Underwater Noise Radiation Optimization of a Scientific Investigation Ship

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Based on Finite Element Method and Statistical Energy analysis, this paper examines and optimizes mechanical noise of a scientific investigation ship under typical working conditions, and a noise reduction plan is presented. Considering that hydrodynamic noise is relatively slight within maximum cruising speed, the underwater radiated noise of a scientific investigation ship under certain working conditions is discussed. Study shows that the main engine is the main noise source of mechanical noise. To meet the demands design criterion of underwater noise radiation, vibration isolation and noise reduction measures is also studied and an optimized design plan is obtained.

Key words: Finite Element Method; Statistical Energy analysis; scientific investigation ship; underwater noise radiation; optimization.

1 Introduction

Marine scientific investigation ships are used for marine scientific research, application research and surveying or exploration. Scientists take a scientific investigation ship to target ocean district, obtain accurate data and sample data to unlock the secrets of oceans and serve the survival and development of humans. Acoustic signals are the only physical information can propagate long distances in the water so that the underwater radiated noise of the ship itself must be controled, in addition to marine survey ship, the protection of marine ecosystems and biodiversity also should be balanced, so scientific investigation ships need low noise design.

Currently, on the research of underwater noise radiation characteristic, scholars have made considerable achievements with the following methods in the field of mono hull craft, including statistical energy analysis (YAO, WANG, SUN, PANG. 2011; PANG. 2012), finite element method (ZOU, CHEN, HUA. 2003; ZOU, CHEN, HUA. 2004.), boundary element method (WANG, ZHOU, JI, XIE, MO. 2010; WANG, ZHOU, JI. 2012; ZHOU.1996.), acoustic-structure coupling method (JIN, ZHAN, MIAO, JIA, WAN. 2011; MIAO, QIAN, YAO, HUANG. 2009; MIAO, WANG, JIA, JIN, PANG. 2012; PANG, YAO, MIA, JIA. 2012). In the field of scientific investigation ships, Yang Yong[13] applied double-layer vibration isolation system to the ship diesel generator and single-layer vibration isolation system to the ship parking units and achieved its cabin noise levels. Kong Xiancai[14-15] made a research on engine room ventilation system noise and power systems of a scientific investgation ship and made a number of useful recommendations, but for scientific research ship underwater radiation noise remains to be further studied. Describing fluid as acoustic medium, simulating the propagation of sound wave in fluid field with acoustic equation, then with the help of boundary impedance method or acoustic infinite element method we can better simulate sound wave propagation characteristic in infinite waters. Based on acoustic-structure coupling method, the underwater radiation noise of a scientific investgation ship is researched, which provides a recommendation to the acoustic design of scientific investgation ships.

2 Theoretical basis

2.1 Acoustic equation

For compressible, adiabatic fluid, considering the loss of flow momentum, the equation of small amplitude motion is shown as below:

$$\frac{\partial p}{\partial x} + \gamma\left(x,\theta_i\right)\dot{\imath} \qquad \ddot{} \tag{1}$$

where p = fluid overpressure; x = spatial coordinate; $\dot{\imath}$ = fluid particle velocity; $\ddot{}$ = is fluid particle acceleration; ρ_f = fluid density; γ = volume drag force (ratio of force and speed on per volume element);θ_i
= field variable which is independent with fluid particle position but may be associated with ρ_f and γ .

For inviscid, linear and compressible fluid, the dynamic pressure of acoustic medium is closely related to its volume modulus and volume strain:

$$p = -K_f\left(x,\theta_i\right)\frac{\partial}{\partial x}\cdot u^f \tag{2}$$

where K_f = volume modulus of fluid:

$$p = -K_f \varepsilon_V \tag{3}$$

where ε_v = volume strain

$$\varepsilon_V = \varepsilon_{11} + \varepsilon_{22} + \varepsilon_{33} \tag{4}$$

2.2 Simulation of infinite fluid field

Usually, we simulate infinite fluid field in two ways, namely: boundary impedance method and acoustic infinite element method. The fundamental of the boundary impedance method is a non-reflecting boundary condition (NRBC), which is realized by preventing acoustic energy reflecting on the interface. However, a large enough fluid field is usually required to ensure calculation accuracy in this method. Acoustic infinite element method covers unlimited units on the boundary to realize the simulation of infinite fluid field. These elements can be directly applied on the boundaries of structural or acoustic finite element fluid field, so that we can reduce the fluid field model according to different requirements, thereby reducing the cost of modeling. Both methods are capable to achieve nice results with little difference (Cipolla. 2002; Grote & Keller.1995; Lu, D'Souza & Chin. 2005.).

Through the two methods above, the fluid field consists of acoustic medium meets the Sommerfield condition on the infinite boundary of fluid field:

$$\lim_{r\to\infty} r\left(\frac{\partial p}{\partial r}+jkp\right)=0 \tag{5}$$

2.3 Fluid-structure coupling vibration equation

As structural motion on the fluid-structure interface produces fluid load and sound pressure generates an additional force on the structure at the same time, we must calculate structural dynamic equation and wave equation of fluid field simultaneously. With the help of discrete model, both wave equation and motion equation can be solved to obtain displacement and sound pressure value on the fluid-solid interface.
Assuming that fluid is an ideal acoustic medium, acoustic wave equation is:

$$\nabla^2 p=\frac{1}{c^2}\frac{\partial^2 p}{\partial t^2} \tag{6}$$

where c = sound velocity in the fluid medium; p = instantaneous sound pressure.

Applying Galerkin method, multiplying the sound pressure variation δp and then integrating (6) in the fluid field volume V, we can be able to achieve the equation below:

$$\iiint_V \frac{1}{c^2}\delta p\frac{\partial^2 p}{\partial t^2}dV+\iiint_V\left(L^T\delta p\right)\left(Lp\right)dV=\iint_S \rho_f \delta p n^T\left(\frac{\partial^2 U}{\partial t^2}\right)dS \tag{7}$$

where u = displacement vector on S surface;

$$L^T=\nabla\cdot(\ )=\begin{bmatrix}\frac{\partial}{\partial x} & \frac{\partial}{\partial y} & \frac{\partial}{\partial z}\end{bmatrix} \tag{8}$$

$$L=\nabla(\ ) \tag{9}$$

After discretizing the fluid equation into finite elements and other operations we can fully achieve fluid-structure coupling vibration equation:

$$\begin{bmatrix}M_S & 0\\ \rho_f R & M_f\end{bmatrix}\begin{Bmatrix}\ddot{U}\\ \ddot{P}\end{Bmatrix} [illegible] \begin{bmatrix}\cdots & R^T\\ \cdots & K_f\end{bmatrix}\begin{Bmatrix}U\\ P\end{Bmatrix}=\begin{Bmatrix}F_S\\ 0\end{Bmatrix} \tag{10}$$

where M_s = structural mass matrix; C_s = structural damping matrix; K_s = structural stiffness matrix; M_f = fluid mass matrix; C_f = fluid damping matrix; K_f = fluid stiffness matrix; R = fluid-structure coupling matrix; U = node displacement vector; P = sound pressure vector; F_s = structural load vector.

3 科考船水下辐射噪声预报模型

3.1 科考船简介

The scientific investgation ship model used in this paper has geometric dimensions as below: length L = 103m; breadth B = 9m; draught T = 5.7m. Structure material properties: elastic modulus = $2.1*10^{11}\ pa$, Poisson's ratio λ = 0.3, density ρ = 7800kg/m^3.

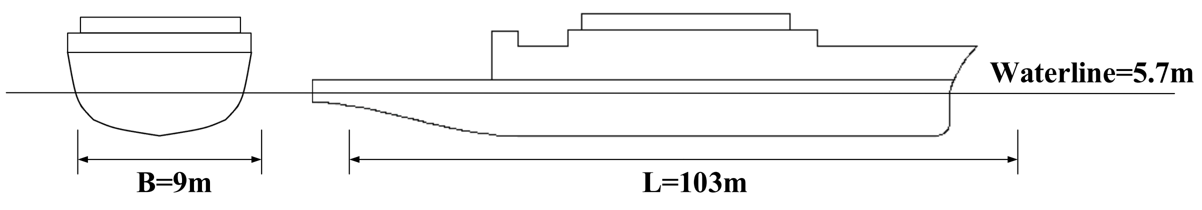


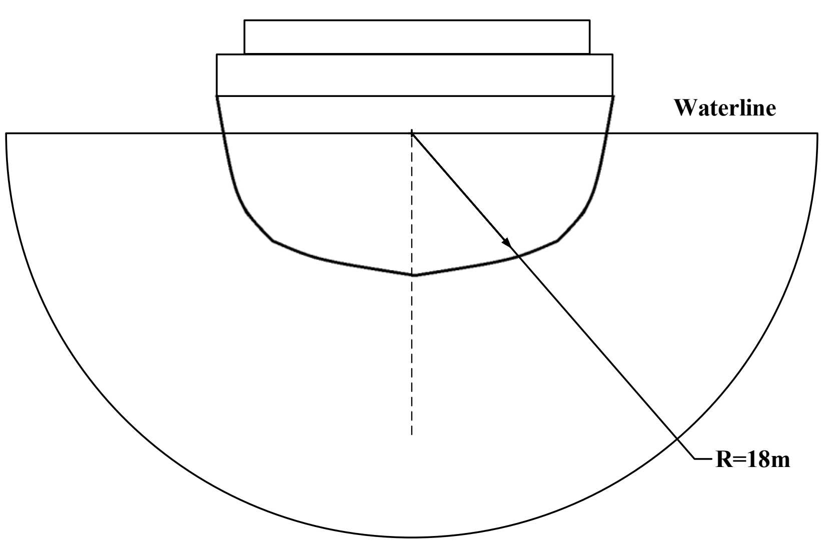
Figure1. The sketch of scientific investigation ship

In order to fully reflect the coupling between the wet surface of the outer hull plates and fluid field, usually we need to determine the minimum radius of the fluid field (PANG. 2012.):

$$R_f \geq \max(D/2+0.2\lambda, 2D) \qquad (11)$$

Where D = the maximum diameter of the structure; λ = the sound wavelength corresponding to the predicted frequency. Here we make 20Hz as the lower limit of frequency with a corresponding wavelength of 5.94m. The width of bodes is about 9m.Then the minimum radius of the fluid field is calculated as 18m according to (11), which is applied.

Fluid material properties: sound velocity c = 1460m/s, density = 1000 kg/m^3.



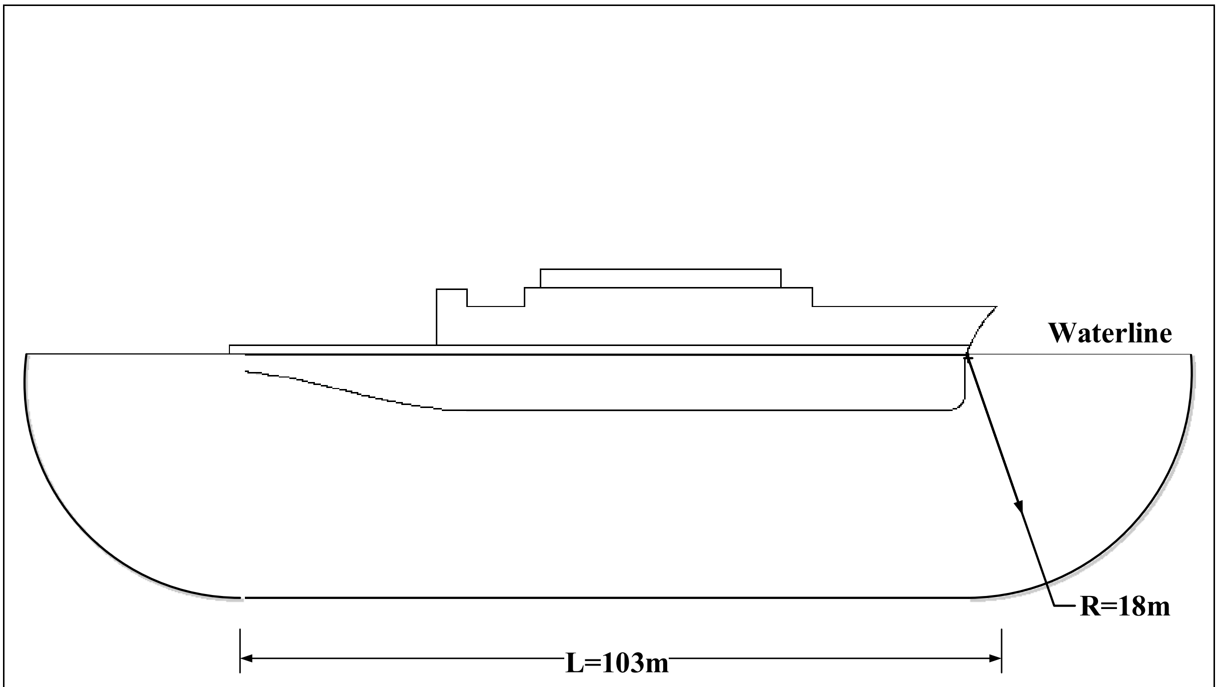


Figure.2 Fluid field of the scientific investigation ship

3.2 激励载荷与考核工况

The main vibration sources of the scientific investgation ship are main engine, auxiliary engine, propulsion motor, steering engine and sewage pump. Considering that different working conditions may occur during practical sail, 3 typical working conditions are installed in this paper, which are mormal condition, maximum condition and minimuncondition. The maximum condition and minimum condition are respectively the vibration velocity level or acceleration level of the normal condition ±5dB. The main engine, auxiliary engine and propulsion motor are given by vibration velocity level, of which the reference is 1e-9m/s; the steering engine and sewage pump are given by acceleration level, of which the reference is 1e-6m/s^2. The main vibrstion sources of the scientific investgation ship under the normal condition can seen from figure 3 as below:

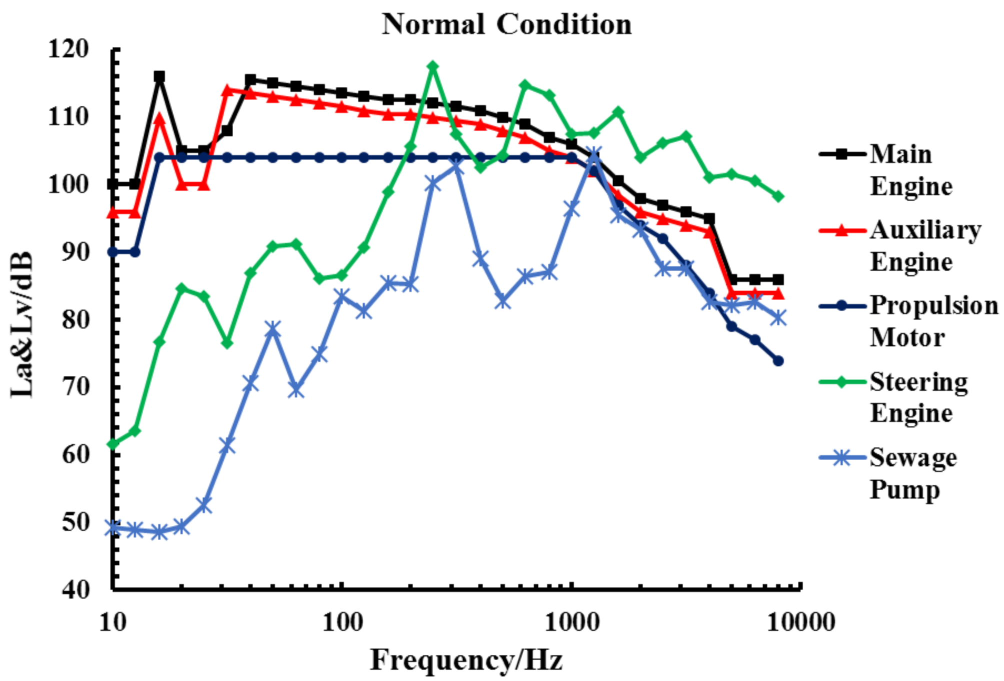


Figure.3 Main vibration sources under normal condition

Considering that the actual total damping loss factor of a ship is difficult to determine, 4 typical total damping loss factor are given as η=0.025%, η=0.1%, η=0.25% and η=1%。The sound pressure level that 1 m far away from the ship body is measured, which is known as the sound source level of a ship.

4 科考船水下辐射噪声评估

4.1 科考船水下辐射噪声标准

According to the International Council for the exploration of the Sea (ICES), expedition ship underwater radiation noise meets ICES209 specifications, namely:

（1）During 1Hz~1kHz, the sound pressure spectral density level does not exceed:

$$L_p \le 135 - 1.66 \log f_{\mathrm{Hz}} \quad (1\text{-}1)$$

（2）During 1Hz~100kHz, the sound pressure spectral density level does not exceed:

$$L_p \le 130 - 22 \log f_{\mathrm{kHz}} \quad (1\text{-}2)$$

The sound pressure level curve is shown in Figure 4:

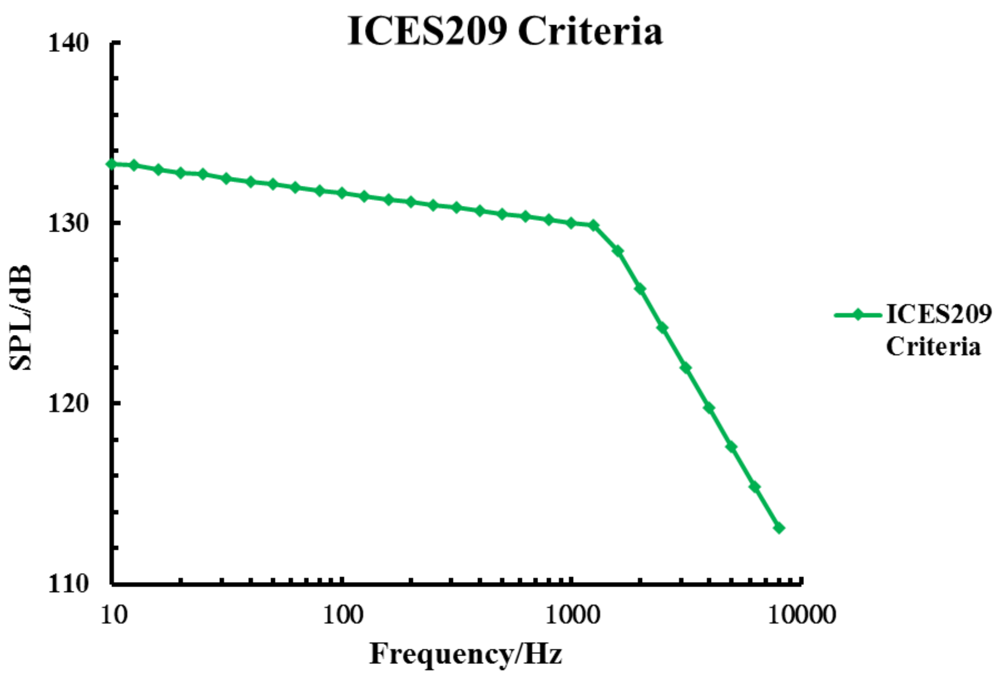


Figure.4 ICES209 Criteria

4.2 科考船水下辐射噪声预报

After calculation, the sound source level under different working conditions and damping loss factor are shown as figure 5~figure 8, in order to facilitate the analysis of the scientific investgation ship source level under the conditions, ICES209 standard curve is given.

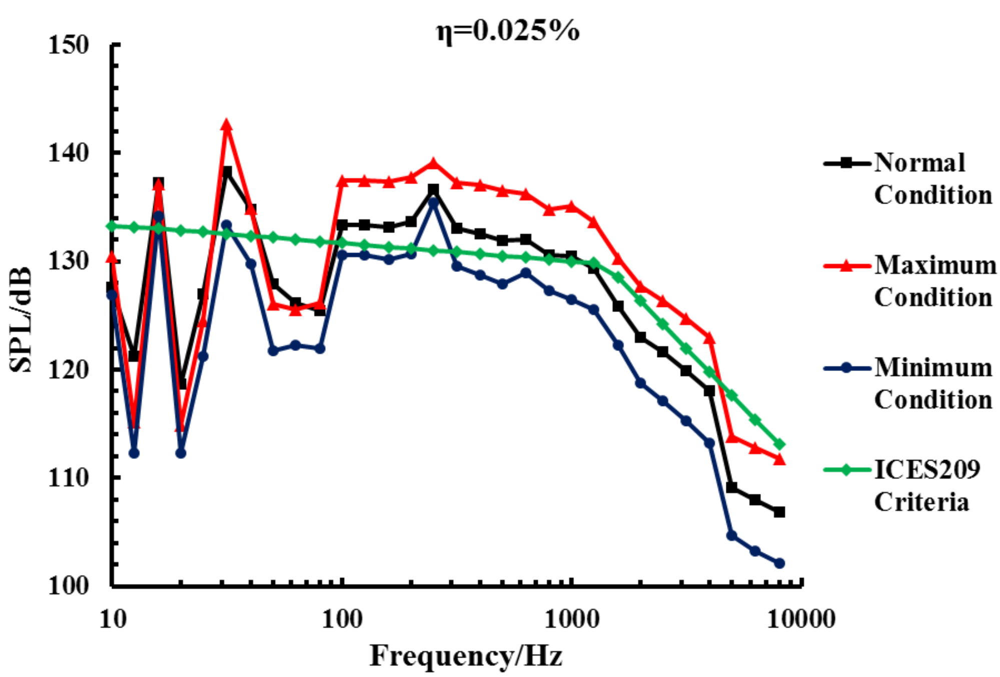


Figure.5 Sound source level under η=0.025%

As can be seen from Figure 5, when damping loss factor η =0.025%, in 16Hz and 31.5Hz under different working conditions scientific ship underwater radiation noise sound level is taken to the peak and are exceeded. After 100Hz in the high frequency band, underwater radiated noise levels decreased gradually, and with increasing frequency, noise level decays rapidly. Basicly minimum condition meet the ICES209 requirements for noise levels, maximum working condition apart from the 12.5Hz, 20Hz and a few other low frequency points, exceeding all of its noise levels, normal condition of 100Hz low frequency band and 1kHz high frequency fundamental meeting targets, but higher frequencies in the 100Hz~1kHz exceeded.

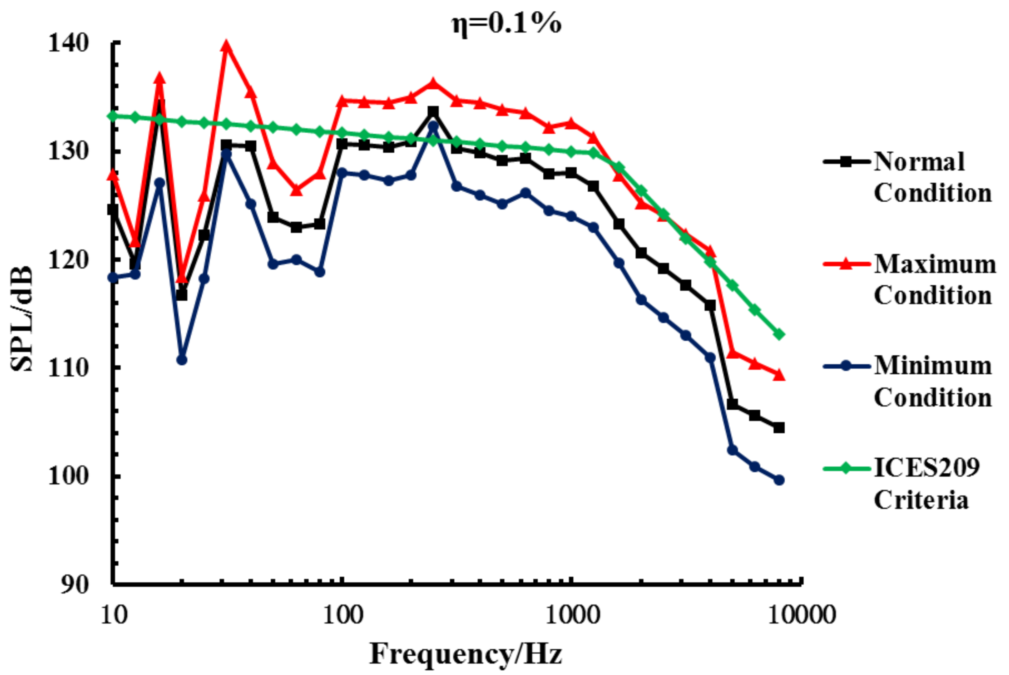


Figure.6 Sound source level underη=0.1%

As can be seen from Figure 6, when damping loss factor η =0.1%, in 16Hz and 31.5Hz under different working conditions scientific ship underwater radiation noise sound level is taken to the peak and are exceeded. After 100Hz in the high frequency band, underwater radiated noise levels decreased gradually, and with increasing frequency, noise level decays rapidly. Basicly minimum condition and normal condition meet the ICES209 requirements for noise levels, maximum working condition apart from the 12.5Hz, 20Hz and a few other low frequency points, exceeding all of its noise levels.

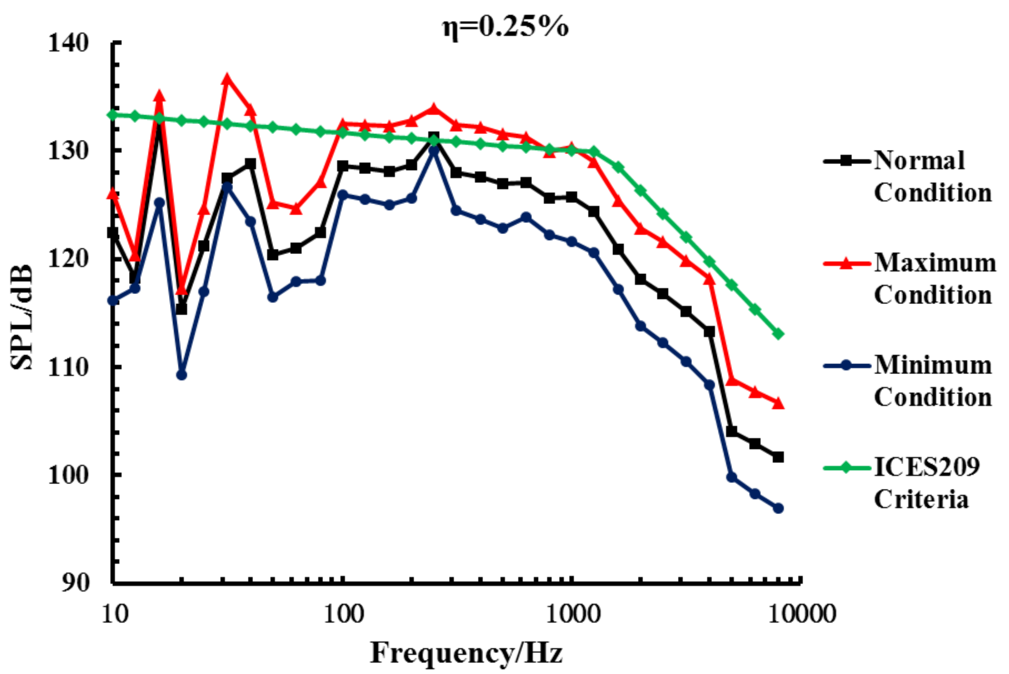


Figure.7 Sound source level underη=0. 25%

As can be seen from Figure 7, when damping loss factor η =0.25%, in 16Hz and 31.5Hz under different working conditions scientific ship underwater radiation noise sound level is taken to the peak and are exceeded. After 100Hz in the high frequency band, underwater radiated noise levels decreased gradually, and with increasing frequency, noise level decays rapidly. Basicly minimum condition and normal condition meet the ICES209 requirements for noise levels, maximum working condition apart from the 12.5Hz, 20Hz and a few other low frequency points, exceeding all of its noise levels.

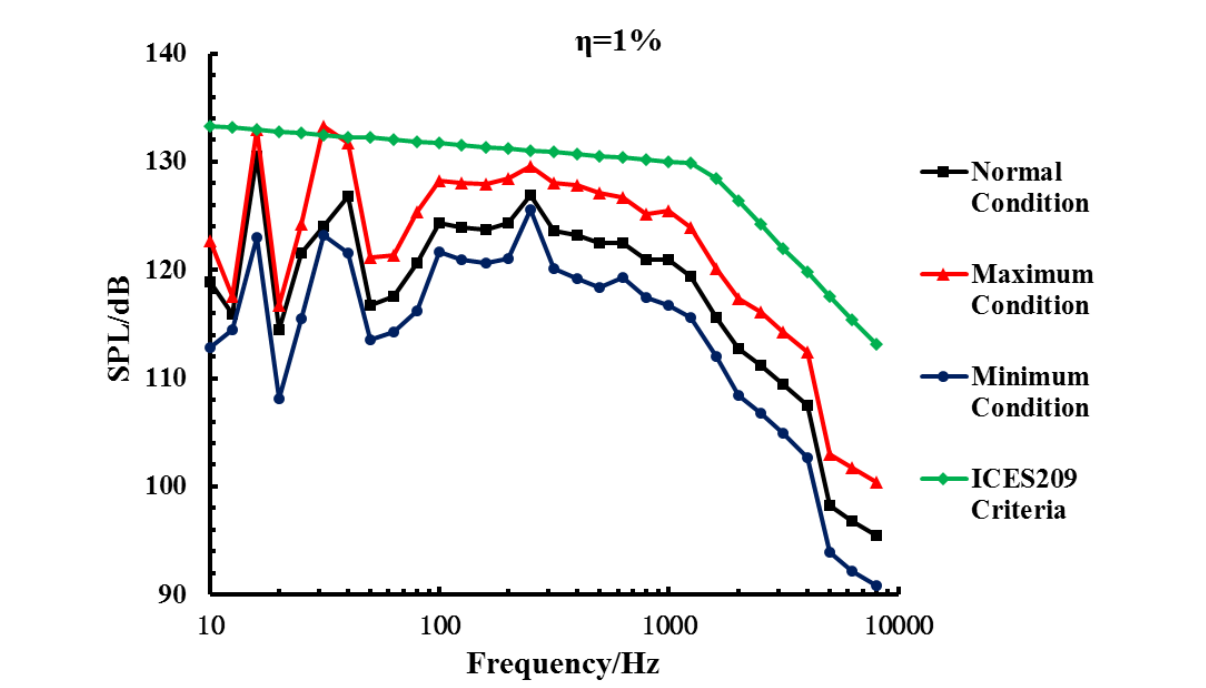


Figure.8 Sound source level underη=1%

As can be seen from Figure 8, when damping loss factor η =1%, in 16Hz and 31.5Hz under different working conditions scientific ship underwater radiation noise sound level is taken to the peak and are exceeded. After 100Hz in the high frequency band, underwater radiated noise levels decreased gradually, and with increasing frequency, noise level decays rapidly. Three conditions meet the ICES209 requirements for noise levels, in's obvious that bigger damping loss factor will decrease the underwater radiation noise significantly.

5 结论

Based on acoustic-structure coupling method, the underwater radiation noise of a scientific investgation ship is researched, some conclusions can be drawn as below:

(1) When damping loss factor is small (η =0.025%), the underwater radiated nlise of the scientific investgation ship will exceed ICES 209 requirements; while structure loss factor is gradually increased, the underwater radiated nlise will gradually decline.

（2）When damping loss factor η≧0.25%，Basicly minimum condition and normal condition meet the ICES209 requirements for noise levels; When damping loss factor η≧1%, maximum working condition will meet the ICES209 requirements.

（3）In three conditions, noises levels in 16Hz, 31.5Hz, 250hz are quite high. Analysis shows that the 3 mechanical noises is mainly caused by the main engine.

(4) The damping loss factors has great influence on the noises, so we can improve the scientific ship loss factor to reduce the underwater noise of the scientific investgation ship.

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