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ABSTRACT

This paper presents a technique of measuring directly the friction losses occuring on a single piston reciprocating compressor mechanism. Special attention is given to the parallel-face thrust bearing in order to define its lubrication-friction mechanism, and to evaluate the friction losses as function of the axial load and the lubrication oil temperature. A specific device, designed for the laboratory measurements, is described in conjunction with the measurement technique. Laboratory measurements have identified the change in the friction losses by replacing the conventional parallel-face thrust bearing by other devices like; ball bearing, tilted-pad thrust bearing, anti-friction rings, and reduced bearing width. Local measurements of the friction losses were made on the thrust bearing, main radial bearing, and the eccentric bearing together with the cylinder piston. The results are compared to the theoretical models. The thrust bearing measurements seem to be the most important, where the theory usually makes poor predictions.

INTRODUCTION

Due to the requirement of developing high efficient household refrigerating compressors, the correct evaluation and consequent reduction of the mechanical friction losses, have become an important issue.

The behaviour of the hydrodynamic journal bearings, can be estimated within a good accuracy based on the hydrodynamic lubrication theory. However, the behaviour of the small parallel-face thrust bearing is not clearly understood.

As reportted in Ref.[1], the theory for the parallel-face thrust bearing is that " the lubricant will experience a rise in temperature due to viscous friction. This rise in temperature produces a slight expansion of the lubricant and increases its volume as it flows through the clearance space. Since the cross-sectional area of the clearance remains constant, a relative constriction of the flow takes place, producing some pressure build-up and load-carrying capacity." This theory is known as the "thermal effect".

The development of our specific study, was motivated in order to find answers for questions like; How does the load on the thrust bearing affect its friction losses? What lubrication mechanism takes place at the thrust bearing? What equivalent friction coefficient should be used for simulation purposes? What change in the friction losses do occur if the conventional parallel-face thrust bearing is replaced by devices like ball bearing, tilted-pad thrust bearing, and anti-friction rings? Answers for those questions will give a strong support for designing the most efficient thrust bearing, considering the cost and power consumption.

The experimental evaluation is based on the friction torque measurement of the unloaded running compressor. A special torque transducer was designed for this purpose.

Reference [2] describes a technique of measuring the actual shaft torque of a rolling piston type rotary compressor, under various operating conditions. The shaft torque was obtained by the torsional angle measurement using an eletromagnetic torque sensor. The overall friction torque was then calculated by subtracting the gas torque from the total load torque.

The overall friction losses can also be calculated, say, from a standard calorimeter test, by measuring the total power input, the speed, the motor winding temperature, and the P-V diagram. The motor efficiency can be measured under the same conditions, and the friction losses are calculated by subtracting the compression power from the shaft power.

The improvement of the technique presented in this paper, is the identification of the local losses. Knowing separately the thrust bearing loss, the main bearing loss, and the connecting rod + cylinder-piston loss, is very important from the design standpoint.

THE MEASUREMENT TECHNIQUE

A general understanding of the measurement technique can be drawn from Figure (1) and Figure (2). Figure (1) gives a general view of the torque transducer with indication of the extensometers location. The compressor pump is attached to the upper ring of the transducer, while the motor is attached to the bottom ring. Since the compressor runs unloaded (without cylinder head), any motor reaction that is measured represents the friction torque.

Special attention has to be given for the extensometers installation. Correct measurements can only be obtained if the strain-gages are located at 180 degrees opposite bending legs, and one strain-gage at both sides of the legs. This makes the transducer be unsensible for the electromagnetic radial forces.

The block diagram of Figure (2), shows the three major branches that represent the torque, speed and temperature measurements, which are the variables of interest.

The transducer was statically calibrated by the dead weight procedure. Any PC output value represents the average of 100 measured data.

EXPERIMENTAL RESULTS

The experimental evaluation was performed on an EMBRACO compressor, with a cooling capacity of 950 Btu/h at the standard LBP check-point operating condition. All tests were run without load on the cylinder, and using 3GS lubrication oil (viscosity of approx. 6.0 cP at 82 [C]). The basic mechanism is shown on Figure (3).

The objective is to identify changes in the friction loss, by changing the oil temperature, and the load, shape and size of the thrust bearing. The local losses at the main bearing, thrust bearing, and eccentric bearing + cylinder-piston, were also measured for the original design in order to compare with the theoretical results.

Thrust Bearing Loss as Function of the Axial Load

The measurements were made at approximately 82 [C] oil temperature, and 3460 [rpm] running speed. The shaft + rotor weight gravity force corresponds to 1187 [gf]. The thrust bearing weight was reduced step by step to the point of total shaft suspension. Figure (4) presents the results for the indicated types of thrust bearings. For all measurements, the standard deviation was less than 0.3 [W], with most of them below 0.15 [W].

Thrust Bearing Loss as Function of the Oil Temperature

The load on the bearing was kept constant and equal to the weight gravity force of 1187 [gf]. The thrust bearing net consumption was obtained from the difference between the loaded and unloaded (shaft suspended) condition. The speed reference for the loaded condition was 3460 [rpm] which increased to about 3475 [rpm] for the unloaded condition.

The results are shown graphically on Figure (5), with the data adjusted to a 2nd order curve by the least square method. The average oil temperature may vary by approx. 1 unit between the tests, for the different bearing types.

THEORETICAL MODELS

The objective of this section is to present the basic theoretical models to evaluate the friction losses of the mechanism.

Thrust Bearing

For the thrust bearing, the friction losses can be estimated by knowing either the friction coefficient (k), or the oil film thickness (h). Therefore, based on the measured friction losses, the objective is to evaluate (k) and (h) considering the boundary lubrication theory and the hydrodynamic lubrication theory respectivelly.

From the boundary lubrication theory, the friction losses are calculated by;

$$E = k\, F\, r\, w \qquad (1)$$

where E represents the measured friction loss in [W]; k the equivalent friction coefficient; F the constant axial load in [N]; r the average bearing radius in [m]; and w the rotational speed in [rad/sec].

For the hydrodynamic lubrication theory, the thrust bearing friction loss can be evaluated by;

$$E = \frac{2\pi\ U\ w^2}{h} \left[\frac{r_o^4 - r_i^4}{4} \right] \qquad (2)$$

where U represents the oil viscosity in [N sec/m2]; h the oil film thickness; ro the bearing outside radius; and ri the bearing inside radius.

The following table presents the equivalent friction coefficient (k) and the equivalent oil film thickness (h) as function of the thrust bearing type. The oil temperature was maintained around 82[C] (oil viscosity around 6.0e-3 [N sec/m2]), and the load was the gravity weight force of 11.87 [N].

Thrust brg type	k	h [um]
Original design	.056	8.2
Brg width reduced by 1/3	.049	5.8
Brg width reduced by 2/3	.085	1.5
Tilted-Pad brg	.04	11.5
Ball brg	.006	-
Orig. design + teflon ring	.051	9.1

Radial Bearings

The friction losses in radial bearings are evaluated by Petroff's equation, which applies the Newton's law of viscosity to journal bearings, and has the following form;

$$E = \frac{2\pi\ U\ L\ R^3\ w^2}{C} \qquad (3)$$

Cylinder-Piston

If hydrodynamic friction is considered, then the Newton's law of viscosity can be applied which ends up with the following equation;

$$E = \frac{4\ U\ D_p\ L_p\ E^2\ w^2}{\pi\ C_p} \qquad (4)$$

with Dp being the piston diameter, Lp the piston length, E the shaft eccentricity, and Cp the cyl-piston radial clearance.

Comparison Between the Measured and Theoretical Data

In the following table we compare the local losses of the original design, for both the measured and theoretical data, considering for both cases an average oil temperature of 82 [C] (viscosity of 6.0e-3[N sec/m2]) and rotational speed of 3460 [rpm].

Friction loss location	Measured	Calculated
Main bearing [W]	12.9 (61.1%)	13.0
Thrust bearing [W]	2.9 (13.8%)	-
Ec. bearing + cyl-piston [W]	5.3 (25.1%)	1.6+4.0=5.6
TOTAL LOSS [W]	21.1	

CONCLUSIONS

- From Figure (4) we identify that hydrodynamic friction is occuring on the thrust bearings of types A , B and D , while type C is approaching boundary friction.

- There is no significant advantage in reducing the axial load in order to reduce the friction loss on the thrust bearing, unless for the case of boundary lubrication.

- In Figure (5) we identify a gain in friction reduction if comparing the tilted-pad thrust bearing to the original design.

- For high oil temperatures (low oil viscosity), the thrust bearings with reduced width tend to present a point of minimum friction loss much earlier than the original design, which indicates that beyond that point the oil film thickness becomes very small, or boundary friction occurrs.

- A friction coeff. of approx. 0.05 was identified considering an oil temperature of 82 [C]. For higher temperatures, or equivalent viscosities, this coeff. will still be reduced, which indicates that the thrust bearing friction coefficient is much lower than the values that were commonly used.

- Figure (5) also indicates that for type B a reduction in the bearing area represented a friction loss decrease, while for type C , with a greater reduction in the bearing area, resulted a friction loss increase. This happens due to the great reduction in the oil film for type C (Figure 6), which has a much greater influence on the friction loss than the reduction in the bearing area.

- A good agreement between the measured and calculated friction losses was obtained for the radial bearings and cylinder-piston, considering the actual running clearances.

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[2] Sakitani, K.; Koiwa, J. and Maekawa, T., "Performance Evaluation of an Hermetic Refrigeration Compressor Using Torque Measurement Technique", 1986 International Compressor Engineering Conference - At Purdue, pp. 228-241.

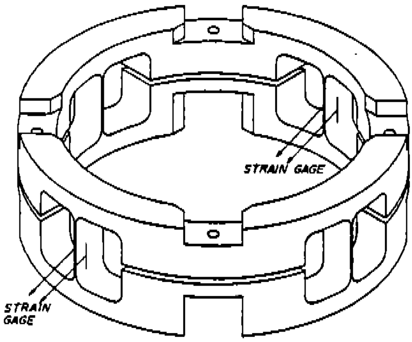

FIG. 1 — GENERAL VIEW OF THE TORQUE TRANSDUCER

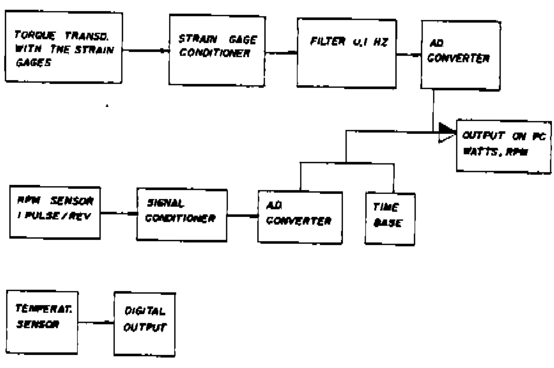

FIG. 2 — BLOCK DIAGRAM OF THE INSTRUMENTATION

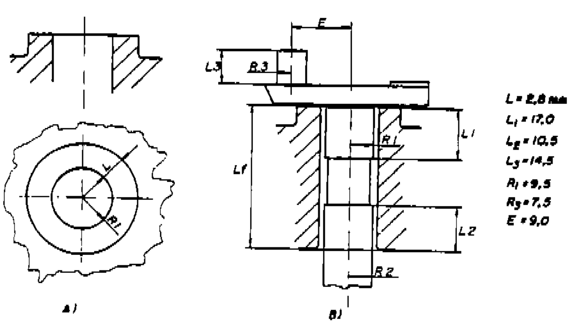

FIG 3 — GENERAL VIEW OF THE MECHANISM
A) THRUST BEARING
B) ECCENTRIC SHAFT/MAIN BEARING

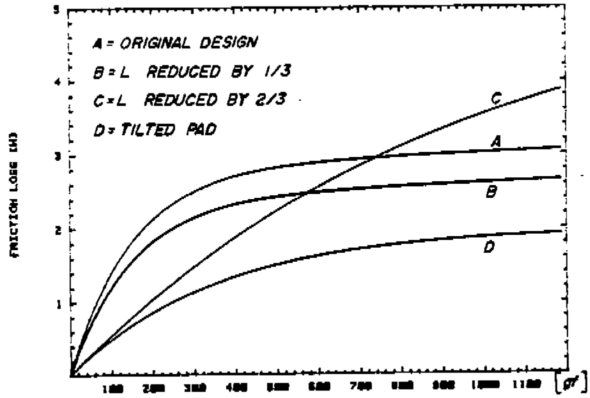

FIG.4 THRUST BRG. LOSS X AXIAL LOAD.

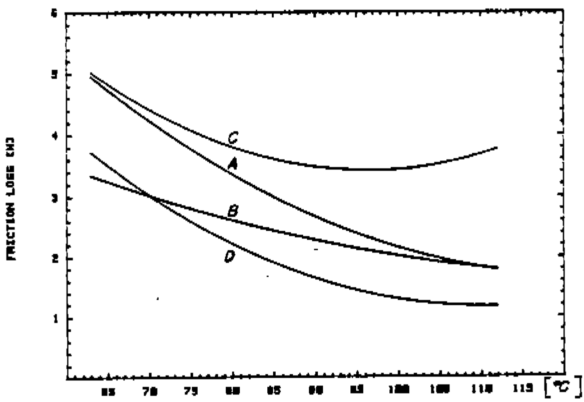

FIG.5 THRUST BRG. LOSS X OIL TEMPERATURE.

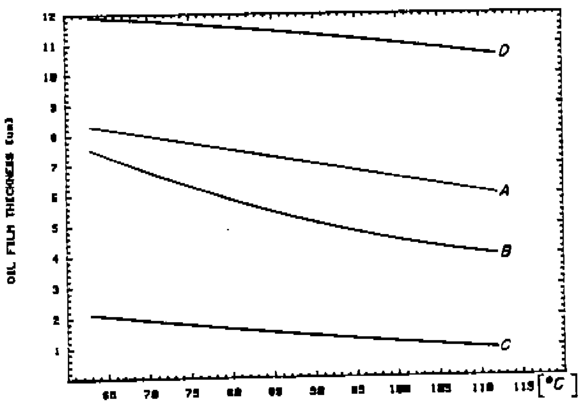

FIG.6 THRUST BRG FILM X OIL TEMPERATURE

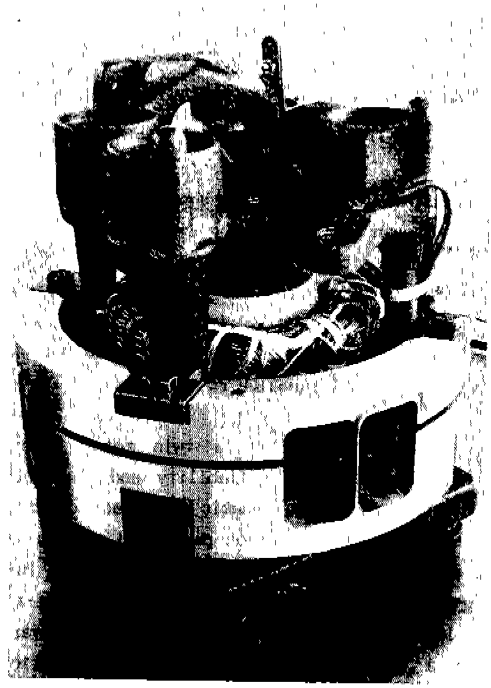

FIG. 7 - COMPRESSOR MECH. MOUNTED ON THE TRANSDUCER

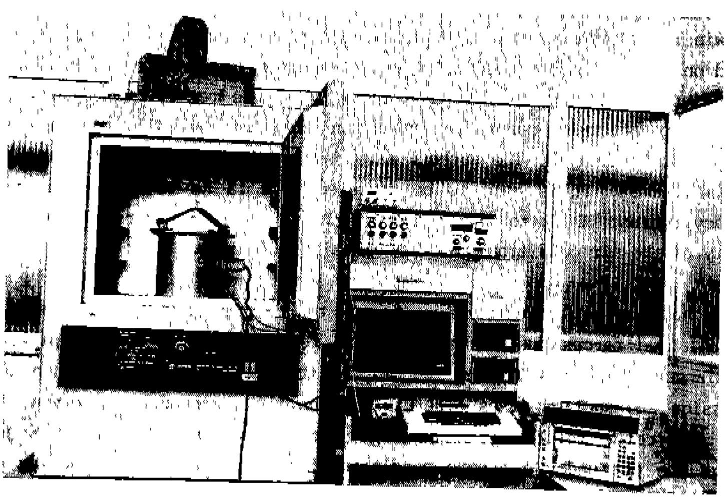

FIG. 8 - GENERAL VIEW OF THE EXPERIMENTAL SET-UP