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HERMETIC COMPRESSOR NOISE CONTROL BY SHELL MODIFICATIONS

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1. ABSTRACT

The noise generated by a hermetically sealed reciprocating compressor reaches the outside environment by traveling through compressor shell. The intensity of the noise reaching outside is known to be sensitive to the geometric configuration of compressor shell.

As part of a noise reduction program, compressor shell has been modeled on ANSYS Finite Element Analysis (FEA) Software and modal analysis was performed on the same. The results from this analysis were then cross checked with the near field sound intensity analysis of the compressor on dual channel FFT sound analyzer, B&K 2144. It was found that there were compressor shell natural frequencies of vibration in 1000 Hz to 3000 Hz frequency range and these were very close to the high intensity peaks in the sound intensity spectrum of compressor obtained by using dual channel FFT analyzer.

Shell shape alternatives were analyzed for their forced response using ANSYS FEA software; shell of modified shape has shown significant improvements in forced response characteristics in the 1000 - 3000 Hz range. A modified shell has been prototyped. An overall noise power reduction of 1.8 dBA has been achieved, without any associated cost penalty.

2. INTRODUCTION

The noise of an air-conditioning compressor (hermetic reciprocating type) used in window-air-conditioning applications, has a direct effect on human beings using appliances fitted with such compressors. A typical layout of such a compressor is shown in Fig. 8.1. All over the world in the hermetic compressor industry; a considerable effort has been made and is being made to understand and to reduce the compressor emitted noise.

Whatever be the noise path inside such sealed compressors, compressor shell vibration characteristics play a vital role on the noise radiated from the compressors. The shell vibration characteristics can be influenced by increasing shell thickness or by modifying its geometry. The major challenge in such noise reduction efforts is that the changes required to be made in the compressor should not increase the compressor cost. Increasing shell thickness is a costly alternative, and hence, the compressor shell geometry modifications, so as to favorably modify the shell vibration characteristics, stand as a more practical alternative. The key here is to bring about a frequency mismatch between the excitation frequencies and the shell natural frequencies of vibration. This approach has been applied to a particular compressor model used on window-air-conditioners. Test results from prototypes made show a significant reduction of 1.8 dBA in sound power levels using ISO: 3745 method of sound power measurement.

3. COMPUTATION OF SHELL VIBRATION CHARACTERISTICS

Shell vibration characteristics, defined by mode shapes and the natural frequencies of vibration of the shell, were computed by using the ANSYS FEA software. See Table 8.1 for specific shell natural frequencies of vibration computed on ANSYS FEA software The modeling efforts were aimed at predicting the shell natural frequencies of vibration as much precisely as possibly without making the modeling too complex. This computation was repeated for standard design, modified geometries as well as thicker shells for comparative purposes. Refer Fig. 8.4 and 8.5 for mode shapes, computed on ANSYS FEA software, at which shell top vibrates for original and modified configurations.

4. PREDICTING SHELL FORCED RESPONSE

The high measured noise levels (the measurements are described in section 6 - for specific values refer Table 8.2) at 1000 Hz and 2000 Hz octave bands indicated that there should be shell natural frequencies of vibration around 1000 Hz and 2000 Hz which are getting excited. However it was realized that manufacturing limitations and costs involved prohibited the change of shell horizontal profile. This meant that only the shell top profile could be changed and those shell natural frequencies of vibration should be looked for at which shell top vibrates. It was expected that modifications in the shell top might bring positive changes in shell side wall vibration characteristics also. Keeping this in mind, the shell top face vibration modes were identified and they were found to be in the 2000 Hz band.

Forced response of the shell was computed on ANSYS FEA software by applying a harmonic pressure pulse from inside with frequencies which were close to shell top vibration mode frequencies and were also seen in FFT response obtained on sound analyzer. See Fig. 8.6 and 8.7 for shell forced response computed on ANSYS FEA software. These were typically in the region of 2 kHz band. This computation was repeated for standard design, modified geometries as well as for thicker shells for comparative purposes.

5. MODIFICATIONS IN THE COMPRESSOR SHELL SHAPE

Based on the work as mentioned above, it was decided to modify the shell geometry so as to push the shell natural frequencies towards higher values and hence make the shell stiffer plus bring about a mismatch between the shell excitation and the shell vibration modes. The various options were studied on ANSYS FEA software and increased shell top radius in place of the original radius brought about a significant increase in the shell top vibration mode frequencies. It is also found that the effect of increased radius is comparable to that of a thicker shell of thickness 30% more. Also, very importantly, the compressor cost remained unaffected and no changes were required to be made in the compressor inside components, thereby eliminating over 30% increase in a critical component cost. The harmonic response of the shell with modified geometry indicated significantly lower levels of vibration amplitudes under excitation from inside the shell at the earlier high vibration frequencies. Refer Table 8.3 for comparison. From Fig. 8.6 and 8.7 it can be seen that shell top configuration modification has also resulted in lower vibration amplitudes at shell sides under shell excitation from inside on ANSYS FEA software. This also contributes to reducing the shell emitted noise.

6. SOUND INTENSITY MEASUREMENTS

The running compressor was put to sound testing in a semi-anechoic chamber. The compressor sound power data indicated that there was high noise at 1000 Hz and 2000 Hz octave bands. Further to this the compressor noise intensity level distribution over shell surface was collected by obtaining near field sound intensity data. For this purpose the compressor shell surface was divided into a grid of 234 numbers of 1 inch side squares and near field sound intensity measurements were made at the center of each such square with the help of B&K 2144 sound intensity FFT analyzer. Refer Fig. 8.3 for sound intensity plots at the shell top at frequencies close to shell natural frequencies of vibration at which shell top vibrates.

7. CORRELATING PREDICTION AND EXPERIMENTATION

The contour plot obtained from the near field sound intensity data was then compared with these mode shapes. Also the shell was subjected to harmonic pressure pulse from inside on ANSYS FEA software, at different frequencies as shared above. At a given frequency, this harmonic response was also compared with the sound intensity contour plot on the shell top surface obtained from experimentation using FFT analyzer. The remarkable resemblance was seen between the shell top sound intensity distribution and the shell response obtained on ANSYS at frequencies of 2112 Hz and 2752 Hz. See Fig. 8.6, 8.7 and Fig. 8.3. This confirmed that the shell top vibration modes are significant noise contributing parameters in compressor emitted noise frequency range.

8. RESULTS

The compressor sound power level, by making changes in its shell geometry has been reduced by 1.8 dBA. The detailed computational and experimentation results and figures are given on page numbers 4 to 8.

9. CONCLUSIONS

1. The natural frequencies of vibration & mode shapes of shell significantly influence noise characteristics of compressors.
2. The compressor noise can be reduced by altering geometry of the shell .

10. REFERENCES

1. Kelly, A. D. and Knight, C.E., "Dynamic Finite Element Modeling and Analysis of Hermetic Reciprocating Compressor," Proceeding of 1994 International Compressor Engineering Conference at Purdue, West Lafayette, 1994.
2. Baxa, E. D., "Noise Control in Internal Combustion Engines," Robert E. Krieger Publishing Company, Malabar, Florida, 1989.
3. Hamilton, J. F., "Measurement and Control of Compressor Noise," Ray Herrick Laboratory, Purdue University, USA, 1988.
4. ISO 3745.

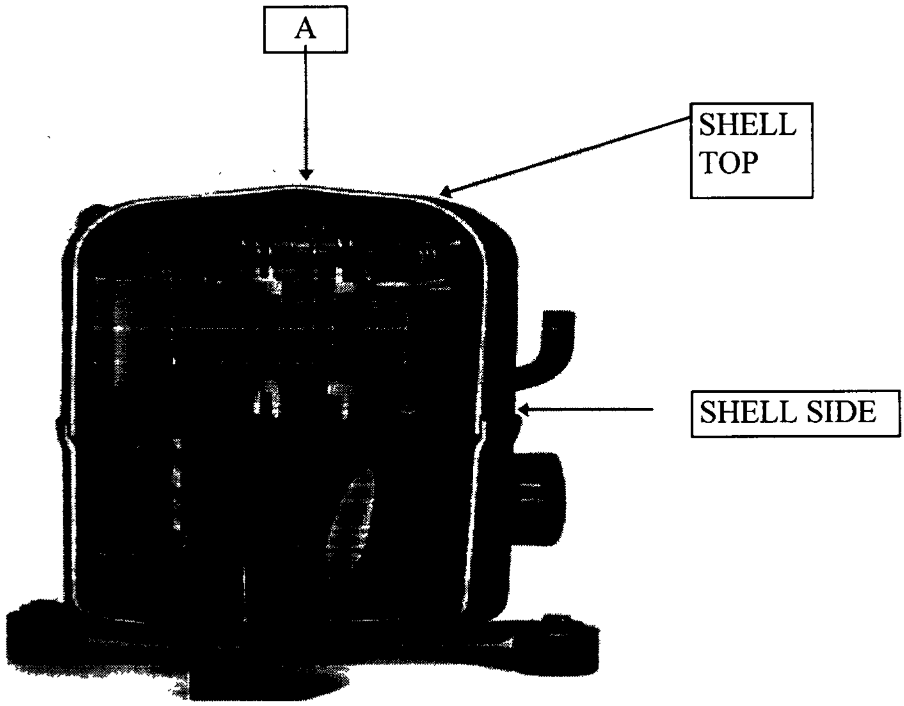
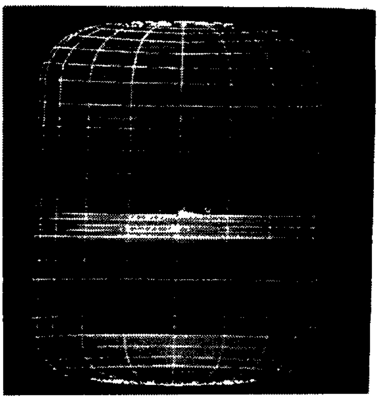
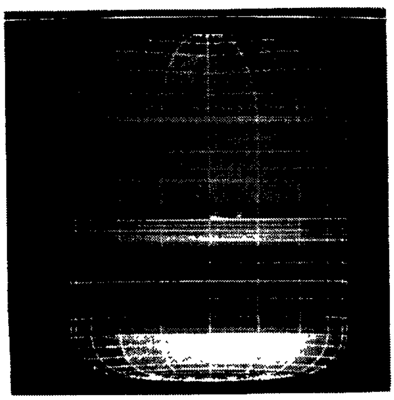


FIG. 8.1 A TYPICAL COMPRESSOR LAYOUT - CUT VIEW

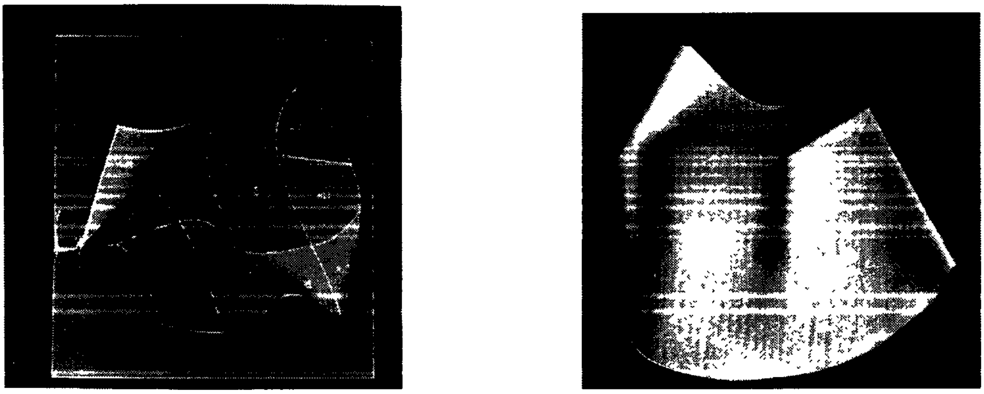


a. ORIGINAL CONFIGURATION



b. MODIFIED CONFIGURATION

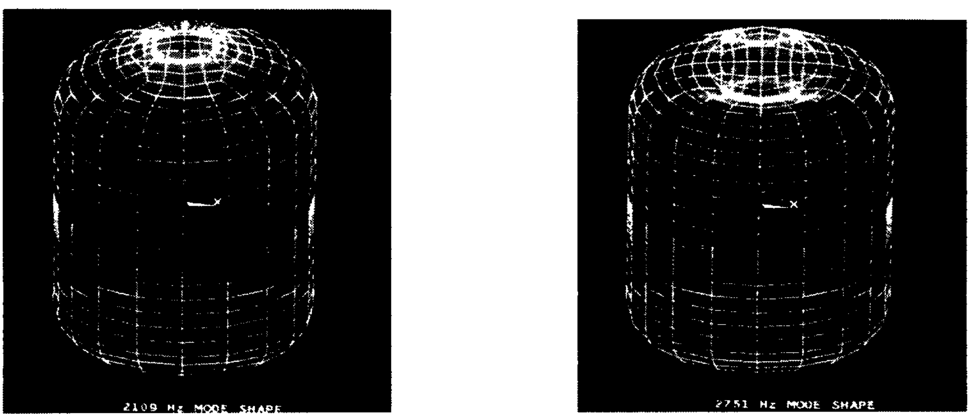
FIG. 8.2 SHELL CONFIGURATIONS



a. AT 2112 Hz

b. 2752 Hz

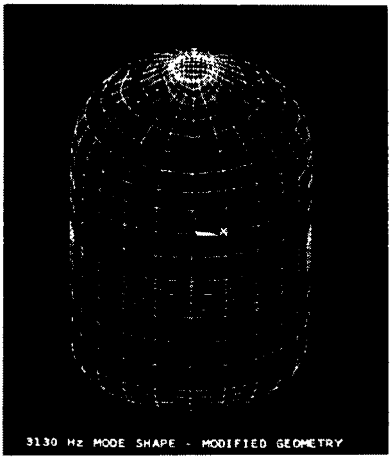
FIG. 8.3 SOUND INTENSITY PLOTS OVER SHELL TOP



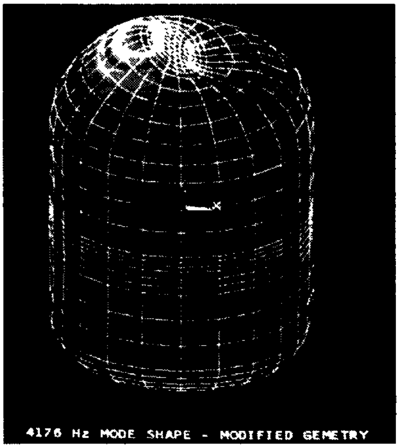
a. 2109 Hz

b. 2751 Hz

FIG. 8.4 SHELL TOP VIBRATION MODE SHAPES
ORIGINAL CONFIGURATION

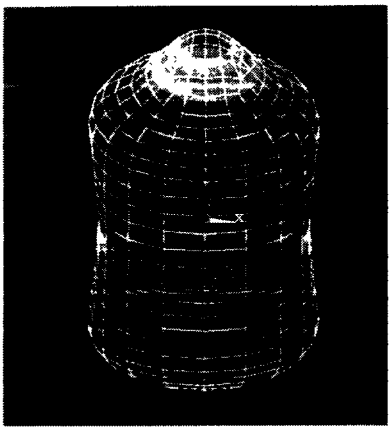


a. 3130 Hz

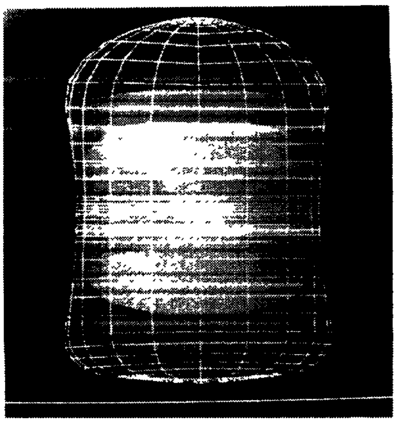


b. 4176 Hz

FIG. 8.5 SHELL TOP VIBRATION MODE SHAPES
MODIFIED CONFIGURATION

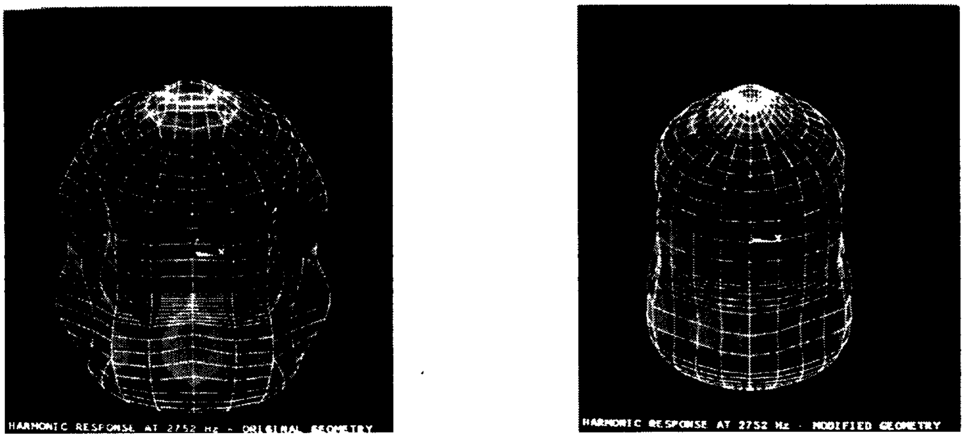


a. ORIGINAL CONFIGURATION



b. MODIFIED CONFIGURATION

FIG. 8.6 SHELL FORCED RESPONSE COMPARISON - AT 2112 Hz



a. ORIGINAL CONFIGURATION b. MODIFIED CONFIGURATION

FIG. 8.7 SHELL FORCED RESPONSE COMPARISON - AT 2752 Hz

TABLE 8.1

NATURAL FREQUENCIES (IN Hz) OF VIBRATION OF THE ORIGINAL AND THE ALTERNATIVE SHELLS AS COMPUTED ON ANSYS FEA SOFTWARE			
MODE NUMBER	ORIGINAL SHELL	THICKER SHELL	MODIFIED GEOMETRY SHELL - TOP RADIUS INCREASED
9	1639	2129	1716
10	1766	2130	1865
11	2109 - SHELL TOP VIBRATION MODE	2374 - SHELL TOP VIBRATION MODE	2228
12	2193	2481	2255
18	2501	2972	2884
19	2751 - SHELL TOP VIBRATION MODE	3043	2908
20	2805	3341 - SHELL TOP VIBRATION MODE	3106
21	2806	3345	3130 - SHELL TOP VIBRATION MODE
36			4068
37			4176 - SHELL TOP VIBRATION MODE

TABLE 8.2

SOUND POWER LEVEL OF MODIFIED CONFIGURATION SHELL COMPARED WITH ORIGINAL SHELL				
FREQUENCY Hz	ORIGINAL SHELL	MODIFIED SHELL	ORIGINAL SHELL	MODIFIE D SHELL
63				
125		49.4		48.5
250	51.4	48.2	51.8	51
500	48.5	44.6	48.3	45.5
1000	56.6	55.7	59.9	58.8
2000	58.3	56.4	59.3	56.6
4000	55	54.3	56.5	52.8
8000	49.5	50.2	48.1	48.8
A-WEIGHTED	62.8	61.1	63.9	62.0

AVERAGE NOISE REDUCTION ON TWO PROTOTYPES : 1.8 dBA

TABLE 8.3

SHELL TOP VIBRATION AMPLITUDES UNDER INSIDE PRESSURE PULSE EXCITATION COMPUTED ON ANSYS AT POINT A IN FIG. 8.1			
FREQUENCY OF EXCITATION, Hz	ORIGINAL SHELL CONFIGURATION	ORIGINAL SHELL WITH INCREASED THICKNESS	MODIFIED SHELL CONFIGURATI ON
2112	1460E-05	10.3E-05	2.62E-05
2752	7.77E-05	7.45E-05	6.61E-05