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DESIGN TECHNIQUES AND RESULTING STRUCTURAL MODIFICATIONS USED TO REDUCE HERMETIC COMPRESSOR NOISE

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ABSTRACT

This case history explains the methodology and results of a design process used to reduce noise in hermetically sealed scroll compressors through upper shell design. Noise sources were identified and localized through the use of acoustic intensity and time domain analysis. It was determined that a large contributor to overall compressor noise was the upper shell/discharge plenum assembly. Modal analysis of the upper shell revealed that its fundamental modes of vibration corresponded with frequencies of high noise energy content. The upper shell was modeled using finite element methods. The model was fine-tuned to match the results of experimental modal analysis. The finite element model was then used to obtain a stiffer upper shell design, thereby raising the resonant modes of the structure and reducing the dissipation of energy through sound emission. New top shells were fabricated and tested. A comparison between the old and new upper shells showed a decrease in sound energy in upper shell resonant modes, and an overall noise reduction.

INTRODUCTION

Scroll compressors continue to replace reciprocating technology in residential and small commercial air conditioning applications. Reduced part counts and promises of higher reliability, efficiency and performance are driving factors in this change.

As sound levels become a major measure of product quality, compressor manufacturers intensify efforts to reduce compressor noise. This effort to lower compressor sound level sparks new emphasis on design analysis and evaluation methods used for a successful noise reduction program. This paper presents the methodology and results of a design analysis and evaluation directed at reducing noise on an HVAC scroll compressor, of the type shown in Figure 1, by the use of a number of empirical and analytical techniques.

PRELIMINARY INVESTIGATION AND PROBLEM IDENTIFICATION

Preliminary testing of the Carrier SC and SR type scroll compressors revealed broad band noise predominant in the 0.5 kHz to 8.0 kHz third octave bands (see Figure 2: 1/3rd Octave Band Data for SC). An overall reduction of around 3 dBA was desired.

Acoustic intensity and filtered microphone sweeps indicated that a major source of radiated noise was the compressor top shell in the 2 kHz to 5 kHz frequency range. Frequency response measurements on the top shell also indicated that this portion of the compressor resonated in the 2 kHz to 5 kHz frequency range when dynamically excited by an impulse hammer.

While the top shell was identified as a prime radiator of acoustic energy, the source of this energy was still unknown. A test used effectively in the past [1] compares the time-domain near field and accelerometer data to a crankshaft position indicator signal from a proximity probe inside the compressor. The variation in magnitude of the filtered time domain signal may reveal whether the source is event-based or steady state.

Event-based sources, such as forced pressure pulsations or mechanical impact, will be revealed by a large and regular variation in the time domain signal over the course of a single shaft revolution. Comparison of the signal with the crankshaft position indicator will reveal the relative position of mechanical components at the time of the event. This will serve as a guide to possible sources. Steady state sources, such as uniform flow-induced noise and mechanical rubbing action, will result in a more steady time domain signal.

The time domain signal was found to vary over the course of a shaft revolution, a sign of an event-based source. The shell surface and the near field were most active for short periods of time during a single crankshaft revolution (see Figure 3: Sample Time Domain Data). The timing of the greatest of these impulses corresponded with the beginning of the discharge porting process.

A working hypothesis was that gas pulsations from the discharge process were exciting mechanical resonances of the upper shell. Variation in mechanical loading between the scroll vane walls at the discharge porting point was also a candidate energy source. While the frequency content of an impulse energy source may vary depending on the intensity and duration of the impulse, the overall content generally falls off with increasing frequency. Increasing the stiffness of the structure will shift its resonance frequency into a zone where there is proportionally less impulse energy to excite it.

While portions of the investigation were directed to investigating methods to verify and reduce the possible impulse-based sources, a large part of the effort shifted to consider the top shell design.

MODAL AND FINITE ELEMENT ANALYSES

The original top shell (shown in Figure 11 and which will be referred to as the "old design") was analyzed with impact response (inertance) type experimental modal analysis to determine its resonance mode shapes and frequencies. The results of this analysis are shown in Figs. 4, 5 and 6. This information was used to confirm the hypothesis that the top shell resonance modes corresponded to the high noise frequency regions shown in Figure 2, and also as input used to tune and validate the accuracy of a finite element based model. Figures 7 and 8 illustrate the modal predictions of the model. The table below gives a brief comparison of the results for the lower modes of the older design:

Old Shell Design Modal Frequencies

Mode	1	2
Experimental	3.25 kHz	4.75 kHz
Analytical	3.58 kHz	4.95 kHz

With reasonable verification of the analytical model, it can be used to predict the effect of design changes on modal frequencies and shapes. The objective is simply to raise the modal frequencies of the top shell to as high a level as possible, away from the region of peak energy content of the impulse source. Secondary objectives include no increased use of material, no net increase in overall size, compatibility with the connecting structural elements, and provision of sufficient pulse volume to damp out the low frequency (58 Hz) discharge pulsation.

The design that would allow for maximized shell stiffness with no penalty of increased mass or increased top shell height required that the curvatures on the old top shell design be increased severely. As can be seen in Figure 12, the resulting design is a more "domed" shape structure than the old design. Figs. 9 and 10 illustrate the analytically determined mode shapes for the first and second modes. In addition to increasing the modal frequency, the areas of maximum displacement, which approximate the radiating areas, are somewhat smaller in size.

The analytical and experimental frequencies of the new design shell are summarized below:

New Shell Design Modal Frequencies

Mode	1	2
Experimental	5.83 kHz	6.08 kHz
Analytical	5.37 kHz	5.49 kHz

RESULTS

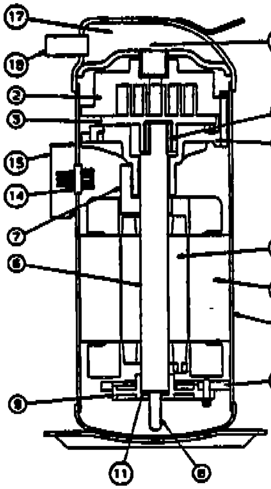
A prototype model of the new domed top shell was made and comparison tested on a running compressor to evaluate its effectiveness in reducing radiated sound. Test results show the new top shell alone contributed an approximate 1.7 dBA noise reduction in both the SC and SR type compressors. In Figures 13 and 14, A-weighted 1/3 octave band sound pressure levels are compared for the old top shell design and the new domed top shell. In the 1/3 octave bands centered at 3.15 kHz-6.3 kHz, corresponding to the first three modes of vibration of the old design top shell, a large decrease in radiated sound energy is noticeable.

CONCLUSIONS

This work demonstrates that the integration of the design and evaluation techniques of time domain and frequency response measurements, experimental modal analysis, and finite element analysis can be combined logically and effectively in reducing compressor noise. These tools can substantially decrease the time and cost of development by guiding engineering efforts in productive directions, reducing the number of prototypes and design iterations required to develop a product. In this particular example, a computer model was confirmed with empirical data and then used to generate a new design which could be fabricated with confidence that the desired goals would be achieved.

REFERENCES

1. Bush, James W., and Neville, Donald G., "Identificaiton and Reduction of Rotary Compressor Pure Tone Noise Sources Using Random Noise Excitation," Proc. Purdue Compressor Conference, 1990, pp. 780-788.



LEGEND

1 Mid-Shell
2 Fixed Scroll
3 Orbiting Scroll
4 Slider Block
5 Crank Case
6 Shaft
7 Upper Counter-weight
8 Oil Pickup Tube
9 Lower Bearing
10 Lower Bearing ring
11 Thrust Washer
12 Rotor
13 Stator
14 Terminal
15 Terminal Fence
16 Discharge Port
17 Discharge Plenum
18 Discharge Tube

Fig. 1: Carrier Scroll Compressor

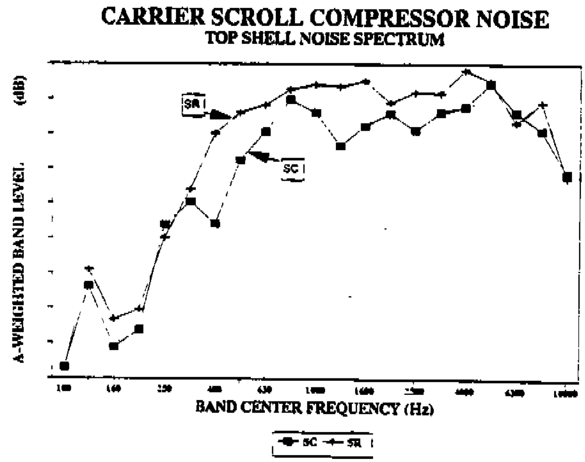


Fig. 2: 1/3rd Octave Band Noise Spectrum. Predominant Noise problems occured between 500Hz and 5000Hz

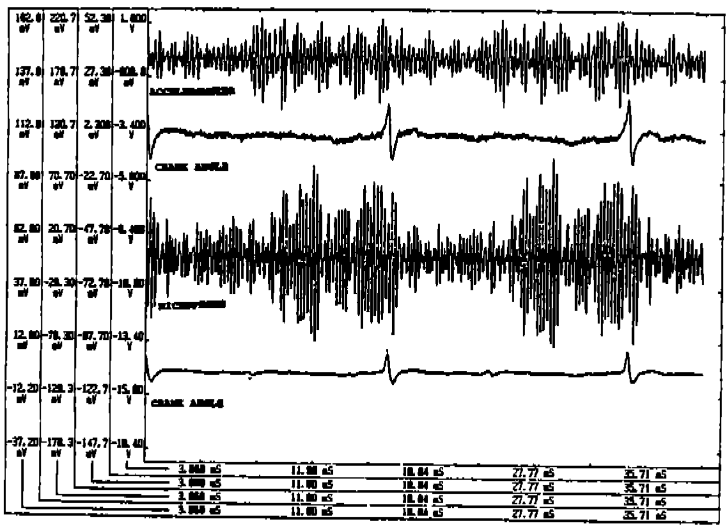


Fig. 3: Measured Microphone and Accelerometer Time Domain Response as a Function of Compressor Crank Angle

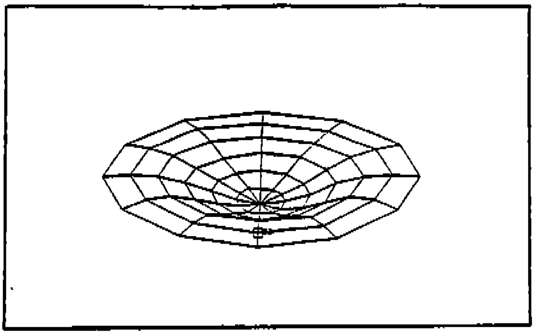


Fig. 4: Old Top Shell.
First mode, 3.25kHz

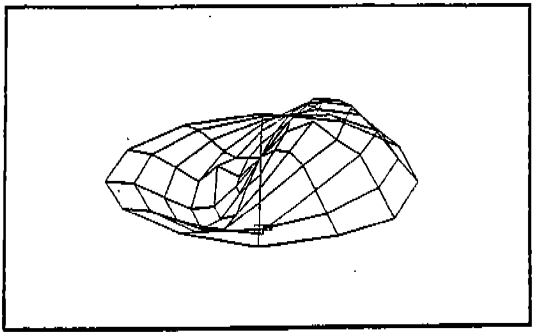


Fig. 5: Old Top Shell.
Second Mode, 4.75kHz

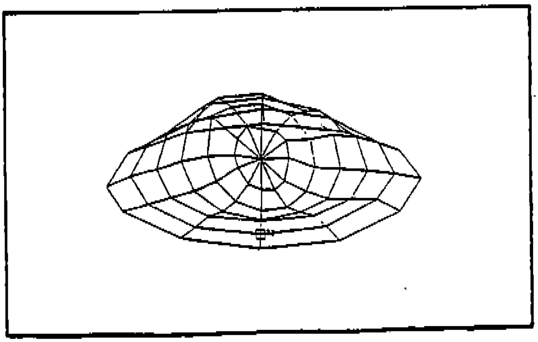


Fig. 6: Old Top Shell.
Third Mode, 5.68kHz

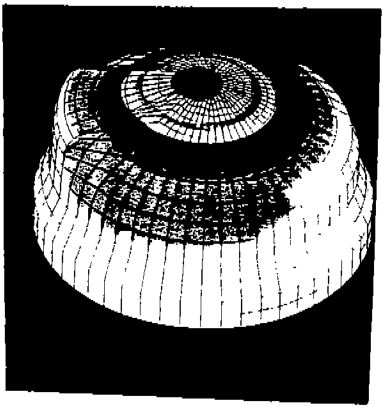


Fig. 7: Old Top Shell FEA Model.
First Mode 3.58kHz

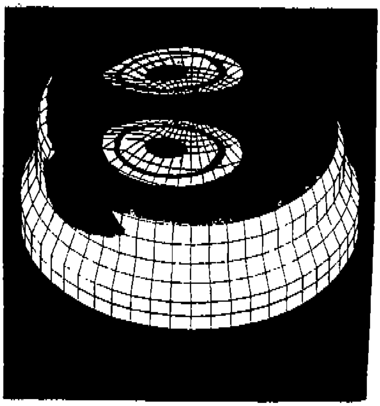


Fig. 8: Old Top Shell FEA Model.
Second Mode 4.95kHz

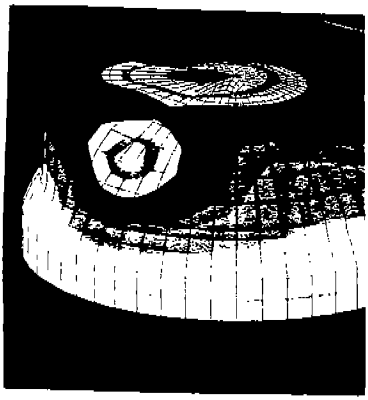


Fig. 9: New Top Shell FEA Model.
First Mode 5.37kHz

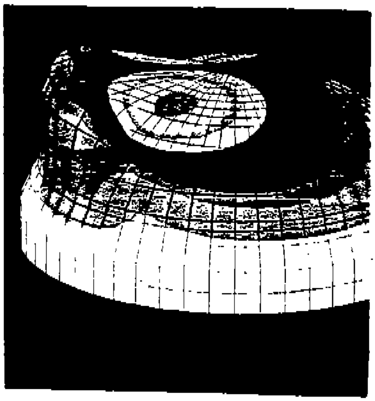


Fig. 10: New Top Shell FEA Model.
Second Mode 5.49kHz

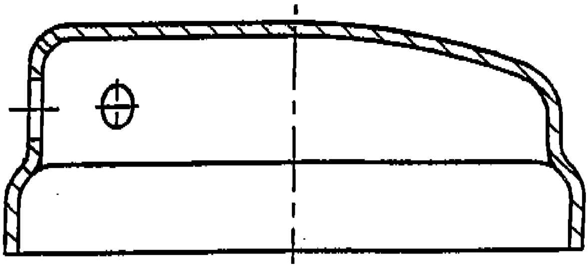


Fig. 11: Old Top Shell Cross-Section

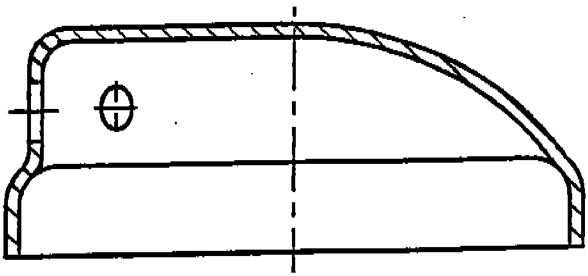


Fig. 12: New Top Shell Cross-Section

SC SCROLL COMPRESSOR NOISE
TOP SHELL NOISE SPECTRUM

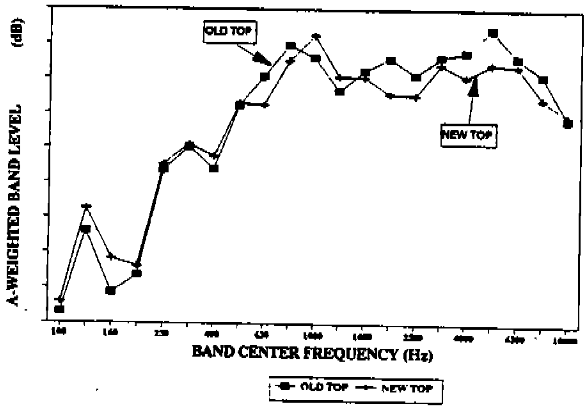


Fig. 13: 1/3rd Octave Band Spectrum for SC Compressor. Notice noise reduction in frequency range of interest.

SR SCROLL COMPRESSOR NOISE
TOP SHELL NOISE SPECTRUM

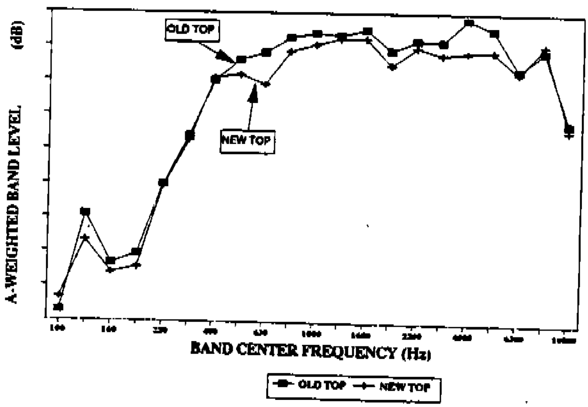


Fig. 14: 1/3rd Octave Band Spectrum for SR Compressor. Notice the difference in noise reduction from the SC type compressor