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Estimation of Bearing Load of Rolling Piston Type Rotary Compressors Under High Speed Operation

M. Sugiyama

T. Itami

M. Kubo

ESTIMATION OF BEARING LOAD OF ROLLING PISTON TYPE ROTARY COMPRESSORS UNDER HIGE SPEED OPERATION

Tsugio Itami,* Masahiro Kubo, Makoto Sugiyama

TOSHIBA CORPORATION Fuji Works
336,Tadehara, Fuji, Shizuoka, 416, Japan
*TOSHIBA R&D CENTER, 1, Komukai, Toshibacho
Saiwaiku, Kawasakishi, Kanagawa, 210, Japan

ABSTRACT

In today's air-conditioner, the rolling piston type rotary compressors are widely used. Induction motors are generally used to drive these types of compressors. Therefore,the maximum operating speed is about the same as the electric source frequency (example, 50Hz or 60Hz).

We are now able to vary the rotational speed of the rolling piston type rotary compressor with the aid of an inverter control that controls the power source frequency instead of using an on - off control. The purpose of this is to increase heat pump capacity during high speed operation when the load is large and decrease heat pump capacity during low speed operation when load is small.

The rolling piston type rotary compressors, which have a rather large degree of eccentric mass at the roller and the eccentric section of the shaft, would have problems such as the distortion of the shaft and its support system at high speed operation and would cause too much bearing load.

We are going to discuss the problems of bearing loads of compressors during high speed operation.

Countermeasures for reducing compressor bearing loads during high speed operation are as follows ;

(1) Reduction of eccentric mass
(2) Reduction of balancer weight of rotor
(3) Increase shaft rigidity
(4) Reduction of rotor stack hight
(5) Reduction of discharge pressure

NOMENCLATURE

U1: rotor lower end balancing weight

U2: rotor upper end balancig weight

U0: roller weight and shaft eccentric weight

L1: length between U1 and U0

L2: length between U2 and U0

Fg: compressing gas load

@: a rotational degree of shaft from blade

V: compressing displacement volume

H: cylinder height

Rr: inside radius of cylinder

Ra: outside radius of roller

Ps: pressure of suction

Vd: displacement volume

INTRODUCTION

Recently, in Japan the inverter air-conditioners which have been made variable with the aid of an inverter control that controls the power source frequency get a large percent (particulary heat pump air- conditioner).
The inverter air-conditioner has been on the Japanese market for over six years.

Although the inverter air-conditioner cost more than a non-inverter air-conditioner, user will usually buy the inverter air-conditioner. The reason for this is that in the winter when it is very cold outdoors, the inverter air-conditioner is able to have a large heat pump capacity and the SEER (Seasonable Energy Efficiency Ratio) can be improved by use of the heat pump thereby reducing maintanance cost.

As the capacity of the inverter compressor increases during high speed operation, we can make the compressor small,light weight and reduce cost.

However, operating the compressor at hight speed will increase the number of problems that will occur. The range of the compressor is limited by the capacity and efficiency of the air conditioner and weather conditions.

When we use a new compressor , we use several test to confirm its reliability. One of the most important test is the break up test in order to find the exact limit of the compressor. This is an easy way to judge the reliability of a new compressor.

The limit of the inverter compressor is judged by the maximum frequency,maximum discharge pressure,maximum temparature, etc.
We state that the problem in a compressor is related to the running conditions of the actual compressor under real life testing, but the aim of this report is to investigate the shaft and bearing only. We will not discuss problems with bearing clealance, variety, etc.
When we increase compressor rotational speed, the rotor amplitude increases and the rotor will touch the stator. Also, the main bearing upper load is large. By increasing discharge pressure at high speeds,the bearing load will be so increased that sub-bearing and main bearing lower end will show wear.

SAMPLE MODEL

We call a compressor which has variable rotational speed an inverter compressor. An inverter air-conditioner is shown in fig. 1.

The speed of the inverter compressor is changed by the load.
This is carried out by a sensor which senses the temperature in the room and sends a signal to the inverter system and adjusts the frequency of the compressor.

Figure 2 shows a rolling piston type rotary compressor. The upper part is the motor (rotor , stator) and the lower part is the compressor (bearing , cylinder) . The shaft is supported by two journal bearings (main bearing and sub bearing),the rotor is attached to the shaft,and the compressor section has the roller and shaft eccentric mass. Shaft unbalance mode shown in fig.3. Unbalance weights are attached to the upper part of rotor and balance weight U2,lower rotor and balance U1,and roller shaft eccentric mass U0.

The relation of U0, U1, and U2 is;

$$U1= U0 \quad L2/(L2 -L1)$$
$$U2= U0 \quad L1/(L2 -L1)$$

ANALYSIS MODEL

We use the matrix of two layers transmission for analysis. That is to say we made the first layer the oil spring with the shaft and bearing, and the second layer the fixed condition with the bearing husing, including the bearing and the cylinder. In the first layer the sub shaft has one oil spring and the main shaft has two (because of the difference in the length of the bearing) .
We decided on the number of oil springs for the experiment data through a compressor turnning test.
From this test we determined the natural frequency and the dangerous speed of the compressor.
The turnning test was carried out by measuring the response of the running compressor when it started to vibrate as shown in fig.4. In fig.5 we show an example of the experiment.

ROTOR AMPLITUDE

We can easily understand that the rotor amplitude is large during high speed operation. The first problem that we must solve is the rotor making contact with the stator during high speed. Namely, we must make the rotor amplitude smaller than the rotor gap between the stator and the rotor.

The experimental equipment for measuring rotor amplitude is shown in fig. 4. We measured the rotor amplitude directly with a gap sensor.

We show the results of our experiment and analysis with the matrix of two layers transmission in fig. 5.
In figure A and B we changed the rotor unbalance weight U1 , U2 as shown in table 1, and the shaft material of compressors C and D (Shaft young's modulus) as shown in table 2.
We are satisfied with the fact that the experiment data were nearly equal to the calculated data.
In fig. 8, calculation for shaft young's modulus, rotor length and rotor amplitude are shown.
That is to say,in order to reduce rotor amplitude;

(1) Reduction of unbalance weight
(reduce weight of U1, U2)
(2) Reduction of eccentric mass (shaft, roller)
(reduce weight of U0)
(3) Reduction of rotor stack height
(to decrease L2 - L1)
(4) Increase shaft rigity

Thus we can reduce rotor amplitude during high speed operation, and we can solve the problem of the rotor stator. But we must be careful that compressor casing vibration increases when we reduce the rotor umbalance weight in order to reduce rotor amplitude. The efficiency of the compressor is reduced when the rotor-stator gap was made bigger to keep the rotor gap from making contact with the stator.

BEARING LOAD

In the journal bearing, bearing load capacity is determined by shaft diameter, bearing length, shaft-bearing clearance and lubrication system. This report shows the calculated data of bearing load in connection with shaft rigity, rotor length and rotor unbalance weight. Bearing load becomes large when discharge pressure is large during high speed operation.
The only way to improve bearing reliability is to reduce this load.

In figures 9 and 10, shaft young's modulus, rotor length and bearing load of the main bearing upper and lower end are shown. In figures 11 and 12, rotor unbalance weight and shaft young's modulus, rotor length and bearing loads ratio of the main bearing upper end and lower end are shown.

We can easily see from these figures that reduction of rotor length and reduction of rotor upper balance weight are effective for reducing bearing load in the main bearing upper end, but hardly effective for reducing the lower end. Increasing the rigity of the shaft is effective in reducing main bearing upper and lower loads. We can increase shaft rigity by using a larger shaft and/or good material in the shaft. And at the same time we must consider the bearing material for the shaft to be congenial.

In figures 13 and 14, the compressor rotational speed and bearing loads of main bearing upper and lower end are shown. We can state the same theory above from figure 13 and 14.

Reduction of rotor length is effective in the reduction of rotor amplitude, but the necessary compressor torque must be taken into consideration.

LOCUS OF SHAFT

It is important to know the strength of the oil film and the bearing load capacity in the journal bearing. Because this is an important clue to the reliability of the bearing during high speed operation. In the journal bearing, an oil groove is formed on the bearing or the shaft. Generally, in the position where the oil groove locates, the permissible bearing load capacity is small.
The data for judging the reliability of the bearing under large load can be obtained from the locus of the shaft and its width.
Below we show the calculation data from the analysis model used above and the locus of the shaft for the experiment.

The compressor model is shown in fig. 15 . In analysis, we set the gas loads as the constant circulated loads to the crank shaft. We determined the circulated loads from compressing gas loads.
Compressing gas loads (Fg) is shown by;

$$Fg(@)=Ps(V(0)/V(@))H\ Rr\ Sin(1/2(2\pi-@))$$

total displacement volume is given by;

$$Vd=\pi H(Rr^2-Ra^2)$$

In fig. 16, the balancing mode of the shaft is shown. In fig. 17, the calculated data is shown. We found that in the main bearing upper end the low tension range was between 10° - 90°, and in the main bearing lower end eccentric was 100°.
Next, we show the locus data of the shaft in the actual experiment. We could find the locus by measuring the clealance of the shaft and the bearing with the gap sensor buried in the bearing. The results are shown in fig. 18. In figure 18, the shaft was nearly eccentric through-out its whole circumference in the main bearing upper end, and its eccentric was 0° -100° in the main bearing lower end. That is to say, the locus of the shaft is different between main bearing lower end and upper end.

We could have a better idea of fig. 17 and 18.

(1) The main bearing lower end
is mostly controlled by compressed gas load
(2) The main bearing upper end
is mostly controlled by the inertial load
of the rotor upper end unbalance mass

(In fig. 15, a load direction taken to the crank shaft is $\theta/2$, a inertial load caused by the rotor upper end balancer is θ, a inertial load caused by the rotor lower end is $(\theta - \pi)$ and a inertial load caused by the crank unbalance weight is θ.)

Reduction of bearing load is made by the eccentric locus rate smaller coupled with bearing load capacity.

CONCLUSION

Countermeasures for reducing compressor bearing loads at high speed operation will be follows;

(1) Reduction of eccentric mass

(2) Reduction of balancer weight of rotor

(3) Increase shaft rigity

(4) Reduction of rotor stack height

(5) Reduction of discharge pressure

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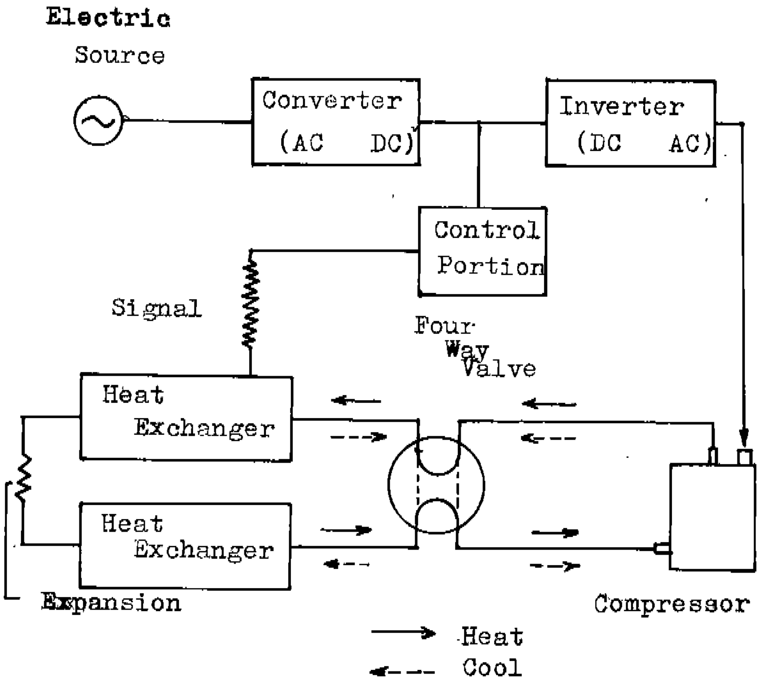


Fig.1 Inverter Air Conditioner

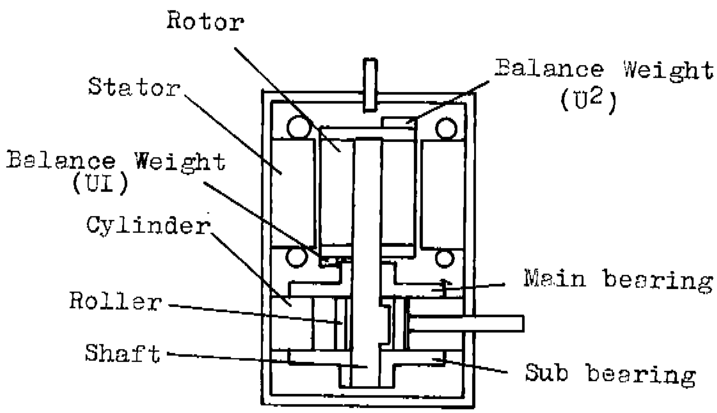


Fig.2 Section of Compressor

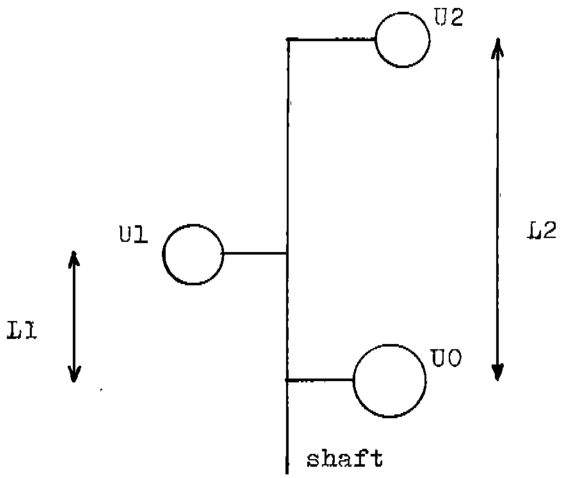


Fig3. Balancing Mode of Shaft

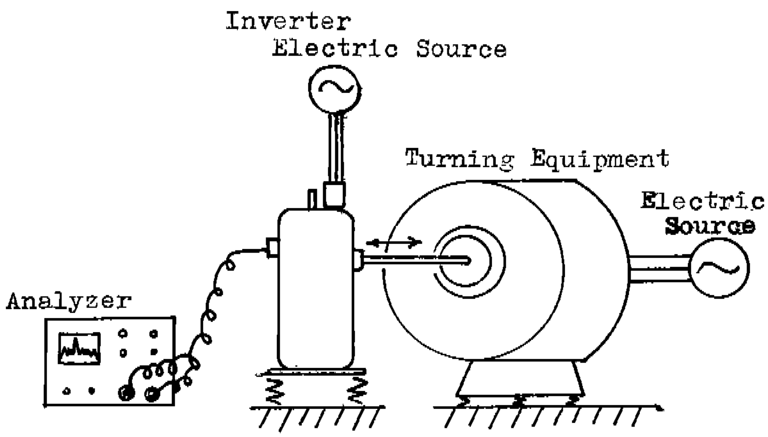


Fig4. Turning Test Equipment

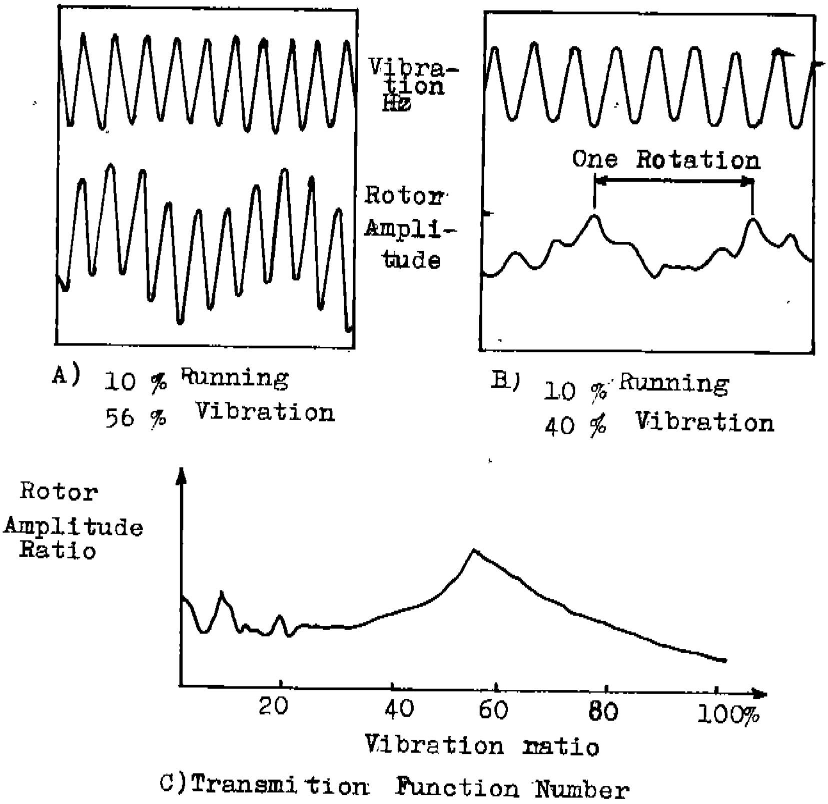


Fig.5 Example of Turning Test

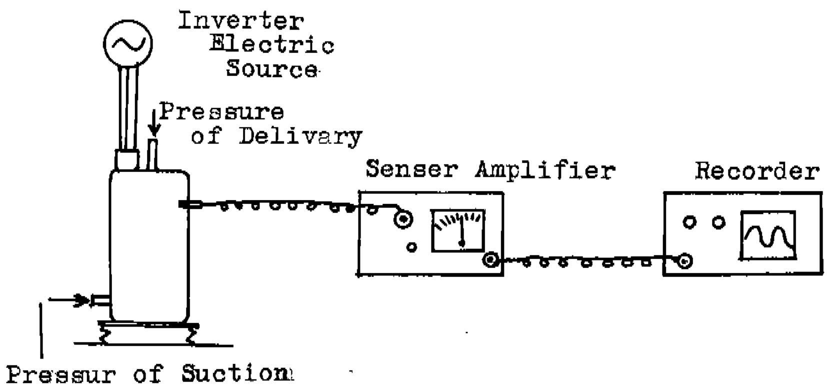


Fig.6 Equipment for Mesuring Rotor Amplitude

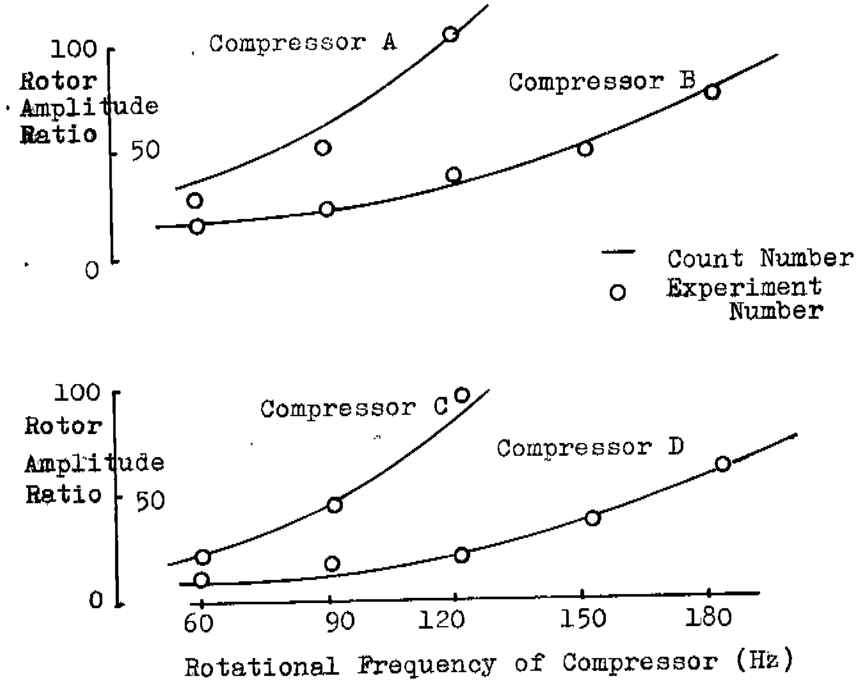


Fig.7 Comparison Experiment with Analyzing

	U1(gr)	U2(gr)
Compressor A	85	25
Compressor B	75	15

Table 1 Weight of U1,U2 in compressor A,B

Compressor C	10500
Compressor D	16500

Table 2 Shaft Young Modulus in Compressor C,D

In Fig.8,9,10 Rotational Speed 180 rps

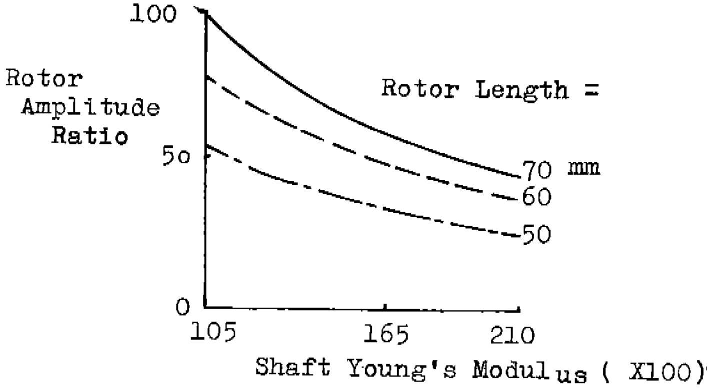


Fig.8 Shaft Young's Modulus (1)
and Rotor Amplitude Ratio

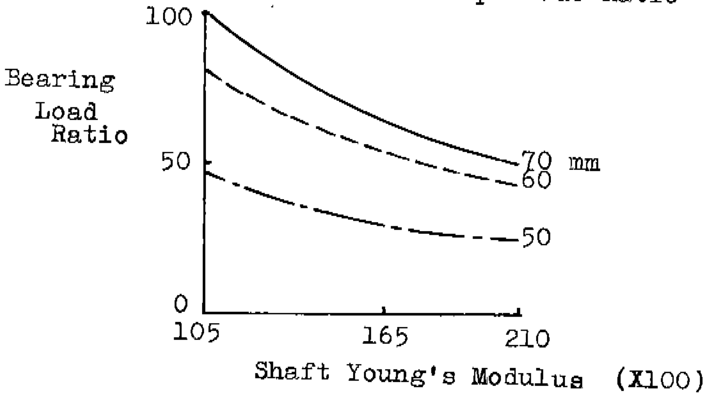


Fig.9 Shaft Young's Modulus (1)
and Main Bearing Load(Upper End)

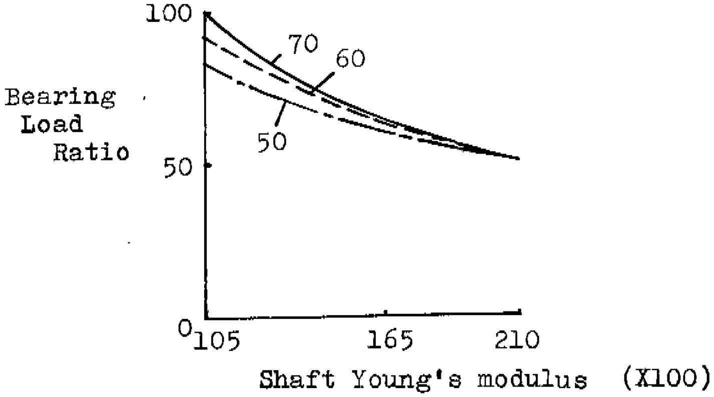
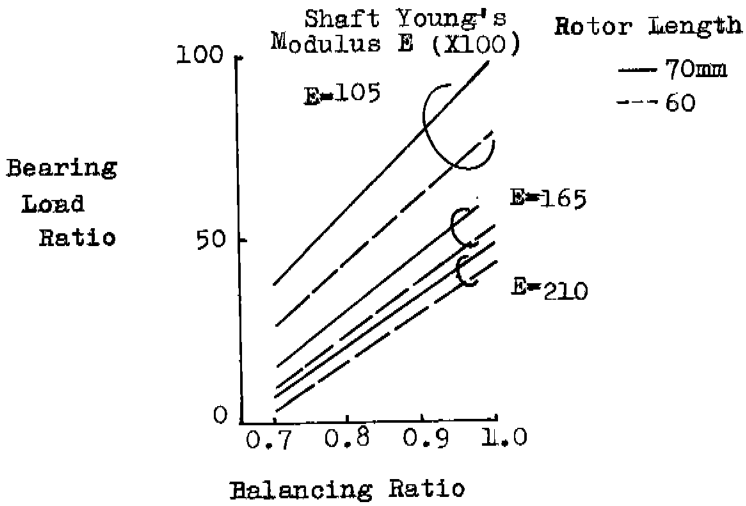


Fig.10 Shaft Young's **Modulus** (1)
and Main Bearing Load Ratio
(Lower End)

In Fig.11,12 Rotational Speed 180rps



(1)
Fig.11 Balancing and Main Bearing Load **Ratio**
(Upper End)

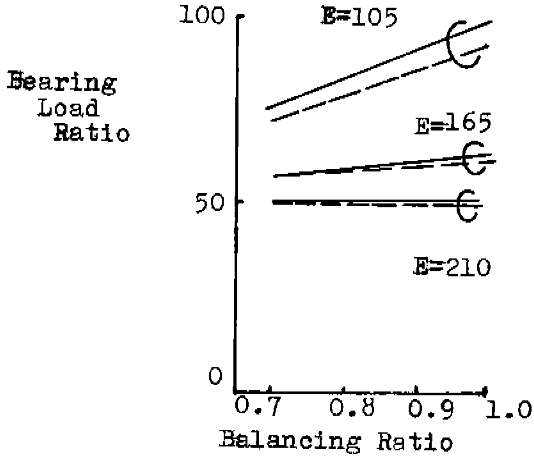


Fig.12 Balancing and Main Bearing Load Ratio (1)
(Lower End)

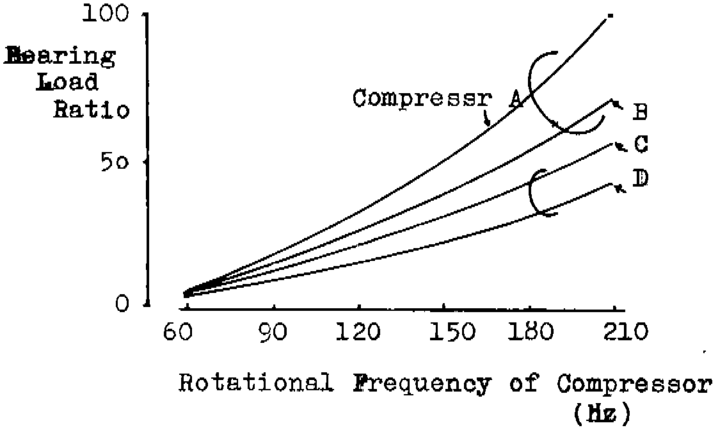


Fig.13 Compressor Rotational Speed
and Bearing Load Ratio
(main bearing upper end)

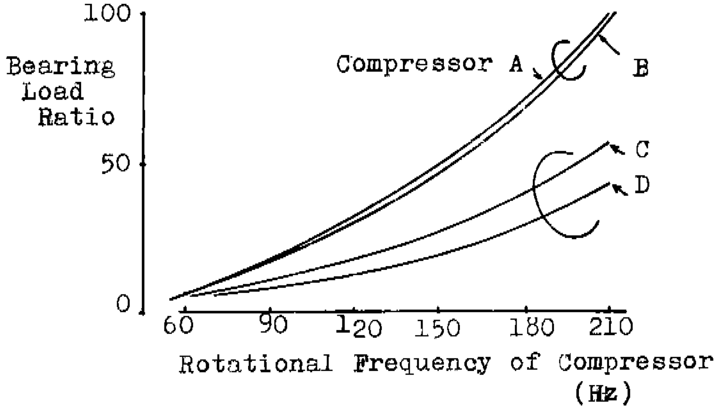


Fig.14 Compressor Rotational Speed
and Bearing Load Ratio
(main bearing lower end)

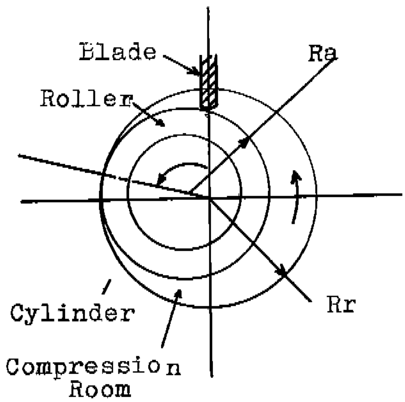


Fig.15 Compressor
Model

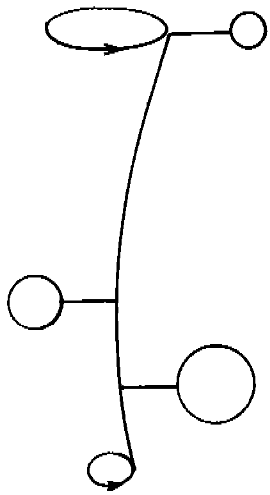


Fig.16 Balancing Mode of Shaft
Under High Speed Operation

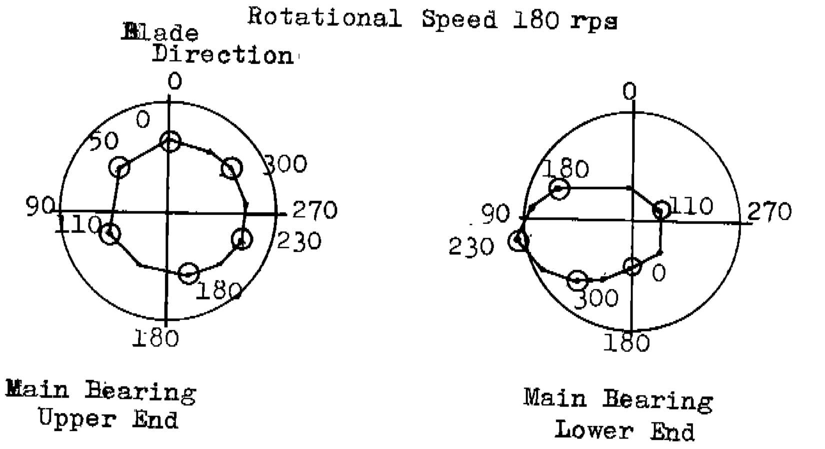


Fig. 17 Locus of Loads effcted to Main Bearing
in Analysis Under 180 rps

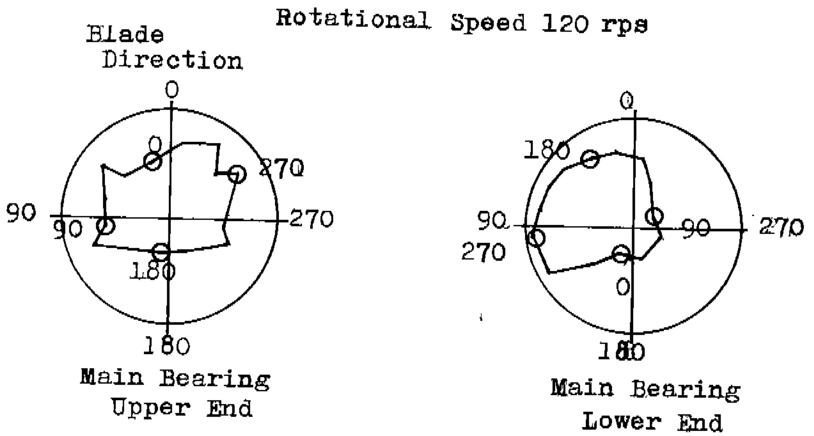


Fig. 18 Shaft Locus in Experimnt
Under 120 rps