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INFLUENCE OF THE HOUSING ON THE NOISE EMITTED BY A RECIPROCATING COMPRESSOR

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Abstract

The influence of the housing on the noise emitted by a typical reciprocating compressor for domestic appliances is analyzed. Sound pressure and sound intensity measurements were conducted to analyze the noise properties emitted by the compressor with and without the housing. The elastic properties of the compressor housing were determined experimentally and by a finite element model. The results show that the dominant noise frequencies correspond to the eigen frequencies of the housing. Simple stiffening of the housing is shown to reduce amplitude and modify the frequency content of the noise recorded outside the compressor.

INTRODUCTION

Noise is one of the most important criteria on today's market of compressors for domestic appliances. In order to reduce the noisiness of a compressor, one has to investigate both the noise sources and the noise transmission paths (Norton, 2003; Moyne, 2005). In general, noise sources depend on the physical principle employed for compression, while the transmission paths depend on the design of the compressor (Moyne, 2005).

In this article, the noisiness of a single-piston reciprocating compressor is investigated. The compressor consists of a compressor unit mounted in a housing. The dominant noise sources within the compressor unit appear to include cooling media pulsations in the muffler and along the pipes and chambers, inlet/outlet valve vibration, etc. Noise emitted within the housing is then transmitted through the housing to the outside, where it is heard by the owner of the domestic appliance in which the compressor is used. Since the housing acts as a noise transmitter, it is expected that its elastic properties affect the properties of noise outside the compressor.

The aim of the article is to examine the influence of the housing on the noise outside the compressor. For this purpose, sound pressure and spatial distribution of sound intensity are recorded and compared for a reciprocating compressor with and without a housing. The transfer function of the housing is determined experimentally and by a finite element model. The analysis shows that the dominant frequencies of noise of the compressor with housing mainly correspond to the eigen frequencies of the housing. A simple reinforcement of the compressor housing indeed modifies the frequency content of the noise and reduces the amplitudes of the dominant noise frequencies.

EXPERIMENTS

In order to determine the influence of the housing on the noise emitted by the compressor, experiments were conducted using two reciprocating compressors: one with a housing and one without. The compressor with a housing was operating under realistic conditions. In the compressor without a housing, the suction and exhaust pipe had to be opened, which prevented normal operation. However, it was assumed that the open pipes did not significantly change the properties of noise emitted during the operation of the compressor. The experiments were conducted in a room designed for standard sound power level measurements of compressors for domestic appliances.

Sound pressure and sound intensity were recorded for both compressors using a microphone and a sound intensity probe. The microphone was mounted approximately 25 cm directly above the compressor. During a 2 min period of compressor operation, 10 sound pressure signals were recorded at a sampling frequency of 16 kHz, lasting 4 sec each. The signals were stored for subsequent frequency analysis.

Figure 1 – Experimental setup for sound intensity measurement

For sound intensity measurement, a probe with a face-to-face arrangement of the pressure microphones was employed. A 12 mm solid spacer between the microphones was used, allowing quality measurements of signals with frequencies up to 5 kHz. To determine the spatial distribution of sound intensity, the outer form of the compressor was modelled by a virtual square block represented by a thin wire frame. Each of the five reference planes of the block contained 16 evenly distributed reference points. The distance between the reference planes and the compressor surface was approximately 15 cm. At each reference point, 6 pairs of sound pressure signals were recorded at a sampling frequency of 20 kHz, lasting 4 sec each. The sound intensity was then calculated from the imaginary part of the cross-spectrum of a pair of sound pressure signals (Crocker, 2003; Fahy 1995). The sound intensity probe was always oriented perpendicularly to the reference plane, and thus the sound intensity calculated and analyzed in this article is in fact a component of the sound intensity vector normal to the reference plane.

The modal properties of the compressor housing were determined experimentally and by a finite element (FE) model. Experiments involved impact excitation of the housing by an instrumented hammer at several points evenly distributed along the housing cover (Figure 2). The response of the housing was recorded by an accelerometer mounted on top of the cover. At each excitation point, 10 excitation-response signal pairs were recorded at a sampling frequency of 25 kHz, lasting 1 sec each. The recorded signals were then used to calculate the transfer function of the housing cover (Ewins, 2000).

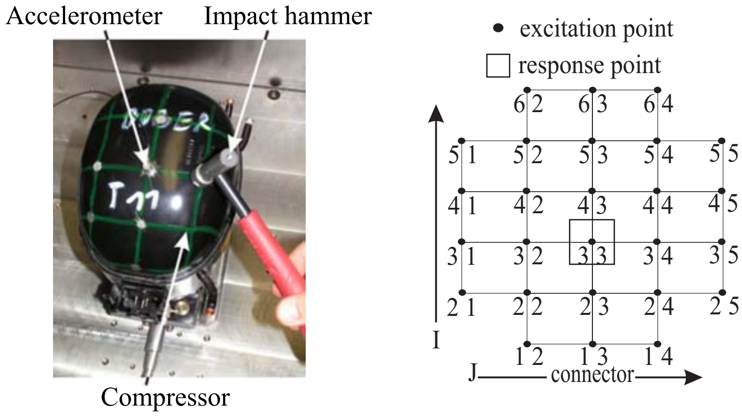

Figure 2 – Experimental setup for transfer function measurement (left), and the distribution of excitation points and the response point

Eigen frequencies and eigen modes of the housing were also calculated by an FE model using a commercial FE solver.

RESULTS

Sound pressure

Let us first consider the sound pressure of noise emitted by the compressor with and without a housing. The average amplitude spectra of the recorded sound pressure signals for the two compressors are shown in Figure 3. It is obvious that the average amplitude of noise emitted by the compressor with a housing is much lower than without a housing. A difference in frequency content of the two noises can also be observed. In the noise of the compressor with a housing, the dominant spectral peaks are found at frequencies 4.2 kHz, 5.3 kHz and 5.7 kHz. In the noise of the compressor without a housing, these spectral peaks are not observed at all, while there are several dominant peaks in the range from 50 Hz to 2 kHz. In both cases, the spectral peaks are mostly located at multiples of 50 Hz, which corresponds to the fundamental operating frequency of the compressor. These results therefore indicate that the housing not only reduces the amplitude of noise emitted by the compressor within the housing, but it also markedly changes the frequency content of the noise. The analysis of spatial distribution of sound intensity shows some additional influences of the housing on the noise properties.

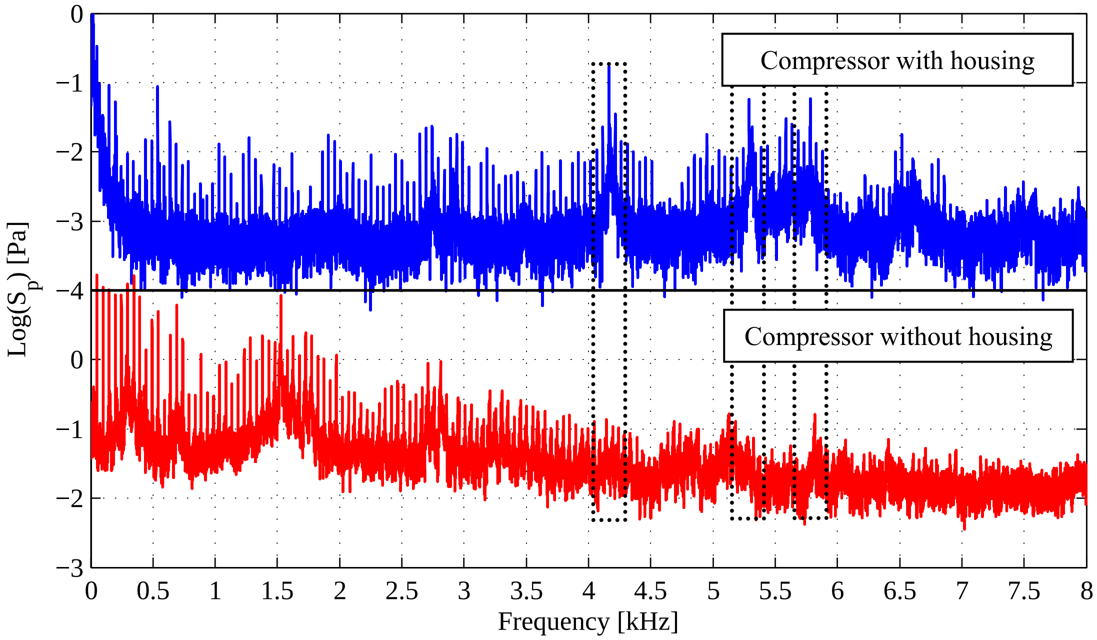

Figure 3 – Average amplitude spectra of the sound pressure of noise emitted by the compressor with (top) and without housing (bottom)

Sound intensity

Figure 4 shows the spatial distribution of sound intensity for the compressor with and without a housing. The five fields with 16 squares each correspond to the five reference planes around the compressor, as illustrated in the top right corner of the figure. The sound power levels L_{Wi} above the five fields denote the noise power emitted through the five reference planes. The boxed sound power level L_W stands for the total sound power level from all reference planes.

The compressor with a housing emits most of the noise through the top (5) and right (2) reference planes. The highest sound intensity amplitudes are found at the top plane (5). The spatial distribution of sound intensity for the compressor without a housing is significantly different. Most of the noise is emitted from the bottom-right (2) and back (3) planes, where the muffler and inlet and outlet pipes are located. The total sound power levels emitted by the compressor with and without a housing are 34.1 dB and 79.3 dB, respectively, indicating a tremendous damping effect of the housing. However, the differences between the spatial distributions of sound intensity of the compressors with and without a housing also show that, apart from decreasing the noise amplitude and modifying its frequency content, the housing also spatially rearranges the noise emitted by the compressor.

Results of sound pressure and sound intensity analyses therefore indicate that the compressor noise recorded outside the housing is significantly affected by the housing. In order to better understand the influence of the housing on noise, it is necessary to determine the modal properties of the housing.

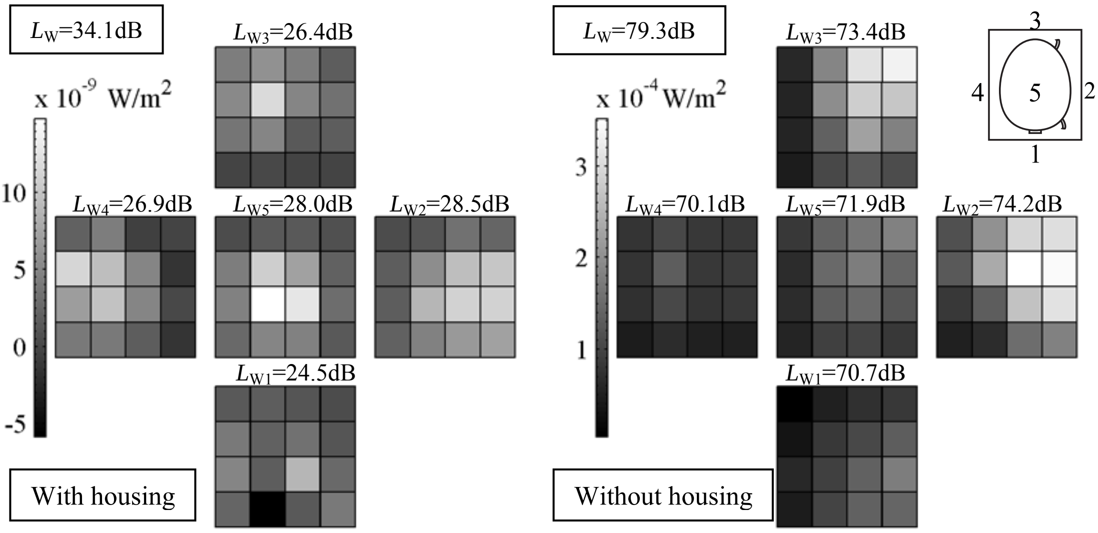

Figure 4 – Spatial distribution of sound intensity in the frequency range from 200 Hz to 8 kHz emitted by the compressor with (left) and without housing (right). Note that the amplitude scales differ by several orders of magnitude

Transfer function of the housing cover

The transfer function of the housing cover was determined experimentally and by the finite element (FE) method. Figure 5 shows the amplitude of the recorded transfer function at the top of the housing cover (point 33 in Figure 2). The highest spectral peaks appear at frequencies 4.2 kHz, 5.2 kHz, 5.6 kHz and 6.5 kHz, which represent the eigen frequencies of the housing. Similar frequencies were obtained by the FE method. These frequencies closely match those of the dominant spectral peaks in the spectrum of noise emitted by the compressor with a housing (Figure 3).

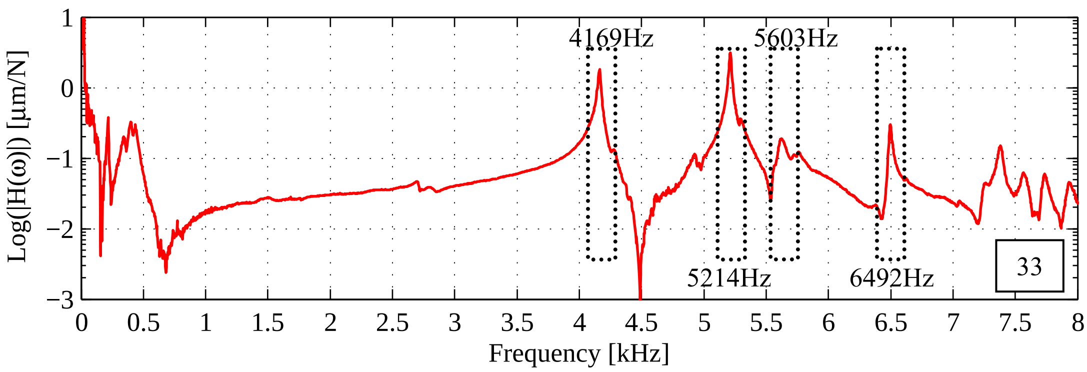

Figure 5 – Amplitude of transfer function amplitude of the housing cover at the cover top

Figure 6 shows the reconstructed and FE-calculated shapes of the dominant eigen mode of the housing cover. The two mode shapes agree very well. It can be observed from the mode shapes that the location of the maximal housing displacement corresponds to the location of the highest sound intensity amplitudes (Figure 6). It therefore seems reasonable to stiffen the housing in order to decrease the amplitude of noise emitted by the compressor through the housing (Moyne, 2005).

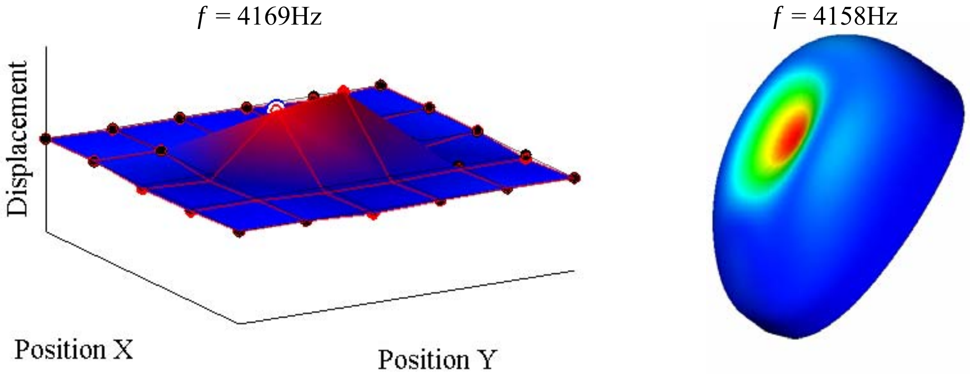

Figure 6 – The reconstructed (left) and the FE-calculated (right) shapes of the dominant eigen mode at a frequency of 4169Hz

SIMPLE MODIFICATION OF THE HOUSING

There are several ways to stiffen the housing, such as modifying its design, increasing its thickness or attaching stiffening ribs to it. For this investigation, the last way was chosen, and two crossed ribs were welded to the top of the housing (Figure 7). The location of the ribs was determined based on the analysis of the eigen-mode shapes and the spatial distribution of sound intensity, while the shape of the ribs was chosen to be as simple as possible. The influence of such a modification was evaluated by analysis of sound pressure and the sound intensity distribution of noise emitted by the compressor with a modified housing.

Figure 7 – Compressor with modified housing

Figure 8 compares the amplitude spectra of the sound pressure for noise emitted by the compressors with the original and modified housings. Modification of the housing decreases the amplitudes and changes the frequency of the dominant spectral peaks of noise. The dominant spectral peak is shifted from 4.1 kHz for the original housing to 3.9 kHz for the modified housing, while its amplitude is significantly decreased. Analysis of the transfer function of the modified housing also confirms that the frequencies of the dominant spectral peaks of noise still lie close to the eigen frequencies of the housing.

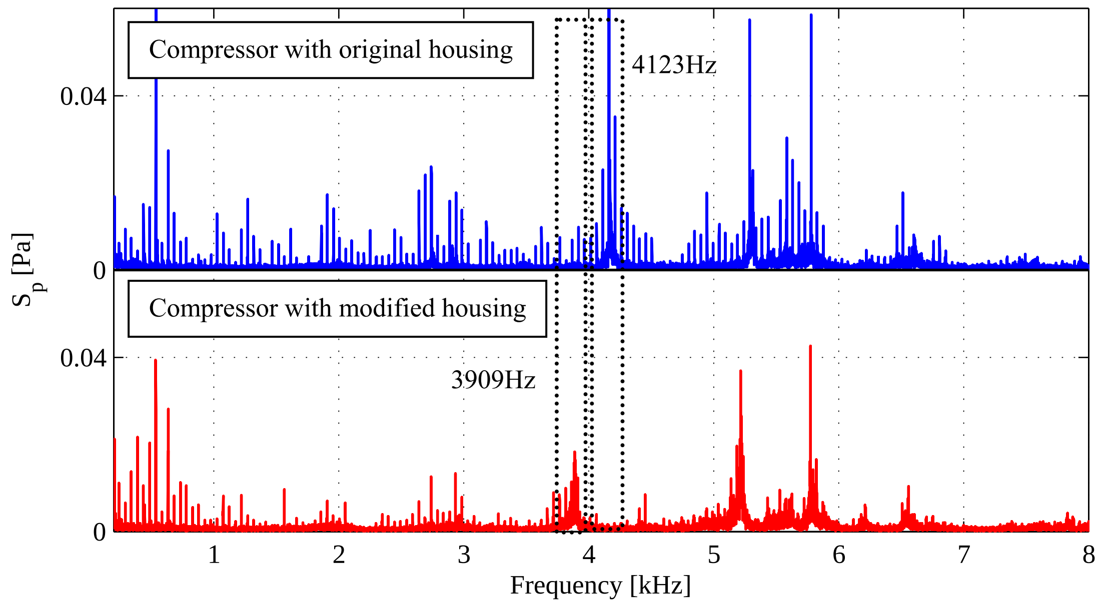

Figure 8 – Average amplitude spectra of the sound pressure of noise emitted by the compressor with original (top) and modified housings (bottom)

The spatial distributions of sound intensity emitted by the compressors with original and modified housings are compared in Figure 9. The housing modification decreased the sound power levels at the right (2), left (4) and top (5) reference planes, while the total sound power level decreased from 34.1 dB to 33.1 dB. Detailed analysis revealed that the largest decrease in power level occurs in the frequency range from 3.5 kHz to 4.3 kHz (Jerič, 2005), which contains the dominant spectral peak of the noise.

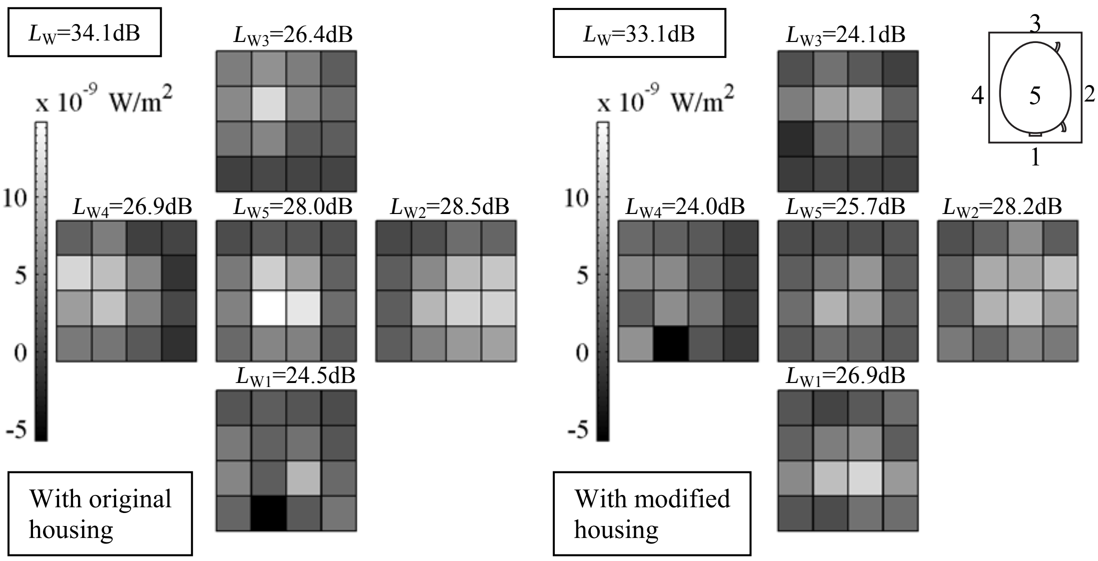

Figure 9 – Spatial distribution of sound intensity in the frequency range from 200 Hz to 8 kHz emitted by the compressor with (left) and without housing (right).

CONCLUSIONS

The influence of the housing on the noise emitted by a reciprocating compressor was analyzed in the paper. Analysis of recorded sound pressure and spatial distribution of sound intensity emitted by the compressor with and without the housing revealed that the housing significantly affects the noise emitted inside and transmitted through the housing. The housing not only dampens the noise amplitude, but it also modifies its frequency content and the spatial distribution around the compressor. The dominant spectral peaks of noise emitted by the compressor with a housing are located close to the eigen frequencies of the housing. The locations of the dominant sound intensity amplitudes match the locations of displacement maxima determined by the eigen modes of the housing. In order to stiffen the housing, two ribs were welded to the top of the housing cover. This modification decreased noise amplitude and changed the frequencies of the dominant spectral peaks of noise, while the sound power level of noise radiating from the compressor was decreased from 34.1 dB to 33.1 dB. These results show that a reduction of compressor noise can be achieved by modifying the housing. We believe that more advanced yet inexpensive housing modifications could lead to considerable reductions of noise emitted by a reciprocating compressor in domestic appliances.

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