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THE NOISE SOURCE IDENTIFICATION OF A RECIPROCATING REFRIGERATION COMPRESSOR

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Abstract: In this study, the vibration and noise characteristics of the stationary refrigerant hermetic reciprocating compressor shell system are investigated experimentally by three different techniques namely the modal analysis impact excitation, measurement of sound power level and acoustic intensity measurement. The hermetic shell of the compressor is the most important component between the noise sources and the receiving human ear. In order to prevent and reduce such noise, sound power level and acoustic intensity are measured for the compressor, and the results of these measurements, the noise radiation characteristics of the hermetic reciprocating compressor are identified. Also the experimental modal analysis is applied to the hermetic shell of the compressor to identify noise sources.

1. INTRODUCTION

Recently, noise and vibration problems are gaining increasing importance in the electrical home appliances and the trend is to manufacture lighter and higher quality goods. It is complicated to find the generating source and propagating path of noise in the compressor. Fractional horsepower reciprocating compressors are commonly found in HVAC systems such as household refrigerators and air conditioners. Fig.1 shows a typical reciprocating piston compressor. The compressor noise is a major contributor to the overall noise level of the domestic refrigerator. The periodic operation of the compressor and noise radiated during operation has been deemed to be a considerable problem by the consumer, especially during the quite hours of the night. As a result, there is an increased need for understanding the noise and vibration characteristics of these compressors for the purpose of noise control.

In the reciprocating piston compressors, the compression process is the origin of all noise radiated from the compressor. Suction and discharge gas pulsations and compressor mechanism vibrations are all caused by the compression process and are the main noise sources in the compressor. The noise can be transmitted through four paths, the refrigerant gas path, the discharge tube path, the suspension system path and the lubrication oil path. All noise paths include the compressor shell causing either direct noise from the compressor, or indirect noise because of the excitation of the surrounding structure by the compressor.

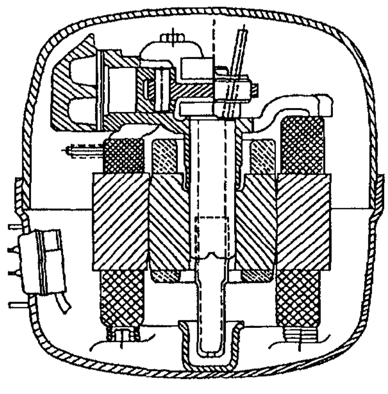

Fig.1. Reciprocating Refrigeration Compressor.

In the present work, the dynamical characteristics of the stationary compressor shell system were experimentally determined using modal analysis and impact excitation. The goal of this study is to identify and evaluate the contribution of the shell's dynamical characteristics in the transmission and amplification of compressor noise for the understanding of approaches to reduce the direct noise radiation from the compressor shell. In order to prevent and reduce such noise, sound power and acoustic intensity are measured for the compressor, and the results of these measurements, the noise radiation characteristics of the compressor are identified.

2. APPLICATION OF EXPERIMENTAL MODAL ANALYSIS TECHNIQUE

To identify the structural resonances of the compressor, an impact test of the compressor shell is done. The impact technique uses a hammer equipped with a force transducer and an accelerometer attached to the test structure. Compared to steady state sinosoidal excitation where the structure is driven at only one discrete frequency uses an impact force where the structure is excited over the broad band frequency range as it is struck by the hammer. Both signals, the signal of the hammer and the response signal of the accelarometer are measured simultaneously. The fast Fourier Transform (FFT) of the acceleration divided by the FFT of the input force yields the Frequency Response Function (FRF) for a particular set of two points where the structure is impacted and the acceleration is measured. The Frequency Response Function is of complex form and contains amplitude ratio and phase information for each frequency band in the spectrum [1].

Once the set of data for a structure with n measurment points is obtained the model coefficents can be extracted from the measured transfer functions by a suitable algorithm and the mode shapes displayed. The procedure is valid for a linear where the Maxwell-Beti reciprocity principle holds. To obtain the required data set either the acceleration can be measured at a fixed point and the structure is impacted at all the points or vica versa. The fixed point should be chosen at a location where no modal line of any of the important modes passes through.

Modal analysis software is used to calculate the natural frequencies of the compressor shell and to display the natural modes of vibration. One advantage of using the modal program is the possibility to analyze the stored data any time after the measurement is complated. Since there exist a large number of modes within the frequency range of interest, this analysis provides valuable information on the contribution of the relevant mode to the overall noise level of the compressor shell. The Coherence Function is also computed. The Coherence Function serves as a criterion to judge the validity of the measurements. If The Coherence Function is not equal to unity a weak signal to noise ratio, leakage, nonlinear effects or an output signal caused by influences other than the measured input are possible reasons. Some of natural modes of vibration of the compressor shell obtained through the modal analysis are shown schematically in Fig.2.

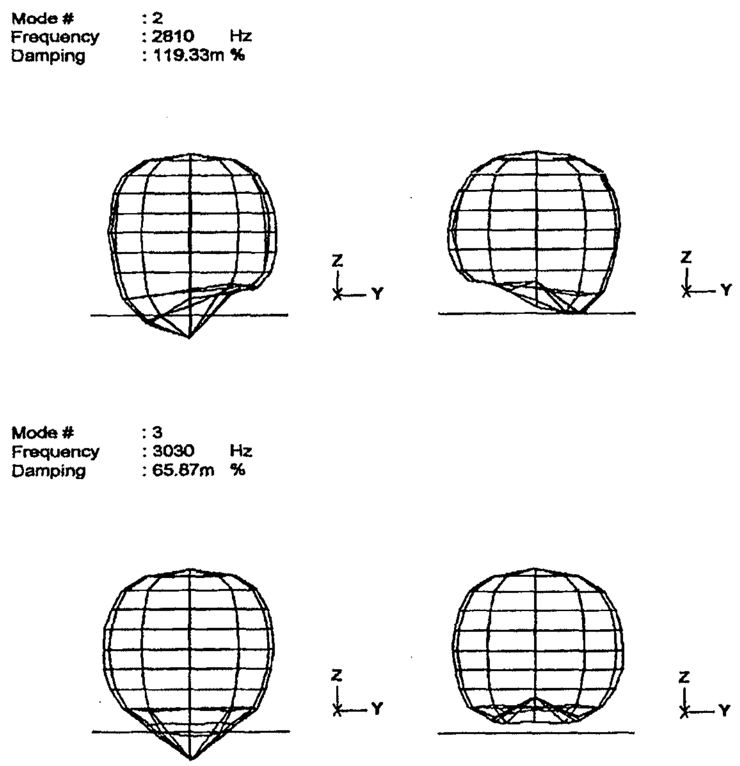

Fig.2. The Natural Frequencies and the Corresponding Natural Modes of the Compressor Shell.

3. THE MEASUREMENT OF SOUND POWER AND ACOUSTIC INTENSITY

Both the measurements of sound power and acoustic intensity were carried out at the steady state operating conditions of the compressor. Sound power measurements were made with real-time dual-channel frequency analyzer. The analyzer is available for calculation of sound power according to ISO 3745 standart [2].

The sound power measurements were made in a semi-anechoic room. The measurement surface on which the measuring points are located is a hemisphere enveloping the compressor. The results are represented in Table 1 and typical frequency spectrum plot of the compressor is shown in Fig.3.

Table 1. Sound Power Measurements.

Segment	Point	Area [m²]	Sound Pressure Level dB	Sound Pressure Level dBA	Sound Power Level dB	Sound Power Level dBA
1	1	0,3534	41,3	40,6	36,8	36,1
1	2	0,3534	39,8	39,3	35,3	34,7
1	3	0,3534	40,5	39,9	36	35,4
1	4	0,3534	40,9	40,3	36,4	35,8
1st Segment		1,4136	46,7	46,1	42,2	41,6
2	1	0,4241	36,6	36,2	32,8	32,5
2	2	0,4241	35,8	35,4	32,1	31,7
2nd Segment		0,8482	39,2	38,9	35,5	35,1
3	1	0,3534	37,4	36,9	32,9	32,3
3	2	0,3534	38,2	37,6	33,6	33,1
3	3	0,3534	36,4	36	31,9	31,5
3	4	0,3534	37,3	37,1	32,8	32,6
3rd Segment		1,4136	43,4	42,9	38,9	38,4
4	1	0,6786	37,3	36,8	35,6	35,1
4	2	0,6786	37,3	36,7	35,6	35,1
4th Segment		1,3572	40,3	39,8	38,6	38,1
Total		5,0326			45,4	44,9

85 points over the prismatic surface around the compressor are selected and the average intensity level is measured. The sound power is then calculated by multiplying by the area of the surface by the average intensity.

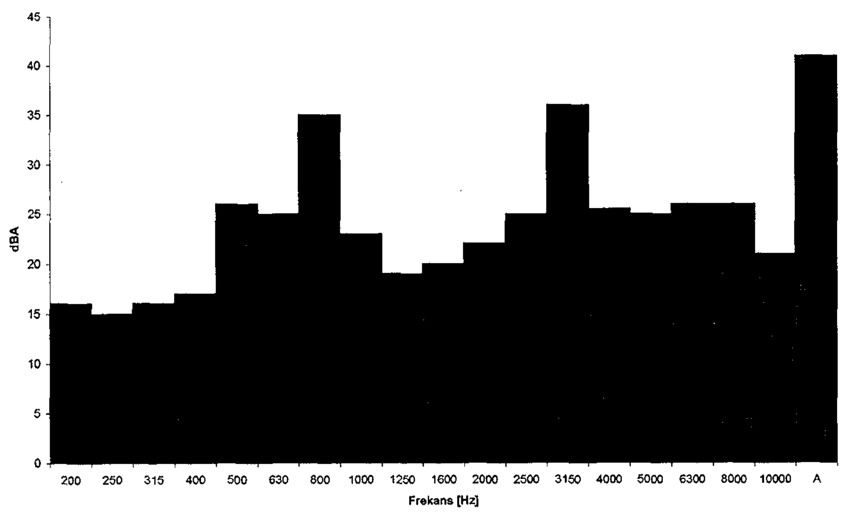

Fig.3. Sound Power Frequency Spectrum.

Corrections to account for air density variation are automatically performed by entering the values of temperature and pressure. All measurements are performed with 12mm spacer between the two measuring microphones. The results are represented by contours as shown in Fig.4.

4. EXPERIMENTAL RESULTS AND CONSIDERATIONS

The frequency response characteristics of compressor shell vibration shows relatively high peak values between 2.8kHz and 3.5kHz and above 4.5kHz. At this frequency band, natural modes of the lower part of compressor shell are appeared as shown in Fig.2 which is the principle noise source. The experimental results of sound power under the steady state operating conditions of the compressor are shown in Table 1. The measurements of overall sound pressure levels were conducted in the semi-anechoic chamber and the overall sound power level is computed. In order to find the response characteristic of sound power, center frequency levels are shown in Fig.3 using 1/3 octave band frequency spectrum. From this figure, it can be seen that the dominant center frequencies are 800Hz and 3150Hz. Contour plots of sound intensity both in the full measuring frequency range and in between 2828Hz and 3563Hz are shown in Fig.4 to identify in more detail the compressor noise directivity pattern. The highest level of 3150Hz noise is influenced by the natural modes on the lower part of compressor shell. In the vicinity of 800Hz the mounting vibration of the compressor motor within the casing shell becomes dominant. This effect can not be observed by the modal analysis of the shell. It is an operational deflection shape. This effect around 800Hz contribute to the sound power at 800Hz.

5. CONCLUSION

Since noise radiated by the compressor in a refrigerator accounts for a major portion of the total noise emitted from the refrigerator it is desirable to control the compressor noise. The first step in solving noise problems is to identify the sources and transmission paths of the noise. In the present study an attempt is made by using three different experimental techniques namely the modal analysis impact excitation, measurement of sound power level and acoustic intensity measurement in a reciprocating refrigeration compressor. Data analysis was emphasized in the frequency ranges with strong noise. From the considerations of the experimental results, following conclusions are obtained. The dominant frequency bands in 1/3 octave band are 800 Hz and 3150Hz. The main source of noise is the shell vibration, which is generated from the lower part. And this vibration produce the structure borne sound, leading to the solid body motion of the shell influenced mounting. Once the dominant noise sources and transmission paths are determined, noise control strategies can be identified to eliminate or reduce the noise emitted from these sources by either reducing the source level or modifying the transmission paths.

A few conclusive remarks regarding the general concept and how to organize a noise study most efficiently are appropriate. Most important, one single type of experiment (e.g. investigation of the dynamical characteristics of structure) seldom produces the information needed to fully explain the noise generation mechanism. It merely provides a portion of the true picture. A characteristic problem of noise analysis is the inordinate amount of data that can be produced by any single type of experiment. The temptation is strong to elaborate too much on the large quantity of produced measurement results instead of ignoring the redundant results. To focus on significant data is only possible if further pieces of information are avaliable. For this reason it is essential that a variety of different pieces of information is acquired at the outset of the project by a scanning process using simple measurements. A rough overall picture of the mechanism can be gained by combining the information from different experiments. Once an efficient approach is well defined, then sophisticated experiments can be undertaken and a through data analysis of the details becomes meaningful.

6. REFERENCES

[1] D.J. EWINS, Modal Testing: Theory and Practice, John Wiley, New York, 1991.

[2] ISO 3745, Acoustic-Determination of Sound Power Levels of Noise Sources-Precision Methods for Anechoic and Semi-Anechoic Rooms, Geneva, Switzerland, 1977.

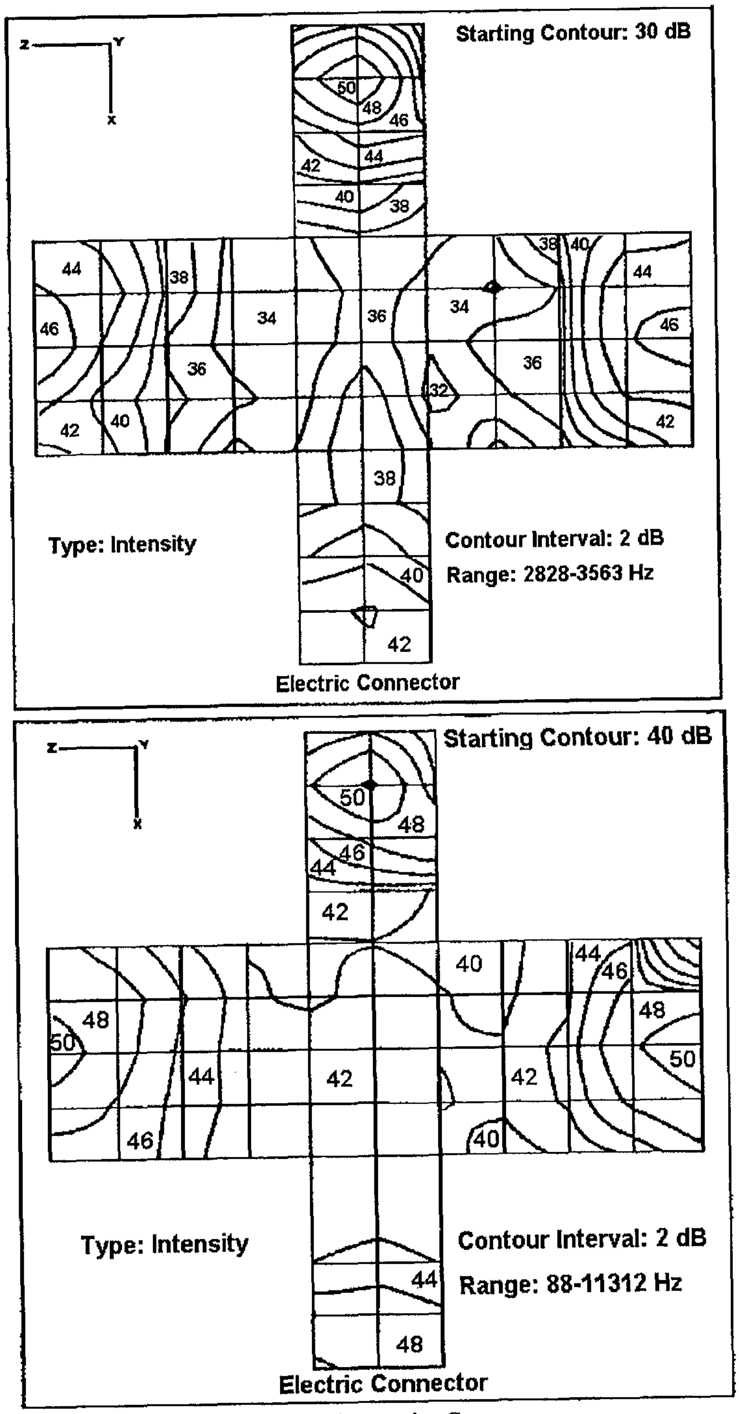

Fig.4. Sound Intensity Contours.