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AN INVESTIGATION OF VIBRATION CONTROL FOR VIBRATORY MACHINES ON A TWO-DIMENSIONAL SUPPORTING STRUCTURE USING SYNCHROPHASING

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In this paper, we investigated the application of synchrophasing technique in controlling vibration of vibratory machines on a two-dimensional structure. Two cylinder machine models of even-mass distribution with four symmetrical supports installed on a periodic supporting structure model were analyzed. The main purpose of this research is to minimize the total structure-borne sound power transmitted from the machines to the supporting structure. In order to determine the structure-borne sound power, source mobility of the two machines was calculated theoretically. In addition, point mobility and transfer mobility between installation points on the supporting structure were calculated by the finite element simulation. Finally, the optimum phase angles between the two machines were searched for different frequencies of vibration. Optimized structure-borne sound power transmission was compared with structure-borne sound power transmission without control. The results confirm that synchrophasing technique can be an effective control technique in suppressing structure-borne sound power transmission of machines on a two-dimensional supporting structure.

1. Introduction

There are various kinds of noisy machines installed in buildings, such as chillers, pumps, boilers, air compressors, and other kinds of vibratory machines. Structure-borne sound power can be transmitted from these kinds of machines to the wall and the floor structure, which is eventually emitted as sound into indoor spaces and disturb residents. Structure-borne sound transmission of vibratory machines are therefore being analysed by many researchers and engineers [1].

Mak and Yun [2, 3] investigated the structure-borne sound power transmission from two coherent machines to a floor structure in buildings and a dual-layer coupling floor structure, respectively. Analysis results showed that the structure-borne sound power from the two coherent machines can be magnified at some frequencies, which means that the coupling effects exist. Since the coupling effects lead to the magnification of the total power, it can be utilized to reduce total power by appropriate control measures. Therefore, the controlling strategy of changing the phase difference of electrical supply between the two coherent machines, which called synchrophasing control technique was tested. It can be a supplementary controlling measure to suppress the total structure sound power transmission from the two machines besides traditional passive vibration control method, such as: passive vibration isolation and passive vibration absorption. In reality, it can be found that some machines are installed coherently. For example, two coherent fans are installed in some air-conditioning outdoor machines. In this kind of coherent machines system, synchrophasing technique can be an appropriate choice for that it is easy to be applied and only the phase difference of electrical supply between the two machines need to be adjust.

Synchrophasing control technique was first investigated in controlling propeller noise of aircraft [4-6], which can be dated back to 1980s. It has been investigated in noise control solutions for more than thirty years. However, the researches in vibration control solutions were started in recent years. Dench et al. [7] demonstrated that synchronisation is an effective vibration control technique in decreasing vibration transmit to a one-dimensional structural through theoretical analysis and experiment. Yang et al. [8, 9] test the possibility of control vibration transmission of multiple rotating machines installed on a floating raft through experiments. In those papers, the square of velocity was selected to be the cost function. While, Mak and Su [10-12] pointed out that the total structure-borne sound power transmission is closer than the transmitted forces to the sound radiation as it considers the interaction and phase difference of motion between the complex vibratory source and the receiving floor structure. Thus, the structure-borne sound power transmission may be a better index in assessing the control effect. In this research, the structure-borne sound power was selected to be the cost function in assessing the vibration control effect and searching for the optimum phase difference.

A cylinder machine model with four installation points installed on a supporting structure were used in this research. The mobility method is utilized for modelling the structure-borne sound power transmission from the two machines to the supporting structure. The structure-borne power transmission in the vertical direction was analysed. To calculate the structure-borne sound power transmission, source mobility of the machine model, point mobility and transfer mobility of the supporting structure were investigated.

2. Theory

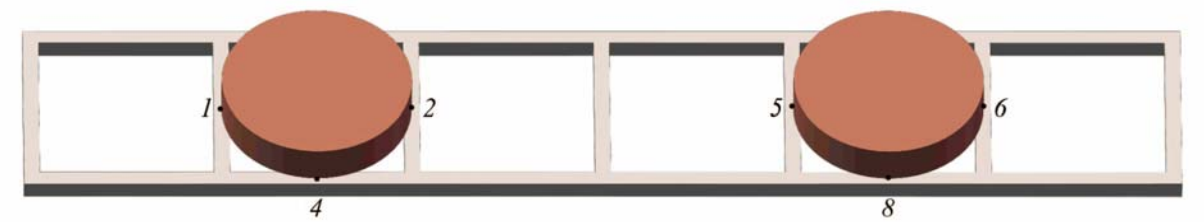

Figure 1: The schematic of the supporting structure with the two machines.

A schematic in Fig. 1 presents two cylinder machines installed on a supporting structure. The total structure-borne sound power, which represents the structure-borne sound power transmission, is selected to be the cost function. By comparing the total structure-borne sound power transmit from the two machines to the supporting structure, the effectiveness of this control technique can be validated. The source mobility of the machines, the point mobility and transfer mobility between installation points on the supporting structure were analysed.

2.1 Source mobility

A cylinder machine model with four symmetrical installation points (*p1*, *p2*, *p3*, *p4*) were analysed in this research, as shown in Fig. 2. The radius of the machine model is *R*, and the height is *H*. The source mobility matrix of this machine model is expressed by a 4×4 matrix for the installation points 1, 2, 3, and 4, as follows:

$$[Y_S] = \frac{1}{jwM_m}\begin{pmatrix} 1+\alpha^2 & 1-\alpha^2 & 1 & 1 \\ 1-\alpha^2 & 1+\alpha^2 & 1 & 1 \\ 1 & 1 & 1+\alpha^2 & 1-\alpha^2 \\ 1 & 1 & 1-\alpha^2 & 1+\alpha^2 \end{pmatrix} \tag{1}$$

where w denotes the angular frequency, M_m denotes the mass of the cylinder machine model, and $\alpha^2 = 3R^2 \big/ \left(3R^2 + 4H^2\right)$.

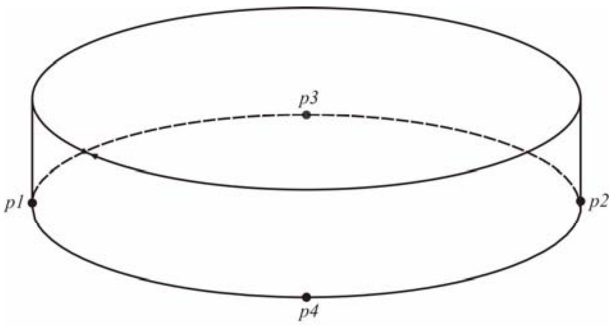

Figure 2: Model of the cylinder machine source with four installation points.

Four spring isolators were installed between each machines and the supporting structure to suppress the vibration. It is assumed that the central frequency of vibration of the machine is f_0=15Hz. The mobility of the four spring isolators under each machine can be expressed by a mobility matrix as follows:

$$[Y_K] = \frac{jw}{K_s}\begin{pmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{pmatrix} \tag{2}$$

where $K_s = (2\pi f_0)^2 \times M_m / 4$ stands for the axial stiffness of each spring isolator.

2.2 Transfer mobility and input mobility

The point mobility and transfer mobility can be determined by using the formula below:

$$Y_{ij} = V_i \big/ F_j \qquad i, j = 1, 2, \cdots, 8 \tag{3}$$

which V_i denotes the velocity of the i^{th} point on the supporting structure, F_j denotes the force applied on the j^{th} point, Y_{ij} denotes the transfer mobility from point i to point j if $i \neq j$, and Y_{ij} denotes the point mobility of point i if $i = j$.

For the two machines, the corresponding transfer mobility and point mobility can be expressed by a 4×4 mobility matrix, as follows:

$$[Y_{F1}] = \begin{pmatrix} Y_{11} & Y_{12} & Y_{13} & Y_{14} \\ Y_{21} & Y_{22} & Y_{23} & Y_{24} \\ Y_{31} & Y_{32} & Y_{33} & Y_{34} \\ Y_{41} & Y_{42} & Y_{43} & Y_{44} \end{pmatrix} \quad \text{and} \quad [Y_{F2}] = \begin{pmatrix} Y_{55} & Y_{56} & Y_{57} & Y_{58} \\ Y_{65} & Y_{66} & Y_{67} & Y_{68} \\ Y_{75} & Y_{76} & Y_{77} & Y_{78} \\ Y_{85} & Y_{86} & Y_{87} & Y_{88} \end{pmatrix} \tag{4}$$

2.3 Total Structure-borne sound power transmission

Assuming that the free velocity vectors of the machines are:

$$\left[V_{Si}\right] = V_i[1,1,1,1]^T \qquad i = 1,2 \tag{5}$$

where i stands for the numbers of the machines. Then, the dynamic force transmitted from the machines to the supporting structure at the four installation points can be calculated by:

$$\left[F_i^{'}\right]=\left(\left[Y_{Si}^{'}\right]+\left[Y_K\right]+\left[Y_{Fi}\right]\right)^{-1}[V_{Si}] \qquad i=1,2 \tag{6}$$

where Y_{Fi} denotes the matrix of floor mobility. If the coupling effects between the two machines were not considered, the power transmitted to the supporting structure is given by:

$$P_{wc}=\sum_{i=1}^{2}\mathrm{Re}([F_i^{'}]^{*\mathrm{T}}[Y_{Fi}][F_i^{'}]) \tag{7}$$

If the coupling effects were considered, the dynamic force transmitted from the machines to the supporting structure at the eight contact points is given by:

$$[F_T]=\left(\begin{bmatrix}Y_{s1} & O\\ O & Y_{s2}\end{bmatrix}+\begin{bmatrix}Y_{K1} & O\\ O & Y_{K2}\end{bmatrix}+\begin{bmatrix}Y_{F11} & Y_{F12}\\ Y_{F21} & Y_{F22}\end{bmatrix}\right)^{-1}\begin{bmatrix}V_{s1}\\ V_{s2}\end{bmatrix} \tag{8}$$

The total power transmitted from the two machines to the supporting structure is given by:

$$P=[F_T]^{*\mathrm{T}}\,\mathrm{Re}\left(\begin{bmatrix}Y_{F11} & Y_{F12}\\ Y_{F21} & Y_{F22}\end{bmatrix}\right)[F_T] \tag{9}$$

where Y_{s1} and Y_{s2} denotes the source mobility matrixes of the two machines, $Y_{F11}=Y_{F1}$ denotes the mobility matrix for the installation points 1, 2, 3, 4 for the 1st machine, and $Y_{F22}=Y_{F2}$ denotes the mobility matrix for the installation points 5, 6, 7, 8 for the 2nd machine, whilst $\left[Y_{F12}\right]=\left[Y_{F21}\right]$ are mobility matrices between the installation points 1, 2, 3, 4 and 5, 6, 7, 8.

3. Analysis

The radius of the two machines in this research is R=0.13m, and the height is H=0.05m. The rectangular supporting structure is plotted in Fig. 3. It is made of steel. The dimensions for the rectangular supporting structure are length L=1.46m, width W=0.26m, and thickness d=0.01m. There are six squares with length l=0.22m and interval d=0.24m apart. The parameters for the supporting structure are density ρ=7.8×10^3kg/m^3, Young's modulus E=2.0×10^{11}N/m^2, Poisson's factor μ=0.2, and loss factor η=0.05. There are eight installation points on the supporting structure, which used for installing the two machines. The two sides with width W and thickness d were set to be fixed in the finite element analysis. The frequency band of 0-200Hz was analysed.

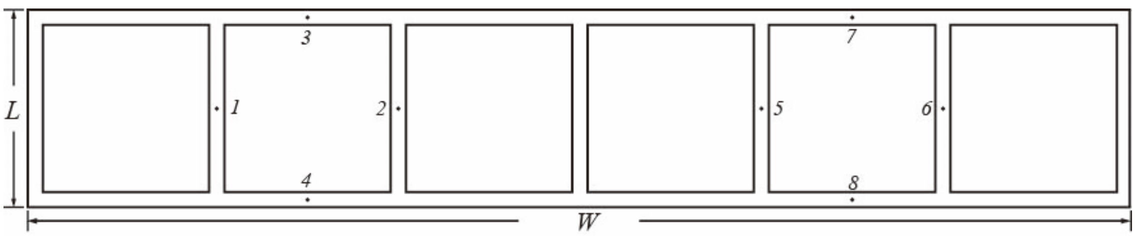

Figure 3: The schematic of the supporting structure.

The finite element analysis of the supporting structure was conducted by using COMSOL. In the analysis, the structure mechanics module was utilized, a three-dimensional model was build, and the frequency response of the supporting structure was calculated in the frequency band between 0Hz and 200Hz. With the simulation results, the required point mobility and transfer mobility on the supporting structure can be derived by applying Eq. (3). As an example, the real part and image part of transfer mobility between the 4th installation point and the 2nd installation point is plotted in Fig. 4. It is shown that there are four resonance frequencies in the two mobility-frequency curves. The magnitude of the first and the second resonance peak is significantly larger than the third and the four resonance peak in the two frequency-mobility curves.

Assuming that the free velocity of the two machines is a constant value in the frequency band being analysed. With the derived point mobility, transfer mobility, and the assumed free velocity, the total structure-borne sound power can be calculated by applying Eqs. (8) and (9). First, the total structure-borne sound power transmission of the two machines was calculated and the frequency-power curve was plotted in Fig. 5. Then, the minimum total structure-borne sound power transmission of the two machines were searched and the frequency-power curve was plotted in Fig. 5.

By comparing the two curves in Fig. 5, it can be found that the control effect of changing the phase difference between the two machines is effective. In most part of the frequency band, the difference of magnitude between the two curves is larger than 5dB. It means that by setting the phase difference value between the two machines, the structure-borne sound power transmission will be decreased significantly at the target frequency.

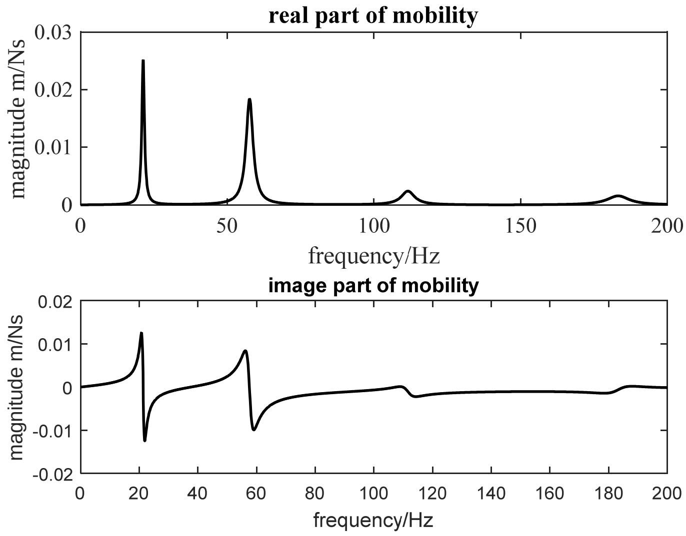

Figure 4: The transfer mobility between installation point 4 and installation point 2.

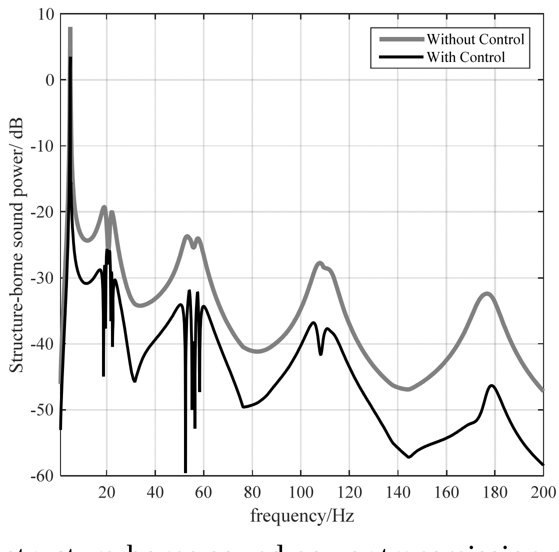

Figure 5: The total structure-borne sound power transmission of the two machines.

The optimum phase difference between the electrical supply of the two machines in the analysed frequency band were searched under the criteria of obtain the minimum structure-borne sound power. The frequency-phase curve is plotted in Fig. 6. It shows that the optimum phase difference value changed slowly in most part of the curve, except for the frequencies around the resonance frequencies. Thus, when setting the phase difference corresponding to a target frequency, the structure-borne sound power transmission of a frequency band around the target frequency will decrease to some extent.

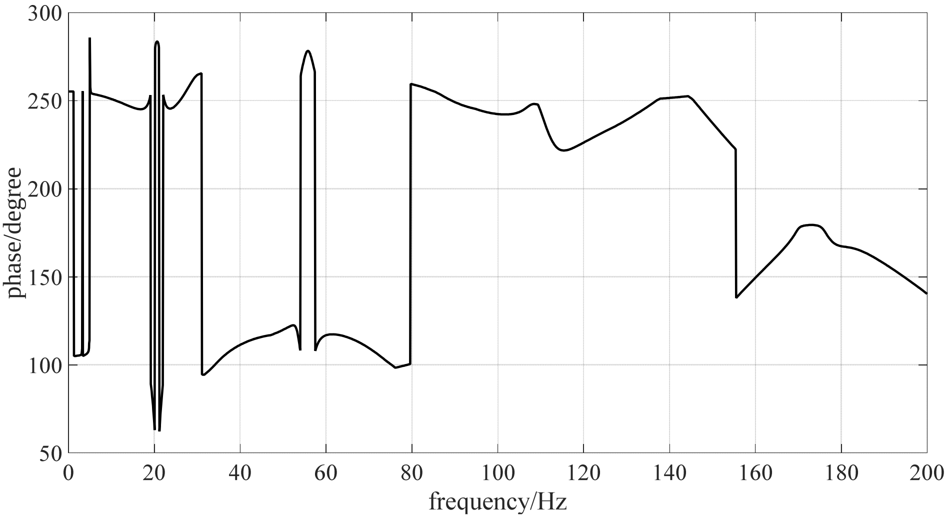

Figure 6: The optimum phase angles between the two machines.

4. Conclusions

The application of synchrophasing control technique in controlling structure-borne sound power transmission of two vibratory machines on a supporting structure was analysed. First, the mobility method was utilized in calculating the structure-borne sound power transmission from the two coherent vibratory machine to the supporting structure. Then, the transmitted structure-borne sound power with and without utilizing this technique was compared, and the optimum phase difference between the two machines was searched. The results show that the synchrophasing technique is effective in reducing structure-borne sound power transmission.

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