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THE EFFECT OF REFRIGERANT DISSOLVED IN OIL ON JOURNAL BEARINGS RELIABILITY

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ABSTRACT

This paper presents how the refrigerant dissolved in oil affects on Journal bearings reliability. The bearing model test is performed using the hermetic test apparatus. The oil film pressure is measured under several types of bearing grooves, L/D ratios and mixing ratios of refrigerant to oil. As a result, the means by which the refrigerant dissolved in oil causes damage to journal bearings is made clear. That is, the fundamental problem is that the refrigerant gas, which evaporates and accumulates in the bearing clearance, disturbs the oil supply into the bearing. Also, it is found that the bearing groove, which is open to the bearing side, is effective to solve this problem.

INTRODUCTION

As far as the compressor for refrigerators or air-conditioners is concerned, the oil, in which refrigerant is dissolved, is necessarily used for journal bearings as lubricant. The necessary use of refrigerant dissolved oil characterizes compressor bearings and sometimes causes damage to them.

It is well-known that the bearing is apt to be damaged under the liquid-back condition whereby a lot of refrigerant is dissolved in oil. (1) The cause of the damage is supposed to be the lower viscosity of lubricant by more refrigerant dissolved in oil, or the compressibility of lubricant by the refrigerant bubbles. Each is certainly one of the causes, however, the fundamental cause is uncertain. The means by which the refrigerant dissolved in oil causes damage to journal bearings is not clear and needs to be clarified.

In this research, the bearing model test is applied as a method of making the means clear.

SYMBOLS

L/D : Ratio of bearing length to diameter

N : Rotational speed (s^{-1})

P : Oil film pressure (MPa)

Pa : Pressure in the apparatus (MPa)

Po : Oil supply pressure (MPa)

W : Load (N)

θ : Bearing angle (°)
[The center of oil supply hole is located at $\theta = 0$]

θw : Direction of load (°)

MODEL TEST

Test Apparatus

The test apparatus is shown in Fig. 1. It has hermetic structure with a mechanical shaft seal so as to simulate the compressor bearing. The main shaft is revolved using a motor driven by an inverter. A torque detector is installed between the motor and the main shaft to measure the rotational speed as well as the torque. The test bearing is statically loaded using a pair of coil springs. Their direction is at right angle to each other. The magnitude of the load is determined by the displacement of the coil springs. The lubricant, refrigerant-dissolved oil, is supplied from a reservoir to the test bearing. An oil pump is used to return the lubricant to the reservoir.

The main shaft has a diameter of 30 mm and is made of steel. The test bearing has a clearance of 30 μm and is made of bronze. The seven types of bearings are prepared, as shown in Fig. 2. They differ in L/D ratio and the type of grooves.

Oil Film Pressure Measurement

The oil film pressure is the most important characteristic of a journal bearing. It is measured at the mid-plane of the test bearing. A pressure sensor is mounted on the main shaft, as shown in Fig. 3, and connected to an amplifier by way of a slip-ring. The sensor is a semi-conductor strain-gauge type, and its diameter is 3 mm.

The rotational angle of pressure sensor is measured by an eddy-current gap sensor on the test bearing, which detects the slit on the main shaft at the same angle as the pressure sensor.

These data are recorded on a cartridge tape in digital form using a wave memorizer and personal computer, and they are processed using a desktop computer and XY plotter.

Test Condition

Refrigerant and oil are put into the test apparatus so as to be mixed at the regular ratio. The mixing ratios of refrigerant to oil are 0 and 20 weight-%.

The refrigerant gas pressure in the apparatus Pa is approximately 0.4 MPa in the case of 20% refrigerant. On the other hand, in the case of 0% refrigerant, nitrogen gas is fed into the apparatus so that the Pa may be adjusted to 0.4 MPa. The oil supply pressure Po is set 0.05 MPa higher than the Pa by controlling the valve on the connecting pipe.

The other conditions are as follows:
Rotational speed ; N = 20 s^{-1}
Load ; W = 2.8 kN (on type 1 to 6) 1.4 kN (on type 7)
Direction of load ; $\theta_w = 180°$

RESULTS AND DISCUSSION

Fig. 4 shows the oil film pressure distributions on the type 1 bearing. This bearing has L/D of 2 and no groove. The pressure profile of 20% refrigerant has a narrower width and higher peak than that of 0% refrigerant. This is the typical pressure profile of starved lubrication. (2) It is known that the bearing lubrication is changed into the starved condition by refrigerant dissolved in oil.

The cause of starvation is thought to be that the refrigerant gas, which evaporates and accumulates at the cavitation region of the bearing clearance, disturbs the oil supply into the bearing, as shown in Fig. 5, for the pressure around the oil supply position is almost the same as oil supply pressure in the case of 20% refrigerant, while it is below the oil supply pressure in the case of 0% refrigerant.

In the case of the type 2 bearing which has the oil supply groove, the same phenomena appear as in the type 1, as shown in Fig. 6. The starvation degree of the type 2 is less than that of the type 1. This is the effect of the oil supply groove, but it is not a fundamental solution.

Fig. 7 indicates the oil film pressure distributions on the type 3 bearing. This bearing has the groove, which is not connected to the oil supply hole and open to the bearing sides. Both the pressure profiles of 0% and 20% refrigerant are almost the same. This means that this groove removes the refrigerant gas accumulating at the cavitation region of the bearing clearance. It is found that this groove is effective for full lubrication. This type of groove shall be named "the gas removal groove".

Comparing the pressure profile of 0% refrigerant on the type 3 with that on the type 1, it is found that the lubricating condition on the type 3 is more starved. This reason is considered to be the prevention of oil circulation by the gas removal groove, as shown in Fig. 8.

Fig. 9 shows the oil film pressure distributions on the type 4 bearing, which has grooves for both oil supply and gas removal. In this case, starved lubrication doesn't occur, even if refrigerant is dissolved in oil. It is found that both grooves function well.

Fig. 10 indicates the oil film pressure distributions on the type 5 bearing. It has only one groove, whose lower edge is connected to the oil hole, and upper edge is open to the bearing side. The result is almost the same as that of the type 4. It is found that this type of groove can combine the functions of the oil supply and the gas removal groove.

Fig. 11 and Fig. 12 respectively show the oil film pressure distributions on the type 6 (L/D=1) and the type 7 (L/D=1/2). Since both bearings have no groove, the pressure distributions on 20% refrigerant are somewhat starved lubricating profiles. They are not as remarkable as in the type 1. This is because the smaller L/D ratio, the more refrigerant gas tends to go out the bearing sides.

CONCLUSIONS

(1) The means by which the refrigerant dissolved in oil causes damage to journal bearings is made clear. That is, the fundamental problem is that the refrigerant gas, which evaporates and accumulates in the bearing clearance, disturbs the oil supply into the bearing.

(2) This problem is more remarkable in bearings of higher L/D ratio.

(3) The gas removal groove is effective to solve this problem.

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REFERENCES

(1) Kawamura M., Tsuchiya K., Osada Y. and Kawahara H. "The experimental study for the lubrication of the crankshaft and journal bearings on multi-cylinder hermetic compressor" Refrigeration, Vol. 50, No. 573, 1975, pp 33 - 40.

(2) Articles A. and Heshmat H. "Analysis of starved journal bearings including temperature and cavitation effects" Trans. of ASME, Journal of Tribology, Vol. 107, No. 1, 1984, pp 1 - 13.

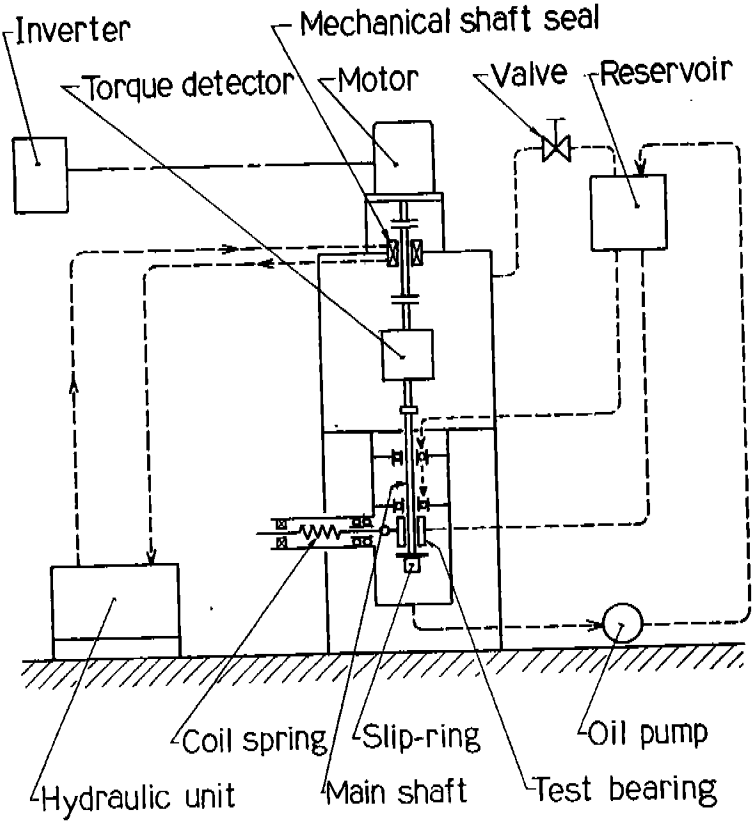


Fig.1 Test apparatus

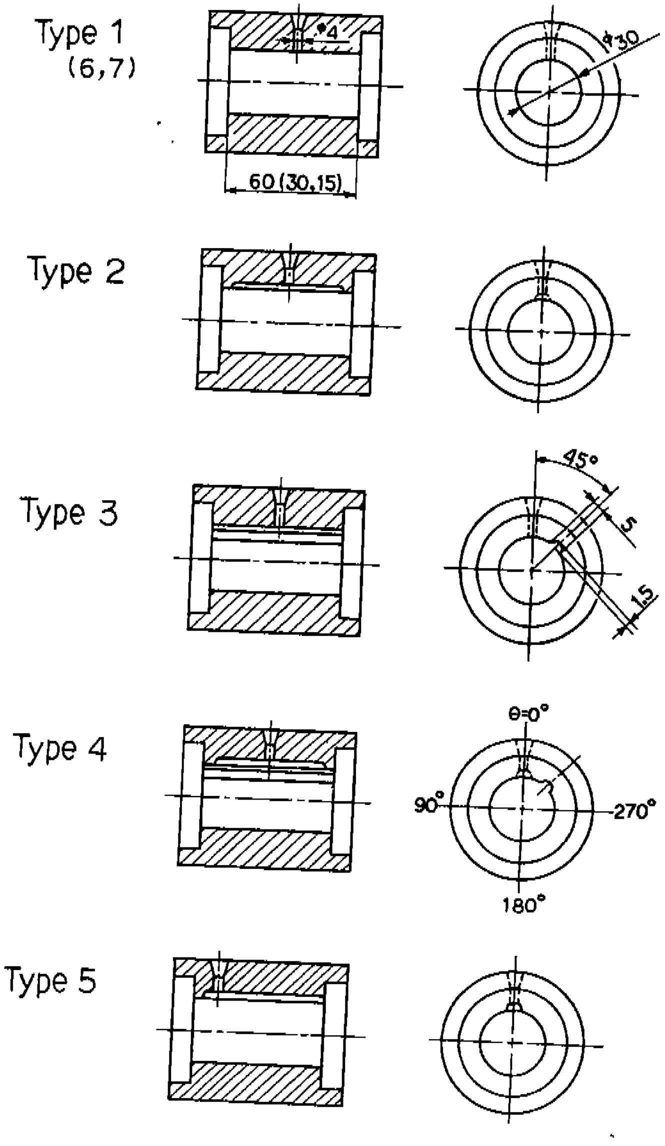


Fig.2 Test bearings

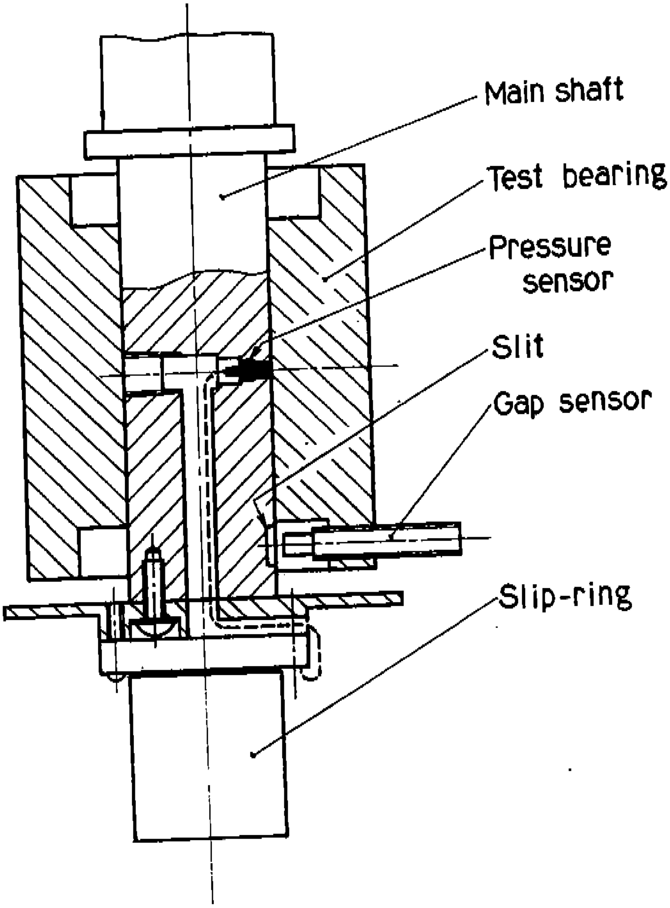


Fig.3 Oil film pressure measurement

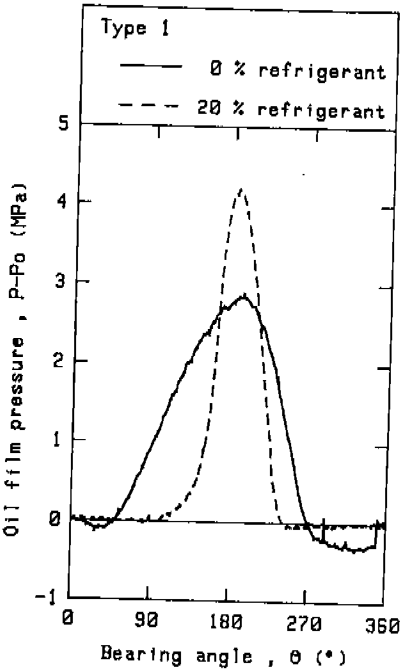


Fig.4 Oil film pressure distributions on the type 1 bearing

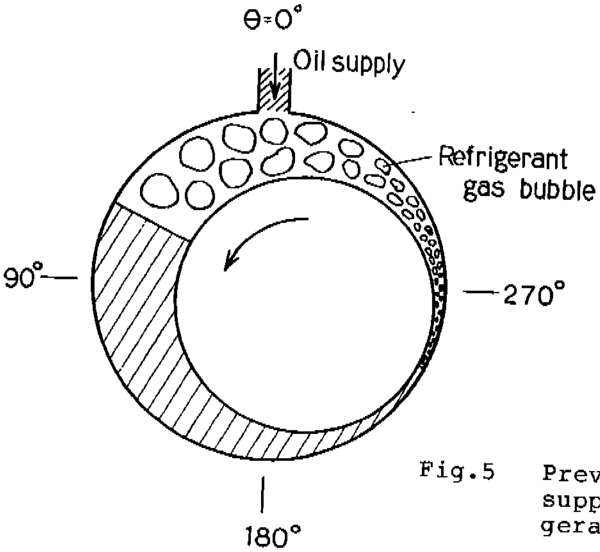


Fig.5 Prevention of oil supply by refrigerant gas

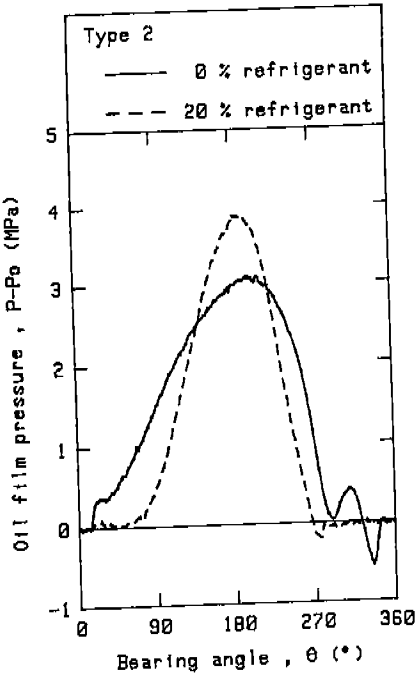


Fig.6 Oil film pressure distributions on the type 2 bearing

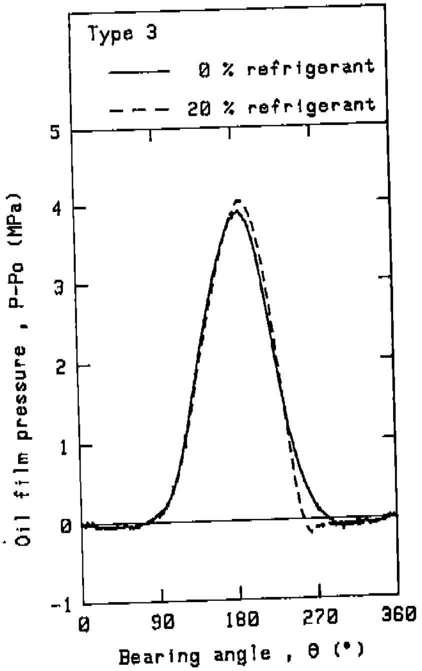
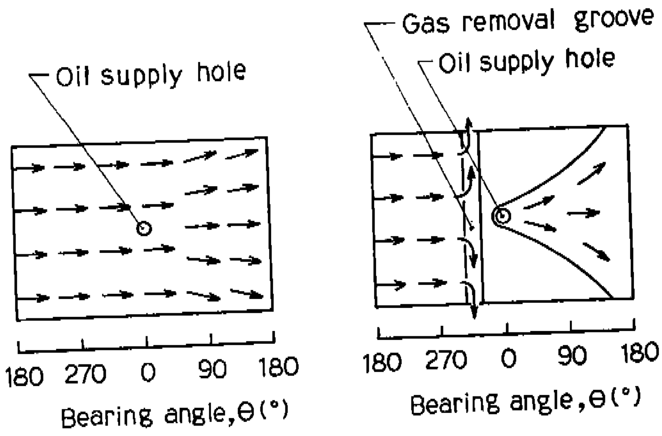


Fig.7 Oil film pressure distributions on the type 3 bearing



Type 1

Type 3

Fig.8 Oil flow in the bearing clearance

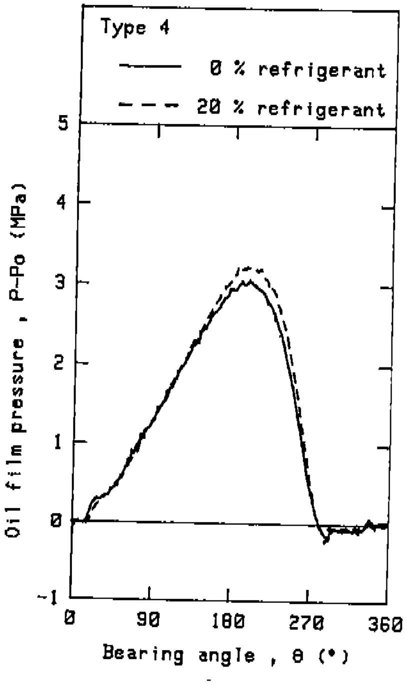


Fig.9 Oil film pressure distributions on the type 4 bearing

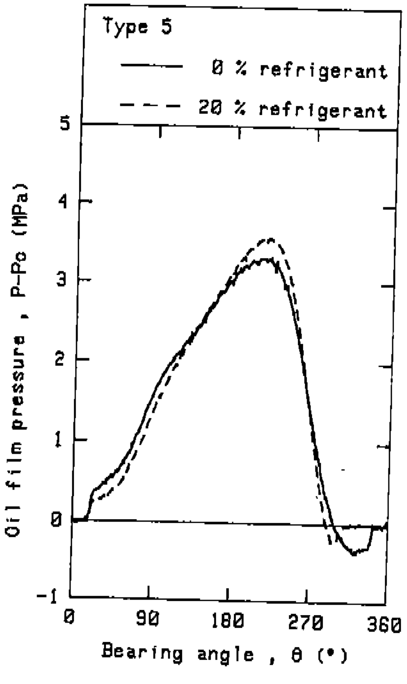


Fig.10 Oil film pressure distributions on the type 5 bearing

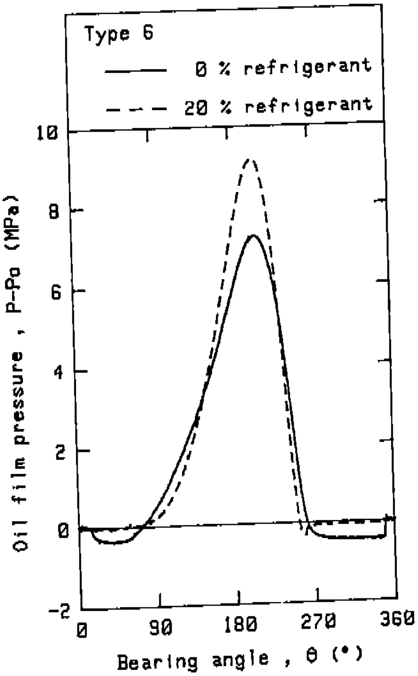


Fig.11 Oil film pressure distributions on the type 6 bearing

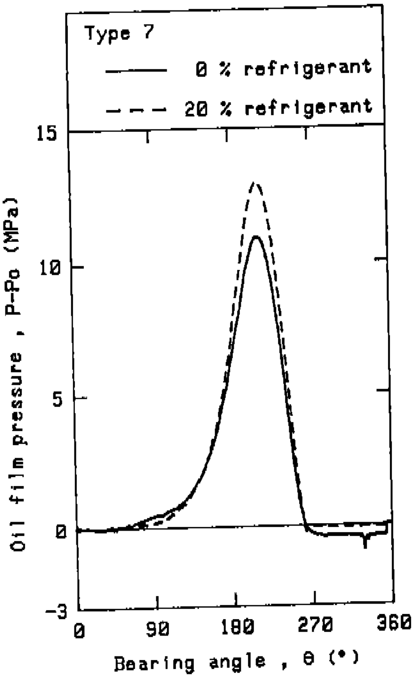


Fig.12 Oil film pressure distributions on the type 7 bearing