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1. ABSTRACT

The noise of a small heremetic compressor used for household refrigerators and other equipment is mostly radiated by vibration of compressor shell. Therefore, an effective approach to the noise improvement is to provide a compressor shell structurally less responsive to the vibration source.
By applying finite element method (FEM) and MODAL analysis techniques to the steel compressor shell, we obtain these two findings:
1. The node of a principal mode on a spherical surface shall be fixed regardless of curvature and is located on the same center angle.
2. Strain hardening exponential(n-value) and Lankford value (r-value) of the steel material especially effected the sound radiated from the compressor shell.
The compressor noise level is reduced about 3 dB(A) by using these two characteristics.

2. INTRODUCTION

Environmental noise has significant mental and physical impact on human comfort. More attention has been focused on the noise characteristics than other characteristics of compressors. In general,compressor for refrigerator has two types, one is reciprocating piston types, and the other is rolling piston type rotary compressor. Table 1,2 and Figure 1,2 show noise generation diagram and mechanical constructions of each type. In the case of reciprocating piston type compressor, the compression mechanism itself is the principal source of noise. Noise generated by a compression mechanism may propagate through the refrigeration gas in the shell and radiate externally through the shell. Vibration from the compression mechanism may resonate with compressor components, thus transmitting resonant noise externally via the shell. The vibration may also propagate to the shell via the compressor discharge and suction system and support system and resonate with the shell, thus transmitting resonant noise from shell. The noise generation mechanism of rotary compressor is somewhat a different from the case of reciprocating piston type compressor. It is like the noise generation mechanism of a single phase induction motor. This is due to direct mechanical fixing of the motor stator and bearing on the compressor shell. In the case of rotary compressor, the following motor bearing properties might have considerable effects:
1. Fixing of the motor bearing on the case shell
2. Rotor shaft and bearing rigidity.
3. Clearance between shaft and bearing.

These three factors generate 4 to 6 kHz sliding noise and noise at the natural frequency of the bearing and case shell system. It was made clear that the shell could have a substantial effect on compressor noise.
The transfer function of a shell and noise of the compressor are shown in Fig.3 for comparison. It is clear that the natural frequencies of the shell are reflected on the compressor noise characteristics. It is therefore considered necessary to design a highly rigid shell to reduce compressor noise. This report summarizes the result of basic experiments concerning the shell shape and noise interrelation as well as shell steel materials and noise interrelation.

3. SHELL CURVATURES AND THE NATURAL FREQUENCY OF SHELL

A compressor shell, in general, consist of continuous curved surfaces. There are some blend point of different curvatures on the shell surface. Surface curvatures is therefore very important shell design criteria. The natural frequency and mode have been calculated and experimented on for the models in Table 3 and Fig.4 using FEM method and MODAL approach. The models are three variants having the same flange of the same diameter but width 125mm, 75mm and 68.8mm shell diameters. Model R1 is flatter than model R2 and R3. Model R1 has 20mmR curvature at the joint of the flange. Results of MODAL experiments and FEM calculations are shown in Fig.5 and Table 4. Principal mode is irrelevant to surface curvatures and is produced by top plane movement in the vertical direction. Fig.6 shows curvature vs. principal natural frequency. Characteristic values tend to become saturated at higher curvatures. It has therefore been concluded that a curvature increase would not help increase principal natural frequency, though curvature increase may increase shell rigidity and shell size. Fig.7 shows model cross sections of MODAL experiments in principal mode. Principal mode is the oscillation in the direction vertical to the shell top surface with two nodes in the center. Feg.8 shows the locations of two nodes in respect to curvature center of models. Nodes of model R2 and model R3 are located on the same angle at the center of each model curvature as shown in this figure. Model R1 nodes are located at the joints of the 125mmR curvature surface and 20mmR curvature surface. It is thus concluded that;

1. The node of a principal mode on a spherical surface shall be fixed regardless of curvature and is located on the same center angle.
2. When the point of contact of curvatures are located within the center angle as noted above, those points will be the nodes of the natural frequency mode.

It is then possible to design a highly rigid shell by taking advantage of the above findings to optimize curvature vs. locations of curvature blend points. A practical example of an improved shell design is shown Fig.9. In this Figure 1,2,3 and 4 represent FEM result, transfer functions, result of MODAL experiments and result of compressor noise measurement respectively. It is then possible to design a highly rigid shell by taking advantage of the above findings to optimize curvature vs. locations of curvature discontinuity points.

A practical example of an improved casing shell design is shown in Fig.9. In this Figures 1,2,3 and 4 represent FEM estimation, transfer functions, results of MODAL experiments and results of compressor noise measurement respectively. Model A and model B respectively represent a present shell and improved shell. Model A has 2.1 kHz 2.72kHz resonance frequency at its shell bottom surface and shell top surface respectively. The improved shell design by using of above findings factor gets the maximum space and shifts resonance frequency. As shown in Figure, the 2. 1kHz at the shell bottom surface is shifted to 1.7kHz by increasing damping. Likewise, the 2.7kHz at the shell top surface is shifted to 5.26kHz. Compressor noise showed a decrease of 6dB. Thus, it has been confirmed that natural frequencies are controllable within a certain limit by taking advantage of the 3 points explained above.

4. SHELL MATERIAL CHARACTERISTICS AND COMPRESSOR NOISE ☆

The compressor shell material has been often reviewed from various points of view. The shell iron material characteristics and compressor noise have been studied very little, however. So this item introduce the relationship between shell steel material characteristics and compressor noise. Steel plate characteristics are represented in general by five parameters as listed in Table 5. So five sample steel plates having different combination of those parameters were screend for producing compressor shells to test compressor noise. Fig.10 shows sample numbers, each parameter values, transfer function curves of the shells so produced, and measured compressor noise. The differences in material characteristics are irrelevant to the transfer function curves at each shells. Representative oscillation mode is shown in Fig.11. But substantial differences have been found in compressor noise depending on the shall steel material. It has been thus concluded that material characteristics have substantial influence on compressor noise. However, there is still a question with respect to the interrelations between material parameters and compressor noise. Fig.12 shows the relationship between EL, TS, and YP parameter s and compressor noise level. It is clear that those parameters are least interrelated to the compressor noise level. Fig. 13 shows $n \times r$ values in contrast with the compressor noise level and it is clear that a proportional interrelation exists between compressor noise level and $n \times r$ values. Fig.14 shows $n \times r$ values in contrast to mechanical properties of steel plates. The rate of decrease in plate thickness (preforming plate thickness-postforming plate thickness / preforming plate thickness) and hardness are closely interrelated with $n \times r$ values. Fig. 15 shows $n \times r$ values in contrast with the shell damping. Damping is also closely interrelated to $n \times r$ values. It is concluded that by increasing $n \times r$ values, the rate of plate thickness decrease during shell forming increases, thus damping also increases so that the noise level can be reduced even for shell which has the same natural frequency. It is considered possible to reduce the compressor noise level by controlling $n \times r$ values.

☆ Cooperation was offered by Messrs. SUMITIMO METAL INDUSTRIES Co., Ltd with respect to this item.

5. CONCLUSION

Compressor noise is closely related to the material and configuration of the shell. This study has conformed the following:

1. Curvatures and natural frequency of shell

(1) The lowest natural frequency value tends to saturate as shell curvature increases.

(2) The mode of a principal mode on a spherical surface shall be fixed regardless of curvature and is located on the same center angle of each curvature.

2. Steel sheet characteristics and compressor noise

(1) Of the various steel characteristics, the n × r value is closely related to compressor noise and greatly affects the damping rate of the shell. With respect to item 1, effects of flange diameter may have to be taken into consideration. Using a model with a shell configuration redesigned according to finding of this study, compressor noise was reduced 3 dB(A). Appreciation is acknowledge for the cooperation offered byMessrs. SUMITOMO Metal Industries Co., Ltd.

REFERENCES

1. "Thermophysical Properties of Refrigerants, " Japan Association of Refrigeration

2. Tojo, k., Saegusa, S., Machida, S., Hirata, T., Sudo, M. and Tagawa, S., "Noise Reduction of Refrigerator Compressors, " Proceedings of 1980 Purdue Compressor Technology Conference, July 1980, pp. 235-242.

3. David C. Lowery, "An Improved Shape for Hermetic Compressor Housings." Proceeding of 1984 Purdue Compressor Technology Conference

Table 1
NOISE GENERATION DIAGRAM
(Reciprocating Type)

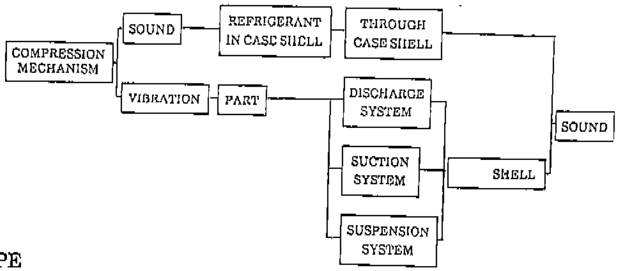


Fig. 1
RECIPROCATING TYPE

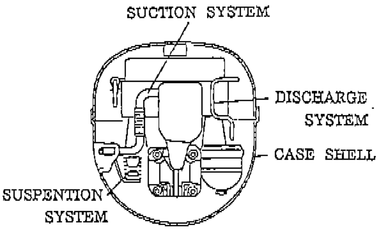


Fig. 2
ROTARY TYPE

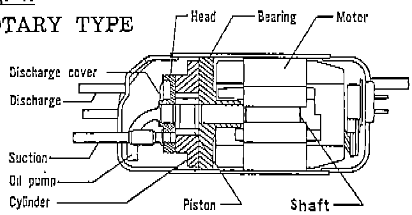
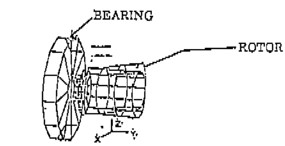


Table 2
NOISE GENERATION DIAGRAM
(Rotary Type)



VIBRATION MODE OF BEARING SYSTEM

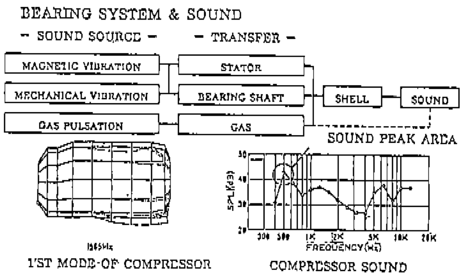


Fig. 3

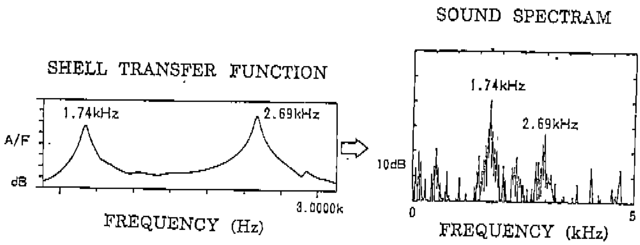


Table 3 TEST MODEL

MODEL	R 1	R 2	R 3
FRANGE (mm)	144	144	144
THICKNESS (mm)	2.5	2.5	2.5
DIA. R (mm)	125	75	68.8
WEIGHT (g)	578	670	755

Fig. 4 TEST MODEL

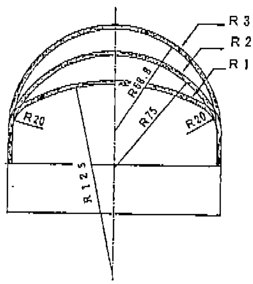


Fig. 5 MODAL EXPERIMENTS

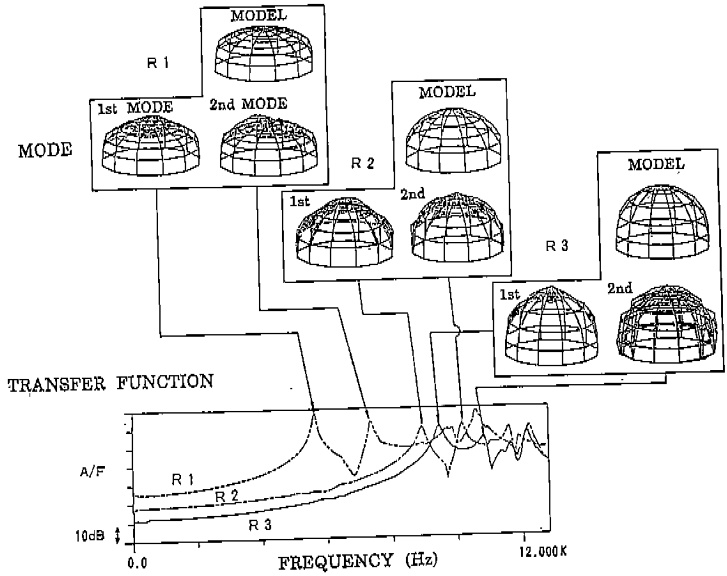


Table 4 MODAL, F.E.M RESULT

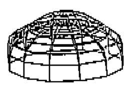
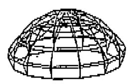
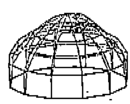
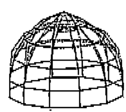
	MODEL	R 1	R 2	R 3
F・E・M	1st MODE			
	FREQUENCY (kHz)	5.80	8.90	9.43
	WEIGHT (g)	562	666	741
MODAL	1st MODE			
	FREQUENCY (kHz)	5.81	8.93	9.45

Fig. 7 CROSS SECTION OF 1st MODE

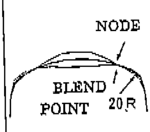
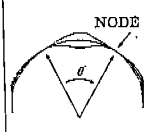
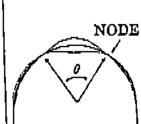
	MODEL	R 1	R 2	R 3
MODAL	1st MODE			

Fig. 6 CURVATURE - NATURAL FREQUENCY

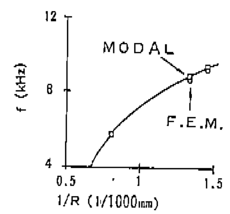


Fig. 8 MODE - NODE

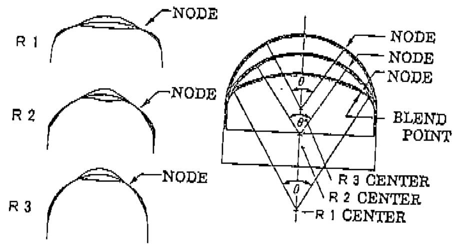
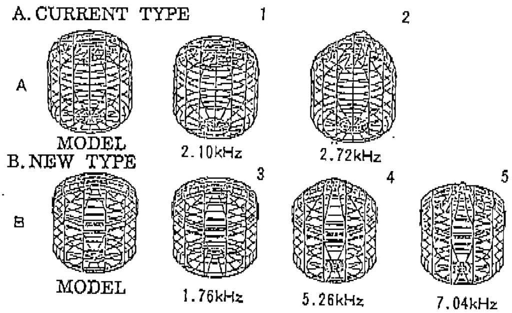
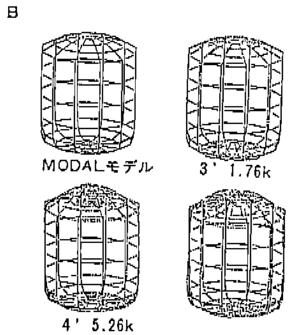


Fig. 9 IMPROVEMENT OF SHELL

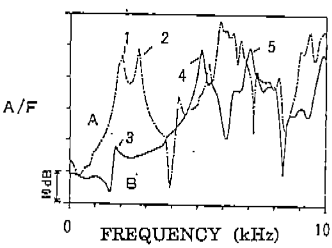
1. F. E. M. RESULT



3. MODAL RESULT



2. TRANSFER FUNCTION



4. SOUND SPECTRAM OF A THIRD-OCTAVE BAND

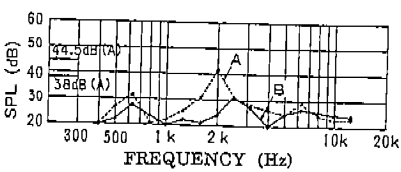
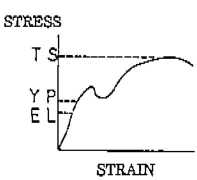


Table 5 CHARACTERISTICS OF STEEL



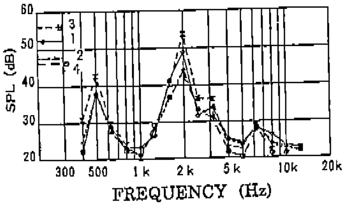
YP…YIELD POINT
TS…TENSILE STRENGTH
EL…ELASTIC LIMIT
n VALUE… STRAIN HARDNING EXPONENT
r VALUE…LANKFORD VALUE

Fig. 10 RESULT OF STEEL MATERIAL AND COMPRESSOR SOUND

1. STEEL MATERIAL AND COMPRESSOR SOUND

STEEL NO.	THICK. mm	YP Kg/mm²	TS Kg/mm²	EL %	n	r	S.P.L. (dB(A))	fo (kHz)
1	2.92	20.3	33.3	50.0	0.231	0.92	52.3	2.16
2	2.93	20.7	34.1	47.8	0.216	1.04	47.7	2.16
3	3.01	28.0	37.3	45.8	0.211	0.88	56.0	2.18
4	2.92	21.9	34.1	48.2	0.223	1.02	46.3	2.16
5	2.96	16.7	28.2	48.4	0.234	0.55	———	———

2. SOUND SPECTRAM OF A THIRD-OCTAVE BAND



3. TRANSFER FUNCTION

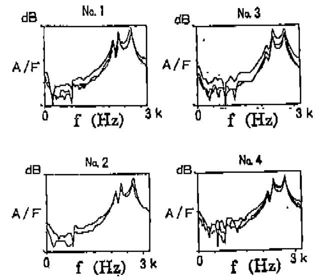


Fig. 11 MODE FIGURE

MODE

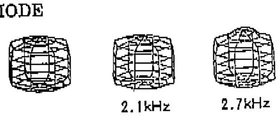


Fig. 12 EL, TS, YP - SOUND LEVEL

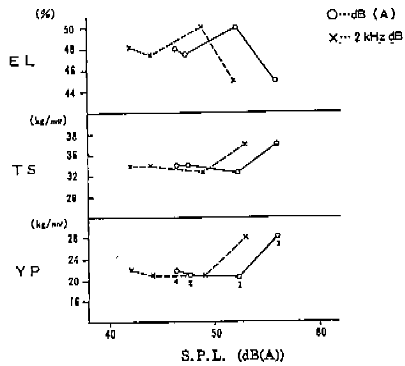


Fig. 13 n×r - COMPRESSOR SOUND LEVEL

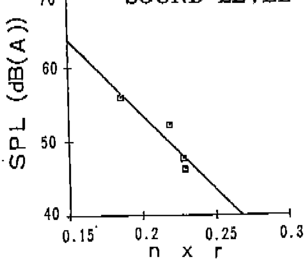


Fig. 14 n×r - MECHANICAL PROPERTIES

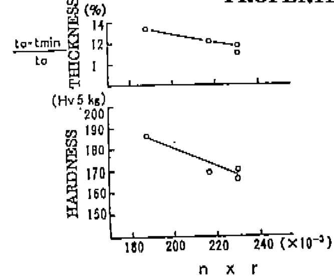


Fig. 15 n×r - SHELL DAMPING