Vibration Isolation for Noise Control in Residential Condensing Units

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VIBRATION ISOLATION FOR NOISE CONTROL IN RESIDENTIAL CONDENSING UNITS

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ABSTRACT

Resonances in the compressor mounting feet and the base pan of the condensing unit can reduce the effectiveness of simple elastomeric isolators resulting in unacceptable noise levels. For this case, the simple single-degree-of-freedom transmissibility model is no longer valid and a multi-degree-of-freedom model should be used. In this paper a case study is presented which illustrates how a three-degree-of-freedom transmissibility model can be used to make design decisions for reducing noise levels in a residential condensing unit when the higher frequency system resonances are important.

INTRODUCTION

The condensing unit represents one of the major sources of noise in a residential HVAC system. A typical condensing unit consists of a set of condensing coils and associated tubing, a sheet metal cabinet and base pan, a fan, and a compressor which is mounted to the base pan using a set of simple elastomeric isolators. The primary emphasis of many programs for reducing condensing unit noise is to reduce the airborne noise from the fan and the compressor. Large reductions in fan and compressor noise can, however, expose structure borne noise problems which might otherwise be masked by the higher background noise levels. The structure borne noise problems can then become the dominant sources in the system and further noise reduction can only be achieved by addressing these sources.

In a typical condensing unit, one of the major sources of structure borne noise is the mechanical vibration transmitted from the compressor to the base pan. The mechanical transmission problem is typically addressed using vibration isolation techniques based on a transmissibility model for a single-degree-of-freedom (SDOF) system. The SDOF model and the associated transmissibility curve are shown in Figure 1. For this case the mass represents the mass of the compressor and the stiffness and damping are associated with the isolator. This method is generally effective for dealing with the low frequency components associated with the running speed but may not be sufficient for dealing with higher frequency components which can be important from a noise control point of view.

When using vibration isolation for noise control in a typical condensing unit, there are two complicating factors which should be considered: internal machine resonances and base flexibility. Internal machine resonances refer to the flexural motion of the compressor mounting feet as opposed to the overall rigid body motion of the compressor and base flexibility refers to the fact that the base pan of the condensing unit is not a rigid structure and is considerably less massive than the compressor itself. For this case, the transmissibility model should be developed using a multi-degree-of-freedom (MDOF) model. The transmissibility curve for a simple three-degree-of-freedom system is shown in Figure 1. In this simple model two additional degrees-of-freedom are added; one to represent the dynamics of the compressor mounting feet and one to represent the dynamics of the base pan. As illustrated in Figure 1, the added degrees of freedom introduce additional peaks in the transmissibility curve which correspond to the higher resonant frequencies of the combined system. At these frequencies, the transmissibility will be much higher than predicted using a simple SDOF model. Although the transmissibility is less than one, the peaks are still important because the sheet metal base pan is, in general, a very good radiator of noise and only a small amount of mechanical power is required to generate a large amount of sound power. The decrease in performance of the isolation system and the subsequent increase in noise levels will be more pronounced when the dominant excitation frequencies from the compressor, the internal machine resonances, and the resonances associated with the base flexibility are all close to one another.

From a quantitative point of view, the use of a MDOF model for the analysis and design of an isolation system can be difficult because there are usually a large number of resonances associated with the mounting feet and base pan and so the transmissibility curve will be more complicated than the one shown in Figure 1. The simple MDOF
transmissibility model can, however, be used to both gain a better understanding of the use of vibration isolation for noise control and as a basis for making better design decisions. In this paper, a case study is presented which illustrates how a better understanding of vibration isolation can be used to reduce noise levels in a residential condensing unit.

**PROBLEM DESCRIPTION**

A manufacturer of residential condensing units was involved in a program to reduce the overall noise levels in one of the product lines. A quieter fan was installed in the unit and the compressor was mounted in an acoustically lined enclosure. The objective was to have the overall sound power governed by the sound power of the fan. A sound level rating was established for the product in accordance with the American Refrigeration Institute's (ARI) Standard 270. When the units were audited per the ARI standard, a peak was observed in the 800 Hz 1/3 octave band as illustrated in Figure 2. The high 800 Hz 1/3 octave band, which was 6 dB above the adjacent bands, resulted in an unacceptable overall sound level.

To diagnose the source of the problem it is necessary to identify the primary noise sources for a typical condensing unit. These sources are summarized in Table 1. The sources can be separated into two classes: component noise sources and system interaction noise sources. A component noise source refers to the airborne noise from a component which would be present even if the component was operated outside of the assembled unit. A system interaction noise source refers to a source which arises primarily as the result of an interaction between the various system components.

Table 1. Primary Noise Sources in a Typical Condensing Unit

<table>
<thead>
<tr>
<th>Component Noise Sources</th>
<th>System Interaction Sources</th>
</tr>
</thead>
<tbody>
<tr>
<td>- airborne noise from the fan.</td>
<td>- structure borne noise transmitted from the fan through the fan mounting points.</td>
</tr>
<tr>
<td>- airborne noise from the compressor.</td>
<td>- fluid/structure interaction noise associated with fan flow field.</td>
</tr>
<tr>
<td></td>
<td>- fluid/structure interaction noise associated with refrigerant flow from the compressor.</td>
</tr>
<tr>
<td></td>
<td>- structure borne noise transmitted from the compressor through the tubing.</td>
</tr>
<tr>
<td></td>
<td>- structure borne noise transmitted from the compressor through the mounting points.</td>
</tr>
</tbody>
</table>

Based on an understanding of the primary noise sources and the fact that the compressor is mounted in an acoustically lined enclosure, the problem is most likely related to either the fan or one of the compressor related system interaction sources. A measurement of the intensity distribution of the 800 Hz 1/3 octave band noise showed that the noise was radiated almost exclusively from the base pan of the unit. This would suggest that the noise problem is probably not related to the fan but is structure borne in nature where the vibration energy is transmitted from the compressor to the base pan through the compressor mounting points with the base pan in turn radiating the noise.

A diagnostic procedure was developed to test the structure borne hypothesis. In step (1) the sound power of the unit was measured with only the fan running. In step (2) the sound power of the unit was measured with only the compressor running. To differentiate between the different compressor related interaction sources, the condenser coils were bypassed and the compressor was run using a simple run around stand. In this configuration the compressor is connected to the condensing unit only at its mounting points. In step (3), the compressor is removed from the unit and tested alone. The compressor operating conditions for steps (2) and (3) are the same as for the total unit operating at the standard 95° ambient ARI rating point. The measured sound power levels for the different steps are shown in Figure 3. A comparison of Figures 2 and 3 clearly shows that the peak in the 800 Hz 1/3 octave band is the result of structure borne noise transmitted from the compressor to the base pan through the compressor mounting points.

For this particular unit, the 800 Hz 1/3 octave band measured in step (2) is about 10 dB greater than that measured in step (3). In other words, the structure borne noise, which is radiated from the base pan, is significantly higher.
greater than the airborne noise which is radiated from the compressor itself. This is important to note because the structure borne noise is what the manufacturer of the condensing unit sees while the airborne noise is what the compressor manufacturer sees.

**PROBLEM SOLUTION**

Based on the earlier discussion, the key parameters for the structure borne vibration problem are the compressor related excitation frequencies, the fundamental resonant frequency of the isolation system, the resonant frequencies of the compressor feet, and the resonant frequencies of the base pan. Critical system frequencies, which were obtained using experimental modal analysis techniques, are listed below.

1. The primary excitation from the compressor occurs at the running speed which is approximately 60 Hz. A secondary excitation frequency, which is related to the specific design and principal of operation of the compressor, occurs at about 800 Hz.

2. The fundamental frequency of the isolation system occurs at about 20 Hz.

3. The first resonant frequency of the compressor mounting feet occurs at approximately 825 Hz. At this frequency the foot bends as a simple cantilevered beam.

4. The base pan has a resonant frequency at about 800 Hz. The regions of maximum displacement at this frequency correlate well with the regions of maximum sound intensity.

From the data listed above, it can be concluded that the isolators are well designed relative to the running speed but in the 800 Hz 1/3 octave band a coincidence problem exists which significantly reduces the effectiveness of the isolation system. There are three general approaches which can be used to improve the high frequency isolation characteristics of the system: reduce the overall transmissibility by reducing the stiffness of the isolator, reduce the transmissibility near 800 Hz by adding damping, or reduce the transmissibility near 800 Hz by eliminating the coincident frequencies. Here the simple three-degree-of-freedom transmissibility model can be used to help assess the impact of changes in the system parameters on the overall transmissibility.

Transmissibility is defined as the ratio of the applied force $F_C$ (excitation force from compressor) to the force transmitted through the isolator to the base pan, $F_T$. The transmitted force $F_T$ is given by the difference in the displacements of the compressor foot $X_2$ and the base pan $X_3$ times the stiffness and damping of the isolator. Solving the system of equations for the model in Figure 1 yields the following equation for the transmissibility:

$$TM = \frac{k_1k_2k_3 - k_1k_2m_3\omega^2}{[k_1k_2k_3 - (k_1k_2m_3 + k_1k_2m_2 + k_1k_3m_2 + k_2k_3m_1 + k_1k_2m_1 + k_1k_3m_1)\omega^2] + (k_1m_1m_3 + k_1m_2m_3 + k_2m_1m_3 + k_2m_2m_1 + k_3m_1m_2)\omega^4 - m_1m_2m_3\omega^6}$$

where $k_n = k_n(1 + C_n)$ \( n = 1, 2, 3 \)

Here $m_1$ is the mass of the compressor, $m_2$ the modal mass for the first mode of the compressor foot, and $m_3$ is the modal mass for the 800 Hz mode of the base pan. $K_1$, $K_2$, $K_3$, $C_1$, $C_2$, and $C_3$ are the stiffnesses and loss factors for the first mode of the compressor foot, the isolator, and the 800 Hz mode of the base pan respectively.

Approximate values for the system parameters were obtained from the experimental data. Because of the approximate nature of the model, only relative comparisons can be made and the model is only valid at frequencies near 800 Hz. A summary of the effect of various design changes on the transmissibility is given in Table 2.
Table 2 Summary of Effect of Design Changes on Transmissibility

<table>
<thead>
<tr>
<th>Change</th>
<th>Reduction in Transmissibility (dB)</th>
</tr>
</thead>
<tbody>
<tr>
<td>by factor of 2</td>
<td>by factor of 4</td>
</tr>
<tr>
<td>Reduce Isolator Stiffness</td>
<td>1.5</td>
</tr>
<tr>
<td>Increase Isolator Damping</td>
<td>7.0</td>
</tr>
<tr>
<td>Increase Damping in Foot Alone</td>
<td>2.5</td>
</tr>
<tr>
<td>Increase Damping in Base Pan</td>
<td>&lt; 1.0</td>
</tr>
<tr>
<td>Increase Foot Stiffness</td>
<td>15.0</td>
</tr>
<tr>
<td>Increase Base Pan Stiffness</td>
<td>2.0</td>
</tr>
<tr>
<td></td>
<td></td>
</tr>
</tbody>
</table>

The simplest solution would be to modify only the isolators. The major problem with this solution is that it is difficult to achieve either the required stiffness or damping at a reasonable cost using the current elastomeric isolator design. In addition, both modifications result in a large increase in the static deflection which then requires a tubing design change as well as the need to control excessive deflections during shipping.

Any other solution will involve a change to either the compressor feet or the condenser base pan. From the table, the most effective solution would be one that detunes the coincident frequencies. Given the basic characteristic of the transmissibility curve and the results listed in Table 2, this is best accomplished by moving the first resonant frequency of the compressor feet to a higher frequency. Because of the simple nature of the first resonant mode of the compressor feet, this modification is relatively easy compared to a modification to the base pan. In addition, the general trend in the industry is to reduce the overall noise levels in the condensing unit, so a similar structure borne problem could occur in other units in the future. Therefore, the optimal solution would be a modification to the compressor feet.

Finite element analysis techniques were used in conjunction with experimental modal analysis to redesign the compressor feet. The objective was to move the first natural frequency to a frequency above the 800 Hz 1/3 octave band without adversely affecting current manufacturing/assembly processes. In addition, the modified design had to represent a drop in replacement for the manufacturer of the condensing unit. In the final design, the higher frequency was achieved by increasing the stiffness at the base of the foot and decreasing the mass at the end of the foot. The original and modified designs are illustrated in Figure 4. The effectiveness of the new design is illustrated in Figure 5 where the sound power levels for the two different designs are compared. Structure borne noise in the 800 Hz 1/3 octave band was reduced by 10 dB so that the resulting sound power level was governed by the fan.

SUMMARY AND CONCLUSIONS

This study illustrates the need for a better understanding of structure borne noise and the methods for controlling it. It also points out that more sophisticated analytical and experimental tools may be necessary to achieve an optimal solution. In this case acceptable sound power levels were obtained by modifying only the compressor. If the sound levels were to be reduced further, it would probably be necessary to also modify either the isolators or the base pan. In general, as the overall levels of the component noise sources are decreased, the importance of the interaction sources will increase. This is significant because the interaction noise sources may not be apparent to the component manufacturer as they are system problems. Also, because of the large number of possible system designs, it is usually not possible for the component manufacturer to account for all possible interaction effects. The interaction effects are most effectively dealt with by the system manufacturer in cooperation with the component manufacturer. Therefore, it is important for both parties to share basic design philosophies and design goals so that adverse interaction effects can be avoided where possible and timely and cost effective solutions can be developed when problems do occur.

REFERENCES


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Figure 1. Single and Multi Degree-of-Freedom Transmissibility Models

Figure 2. Illustration of Sound Problem in 800 Hz 1/3 Octave Band
Figure 3. Summary of Sound Power Spectra From Diagnostic Tests

a) Original Design  

b) Modified Design

Figure 4. Original and Modified Design of Compressor Feet

Figure 5. Sound Power of Total Unit with Original and Improved Foot Design