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ABSTRACT

Compressor is a major noise source of air-conditioner. Especially, its low frequency band noise below 1000Hz is very important because it will not be attenuated by passing through the cover panel and heat exchanger in air-conditioner. The factors affecting the low frequency band noise are studied by geometric similarity along with several experiments, and the low frequency noise is closely related with the discharge holes of muffler as well as the cavity of lower shell. The low frequency band noise is significantly reduced by proposed designs.

CHARACTERISTICS OF LOW FREQUENCY BAND NOISE

Sound tests are carried out for compressors with different frame sizes, which are 15 frame with capacity range of 1465 through 4000W, 20 frame of 3500 through 6300W, and 30 frame of 6150 through 8550W. Their sound spectra measured in an anechoic room at ARI condition, 6.37kgf/cm^2 suction , 21.87kgf/cm^2 discharge pressure is, and 18.3℃ suction , are given in Figure 1. There is a strong pure tone below 1000Hz in each frame; they are 750Hz in 15 frame, 650Hz in 20 frame and 570Hz in 30 frame. Considering the fact that the pure tone shifts to the higher frequency by the change of suction gas temperature, it may well be related with the cavities inside compressor.

SOUND INTENSITY FROM COMPRESSOR SHELL

Narrow band sound intensities are measured over the surface of 15 frame-compressor shell including accumulator. Contour map on the pure tone below 1000Hz is generated on the surface of compressor shell only. Figure 2 shows the intensity contour over the compressor shell at 750Hz, which is the dominant frequency in the sound pressure spectrum in Figure 1. In addition, the intensity contour map seems to be caused by the di-pole cavity mode, which occurs inside compressor shell. Considering high radiation efficiency of di-pole mode, it should have an important effect on the pure tone noise at 750Hz.

CAVITY MODE ANALYSIS USING FEM

The finite element method (FEM) is used to predict the cavity modes in the cavity inside compressor shell for 15, 20 and 30 frame. The shape of di-pole mode for 15 frame is given in Figure 3, which is same as the shape of intensity contour map. This shows that internal cavity with di-pole mode should be closely related with the pure tone noise below 1000Hz.

GAS PULSATION INSIDE COMPRESSOR

Gas pulsations are measured using pressure transducers inside compression chamber cavities, inside muffler, and compressor lower shell. Their spectra for 20 frame compressor at ARI condition are given in Figure 4. The spectrum of gas pulsation inside cylinder is smoother than that of the others below 1000Hz although it has much higher magnitude. As gas flows from the compression chamber to the lower shell cavity passing through muffler, the magnitude of the gas pulsation goes down but it has a peak at around 650Hz inside lower shell. In addition, the peak of the gas pulsation inside lower shell cavity has the same frequency as pure tone noise below 1000Hz in 20 frame compressor.

There are several methods to reduce the low frequency peaks of gas pulsation inside lower shell. One of them is to prevent low frequency cavity modes inside lower shell, and another one is to design muffler which can attenuate the gas pulsation with low frequencies. The latter is much easier to implement.

SOUND INTENSITY FROM MUFFLER

To see how sound is transmitted through a muffler, narrow band sound intensities over the top of muffler are measured by exciting the muffler cavity with white noise. The white noise with frequency band up to 4000Hz in air is supplied through discharge port in the upper journal bearing as in Figure 5. The intensity tests are performed in air and converted to R22 at ARI condition considering the speed of sound. Existing mass production mufflers in each frame are tested and some of intensity maps are given in Figure 6.

Sound from the muffler radiates through the following paths. One of the paths is the clearance between muffler and the hub of journal bearing, and low frequency band noise below 1000Hz is mainly transmitted through this path. Another path is the discharge hole of muffler. The low frequency pure tone noise, which is related with the strong pure tone noise below 1000 Hz in Figure 1, radiates through this path. Also, the high frequency sound over 1500Hz mainly radiates through the path. The other path is the surface of muffler that radiates sound by the vibration of muffler surface due to the cavity modes inside muffler, and the sound passing through the path is mainly in 800Hz ~ 1000Hz range.

Therefore, in order to attenuate the low frequency band noise, it is important 1) to reduce the clearance between the center hole of muffler and the hub of journal bearing as well as 2) to optimize the location of muffler discharge holes. A new muffler with two holes that face each other in the opposite direction with 180 degree out of phase is proposed. And, the muffler has smaller clearance between the center hole of muffler and the hub of journal bearing than that of the existing muffler.

Sound intensities for the newly designed mufflers in each frame are measured, and some of intensity maps are given in Figure 6. While both mufflers have pure tones at very close frequencies, the intensity contour maps are very different at the close frequencies. Also, the newly designed muffler has much lower intensities below 1000Hz than the existing muffler.

VERIFICATION TEST OF NEW MUFFLER

Narrow band sound intensities over the surface of 15 frame compressor shell with the new muffler are measured at ARI condition. Contour map on the low frequency band below 1000Hz was then generated on the surface of compressor shell only. Figure 7 shows the intensity contour over the compressor shell at 750Hz that is still dominant frequency. The measurements were made using the existing mass production muffler. The shape of the intensity contour seems to be di-pole mode, but it is different from the shape of intensity contour shown in Figure 2 and its magnitude is reduced significantly.

Finally, narrow band sound spectrum of compressors with the newly designed muffler are measured at ARI condition. And, they are compared with the spectra of the compressors with the existing muffler.
Figure.8 shows that low frequency band noises below 1000Hz is significantly reduced for all frame sizes.

CONCLUSION

The factors affecting the low frequency band noise are examined by geometric similarity along with several experiments. In this study, the low frequency band noise is closely related with the muffler as well as the cavity of lower shell. While it is hard to change the cavity of lower shell, it is relatively easy to change the design of muffler. Therefore, in the study of the mufflers, it was found that low frequency band noise below 1000Hz is affected by not only the clearance of the center hole of muffler and the hub of journal bearing but also the location of muffler discharge holes. By adopting the newly designed muffler, the low frequency band noise is significantly reduced.

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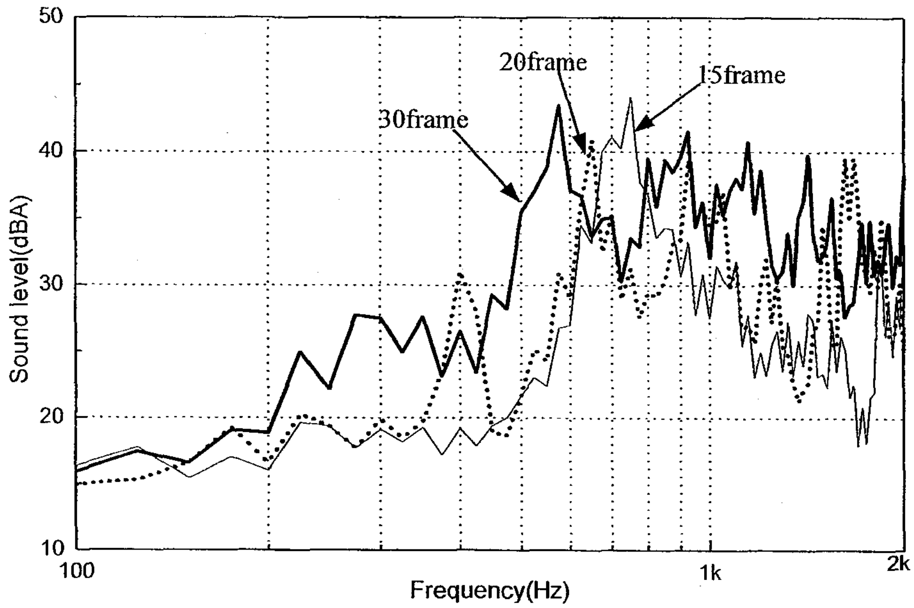


Fig.1 Noise Spectra of Rotary Compressor

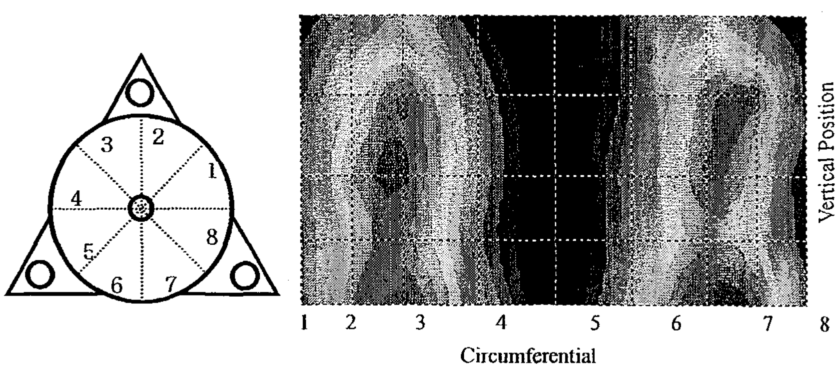


Fig.2 Sound Intensity of 15 Frame Compressor at 750Hz.

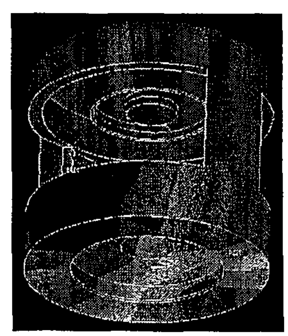
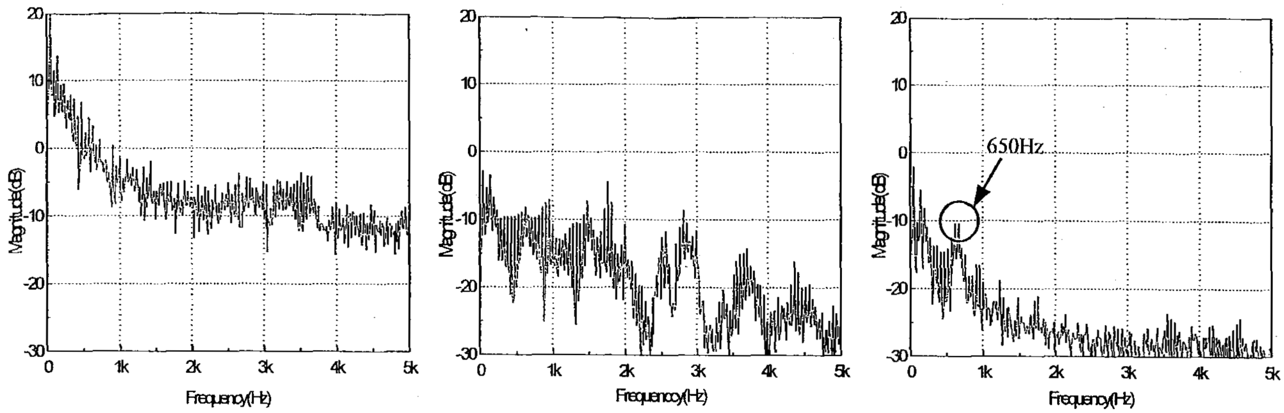


Fig.3 Cavity FEM Analysis of 15 Frame Compressor at 700Hz



(a) Compression chamber (b) Muffler Cavity (c) Low Shell Cavity

Fig.4 Gas pulsation Spectra

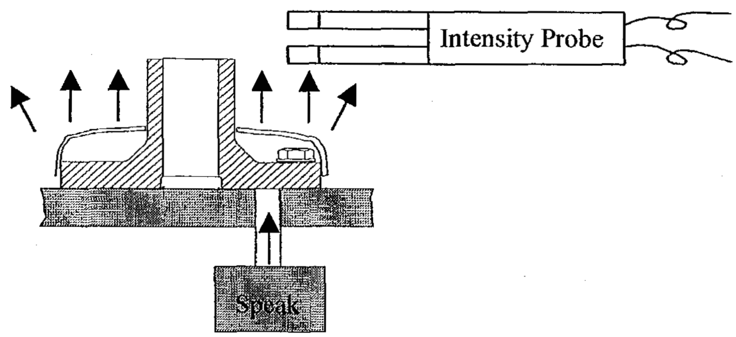


Fig.5 Schematic Test Set-up for Muffler Bench Test

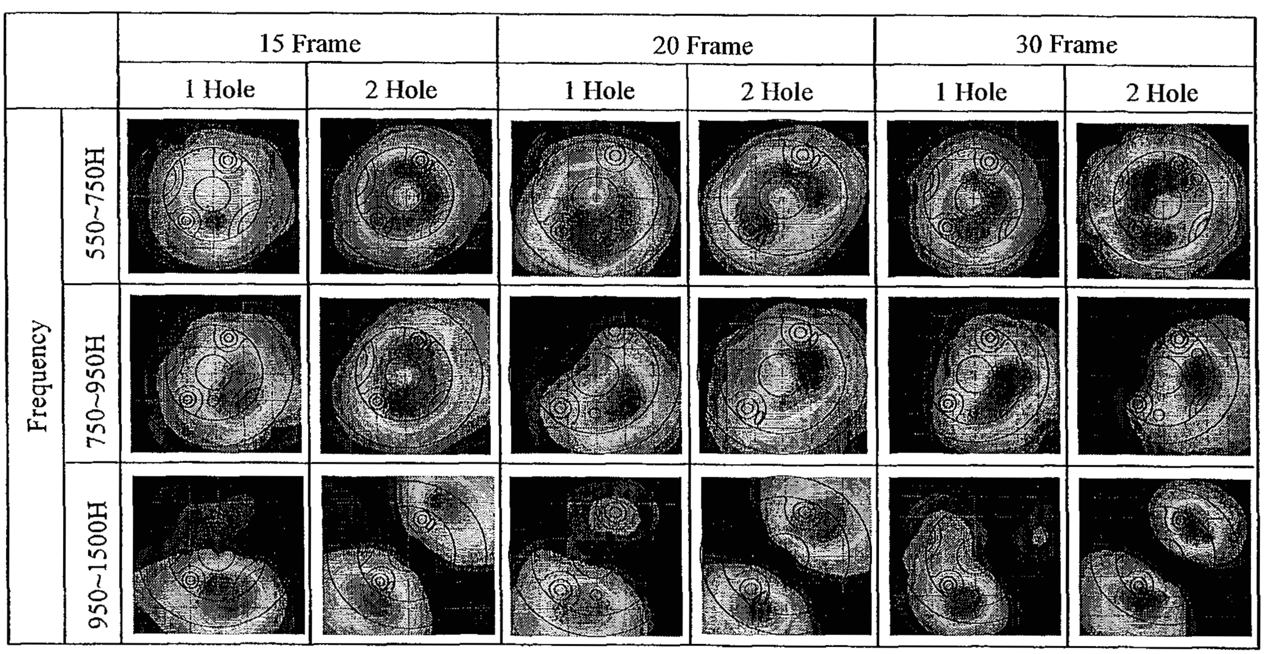


Fig.6 Sound Intensity of Discharge Muffler

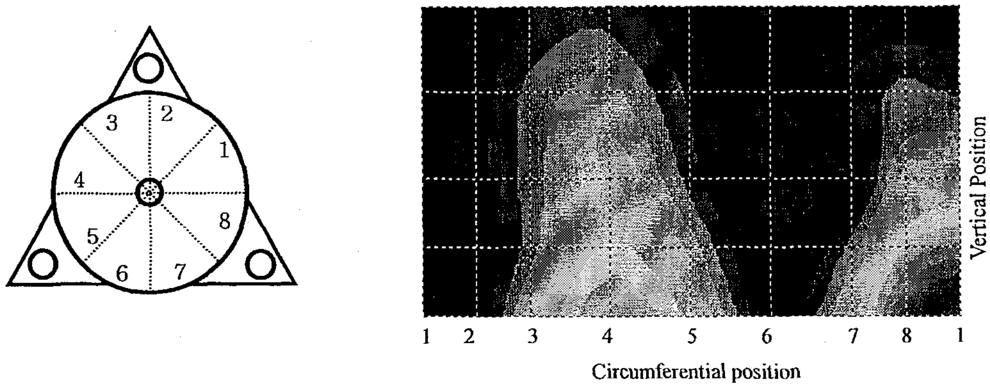
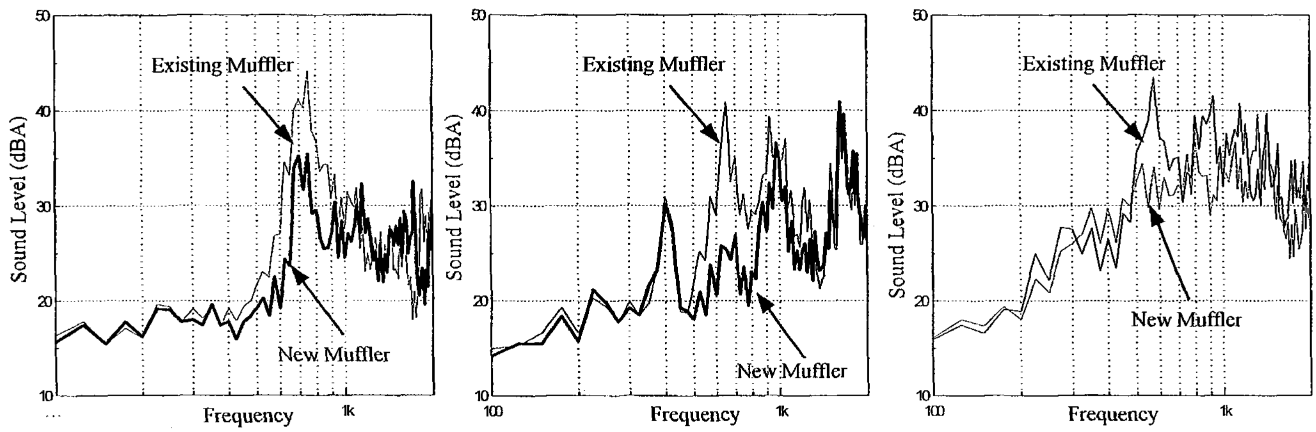


Fig.7 Sound intensity of 15 Frame Compressor newly designed Muffler adopted at 750Hz



(a) 15 Frame (b) 20 Frame (c) 30 Frame

Fig.8 Noise Spectra of Compressor with the Newly Designed Muffler