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DYNAMIC ANALYSIS OF A COMPRESSOR MOUNTING SYSTEM

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ABSTRACT

Compressors are a primary cause for numerous HVAC noise and vibration problems. Typically, vibration energy is transmitted from the compressor to the chassis through piping and isolation mounts. The design of these attachments is thus of paramount importance in determining the dynamic performance of a HVAC unit. Below certain frequencies, compressor vibration can be well described as a rigid-body having six degrees of freedom. In this paper, a procedure for complete determination of the compressor vibration is described based on the rigid-body model. It can be used to reduce the number of measuring points in a vibration monitoring test. Further, a dynamic compressor system model is presented that allows to predict force transmissibilities and the responses of the compressor to the applied forces. This model has been applied to determine the dynamic characteristics of a compressor system.

1. INTRODUCTION

HVAC units are mostly used to create a comfortable environment. However, the result can be adversely affected by the amount of noise and vibration inherent in the HVAC unit. Sound and vibration levels have become an important design concerns for a HVAC system design. Consequently, customer satisfaction has become a consistent driving force to reduce product noise and vibration levels. The cause of most of noise and vibration problems with HVAC systems can be traced to the compressors. While the noise directly radiated from the compressor can be effectively attenuated by using a sound shield, its effects may not be significantly eliminated because vibration and acoustic energies can be transmitted from the compressor to the rest of the unit through its attachment points such as mounts and piping. The design of these attachments is thus of primary importance in determining the sound and vibration performance of a HVAC unit. However, a good design often requires a full understanding of the dynamics of the compressor system, consisting of the compressor, its mounts and its attachment points to the HVAC unit. Even though the compressor can be treated as a rigid-body, there is no easy solution to the dynamics of the compressor system due to the complexity of the problem. Traditionally, the location of natural frequencies is typically used as a guideline for designing the isolation mounts. This procedure is often simplified further by modeling the compressor as a single-degree-of-freedom system. Consequently, there is a risk that the shift of the natural frequencies can not be captured due to the coupling of the mode shapes. In this paper, a procedure for completely determining the vibration of the compressor is described based on a rigid-body model. It can be used to reduce the number of measuring points in a vibration monitoring test. Further, a dynamic model of the compressor system is presented. This model allows prediction of force transmissibilities and the compressor response to the applied forces. This model has been applied to determine the dynamic characteristics of a compressor system.

2. DESCRIPTION OF COMPRESSOR VIBRATION

A compressor shell is usually hard enough for the compressor to be considered a rigid body and its vibration can be determined from rigid-body dynamics in a certain frequency range. In the rigid body model, the compressor motion can be conveniently described by six components (3 translations and 3 rotations) of the displacement at a reference point such as its center of gravity (C.G.). Thus described, the motion at any desired point is obtained from:

$$u = u_c + \omega \times r \qquad (1)$$

where $\boldsymbol{u}$=[u, v, w]^T and $\boldsymbol{u}_c$=[u_c, v_c, w_c] ^{T} are respectively the linear velocities at the point (x, y, z) and the C.G. (x_c, y_c, z_c), $\boldsymbol{\omega}$=[α, β, γ] ^{T} is the angular velocity of the rigid body, and $\boldsymbol{r}$=[x-x_c, y-y_c, z-z_c] ^{T}. Since it is usually difficult to measure the rotational vibrations directly, they are determined indirectly from the translational accelerations measured at some specified points. For example, for the transducer configuration shown in Figure 1, Eqn. (1) can be expanded as:

$$\begin{Bmatrix} u_1 \\ v_1 \\ w_1 \\ u_2 \\ w_2 \\ w_3 \end{Bmatrix} = \begin{bmatrix} 1 & 0 & 0 & 0 & z_1 & -y_1 \\ 0 & 1 & 0 & -z_1 & 0 & x_1 \\ 0 & 0 & 1 & y_1 & -x_1 & 0 \\ 1 & 0 & 0 & 0 & z_2 & -y_2 \\ 0 & 0 & 1 & y_2 & -x_2 & 0 \\ 0 & 0 & 1 & y_3 & -x_3 & 0 \end{bmatrix} \begin{Bmatrix} u_c \\ v_c \\ w_c \\ \alpha \\ \beta \\ \gamma \end{Bmatrix} \tag{2}$$

Once the accelerations on the left side of Eqn. (2) have been specified, the vibration at the C.G., and hence the vibration of the whole compressor, can be fully determined by solving Eqn. (2), provided that the coefficient matrix can be inverted. This provision is easily met by proper selection of transducer locations.

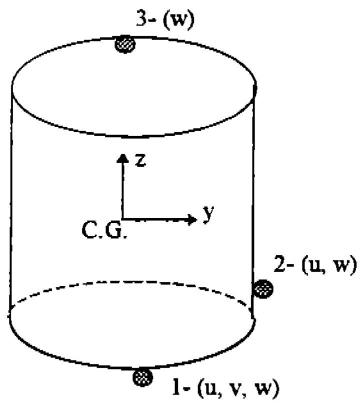

Figure 1. An example configuration of transducer placement

3. DYNAMIC MODEL OF A COMPRESSOR MOUNTING SYSTEM

As discussed in the preceding section, the compressor vibration can be treated as a rigid-body having six degrees of freedom. In order to address practical concerns in designing of a compressor mounting system, one has to be able to predict the response of the compressor to the applied forces. Doing this requires that a solution be sought to the set of six equations that govern the compressor vibration, only two of which are given below:

$$\sum_i k_{xx}^i u_c + \sum_i k_{xy}^i v_c + \sum_i k_{xz}^i w_c + \sum_i \left(k_{xz}^i a_y - k_{xy}^i a_z\right)\alpha + \sum_i \left(k_{xx}^i a_z - k_{xz}^i a_x\right)\beta + \sum_i \left(k_{xy}^i a_x - k_{xx}^i a_y\right)\gamma - \omega^2 m u_c = F_x \tag{3a}$$

$$\begin{aligned} &\sum_i \left(k_{xz}^i a_y - k_{xy}^i a_z\right)u_c + \sum_i \left(k_{yz}^i a_y - k_{yy}^i a_z\right)v_c + \sum_i \left(k_{zz}^i a_y - k_{yz}^i a_z\right)w_c + \sum_i \left(k_{yy}^i a_z^2 + k_{zz}^i a_y^2 - 2k_{yz}^i a_y a_z\right)\alpha \\ &\quad + \sum_i \left(k_{xz}^i a_y a_z + k_{yz}^i a_x a_z - k_{yz}^i a_z^2 - k_{zz}^i a_y a_x\right)\beta + \sum_i \left(k_{xy}^i a_y a_z + k_{yz}^i a_x a_y - k_{zz}^i a_y^2 - k_{yy}^i a_z a_x\right)\gamma \\ &\quad - \omega^2 I_{xx}\alpha + \omega^2 I_{xy}\beta + \omega^2 I_{xz}\gamma = M_x \end{aligned} \tag{3b}$$

with

$$k_{ij}^{l} = k_a^{l}\lambda_{ia}\lambda_{ja} + k_s^{l}\lambda_{is}\lambda_{js} + k_t^{l}\lambda_{it}\lambda_{jt}$$

in which λ_{ia} are the direction cosines of the a-th spring, k_a^{l} is the stiffness of the a-th spring at the l-th mounting point, a_i is the projection onto the i-axis of the distance from a mounting point to the compressor C.G., and I_{ij} are the moments of inertia of the compressor. These definitions are more easily understood when one considers that a real isolation mount can be generally viewed as a set of three simple springs: one in the axial direction and two in the transverse directions. The axial spring maintains the axial stiffness of the isolation mount and both transverse springs maintain the radial stiffness. The absolute orientations of the transverse springs are not necessary as long as they are orthogonal to each other. In this study, the first transverse spring is such orientated that it lies in the plane defined by the z-axis and the axis of the mount. Translational motion in the direction of the x-axis and rotation about the x-axis are addressed by Eqns. (3a) and (3b). The remaining four equations can be readily obtained by making use of the symmetry of the dynamic equations. Examining Eqn. (3) reveals that the six components of motion are generally coupled. Consequently, the natural frequencies of the dynamic system are not only determined by the spring rates of the mounts, but also by their positions with respect to the C.G. of the compressor. Damping effects are ignored for simplicity in Eqn. 3. These effects, however, can be included in the analysis by expressing the spring rates as complex numbers.

Though Eqn. (3) can be solved repeatedly at each frequency, it is preferable to use the modal superposition technique, in which the physical variables are expressed as a linear combination of the normal modes of the dynamic system. By so doing, Eqn. (3) can be decoupled and solved independently. The resulting expression for the vibration at the C.G. can be written as:

$$u_c = \sum_i \frac{\psi_i^T F \psi_i}{\omega_i^2 - i\omega\zeta_i - \omega^2} \qquad (4)$$

where ψ_i is the i-th mode, ω_i is the corresponding resonance frequency in radians, and ζ_i is the critical modal damping ratio. The problem of solving Eqn. (3) is therefore reformulated to determining the modal parameters of the compressor system.

4. RESULTS AND DISCUSSIONS

We will first show some results related to the procedures described in Section 2. The transducer placement shown in Figure 1 is adopted here. A total of seven channels of acceleration data was recorded simultaneously: Six of them were used to determine the motion at the C.G. and the seventh was used for validation purpose. The results in Figure 2 show a remarkable agreement between the measured and predicted real-time acceleration in the x-direction at location 3 that has been selected arbitrarily as the checking point. Such time-domain data manipulation requires a multi-channel (at least 6-channel) data acquisition capability. However, noting that the form of Eqn. (2) will not be changed by Fourier transformation, a simpler procedure can then be established by using spectral data. This is accomplished by measuring the frequency response function or cross-power between a roving and a fixed transducer. As a result, instead of 6-channels, only two channels are required. Shown in Figure 3 is the spectral comparison of the results previously given in Figure 2. It is clear from Figure 3 that the upper bound of the rigid-body model is at about 1 kHz in this particular problem. Since the rigid-body model is essentially limited to a relatively low frequency range, the roving transducer can be used with a magnetic mount.

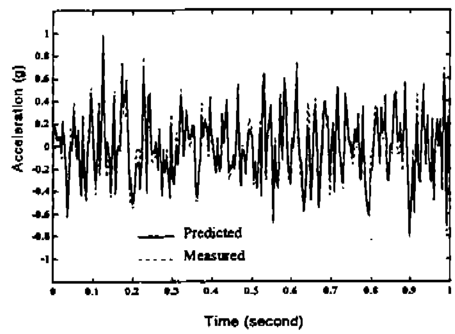

Figure 2. Comparison of accelerations in time domain

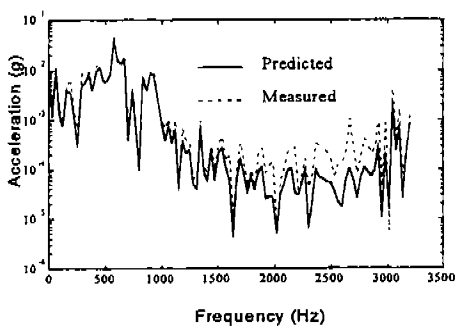

Figure 3. Comparison of accelerations in frequency domain

The above results indicate that, within a certain frequency range, the compressor vibration can be fully specified from only a 6-point vibration measurement. When the six components of motion at the center of gravity of the compressor are known, the vibration at any point such as the connection points to the HVAC unit is readily determined. Therefore, this procedure can be used to reduce the number of measuring points in a vibration monitoring measurement. However, in designing a compressor mounting system a more important question to answer is what defines a good mounting system. It may sound like an easy question. Practically, however, there may not be a simple answer. When determining the effectiveness of a mounting system, one has to know how a compressor responds to the unbalanced forces acting on it. Thus, Eqn. (4) needs to be solved to determine the dynamic characteristics of a mounting system. At this point, one may have already noticed that we have by far only shown some vibration data without (actually with no need of) mentioning the applied forces and the mounting configuration. The data shown in Figures 2 and 3 were collected from a compressor shaker test. In that experimental, the compressor was mounted on three equally spaced isolation mounts in a triangular configuration, represented by nine springs in the model. A shaker was used to apply a random force in the x-direction, 1.5" above the compressor C.G. The mass moments of the compressor were determined experimentally. The responses predicted from Eqn. (4) are compared in Figures (4) and (5) with those measured directly. The model captures the dynamic characteristics of the compressor reasonably well. The discrepancy may be primarily attributed to the fact that the dynamic system is not very "stable" due to extreme softness of the mounts in the radial direction. This softness will have a tendency to cause excessive motion which may lead to severe nonlinearities in the dynamic system. Regarding the assessment of the isolation performance of the mounting system, one of the meaningful variables to be considered is the force transmissibility. From a vibration control point of view, it is desirable that the transmitted forces through the isolation mounts be as small as possible. However, it should be pointed out that this criterion must be specifically related to the frequency range of interest. Therefore, when comparing different mounting designs, one has to first define the interested frequency range based on knowledge of the unbalanced forces. With the responses known at each mounting point (making use of Eqn. (1)), the force transmissibilities can be determined easily. For example, plotted in Figure 6 (solid line) is the x-component of the force transmitted through a single mount. Also shown in Figure 6 are two other curves corresponding to two different mounts in terms of axial stiffness. This plot is to show how easily different mounting designs can be evaluated by using the dynamic system model. In a real-world design, one may have to consider several design criteria simultaneously. Accordingly, it is more advantageous to use this model to understand the dynamics of the compressor system.

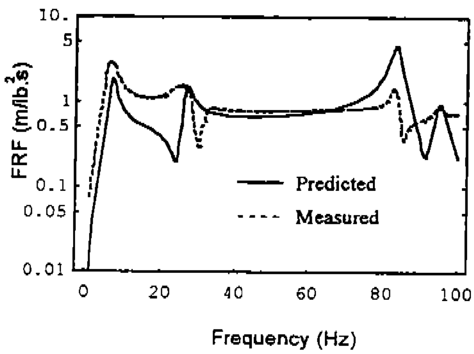

Figure 4. Accelerations in x-direction at location 2

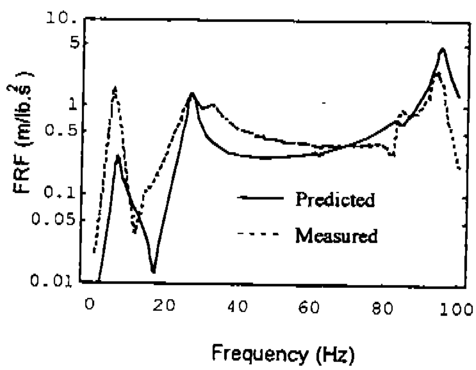

Figure 5. Accelerations in y-direction at location 1

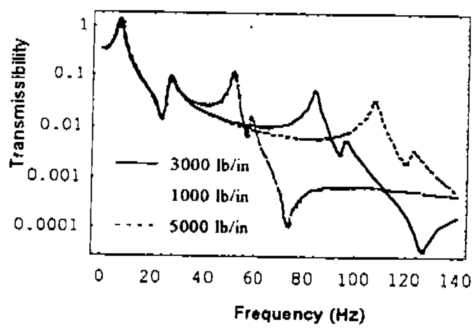

Figure 6. Transmitted force F_x through an isolator

5. CONCLUSIONS

A rigid-body model has been successfully used to completely describe the compressor vibration and hence reduce the burden of collecting massive data in some standardized vibration tests. The validity of this model will not be affected by the types of compressors, mount parameters, locations of attachments, and so on. However, the cut-off frequency should be known if this model is applied to a relatively high frequency range. The dynamic model of a compressor system has been presented. Forces transmissibilities and vibrational responses of the compressor to the unbalanced forces can be readily determined from this model. This simulation capability allows us to examine the effects of a specific design parameter on the isolation performance of a mounting system that would otherwise be difficult to obtain experimentally. With this analytical tool, one can effectively avoid the conventional trial-and-error approach in designing a compressor mounting system.

6. ACKNOWLEDGMENTS

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7. REFERENCES

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