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INTRODUCTION

While the needs for saving resources and energy have been becoming serious, there exists a need to improve the performance and quality of refrigerator and air-conditioning apparatus. Improvement of compressor performance is one of the major tasks in the total energy saving of household refrigerator. A new light weight refrigerator compressor has been developed, which has given an excellent performance. However, as the result of light weight design, compressor noise and vibration increased, and great efforts have been made to solve the problem of noise reduction.

A closer look at the noise produced by the compressor revealed that the peaks of the noise were in the 500 Hz range and the 2000 Hz range, and that the peaks had high correlations with the acceleration of the shell. The noise generation mechanism was then investigated by analyzing factors influencing the noise level, by examining dynamic characteristics of each component, and by estimating the relation between the noise and vibration. The paths along which the acoustic energy flows from the source to the shell were divided into two; (1) the air borne path (cavity resonance, etc.) and (2) the suspension system, and solid path (tubing, etc.). Noise reduction was attempted by improving the characteristics of the above two paths and the structure of the shell and by equipping a resonant type suction silencer.

This paper reveals some new techniques for noise reduction of household refrigerator compressor, which imposes two incompatible requirements, namely light weight and low noise.

OUTLINE OF NOISE REDUCTION METHOD

Figure 1 shows a cut-away view of the refrigerator compressor, which is a hermetic reciprocating type with a direct drive two-pole induction motor. The operating speed of the compressor is 50 Hz and 60 Hz. The compressor is designed with internal mufflers at the intake and discharge, with internal spring suspensions between the frame and the outer shell, and with a flexible tube to direct the discharge gas to the outside of the shell.

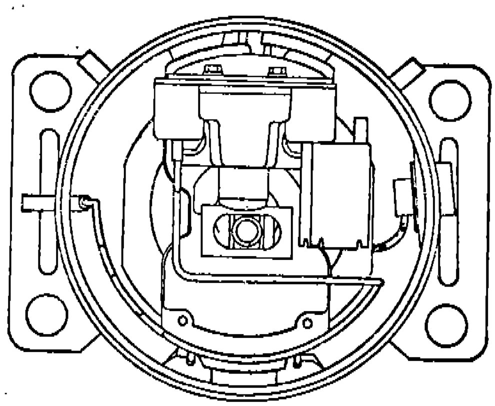

Figure 1. Refrigerator compressor

The hermetic shell housing the compressor mechanism is the principal radiator of noise from a typical compressor. Vibration excitation of the shell takes place in two ways: distributed excitation and point excitation. Figure 2 shows the noise generation mechanism of the compressor. Pressure change in the cylinder is the substantial source of the compressor noise and vibration, which are conducted from the compressor mechanism to the shell through complicated transmission paths. Figure 3 shows the noise reduction program of the compressor. The items enclosed with thick lines are;

1) Enumeration of factors influencing noise
   list the factors which have considerable effects on the compressor noise level.
2) Investigation of compressor noise

investigate the compressor noise and vibration in detail under normal operating conditions.

3) Examination of Dynamic Characteristics
   examine the dynamic characteristics of each component and compare changes in the level of sound output with changes in the suspected each component.
4) Estimation of Noise Sources
   according to the result of above analysis,select the factors which have the most significant effects on the compressor noise level.
5) Countermeasures
   devise countermeasures and make a trial.

COMPRESSOR NOISE

Compressor sound measurements were made in an anechoic chamber and the load stand was put outside the chamber. A microphone was positioned at a distance of 0.3 meters from the compressor shell. The compressor sound spectra shown in Figure 4, in addition to the broad band noise, consist of a large number of pure tone components which constitute harmonics of the rotational frequency. There are two major frequency ranges with high sound pressure amplitudes in the sound spectrum of the Prototype compressor. These peaks had high correlations with the acceleration of the compressor shell. The noise in the former range, named 500 Hz range, is produced by the solid body motion of the shell. The noise in the latter range, named 2000 Hz range, is produced by the local deformation of the shell.

Noises in these ranges were examined in detail. As the result, it was found that the sound pressure amplitude at 500 Hz range is sensitive to the operating conditions, i.e. gas temperature within the shell cavity, rotational speed, etc.
It is to be expected that the generation of the cavity resonance causes substantial fluctuations in the pressure acting on the surface of the shell causing it to vibrate and radiate sound. The noise at 2000 Hz range is considered to be caused by many factors shown in Figure 2. Those factors were examined by comparing changes in the level of the sound output with changes in the suspected sources and transmission mechanism. If the changes can be related and explained in terms of acoustics and vibration theory, the factor will be identified.

500 Hz RANGE NOISE

It is assumed that the sound pressure amplitude at 500 Hz range is increased by the generation of cavity resonance. In order to confirm the effect of cavity resonance, the following experiments were made.

1) Effect of the properties of refrigerant
   It is to be expected that any change in sonic velocity will produce a corresponding change in the cavity resonant frequency. To confirm this, air was substituted for Refrigerant 12 within the shell cavity

2) Temperature sensitivity
   As the temperature of the gas within the shell rises, the sonic velocity of the gas increases causing an increase in the cavity resonant frequency. Gas temperature was varied, and the effect was examined.

3) Rotational frequency
   Except in the case where strong broadband excitation exists, cavity resonance in compressor will occur only when it coincides with a rotational frequency harmonic.

Figure 5 shows the resonant frequency variation due to temperature change from start up to normal operating condition. Sound measurements in this case were made when the rotational frequency of the compressor was at the level producing the highest amplitude of the harmonic at 500 Hz range. The experimental result agrees well with the calculated value obtained by taking into account the change in sonic velocity. Figure 6 shows the sound amplitude variation with compressor rotational frequency.

CAVITY RESONANCE

Consider an annular cavity shown in Figure 7 filled with gas. Resonant frequency is given by the following equation.

$$f_{m,n,s} = \frac{C}{2\pi}\sqrt{\left(\frac{V_{m,n}}{a}\right)^2 + \left(\frac{s\pi}{l}\right)^2}$$

$$C = \sqrt{g.K.R.T}$$

where

- C : sonic velocity
- m : node circle number of radial mode
- n : node line number of circumferential mode
- s : node number of axial mode
- l : height of annular cavity
- a,b : outer and inner radii of annular cavity
- R : gas constant
- T : temperature
- K : specific heat ratio
- g : acceleration of gravity

neglecting the coupled vibration

$$V_{m,n} = \sqrt{\left(\frac{m\pi}{1-b/a}\right)^2 + \left(\frac{2n}{1-b/a}\right)^2}$$

For regular cavity shapes (e. g. cylinder, ideal annular cavity, etc.), the natural frequencies and mode shapes can be determined by solving the governing equation directly and they agreed well with the experimental results.
The actual compressor cavity, however, is not a true annulus, and we cannot always obtain a good agreement between the theoretical values and the experimental results. Figure 8 shows the actual compressor cavity resonant frequency and the calculated one. The actual compressor cavity circumferential resonance lies between the cases of $b/a = 0$ and $b/a = 1$.
Traversing a probe microphone in the actual compressor cavity, resonant frequencies and mode shapes were investigated.
As the result of the investigation, it was found that the amplification of the 500 Hz range peak was caused by the first circumferential resonance of the gas within the shell cavity.
The sources which bring out the cavity resonance are considered to be the following;

- Suction gas pulsation causes the viberation of the gas within the compressor shell.

- Vibration of the compressor mechanism excites the gas directly or is transmitted to the compressor shell which in turn excites the gas in the shell.

SUCTION GAS PULSATION

A test scheme utilizing an actual compressor was designed to estimate the effect of suction gas pulsation. As shown in Figure 9.1 suction line is connected directly with the suction pipe of the cylinder, and not opened to the cavity of the shell. In the case where suction gas pulsation was eliminated, sound pressure amplitude at 500 Hz range was reduced by more than 15 dB.
It is concluded that suction gas pulsation is one of the exciting sources which have significant effects on the cavity resonance.
It was attempted to equip the suction line with a Helmhortz type or side branch type resonator to reduce the gas pulsation at a certain limited frequency range.
Characteristics of the silencer was investigated theoretically and experimentally.

VIBRATION OF THE COMPRESSOR MECHANISM

The effect of vibration of the compressor mechanism on the 500 Hz range noise was investigated using a static compressor in which a cylinder was actuated with a shaker. The test was devised to determine whether or not the vibration of the compressor mechanism excites the compressor cavity and the shell to generate a sound field comparable to that actually produced by the compressor.
This set up provided excitation, in the form of vibration of the compressor mechanism, to the compressor cavity and the shell.
The test results are shown in figure 10.
Comparing them with the sound field which is accoustically produced by the compressor acceptable acceleration amplitude of the mechanism can be seen.
Another countermeasures against cavity resonance of the shell can be taken in steps as follows.

- Dissipators such as baffles or fibrous materials can be placed in the cavity within the compressor shell to damp resonant vibration.

- By changing the shape of the compressor shell and mechanism the resonant frequency of the shell cavity can be shifted to the region where it does not coincide with the rotational frequency harmonic.

2000 Hz RANGE NOISE

In most cases, sound pressure amplitude at 2000 Hz range determines the over all sound level of the compressor. It is caused by many factors.
Suction gas pulsation is one of the sources of 2000 Hz range noise. Usually a compressor is designed with internal silencers at the intake to reduce the suction gas pulsation to an acceptable level. Therefore, the effect of vibration of the compressor mechanism on the sound pressure amplitude at 2000 Hz range is described in this paragraph.

Compressor mechanism within the shell is excited by the pressure change in the cylinder during a compression process. In order to determine the vibration source of the compressor mechanism, the compressor which had accelerometers attached to the piston and the cylinder head was operated in a bolted shell.
The accelerometer would pick up all the excitations received at the two locations.
A displacement transducer was connected to the valve stopper to monitor the opening and closing of the discharge valve.
Figure 11 shows the result of above test.
The discharge pressure was kept at 1.17×10^6 Pa and the suction pressure at 9.8×10^4 Pa. The peaks at the crest of the wavy trace occurred at the end of the

discharge stage. This indicated that the peak excitation was triggerd by the opening of discharge valve. At the point where the discharge valve opens, a sudden pressure change occurs in the cylinder.
As the result of the sudden pressure change piston and other compressor mechanism are excited to cause vibration and noise.
This vibration is carried through the tubing of discharge line and the internal spring suspension to the shell which is a principal radiator of the compressor noise.
In order to reduce the noise level caused by the vibration of compressor mechanism, following countermeasures can be taken.

- Reduction of exciting forces
- Reduction of vibration of the compressor mechanism
- Improvement of characteristics of the transmission paths
- Improvement of characteristics of the compressor shell.

CHARACTERISTICS OF THE COMPRESSOR SHELL

The compressor mechanism within the shell is excited to cause vibration by the pressure change in the cylinder during a compression process. The vibration of the compressor mechanism is carried through the tubing of discharge line and the internal spring suspension to the compressor shell which is a principal radiator of the compressor noise. There was a high correlation between sound pressure amplitude and acceleration amplitude of the shell surface. Figure 13.1 shows a dynamic characteristics of the compressor shell. At the 2000 Hz range, the shell had the 3rd order resonance which causes a high amplitude vibration. Therefore, it is necessary to raise the natural frequency of the shell or take effective measures to damp the resonant vibration. In order to make higher the stiffness of the shell and raise the natural frequency, it is effective to make the shape of the compressor shell as close as possible to the perfect sphere. The effect of stiffeners and curvature of the shell was analyzed theoretically by FEM. Applying the results of theoretical analysis, an improved model which was semi-sperical in shape was made. The natural frequency and mode shape were measured experimentally.

Figure 13.2 shows the dynamic characteristics of the improved model, and Figure 12 shows the natural frequencies of the shell obtained by the theoretical analysis and the experimental measurement. A good agreement was obtained between these two results. In the case of improved model, natural frequencies were shifted to the high region in frequency and the characteristics at 2000 Hz range were advanced considerably.
Holographic interferometry was used to observe the principal mode shapes of the oscilating compressor shell. Figure 14 shows the typical mode shapes of the shell. As seen in this figure, node points and antinode points of the actual compressor shell can be located exactly.

CHARACTERISTICS OF THE TRANSMISSION PATH

The discharge pipe which directs the discharge gas to the outside of the shell and the internal suspension are the possible paths for transmitting the compressor mechanism vibration to the shell. The effects of the transmission paths on the generation of compressor noise were estimated by investigating the transfer characteristics of each path from the compressor mechanism to the shell. The internal suspension is usually adequate for reducing the vibration to an acceptable level.

Characteristics of the discharge pipe was simulated in an component test and theoretical analysis by FEM. Figure 15 shows the experimental result. One end of the pipe was attached to the shell by welding and the other end of the pipe was excited by an actuator. In this figure, lines connecting the resonant points on the transfer impedance curve of the discharge pipe from one end to the other is shown. Transfer characteristics of the pipe was improved by decreasing the stiffness of the system and by making short bend near the exciting point to reduce the vibration of the pipe and to shift the resonant frequency to the region not including rotational frequency harmonics.

RESULT OF IMPROVEMENT

As the result of investigation, the noise generation mechanism was clarified and some countermeasures were taken. Figure 16 shows the sound pressure amplitude of the improved type compressor compared with that of the prototype and the conventional type. Noise was reduced by approximately more than 10 dB(A) in the improved type compressor compared to the prototype compressor and by 2 dB(A) compared to the conventional type compressor

CONCLUSIONS

A close look was taken at the noise produced by the refrigerator compressor. The major peaks in the sound spectrum were found in the 500 Hz range and the 2000Hz range, each having high correlation with the acceleration of the shell. The paths along which the acoustic energy flows from the source to the shell were classified into two; (1) the air borne path (cavity resonance, etc.) and (2) the internal suspension system and solid path (tubing, etc.) between the pump and the shell. Noise was reduced by improving the characteristics of above two paths and the structure of the shell and by equipping a resonant type suction silencer. The result of the present investigation can be applied to estimate relevant parameters influencing the generation of the refrigerator compressor noise.

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2. M.C.C. Tsao, "On the Mechanism of Noise by a Hermetic Reciprocating Compressor with Reed Valves", Symposium of Noise and Fluid Engineering, 1977
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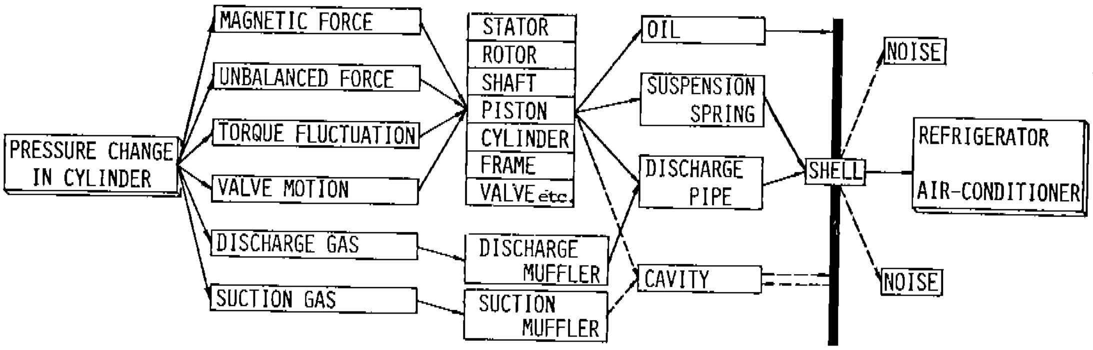


Figure 2. Noise generation mechanism of the compressor

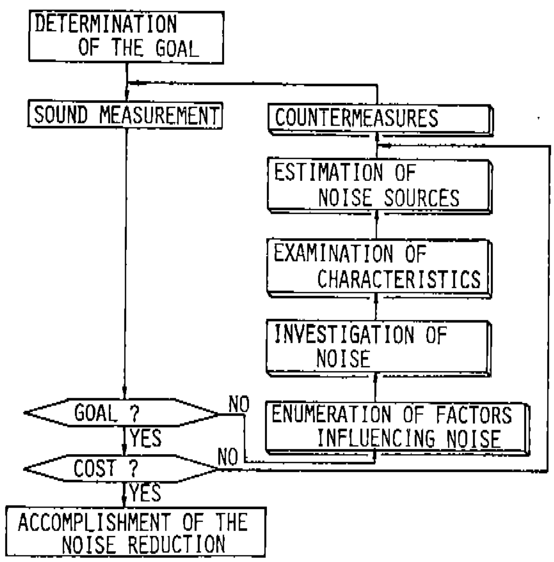


Figure 3. Noise reduction program of the compressor

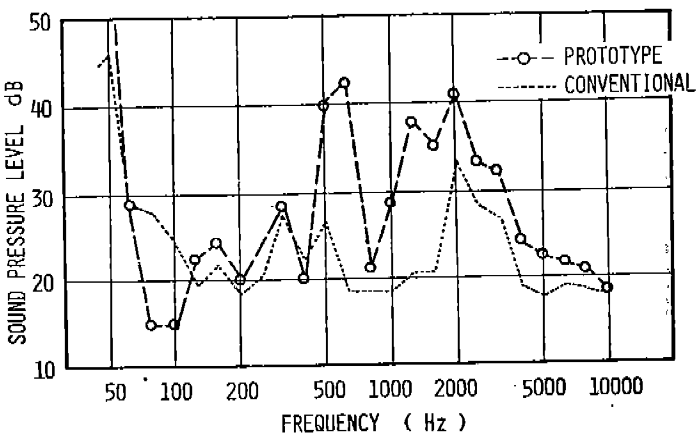


Figure 4. 1/3 octave band compressor sound spectra of prototype and conventional type

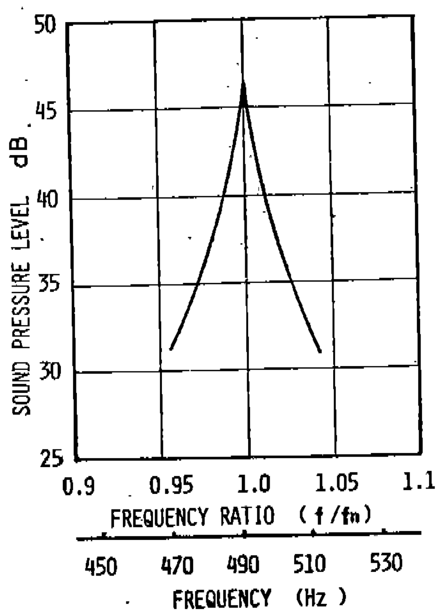


Figure 6. Sound amplitude variation with compressor rotational frequency

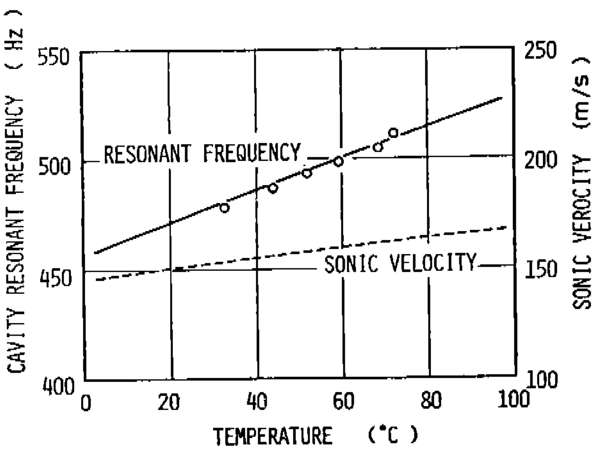


Figure 5. Resonant frequency variation with temperature from start up to normal operating condition

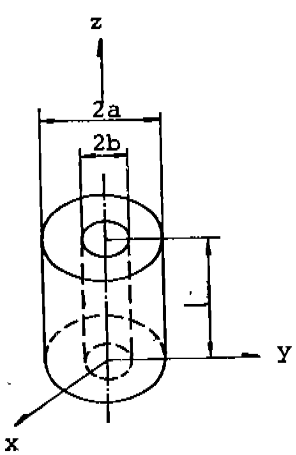


Figure 7. Annular cavity

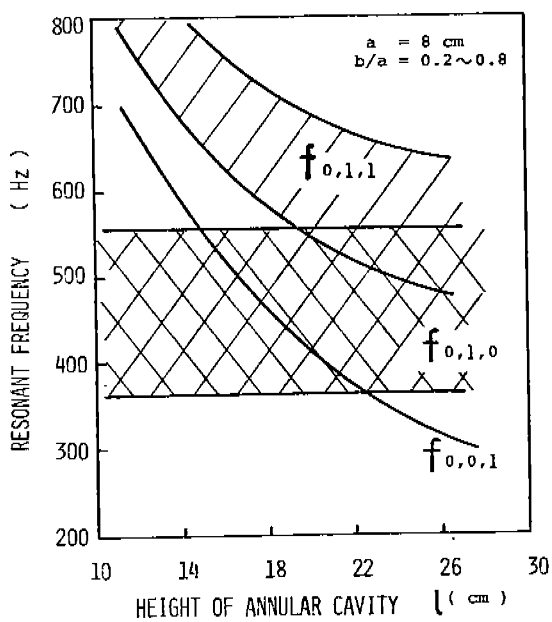


Figure 8. Cavity resonant frequency

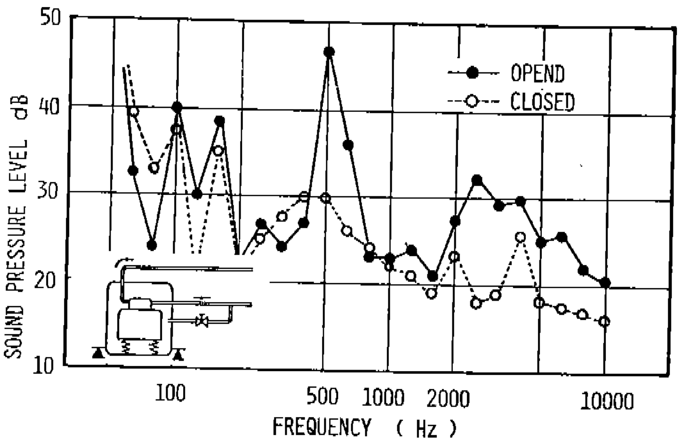


Figure 9. Effect of suction gas pulsation

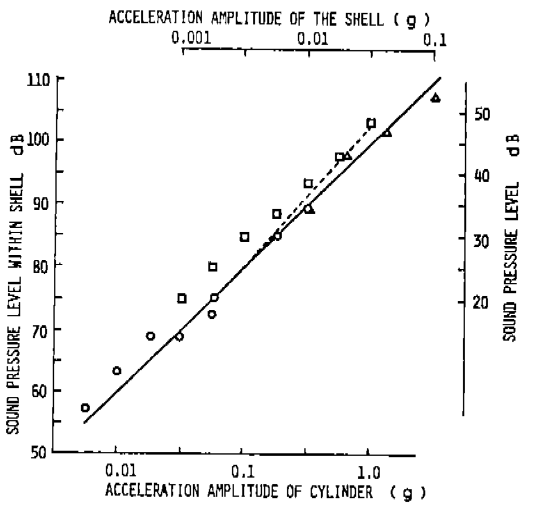


Figure 10. Effect of vibration of compressor mechanism on the 500Hz range noise

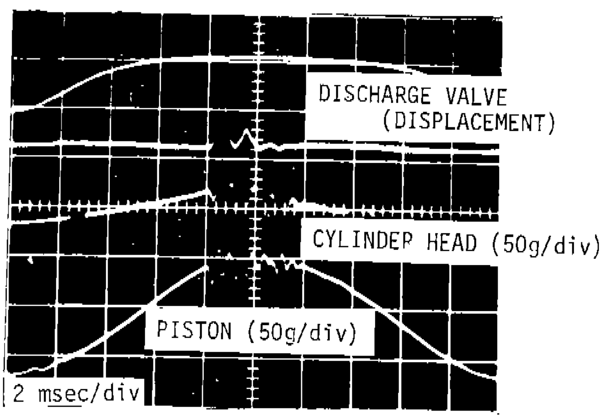


Figure 11. Typical osciloscope trace of compressor mechanism vibration

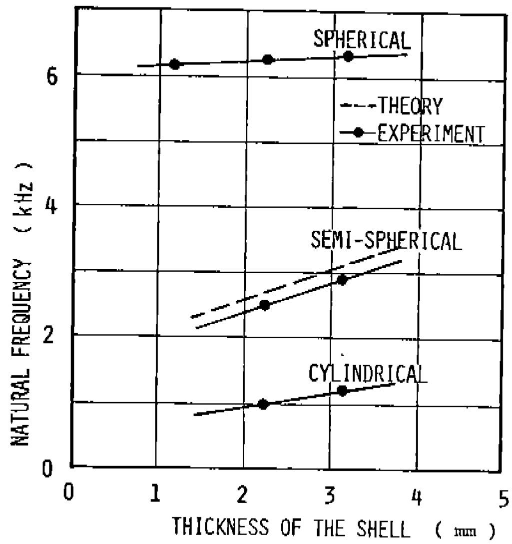
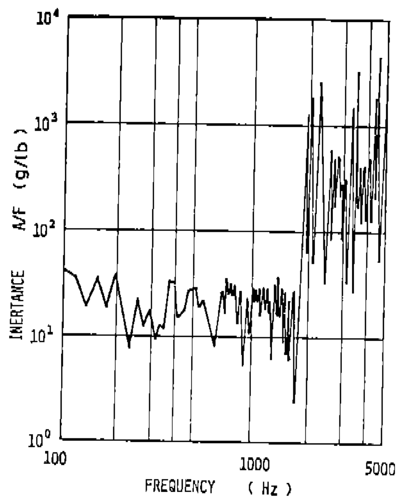
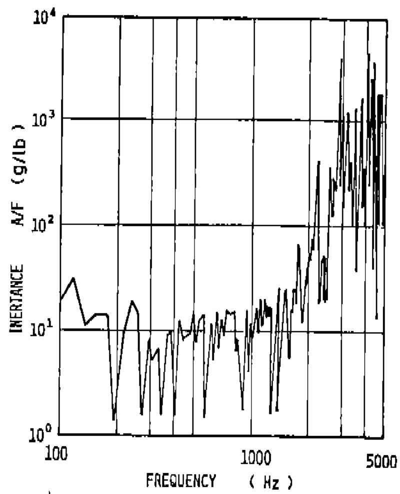


Figure 12. Natural frequency of the shell

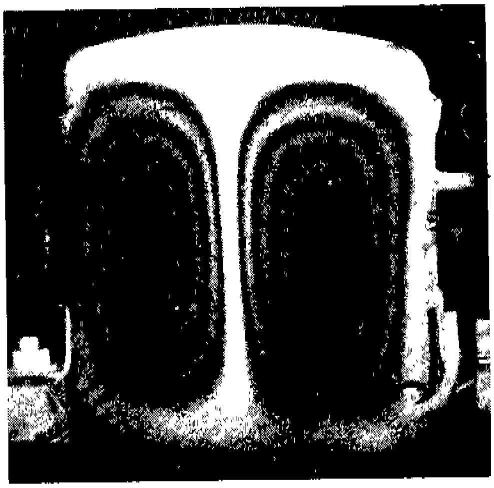


13.1. Cylindrical (Prototype)

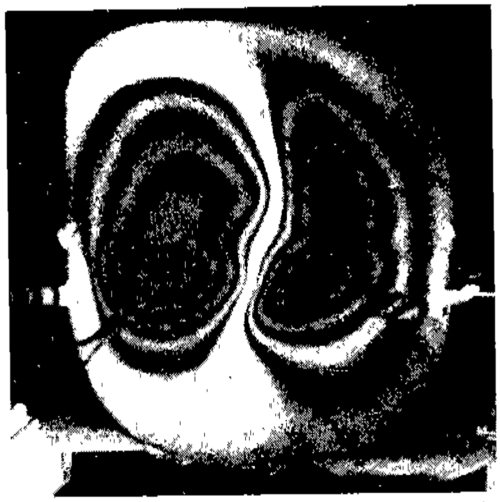


13.2. Semi-spherical (Improved Type)

Figure 13. Characteristics of compressor shell



14.1. Cylindrical (Prototype)



14.2. Semi-spherical (Improved Type)

Figure 14. Typical interference pattern of oscilating compressor shell (the 2nd order resonance)

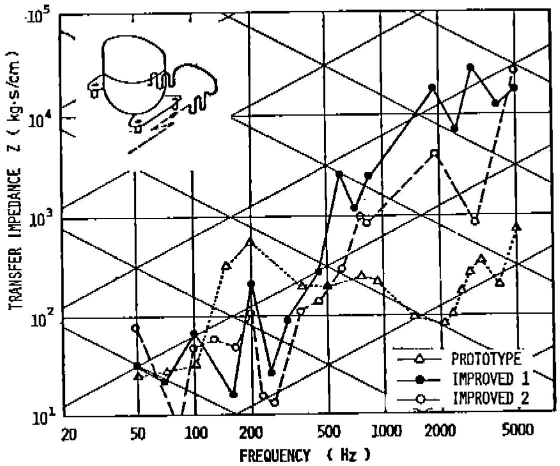


Figure 15. Characteristics of transmission path

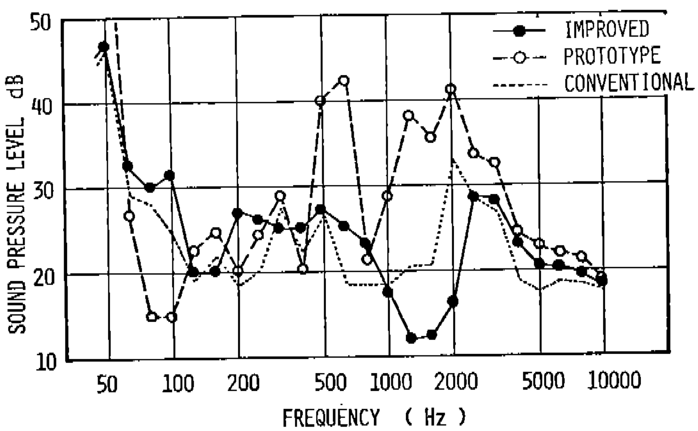


Figure 16. 1/3 octave band compressor sound spectra before and after improvement