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DEVELOPMENT OF A LOW NOISE ROTARY COMPRESSOR

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ABSTRACT

We developed a rotary compressor for use in domestic air-conditioners having low noise, high efficiency and high reliability. In developing this rotary compressor, noise generation mechanism having three factors, noise generating source, noise transfer path, noise radiation, was analyzed and noise reduction in the respective factors was studied. As improved specifications, we optimized a silencer and a discharge route in the noise generating source, developed a new method for fixing the compression elements to the casing in the noise transfer path, and reviewed rigidity of the constituent elements of the casing in the noise radiation, and as a result, we realized a new low noise rotary compressor of the globally top level.

INTRODUCTION

The global demand for air-conditioners has been dominated by the USA and Japanese markets, but recently the market for air-conditioners has rapidly grown in other parts of the world. Especially the prevalence of air-conditioners in the Asia and the Chinese has sharply increased, and window type air-conditioners, whose ready installment is preferred in view of housing circumstances, are prevalent.

Window type air-conditioners are installed near living spaces, and their operating noises directly reach their users. Reduction of these noises is increasingly required of the window type air-conditioners. Among such noises, noises caused by the mounted compressors occupy not low ratios. In addition, energy-saving is also required in view of electric power supply circumstances, legal restrictions, etc. Thus compressor are with low noises and high efficiency required by the market.

Here, the noise reducing technique developed for use in the low noise rotary compressor will de presented.

PRODUCT CHARACTERISTIC

The cross-section of the rotary compressor designed for use in air-conditioners in a 2500 W rated cooling capacity class is shown in Fig. 1. The motor stator is fixed directly to the casing, and the compression elements are fixed to the casing through the frame, whereby deformation of the precision parts are prevented to thereby ensure high efficiency, and furthermore the thrust supporting structure is provided by the lower end of the crankshaft. As a result, high reliability is realized. In addition, the ultra-small-sized overload protector is incorporated in the casing, which permits the air-conditioner to be operated without trouble under any electric power circumstances of countries the air conditioner is operated.

NOISE REDUCING TECHNIQUE

1. Sound power level

The result of a 1/3 oct. band analysis of the rotary compressor before improved, which was an object of the noise reduction is shown in Fig. 2. The frequency bands which contributed to deterioration of overall value of this rotary compressor were 800 - 1600, 2500, and 4000 Hz band.

Then, to make clear the radiation positions and directions of the respective frequency bands, power levels were given by sound intensity methods. This rotary compressor had a higher power level by more than 10 dB at the compressor body than at the accumulator unit, and so the compressor body alone was to be improved.

Fig. 3 shows ratios of the respective radiation positions for the respective frequency bands.

What should be noted here is as follows.

(1) The ratios of overall value for the following three parts are about 50% for the side of the compression elements, 30% for side of the motor stator, and the remainder for the top of the casing.
(2) The highest radiation positions for the following frequencies is noted.
  a) 800 Hz band : the side of the motor stator
  b) 1000 - 1600 and 4000 Hz band : the side of the compression elements
  c) 2500 Hz band : the top of the casing

2. Noise generating source

The refrigerant gas flow path of this rotary compressor comprises a suction port, a compression chamber (a cylinder), a discharge port of the cylinder, a muffler, a discharge port of the casing. Refrigerant gas pressure pulsation at suction port is small, and the constituent elements of the compression chamber have high rigidity. Accordingly, due to refrigerant gas pressure pulsation at the suction port are very few.

To study relationships between compression timings and pressure pulsation in refrigerant gas flow path : inside the cylinder, the muffler and the casing, short time frequency transfer spectrum analysis was conducted. Fig. 4 shows the result of frequency analysis of one round compression stroke. The occurrence timings of major noise frequencies are synchronous with the start of discharge from the cylinder. Here, among these frequencies, the objects to be improved were limited to the following two typical points.

(1) The pressure pulsation generated inside the cylinder which rose from 1000 Hz to 5000 Hz caused by the change of compression volume. Among these, the pressure pulsation in 4000 - 5000 Hz was amplified toward the muffler, the interior of the casing. Therefore, it effective and significant to make countermeasures to the noise generating source.
(2) The noises near 1600 Hz are very similar to the generation pattern of the pressure pulsation in the muffler, and the noise are caused by pressure pulsation in the muffler.

For the noise reduction at the noise generating source,

(1) A silencer is provided inside the compression chamber as means for improving the noise generating mainly at 4000 - 5000 Hz
(2) The transfer length inside the muffler is changed as means for improving the noise generating mainly at near 1600 Hz

As the silencer to be disposed inside the compression chamber, the use of a Helmholtz type resonator was discussed. The resonator, however, must have large dimension for pressure pulsation of low frequencies, and adversely has a larger top clearance volume which will lead to COP drop. It was noted that the Helmholtz resonator was used for pressure pulsation of high frequencies. Fig. 5 and 6 shows relationship among resonance frequency (which changed by volumes of the resonator chamber) , noise attenuation amounts and COP drop amounts. Based on this result, the resonance frequency is set in 4000 Hz, because the effect of most decreasing the noise is obtain and the COP drop could be minimized in this case.

As concerns specifications of the muffler, mufflers having different transfer length were prepared beforehand, and driving sounds were measured in the air on simplified models. The measured driving sounds were converted into ratio of sound velocity, and the sound velocity ratios were taken as driving sound in the refrigerant gas, and those of the mufflers which are good in 1600 Hz band were chosen.

3. Noise transfer path

The main movable compression elements of rotary compressor consist of a crankshaft, a roller, a vane. The three elements are moved integrally with one another in a small gap between a bearing and a cylinder, lubricated by oil. The cylinder is mechanically fixed to the casing through the frame. Small excitation forces of three elements are easily transferred to the casing. Transfer frequencies of the excitation forces are substantially equal to natural frequencies of the compression elements.

In clearing the noise transfer path, first, actions of the compression elements in actual operation were measured and studied in connection with the transfer elements. According to this study, in the 2500 Hz band, where the power level is highest, it was confirmed that compression elements are moved by a resultant force from a force in the horizontal and vertical direction.

The compression elements with the motor rotor is secured to the frame, frequency response function of the respective elements at their response points were given by modal analysis. The cylinder was the reference point, and the respective elements were moved to give their response points. According to the result of the modal analysis, the vibrations of frame are larger than those of the compression elements in a wide range of frequencies expect 1350 and 1390 Hz. To give a typical example, the vibration mode at 2780 Hz represents in Fig. 7. It is found that the frame is more deformed than the compression elements. Also according of the sensitivity analysis, based on modal parameter given by a modal analysis, it found that the frame needs improvement in strength.

For the rigidity improvement of the frame, an optimum shape of the frame was discussed based on computation using a finite element model. The points of the shape of the frame were centered on (1) plate thickness, (2) fixed point number and positions of the casing and (3) mounting direction. The shape shown In Fig. 8 has the most improved natural frequency was given. And fig. 9 shows the comparison of frequency response function level between the conventional and the improved frame. It was found that the vibration level at all the frequencies around 2500 Hz band and the strength improvement was achieved by incorporated the improved frame.

4. Noise radiation

Based on the result of a running mode analysis in which vibration patterns of the respective elements of the casing in actual operation, points of the casing to be reinforced were studied. The vibration patterns of the casing of this rotary compressor are classified as follows.

(1) 200 - 500 Hz : Whirling of the casing itself and vibration of the motor stator
(2) 800 - 1200 Hz : Vibration of the base
(3) 1200 - 1700 Hz : Vibration of top and bottom of the casing and motor stator
(4) 1700 - 2500 Hz : Vibration of top and bottom of the casing and the base
(5) above 2500 Hz :Vibration of top and bottom surface, compression elements and base

Based on this result, as made in the noise transfer path, sensitivities of the rigidities of the elements to frequencies were computed, and it was found that reinforcement of the casing can be effectively conducted in the following sequence.

(1) Bottom of the casing : Substantially all range of 800 - 5000 Hz
(2) Top of the casing : 1000 - 1600 Hz and 2400 - 2800 Hz
(3) Base : Substantially all range of 800 - 5000 Hz
(4) Sides of the casing :
   a) Motor stator : below 860 Hz
   b) Compression elements : 3300 Hz and above 5400 Hz

The thickness of the bottom of the casing was increased to the balance of the casing as a whole, whereby the noise was decreased.

5. Summary

The improvement items in connection with the noise generating source, noise transfer path and noise radiation were incorporated in the rotary compressor, and as shown in the 1/3 oct. band analysis of Fig. 10, the noise was reduced by about 10 dB in overall value.

CONCLUSION

We made clear the noise generation mechanism of the rotary compressor, which includes three factor, noise generating source, noise transfer path and noise radiation by the following methods. Based on the results of these analysis, preference of the noise reduction can be given to the parts of the rotary compressor, and parts to be improved effectively to reduce noises can be located.

(1) Noise generating source : Short time frequency transfer spectrum analysis
(2) Noise transfer path : Modal analysis
(3) Noise radiation : Sound intensity method and running mode analysis

Based on the located improvement points, we developed the following improvement items.

(1) Noise generating source: Optimization of the silencer and the discharge path
(2) Noise transfer path : Development of the new fixing method of the compression elements to the casing
(3) Noise radiation : Review of rigidity of the casing

By incorporating these improvement items, we have successfully realized a low noise rotary compressor of globally top level.

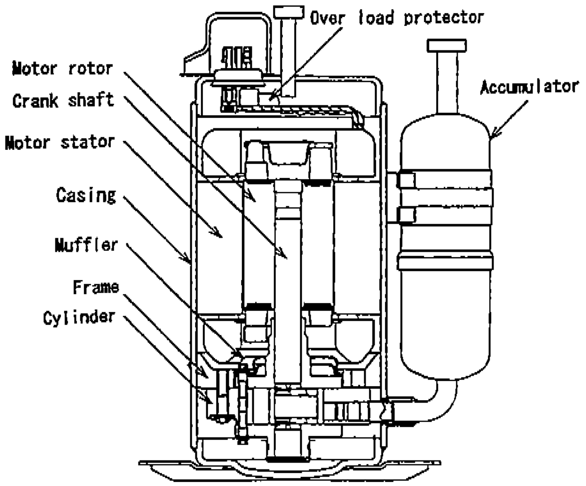

Fig.1 Cross-section of newly developed rotary compressor

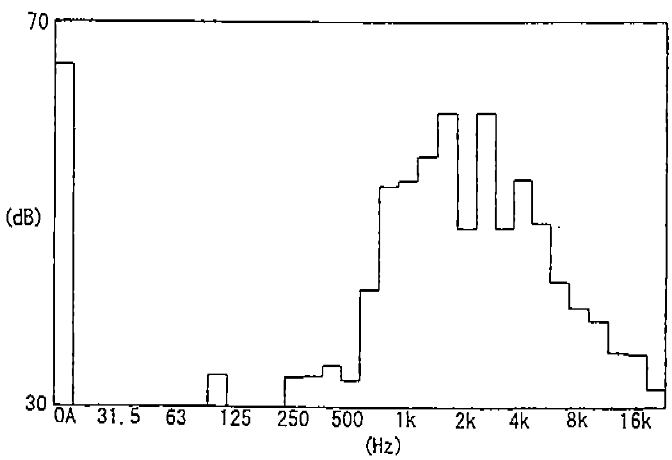

Fig.2 Noise power lebel (conventional)

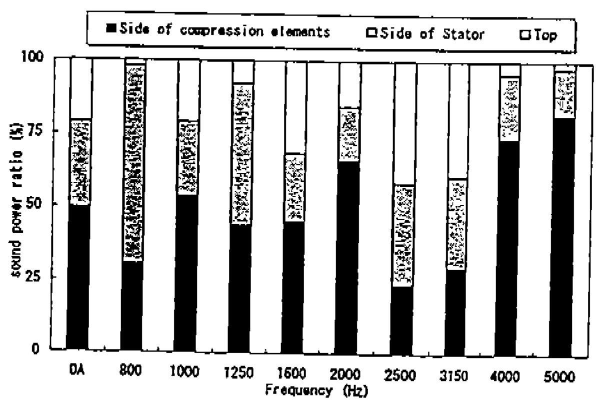

Fig.3 Sound power ratio (conventional)

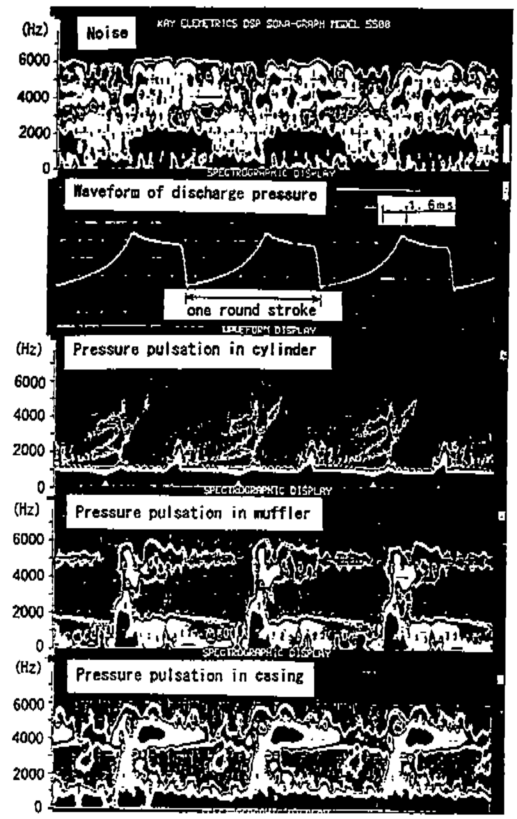

Fig.4 Analysis of pressure pulsation (one round compression stroke)

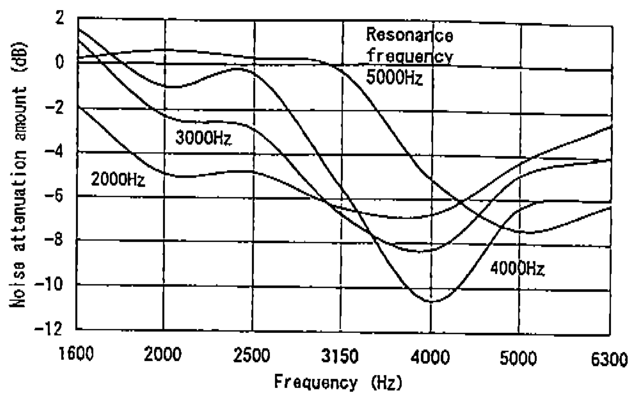

Fig.5 Resonance frequency and noise attenuation amount

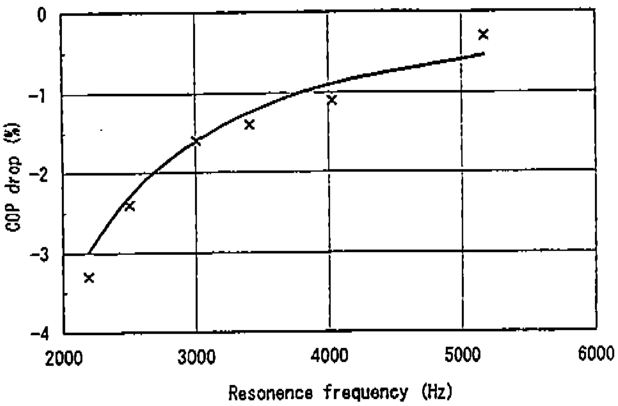

Fig.6 Resonance frequency and COP drop

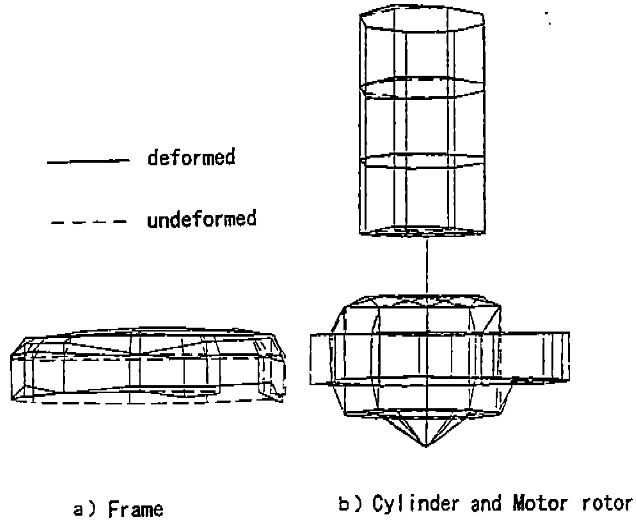

Fig.7 Mode animation (at 2870 Hz)

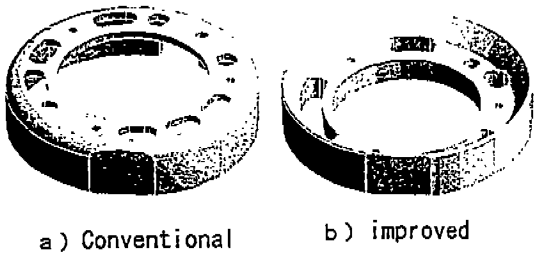

Fig.8 Shape of frame

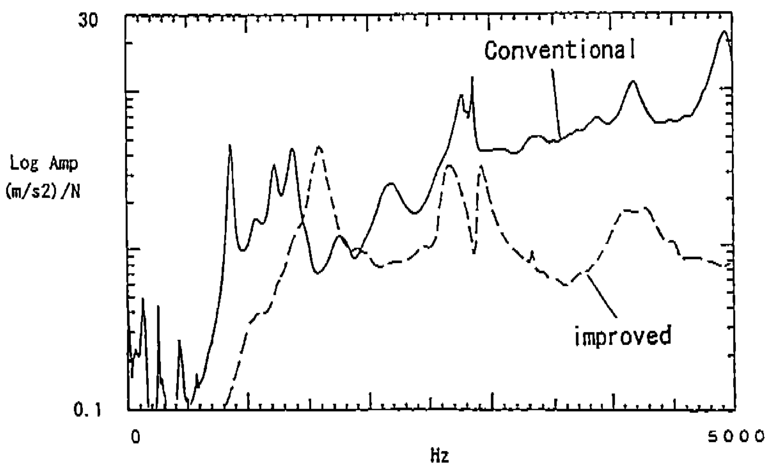

Fig.9 Comparison of frequency responce function

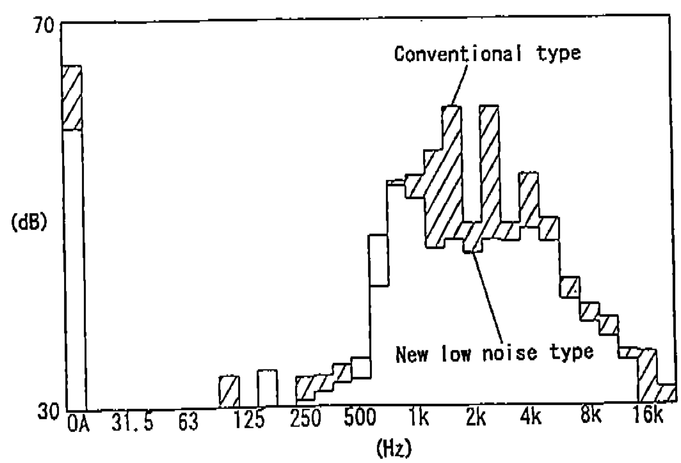

Fig.10 Comparison of power lebel