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NOISE REDUCTION OF ROLLING PISTON TYPE ROTARY COMPRESSOR
FOR HOUSEHOLD REFRIGERATOR AND FREEZER

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ABSTRACT

The noise of a small rolling piston type
rotary compressor (horizontally installed)
for household refrigerator and other ap-
pliances may result from mechanical vibra-
tion, parts impact, gas pulsation and the
electromagnetic force of the motor. The
noise can be reduced by well-designed
structure suitable for vibration isolation
and sound suppression.

8 dB(A) noise reduction of the compressor
was achieved by the application of vibra-
tion and sound proof structure, that were
gained through examination for reducing
the sound radiation from the shell in the
relatively high frequency ranges brought
by several causes of mechanical vibration,
parts impact, gas pulsation and other ex-
citations, and also through the investi-
gation into the cause of noise generation
in the relatively low frequency langes due
to motor electromagnetic force and gas
pulsation.

INTRODUCTION

The type of compressor for household re-
frigerators is now shifting from recipro-
cative to rotary due to many different
reasons. The motor-compressor unit of
rotary compressor is fixed to the shell,
for example, by weld; whereas that of re-
ciprocating compressor is resiliently sup-
ported on the shell. This difference
causes rotary compressor to have a dis-
advantage in noise and vibration compared
with reciprocating compressor. Therefore,
noise reduction of rotary compressor has
become important.

The noise of rotary compressor in the rela-
tively high frequency ranges, originates
in mechanical vibration, parts impact, gas
pulsation and others. These noise sources
are considerably difficult to be specified
definitely. Consequently, controlling each
source is not always easy.

Mechanical vibration and gas pulsation from
any sources are mostly propagated through
the compressor unit, gas flow tubes, and
other propagation paths to the shell, from
which the sound is eventually radiated. In
other words, the shell acts as a final
sound radiator. Therefore, providing the
shell with a suitable structure for vi-
bration isolation can be a significantly
effective means of reducing the compressor
noise.

On the other hand, most of the noises in
the relatively low frequency ranges are
caused by a few sources, such as motor
electromagnetic force and gas pulsation.
The causes of the noise generation is
relatively easy to clear up. Elucidating
the causes and devising a means of noise
control at the sources are indispensable
for noise reduction.

This paper demonstrates the fact that the
sound level on a small rolling piston type
rotary compressor in operation was reduced
by the application of suitable structure
for the vibration isolation and the sound
suppression, which were found by making
examination for reduction of the noise
from the shell in the relatively high
frequency ranges and investigation into
the causes of the noise in the relatively
low frequency ranges with complete compres-
sor.

STRUCTURE OF CONVENTIONAL COMPRESSOR

A cross-sectional view of a conventional
compressor is shown in Figure 1. The re-
frigerating capacity is 191 watts (for op-
eration at 60 Hz power). The gas dis-
charged from cylinder is firstly cooled
outside the shell and then returned into
the shell; i.e., the precooling system is
employed. The main body, motor-compressor
unit is fixed directly to the shell by both
shrink fit of the motor stator and welding
the arms extending in three directions from
the outside of the side housing. The oil
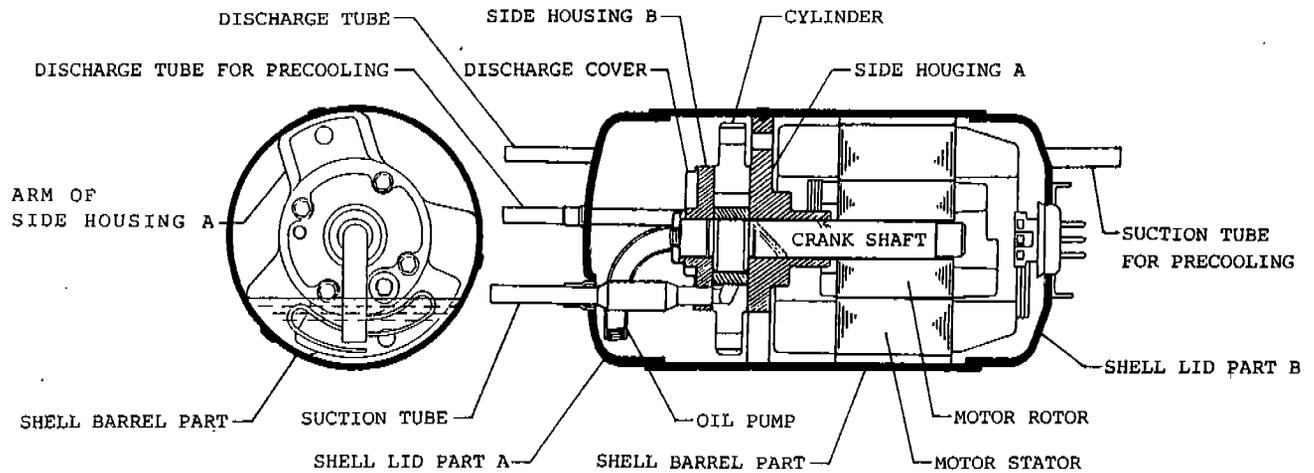pump is consist of the coil spring fixed at

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Fig.1 Structure of conventional compressor.

an end of the crank shaft and the oil feed pipe enclosing the spring. In line with the rotation of the crank shaft, the spring in the pipe rotates and oil rises up through the spiral clearance formed between the spring and inside wall.

NOISE OF CONVENTIONAL COMPRESSOR

Sound pressure level of the compressor noise at significant points were measured inside an anechoic room. The measurements were made under the following conditions:

High-side pressure = 0.98 MPa
Low-side pressure = 0.10 MPa
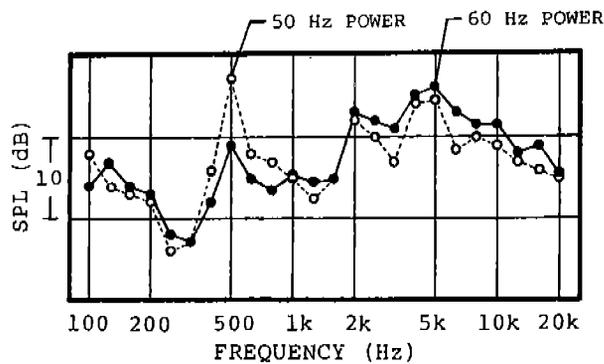Power frequency   = 50 and 60 Hz

Fig.2 Sound spectra of conventional compressor in one-third octave band analysis.

Figure 2 shows sound spectra of a conventional compressor in one-third octave band analysis. The sound pressure level at lots of band in high frequency ranges over 2 kHz is obviously high. In operation with 50 Hz power, the sound pressure level

of the compressor is high at 500 Hz band. From this result the region of unacceptable compressor noise can be divided into two frequency ranges below 1.6 kHz band and over 2 kHz band.

VIBRATION CHARACTERISTICS OF CONVENTIONAL COMPRESSOR SHELL

From the narrow band analysis of the compressor sound spectrum, it was found that the noise contains a lot of frequency components at over 2 kHz. In connection with this fact an attempt was made to examine the dynamic stiffness of the shell in order to reduce the compressor noise by improving dynamic response of the shell acting a final sound radiator.

Figure 3 and 4 show respectively measurement points and spectra of typical driving point mechanical inertance (I=A/F) of the conventional compressor shell which were obtained by sinusoidal-excitation test. The inertance level at central point d between welded point on side housing position and at the points a, b on end position is considerably high comparing with the one at welded point c on side housing position and point e of stator position.

A typical result from measurement of vibration normal mode around the side housing position of the shell barrel is shown in Figure 5. As far as side housing position is concerned, the shell contains a normal mode which is accompanying with nods at three welded points on that position. Thus, it can be supposed that the rigidity of the shell may enable to become high by increase of adherent area between the side housing and the shell.
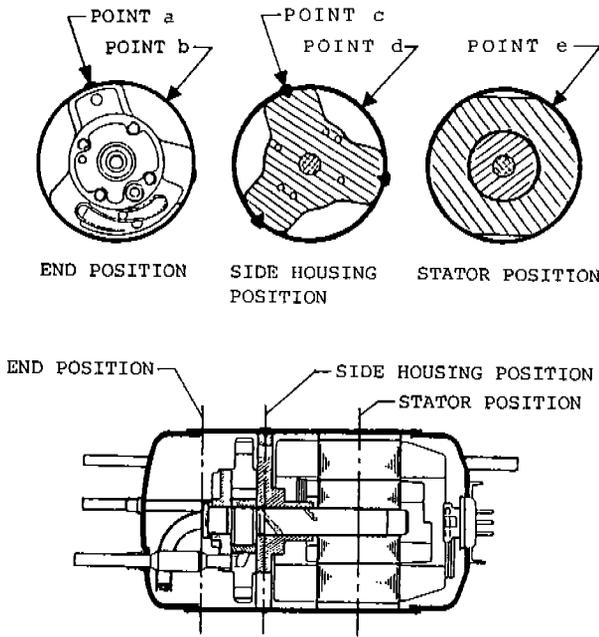
END POSITION   SIDE HOUSING   STATOR POSITION
POSITION

Fig.3   Measurement points of driving point
mechanical inertance of conventional
compressor shell barrel part.
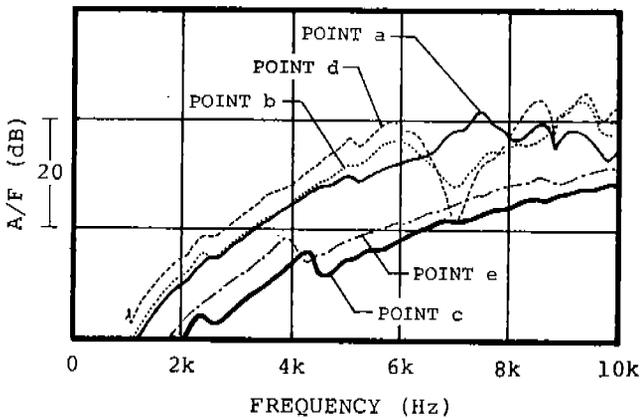

Fig.4   Driving point mechanical inertance
spectra of conventional compressor
shell barrel part obtained by
sinusoidal-excitation test.

From comparison of two inertance spectra at
the end position of the shell barrel, it
can be found that the resonance frequency
at the point b corresponding to point d be-
tween welded points of the side housing
position is slightly higher than the point
a corresponding to welded point c, however,
inertance level thereof is almost the same.
This means that the rigidity in the vicini-
ty of the end position of the shell barrel
is little affected by fixing of the side
housing. Therefore, increased fixing-areas
between the side housing and the shell will
not lead to remarkable result in increasing

the rigidity in the vicinity of the end
position of the shell barrel. It is neces-
sary that the distance between the edge and
side housing joint position of the shell
barrel part is designed as short as pos-
sible to reduce the area where large vib-
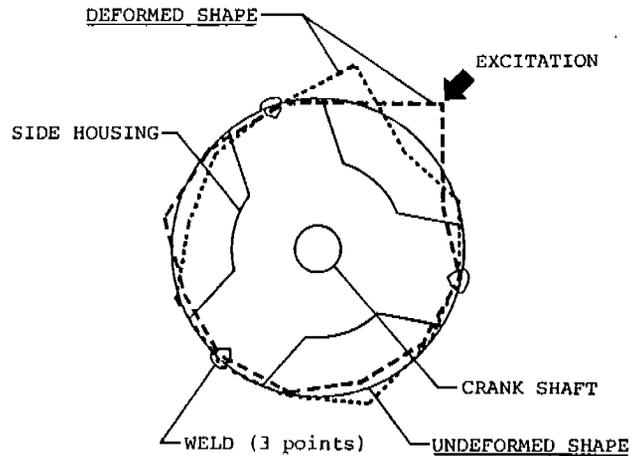ration amplitude may respond against the
same excitation.

Fig.5   5940Hz, a typical vibration normal
mode around side housing position of
conventional compressor shell bar-
rel part.

A COMPRESSOR PROVIDED WITH SHELL IMPROVED IN VIBRATION CHARACTERISTICS

Figure 6 shows a cross-sectional view of a
compressor provided with shell improved in
vibration characteristics (hereafter re-
ferred to as the shell-improved compres-
sor). The side housing is equipped with a
circular flange coupling, all around of
which is tightly contact to the inner
circumference of the shell barrel part and
fixed to the shell by weld. In addition,
the distance between the edge and the side
housing joint of the shell barrel is made
shorter than that of a conventional com-
pressor, and the shell lid part A is made
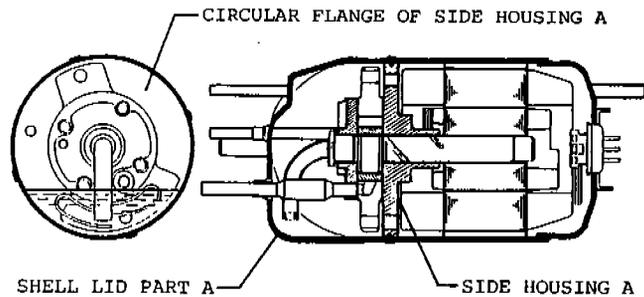sphere as to become stiffer.

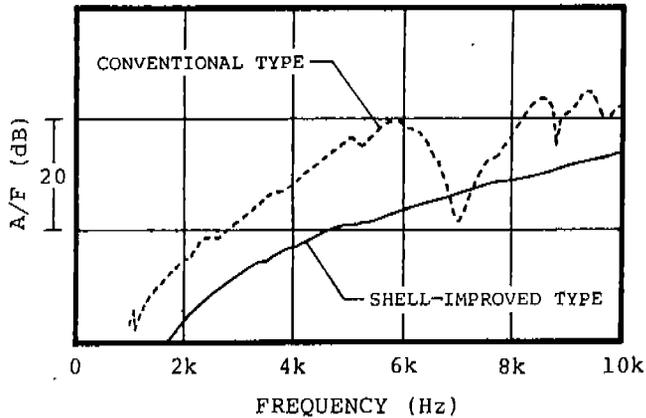Fig.6   Structure of shell-improved compres-
sor.

Fig.7 Driving point mechanical inertance spectra at central point between side-housing welded points on shell barrel part of shell-improved compressor and conventional compressor.
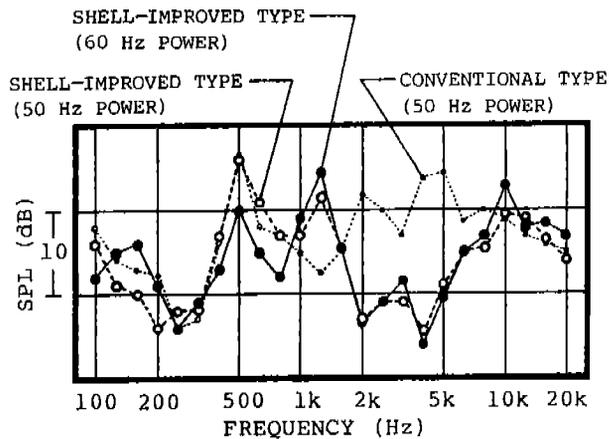

Fig.8 Sound spectra of shell-improved compressor and conventional compressor in one-third octave band analysis.

Figure 7 shows driving point mechanical inertance spectra at the central point between side housing welded points on the shell barrel part of the shell-improved compressor and the conventional compressor. It was made possible to reduce the vibration amplitude to the satisfactory extent against the same excitation on the shell-improved compressor compared with that on the conventional compressor.

Figure 8 shows one-third octave band sound spectra of the shell-improved compressor and the conventional compressor. In each bands through 2 kHz to 8 kHz it is found that the sound pressure level of the shell-improved compressor is prominently lower than that of conventional compressor. In 1 kHz and 1.25 kHz bands, however, the sound pressure level of the former is high compared with that of the latter. An at-

tempt to survey the cause of this noise was made.

1 KHZ AND 1.25 KHZ ONE-THIRD OCTAVE BAND NOISES

The narrow band sound spectrum of 1 kHz and 1.25 kHz one-third octave band noises generated from the shell-improved compressor is shown in Figure 9. Peak frequencies of this spectrum are 1 kHz and 1.4 kHz. A noticeable change in the sound pressure level of these peak components was caused by slanting the compressor to changing the cavity which is formed by oil surface in the shell. Thus, the generation of these components is regardless as the result from the cavity resonance of the shell internal cavity. The shell-improved compressor has the circular-flanged side housing and sphere shell lid, and consequently differs in shell internal cavity shape from the conventional compressor.
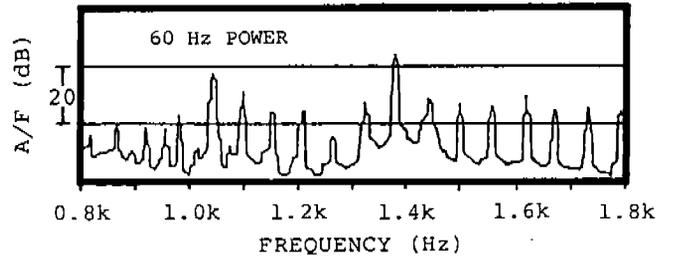

Fig.9 Narrow band sound spectrum of 1 kHz and 1.25 kHz one-third octave band noises generated from shell-improved compressor.

It seems that the complexity of the shell internal cavity dose not allow natural frequencies obtained from a calculation on simple model to accord closely with the actual values. Consequently, vibration normal mode measurements on the shell cavity of the actual shell-improved compressor were made. The sound pressure distribution in the shell internal cavity was measured by radiating sound, from loudspeaker enclosed in a sound-proof box into the cylinder through a tube, which propagates into the shell cavity throughout the hole in the discharge cover.

Figure 10.1 shows a sound spectrum obtained from this experiment at the point of interest in the cavity encircled with the shell lid and barrel, compressor unit and oil. In this cavity significant resonances generate at 1 kHz, 1.4 kHz and 1.6 kHz. These resonance frequencies correspond closely with peak frequencies of compressor noise. This means that 1 kHz, 1.4 kHz and 1.6 kHz standing waves generate in the cavity during compressor operation. Some

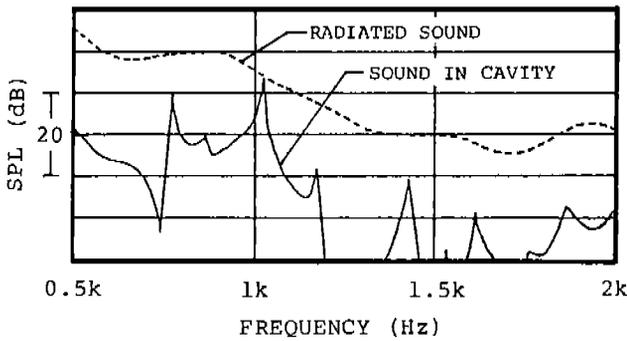of typical vibration normal mode from the experiment related with these resonances are shown in Figure 10.2.

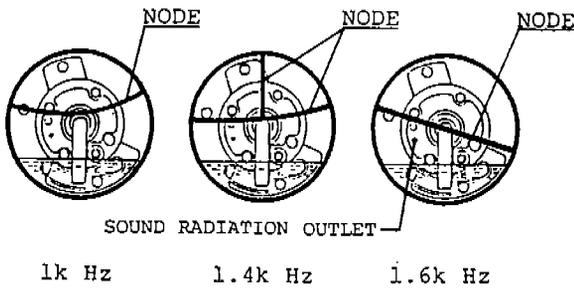
Fig.10.1 Sound spectrum in cavity

SOUND RADIATION OUTLET

1k Hz   1.4k Hz   1.6k Hz

Fig.10.2 Vibration normal modes of cavity

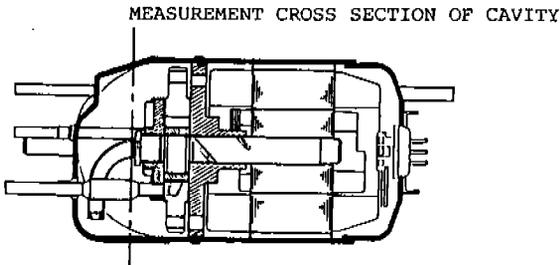MEASUREMENT CROSS SECTION OF CAVITY

Fig.10   Sound spectrum at the point of interest in shell internal cavity of shell-improved compressor and typical vibration normal modes of the cavity, measured by radiating sound into shell through outlet in discharge cover.
Because of in air measurement, frequency conversion to R-12 condition was made.

From measurements of gas pulsation spectrum in the compression chamber of the test compressor where the shell lid parts and barrel part were fastened with bolts, pulsation components was observed in all frequency ranges below 7 kHz. Figure 11 shows the coherence function between the gas pulsation in the compression chamber and the shell vibration of the test compressor. The coherence function exceeds 0.95 at frequencies from about 1 kHz to 1.6 kHz. This means that the causality between gas pulsation and the shell vibration is noticeable.
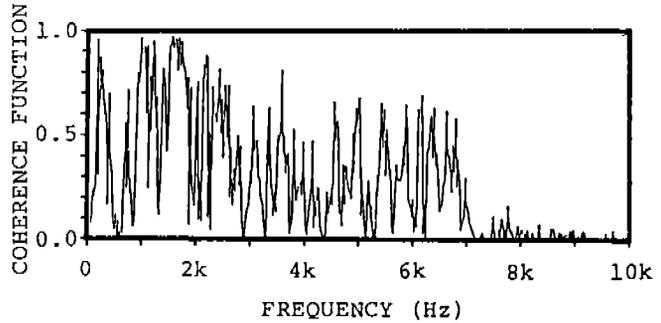

Fig.11   Coherence function between gas pulsation in compression chamber and shell vibration.

From these results it was found that major noise components of the shell-improved compressor in 1 kHz and 1.25 kHz bands are produced by the cavity resonance in the shell due to gas pulsation in the compression chamber. Although the sound pressure level of these noise components can be reduced by some methods for changing the shell internal cavity configuration and modifying the vibration normal mode of the cavity, another noise may generate due to occurrence of new cavity resonance with the modified vibration normal mode. Accordingly, a sound suppression structure capable of attenuating gas pulsation in the frequency ranges from about 1 kHz to 1.8 kHz was equipped locating just behind the gas discharge position from the cylinder.

500 HZ ONE-THIRD OCTAVE BAND NOISE

For a compressor driven with 50 Hz power, 500 Hz one-third octave band noise becomes significant. The narrow band sound spectrum of the shell-improved compressor is shown in Figure 12. The peak frequencies in the range of up to 1 kHz matches with even harmonics of the power frequency. The noise in this range was regarded to be primarily composed of the noise due to electromagnetic force of the motor judging from their peak level. The distribution of the electromagnetic noises has a peak at 500 Hz. Consequently this peak indicates occurrence of some resonance at 500 Hz.
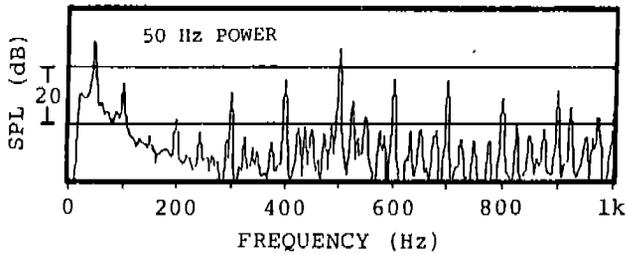
Fig.12 Narrow band sound spectrum of shell-improved compressor below 1 kHz in 50 Hz power operation.
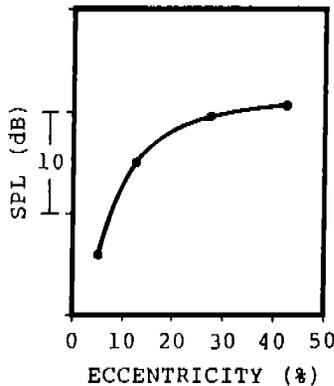

Fig.13 Relation between rotor-stator eccentricity and sound pressure level of 500 Hz electromagnetic noise. Sound pressure level is calculated out by logarithmic mean of upper and side sound pressure levels measured in the radial direction of the shell barrel part. Motor air gap is minimized at the upper side.

The eccentricity of the motor rotor to the motor stator is an important factor governing the generation of motor electromagnetic noise. The eccentricity e is defined by

$$e = (L - \ell)/L \times 100 \ (\%)$$

where L = mean air gap length between the rotor and the stator
       ℓ = minimum air-gap length between the rotor and the stator

Relation between the eccentricity and the 500 Hz electromagnetic noise was surveyed by using a test compressor capable of changing the eccentricity. The relation is shown in Figure 13. The sound pressure level of the 500 Hz electromagnetic noise decreases with the reduction of eccentricity. The sound pressure level sharply decreases when the eccentricity reduces to 5 percent from 12.5 percent. In brief minimizing the eccentricity is an effective

means for reducing electromagnetic noise. Although the eccentricity can be reduced by widening the motor air gap, the motor efficiency is lowered. To provide noise reduction not accompanying such a trade off an investigation was made on the resonance system that will increase prominently electromagnetic noise at 500 Hz.

Tentatively the crank shaft (combined with the rotor) was assumed as the resonance system near 500 Hz because of its large mass. The resonance frequency measured on the crank shaft under inoperative compressor condition showed 570 Hz. It was confirmed that the resonance frequency of the crank shaft was about 500 Hz. Under operating condition, it seems that supporting state of the crank shaft is different from that of inoperative condition due to influence by forces acting to the crank as well as rotation itself. And it is supposed that this difference causes the resonance frequency of the crank shaft be close to 500 Hz under operating condition. In Figure 14 the relation between the electromagnetic noise and the frequency is shown which was obtained in operating condition with various power frequencies. The sound pressure level of the electromagnetic noise becomes maximum level at approximately 490 Hz. From this result it comes to a conclusion that the resonance frequency of the crank shaft in operation is about 490 Hz.
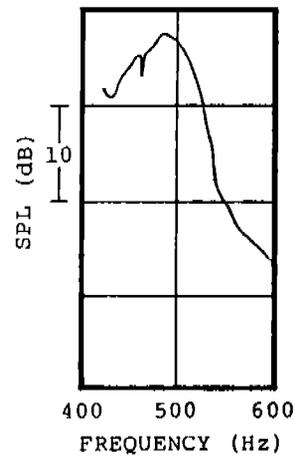

Fig.14 Relation between electromagnetic noise and frequency of shell-improved compressor. Measurement was made in operating condition with various power frequency and by tracking ten times-harmonic of the power frequency.

As described above, the crank shaft itself naturally resonates at approximately 490 Hz in operation. Eventually, it is excited

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effectively by the electromagnetic force of the motor and a noise with high sound pressure level is generated at 500 Hz. Consequently, adjusting the crank shaft resonance frequency for a value being apart from the frequency of the electromagnetic force is an effective means of reducing noise. As one of trial for the adjustment, the position of the motor rotor was moved toward the side housing as shown in Figure 15. The relation between the electromagnetic noise and the frequency of the rotor-moved compressor is shown in Figure 16. The peak of the electromagnetic noise is at about 560 Hz. To the contrary the sound pressure level at 500 Hz becomes lower.

NOISE OF NEW COMPRESSOR

One-third octave band sound spectra of the new compressor on which three improvements stated above are made are shown in Figure 17. The noise level of this new compressor is low over the entire frequency range compared with that of conventional compressors and reduced by approximately 8 dB(A).
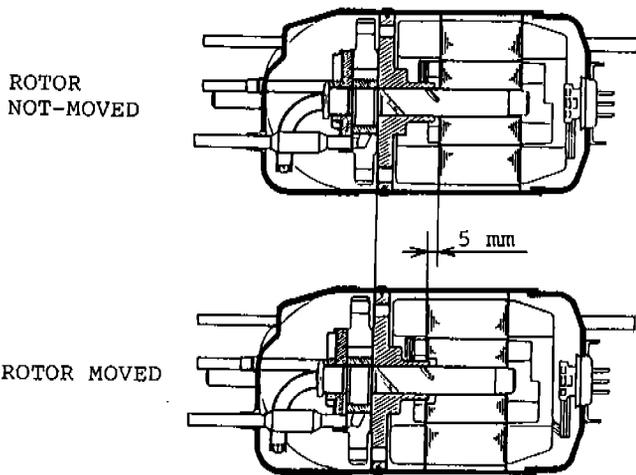
Fig.15   Change of motor rotor position.

Fig.17   Sound spectra of New compressor and conventional compressor in one-third octave band analysis.

CONCLUSIONS

The results from the study are summarized below:

(1) Most of the rotary compressor noise in the relatively high frequency ranges in operation have been able to be reduced by providing the shell with a suitable vibration isolation structure, though it is hard way to work for the reducing these noise against their various sources.

(2) In 50 Hz power operation, the sound pressure level of the compressor electromagnetic noise was high at 500 Hz. To reduce this electromagnetic noise, it is effective to minimize the eccentricity of the motor rotor to the motor stator. On the other hand it has been discovered that this electromagnetic noise generates due to the resonance of the crank shaft (combined with the rotor) by the motor electromagnetic force.
Thus, Reduction of the sound pressure level was achieved by adjusting the crank shaft resonance frequency for a value being apart from the frequency of the motor electromagnetic force.
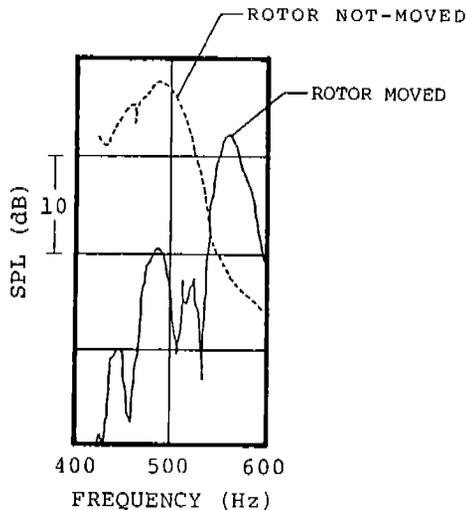
Fig.16   Relations between electromagnetic noise and frequency of both compressor with rotor moved by 5 mm toward side housing and compressor with rotor not-moved.
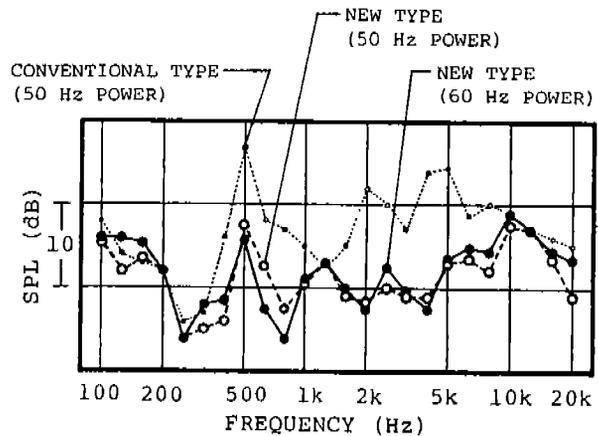
(3) Changing the side-housing and shell
    lid shapes to improve the dynamic
    response of the shell caused the shell
    internal cavity configuration to be
    modified and the sound pressure level
    of 1 kHz and 1.25 kHz one-third octave
    band noises to be increased by gener-
    ation of the cavity resonance in the
    shell internal cavity. Thus, in de-
    signing a compressor and modifying the
    shape of the shell or components it is
    indispensable to predict the vibration
    normal mode of the shell internal
    cavity. These noises were reduced by
    making a sound suppresion structure
    locating just behind the gas discharg-
    ed position from the cylinder.

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