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Characteristic Analysis of Two Level Vibration Reduction System for Construction Machinery

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Abstract. In this paper, the characteristics of the two stage vibration reduction system of construction machinery are analyzed. The vibration characteristics of this kind of construction machinery are analyzed by means of the modal analysis method, which is based on the two degrees of freedom of the suspension construction machinery. The approximate method for calculating the natural frequency of the two stage vibration reduction system of the engineering machinery is obtained, and the general principle of the design of the two stage vibration reduction system is proposed.

Introduction

Vibration reduction is a very important part of modern engineering machinery and equipment, the current textbooks, the first class of vibration reduction system are discussed in detail. Actually, many construction machinery equipment has more than one grade absorber. For example, sitting in mechanical engineering elastic seat, an elastic device is fixed in mechanical engineering frame engine; if the engineering machinery frame elastic suspension as a shock absorber, seat (or engine) following the shock absorber for secondary shock absorber. This system can be simplified as shown in Figure 1 with two level vibration control system. This paper tries to analyze the motion law of the two level vibration reduction system.

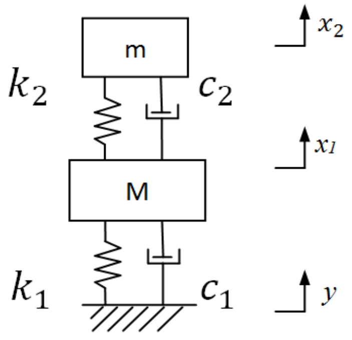

Figure 1

Mathematical Model

In the simplified model shown in Figure 1, M , m respectively for the one or two level of vibration damping quality, k_1 , k_2 respectively for one or two damping spring stiffness, c_1, c_2 respectively for one or two damping, x_1, x_2 respectively, M, m displacement. The displacement of the vertical direction caused by the ground surface is expressed by y, which is regarded as an incentive. In this way, we can establish the mathematical model (1).

$$\begin{cases} M\ddot{x}_1+k_1(x_1-y)+k_2(x_1-x_2)+c_1(\dot{x}-\dot{y})+c_2(\dot{x}_1-\dot{x}_2)=0 \\ m\ddot{x}_2+k_2(x_2-x_1)+c_2(\dot{x}_1-\dot{x}_2)=0 \end{cases} \tag{1}$$

When the damping is ignored, (1) can be simplified as

$$\begin{cases} M\,x_1+(k_1+k_2)\ddot{x}_1-k_2x_2=k_1y \\ m\ddot{x}_2+k_2(x_2-x_1)=0 \end{cases} \tag{2}$$

(2) as matrix form is

$$\begin{bmatrix} M & \\ & m \end{bmatrix}\begin{bmatrix} x_1 \\ x_2 \end{bmatrix}+\begin{bmatrix} k_1+k_2 & -k_2 \\ -k_2 & k_2 \end{bmatrix}\begin{bmatrix} x_1 \\ x_2 \end{bmatrix}=\begin{bmatrix} k_1y \\ 0 \end{bmatrix}$$

Shorthand for

$$M\vec{x}+K\vec{x}=\vec{F} \tag{3}$$

Modal Analysis

The free vibration of $\vec{F}$=0, (3) type can be

$$M\vec{x}+K\vec{x}=0 \tag{4}$$

The characteristic equation of the system is

$$\begin{vmatrix} k_1+k_2+M\omega^2 & -k_2 \\ -k_2 & k_2-m\omega^2 \end{vmatrix}=0 \tag{5}$$

Can be solved

$$\omega_{1,2}^2=\frac{k_1m+k_2(M+m)\mp\sqrt{[k_1m+k_2(M+m)]^2-4Mmk_1k_2}}{2Mm} \tag{6-1}$$

The above formula can also be written as

$$\omega_{1,2}^2=\frac{(k_1+k_2)m+k_2M\mp\sqrt{[(k_1+k_2)m+k_2M]^2-4Mmk_1k_2}}{2Mm} \tag{6-2}$$

This characteristic equation has two positive real roots of ω_1^2 , ω_2^2 , so the system has two frequency ω_1, ω_2. The type (6) in the ω_1^2 , ω_2^2 into (4) solution obtained two modal vector

$$\begin{bmatrix} A_{11} \\ A_{21} \end{bmatrix}=\begin{bmatrix} k_2 \\ k_1+k_2-M\,\omega_1^2 \end{bmatrix}$$

$$\begin{bmatrix} A_{12} \\ A_{22} \end{bmatrix}=\begin{bmatrix} k_2 \\ k_1+k_2-M\,\omega_2^2 \end{bmatrix}$$

Let system for modal coordinates $\vec{q}=[q_1, q_2]T$ analysis method according to the modal, between it and the physical coordinates of $\vec{x}$ should meet the type

$$\vec{x}=\begin{bmatrix} x_1 \\ x_2 \end{bmatrix}=A\vec{q} \tag{7}$$

A is the modal matrix of the system,

$$A=\begin{bmatrix}A_{11}\ A_{12}\\A_{21}\ A_{22}\end{bmatrix}=\left[k_1+k_2{}^{k_2}-M\omega_1^2\ k_1+k_2{}^{k_2}-M\omega_2^2\right] \tag{8}$$

The type (7) into the equation (3) to

$$MA\vec{q}+KA\vec{q}=\vec{F} \tag{9}$$

The type (9) on both sides of the same type to transpose matrix AT is multiplied by A

$$ATKA\vec{q}+ATKA\vec{q}=AT\vec{F} \tag{10}$$

$$Let\cdot Mp=ATMA=\begin{bmatrix}M_{p1} & \\ & M_{p1}\end{bmatrix} \tag{11}$$

$$K_p=ATKA=\begin{bmatrix}K_{p1} & \\ & K_{p2}\end{bmatrix} \tag{12}$$

Where: M_p, K_p are respectively the modal mass matrix and modal stiffness matrix, M_{p1}, M_{p2} respectively for the first order and second order modal mass, K_{p1}, K_{p2} respectively for the first order and second order modal stiffness formula, Plug the type (8), (11), (12) into (10) the vibration equation can be decoupled vibration equation.

$$\begin{cases}M_{p1}\ddot{q}_1+K_{p1q1}=A_{11}k_1y\\M_{p2}q_2+K_{p2\ddot{q}2}=A_{12k_1y}\end{cases} \tag{13}$$

Let y $=A_y$ sin pt， The formula (13) has a stable solution

$$\begin{cases}q_1=\dfrac{A_{11}k_1}{k_{p1}-M_{p1}p^2}A_y\sin pt=\dfrac{1}{M_{p1}}\dfrac{k_1k_2}{\omega_1^2-p^2}A_y\sin pt\\q_2=\dfrac{A_{12}k_1}{k_{p2}-M_{p2}p^2}A_y\sin pt=\dfrac{1}{M_{p2}}\dfrac{k_1k_2}{\omega_2^2-p^2}A_y\sin pt\end{cases} \tag{14}$$

The type (14) into the equation (7) ,the equation (3) to the stable solution

$$\begin{cases}x_1=\left(\dfrac{1}{M_{p1}}\dfrac{k_1k_2^2}{\omega_1^2-p_2}+\dfrac{1}{M_{p2}}\dfrac{k_1k_2^2}{\omega_2^2-p_2}\right)A_y\sin pt\\x_2=\left(\dfrac{\left(k_1+k_2^2\right)}{M_{p1}}\dfrac{k_1k_2^2}{\omega_1^2-p_2}+\dfrac{\left(k_1+k_2-M\omega_2^2\right)}{M_{p2}}\right)A_y\sin pt\end{cases} \tag{15}$$

Discussion on the Characteristics of the Two Stage Vibration Reduction System for Construction Machinery

(1) As with the other two degrees of freedom vibration, the two stage vibration isolation system has two natural frequencies, and the numerical value can be calculated by the formula (6).

(2) The following two points are always established for the engineering machinery with a two level vibration reduction system:

Chassis quality is always far greater than the driver, the quality of the engine, that is, M >>m;

The stiffness of the suspension spring is much larger than that of the driver's seat, and the stiffness of the engine vibration damper is $k_1 >> k_2$.

We ignore the M in the formula (6-1)（M +m）, or the k_2 in the formula (6-2), （$k_1 + k_2$）,can be drawn:

$$\omega_1 \approx \sqrt{k_1/M} \tag{16}$$

$$\omega_2 \approx \sqrt{k_2/M} \tag{17}$$

So, the natural frequency of the two stage vibration isolation system of the construction machinery can be calculated in accordance with the formula (16) and formula (17).

(3) For most construction machinery, we are concerned with reducing the vibration of the engine, the vibration of the driver, which is to reduce the x_2 amplitude. The type (16) and (17) into (15) second type of:

$$x_2 \approx \left(\frac{1}{M_{p1}} \frac{k_1 k_2^2}{\omega_1^2 - p_2} \frac{\left(k_1 + k_2 - M\omega_2^2\right)}{M_{p2}} \frac{k_1 k_2^2}{\omega_2^2 - p_2} \right) A_y \sin pt \tag{18}$$

From the formula (18) can be seen in the following points:

(a) When the vibration frequency of p ground is equal to the natural frequencies of ω_1, any one of the ω_2, x_2 tends to infinity, the system will resonate.

(b) Decreasing the spring stiffness k_1 and k_2 can reduce the, that is, x_2 reducing the system stiffness can improve the effect of vibration reduction.

(c) The increase of the modal mass M_{p2} and M_{p2} can reduce the x_2. Due to the modal mass M_{p2} , M_{p2} is mainly composed of M, m conversion, so, it can be considered to increase the quality of system components can increase the effect of vibration reduction.

(d) It can be seen that when mk_1=Mk_2, the type (18) second brackets is 0, that is to say it can also improve the damping effect. In fact, by the formula (16), (17) the mk_1=Mk_2 is ω_1=ω_2, so the design of construction machinery two damping system, should be roughly equal to two natural frequencies.

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References

[1] Yu Luping (ed.), Analysis of vibration reduction characteristics of track type bulldozer of elastic suspension (construction machinery, 2009) pp. 25-28.

[2] Zhang Yimin(ed.), Mechanical vibration(Tsinghua University press, 2007,3).

[3] Liu Fengqiao and Wen Bangchun, Theory and application of vibration machinery (echanical Industry Press, 1985).

[4] Zheng Beichang (ed.), mechanical vibration (Mechanical Industry Press, 1980).

[5] Fang Tong and Xue Pu, Vibration theory and application (Northwestern Polytechnical University press, 1998).