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A STUDY OF THE VIBRATION REDUCTION OF ROLLING PISTON TYPE ROTARY COMPRESSOR

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ABSTRACT

In general, the vibration of rolling piston type rotary compressors is greater than that of reciprocating compressors because the compressor-motor unit is fixed to the shell. It is therefore necessary that refrigerators utilizing rotary compressors incorporated a vibration-proof design.

This paper refers to the experimental vibration reduction study of rotary compressors (horizontally installed) for household refrigerators and other appliances. The vibration of rotary compressors consists of the rotational vibration caused by the speed variation of the shaft and of the imbalance vibration caused by the mass imbalance in the rotation system. There are various methods for reducing the rotational vibration. This study researched the dynamic damper. It will be shown that the dynamic damper, using a helical extension spring applied to the outside of the shell, is effective in reducing vibration. In regards to the imbalance vibration, this paper researched the influence of the number of correction planes and the accuracy of the balancing.

INTRODUCTION

The vibration of the compressor-motor unit itself is substantially smaller than that of reciprocating compressors because rolling piston type rotary compressors do not have such a large reciprocating inertia force as that of reciprocating compressors. The actual vibration of the shell section, however, is greater than that of a reciprocating compressor. This is because, for a reciprocating compressor, the compressor-motor unit is dampened by means of springs, and on a rotary compressor, in contrast, the vibration of the compressor-motor unit is transferred directly to the shell

In rotary compressors, therefore, the reduction of both noise and vibration are very important. For this reason, in rotary compressors for air conditioning, two cylinder compressors have come into wide use. But in rotary compressors for household refrigerators, it is the real state that the vibration reduction of a compressor itself has not so much been studied.

This paper refers to the result of the experimental study of the vibration reduction method for rotary compressors by the cause of vibration.

STRUCTURE

Fig.1 shows the structure of Matsushita's horizontal rotary compressor for household refrigerators. A motor-stator is shrink-fitted directly to the shell, and the compressor-motor unit is weld-fitted to the shell through the main bearing whose outer circumference is round. The vibration of the compressor-motor unit, therefore, is transferred directly to the shell resulting in the generation of vibration and noise.

VIBRATION OF ROTARY COMPRESSOR

The vibration of a rotary compressor can be divided into 3 forms: (1) rotational vibration caused by the speed variation of the shaft, (2) imbalance vibration caused by the mass imbalance in the rotation system, and (3) reciprocative vibration caused by the reciprocation inertia force due to the vane reciprocating movement.

Firstly, the levels of the respective vibration forms of current rotary compressors (displacement: 5 cm^3 class) were compared with each other by a modal analysis. As shown in Fig.2, the vibration level rate of each vibration form, that is, rotational vibration : imbalance vibration : reciprocative vibration is 4 : 1 : negligible. Rotational vibration is the most serious.

It is therefore the most important to reduce the rotational vibration. There are various methods for reducing the rotaional vibration. This study was made on a dynamic damper as a vibration reducing method, which does not lower the efficiency and maintains compactness and lightness.

In addition, another study was made on the correlation between the accuracy of the balancing and the imbalance vibration, which is considered to be a serious problem where the rotational vibration is reduced.

STUDY OF ROTATIONAL VIBRATION REDUCTION

Application of Dynamic Damper to Compressor

In general, the application of a dynamic damper to a compressor can be expressed as a vibration form of a series of two degrees of freedom. That is, it is expressed as the vibration proof supporting system considered to be a series of one degree of freedom to which a dynamic damper of a series of one degree of freedom is added.

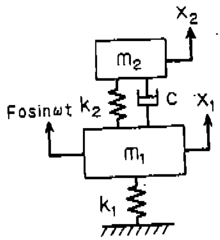
Generally, the vibration damping effect of the dynamic damper can be determined from the following equations.

Natural angular frequency of body

$$\omega_1 = \sqrt{\frac{k_1}{m_1}}$$

Natural angular frequency of dynamic damper (when c=0)

$$\omega_2 = \sqrt{\frac{k_2}{m_2}}$$



Body amplitude ratio

$$M_x = \frac{X_1}{X_0} = \frac{\left(\frac{\omega_2}{\omega_1}\right)^2 - \left(\frac{\omega}{\omega_1}\right)^2}{\left(1 - \frac{\omega^2}{\omega_1^2}\right)\left(\frac{\omega_2^2}{\omega_1^2} - \frac{\omega^2}{\omega_1^2}\right) - \mu \cdot \frac{\omega_2^2}{\omega_1^2}\,\frac{\omega^2}{\omega_1^2}}$$

$$X_0 = \frac{F_0}{k_1} \qquad \mu = \frac{m_2}{m_1}$$

In general, as shown in Fig.3(a), dynamic dampers are used to avoid the resonance frequency which may become a problem. In compressors, however, because K_1 of a vibration proof supporting system is extremely lessened, the compressor may not be resonated at operating frequency. Therefore, the dynamic damper is not used for avoiding resonance but for reducing vibration by letting it damp the vibration of a compressor itself as shown in Fig.3(b).

Study on A Wide Variety of Dynamic Damper

In applying dynamic dampers to compressors, larger vibration damping effect and wider band of vibration damping effect (frequency area in which vibration is damped by adding a dynamic damper) are preferred. The reason for the above is because there is the dispersion of the resonance frequency of the dynamic damper itself and also it may be used for both 50Hz and 60Hz operation. Moreover, it is naturally required to secure reliability. Based on these matters, the authors studied some types of dynamic dampers.

A dynamic damper consists of a mass, spring, and damper, and it is important how to set the spring and damper in particular. Because of a wide variety of dynamic dampers, the following four types were studied: the rubber type, coil spring type, beam type, and helical extension spring type utilizing the lateral bending of a contact spring having initial tension shown in Fig.4. For these studies, an exciter was used in the experiments. Fig.5 shows the vibration damping characteristics (vibration characteristics with dynamic damper / vibration characteristics without dynamic damper) in each type. The experiments revealed that in the rubber type, although a maximum quantity of the vibration damping effect was inferior to that of other types, the effect band is broad and excellent. As shown in Fig.6, however, the variation of resonance frequency to that of temperature is extremely large. This seems to be caused by the variation of the spring constant due to the change of the hardness of the rubber caused by the change in temperature. And this will harmfuly affect the stability and long period reliability of the compressors. The coil spring type provides us with a large maximum damping effect but an effect of a narrow band. In the beam type damper, damping effect characteristics are excellent, but there is a problem in reliability. The damping effect characteristics in the helical extension spring is the most excellent, but even in this type, a problem involves the variation of the resonance frequency due to the exciting force as shown in Fig.7. It is considered that this change occurs because the lateral spring constant exhibits a nonlinear characteristic due to the exciting force, because of the utilization of the lateral bend of the contact spring having an initial tension force. However, by setting the resonance frequency of the damper to the value at the time when the compressor is highly loaded, the vibration can be prevented from amplifying. Since the effect band of this type is broad, it is considered that a comparatively stable vibration damping force can be obtained.

The authors concluded from the results mentioned above that the herical extension spring type is the best considering the stability, reliability, and effect.

Application of Helical Extension Spring Type Dynamic Damper to Compressor

It is the specifications of mass (i.e., weight and shape), the specification of springs (i.e., wire diameters, mean diameter, number of turns and the initial tension force) that is the most important for determining the vibration damping effect and resonance frequency in a helical tension spring type dynamic damper. Fig.8 shows the variation of the vibration damping characteristic while the initial tension varies in the case of the same spring constant. As the initial tension is smaller, the resonance frequency becomes low, and the vibration damping effect also becomes small. This reveals that the initial tension is an important factor for determining a resonance frequency and vibration damping effect.

In addition, there are the following three factors affecting the vibration damping effect other than that mentioned above: the dimension α between the vibration center and the center of gravity of mass, dimension β between the spring fixing position (the position where spring force works to cancel compressor vibration) and the center of gravity of mass, and dimension γ between the center of vibration and the position where the spring is fixed. Fig.9 shows the diagram of the dimension α, β, γ.

Since the rotational vibration reduction is equivalent to the torque variation reduction, the vibration damping effect is proportional to the generated spring force and the dimension γ. Moreover, the generated spring force is proportional to the inertia force of mass which is determined by the dimension α. Consequently, as the vibration damping effect is proportional to the dimension α, γ, the internal installation of a dynamic damper in a compressor, which means the dimension α, γ is smaller, is not effective. In that case, only the vibration damping effect up to 15% is expected according to our research. A large extent of the vibration damping effect is only expected to be achieved when installed externally.

On the other hand, the dimension β is an important factor which affects the buckling of the spring. The generated spring force and the resonance frequency change nonlinerly when the spring is buckled. Therfore, it is necessary to decide the dimension β without the buckling of the spring under all operating conditions.

Effect of Application to Refrigerator in Field

Based on the aforementioned result, the authors evaluated the vibration reduction effect with the helical extension spring type dynamic damper externally installed to a refrigerator in the field. Fig.10 and Table 1 show the result. Excellent vibration damping effect could be obtained, and the degree of reduction reached near the level of imbalance vibration.

The vibration of the rotary compressor of the refrigerator in the field includes the vibration caused by the resonance of the system piping in the radial direction other than rotational vibration. Since when the system piping is resonated, the generated spring force works so as to reduce large rotational vibration in the dynamic damper, the phase relation to the rotational vibration at the position where the dynamic damper is installed becomes important. In the case of the same phase, the vibration in the radial direction is also dampened, but in the case of the reverse phase, the vibration in the radial direction increases.

STUDY OF IMBALANCE VIBRATION REDUCTION

The rotational vibration becomes the most serious problem in the vibration of a rotary compressor. It could be reduced to the extent of the level of imbalance vibration by externally applying the helical extension spring type dynamic damper. So, in the imbalance vibration, we examined the influence of the number of correction planes and the accuracy of the balancing as it pertains to imbalance vibration.

Balancing Concept of Rotary Compressor

Fig.11 shows the arrangement of balance weights in a horizontal rotary compressor. In order to balance the eccentric section of a shaft and piston, in general including the case in our company, two-plane balancing (two balance weights) are used by means of the balance weights M2 and M3. This time, adding the weight M4, three-plane balancing (three balance weights), by which the weight reduction of the balance weights M2 and M3 is possible, were also studied. If the direction of gravity of balance weights is completely same or opposite to the eccentric of the shaft, the ratio of the imbalance moment in each correction plane can be shown in Table 2. It means good balancing that each ratio of the imbalance moment (Cn) gets near 1.

Fig.12 shows the characteristics of the ratio of the imbalance moment (Cn) in two-plane balancing. The weight of M2 is adjusted by the number of steel sheets; and the number of steel sheets determines C3 approximately. And when the number of steel sheets (M2) is constant, C2 is determined automatically by determining C1. That is, in this study the evaluation of the number of steel sheets (M2) and the ratio of the imbalance moment C1 was made.

Balancing Study Specifications

A study was made on both of (1) evaluation of shaft behavior and bearing deformation and (2) vibration evaluation of a compressor, regarding the models different weights and the arrangement of balance weights. The models on which the study was made are as follows:

Model A:two-plane balancing (M2, M3);Balance removal weight(C1=1.3)
Model B:two-plane balancing (M2, M3);Balance adjusting weight(C1=1.1)
Model C:three-plane balancing (M2,M3,M4);Balance adjusting weight(C1=1.1)
Model D:non imbalance (M1 to M4 none, with rotor)

Rotation Characteristics of Rotor

The authors, first of all, measured the shaft behavior and bearing deformation of the shaft and bearing arrangement in the compressor by means of the bearing test machine. Fig.13 shows the structure of the bearing test machine and measuring method. As shown in the Fig.13, the same test bearings (main bearing and sub-bearing) as those of a compressor are fixed to the bearing test machine. In addition, the test shaft is connected to the inverter motor so that the motor may be driven at any speed. An eccentric section is mounted on the test shaft and a rotor is also mounted. An eddy current type gap sensor was used to measure shaft behavior and bearing deformation in the X axis and Y axis directions. Fig.14 shows an example of the measuring result of the rotation locus (position a) of the rotor. As shown in Fig.14, the rotation locus of the rotor becomes like a circle.

As a result of the above-mentioned measurement in the models aforementioned, the following were disclosed.

(1) In a horizontal rotary compressor, even if the balance is adjusted perfectly by two-plane balancing, the rotation of the rotor is 62/80μm(50/60Hz).
(2) The rotation of the rotor is 41/47μm(50/60Hz) by adjusting balance by three-plane balancing. This is 40% reduction.

Imbalance Vibration Characteristics

Fig.15 shows the imbalance vibration measuring results in the compressor of the models A to D. It also shows the imbalance vibration characteristics (60Hz) to the ratio of the imbalance moment C1. As a result, when the ratio of the imbalance moment C1 is 1.1 or less, compared to non imbalance, imbalance vibration is nearly the same level, about 16% to the rotational vibration. But, when the ratio of the imbalance moment C1 becomes 1.3, the imbalance vibration increases up to about 82% to the rotational vibration. These results reveal that imbalance vibration increase on the ratio of the imbalance moment C1, similar to a curve of a secondary degree.

Fig.16 shows the characteristics of the imbalance vibration predicted based on the measuring result of the vibration of compressors of the models A to D. Fig.16 also disclose the following matters: When comparison is made between two-plane balancing and three-plane balancing, it is cleared that both the rotation of the rotor and imbalance vibration in three-plane balancing is smaller than those in two-plane balancing at the same operation frequency. And the difference between the two becomes remarkable during the rotation at high speed in particular. (I and III points ⟶ IV and V points)

When comparison is made between the two at C1=1.1 and 90Hz, the rotation of the rotor is 165/78μm (two/three-plane balancing), and the imbalance vibration is about 29/19% to the rotational vibration. (III point ⟷ V point). Therefore, the imbalance vibration under 90Hz in three-plane balancing is the same level as at 60Hz in two-plane balancing. (I point ⟷ V point)

The collection of aforementioned result are as follows.

(1) The rotation of the rotor and the imbalance vibration of the compressor can be suppressed to a minimum by making the ratio of the imbalance moment Cl within a range of 1±0.1.

(2) Even in three-plane balancing, the imbalance vibration is not improved at an operation frequency of 60Hz or less compared to that in two-plane balancing. That is, even in two-plane balancing, imbalance vibration can be suppressed to a minimum by sufficiently balancing. At an operation frequency of over 60Hz, however, the effect of the reduction of imbalance vibration is large in three-plane balancing.

CONCLUSION

A study was made to reduce rotational vibration and imbalance vibration which become a problem in particular in horizontal rotary compressors for household refrigerators. Regarding rotational vibration, by externally installing a helical extension spring type dynamic damper to a compressor, rotational vibration was reduced to nearly the level of imbalance vibration at an operating frequency of 50/60Hz. Regarding imbalance vibration, it was clarified that at an operating frequency of 60Hz or less in particular, even in two-plane balancing, the imbalance vibration can be suppressed at the same level as in three-plane balancing by conducting sufficient balance adjustment. But three-plane balancing can reduce the rotation of the rotor, more than two-plane balancing, and it is effective for improving reliability between shaft and bearings. In addition, at a higher operation frequency over 60Hz, the reduction effect of imbalance vibration owing to three-plane balancing is large.

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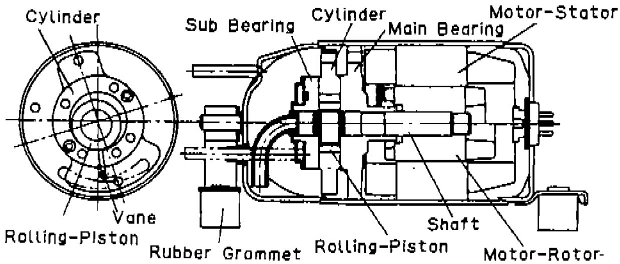


Fig.1 Compressor Internal Structure

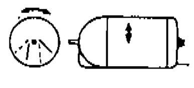
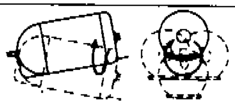
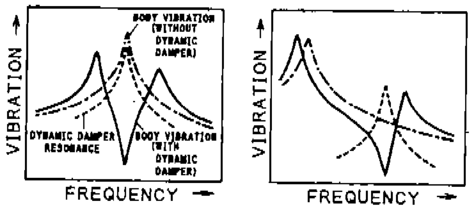
	ROTATIONAL VIBRATION	IMBALANCE VIBRATION	RECIPROCATIVE VIBRATION
VIBRATION FORM			
VIBRATION LEVEL RATE	4	1	NEGLIGIBLE

Fig.2 Vibration Form and Vibration Level Rate



(a) Normal Application of Dynamic Damper (b) Application of Dynamic Damper to Compressor

Fig.3 Dynamic Damper Application

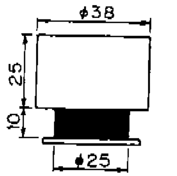
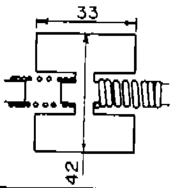
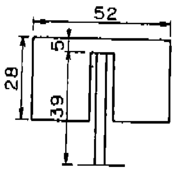
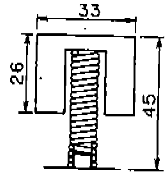
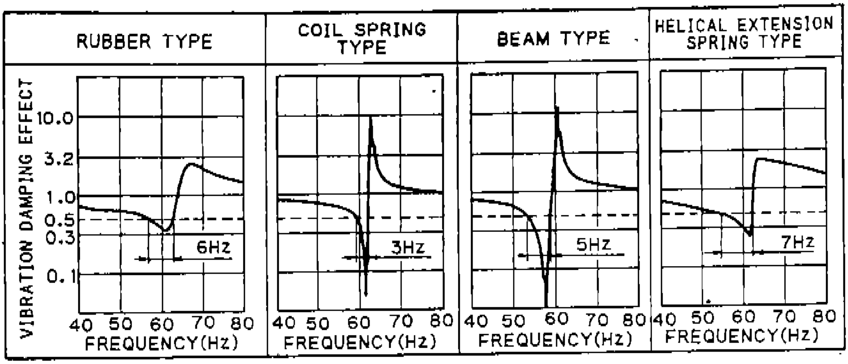
	RUBBER TYPE	COIL SPRING TYPE	BEAM TYPE	HELICAL EXTENSION SPRING TYPE
FORM				
MASS	weight 220g	weight 195g	weight 240g	weight 230g
SPRING & DAMPER	·material :NBR ·hardness:65 (Hs)	·material:piano wire ·diameter of wire :φ1.4mm ·mean coil diameter :φ8.4mm ·number of turns :4	·material:piano wire ·diameter of wire :φ2.9mm ·length :39mm	·material:piano wire ·diameter of wire :φ2.0mm ·mean coil diameter :φ7.2mm ·number of turns :15 ·initial tension :78.4N

Fig.4 Type of Dynamic Damper



*EXCEPT THE INFLUENCE OF INERTIA MOMENT

Fig.5 Vibration Damping Characteristic by Type

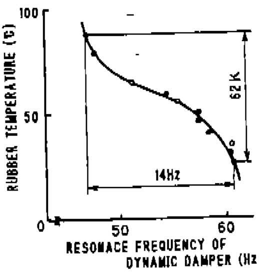


Fig.6 Temperature Variation of Rubber Type--Resonance Frequency Variation Characteristic

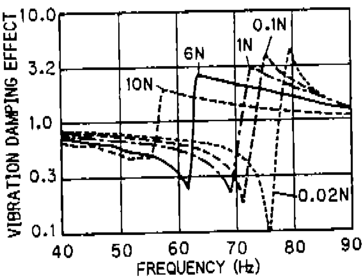


Fig.7 Damping Characteristic Variation Caused by Exciting Force Variation of Helical Extension Spring type Dynamic Damper

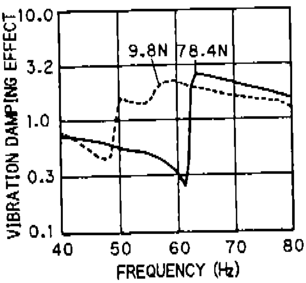


Fig.8 Vibration Damping Characteristic Variation at the Time of Change of Spring Initial Tension

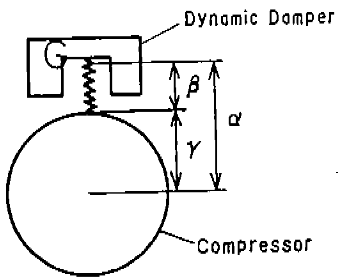


Fig.9 Diagram of Dimension α, β, γ

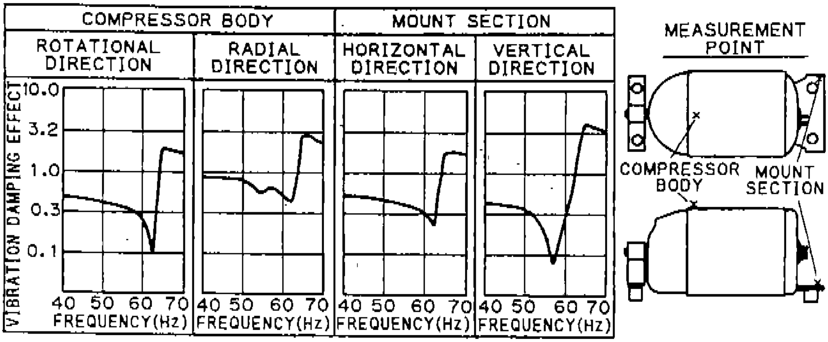
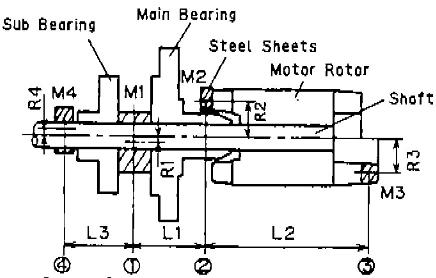


Fig.10 Household Refrigerator in Field Vibration Damping Characteristic in Applying Helical Extension Spring Type Dynamic Damper

Table.1 Household Refrigerator Vibration Damping Effect in Applying Helical Extension Spring Type Dynamic Damper

		IN 50Hz OPERATION	IN 60Hz OPERATION	IN MAX. EFFECT
COMPRESSOR BODY	ROTATIONAL DIRECTION	57%	70%	91%
	RADIAL DIRECTION	17%	47%	55%
MOUNT SECTION	HORIZONTAL DIRECTION	53%	65%	77%
	VERTICAL DIRECTION	63%	80%	92%



[Symbol] M = Eccentric Weight
R = Eccentric Radius
L = Distance between Eccentric Weights
[Subscript] 1 = Eccentric (Shaft, Piston)
2~4 = Correction Planes

Fig.11 Balance Weight Arrangement

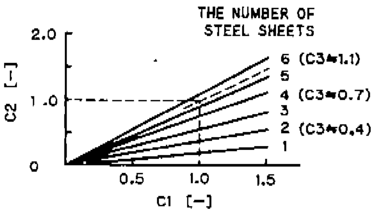
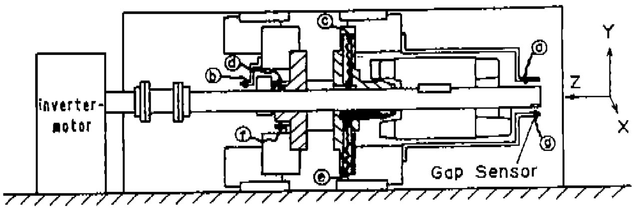


Fig.12 The Ratio of Imbalance Moment Characteristics

Table.2 The Ratio of Imbalance Moment(Cn)

Two-Plane Balancing	$C1 = M3 \cdot R3(L1+L2)/(M2 \cdot R2 \cdot L1)$ $C2 = M3 \cdot R3 \cdot L2/(M1 \cdot R1 \cdot L1)$ $C3 = M2 \cdot R2 \cdot L2/\{M1 \cdot R1(L1+L2)\}$
Three-Plane Balancing	$C1 = \{M3 \cdot R3(L1+L2)+M4 \cdot R4 \cdot L3\}/(M2 \cdot R2 \cdot L1)$ $C2 = \{M3 \cdot R3 \cdot L2+M4 \cdot R4(L1+L3)\}/(M1 \cdot R1 \cdot L1)$ $C3 = \{M2 \cdot R2 \cdot L2+M4 \cdot R4(L1+L2+L3)\}/\{M1 \cdot R1(L1+L2)\}$ $C4 = \{M1 \cdot R1 \cdot L3+M3 \cdot R3(L1+L2+L3)\}/\{M2 \cdot R2(L1+L3)\}$



ⓐ : Measuring of Rotation Diameter of The Rotor Top
ⓑ : Measuring of Shaft Behavior of the Sub Bearing Side.
ⓒ, ⓓ: Measuring of Shaft Locus in the Main, Sub Bearing.
ⓔ, ⓕ: Measuring of Main, Sub Bearing Deformation.
ⓖ : Measuring of Crank Angle.

Fig.13 Test Equipment(Bearing Test Machine)

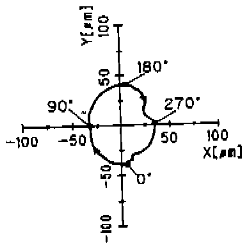


Fig.14 Rotor Top Locus (3500r/min)

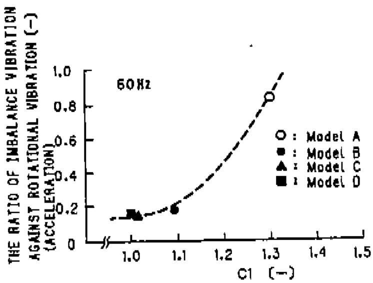
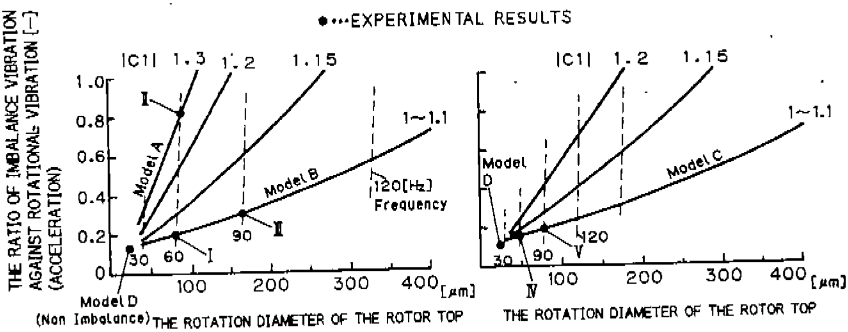


Fig.15 Measuring Result of Imbalance Vibration



(a) Two-Plane Balancing (Steel Sheets : 5pieces)

(b) Three-Plane Balancing (Steel Sheets : 4pieces)

Fig.16 Imbalance Vibration Characteristics