 600 \text{mm} \times 300 \text{mm} \). The shell thickness of equipment is \( t = 20 \text{mm} \). Two vibration isolator support the ends of equipment along the center line of ship cabin model. The stiffness of vibration isolator is \( k = 4 \text{kN/m} \). The boundary condition at the ends of the ship cabin model is simply-simply. Equipment exciting load is \( F = 1 \times \sin(2\pi f) \text{ N} \), which vertically acts on the the center point of equipment. The radius of fluid field is \( R = 6 \text{m} \). 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 \( \Delta f = 5 \text{Hz} \). Damping is \( \eta = 0.05 \). 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.

![Figure 1 – Structure of radiated noise model of a ship cabin](image1.png)

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 Figure.2(b). 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 \( R = 6 \text{m} \). The locations of fluid field sound pressure observation points is shown in Figure.2(b).

![Sketch boundary conditions of calculating model](image2.png)
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.(28).

\[ F(t) = \sum_{i=0}^{N} \sin(2\pi(20 + 5)t) \]  \hspace{1cm} (28)

To take into account that upper limit frequency is 400Hz in this analysis, time exciting loads time spacing is \( \Delta T = 10^{-4} \)s, and the time of ship cabin acoustic radiation underwater reaching steady state, so, the acting time of time domain exciting loads is taken \( T_r = 1.5 \)s, the curve of time domain exciting loads is shown in Figure.3.

For ensuring the effectiveness of the results, the sampling time spacing of vibration and sound pressure observation points is \( \Delta T = 10^{-4} \)s.

### 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 Figure.4. The Figure.5 describe the acoustic radiation distribution of ship cabin model at typical time. The vibration and acoustic radiation contrast curve from time domain analysis and frequency domain analysis are examined in Figure.6.
It can be seen from the Figure 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.

The Figure 5 clearly show the steady state acoustic radiation distribution underwater of the ship cabin model. On the one hand, the acoustic radiation distribution underwater of ship cabin model is different at the different time. The real acoustic radiation underwater of ship in the exciting loads can be obtain through the acoustic radiation distribution of ship cabin model at typical time. So, the acoustic radiation underwater of ship can be actually simulated by the acoustic time domain prediction method. On the other hand, the results of acoustic radiation underwater from acoustic time domain prediction have great difference with the results from acoustic frequency domain prediction. The results of acoustic radiation underwater from acoustic time domain prediction are added by the results at every frequency from acoustic frequency domain prediction. Therefore, the main transmission component and the main transmission way in the concerned frequency points can not be obtained by the acoustic radiation distribution of ship cabin at one time, which is the difference between acoustic time domain prediction and acoustic frequency domain prediction.

In Figure 6, the results from acoustic time domain analysis agree well with 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 close 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.

3.2.2 The analysis of solution efficiency

The solution efficiency of acoustic time domain prediction method is analyzed. Using the same computing hardware configuration, the needed resources of acoustic time and frequency domain prediction method is shown in Table 1.

<table>
<thead>
<tr>
<th>Program</th>
<th>Internal Storage(MB)</th>
<th>CPU solving time(s)</th>
<th>Total solving time(s)</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td>Time domain method</td>
<td>287</td>
<td>15912</td>
<td>16064</td>
<td>Solving time is relative to the minimum mesh size, total number of free degree and calculating time</td>
</tr>
<tr>
<td>Frequency domain method</td>
<td>2010</td>
<td>19497</td>
<td>21140</td>
<td>Solving time is relative to total number of free degree and computer internal storage</td>
</tr>
</tbody>
</table>

*NOTE: Computing hardware configuration is 2G Internal Storage, intel i3 3.3GHz CPU processor. Calculation adopts double CPU for parallel computing.

From the Table 1, no matter the internal storage needing or CPU solving time, the acoustic time domain prediction method is better than the frequency domain prediction method. Especially for real ship structure, due to its calculation scale is greater than the calculation scale in the paper, it will have great difficulty for the frequency domain prediction method to analyzing the model by using general computer. However, it is easy and quick for the time domain prediction method to analyzing the model.

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. Therefore, the acoustic time-frequency complex prediction method of ship advanced in the paper is available. 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.

4. METHOD APPLICATION

4.1 Calculation for the acoustic radiation underwater

The vibration acceleration of gearbox basement of a ship versus frequency have been measured. To facilitate discussion, the paper assumes that the vibration acceleration ray loads of gearbox basement are 1m/s². Loads frequency spacing is 20Hz~400Hz. Gearbox basement locates at the symmetry center line of ship, which has 1/4L distance to the ship-after. The dimension of ship are length 62m, broaden 7.5m, draft 2.5m, displacement 675Ton. The form of a diagram, general arrangement, the typical structure, equipment structure and installation drawings are also detailed.

![Figure 7 – Vibration acceleration of gearbox basement of a ship versus frequency](image.jpg)
After determining the acoustic truncated prediction model of ship structural, the geometric model of ship are constructed. FEM model of truncated prediction of ship structural borne noise are obtained after grid discretion. The fluid field include inner and outer fluid field. Inner fluid field consist of tetrahedron element, due to irregularity in the shape of ship structure. Outer fluid field consist of hexahedral element. The size of structural mesh is 0.2m, and the size of fluid field mesh is 0.35–0.5m. The structural model have 68291 mesh in total, and fluid field model have 405259 mesh in total. FEM model of truncated prediction of ship structural borne noise are present in Figure.8-9.

Sound pressure observation points are setting at the interface of hull center and inner fluid field. The interface distance R=10m to the mid-ship. A series of sound pressure measuring points lay on the interface of hull center and inner fluid field along the direction of ship. Sound pressure observation and measuring points are present in Figure.10.

4.2 Analysis for the acoustic radiation underwater

Figure.11-12 present the sound pressure curves of observation points along the direction of ship. Abscissa x/L is dimensionless ship length. x is the coordinates along the direction of ship, where ship bow is the origin and the direction of ship after is positive. L is ship length. Ordinate is dimensionless sound pressure level $L_p$ that is defined as:

$$L_p = 20 \log \left( \frac{p}{p_0} \right) / a_0$$ (29)

Where $|p|$ is sound pressure amplitude or absolute value of real sound pressure. $p_0$ is reference sound pressure. $a_0$ is constant.
From Figure 11-12, we can obtain the following contents. The acoustic radiation underwater of ship mainly concentrate on the middle and tail area (3/4L) where gearbox basement locates. No matter how the exciting frequency changes, the sound pressure level of cabin where gearbox basement locates is obviously greater than the other areas. The sound pressure level of tail ship area is also relatively high. However, sound pressure level of stem area is low.

With the increasing of exciting frequency, the sound pressure of ship gradually decreases, and acoustic radiation level falls slowly. Because inputting exciting loads is vibration acceleration of gearbox basement. Vibration velocity will decrease with the exciting loads frequency increasing. So, the sound pressure in low frequency is lager, and the sound pressure in high frequency is relatively small.

At low exciting frequency, the acoustic radiation distribution is uniform along the direction of ship. But the nonuniformity of acoustic radiation distribution rises with the exciting frequency increasing. Further more, at the different exciting frequency, the bright spot position of the acoustic radiation underwater is changeable. Because acoustic wavelength vary with the frequency. At low exciting frequency, flexural wavelength in structure and acoustic wavelength in water is relatively lager. Then, the acoustic radiation distribution is uniform. However, with the increasing of exciting frequency, acoustic wavelength decreases, which results into bad uniformity of acoustic radiation distribution.

5. CONCLUSIONS

The paper mainly research on the numerical analysis method about acoustic prediction that is output of truncated prediction model of ship structural borne noise. 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. 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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