1992

Intensity Measurements and Radiated Noise Reduction for a Freon Compressor

J. P. Smith
Virginia Polytechnic Institute and State University

D. H. Kiel
Virginia Polytechnic Institute and State University

C. J. Hurst
Virginia Polytechnic Institute and State University

INTENSITY MEASUREMENTS AND RADIATED NOISE REDUCTION FOR A FREON COMPRESSOR

J. P. Smith, D. H. Kiel, and C. J. Hurst

Department of Mechanical Engineering
Virginia Polytechnic Institute & State University
Blacksburg, VA 24061-0238

ABSTRACT

A set of acoustic intensity measurements were taken over the surface of an operating and loaded Freon compressor. The results showed clear evidence of standing waves in the motion of the sides of the compressor shell at some frequencies. At other frequencies the radiation pattern was less well organized. The results also showed that the top surface of the compressor shell was radiating a substantial amount of noise in some frequency bands. This top surface was very nearly flat, and was connected to the compressor frame by a mounting spring on the surface centerline. Finite element studies of the shell indicated that going to a curved (and therefore stiffer) top should make it less responsive to the spring driving forces. A shell was modified and tested. The results showed a clear reduction in the intensity radiated from the top surface. Experimental results from this investigation will be shown and discussed.

INTRODUCTION

An extensive series of intensity measurements were made on an operating freon compressor as part of an attempt to identify noise paths and to reduce radiated compressor noise.

The measurements were part of a program which included attempts at vibration and noise prediction. The results of these analyses were verified experimentally. For verification purposes, however, the compressor shell was driven with a shaker at a single frequency. Those results will be reported in later papers.

The measurements on the operating compressor were made very close to the shell in hopes of finding areas of high noise radiation. It was expected that the areas of the compressor shell very near the suspension points for the isolated compressor would be very active, and would radiate a great deal of sound. What was found, instead, was that the shell was strongly driven in several of its normal modes. Most of the surface of the can was important in radiating sound. The relatively flat top was a particularly strong radiator.

This paper will begin with a description of the compressor and of the measurement arrangements. Some measurement results on the sides and the top of the compressor shell will be shown and discussed. A proposed simple modification to the shape of the compressor shell reduced the radiated noise levels, and these results will be shown as well. Finally, some observations and lessons learned from this exercise will be discussed.

THE COMPRESSOR

The compressor under test in this investigation was a 5 ton freon compressor hermetically sealed in a shell which was about 10 inches (25 cm) in diameter and 15 inches (38 cm) tall. The horizontal cross-section of the shell was nearly elliptical, while the top and bottom surfaces were nearly flat. The shell was of deep drawn steel, about 0.11 inches (0.27 cm) thick.

Contained within this shell was a two cylinder reciprocating compressor driven by an electric motor.

The compressor was isolated from the shell through three steel springs. Two of these were located low on the sides of the shell and the third was located near the center of the top of the shell. Each lower spring sat on a bracket cantilevered out from about the center of the longer side of the elliptical cross-section.

INTENSITY MEASUREMENT RESULTS ON RUNNING COMPRESSOR

Measurement procedure

Acoustic intensity data were obtained for compressors operating under a load provided by a heat exchanger which maintained the desired operating conditions.

Experimental sound intensity data were obtained at points on a 1 inch (2.54 cm) grid which enclosed the top and sides of the compressor. To allow easier visualization of the intensity radiating from the entire compressor surface the data points over the top of the compressor were examined and plotted independently from the data points on the sides. The curved grid surface over the sides of the compressor was cut along one side and developed to produce a plot for the side intensity data. Figure 1 shows a view of the compressor and the developed measurement grids for the top and sides of the compressor.

The data were obtained with a Bruel & Kjaer type 2520 sound intensity probe and Type 2034 Signal Analyzer. The intensity probe consists of two 0.5 inch (1.27 cm) microphones mounted face to face with a solid spacer in between. The size of the spacer dictated the measurable frequency range. A spacer of .47 inches (12 mm) was used to obtain intensity data between 250 Hz and about 6 kHz. However, most of the intensity data were gathered at frequencies up to 1600 Hz in order to maximize the resolution of the analyzer in the range where sound pressure level data showed that the severest noise problems existed.

The process of obtaining the sound intensity data consisted of holding the intensity probe so that the midpoint of the spacer was 1.6 inches (4.1 cm) from the surface of the operating compressor for each point on the measurement grid. The probe was oriented such that the axis of the probe (along the centerline of the microphones) was perpendicular to the surface of the compressor at each point. The data at each point was averaged in the frequency domain with the signal analyzer and transferred to a PC for storage. Thus an intensity spectrum was obtained for each measurement point over the surface of the compressor.

A typical intensity spectrum measured at a point directly above the top center of the compressor is shown in Fig. 2. The harmonic nature of the forcing function is evident from the intensity peaks located 57 Hz apart, which was the running speed of the compressor. The plot also shows a large peak at 969 Hz. This was a frequency of considerable interest which was found to radiate strongly from the top of the compressor.

Once the measurements were complete a stored intensity spectrum existed for each measurement point. For a given frequency of interest the intensity could then be extracted from the stored intensity spectrum data sets at each point of the measurement grid. This extraction process yielded intensity information at a given frequency for each point of the measurement grid. Plots of this data revealed much about the sound field close to the surface of the compressor and the radiation from each part of the shell.

Intensity data were obtained at each measurement point for both the original (flat topped) compressor and for a modified (domed top) compressor so that the intensity radiation patterns and levels could be compared and the noise reduction resulting from the doming of the compressor top could be determined.

Choices of frequencies to study

Three-dimensional intensity plots over the grid surface at particular frequencies allowed interpretation of the sound field close to the compressor at these frequencies. The sound radiation pattern from each part of the shell surface at the chosen frequencies could be examined to determine the frequency content radiated by each part of the shell. The frequencies chosen to study were frequencies consistently corresponding to peaks in the intensity spectra and also modal frequencies discovered via an impact hammer test. These modes were then correlated with those determined using a finite element program created in ANSYS. Many of the severe noise problems existed at frequencies where the forcing function peaks coincided with the natural mode shapes of the compressor.

Measurements on sides

Many of the sound intensity data plots generated over the sides of the compressor revealed well-defined intensity patterns related to the normal modes of vibration of the compressor shell as determined by a finite-element model. Figure 3 shows a three-dimensional plot of the positive intensities over the sides of the compressor at a frequency of 336 Hz. This plot displays a distinct radiation pattern corresponding to a normal mode at this frequency which coincided with a mode determined by the finite-element model. Figure 4 shows a similarly defined intensity pattern at 854 Hz, corresponding to a higher mode. The lines parallel to the vertical axis of the compressor with low (often negative) intensity levels correspond to nodal lines in the vibration pattern on the surface of the compressor at that frequency.

Figure 5 contains a three-dimensional plot over the sides of the domed-top compressor at 2776 Hz. At higher frequencies, the intensity pattern was much more random as the modes of the shell were more complex and the wavelengths were shorter.

Measurements on top

Figure 6 contains an intensity plot over the top surface of the flat-topped compressor at 969 Hz, which was determined to be one of the frequencies with extremely high sound intensity radiation. Note the uniform radiation pattern. The finite element model determined that the motion of the compressor top at this frequency was similar to that of the fundamental mode of a membrane fixed at its edges, resulting in high sound intensity radiation with a uniform pattern. Thus, there was good correlation between the dynamics predicted with the finite element model and the experimental radiated acoustic intensity.

MODIFICATIONS TO REDUCE TOP RADIATION

In order to reduce the sound radiation from the compressor the top of the compressor was domed using hydrostatic pressure (with the sides constrained). It was expected that this modification would decrease the radiated intensity from the top of the shell by pushing the modal frequencies away from the excitation frequencies.

Figure 7 shows the intensity radiating from the top of the original compressor at a frequency of 1332 Hz. At this frequency the intensity level was 10 dB below that of the flat-topped compressor (shown in Fig. 8).

It turned out that stiffening the shell not only decreased the intensity radiating from the top surface as expected, this change also significantly decreased the noise emitted from the side surfaces. If the side intensity radiation is compared with the original flat-top compressor and new domed-top compressor (see Figs. 3 and 9), there is again a 10 dB drop in radiated intensity.

CONCLUSIONS

The first conclusion reached in this investigation was that the whole shell is important in the acoustic radiation process. Contrary to what might be expected, the areas of intense radiation were not located only near the isolator attachment points. Acoustic radiation patterns consistent with strong normal mode vibration patterns were observed.

Secondly, it was demonstrated that stiffening the shell by eliminating the flat top surface resulted in a quieter compressor. This raised the normal modal frequencies of the top, pushing them away from the exciting frequencies.

ACKNOWLEDGEMENTS

This work was supported by Bristol Compressors, Inc., as part of an investigation including modal analysis, finite element analysis and acoustic prediction work.

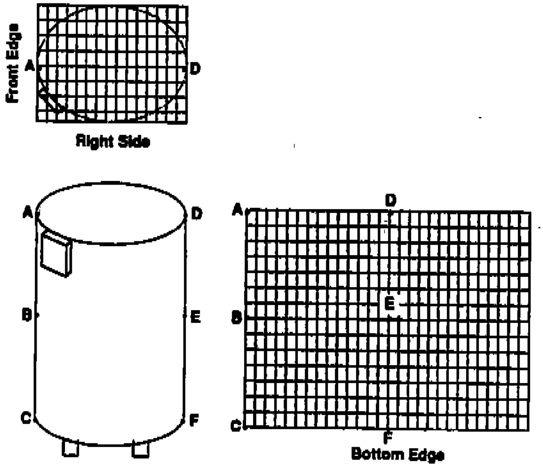

Fig. 1. Sketch of compressor showing top and developed side measurement grids.

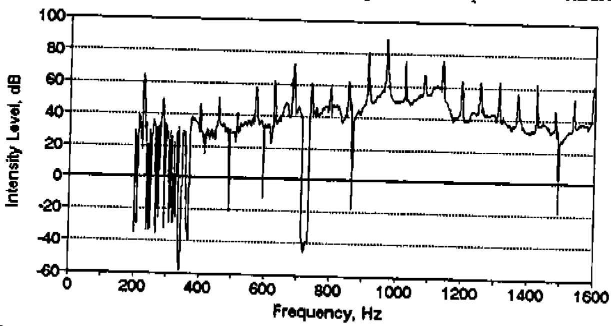

Fig. 2. Intensity spectrum measured at top center of unmodified compressor.

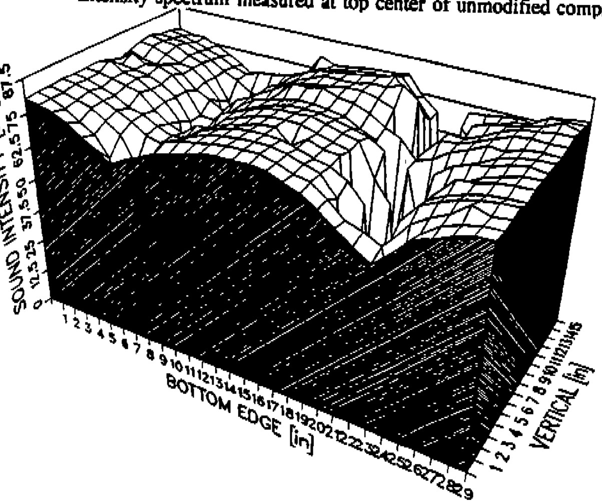

Fig. 3. Intensity plot over the side surface of the unmodified compressor at 336 Hz.

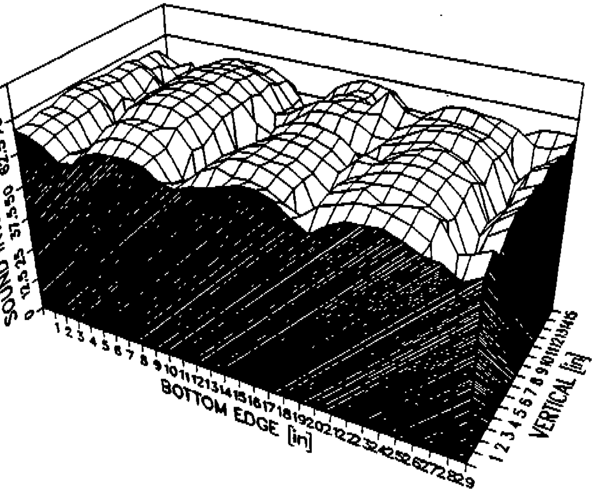

Fig. 4. Intensity plot over the side surface of the unmodified compressor at 854 Hz.

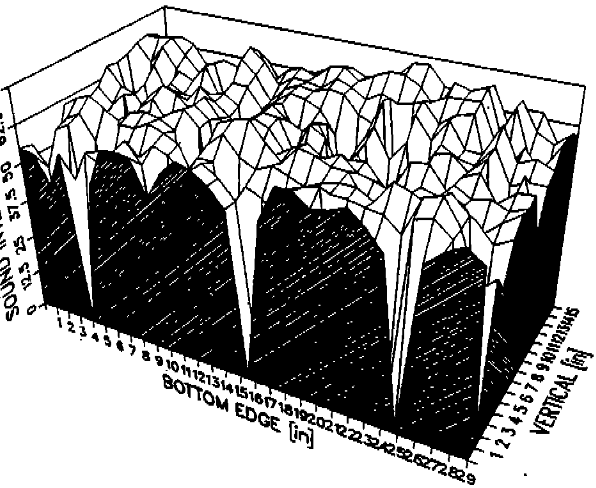

Fig. 5. Intensity plot over the side surface of the modified compressor at 2776 Hz.

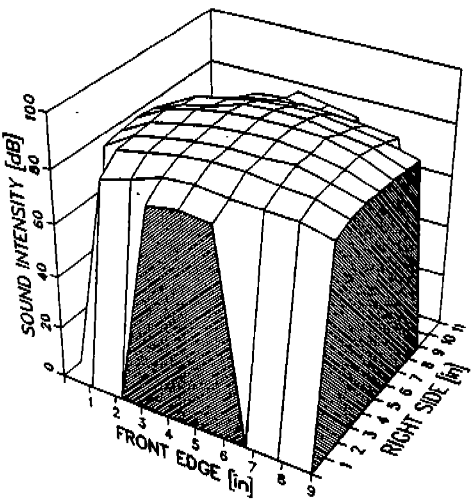

Fig. 6. Intensity plot over the top surface of the unmodified compressor at 969 Hz.

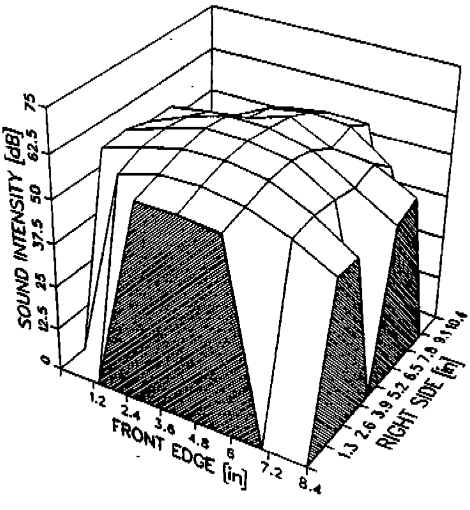

Fig. 7. Intensity plot over the top surface of the unmodified compressor at 1332 Hz.

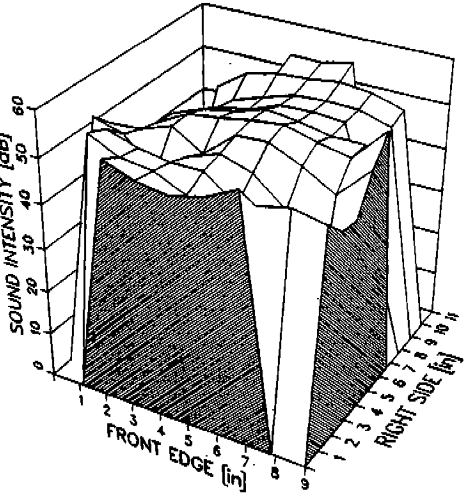

Fig. 8. Intensity plot over the top surface of the modified compressor at 1332 Hz.

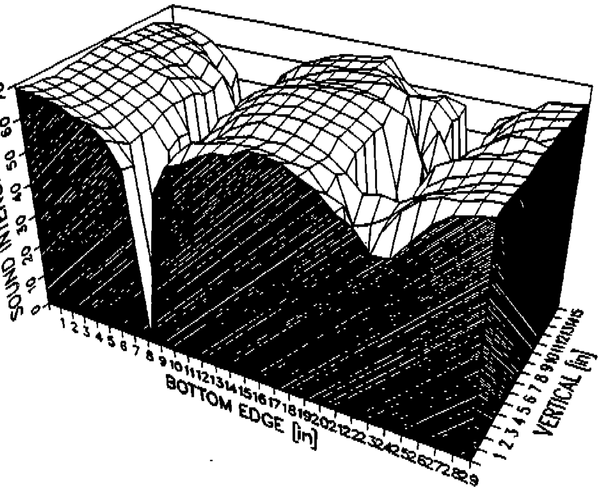

Fig. 9. Intensity plot over the side surface of the modified compressor at 344 Hz.