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Noise Reduction of a High Efficiency Reciprocating Compressor

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ABSTRACT

This paper describes the development of a new reciprocating compressor for HFC-134a employing noise reduction techniques. It is evident that noise increases are mainly at the 500Hz and 2.5~3.15kHz 1/3 octave band sound levels caused by higher efficiency improvements. Among all of the noise factors, 70~80% is due to the suction muffler and compressor shell, as determined by studying noise sources and transmission paths of the compressor’s moving parts.

With respect to the suction muffler factor, the amount of transmission loss at 2.5~3.15kHz 1/3 octave band exerts much influence on sound level. The sound characteristics of the muffler are improved by FEM analysis, and its effect confirmed by noise measurements.

With respect to the compressor shell factor, the resonance characteristics of the shell are analyzed by FEM. In order to reduce the noise due to the shell, the shell stiffness is enhanced.

By using the above techniques, a new reciprocating compressor for household refrigerators with both high efficiency and low noise (by 10dB (A)) is developed successfully.

INTRODUCTION

With increasing concern about protecting the global environment, the requests for energy saving household refrigerators become greater, and the demand for high efficiency compressors in the marketplace will arise. Therefore, we developed a new compressor, with reduced noise and vibration levels without decreasing energy efficiency and without increasing the physical size of the 135~230W (ASHRAE condition) compressor range. Typically, there’s a tendency for the noise level to increase when improving efficiency. If this occurs, the demand for low noise compressors will arise. To resolve the difficulty of noise reduction, this study evaluates the noise sources and transmission paths of the compressor’s moving parts.
NOISE SOURCE

A schematic diagram of a reciprocating compressor for household refrigerators is shown in Fig. 1. Inside the hermetic shell, the motor unit is located in the lower portion of the compressor and compression unit is located in the upper portion. The sound levels of the prototype compressor are shown in Fig. 2. The sound test pressure conditions [R134a] are 1015kPa for discharge and 85kPa for suction. As shown in fig.2, the sound levels are high and mainly at the 500Hz and 2.5k～3.15kHz 1/3 octave bands, with most being at the high frequency bands, generated by higher efficiency improvements. To reduce the sound level of these bands, the noise sources and transmission paths from the compressor's moving parts is investigated. The compressor parts contribution ratio to the sound level is shown in Fig. 3. The contribution ratio of the 2.5k～3.15kHz 1/3 octave band is clear, the suction muffler and shell contributes 70～80% of the total sound level, of which 31% is attributed by shell and 46% is attributed by suction muffler.

SUCTION MUFFLER FACTOR

A schematic diagram of the suction muffler is shown in Fig. 4. The gas enters from the muffler inlet and enters the cylinder through the tail pipe and connecting pipe. The experimental results for the sound transmission loss of the suction muffler with cylinder side noise source are shown in Fig. 5. As shown in fig.5, the transmission loss at 2000～3500Hz is small for the prototype, therefore, it is concluded that the transmission loss at 2000～3500Hz exerts much influence on the 2.5k～3.15kHz 1/3 octave band. The FEM results of the suction muffler’s sound characteristics are shown in Fig. 6. Fig.6 shows the peak and nodal points of the mode in light and dark color. The peak of the mode exists in the open end of the tail pipe at the 2100Hz mode, and the resonance created by the tail and connecting pipes inside the muffler at the 2800Hz mode. By adjusting the length, diameter and position of the tail/connecting pipes, so that the open ends of the pipes can be located at nodal points of the resonance mode inside the muffler, it is possible to improve the transmission loss at 2000～3500Hz as shown in Fig. 5. After using a shorten tail pipe to relocate the open end to the nodal point of the resonance mode, the 500Hz 1/3 octave band increased, in spite of no difference being observed in the transmission loss at 500Hz. Therefore, by maintaining the original tail pipe length and adding an L-shape, and moving the open end to the nodal point of the resonance mode at 2000～3500Hz, the 500Hz 1/3 octave band can be
reduced. This indicates the reduction effect of the 500Hz 1/3 octave band at which the shell cavity resonance occurs due to pulsation from the muffler is determined by the diameter, length of the tail pipe.

Furthermore, the tail pipe has a dampening effect that reduces pulsation from the muffler in addition to the transmission loss of the muffler. On the other hand, the BEM result of the compressor shell cavity resonance is shown in Fig. 7. The peak and nodal points of the mode are in light and dark color. It shows that the shell cavity resonance, occurs due to pulsation through suction muffler, is prevented with the suction muffler inlet location on the nodal point of the 540 Hz resonance mode.

SHELL FACTOR

The experimental results of the resonance characteristic on the compressor shell are shown in Fig. 8. As shown in Fig. 8, the resonance peaks exists at 2800 and 3300Hz for the prototype. These resonance peaks are the causes of the 2.5～3.15kHz 1/3 octave band that mainly contributes to the increased noise. The FEM result of the shell resonance mode for the prototype is shown in Fig. 9. The resonance mode is at 3100Hz, and the nodal point of the resonance mode is located at the flange. This is the location where the upper and lower half shells meet. The peak point of the resonance is located below the flange due to the high stiffness at the flange. This is expected since the shell thickness at the flange is about 2 times that of the upper and lower half shells. Therefore, the resonance peak at 2800Hz shifted to 3000Hz and the peak level reduced, as shown in Fig. 8, as a result of moving the flange near the resonance nodal point. Also, the resonance peak at 3300Hz shifted to 3700Hz. So, by moving the flange location, the shell stiffness is enhanced and resonance peaks improved.

RESULTS OF THE SOUND LEVEL

The experimental results of the sound level for the prototype and new product are shown in Fig. 10. The new product reduced 10～15dB at the 500～630Hz, 2.5k～3.15kHz 1/3 octave band and reduced about 10dB at All Pass sound with the improvement of suction muffler and shell as mentioned above. Although the high frequency sound level at 4kHz and above is also reduced, it is caused by the enhanced shell stiffness that separates the excitation frequency of the compressor's moving parts from the shell's resonance frequency.
CONCLUSION

The factors for increased noise in a newly developed reciprocating compressor for household refrigerators are determined. It is evident that the 500Hz 1/3 octave band sound level is the shell cavity resonance due to the pulsation from the suction muffler, and the 2.5k~3.15kHz 1/3 octave band sound level is the shell resonance determined by transmission loss of the suction muffler.

With respect to the suction muffler, the diameter, length, positions of the tail and connecting pipes inside the muffler are determined by the sound characteristics of the muffler as investigated and analyzed by FEM.

With respect to the shell, the shell stiffness is enhanced by moving the flange, which is the location where the upper and lower half shells meet, as a result of the shell resonance characteristic as investigated and analyzed by FEM.

By using the above techniques, a new reciprocating compressor for household refrigerators with both high efficiency and low noise (by 10dB (A)) is developed successfully.

REFERENCES


Fig. 1  Schematic diagram of reciprocating compressor

Fig. 2  Sound level of prototype compressor
Fig. 3 Contribution ratio of the compressor parts in sound level (2k～3.15kHz band)

Fig. 4 Schematic diagram of suction muffler

Fig. 5 Transmission loss of suction muffler
Fig. 6 Acoustic mode inside suction muffler predicted by FEM model

Fig. 7 Cavity resonance mode of compressor predicted by BEM model
Fig. 8 Resonance characteristic of shell

Fig. 9 Resonance mode of shell at 3100Hz predicted by FEM model

Fig. 10 Sound level of compressor