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Effect of longitudinal vibration of fluid-filled pipe with elastic wall on sound transmission character

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Abstract: When one end of a fluid-filled pipe with an elastic wall is fixed and a harmonic force effect acts on the other end, a steady longitudinal vibration will be produced. Compared to the pipeline resonance mode, the amplitude of the steady longitudinal vibration of an elastic pipe is greater, and the effect on the sound is also greater. The study of the steady longitudinal vibration of pipes can better describe the effects of fluid-filled pipelines on the radiation sound field of the pipe opening. Through the contrast between the analysis calculation of the equivalent beam model and the experimental results, the accuracy of the equivalent beam model for the calculation of the steady longitudinal vibration of pipelines is verified, and a method of isolating the steady longitudinal vibration state is proposed and verified.

Key words: fluid-filled pipe with elastic wall; sound transmission characteristics; steady longitudinal vibration

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0 Introduction

Elastic fluid-filled pipe systems have been more and more extensively applied in daily life, as well as in ships and aircraft industries. So the vibration of elastic pipes attracts much attention from the researchers. Besides the vibration induced by the pipe-transmitted energy at the valves or elbows, the pipe itself will also trigger vibration due to external forces. Such vibration could not only cause system fatigue and shorten the lifespan of the pipes and other elements, but also bring in noise pollution to some extent.

The experts and scholars at home or abroad have conducted a lot of investigations to understand the pipe vibration characteristics more profoundly. As early as in 1945, Rayleigh[1] looked insight on the bending and stretching vibrations of the cylindrical shells, and derived the free vibration frequency of the infinitely-long cylindrical shell in vacuum. In 1992, Makrides et al. [2] analyzed the vibration modes of the pipes through the liquid-elastomer coupling and the gas-elastomer coupling. Zhang et al. [3-6] also conducted some studies about the dynamic characteristics analysis and calculation method of fluid-filled pipe. In 2006, Zeng et al.[7] synoptically reviewed the vibration characteristics of fluid-filled pipes. In 2009, Jin et al.[8] analyzed the transfer matrix method of fluid-filled pipes. And in 2013, Wang et al. [9] explored the coupling effects between the fluid-filled pipe and the supporting system, and derived the motion equations for the coupling system based on finite element equation method.

This paper utilizes equivalent beam model to analyze the particle motion equation in the fluid-filled pipe, and gives the resonance frequency for the steady longitudinal vibration when one end of the fluid-filled pipe is fixed and the other undergoes the harmonic force. Through experimental analysis, the correctness and deviation of caclulating the longitudinal vibration of fluid-filled pipe by the beam model are verified. In addition, with many considerations

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being combined, this paper proposes the method for isolating the pipe longitudinal vibration, and validates it experimentally.

1 Particle motion equation of the equivalent beam model

Because the elastic fluid-filled pipe's length is usually far greater than its diameter, the longitudinal vibration model of rod is adopted for the calculation, with the vibration of the pipe itself being taken into account.

The longitudinal vibration equation for a rod can be illustrated as below:

$$\frac{\partial^2 u}{\partial t^2} - c^2 \frac{\partial^2 u}{\partial z^2} = 0 \tag{1}$$

where $c = \sqrt{\dfrac{E}{\rho}}$ indicates the propagation speed of rod longitudinal vibration; E is Young's modulus; ρ is rod density; u is longitudinal displacement; t is time factor; and z is longitudinal coordinate. The general solution of the vibration equation can be obtained by the variable separation method.

$$u(z, t) = (A\cos kz + B\sin kz)\cos(\omega t - \varphi) \tag{2}$$

In this formula, $k = \omega/c$ is known as wave number; ω is angular frequency; and A, B and φ are constants, which are determined by the initial conditions.

In practical application, the outlet end of the elastic fluid-filled pipe is often fixed onto some kind of supporting structure, while the inlet end will be equipped with a pump or other sound source that can produce harmonic pulsating pressure. The force excitation can be expressed by $F = F_0\cos(\omega t - \varphi)$.

Therefore, the boundary conditions can be expressed as follows:

$$u(z, t)_{z=0} = 0 \tag{3}$$

$$\left.\frac{\partial u(z, t)}{\partial z}\right|_{(x=l)} = -\frac{F_0}{ES}\cos(\omega t - \varphi) \tag{4}$$

where S indicates the cross-section area of the rod, and l means the rod length.

The boundary condition is substituted back to the general solution, then the displacement can be solved out, as follows:

$$u(z, t) = \frac{F_0}{ESk\cos(kl)}\sin kz\cos(\omega t - \varphi) \tag{5}$$

When $kl = (n - 1/2)\pi$, the displacement tends to be infinite. So, if the beam model is fixed at one end and exerted with a harmonic force of $F = F_0\cos(\omega t - \varphi)$, the resonance frequency can be written as:

$$f = \frac{c}{4l}(2n - 1) \tag{6}$$

where $n = 1, 2, 3, \cdots$.

2 Sound transmission characteristics of elastic fluid-filled pipe

In the cylindrical coordinate system, the Helmholtz equation of the elastic fluid-filled pipe can be expressed as:

$$\begin{cases} \nabla^2\Phi_0(r, \varphi, z) + k_0^2\Phi_0(r, \varphi, z) = 0,\ 0 \leqslant r < b \\ \nabla^2\Phi_p(r, \varphi, z) + k_p^2\Phi_p(r, \varphi, z) = 0,\ b \leqslant r \leqslant a \\ \nabla^2\boldsymbol{\Phi}_s(r, \varphi, z) + k_s^2\boldsymbol{\Phi}_s(r, \varphi, z) = 0,\ b \leqslant r \leqslant a \end{cases} \tag{7}$$

where Φ_0, Φ_p, $\boldsymbol{\Phi}_s$ represent the velocity potential functions of the sound field in the pipe-filling liquids, the longitudinal and lateral waves in the pipe wall respectively.

According to the boundary conditions, the particle's normal vibration velocity is continuous on the inner wall of the pipe, and the same is true for the sound pressure in the liquid, the normal stress on the pipe wall, as well as the tangential stress on the outer wall of the pipe. From Ref. [10], the eigen equation of normal wave in the elastic fluid-filled pipe can be derived as below:

$$\begin{vmatrix} a_{11} & a_{12} & a_{13} & a_{14} \\ a_{21} & a_{22} & a_{23} & a_{24} \\ a_{31} & a_{32} & a_{33} & a_{34} \\ a_{41} & a_{42} & a_{43} & a_{44} \end{vmatrix} = 0 \tag{8}$$

Let k_{zmn} be the eigenvalue of the (m, n)-order normal wave, that is, the solution of the (m, n)-order normal wave of the above eigen equation. $k_{zmn} = \sqrt{k_0^2 - \xi_{0mn}^2}$, $\xi_0^2 = \sqrt{k_0^2 - k_z^2}$, $k_0 = \omega/c_0$, k_z is the propagation factor of the normal wave in the fluid-filled pipe. The detailed deduction procedure can refer to Ref. [10]. When k_{zmn} = 0, i.e., $k_0^2 = \xi_{0mn}^2$, it corresponds to a specific frequency f_{mn} of the eigen equation. If above this frequency, the (m, n)-order normal wave will be a propagable wave; while it becomes a non-propagable wave below this frequency, and the specific f_{mn} will be the cutoff frequency of the (m, n)-order normal wave. Solving the eigen equation can get the cutoff frequency f_{mn} of the pipe's normal wave at any order. Above f_{mn}, the (m, n)-order normal wave is a propagable wave in the pipe, while it becomes a non-propagable wave and goes down exponentially below f_{mn}. Therefore, the cutoff frequency f_{01} of the (0, 1)-order normal wave is usually taken as the elastic pipe's cutoff frequency, below which no propagable planar wave exists.

This is because in the fluid-filled pipe, the characteristic impedance of liquid and pipe-wall metal material differs in one magnitude only, and the pipe cannot be taken as rigid surface but as elastic one on the fluid-solid coupling surface where the metal contacts with water. But in air, the conclusion is different, because the characteristic impedance of air and metal differs in about 5 magnitudes and the metal in air can be taken as rigid surface. So, the planar wave can exist in the pipes transmitting sound by air.

3 Influence of longitudinal pipe vibration on sound transmission

To verify the influence of longitudinal pipe vibration on sound transmission, this paper made tests on the thick-wall steel pipes with 2 different sound sources equipped at one end. The thick-wall steel pipes have outer radius of 3 cm, inner radius of 1.5 cm and length of 1.4 m. The two sound sources include: a planar piston transducer assembled with thick-wall pipe as the sound pipe (sound source 1), and a 8150 hydrophone produced by B&K Inc. from Denmark (sound source 2).

Reverberation method was adopted to conduct the experiments. Based on the practical situation, the thick-wall steel pipe is bolted on the side-wall of the reverberation tank, and a rubber ring is placed in the middle for isolating vibration and for water-proof. The sound source is placed on the other end of the pipe and processed for water-proof. After that, an 8103 hydrophone from B&K Inc. is used to measure the sound energy radiated to the reverberation tank from the pipe opening, after the sound source radiation energy transmits through the pipe. The detailed testing structure is shown in Fig. 1.

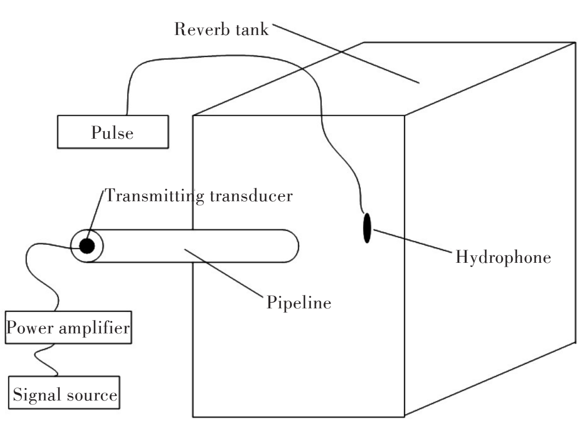


Fig.1 The system of the sound transmission characteristics measurement for the elastic pipeline filled with fluid

According to the previous calculation method, the (0, 1)-order cutoff frequency f_n for the experimental steel pipe has the analytical solution and measured value as listed in Table 1.

Table 1 Analytical solutions and measured values of the pipeline cutoff frequency

	Analytical solution	Measured value
Cutoff frequency f_n /kHz	24.0	15.0

Because the planar wave cannot propagate below the elastic fluid-filled pipe's cutoff frequency, whatever sound source is used, after its radiation energy transmits through the pipe, the energy radiated to the reverberation tank from the pipe opening should have the frequency above the cutoff frequency. The measuring results below the cutoff frequency should be nearly the same with background noise.

The measuring results on the thick-wall steel pipe with the sound source of planar piston transducer (sound source 1) are shown in Fig. 2.

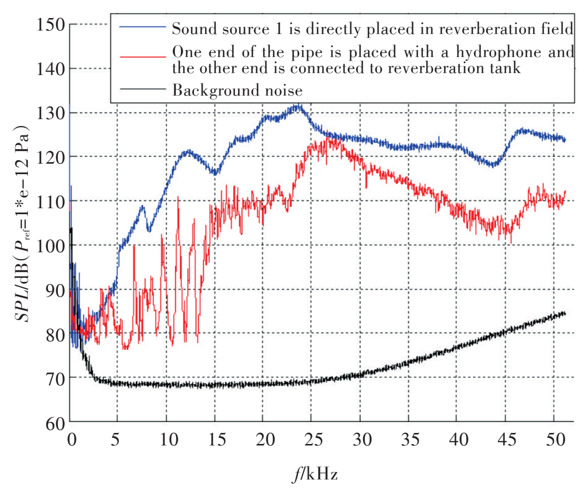


Fig.2 The radiated sound power through the pipeline when one end of the thick pipe is in reverberation field, and the other end is source 1

In Fig. 2, the upper curve represents the measured sound radiation power with the sound source being placed in the reverberation tank, while the lower curve indicates background noise, and the middle curve is the sound energy radiated to the reverberation tank from one end opening of the pipe when the sound source is placed on the other end. It's found that the pipe-opening radiated energy appears to be a continuous spectrum above the calculated cutoff frequency, representing the sound energy radiated to the reverberation tank after passing through the pipe. By contrast, below the cutoff frequency, there are still some discontinuous line-spectral signals. According to analysis, these signals could be induced by the radiated sound energy into the reverberation tank, due to that the longitudinal vibration of the pipe drives the vibration of the side-wall of reverberation tank. To prove this point, we first magnify the measuring results below the cutoff frequency to identify the frequencies of individual peak signals, as shown in Fig. 3.

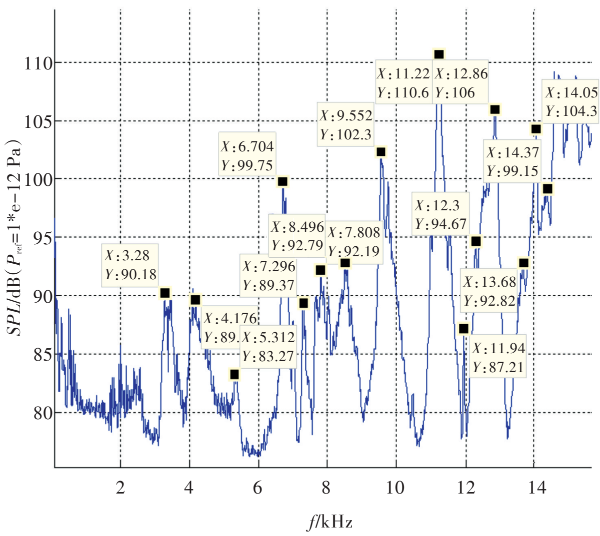


Fig.3 The sound transmission characteristics for elastic pipe with thick wall when $f < 15$ kHz(source 1)

The pipe's resonance frequencies are compared with the peak frequencies of the line spectra, and the results are listed in Table 2.

Table 2 The resonance frequencies and measured values of thick wall pipeline(source 1)

Order	Resonance frequency f_z /Hz	
	Theoretical value	Measured value
7	3 294	3 280
9	4 308	4 176
11	5 321	5 312
14	6 842	6 704
15	7 349	7 296
16	7 856	7 808
17	8 363	8 496
19	9 376	9 552
21	10 390	10 470
23	11 404	11 220
24	11 910	11 940
25	12 417	12 300
26	12 924	12 860
27	13 431	13 680
28	13 938	14 060
29	14 445	14 370

From Table 2, when sound source 1 sits on one end of the pipe and sound energy radiates to the reverberation tank from the pipe opening on the other end, the measured line-spectral peak frequencies below cutoff frequency are basically in accordance with the pipe's resonance frequencies. Meanwhile, as we compare the measured frequencies with the calculated resonance frequencies, we can get the maximal deviation of 3.07% and the minimal deviation of 0.19%, as shown in Fig. 4. Basically, it's sure that the measured discontinuous line-spectral signal below cutoff frequency just comes from the reverberation bank's side-wall vibration, which is in turn caused by the pipe's longitudinal vibration.

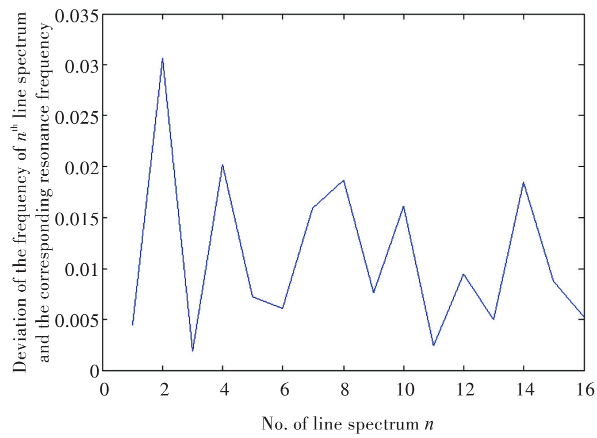


Fig.4 The deviation of the resonance frequency and measured value in thick wall pipe

The sound source is replaced with the 8105 hydrophone (sound source 2) and the same experiments are carried out on the thick-wall steel pipe. The results are shown in Fig. 5.

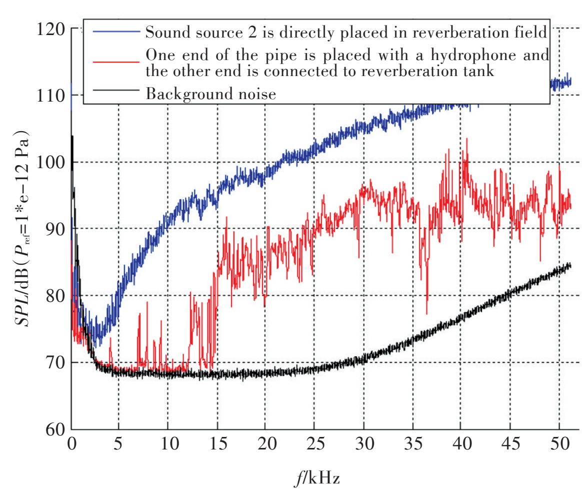


Fig.5 The radiated sound power through the pipeline when one end of the thick pipe is in reverberation field, and the other end is source 2

Again, the upper curve represents the measuring results with the sound source 2 being placed in the reverberation tank, while the lower curve indicates the background noise, and the middle curve is the sound energy radiated to the reverberation tank from one end opening of the pipe when the sound source is placed at the other end. As we can see, there are also some discontinuous line spectra below cutoff frequency. Based on the above analysis, we also magnify the results below cutoff frequency to locate the specific frequencies of these line spectral peaks, as shown in Fig. 6.

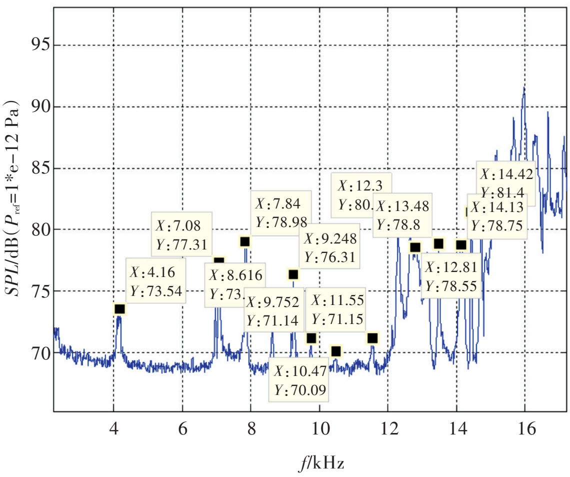


Fig.6 The sound transmission characteristics for elastic pipe with thick wall when f< 15 kHz(source 2)

The frequencies of these line spectra are compared with the resonance frequencies, and the results are shown in Table 3.

Table 3 The resonance frequencies and measured values of thick wall pipeline(source 2)

Order	Resonance frequency f_z /Hz	
	Theoretical value	Measuring value
9	4 308	4 160
15	7 349	7 080
16	7 856	7 840
18	8 869	8 616
19	9 376	9 248
20	9 883	9 752
21	10 390	10 470
23	11 404	11 550
25	12 417	12 300
26	12 924	12 810
27	13 431	13 480
28	13 938	14 130
29	14 445	14 420

Through one-to-one matching, the deviations between the measuring results and the resonance frequency can be obtained as in Fig. 7.

We can see that the maximal deviation is 3.66% and the minimal deviation is 0.17% , which once again proves that the measured signal below cutoff frequency just comes from the pipe's longitudinal vibration.

According to the above experimental analysis, we can know that the pipe's longitudinal vibration has a

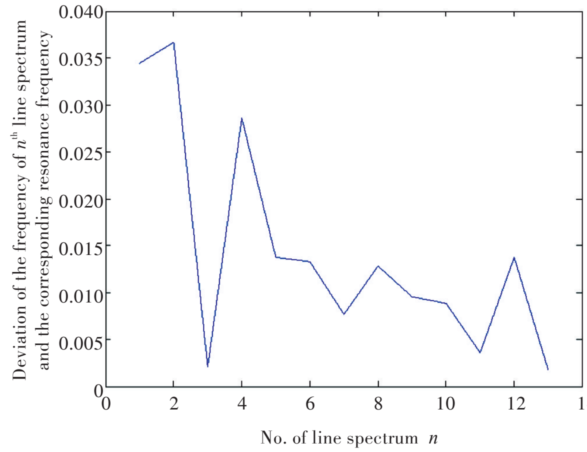


Fig.7 The deviation of the resonance frequency and measured value in thick wall pipe(source 2)

certain impact on the sound transmission characteristics of elastic fluid-filled pipe below cutoff frequency.

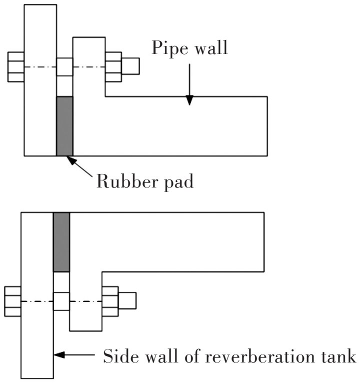
4 Isolation method of the pipe's longitudinal vibration

In the above experiments, although a rubber ring is inserted between the pipe and the reverberation tank side-wall for isolating vibration, the pipe's vibration could still be passed to the reverberation tank side-wall through the rigid connection with bolts. To utterly isolate the pipe's longitudinal vibration, the pipe, instead of being bolted to the reverberation tank, can be glued onto the reverberation tank by soft adhesive. On one side, it can guarantee the water tightness, and on the other side, it eliminates rigid connection between the pipe and the reverberation tank. Fig. 8 shows the two connection methods between the pipe and the reverberation tank.

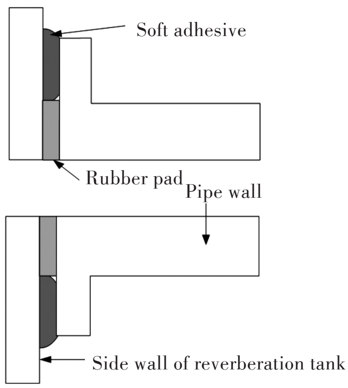
Fig. 8(a) depicts the bolt-connecting way, while Fig. 8(b) shows the soft connection by soft adhesive. We repeat the experiments by adopting the connection method in Fig. 8(b) and using 8105 hydrophone as sound source, and get the measuring results as shown in Fig. 9.

By soft connecting way, the radiated sound power through the pipe when one end of the thick pipe in reverberation field, and the other end is source.

We can see that, with soft connecting way, the elastic fluid-filled pipe's sound transmission effect is in accordance with the previous analysis completely. No planar wave exists any more below the cutoff frequency, and the measuring result nearly coincides with the background noise. The line-spectral signals measured below cutoff frequency are definitely caused by the conjunct vibration of the side-wall re-



(a) Bolt-connecting way



(b) Soft-connecting way

Fig.8 The attended mode between the pipeline and the reverberation tank

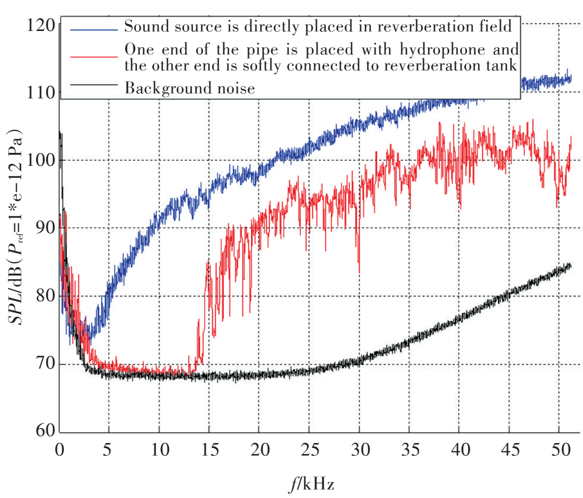


Fig.9 By soft connecting way, the radiated sound power through the pipeline when one end of the thick pipe in reverberation field, and the other end is source

verberation tank, which in turn caused by the pipe's vibration.

5 Conclusions

This paper considers the impact of elastic fluid-filled pipe's longitudinal vibration on the sound transmission characteristics, and uses the equivalent beam model to calculate the elastic pipe. The resonance frequency of the beam model's longitudinal vibration can be calculated. Through the experimental verification of the influence of two vibrating sound sources on the sound transmission characteristics of elastic pipe, it's found that the measured noise characteristic fits very well with the resonance frequency of the pipe longitudinal vibration, proving that the elastic pipe's longitudinal vibration has impact on the sound transmission characteristics below the cut-off frequency. To relieve such impact, this paper proposes to improve the pipe connection way by adopting soft adhesive connection method. It's proved by experimental measurement that this method can effectively eliminate the longitudinal vibration's influence on the sound transmission of the pipe.

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基于航速保持的舵减摇控制方法

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摘　要: 船舶航行受阻力影响引起航速和能量损耗。研究船舶在静水和波浪中的附加阻力,给出船舶航行时的总体航速损失的计算方法。设计带有航速损失约束的自动舵控制系统,依据舵角协同控制方法设计航向和舵减摇滑模控制规律。综合讨论“航向”与“航向+减摇”两种工作情况,包括横摇稳定、航向精度、航速保持、操舵能量消耗。仿真结果表明:该方法可以有效保持航速;从航行经济性的角度,对于同时安装有减摇鳍和自动舵的船舶,不推荐采用舵鳍联合减摇的控制方法。

关键词: 船舶; 运动控制; 舵减摇; 自动舵; 附加阻力; 航速损失; 滑模控制

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纵振动对声传输测量带来的干扰及其避免方法

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摘　要: 弹性充液管道在一端固定,另一端受到谐和力作用时自身会产生稳态纵振动。相比于管道自身模态的谐振,弹性管道稳态纵振动的幅度更大,对于声场的影响也更大。对于管道稳态纵振动的研究可以更好地说明充液管道对管口辐射声场的影响。通过等效梁模型的解析计算及与实验结果的对比,验证了等效梁模型用于计算管道稳态纵振动的正确性,同时,提出一种用于隔离管道纵振动的方法,并通过实验验证了其有效性。

关键词: 弹性充液管道; 声传输特性; 稳态纵振动