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NOISE REDUCTION ANALYSIS ON INVERTER DRIVEN TWO-CYLINDER ROTARY COMPRESSOR

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ABSTRACT

Sanyo has originally introduced 6HP two-cylinder rotary compressors for packaged airconditioners in 1985. The concept of two-cylinder rotary compressor has been recently applied to room airconditioners(RACs) for the purpose of reducing vibration and noise of the unit. However, RACs are demanded to decrease the noise by lowering levels of the compressor noise. Sanyo has newly developed the quieter 1HP two-cylinder rotary compressor for major RACs with inverter system.

This paper describes generating mechanisms of the compressor noise established by analysis using signal processing and computer aided engineering(CAE). In addition, concrete countermeasures are presented for the noise reduction of the two-cylinder rotary compressor.

INTRODUCTION

Sanyo two-cylinder rotary compressor

The construction of Sanyo's 1HP two-cylinder rotary compressor is shown in Fig. 1. This constrution was originally developed for the two-cylinder compressor in 1985. In the process of the development, an investigation was made to determine specifications as shown in Fig. 2 for the decision of the basic construction of the two-cylinder rotary compressor. Finally, Spec. 3 was selected because the span of jounals is much shorter and the whole of pump is much compacter. However, strains of the cylinder according to this specification increased. Therefore, the cylinder configuration was optimized to solve the problem by tacked welding and fitting the cylinder on the shrinkage case. The pump of this compressor is constructed with 1st and 2nd cylinders, upper and lower journals, one middle plate, discharge mufflers, one shaft, two rollers and two vanes. This pump is fixed between the 1st cylinder and the case by tacked welding. The assembling of the pump is an important technique for setting the sweep clearance. The process is as follows:

After setting the 1st cylinder, the pump assembly machine memorize the relative location of the cylinder and roller, and de-assemble parts. Next, after setting the 2nd cylinder as well as 1st cylinder, the machine re-assemble the rest of parts according to the memory.

Two-cylinder rotary compressor noise levels in low frequency band considerably low because compression torque ripple of two-cylinder compressors is 1/3 that of one-cylinder. Fig. 3 gives the comparison of operating circumference vibration levels between the two- and one-cylinder rotary compressors. On the other hand, 600Hz~1.5KHz levels increase being influenced by the interference with gas from each cylinder.

Outline of the rotary compressor noise

1. Noise based on frequency domain

(1) Less then 500Hz

a. The fluctuation of the shaft speed is generated by that of the gas compression torque during one cycle. The noise is caused by the compressor vibration from this phenomenon. The advantage of two-cylinder compressor is that the 1st order of the electro-source frequency is almost eliminated.

b. The electromagnetic noise of motor is generated by the eccentricity of the rotor and ununiformity of magnetisms. Further, the beat noise is affected by the difference between the operating source frequency and shaft revolution.

c. The whole compressor is vibrated as a result of reaction by inertia force which is generated by unbalanced mass.

(2) 500Hz~2KHz

a. Gas resonance noises are generated in the case, mufflers and the accumlator. This noise has clear directivity and goes through the compressor case and unit panels.

b. Counter weighters are placed on the rotor in order to adjust static balance for revolution. However, the shaft is dynamically deformed by the centrifugal force and the electromagnetic force through the rotor and balancers, thus the noise appeared by revolving like a wooden pestle movement. The two-cylinder compressor has advantage for this noise because of no counter weighters required.

c. The noise is generated as a results of up-down moving of the rotor and shaft as a rigid body which is caused differential pressures between the uppper and lower cavities of the stator.

(3) 2KHz-5KHz
Major exciting force is the gas pulsation is generated in compression process. This noise depended on vibration transmission charactericistics of compressor parts. Removing exciting sources is demonstrated the greater noise reduction. also it is found that the improvement of the vibration transmission charactricistics of parts is one of effective countermeasures.

(4) More than 5KHz
This band noise, has peculiarity of transmission waves in solid structures, and is generated by slides and shocks of parts capable of moving.

2. Noise based on time domain

(1) Suction process (0~180 degrees)
The impact vibration waves are observed at crank angles of 0, 90, 180 degrees. These phenomena are interpreted in terms of the change in moving direction of the vane.

(2) Discharge process (180~340 degrees)
Fig. 4 shows casing vibration and gas pulsation in the cylinder chamber pressure curve. The greatest casing vibration level that is mainly generated by the gas pulsation are recognized in this terms. It is suggested that the gas pulsation is the major cause and both waves closely correlate each other.

(3) Other crank angles (350~360 degrees)
The strike vibration of the discharge valve is generated as the valve closes. Further, the impact vibration of the vane is caused by the jump on the roller.

3. Noise dependent on inverter system

The output of inverter system includes harmonics on time domain by chopping. Thus, the electromagnetic force is generated between motor stator and rotor by the output of inverter system interfering with space hermonics of the induction motor. The electromagnetic vibration and noise of motor are generated by this force. The lower frequency noise is generated by the coincidence between the N-order harmonaized frequencies of motor structures vibration characteristics and operating frequency. The higher frequency noise is dependent on the carrier-frequency characteristic of inverter system. The major countermeasures with this noise are as follows:

(1) The air-gap should be maintained equal around the rotor.
(2) The output waveform should be similar to the sine waveform.
(3) The carrier-frequency should be more than 20KHz.

Major causes and countermeasures

<u>Gas pulsations in the cylinder chamber</u>

1. Discussion of generating mechanisms

Gas pulsations are generated in the gas compression process in cylinder, and these frequency peaks are found in wide range from 500Hz to 6KHz. The acoustic particularity in cylinder was measured in the air by using a measuring instrument as shown in Fig. 5. Fig. 6 shows one of the difference sound pressure levels between output and input caused by statical changes of the crank angles. Also, table 1 presents the data converted from measured in the air(160 m/s／340 m/s). The followings were cleared from this experiment.

(1) The 1st resonance frequency is generated by the presence between the discharge port space and the compression room space(Type A). However, these spaces are not regarded as a side resonator after passing point of about 270 degrees. Therefore, experimental values are lower than calculative. This cause appeares that affection of gas leakage on calculation increases as the compression room volume decreased.

(2) The standing wave is generated as one half or one wavelength is regarded as a diameter of the cylinder bore(Type B). This standing wave is produced at an angle around 180 degrees, but it is not clearly observed at other angles.

(3) The standing wave is generated as one half wavelength corresponding the height of the cylinder(Type C). This frequency of the wave does not shift even if the angle is changed.

Next, gas pulsation frequencies in the cylinder chamber were determined from 30Hz to 120Hz under the operating conditions. One of the experimental results is shown in Fig. 7. Table 2 reprsents the data of basic harmonic frequency under each operating conditions. This investigation has provided the following information on the variations of basic frequencies.

(1) As the valve closes, the harmonic frequency slightly changes starting at approximately 1KHz with rise in cylinder pressure under 30Hz~90Hz operating conditions. However, above 90Hz, the harmonic frequency changes starting at approximately 2KHz.

(2) As the valve openes, the change in frequency starts shifting from approximately 500Hz which is the 1st resonance frequency of cylinder chamber at 30Hz and 40Hz. However, in case of the operating condition at more than 50Hz, the frequency begins to change from apporoximately 2KHz, 2.5KHz or 3KHz.

The causes of gas pulsation is estimated to be the follwings from all experimental results.

(1) While the valve closes, the standing wave is generated as the wavelength corresponded to the length of compressible chamber. At high frequency operation, the 2nd order standing wave is generated.

(2) While the valve openes, the resonance frequency is determined by geometric configuration between space of the port and the compression chamber. At lower operating frequencies, the 1st order resonance harmonic frequency is generated. On the other hand, at higher operating frequencies, the several order frequencies are observed.

(3) The harmonic resonance pulsation wave consolidates several order harmonic waves. The order of the emphasized hermonic wave shifts depending on the operating frequency.

(4) As the volume of compression chamber decreases, the resonance wave shifts suddenly the standing wave regarded as the height of cylinder. This fact is revealed by the uncontinuous shift in frequency that is approximately 6KHz around 270 degrees as shown in Fig. 8.

In the cylinder chamber, gas pulsations process is complicated by various causes in the compression process. And, these experimental results confirmed that the maximum exciting source of rotary compressor noise was gas pulsasions.

2. Countermeasure against gas pulsations

The gas pulsation reduction specifically between 3KHz~5KHz provided the marked effect on compressor noise reduction, because the transmission characteristic curve of the case has a number of peaks in this frequency band. Fig. 9 is a bird's-eye view of a muffler which has 3200Hz frequency resonance characteristic. It is located on the wall of cylinder bore. Fig. 10 shows the difference in gas pulsations with and without the muffler. Gas pulsations are reduced in the frequency band which matched the muffler resonance characteristic. Also, it was confirmed that the casing vibration level is lowered by gas pulsation reduction.
The muffler has only one resonance frquency. However, the muffler is significantly provided the effective reduction over wide range. This phenomenon suggests that the muffler resonance frequency shifts to the higher with changing in the gas sound velocity from suction to discharge, and that this resonance muffler behaves as a side branch resonator. It also proves that the proper position of muffler is at an angle just before the valve opens. Although the muffler dose not seem to work after the roller passed through, the major resonace frequencies of gas pulsation generated by re-expansion at the end of compression can be damped by this muffler in the suction process. Hence, the muffler should be located at around 180 degrees. In addition, the droping of energy efficiency ratio(EER) by the muffler located at around 180 degrees is less than that of the muffler located at around the discharge port.

Resonance in the cavity

1. Discussion of generating mechanisms

There are mainly three cavities in the compressor(see Fig. 1). As gas pulsations are amplified dependeing on the nature of these spaces, the noise band between 500Hz and 2KHz includes a number of resonance frequency peaks.

(1) Calculation by the Boundary Element Method(BEM)
The BEM models for these cavities as illustrated in Fig. 11 are created by computer system. The zone 1, zone 2 and zone 4 represents the space in the discharge muffler(What we call Cup), the lower space of the stator and the

upper space of the stator respectively. And the zone 3 represents a passage(including air-gap and slots) between zone 2 and zone 4. The calculated resonance frequencies as shown in table 3 are given by the acoustic analysis using this BEM model. Fig. 12 shows a typical resonance mode as a contour map of the zone 2, and Fig. 13 shows the comparsion of acoustic characteristics as sound pressure levels of each zone.

(2) Calculation by the wave equation

The three-dimensional steady-state wave equation is given by

$$\frac{\partial^2\phi}{\partial r^2}+\frac{1}{r}\times\frac{\partial\phi}{\partial r}+\frac{1}{r^2}\times\frac{\partial\phi}{\partial\theta^2}+\frac{\partial^2\phi}{\partial z^2}=\frac{1}{c^2}\times\frac{\partial^2\phi}{\partial t^2} \quad \cdots \quad (a)$$

where ϕ=velocity potential

c=sound velocity

(r, θ, z)=cylindrical coordinate system

The solution of equation (a), assuming that the boundary condition is a double cylindrical enclosure, is given by Bessel function. The results of calculaion by using the Bessel's numerical table are revealed in table 3 as well as BEM results.

(3) Acoustical experiment

Sweeped sine waves between 1 Hz and 5KHz is excited in these cavities of the compressor. Each acoustic charateristics as shown Fig. 14-a, -b and -c are obtained from this experiment in the air. The (0,1), (0,2) mode frequencies in variety of resonance modes is shown additionally in table 3 as the data converted into refrigerant state from data based on experiment in the air.

It was found that values of resonance frequency in the cavity can be estimated by calculations. Especially, there is a possibility to estimate an acoustic characteristics in space having complicated enclosures by using BEM analysis.

2. Countermeasures against resonace in cavity

(1) Cup (zone 1)

Improvement of the both cups is as follows.

a. Increasing volume of cup to a great extent.
b. Making tighten sealing between cups and jounals.

The upper cup is specifically improved by the following methods.

c. Making two discharge holes on cup at the position that the maximum amplitude of the standing wave appeares in the cup. The standing wave has one wavelength as one circumference of the cup.
d. Designing the ratio of each hole area to be equal to inverse proportion of the ratio of each length from discharge port on the journal so that each exciting force is equal to the sound pressure times the area for the zone 2.

(2) Lower space of the stator(zone 2)

The gas pulsation phase lag between the upper and lower discharge port increases the resonance noise level. Therefore, passages from each discharge port were completely separated, and the lower passage was longer than the upper. Phase of the lower gas pulsation coincides in the zone 2 with that of the upper gas pulsation. The main resonance frequency is greatly damped by the design which the difference between the uppper and lower passage length is to be one wavelength of the main resonance frequency.

(3) Upper space of the stator(zone 4)

It seems that resonance in zone 4 is hard to be generated and it has only (0,1) mode. Also, the sound level is lower than that in other zones. The reason is that gas pulsation is damped while gas pass through the zone 3, and the outlet geometrical configuration of zone 3 is like a torus. Therefore, the inlet pressure wave symmetrically flows into the zone 4 with the same phase.

<u>The transmission of acoustic waves in solid structures</u>

1. Discussion of the casing vibration transmission characteristics

Vibration of air caused by the vibration of casing is a source of noise. On the other hand, the casing vibration source is gas pulsation inside some cavities. Noise frquencies between 3KHz and 5KHz coincide in peaks with the casing vibration characteristics. The lower case of the compressor is investigated by using the Experimental Modal Analysis(MODAL) with measureing Frequency Response Functions(FRFs). One of FRFs shows in Fig. 15. Fig. 16 shows one of the mode shape by the MODAL. Also, the calculated transmission characteristics curve of casing can be estimated by Finite Element Method(FEM) analysis using damping ratios extracting from the MODAL data. Fig. 17 shows one of the calculated FRF curves of casing at a coordinate point. Agreement between calculated and experimental FRF curves are mached well in a high frquency range.

The difference of levels between the casing characteristics with cylinder and without cylinder is important because the difference is caused by the vibration mode which appeares at the edges of cylinder(see Fig. 16).

2. Countermeasures against transmission in solid structures

The best countermeasure is to eliminate causes like gas pulsations. Although isolation of transmission path is difficult, most effective countermeasures are as follows:

(1) Changing the cylinder configuration to full-round shape from partial-round shape which some parts of the cylinder interferes to the case inside.
(2) Using heavy damping materials for pump parts.

This series of two-cylinder compressors have achieved our goal of noise reduction by damping gas pulsations in the cylinder chamber. However, in next series of products will be taken the full-round shape cylinder in the specifications. In this series, there are following countermeasures.

(1) Shortening the distance between the stator and cylinder.
(2) Thickening the compressor casing.
(3) Increasing the area of casing touched on the cylinder.

<u>Other causes and countermeasures</u>

1. Noise caused by up-down moving of the shaft

The differencial pressure is generated between upper and lower spaces of the stator as the refrigerant gas flow increased corresponding to larger capacity of compressor. This fact is typically recognized on compressors discharge gas out of the discharge port of upper jounal. Then, the shaft moves up and down as rigid motion. And, noises in the relation from 1KHz to 1.5KHz are brought by being excited up-down mode of the pump. This noise is amplified by electromagnetic force of the motor if the magnetism is used as counter weighters. The countermeasures are the followings.

(1) Prevention from generating the degree of differencial pressure.
   a. Making some holes passing through the rotor core.
   b. Partially cutting the circumferential edge of the stator.

(2) Restraint of up-down moving of the shaft
   a. Setting the rotor elevated with proper bias to the stator so that the shaft is pushed down by electromagnetic reaction force.
   b. Controling strictly morphological precision of the shaft thurust is gone up so that the shaft is liberated from journal faces by concave-convex on the shaft thurust in each cycle.

2. Noise caused by impact of moving parts

(1) Vane

The vane contacts to the cylinder vane slot and the roller and the vane-shockwaves under the operating can be observed at the same crank angles. From this observation, the vane can not perfectly follow the motion of the roller under certain operating conditions, such as light load. Countermeasures are the followings.

a. Changing the vane material from steal to carbonic because smaller inertia force provides the inverter compressor with larger effect on better vane movement, and carbonic soft material decreases shock by its large damping ratio.

b. Reduction strain of the cylinder vane slot.
   ① Selecting proper positions of tacked welding intending to slightly widen the opening of the vane slot.
   ② Keeping gap between case and cylinder for shrinkage-fit is more than zero.

(2) Valve

The vibration is generated when the valve slapes on the discharge port. A countermeasure is to keep the port position is lower than the valve seat. Fig. 18 shows back pressure with the differencial height between the port and valve seat, which was calculated with FEM. The low-port specifications(Height is minus quantity) is required higher back pressure than the high-port specifications(Height is plus quantity). The valve to closing speed is in propotion to back pressure when the valve has touched to the port. Therefore, the valve closing speed of the low-port specifications decrease and the shock is reduced under the same back pressure condition.

3. Noise of the accumulator

The accumulator is one of causes of the compressor noise. The vibration of accumulator shell and resonance inside the accumulator are generated by the compressor vibration and suction gas pulsations. Fig. 19 shows one of measured FRFs at a coordinate point of the accumulator shell simulated by the MODAL analysis.

(1) Low frequency peaks indicate the whole compressor vibration modes.
(2) High frequency peaks indicate the shell vibration of expands and contracts to the radial direction.

Fig. 20 shows the experimental acoustic characteristics inside the accumulator in the air. Causes of the noise generation are described in the followings by using the data has been converted into refrigerant.

(1) 500Hz peak is a standing wave which has one wavelength equal to shell height.
(2) 1400Hz and 2100Hz peaks are the 1st and 2nd resonance frequencies to the circumferential direction.
(3) 1600Hz peak is a standing wave which has one wavelength equal to height of the partition plate.

These causes directly generate noise, and indirectly generates the noise matched the FRF of the accumulator shell. It is the most important that harmonic frequency of the structure should not agree with resonance frequency in the cavity. Countermeasures are the fowllings.

(1) For resonances inside the cavity
  a. Changing the location of the screen and partition plates.
  b. Modifying the diameter and length of the accumulator shell.

(2) For the transmission characteristics of the accumulator shell
  a. Changing the material and thickness for increasing mass.
  b. Modifying the shell diameter for increase stiffness.
  b. Adding another partition plate to shift vibration mode frequencies.

The above two countermeasures are related to each other. The specifications of these countermeasures were carefully determined by considering the relations between both.

CONCLUSION

The main comparsion of 1/3 octave band sound pressure spectrums between an early prototype and final as demonstrated in Fig. 21.

(1) Countermeasures for resonance in cavities were achieved by reducing 630Hz-~1KHz levels.
(2) The effect of muffler in the cylinder chamber contributed to the reduction of 3KHz~6KHz levels to a great extend.

Noise causes and mechanisums of noise generation for the rotary compressor are clarified in this paper. The following subjects are left to be solved in the future.

(1) How can we establish new specifications taking into these countermeasures without increasing the cost?
(2) How can we carry out a number of noise countermeasures by considering high EER, min. cost and max. productivity for developing new compressors?

These new analyses like CAE techinics will provide a time saving and low cost means for the optimum and speedy research especially for the development of new compressors.

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Table 1 Values of acoustic experiment in the cylinder chamber

Crank angle (degree)	Type A (Hz)	Type B (Hz)		Type C (Hz)
90	341 (376)*	No clear	No clear	No clear
120	400 (403)*	1,812	No clear	2,965
150	449 (445)*	1,800	No clear	2,976
180	506 (514)*	1,976	No clear	2,988
210	612 (619)*	1,800	3,705	2,988
240	788 (813)*	1,800	No clear	3,024
270	941 (1,183)*	1,800	No clear	3,024
300	1,165 (2,138)*	No clear	No clear	3,047

*:values in paren are calculation

Table 2 Emphasized resonance frequencies under the operating

Operating frequency (Hz)	Suction process (valve closes) (Hz)	Discharge process (valve openes) (Hz)
30	1,000~	520~
40	1,130~	510~
50	1,140~	1,820~
60	1,200~	2,500~
70	1,160~	2,380~
80	1,200~	2,560~
90	1,180~	2,860~
100	1,820~	2,380~
110	1,920~	2,780~
120	1,820~	2,780~

Table 3 Resonance frequencies in the cavites

Zone	Mode	Calculation (Hz)		Experiment (Hz)
		use BEM	use Wave eq.	
1 (Cup)	(0,1)	1,200	1,186	1,270
	(0,2)	–	2,220	2,186
2 (Lower)	(0,1)	650	657	658
	(0,2)	1,450	1,330	1,226
4 (Upper)	(0,1)	850	926	891
	(0,2)	–	2,144	2,108

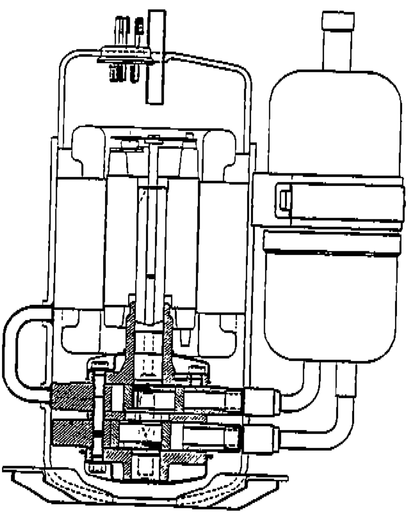

Fig.1 Cross-sectional view of Sanyo two-cylinder compressor

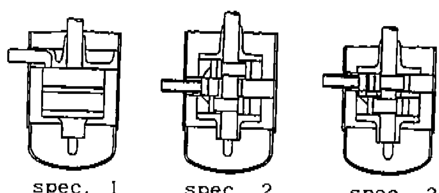

Fig.2 Pump specifications of two-cylinder rotary compressor

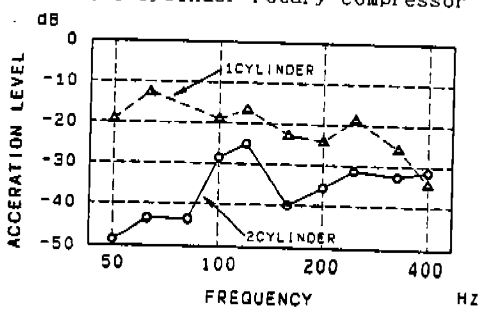

Fig.3 Comparison of vibration level

Discharge pressure(Pd)=2.1MPa
Suction pressure (Ps)=0.5MPa

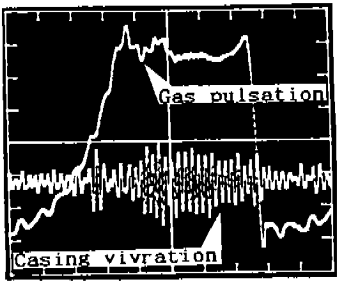

Fig.4 Gas pulsation in cylinder and casing vibration(60Hz)

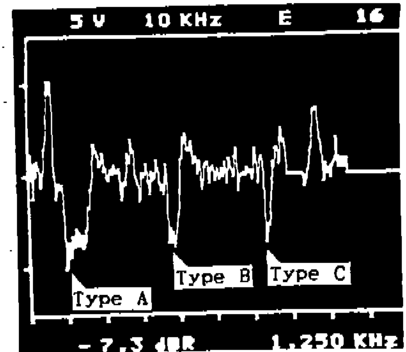

Fig.6 Measured "Input-Output" sound pressure level(150 deg.)

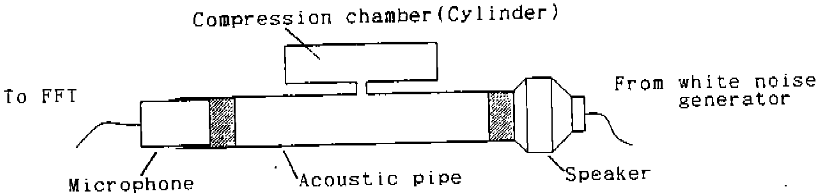

Fig.5 Acoustic measuring instrument

Pd=1.2MPa, Ps=0.6MPa

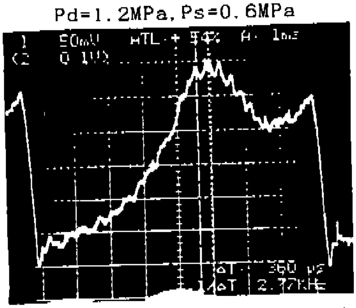

Fig.7 Pulsation under operating condition of 120Hz frequency

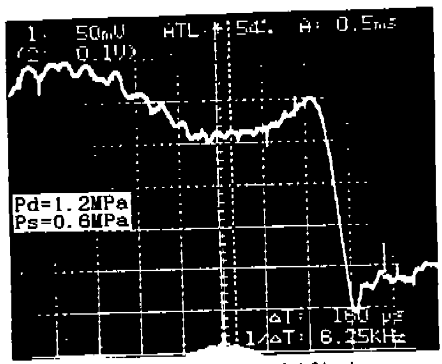

Fig.8 Uncontinuous shift in frequency (120Hz)

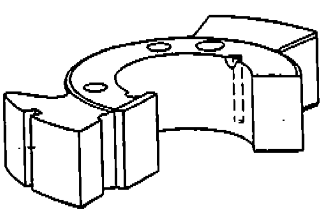

Fig.9 Schematic view of muffler

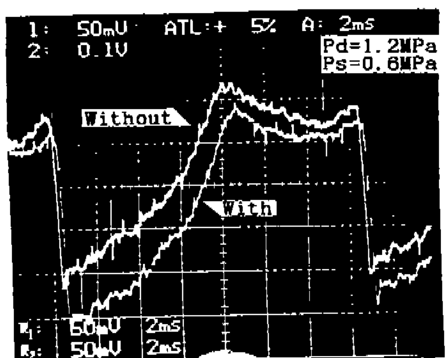

Fig.10 Comparsion of gas pulsation with muffler and without (70Hz)

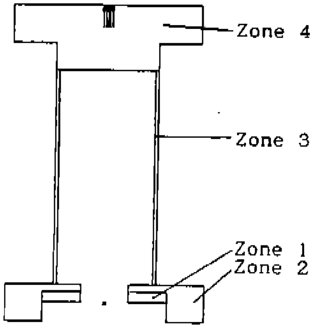

Fig. 11 BEM model for cavites

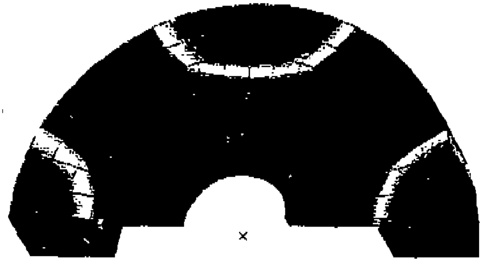

Fig. 12 (0, 2) mode in Zone 2

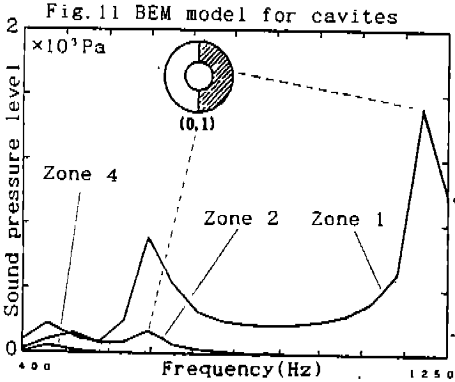

Fig. 13 Calculated sound pressure levels in each zone by BEM

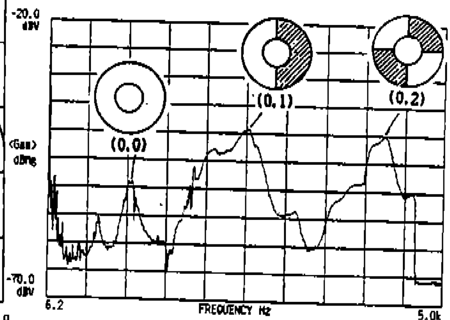

Fig. 14-a Sound pressure spectrum in Zone 1

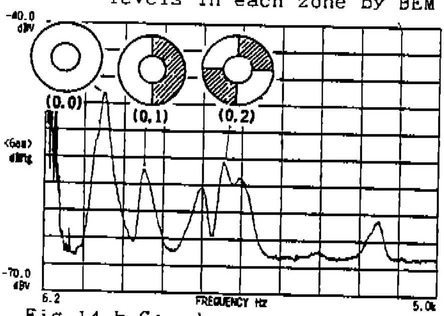

Fig. 14-b Sound pressure spectrum in Zone 2

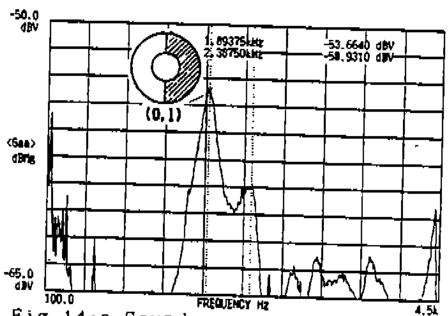

Fig. 14-c Sound pressure spectrum in Zone 4

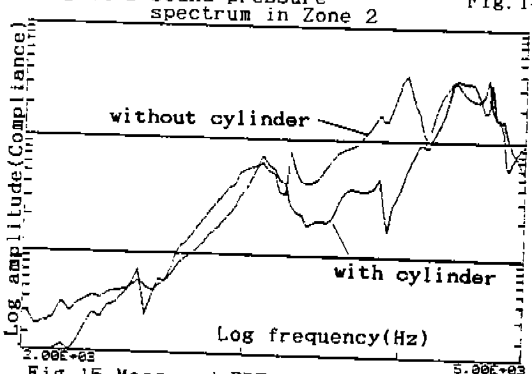

Fig. 15 Measured FRF of comp. case

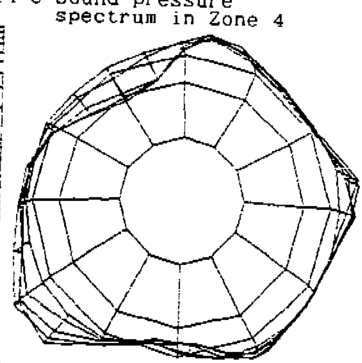

Fig. 16 Modal shape at 3122Hz

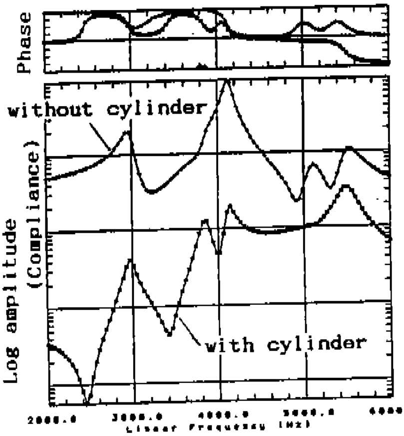

Fig. 17 Calculated FRF of compressor case

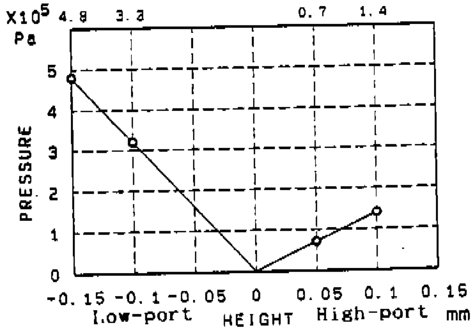

Fig. 18 Plot of back pressure vs. differencial height

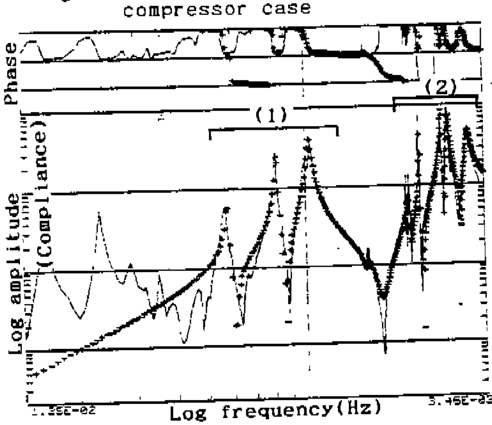

Fig. 19 Measured FRF of accumulator shell

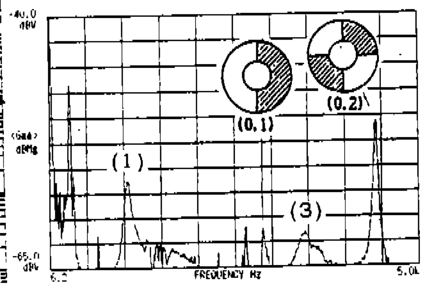

Fig. 20 Sound pressure spectrum in accumulator

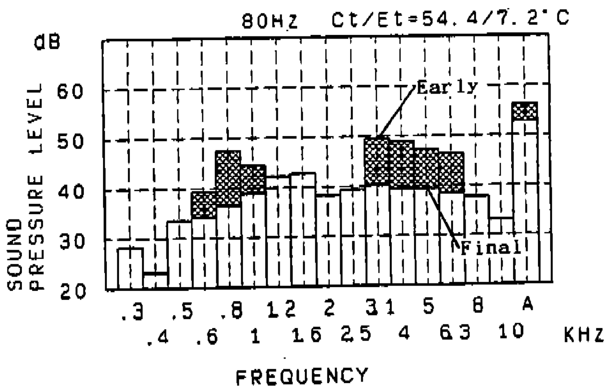

Fig. 21 Comparsion of 1/3 octave band sound pressure spectrum between early prototype and final