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The Effect of the Dome Shape of a Hermetic Compressor Housing on Sound Radiation

by

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ABSTRACT

The sound level of air conditioning units is becoming increasingly more important to the OEM unit manufacturer and the end user of the product. Invariably, the compressor is a significant contributor to the overall sound. This paper will discuss the analysis techniques and ultimate results of redesigning a hermetic compressor housing with the goal of lowering the sound level.

The redesign process consisted of initial finite element analysis to predict resonance frequencies, construction of prototypes, and verification by sound testing and experimental modal analysis. The experimental work showed that the first resonance of the dome increased from 1380 to 2370 Hz. The overall sound pressure level of the compressor was reduced by as much as 2.5 dBA. Even though the finite element analysis did not agree with the experimental modal results, it was a valuable tool for visualizing the benefits of an improved housing shape.

INTRODUCTION

Since the compressor housing is the ultimate radiator of sound, it follows that the shape of the housing is a very important factor in reducing the sound level. The vibration of the housing surface is of course the source of all the sound heard by the listener. The object in designing a compressor housing would then be to make it resistant to excitation. By definition, a stiff structure deflects less per unit force. It follows that the surface velocities are also lower, which translates into less intense sound radiating from the surface. Dynamic forces are transmitted through the suspension springs and the discharge shock loop. These forces are also transmitted in the form of pressure fluctuations acting over the inner surface of the housing. In general, the magnitudes of these forces and pressures decrease as the frequency increases. Also, the housing radiates sound most effectively at frequencies at which it is resonating. Thus, in designing a stiff housing, the resonance frequencies are shifted upwards away from the high amplitude forces.

This paper is written to illustrate some of the techniques available to the engineer for determining the effect of dome shape and to aid in designing a compressor housing for reduced sound. Results of finite element modeling conducted on a personal computer will be presented. To verify the results of the mathematical model, simplified experimental modal analyses (EMA) were run, concentrating on the top surface. Finally, sound testing of the compressors in the new housing is presented to show the ultimate benefit of the new housing design in comparison with the original design.
Finite Element Modeling of Housing Dome Shapes

First, a finite element model (FEM) was constructed of the original compressor housing. The top surface has a spherical radius of 17 inches. To predict the increase in stiffness that would be achieved, a proposed housing shape was also modeled in which the radius of the upper housing was reduced to about 3.25 inches. Also, the dome was modeled to blend smoothly into the sides of the housing, without an abrupt bend or sharp radius. These models are shown in Fig. 1. The final design of the improved housing top had a spherical radius of 5.5 inches and the top did not blend as smoothly into the sides of the housing. This was a compromise solution which limited the height of the compressor.

The model of the original housing predicted the first resonance frequency which consisted of predominately top motion at 1978 Hz, the only one in the first 18 modes. The deflected shape is shown in Fig. 2. The model of the domed-top housing did not predict a resonance with significant top motion in the first 18 modes. The PC-based analysis was limited by available hard disk space and RAM memory.

To reduce the size of the model and attempt to concentrate on the vibration of the top of the housing, the finite element models shown in Fig. 3 were used. The nodes along the boundary where the top was severed from the housing were constrained against translation in all directions. Realizing that the smaller models do not accurately model the entire housing, only vibration modes which showed no rotation at the nodes along the constrained boundary were considered. The first two resonance modes for the original housing top are shown in Fig. 4, occurring at 1973 Hz and 2266 Hz. Notice that the full model and the reduced model agree on the first resonance. The agreement on the second mode would have to be verified through further analysis of the full model to higher frequencies (after the computing deficiencies are resolved). The first two modes for the domed-top model were predicted at 4138 Hz and 4410 Hz. These vibration shapes are shown in Fig. 5.

As will be seen in the following section, the finite element models did not agree well with the experimental modal testing. It is obvious that the domed-top model does not accurately depict the actual design which was modal and sound tested. Further study of all of the factors involved, such as variations in material thickness and dimensional differences between the model and the actual structure, will be required to make the models more useful and informative tools.

Experimental Modal Analysis of Housing Domes

Normally, an experimental modal analysis of a structure involves marking the shell with many points at which the acceleration would be measured while it is being driven at fixed location. The denser the measurement points the higher the frequency of the mode shape that can be confidently estimated. However, the higher the density, the greater the amount of data in the form of frequency response
functions (FRF) which must be processed. This procedure is very time-consuming and computer-intensive. On the other hand, there are new laser-based methods for measuring the surface velocities of a structure and determining mode shapes, reducing the time for a modal test. However, these systems are very cost prohibitive.

A much simpler setup can be used to determine the first two or three resonance frequencies of the dome without the expense of a modal analysis software package. With an impulse hammer to excite the shell, an accelerometer to measure the response and a 2-channel narrow-band frequency analyzer to calculate the frequency response functions, a quick understanding of the vibration shapes of the shell can be obtained.

For this study, seven points were marked on the top of the housings as shown in Fig. 6. They were chosen in such a way that the first three resonance mode shapes of the top could be determined. These modes are shown schematically in Fig. 7, along with their corresponding resonance frequencies. The "+" sign in the first mode shape means that all points around the surface would be moving together, in phase with one another. In the second and third modes, points on opposite sides of the dashed lines would be moving out of phase with one another. Points lying along the dashed line would be essentially stationary if the structure is driven at the frequency corresponding to that shape.

The transfer function and phase plots shown in Figs. 8 and 9 pertain to the original housing top. Figure 8 refers to two points on the same side of the major axis and Fig. 9 refers to points positioned diagonally on opposite sides of the major axis. In all cases, the acceleration was measured at point A. The plots have the frequencies marked which have been identified as the first three resonance frequencies which show considerable top motion. The peak around 1000 Hz appears to be a mode which consists of mostly side motion, so it will be excluded from the discussion.

Notice in Figs. 8 and 9 and that the phase relationship is 90° at 1380 Hz, as it was for all 7 points, which indicates that all points are moving together. At 2070 Hz, the phase between points A and E is 90°. Between points A and B, the phase is 180°. Points A and E are moving in phase and points A and B are moving out of phase at this frequency. Further, at 2360 Hz, the phase between points A and E is 180° and between points A and B it is also 180°. Observing this, and the other frequency response functions between the other points in Fig. 6, we can conclude that the first three modes of the top occur at about 1380, 2070 and 2360 Hz, respectively, taking on the shapes on Fig. 7. Notice also that the magnitude of these resonance peaks is about $10^{-20}$ g/lb.

The same observations can be made on the transfer functions and phase plots for the improved housing top shown in Figs. 10 and 11. However, I suspect that the shape of the modes in the domed top do not conform as precisely to the shapes in Fig. 7 as does the flatter top of the original housing. Perhaps a more important observation is that the noticeable shift in the resonance frequencies, but the response magnitude below 2000 Hz is now less than 1 g/lb. Also, the magnitude of the resonance peaks does not exceed 10 g/lb. This means that more force is required to achieve the same surface velocity, which results in lower sound radiation.
Sound Comparison

The ultimate goal of this study is to reduce the sound level of the compressor. An example of the resulting sound data is shown in Fig. 12. The most obvious effect of the new housing shape is the reduction in sound energy in the 1250 and 1600 Hz bands, which were reduced by 8 dB and 3 dB. This resulted in an overall sound level reduction of 1.4 dBA in this application. In some tests with this new housing design, as much as 2.5 dBA reduction in sound pressure level was achieved.

CONCLUSIONS

Finite element models of compressor housings were constructed and analyzed. To reduce the size and complexity of the models, the top portions were removed from the housings and modeled independently. The full and reduced models agreed very well on the frequency of the first resonance mode of the original housing. As an additional illustration of the analytical methods available, experimental modal analyses were run on the original housing and the final design of the new housing with the improved top. The experimental work did not agree well with the FEA, but it did show that the actual resonance frequencies would be shifted upward at least 1000 Hz. This conclusion is supported by the sound testing where as much as 2.5 dBA reduction was realized.
Fig. 1. Finite element models of a) original and b) improved-dome housings.

Fig. 2. Exaggerated deformed shape of the first resonance mode of the original housing which shows top motion (1978 Hz).
Fig. 3a. Reduced finite element model of original housing top.

Fig. 3b. Reduced finite element model of improved housing top.
Fig. 4a. Deformed shape of the first resonance mode of the reduced model of the original housing top (1973 Hz).

Fig. 4b. Deformed shape of the second resonance mode of the reduced model of the original housing top (2266 Hz).
Fig. 5a. Deformed shape of the first resonance mode of the reduced model of the improved housing top (4138 Hz).

Fig. 5b. Deformed shape of the second resonance mode of the reduced model of the improved housing top (4410 Hz).
Fig. 6. Measurement points for experimental modal analysis of the housing tops.

Fig. 7. Schematic representations of the first three resonance modes of the housing tops.
Fig. 8. Frequency response function between force at point E and acceleration at point A on the original housing top.

Fig. 9. Frequency response function between force at point B and acceleration at point A on the original housing top.
Fig. 10. Frequency response function between force at point E and acceleration at point A on the improved housing top.

Fig. 11. Frequency response function between force at point D and acceleration at point A on the improved housing top.
Fig. 12. A 1/3-octave sound spectrum showing the results of a sound test comparing the original housing and the improved housing.