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Research on Vibration Characteristics of the Pedestal and Equipment Systems with Variable Parameter

Hailong Chen [a], Zhonghui Qiu [b], Fuzhen Pang [c], Lv Zheng

College of Shipbuilding Engineering, Harbin Engineering University, Harbin 150001, China

[a]chenhailong@hrbeu.edu.cn, [b]qiuzhonghui01@163.com, [c]pangfuzhen@yahoo.com.cn

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Abstract. According to the relation between exciting force and the pretightening force, the parameter of pedestal and equipment systems can be variable or invariable. The differential equation for forced vibration problem of the system has been established. The condition whether piecewise linear vibration will happen has been found. The result shows that the system will appear multi-frequency vibration response on the condition of single-frequency turbulence force. The results in this paper can be applied to the installation of the equipment.

Introduction

In actual engineering field, the study on the complex vibration induced by the equipment connected with marine pedestal through bolts with pretightening force has received extensive concern. When the exciting force doesn't beyond the limitation that will cause parameter variable in one vibration period, the phenomenon of the micro-amplitude vibration on the equipment can be observed，that is，the reciprocating tension and compression of the pedestal panel is at small displacement amplitude[1]. Thus, the system stiffness always can be tread as the pedestal panel stiffness and the whole vibration system can be approximately regarded as linear vibration system[2-3]. Now the forcing term of the vibration equation performs a leading role in the vibrational solution. As the driving force exceeds the limitation, for part of the vibration period, the connected bolts will be pulled at large displacement size and meanwhile the pedestal panel will be pressed by the equipment. Due to the difference between the pedestal stiffness and the bolts stiffness, the whole system stiffness will jump in different time period and the vibration system then can be treated as piecewise linear system[4]. In addition to the forcing term, the vibrational solution contains the complicated parameter terms such as natural vibration frequency.

The motion differential equations

Based on structural dynamics, this paper systematically analyses the vertical vibration problem of the pedestal and equipment system. When the sine exciting force acts on the mass center of equipment, the vibration is typical and can be considered to be single-degree-of-freedom. According to Newton's second law, the differential equations of non-damping vibration of the linear system and the piecewise linear system are established.

When the exciting force amplitude holds within limits, the vibration displacement of the equipment is within the prestrain of the bolts. The system stiffness remains the stiffness of the pedestal panel throughout. The system can be considered to be linear. The motion equation is：

$$m\ddot{x}+\left(k_1+k_2\right)x=F\sin\omega t \qquad (1)$$

Once the exciting force amplitude exceeds the limits valuation, the pulling displacement amplitude is greater than the prestrain and the tension of the connected bolts will occur in the vibration period. The system stiffness consists of the pedestal panel stiffness and the bolts stiffness. The system becomes piecewise linear. The following is schematic diagram：

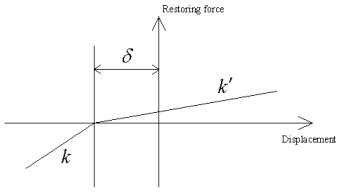

Fig.1. Relation curve of restoring force and displacement

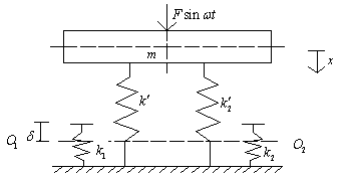

Fig.2. Schematic diagram of variation parameter system

Fig.1 shows the relation curve of restoring forceand displacement. δ is the prestrain of the connected bolts. In Fig.2, the significance of the symbols is as listed below: O_1, O_2, are the static equilibrium position when the certain preload is applied on the connected bolts; x_1, x_2 are the respectively vertical vibration displacements of two connected bolts and downward direction is defined positive； m stands for the mass of equipment; $F\sin\omega t$ stands for the exciting force;

When the tensile strain of the bolts exceeds prestrain（prestress can generally reach 50% ~ 70% of material yield limit），the system stiffness will become the stiffness of the connected bolts. The motion equation is：

$$m\ddot{x} + \left(k_1 + k_2\right)x = F\sin\omega t \qquad x > \delta \tag{2}$$

$$m\ddot{x} + \left(k_1' + k_2'\right)x = F\sin\omega t \qquad x < \delta \tag{3}$$

k_1', k_2' are the stiffness of the bolts. Because the system vibration is cyclical，equation（2）and equation（3）will appear alternately in the vibration period.

The vibration characteristics of the constant parameter system

The motion equation of the constant parameter system is equation（1）. By solving, the natural frequency of the system is $p^2 = 2k/m$. The frequency ratio is $\gamma = \omega/p$. Based on the vibration theory, the general solutions for displacement and speed of equation（1）are:

$$x = B\sin pt + D\cos pt + \frac{F}{2k}\frac{1}{1-\gamma^2}\sin\omega t \tag{4}$$

$$\dot{x} = Bp\cos pt - Dp\sin pt + \frac{F}{2k}\frac{\omega}{1-\gamma^2}\cos\omega t \tag{5}$$

Where the coefficient B, D can be found based on the initial conditions. If the initial displacement and speed are set as $x_0, \dot{x}_0$, the general solution is:

$$x = \frac{\dot{x}_0}{p}\sin pt + x_0\cos pt + \frac{F}{2k}\frac{1}{1-\gamma^2}\left(\sin\omega t - \frac{\omega}{p}\sin pt\right) \tag{6}$$

In most cases the initial condition is assumed to zero so as to simplify the analysis. Through the formula（6）,the condition is obtained that when $\delta < \min(x)$ is fulfilled, namely:

$$\frac{F_{pre}}{k} < \min\left(\frac{F}{2k}\frac{1}{1-\gamma^2}\left(\sin\omega t - \frac{\omega}{p}\sin pt\right)\right) \quad (7)$$

the piecewise linear situation of the variable parameter system will not appear.

To analyses the vibration characteristics of the system, as samples, the system parameters are shown as follows: $k' = 10\text{N/m}$ $k = 20\text{N/m}$,, $m = 10\text{kg}$, $F = 10\text{N}, \omega = 1\text{rad/s}$ .The distance from connected bolts to equipment is $l = 1\text{m}$. Based on the above system parameter, the natural frequency is $p_1 = 2\text{rad/s}$ and frequency ratio is $\gamma = \omega / p_1 = 1/2$ .The vibration curve is shown in Fig.3:

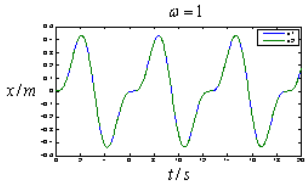

Fig.3 The vibration curve of the constant parameter system

Because the frequency ratio $\gamma = \omega / p_1 = 1/2$ is integer multiple, the vibration curve is periodic in Fig.3. By observing, the vibration period is 2π s and the displacement amplitude is 0.43m which acts as the limitation whether piecewise linear situation will occur.

The vibration characteristics of the variable parameter system

When the limitation condition can't be fulfilled, two kinds of system stiffness will exist in the vibration equation. When $x < \delta$, equation （3）takes effect and the natural frequency of the system is $p'^2 = 2k'/m$ and the frequency ratio is $\gamma' = \omega / p'$. The general solution for displacement of equation（3） is:

$$x = \frac{\dot{x}_0}{p'}\sin p'(t-t_0) + x_0\cos p'(t-t_0) + \frac{F}{2k'}\frac{1}{1-\gamma'^2}\left(\sin\omega(t-t_0) - \frac{\omega}{p'}\sin p'(t-t_0)\right) \quad (8)$$

In the equation， $x_0, \dot{x}_0$ are the displacement and speed of last current state and t_0 is the hour when the stiffness changes. They will act as the initial conditions and the initial hour on next stage. Because of the periodic variation of stiffness, the displacement (δ), speed and t_0 of last current state have periodic effects on the vibration solutions on next stage.

For the sake of comparison, the system parameters is identical to the above. When $\delta > \min(x)$, namely $\frac{F_{pre}}{k} > \min\left(\frac{F}{2k}\frac{1}{1-\gamma^2}\left(\sin\omega t - \frac{\omega}{p}\sin pt\right)\right)$, the displacement amplitude exceeds the prestrain of the connected bolts （ $\delta > -0.43$ ） . The vibration of the equipment can be treated as the piecewise linear system. When the prestrain is set as $\delta = -0.3\text{m}$, the natural frequency of the second case of the system is $p_2 = 1.41\text{rad/s}$ and the frequency ratio is $\gamma = 0.71$. The vibration curve is shown in Fig.4:

（a） $\omega = 1$ rad/s （b） $\omega = 1.4$ rad/s

Fig.4 The vibration curve of the variable parameter system

In Fig.4（a）, the horizontal line stands for the location of the prestrain of the connected bolts. Compared with Fig.3, vibration situation is the same on the initial stage. But when the displacement exceeds the prestrain of the bolts, the system stiffness jumps and the following vibration curve changes correspondingly. Fig.4 shows that the displacement amplitude is larger and the vibration is asymmetry. In order to analyses the vibration characteristics of the variable parameter system, when the frequency of the exciting force is close to the natural frequency of the second case($\omega = 1.4$ rad/s), the vibration curve is shown in Fig. 4（b）. The phenomenon is observed：when the frequency of the exciting force is close to the natural frequency p_2， the displacement amplitude will increase upward and there is a clear distinction between two frequencies of the variable parameter system.

Conclusions

In this paper, the vibration characteristics of the variable parameter system are analyzed. The results show that when the displacement amplitude exceeds the prestrain of the connected bolts, the vibration of the equipment is piecewise linear system. The dynamical response under the single-frequency turbulence force is multifrequency. The value of the prestrain should be well controlled so that it exceeds the displacement amplitude. When the installation of equipment can't meet the requirements of the pretightening force, ω should stay away from the natural frequencies, p_1 and p_2 ,so that the harmful resonance phenomena can be eliminated in the working range of oscillation frequency.

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References

[1]. Jazar,G.N.,Mahinfalah,M.; Deshpande,S: Proceedings of the Institution of Mechanical Engineers, Part K: Journal of Multi-body Dynamics Vol. 221(2007), p. 441-449.
[2]. Huidong Xu, Jianhua Xie: Journal of Vibration and Shock Vol. 27(2008), p. 20-24. In Chinese
[3]. Fuzhen Pang, Botian Yu, Danzhu Yu, Xiongliang Yao: SHIP ENGINEERING Vol. 33(2011), p. 36-40. In Chinese
[4]. Wei Wang, Qichang Zhang, Ruilan Tian: Journal of Vibration and Shock Vol. 27(2008), p. 130-133. In Chinese