HIGH EFFICIENCY FLUID FILM THRUST BEARINGS FOR TURBOMACHINERY

by
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Bernard S. Herbage received his BSME from Bradley University where he was also a member of the honorary fraternity of engineers, Tau Sigma. The first ten years of his engineering career were spent with Allis Chalmers in Milwaukee, Wisconsin in the mechanical design section of the Thermal Bower Department. During this time Mr. Herbage devoted considerable time in the design and testing of thrust bearings with much emphasis on thrust bearing load capacity and efficiency.

After leaving Allis Chalmers, he continued to pursue his interest in fluid film bearings by moving to Waukesha Bearings Corporation as chief engineer for four years. There he worked with virtually every major high speed rotating machinery manufacturer in the design and application of fluid film thrust and journal bearings.

Shortly after moving to Texas, Mr. Herbage helped found Centritech, an end-user oriented company specializing in the design and manufacture of fluid film bearings, with emphasis on problem solving. He is a registered professional engineer in the State of Texas and a member of the Turbomachinery Symposium Advisory Committee.

ABSTRACT

Fluid film thrust bearings in use on high speed high capacity turbomachinery absorb a great amount of energy in performing their task of positioning rotors. A review of thrust bearing fundamentals along with the latest design concepts briefly outline how thrust bearing performance can be substantially improved. The major improvements come from selection of materials and methods of lubrication.

INTRODUCTION

Fluid film thrust bearings used in high speed high capacity turbomachinery absorb a great amount of energy in performing their obvious task of positioning rotors. By careful analysis and the latest design criteria, we are now able to design more efficient thrust bearings, saving hundreds of horsepower, and thousands of dollars with no sacrifice in safety or reliability.

As a typical example of energy absorbed by thrust bearings, let us look at a syngas compressor train in an ammonia plant. This particular train has three centrifugal multistage compressors and two steam turbines. There are five active thrust bearings which have a combined power loss of 790 horsepower. The two drive turbines produce approximately 25000 horsepower meaning that the thrust bearing losses are three percent of the total power requirements of the train. One can quickly appreciate that, if the power loss in the thrust bearings could be cut by one-third, the efficiency of the unit would be increased by one percent.

Fluid film thrust bearings are designed in a wide range of types from the simplest flat land thrust bearing (Figure 1) to the tapered land thrust plate (Figure 2) to the tilting pad type (Figure 3) all the way through to the sophisticated tilting pad, self-equalizing, directed lube design (Figure 4).

The selection of a particular thrust bearing design is based on continuous thrust loads, transient thrust loads, unit speed, alignment considerations and the critical nature of the machines application.

The scope of this discussion will direct itself to the higher load, 250 to 1000 psi; higher speed, 10000 feet per minute mean rubbing speed; and more critical machines using tapered land or tilting pad thrust bearings. How energy loss can be reduced without sacrificing reliability in a safe monitored way is the essence of this paper.

A brief review of fluid film thrust bearing fundamentals will possibly be of help to those machinery users and designers who do not get involved with thrust bearings on a day to day basis. Only hydrodynamic thrust bearings will be discussed.

In the cross section of a thrust bearing (Figure 5) the thrust collar attached to the rotor is moving with respect to the stationary tapered land thrust plate. A non-compressible fluid is forced through the assembly between the thrust plate and the collar and back to drain. The viscous shear forces between the oil and the adjacent surfaces force the fluid into the converging wedge established by the machined surface of bearing and the flat surface of the thrust collar. Ignoring side leakage, all the oil going into the leading edge has to exit the trailing edge and since oil is incompressible a pressure is developed to induce a higher flow rate at the trailing edge. In fact, what also happens is that the pressure builds up to separate the collar from the stationary thrust bearing and creates an oil film at the trailing edge of each land sufficient to produce an equilibrium in oil flow.

If, instead of using a fixed tapered bearing, we use a tilting pad bearing (Figure 6) with a central pivot, the same hy-

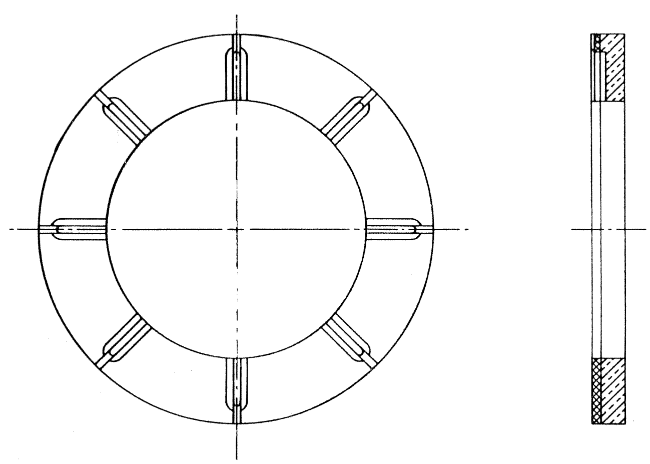


Figure 1. Flat Land Thrust Bearing.

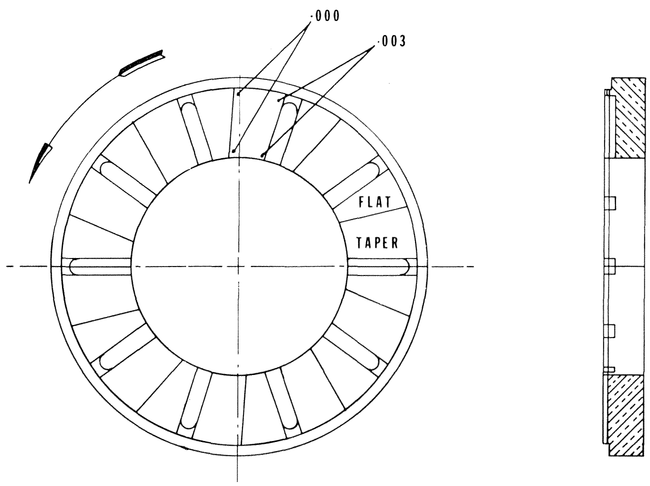


Figure 2. Taper Land Thrust Bearing.

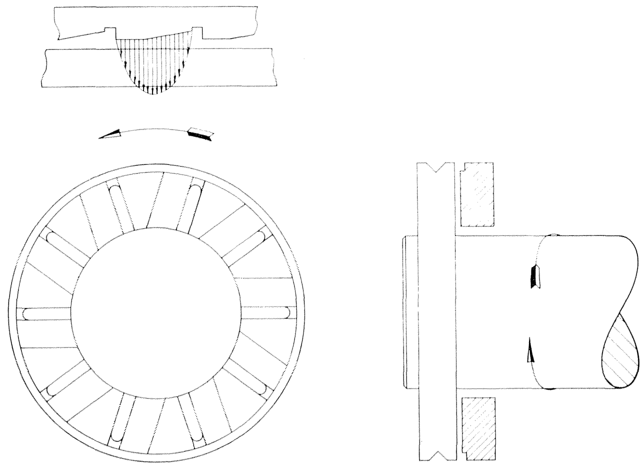


Figure 5. Oil Film Pressure Profile.

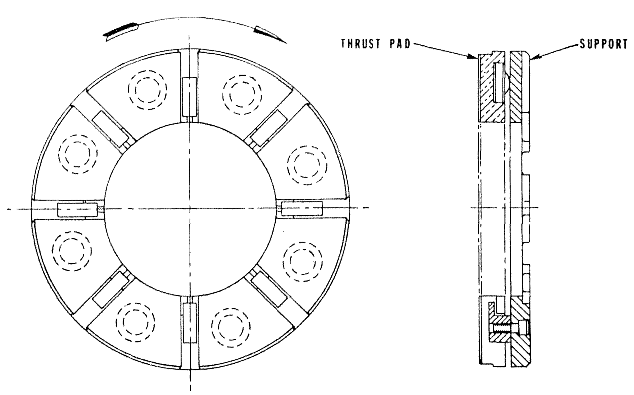


Figure 3. Tilt Pad Thrust Bearing.

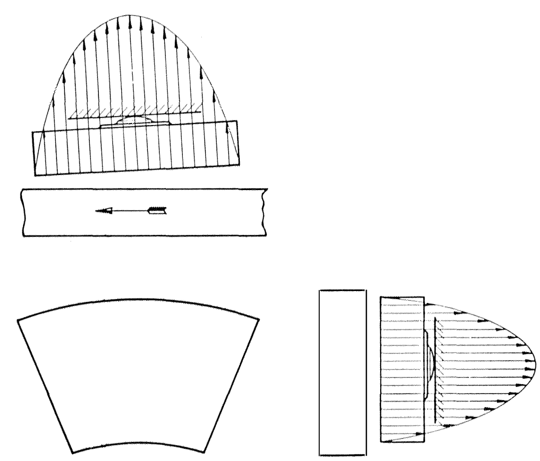


Figure 6. Tilt Pad Pressure Profile.

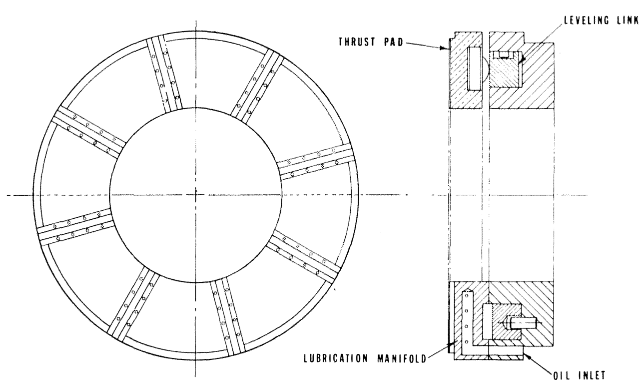


Figure 4. Self-Equalizing Thrust Bearing with Tilt Pads and Directed Lubrication.

drodynamic process occurs. In this case the pad tilts to produce the wedge necessary to develop sufficient pressure to support the imposed thrust load.

These simple illustrations are intended only to point out that viscous shear is used to separate moving and stationary parts. Viscous shear stated in layman's terms is friction.

For purpose of a general understanding the power loss in a simple ring type thrust bearing [1] of outside radius R_2 in inches and inside radius R_1 is

$$Hp = 2.61 \times 10^{-6} \mu \frac{N^2}{t} (R_2^4 - R_1^4)$$

Where μ = Viscosity lb-sec/IN²
N = Shaft speed RPM
t = film thickness, inches

In other words, the loss in a thrust bearing is directly proportional to the fourth power of the radius.

This is our first clue to potential energy savings in a thrust bearing. That is, for a given thrust load the thrust bearing should be designed with as small a diameter as possible.

The two major factors dictating the size of a thrust bearing have been pointed out by numerous bearing experts but are worth repeating here. They are:

1. Bearing surface metal temperature
2. Minimum fluid film thickness

In conventional steel backed babbitted thrust bearings the metal temperature of babbitt is the predominating control. From the pressure profiles shown on Figure 6, it can be seen that a high pressure zone exists approximately in the center of the pad. This is also one of the hottest zones in the oil film passing through the bearing. Steel being a relatively poor conductor resists the flow of the heat from the bearing surface to bearing housing and to the oil. Heat concentrated and combined with this high pressure zone causes plastic flow or melting of babbitt at fairly low specific loads. Typically, machines are designed for unit loads on thrust bearings of 250 psi to 400 psi. Tests have shown that failure of a steel backed bearing of rather sophisticated design can occur at as low as 650 psi.

Although tilt pad bearings dissipate more heat by being bathed with cool oil on the sides and back, they lose some of that gain by the thermal distortion of the pad due to temperature gradients across the pad. In this exaggerated illustration (Figure 7), it can be seen that the pad simulates a bimetallic strip. The pad surface toward the babbitt side expands due to heat input while the back side stays relatively cool. Consequently, a crown forms at the point of maximum pressure over the pivot and progressively the pressure profile distorts to increase the specific load at the high pressure zone, the film thickness decreases, the temperature goes up and either equilibrium occurs or exponential failure results.

So, here we have a clue to increasing load capacity of thrust bearings permitting us to use a smaller diameter and reduce energy requirements; namely, use backing materials with better thermal conductivity.

Noted bearing authorities, Bahr [2] and Gardner [3], have tested several of these better materials. The author has also tested and found that with proper mechanical design to compensate for differences in strength and stiffness these high conductivity metals can be used to produce thrust bearings with thrust capacity well over 1000 psi. Figure 8 is typical of bearings of this type, showing the difference in thrust capacity for a steel backed shoe and one having a high thermal conductivity backing. This curve is based on testing experience of the author and is meant only as a general guideline.

As an example of the significant energy savings that can be achieved on high capacity thrust bearings operating at high speeds, let us investigate a steam turbine using a 10½ inch diameter steel backed tilt pad thrust bearing operating at 11000

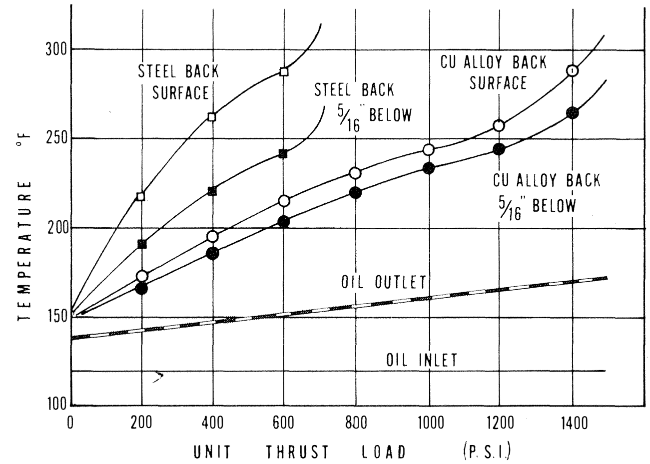


Figure 8. Unit Thrust Load (psi).

rpm. This bearing has a 5¼ inch bore and a thrust area of 55 square inches. As shown on Table 1, at a thrust load of 27500 pounds (500 psi) the power required to operate this thrust bearing is 190 horsepower. This is actual horsepower in a flooded cavity. Designing for a thrust load of 750 psi using high conductivity backing material would require 36.7 square inches of area instead of 55. The 5.25 bore has to be maintained requiring an outside diameter of 9 inches. The actual power required to operate this 9 inch thrust bearing is 130 horsepower or 60 horsepower less than the 10½ inch bearing.

Most thrust bearings in high speed turbomachinery operate in a flooded cavity; either throttling on the inlet and overflowing on top outlet or throttling on the outlet with a relatively unrestricted inlet. This flooded cavity is necessary to assure a full supply of oil and adequate cooling to all pads while maintaining satisfactory thrust shoe metal temperatures and bulk oil discharge temperatures. It has been found that an unexpected additional power loss over calculated viscous shear loss occurs in a flooded thrust cavity due to viscous pumping at the periphery of the thrust collar and churning of oil between adjacent thrust pads.

Considerable design and experimental work has been conducted by New [4] on the concept of injecting lubricant between the thrust pads in an effort to more efficiently utilize the lubricant and minimize these parasite losses of churning and viscous pumping. Applying this knowledge and experience to the 10½ and 9 inch thrust bearings sited previously — a considerable savings in energy can be realized by using the patented directed lubrication principles. As shown in Table 1, the power loss in a 10½ inch thrust bearing with directed lubrication at 11000 rpm and 500 psi load is 125 hp compared to the 190 hp with a conventional flooded cavity. This is in fact a savings of 65 hp. The 9 inch thrust bearing with directed lubrication at 11000 rpm and 750 psi loading has a power loss of 80 hp compared to the 130 hp with a flooded cavity. Going all the way on this unit from a 10½ inch flooded bearing to the 9 inch directed lube bearing would result in a savings on one thrust bearing of 110 hp.

In order to realize all the energy savings possible, using directed lubrication, it is necessary that the flooded cavity be modified or redesigned in such a way as to freely drain away the lubricant from the immediate vicinity of the thrust collar. Figure 9 shows a typical thrust end assembly. Three 1 inch drain holes in the cavity around the thrust collar result in a

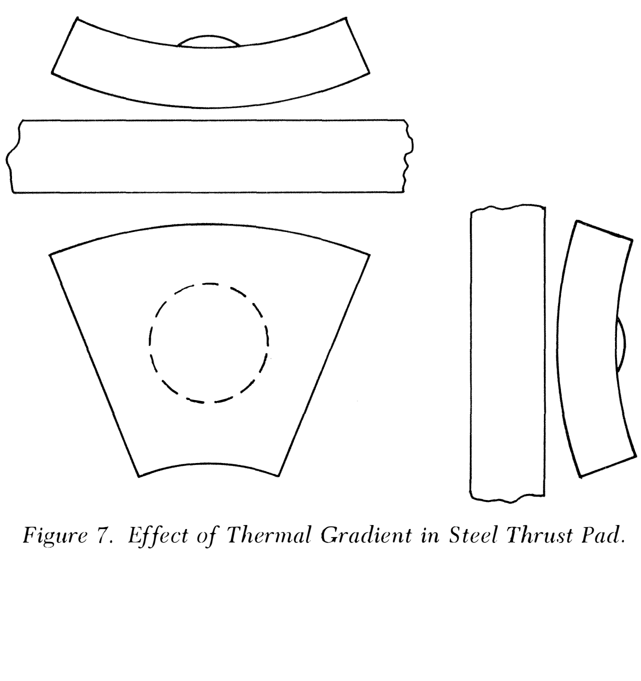


Figure 7. Effect of Thermal Gradient in Steel Thrust Pad.

TABLE I

BEARING SIZE AND TYPE	THRUST AREA SQUARE INCHES	THRUST LOAD POUNDS	UNIT LOAD PSI	POWER LOSS HORSEPOWER
10-1/2" diameter steel backed flooded cavity	55	27.500	500	190
9" diameter copper backed flooded cavity	36.7	27.500	750	130
10-1/2" diameter steel backed directed lube	55	27.500	500	125
9" diameter copper backed directed lube	36.7	27.500	750	80

velocity through the drains of 1.8 feet per second. This is satisfactory although if sufficient space is available a velocity of one foot per second would be preferred. Other examples of directed lubrication thrust bearings are shown in Figures 10, 11, and 12.

The obvious questions that arise when a change is made to the design of a critical unit are:

"What is the risk involved?" and "How can we be sure safety and reliability are not sacrificed?"

Fortunately, we have at our disposal these days accurate and reliable instrumentation to monitor thrust bearings sufficiently to assure safe continuous operation and to prevent catastrophic failure in the event of an upset to the system.

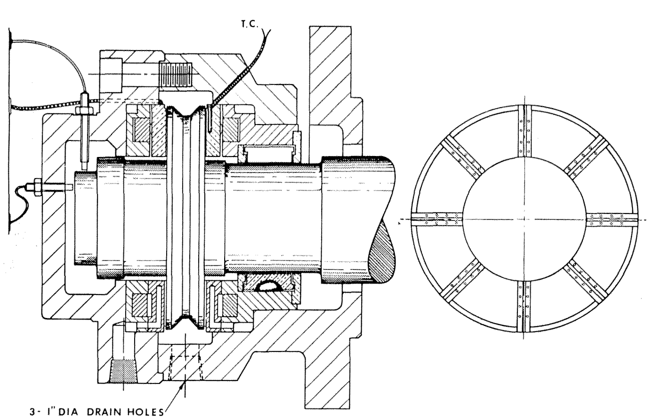


Figure 9. Directed Lube Thrust Bearing for Centrifugal Compressor.

Temperature sensors such as RTD's (Resistance Temperature Detectors), Thermocouples, and Thermistors can be installed directly in the thrust bearing to measure metal temperature. The installation shown in Figure 13 has the RTD puddled into the babbitted surface. It is in the most sensitive zone of the shoe 75% from the leading edge and 50% radially. The position of the sensor is critical in establishing the safe operating limits. As long as the probe is generally in the zone of maximum temperature, it will be highly sensitive to load, although the level of temperature may vary considerably as can be seen on the table in Figure 14. The temperature is also dependent on the pad backing material. At 500 psi load the center sensor at A-II registers 200°F while the sensor at B-I registers 280°F in a steel backed bearing. Again, these temperatures are typical and will vary with size, type, speed and lubrication from bearing to bearing. The difference in a copper backed bearing can be seen to be quite significant, with A-II reading 185°F and B-I reading 205°F. The position of the sensor with respect to the surface is less significant in this bearing than in the steel backed. Again, position in the sensitive zone is important in establishing safe operating limits of temperature.

Axial position proximity probes are another means of monitoring rotor position and the integrity of the thrust bearing. A typical installation is shown in Figure 15. In this case two positions are being monitored, one looking at the thrust runner and one looking at the end of the shaft near the centerline. This method detects thrust collar runout and also rotor movement. Bently [5] gives a description of a proximitor probe for thrust measurement as a "non-touch, eddy current, electronic micrometer" in effect justifies and endorses their use as an accurate, reliable protection tool.

It is the author's opinion that a critical installation should have the combination of metal temperature sensors in the thrust pad along with axial proximity probes. If metal tempera-

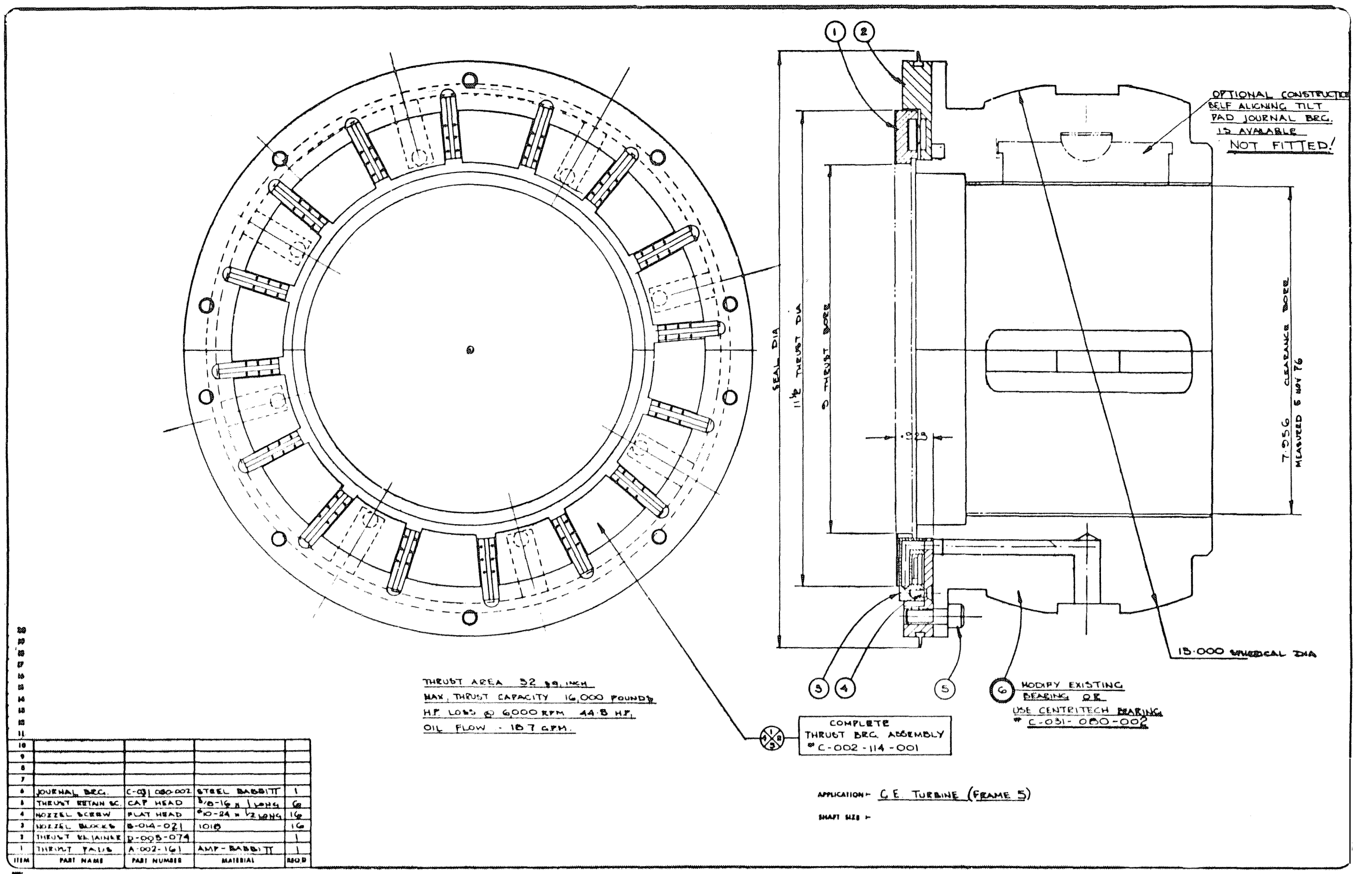


Figure 10. Tilt Pad Thrust Bearing.

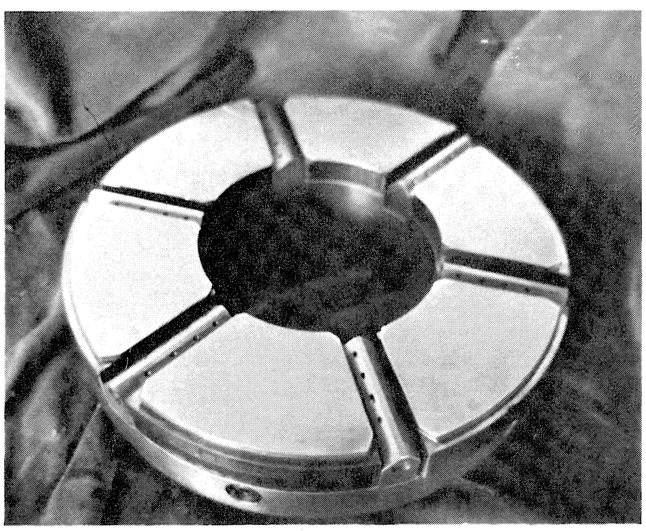


Figure 11. 10½ Inch Self-Equalizing Directed Lubrication Tilt Pad Thrust Bearing.

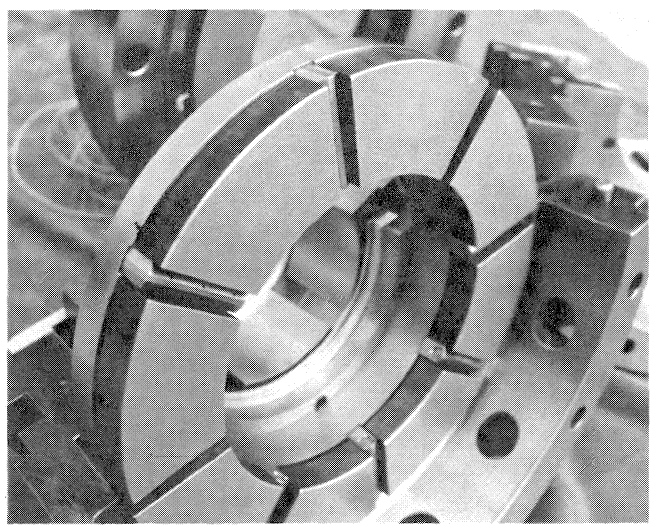


Figure 12. 8-Inch Self-Equalizing Directed Lubrication Tilt Pad Thrust Bearing. (Note large discharge holes to evacuate drain cavity.)

tures are high and the rotor shows significant movement, then thrust bearing failure should be anticipated.

Because of the many variables which often combine to establish the thrust load, it may become desirable and necessary to measure the actual thrust load. This can be done rather conveniently on larger bearings (5 inch minimum) by the use of strain gauge load cells. The use of load cells in conjunction with metal temperature sensors would be a useful tool in establishing guidelines on typical bearing applications correlating metal temperature versus bearing load.

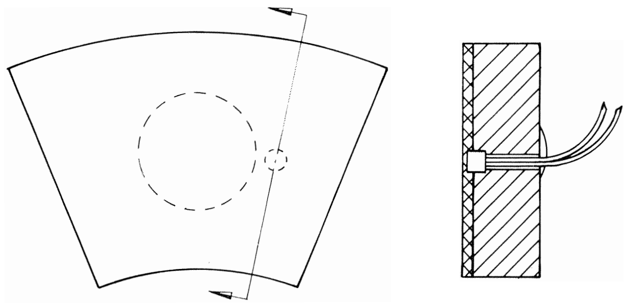
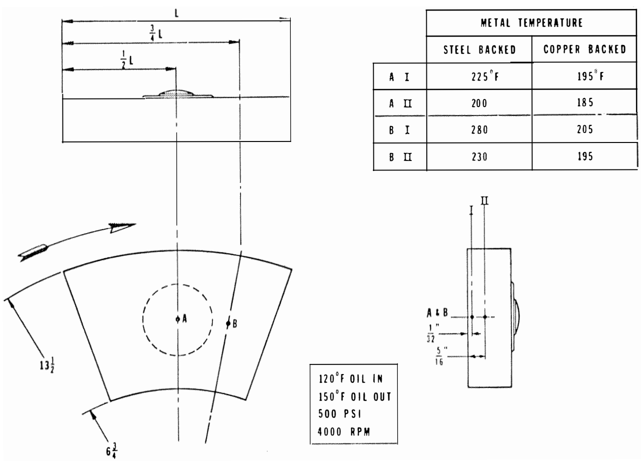


Figure 13. RTD Installation in Thrust Pad.



	METAL TEMPERATURE	
	STEEL BACKED	COPPER BACKED
A I	225°F	195°F
A II	200	185
B I	280	205
B II	230	195

Figure 14. Temperature Distribution in the Pad.

The steps required to increase the efficiency of fluid film thrust bearings have been outlined. Conservation of energy while maintaining machinery reliability can be achieved by simple applications of improved bearing designs and use of readily available updated bearing technology.

Substantial improvements in current thrust bearing performance can be obtained without radical redesign of the turbomachinery in use today. The essential ingredient necessary for realizing these improvements is for responsible individuals to take that bold step of breaking away from tradition and making the change.

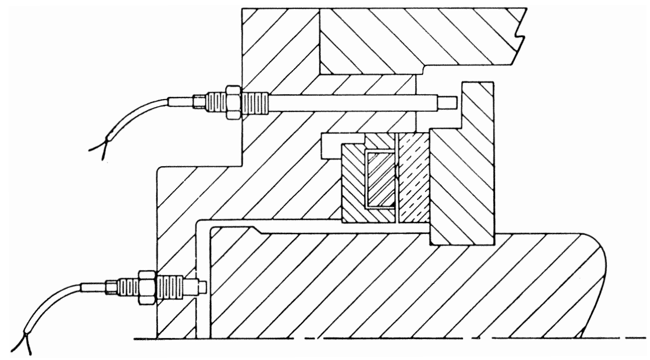


Figure 15. Typical Axial Proximity Probe Installation.

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