HIGH SPEED JOURNAL AND THRUST BEARING DESIGN

by

B. S. Herbage

President and Technical Director
Centritech Corp.

Centritech Corporation was founded by Bernard S. Herbage to apply his many years of experience in O.E.M. bearing design to the growing needs of the high performance bearing replacement market. He was with Allis-Chalmers for ten years, ultimately becoming Senior Engineer Supervisory in the mechanical design section, with responsibilities for the design of high speed journal and thrust bearings. He then joined Waukesha Bearing Corporation, where he spent six years. Initially, he was Chief Engineer and ultimately, General Manager. He is a graduate mechanical engineer and a registered professional engineer. At Centritech, Mr. Herbage is assisted by an able staff of design and manufacturing personnel.

There are many approaches to the design of thrust and journal bearings for high speed turbomachinery, ranging from the simplest flat thrust bearings and plain journal bearings to the very sophisticated multiwedge designs for both thrust and journal bearings. It is the intent of this paper to stress some fundamentals by which bearing designs can be selected which match the performance requirements of various types of turbomachines.

The scope of this paper will be limited to design of fluid film bearings and will therefore not include any discussion on the use of rolling element bearings nor solid film lubricated bearings.

Fluid film bearings should be designed to meet the operating requirements of the machine if at all possible. In some cases, an economic decision dictates varying or compromising the design of the particular machine to accept a less expensive and less sophisticated bearing design. An argument on the original reasoning versus the long term effect of such decisions is beyond the scope of this discussion.

FACTORS AFFECTING BEARING DESIGN

Many factors enter into the final selection of the proper design of bearings. Some of these affect both the journal and thrust bearing design, while the rest affect one or the other.

1. Shaft misalignment through the bearing area.
2. Maximum, normal and minimum loads expected and the duration of each.
3. Maximum and minimum shaft speeds.
4. Critical nature of the turbomachine's operating function.
5. Viscosity of the lubricant to be used.
6. Minimum practical inlet oil temperature.
7. Temperature gradient in or adjacent to the shafting at the bearing.
8. Contamination of the lubrication system, either chemical or mechanical.
9. The amount of axial movement that can be tolerated within each machine.
10. The effect of friction loss in the bearings on the efficiency of each machine.
11. Lateral critical speed analysis.
12. Maximum vibration roughness anticipated during the life of the bearings.
13. Space available.

Volumes of books and hundreds of papers have been published on the various facets of fluid film babbitted bearing design, covering completely or partially all of the factors just listed. This paper will concentrate on those factors which are most often overlooked or simply ignored in the selection of the proper bearing to provide the greatest potential for long life trouble free performance.

MISALIGNMENT

The amount of misalignment that can be tolerated by a bearing is a function of the load being applied and the ability of the bearing to conform to the misalignment.

As a solution to this problem in the design of journal bearings, Stokely and Donaldson (1) have presented a straightforward guideline to "quantitatively assess" the effect of misalignment in journal bearings. As a rough criteria for design, they indicate that a conservative approach is to assume the shaft tilts about a fixed point in a plane at the bearing midlength and design so as not to let the shaft contact the end of the bearing within the minimum film thickness, determined by assuming perfect alignment. Referring to Figure 1, it can be readily appreciated that for the same h(min), a shorter bearing can absorb more misalignment than a longer bearing.

If a greater amount of misalignment is expected than that defined in Figure 1, the designer can either refine his analysis by further application of theory as proposed by Stokely and Donaldson, or he can provide a self aligning feature in his bearing design. Such designs as a spherical support housing or a self aligning tilting pad journal bearing as developed by the author's company may be used.

The effect of misalignment on thrust bearings is in some ways similar to journal bearings, and yet is uniquely different. As seen in Figure 2, thrust collar misalignment loads up one segment of the thrust bearing arc and

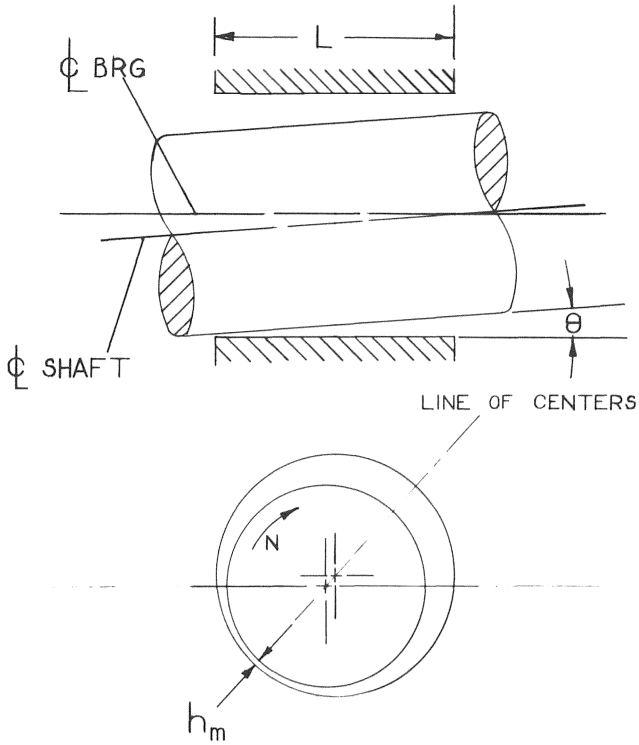

Figure 1. Journal Bearing Misalignment.

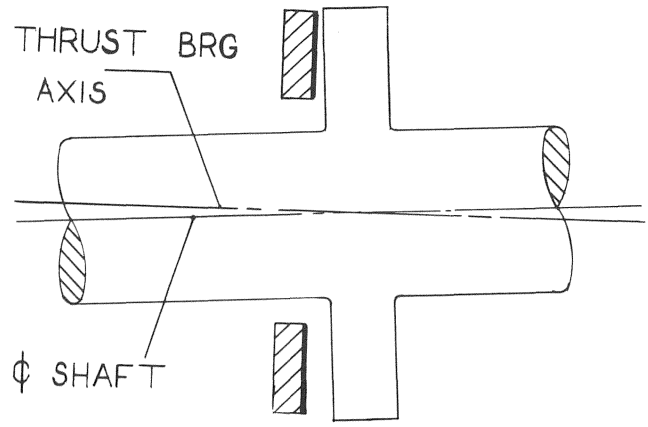

Figure 2. Thrust Bearing Misalignment.

unloads the opposite segment. This effect is more pronounced the higher the thrust load and the less flexible the thrust bearing support structure.

Figure 3 shows four types of thrust bearings commonly used in turbomachinery.

The first of these, Figure 3A, is commonly called a tapered land bearing. When this bearing is designed with the proper taper for the operating speed, for the

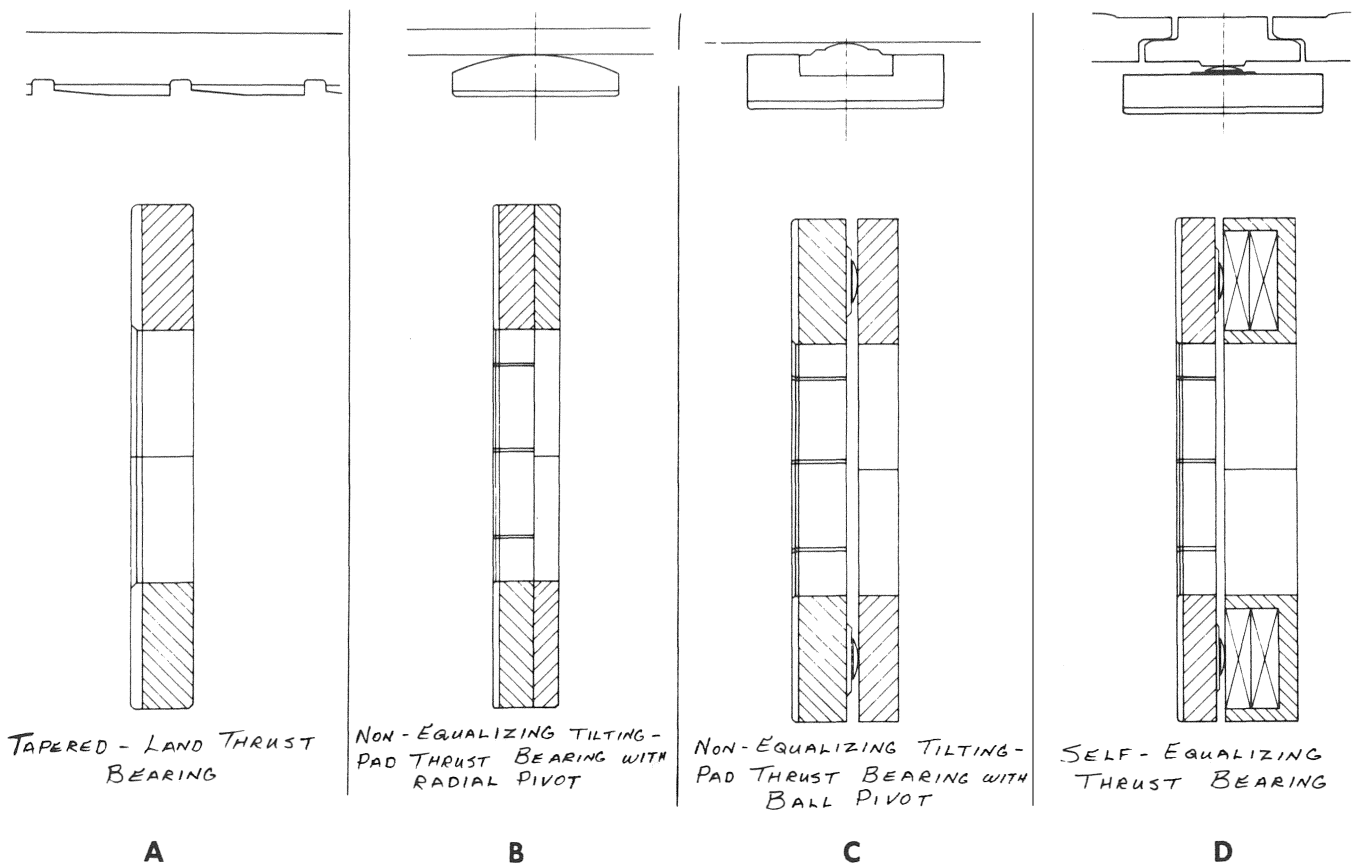

Figure 3. Various Thrust Bearing Designs.

viscosity of oil used and the maximum expected load, it can support a load equal to a tilting pad thrust bearing. It can even match the load carrying capacity of a self-equalizing tilting pad thrust bearing, if perfect alignment is maintained between the thrust collar and the thrust face.

The design shown in Figure 3B is a non-equalizing tilting pad thrust bearing that pivots on the back of the pad along a radial line. In some instances, this style is designed with an offset pivot position to obtain optimum load carrying capacity. This offset makes the bearing uni-rotational.

Straightforward calculations show that a variation in film thickness between diametrically opposite sides of a 6″ to 10″ thrust bearing in the order of magnitude of .004 of an inch will virtually unload the open side and severely overload the opposite side. It is obvious that a bearing that cannot pivot freely with some self-equalizing feature will not be able to accommodate a significant degree of misalignment.

The third design, Figure 3C, is a non-equalizing tilting pad bearing with the pads supported on spherical pivot points. This allows each pad to pivot in any direction, therefore misalignment is not as serious in this design as in the cases of Figures 3A and 3B. A condition known as hot oil carryover, that causes premature failure in thrust bearings, is relieved somewhat in misaligned non-equalizing thrust bearings resulting in more tolerance to misalignment than would normally be expected by calculation.

Figure 3D is an example of the well known kingsbury self-equalizing thrust bearing. Except for the friction at the leveling link surfaces, this bearing virtually eliminates the problem of misalignment in thrust bearings. The major drawback with this design is that standard designs of this type require more axial space than a non-equalizing thrust bearing.

Within the space available, the optimum thrust bearing should be installed. It may be necessary to make rather drastic variations from conventional stereotyped designs to fit the space, but this has been done quite often with great success. The author's company recently supplied a 7″ self-equalizing thrust bearing to fit in the same space as a non-equalizing design in an axial length of 1.250 inches.

MAXIMUM AND MINIMUM SHAFT SPEEDS

Bearing design begins to become a problem as turbomachinery surface speeds exceed 4000 feet per minute. Below this speed, many machines still use oil ring lubricated bearings. It is not recommended that oil ring lubricated bearings be used beyond 5000 feet per minute, even with special oil ring design and the addition of oil scrapers.

Most turbomachinery bearings are designed for pressure lubrication. Up to a surface speed of 12000 feet per minute, the performance of pressure lubricated journal bearings can be calculated quite accurately. Beyond this speed, the flow of oil in the clearance space tends to deviate from laminar and turbulence begins to affect bearing performance. Turbulence and hot oil carryover can cause severe problems at speeds beyond 18000 feet per minute. In journal bearings, special contouring of the bearing surfaces are employed to minimize the losses generated and maintain safe operating temperatures. In thrust bearings, the thrust collar acts as a viscosity pump and generates excessive heat at the periphery. This loss can be greatly reduced by the installation of a control ring with tangential discharge orifices.

In journal bearings, the transition speed (N_T) between laminar and turbulent behavior is

$$N_T = 1.57 \times 10^3 \frac{v}{D^{1/2} \; C^{3/2}} \text{ RPM}$$

where

v = Kinematic Viscosity
D = Diameter
C = Diametrical Clearance

In thrust bearings, this transition is approximately at a Reynolds number of 1000.

At the higher speeds, it is not practical to limit the oil discharge temperature to 160°F and therefore a chart as shown in Figure 4 can be used as a more reasonable limit.

FACTORS AFFECTING JOURNAL BEARING DESIGN

The simple selection of bearing designs based on speed alone can lead to problems on high speed rotating equipment. The following are some of the factors often ignored, or not considered for economic reasons.

1. The potential for high vibration caused by the phenomenon known as oil whirl.
2. The potential for high fatigue loads due to rotor unbalance.
3. Misalignment of the shaft in the bearing, leading to wiping and/or fatigue failure. (Discussed previously).
4. The influence of bearing stiffness on the critical speeds of the shaft.

Because of the limited time allotted for this presentation, and the volume of material which has already been written on a variety of plain journal bearings, I would like to present a case for the virtual exclusive use of tilting pad journal bearings in high speed turbomachinery.

Normally the tilting pad journal bearing is considered when shaft loads are light, because of its inherent ability to resist oil whirl vibration. However, this bearing, when properly designed, has a very high load carrying capacity. It has the ability to tilt to accommodate the forces being developed in the hydrodynamic oil film, and therefore operates with an optimum oil film thickness for the given load and speed. This ability to operate over a large range of load is especially useful in high speed gear reducers, with various combinations of input and output shafts.

A second important advantage of the tilting pad journal bearing is its ability to accommodate misalignment of the shaft. Because of its relatively short length to diameter ratio, it can accommodate minor misalign-

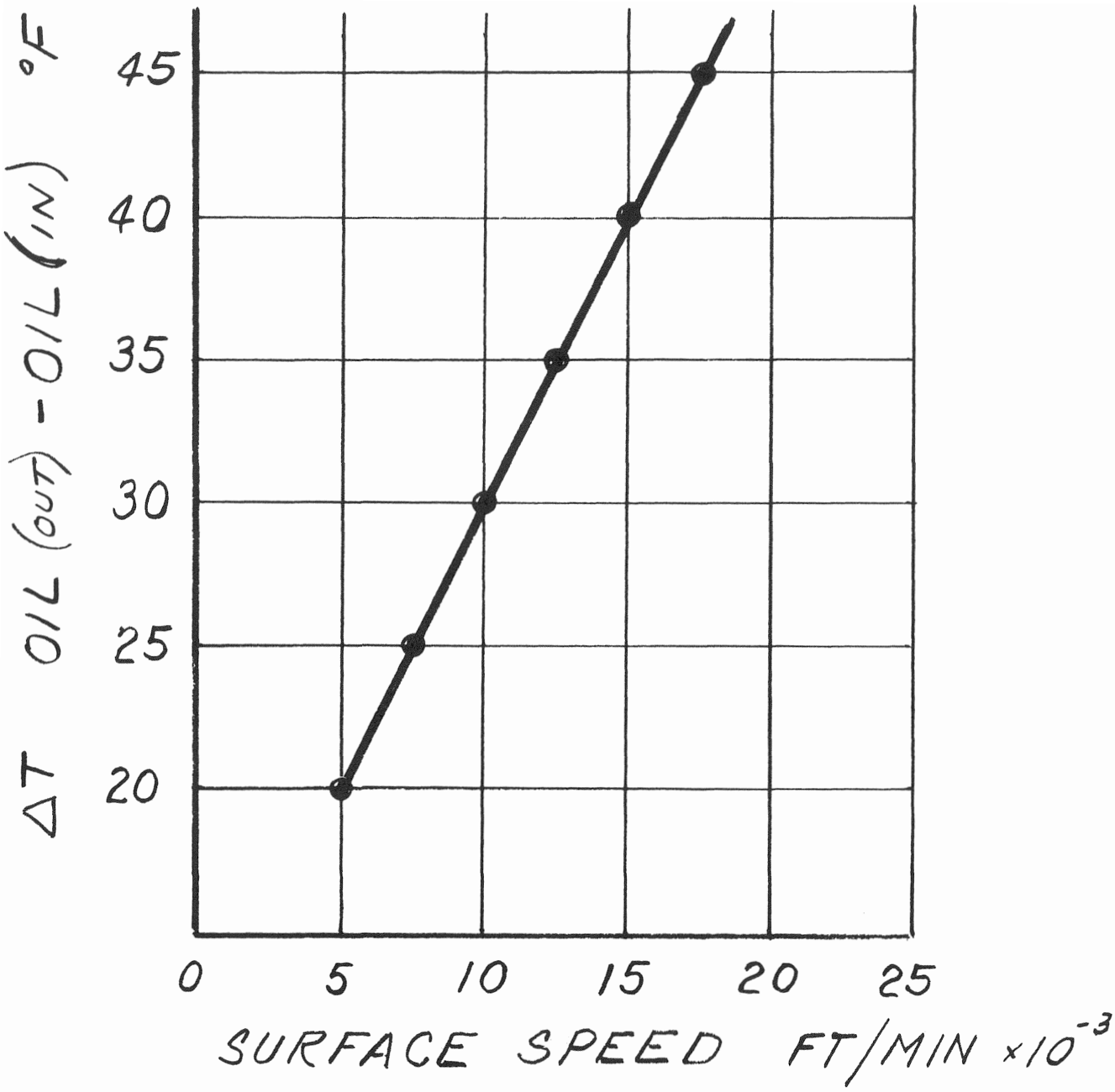

Figure 4. Discharge Temperature Criteria.

ment quite easily. Where moderate to severe misalignment is expected, or is a distinct possibility, special designs can be produced in which each pad can pivot in any direction to completely accommodate major misalignment. CENTRITECH ball and socket designs of this type have been in operation for over eight months. Interim inspections have shown that these bearings are operating successfully and have solved a chronic vibration and misalignment problem.

A third category to investigate is the one of fatigue failure. Whenever vibration occurs in a high speed machine, the potential for fatigue failure of some part of that machine is present. Since bearings are an integral part of each machine, they are also susceptible to fatigue failure.

All rotating machines vibrate when in operation. Whether the bearings fail due to fatigue loading depends on the level of vibration, and the ability of the bearings to resist cyclic stresses developed. It remains the responsibility of the end user to determine the level of vibration his unit can tolerate. A useful chart for judging this level is a vibration severity chart, shown in Figure 5. The readers are advised to review such a chart published in Safety Maintenance, March, 1966.

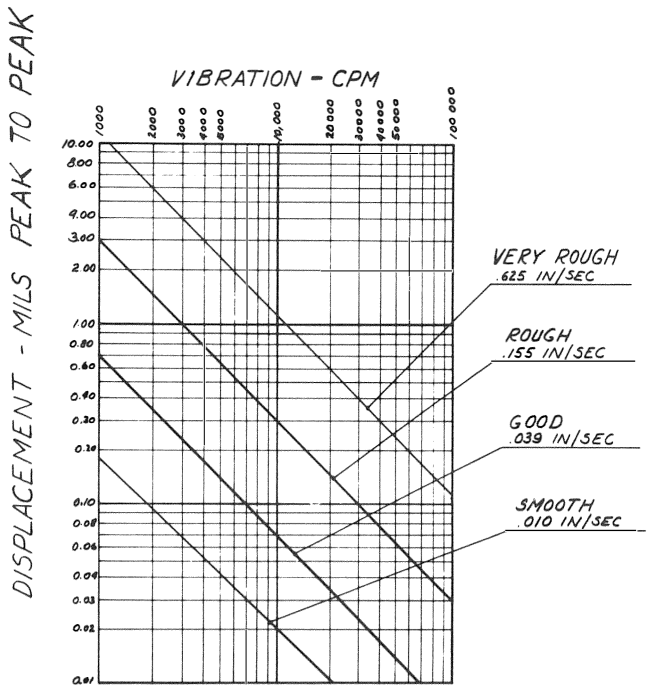

Figure 5. Vibration Severity Chart.

While vibration displacement is a useful criteria in judging a machine, a more useful tool is the measure of the maximum velocity attained during a peak to peak oscillation.

There are some general rules of thumb which can be used to judge the performance of a machine. These readings are of the bearing cap and a more precise definition is needed to correlate these readings to shaft movement for each particular component. These are shown in the following table:

GENERAL MACHINERY VIBRATION SEVERITY TABLE

		Amplitude Mils	Velocity In/Sec
2,000 RPM	Smooth	.093	.010
	Rough	1.60	.157
	Very Rough	6.00	.628
4,000 RPM	Smooth	.05	.010
	Rough	.78	.157
	Very Rough	3.0	.628
10,000 RPM	Smooth	.02	.010
	Rough	.30	.157
	Very Rough	1.20	.628
20,000 RPM	Smooth	.01	.010
	Rough	.15	.157
	Very Rough	.6	.628

If a high speed rotating machine operates for any length of time at the "very rough" vibration level, it can be assumed that the bearings will experience severe pounding and ultimate fatigue failure. Whether this level of vibration is expected by the idiosyncrasies of the original design, or is actually experienced in the field during operation is not the issue. For successful operation, a decision must be made to install a bearing with the maximum fatigue life characteristics.

It has long been known that a bearing with thick babbitt does not have a good fatigue life expectancy. Figure 6 is a much publicized chart of the fatigue life of a babbitted bearing, as it varies with the babbitt thickness. Bearings with a babbitt thickness greater than .010 of an inch cannot resist the pounding produced at the "very rough" vibration levels. In trying to increase the fatigue life of a bearing operating in the "very rough" regime, the tilting pad journal bearing offers the greatest amount of flexibility in the necessary design variables.

1. A self-aligning ball and socket design can be selected to obtain optimum alignment and *minimum unit load.*

2. A high thermal conductivity backing material can be readily used to dissipate the heat developed in the oil film, and therefore maintain *maximum strength* of the babbitt.

3. A thin babbitt layer can be centrifugally cast to the bearing backing material to obtain the uniform thickness of no greater than .005 of an inch. By use of adjustable shims in the pad supports, minimum bearing clearances can be maintained without machining great variations in babbitt thickness.

4. By varying the contour of the bearing pad, a certain degree of preload can be designed into the bearing, which, if applied properly, can reduce the amplitude of shaft vibration.

Many high speed rotating machines operating in industry today were designed many years ago. At that time, and even today, much is being learned concerning the exact effect of support stiffness on the various critical speeds of shafts. One thing has definitely been determined; the actual critical speeds of shafts are influenced by the bearing's oil film thickness. The bearing's stiffness varies with the oil film thickness, so that the critical

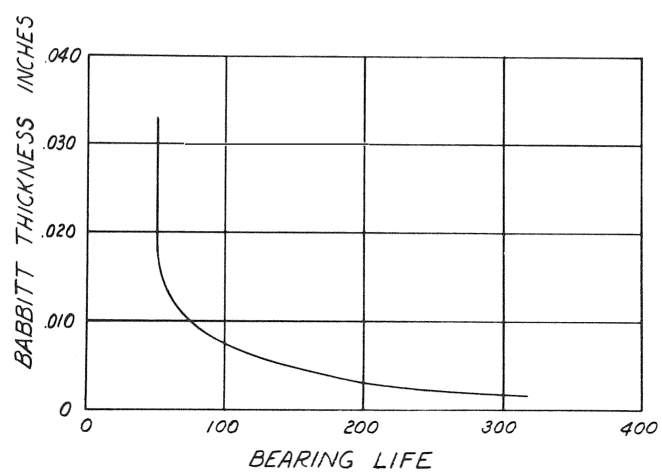

Figure 6. Babbitt Fatigue Characteristics.

speed is influenced to a certain degree directly by oil film thickness. Again in the area of critical speeds, the tilting pad journal bearing has the greatest degree of design flexibility. There are now sophisticated computer programs in use and being expanded, to show the influence of various load and design factors on the stiffness of tilting pad journal bearings. Figure 7 shows a tilt-pad bearing designed to increase horizontal stiffness. By using an alternate internal ring the number of pads can be increased to five.

The following variations are possible.

1. The number of pads can be varied, from three to any practical number.
2. The load can be directed either directly on a pad, or to occur between pads.
3. The unit loading on the pad can be varied by either varying the arc length, or the axial length of the bearing pad.
4. A parasitic preload can be designed into the bearing by varying the circular curvature of the pad with respect to the curvature of the shaft.
5. An optimum support point can be selected to obtain a maximum oil film thickness.

FACTORS AFFECTING THRUST BEARINGS DESIGN

The principle function of a thrust bearing is to resist the unbalanced thrust developed within the working elements of a turbomachine and to maintain the rotor position within tolerable limits.

After an accurate analysis has been made of the thrust load, the thrust bearing should be sized to support this load in the most efficient method possible. Many tests have proven that thrust bearings are limited in load capacity by the strength of the babbitt bearing surface in the high load and temperature zone of the bearing. In normal steel backed babbitted tilt pad thrust bearings,

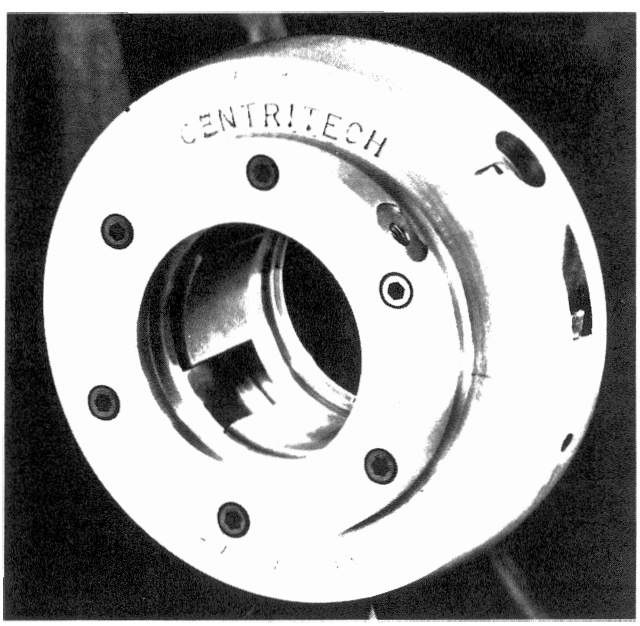

Figure 7. Centritech Multipad Journal Bearing.

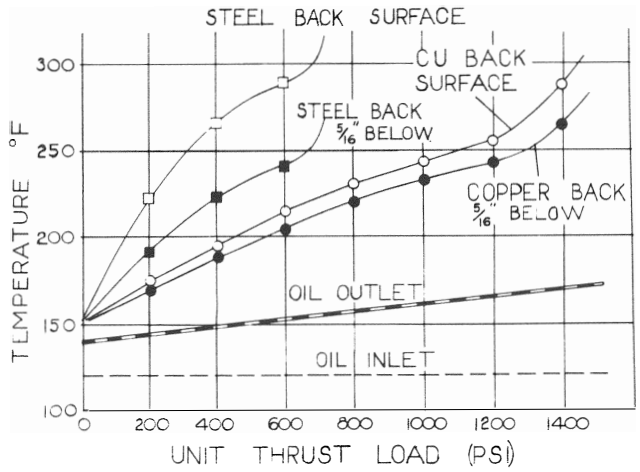

Figure 8. Thrust Bearing Temperature Characteristics.

this capacity is limited to between 250 to 500 psi average pressure. It is the accumulation of temperature at the surface and pad crowning that causes this limit.

The thrust carrying capacity can be greatly improved by maintaining pad flatness and by removing heat from the loaded zone. By the use of high thermal conductivity backing materials with proper thickness and properly supported, the maximum continuous thrust limit can be increased to 1000 psi or more. This new limit can be used to either increase the factor of safety and improve the surge capacity of a given size bearing or to reduce the size of the thrust bearing and consequently the losses generated for a given load.

Since the higher thermal conductivity material (copper or bronze) is a much better bearing material than the conventional steel backing, it is possible to reduce the babbitt thickness to the order of .010 to .030". This will limit the axial movement in the event of a babbitt wipe out and yet allow sufficient movement to record a significant axial movement on an axial position indicator.

Along with an axial position indicator, the use of embedded thermocouples and RTD's will signal distress in the bearing if properly positioned and religiously recorded.

In a change from steel backing to copper backing, a different set of temperature limiting criteria should be used. Figure 8 shows a typical set of curves for the two backing materials. This chart also shows that drain oil temperature is a poor indicator of bearing operating conditions in that there is very little change in drain oil temperature from low load to failure load.

SUMMARY

Poor bearing selection can result in severe limitations on turbomachinery performance. On the other hand, bearings cannot, by themselves, correct inherent shortcomings of a machine's design. By the selection of the optimum bearing to suit the performance requirements of the machine, and by the use of established monitoring systems with accurate reference guidelines, the performance of a machine can be carefully controlled.