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NOISE REDUCTION OF HERMETIC COMRESSOR
BY IMPROVEMENT ON ITS SHELL SHAPE

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ABSTRACT

The noise of a small hermetic reciprocating compressor used for household refrigerators and other equipments is mostly radiated by the vibration of the compressor shell. Therefore, an effective approach to the noise problem is to provide a compressor shell structurally less-responsive to the vibration source.

This paper discusses the process in which the reduction of compressor noise has been achieved by restructuring the compressor shell and decreasing the amplitude of vibration.

The current compressor shell whose cross-section forms a circle is not fixed in the direction of principal axis of inertia because of its sectional symmetry. And so it is difficult to restrain the vibration amplitude by changing the support positions of the motor-compressor unit. Moreover, the dynamic stiffness with respect to normal mode of vibration is relatively low, because the shape itself is simple. In the shell with such a simple section shape, all of the forced vibrations close to the resonance frequencies of the shell are amplified effectively. In addition, the superimposition of each vibration mode tends to increase the amplitude. On the other hand, in the case of a shell whose cross-section is asymmetrical, its amplitude can be expected to be lower if all the other conditions are equal.

The writers have experimentally proved the fact that a properly selected cross-sectionally asymmetrical shell is much smaller in its amplitude of the vibration caused by the operation of the motor-compressor unit than the current shell. The application of the asymmetrical shape to the shell has made it possible to reduce the thickness of the shell 1 mm thinner than the current one. Yet the noise of the compressor has decreased by 6 dB(A).

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INTRODUCTION

It is extremely difficult to identify the noise source, particularly in relatively high frequency range, generated during the operation of small hermetic reciprocating compressor used for household refrigerators and other equipments. So there is hard way to work for the reduction of the compressor noise at its source.

The noise source aside, however, the vibration of the compressor itself is mostly propagated via such propagation passes as the suspension springs, discharge tube, refrigerant gas and oil, etc., and eventually make the shell vibrate, resulting in the radiation of noise. Therefore, the shell which is a final acoustic radiator has a direct influence on the compressor noise.

It is pointed out that, as a matter of fact, a spectrum indicating the frequency response of the shell has been quite in accord with that of compressor noise. Consequently, it can be expected that an efficient method to reduce the compressor noise appreciably by improving the shape of the shell is to provide a shell structurally of small amplitude against the same vibration source. This study mainly tries to reveal that, in the shell whose cross-section is circular, all of the vibrations are amplified very effectively because of its simple shape and that, in fact, much of compressor noise has been reduced by reconsidering the shell shape.

CHARACTERISTICS OF CURRENT COMPRESSOR NOISE AND SHELL VIBRATION

The compressor shell having a shape, as shown in Fig. 1, now being widely used for household refrigerators has picked up for experiment. As seen from Fig. 1, the shell perfectly circle in cross-section is very simple in shape. The sound pressure level spectrum (Fig.3) of the compressor used this type of shell, and the shell's driving
point mechanical compliance (Fig. 4) have been measured each at the position as shown in Fig. 2. The frequency response of the shell has been measured with the motor-compressor unit resiliently supported under the normal condition.

The peak frequencies of the compressor noise spectrum are in full accord with the peak frequencies of the shell vibration on the frequency response curve. From this it is clear that the compressor noise characteristic mostly can be determined by the shell vibration characteristic.

Now, the first resonance frequency lies within a relatively low frequency range, 1900 Hz - 2000 Hz, in a frequency response curve of the shell. Besides, most of the resonance peak levels are high within the measured range. Like this, the dynamic stiffness of this shell is considerably low.

It is possible to reduce the number of resonance frequencies in the audio frequency range, for example, by giving curves to the shell at its straight portion or spherical surfaces at its plane section to increase the dynamic stiffness as a result. However, it is hard to reduce its peak level by the method as mentioned above.

VIBRATION MODE OF CURRENT SHELL

Fig. 6 shows the driving point mechanical compliance spectra of the current shell measured at three different positions, as shown in Fig. 5. They are very similar to one another. Namely, this shell has identical dynamic stiffness at all positions in a cross-section. This means that the direction of inertia principal axis is unsteady, because the shell is axisymmetry in a cross-section, and because all of the normal modes of vibration are amplified very effectively.

It is possible to reduce the number of resonance frequencies in the audio frequency range, for example, by giving curves to the shell at its straight portion or spherical surfaces at its plane section to increase the dynamic stiffness as a result. However, it is hard to reduce its peak level by the method as mentioned above.

AMPLIFICATION BY SUPERIMPOSITION OF VIBRATION MODES

As mentioned above, in the current shell whose shape is axisymmetrical in a cross-section, its amplitude is increased by the superimposition of identical vibration modes. On the contrary, the experiment shows that in the asymmetrical shell having plane surfaces at A and C portions as shown in Fig. 9, such an effect seldom occurs.

Fig. 11 illustrates the principal normal mode vibration levels of both current and new shells at position C when each of the shells is excited at different positions A and B in the cross-section (Fig. 10). All vibration normal modes of the current shell are effectively amplified with the exciting positions as a loop. Therefore, when the shell is excited at A and B positions at the same time, each identical vibration mode excited effectively is superimposed each other, resulting in considerable increase in amplitude. Then, as shown in Fig. 11, at 2000 Hz, the vibration level at position C decreases in the in-phase excitation, but increasing in the anti-phase excitation. Likewise the vibration level changes at 2500 Hz and 3900 Hz.

On the contrary, in the case of the shell whose shape has been changed to be asymmetrical in a cross-section, the direction of inertia principal axis is fixed and because of this the positions of modes of almost all vibration normal modes are determined. So, each of the vibration normal modes has its own effective excitation position. For this reason, even if they are excited simultaneously at different positions, the superimposition of the same modes effects little change in amplitude, because the natural frequencies excited effectively at their respective positions are different. As shown Fig. 11, the vibration levels at C position hardly change even when excited at A and B positions.
As mentioned above, if the shape of the modes by the finite element method has amplitude of the vibration is hardly determined by themselves. As a result, the amplification of the vibration is hardly given rise by the superimposition of the same modes, nor is it possible to excite all of the vibration normal modes effectively at any given position.

The examination of the vibration normal modes by the finite element method has revealed that the vibration modes higher than 5 kHz mostly come from the deformation of the shell in vertical direction. For the deformation in vertical direction, the telescope welding is preferable to the flange welding as in the current shell. Besides, the stiffness becomes higher by making all phases of the shell spherical. Taking the abovementioned into consideration, we have selected a completely new shape of the shell, as shown in Fig. 12. It is made asymmetrical not only in the cross-section but also in the vertical section. Moreover, all phases of this shell are made spherical except for the surface on which the relay box is fixed.

Characteristics of New Compressor Noise and Shell Vibration

The new shell in Fig. 12 is 1 mm thinner than the current shell and houses the same motor-compressor unit as used in the current shell. In this new shell, the spectrum of the driving point mechanical compliance changes according to the change of measurement position. Namely, because of fixed node positions of the vibration modes, each loop of the vibration normal mode is not concentrated on an exciting position and is dispersed in the fixed positions, different from the current shell. Accordingly, the dynamic stiffness of every position of this shell is extremely high and the amplification of the same modes is hard to occur on the shell.

Fig. 13 shows the spectrum of the driving point mechanical compliance of this shell at the position in Fig. 12. In this frequency response curve, peak levels of the resonance are generally low and the first resonance frequency is over 3 kHz. So, the dynamic stiffness of this shell has become extremely high. On the other hand, the sharpness of the resonance peak is reduced and so the damping is seemingly effective on this shell.

As a result, the shell vibration caused by the same motor-compressor unit is very small in comparison with current shell and the acoustic energy radiated from the shell is reduced, then compressor noise level is also reduced considerably. Fig. 14 shows the sound pressure level spectrum of this compressor at the measurement position shown in Fig. 12. The noise level of this compressor is lower 6 dB(A) than the current one. In addition, it is worthy of our notice that the new shell is 1 mm thinner than the current one.

Conclusion

Control of the shell vibration characteristic is very effective to reduce the hermetic compressor noise.

In a shell which has a symmetrical shape in a cross-section, because of its symmetry, the direction of the inertia principal axis is not fixed. For that reason, all of the forced vibration close to the resonance frequency are effectively amplified.

On the other hand, because a proper asymmetrical shape in a cross-section determines the direction of the inertia principal axis, the node of the vibration normal mode is fixed. Thus, among all forced vibrations close to the resonance frequencies, those effectively amplitude are quite limited in number. In addition, the amplification of the same vibration modes is also hard to occur.

In consequence, for the same motor-compressor unit, the vibration of the shell induced by the compressor operation is very small in comparison with a shell whose cross-section is symmetrical. So, the acoustic energy radiated from the shell is reduced, and then radiation noise from compressor is greatly reduced.

Especially, as far as we judge from the experimental results, the damping is seemingly effective. This phenomenon may be interpreted as follows: Since the shell repeats deformation by its vibration, the direction of inertia principal axis is slightly moved. Particularly, in the neighborhood of resonance frequencies, its proportion to increase in deformation of the shell, the change of direction of inertia principal axis becomes effective. Thus, the normal mode of vibration also changes, therefore, the forced vibration is hard to amplify effectively.

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Fig. 1 Shape of current shell

Fig. 2 Measurement positions of current shell compressor noise and its shell driving point mechanical compliance.

Fig. 3 Sound spectrum of current compressor in narrow band analysis.

Fig. 4 Relation between current shell driving point mechanical compliance and frequencies.
Fig. 5 Measurement positions of current shell driving point mechanical compliances in cross-section.

Fig. 6 Relation between current shell driving point mechanical compliance and frequencies at position A, B, and C in cross-section.

Fig. 7 Vibration modes of current shell, when excited by shaker.
Fig. 8 Current shell normal mode of vibration calculated by finite element method.

Fig. 9 Shape of asymmetrical shell for experiment using two shakers.

Fig. 10 Excitation and vibration measurement position for experiment using two shakers.

Fig. 11 Vibration levels at position C in Fig. 10 when excited by two shakers.
Fig. 12 New shell shape

Fig. 13 Relation between new shell driving point mechanical compliance and frequencies.

Fig. 14 Sound spectrum of a third-octave Band analysis.