FLUID FILM BEARINGS
FUNDAMENTALS AND DESIGN CRITERIA AND PITFALLS

by

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In 1956, he founded his company, Abramovitz Associates, Inc., which specializes in consulting, and the design and manufacture of special fluid film bearings. There was a 3 year interlude where he was director of a jointly owned division of Industrial Tectonics, Inc., and oversaw a staff that was responsible for development, manufacture and sales of bearings and special machine products. Earlier, in 1950, he founded and directed the Friction and Lubrication Section of The Franklin Institute Laboratories.

He received a B.S. degree in M.E. from Drexel University in 1948, and an M.S. degree in M.E. from the University of Pennsylvania in 1953.

Mr. Abramovitz has lectured extensively, holds a number of patents, and has published over 20 papers on fluid film bearings. He is an elected member of several honorary societies, and is a member of a number of professional societies where he held chairman positions responsible for divisions and national conferences.

ABSTRACT

The presentation will cover fluid film bearing fundamental theory and design criteria. Since the subject is extensive, the design portion will be restricted to the hydrodynamic (self-acting) bearing mode using liquid lubricants. This is the condition found in the majority of turbomachinery in the field, and is of most general interest.

The four basic fluid film bearing modes and principles of bearing operation will be described. The various types of thrust and journal bearings will be identified and discussed as to their operation and use.

The design criteria will cover such subjects as load capacity, power loss, temperature, speed, turbulence, misalignment, and vibrations. This will include the most recent available information for obtaining maximum bearing performance. Many references will be included for more detailed study.

The concluding portion will deal with the pitfalls and problems in applying the bearing design principles and types, with actual field examples. This will cover operating conditions, bearing type selection, and physical design and installation, what must be done or avoided to translate a promising bearing application into a practical and successful reality.

INTRODUCTION

This paper, presented as a tutorium, is intended to give an overall view of fluid film bearing design. Even as a survey, this is a relatively short presentation for this extensive field. It was therefore necessary to restrict the scope to conditions that would normally be found in the majority of turbomachines. It is hoped that the selected material will not only be informative, but also useful to the machinery user for evaluating fluid film bearings in his equipment. Also, coupled with the 47 references cited, this paper should provide a basis for the design and application of this class of bearing.

The various modes and principles of operation will be described. An appreciation of the principles permit the designer to do more than just blindly follow a set of rules. Since the selection of the type of bearing can be equally as important as the theoretical design, the many types of thrust and journal bearings will be identified and discussed as to their operation and use. The design information will in some cases be immediately useful as a design guide, whereas, in other instances, a trend will be shown, pointing to optimum performance. Everything presented will be referenced for more detailed study. The presentation will conclude with a discussion of the practical design pitfalls and problems. It is the writer's experience that an excellent analysis and the use of optimum design data can become worthless when certain operating conditions are not taken into account, the wrong type of bearing is selected, and important physical design and installation details are neglected. This is a very critical area of design that can mean the difference between success and failure.

As noted, this paper will be restricted in content. It will only cover liquid lubricants, and the hydrodynamic mode of operation. However, all four modes will be covered in the section on principles. Although much of the design information deals with oil as the lubricant, a section is devoted to water lubrication as a class of low viscosity corrosive liquids. This class of lubricants should not be considered as esoteric, since, for many machines, the use of the process fluid as the lubricant has many advantages and has been proven to be successful and reliable.

There are many excellent references for subjects outside the scope of this paper. Compressible lubricants such as air and other gases are covered in [1, 2, 3, 4, and 5]. Reference [6] is a very good hydrostatic lubrication design manual. Text books that cover theory and design for the complete field are [7, 8, 9, 10, and 11]. A wonderful text book is [12]. It has an extraordinary number of references, and each topic not only shows the results of the theory and research, but how our present understanding has been built up from one discovery to the next. References [13] and [14] give a broad view of fluid film bearing applications, with many examples.

It seems appropriate to show where fluid film bearings fit into the bearing field.

BEARINGS

ROLLING ELEMENT — SLIDING SURFACE

- ROLLING ELEMENT
  - BALL
  - ROLLER
- SLIDING SURFACE
  - BOUNDARY
  - MIXED FILM
  - FLUID FILM

The three classes of sliding surface bearings are really bearings operating with different amounts of metal to metal separation. Boundary lubrication is a very specific field, involving friction and wear, and the chemistry and physics of materials and lubricants. The mixed film class is esentially a marginal fluid film. Full fluid film lubrications is the subject of this presentation.

PRINCIPLES

Fluid film lubrication exists when there is a full film of fluid that completely separates the surfaces of the two members that comprise the bearing. This separation can only be achieved if the fluid in the clearance space is pressurized to the extent that the fluid pressure forces balance the bearing load and maintain equilibrium. This means that the fluid must be continuously introduced into and pressurized in the film space.

Bearing Modes

Figure 1 schematically shows the four modes of fluid film lubrication. The hydrodynamic mode, (a), is where the pressure is self-induced by the relative motion between the two bearing member surfaces, and the film is a wedge shape configuration. It is also called a "self-acting" bearing. The hydrostatic mode, (b), achieves its separation and load capacity by the introduction of externally pressurized lubricant flow. The squeeze film bearing, (c), derives its load capacity and separation from the fact that a viscous fluid cannot be instantaneously squeezed out from between two surfaces that are approaching each other. Reciprocating machinery with piston-pin-type bearings are a good example. The hybrid bearing, (d), can combine any of the first three modes, although a combination of (a) and (b) is the most common.

Hydrodynamic bearing operation depends on the velocity of the bearing member as well as the existence of a wedge

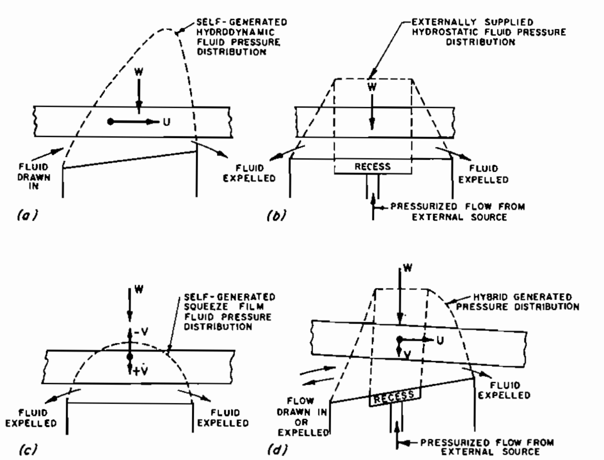

Figure 1. Modes of Fluid-film Lubrication; (a) Hydrodynamic, (b) Hydrostatic, (c) Squeeze Film, (d) Hybrid (from reference 14).

shape configuration. This means that the bearing surface must be physically made to form a film wedge. The journal bearing forms a natural wedge that is inherent in its design. This is shown in Figure 2 (b), where the displacement of the journal due to load produces a wedge shape supporting film. This figure also displays, in the two views, the three dimensional pressure distribution that can be applied to all modes.

The Fluid Film

With so much emphasis on the fluid film and the amount of separation as the basic operational criteria, an understanding of this microscopic area is essential. In fluid film bearings we are dealing with film thicknesses that range from 0.0001 to 0.0100 inches, depending on the mode, lubricant, and application. Practically, films usually range from a minimum of 0.0003" using gases, to 0.0080" for hydrostatic oil lubricated bearings. The exception is with oil squeeze films, where the capacity to take extremely high reversing loads with no bearing harm indicates that the minimum instantaneous films could be 0.0001", or below.

Figure 3 shows enlarged views of metallic bearing surfaces. Regardless of surface finish, peaks and valleys make up the entire surface. In general, the average asperity height may be from 5 to 10 times the RMS surface finish reading. Figure 3 (A) shows relative separation for full fluid film, mixed film, and boundary. Figure 3 (B) and (C) are sections showing the various types of "contamination." When a metal surface is abraded, an oxide film will form almost immediately. In addition, boundary and extreme pressure oil additives are "contaminants" that form beneficial surface films. These films are usually only several molecular layers thick, and are therefore quite small compared to the size of the peaks and valleys. Some nonmetallic materials, such as Graphite and Teflon, inherently possess beneficial boundary lubricated surfaces. Reference [15] is one of many that can be used for further study.

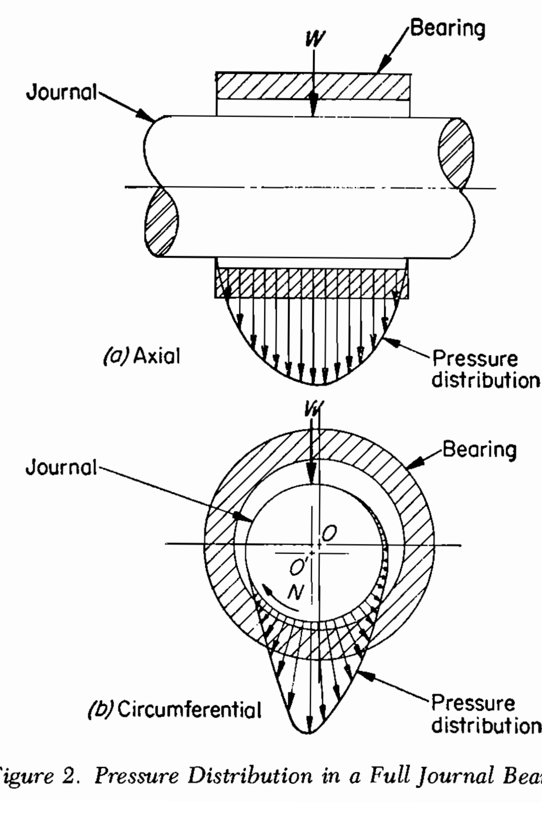

Figure 2. Pressure Distribution in a Full Journal Bearing.

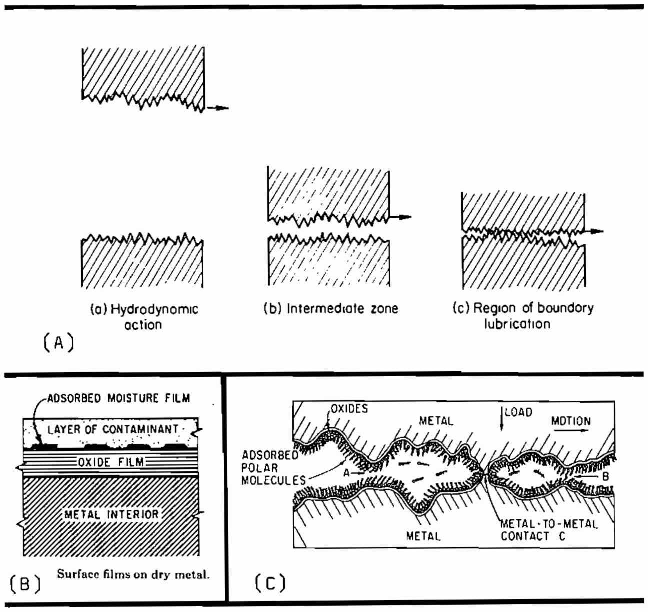

Figure 3. Enlarged Views of Bearing Surfaces.

A classic way of describing film separation, or the "state of health" of a bearing, is by plotting a curve of ZN/P versus coefficient of friction. It takes the form of Figure 4, where Z is the lubricant viscosity in centipoise, N is the journal speed in rpm, and P is the projected area unit loading in psi. It is more easily analyzed if only one variable, for example N, is varied, with the other two variables held constant. Note that when a full fluid film is achieved, the friction is at the lowest point, after which it increases due to the increasing lubricant shear forces.

If a full fluid film exists, the bearing life is almost infinite. The "almost" is tied in to such factors as lubricant breakdown, shock loading, erosion of bearing surfaces, fretting of bearing components, and other practical considerations.

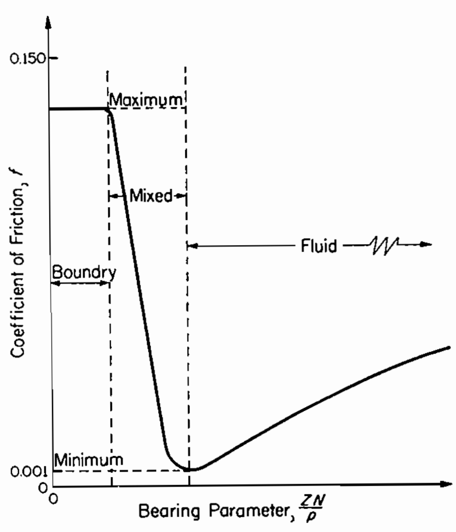

Figure 4. Classic ZN/P Curve.

BEARING TYPES

Hydrodynamic bearing geometries must be receptive to the formation of a film pressure wedge. So many types of bearings have been developed over the years to meet special needs, that it would be virtually impossible to catalogue every variation which designers have evolved. Therefore, the types in most common use will be identified and discussed, along with some special designs that have now become popular.

Thrust Bearings

1) Moveable Type

The most reliable and best overall performing type of thrust bearing is the tilting-pad (shoe) bearing, Figure 5. The pad rests on a pivot and is free to incline to any angle that will give the optimum wedge for the speed, load, and lubricant viscosity conditions. Any number of pad segments can be used, mounted with their pivots on a flat base. Where the developed fluid film is unusually thin, and misalignment can be expected, an equalizer system, Figure 5 (B), is used. The series of leveling plates enables each pad to carry an equal share of the total load, and some runner misalignment can be tolerated. All pivot points, including the pad pivot, are normally hardened to increase sensitivity and minimize fretting. An intermediate design, Figure 5 (C), is where the pads and pivots are mounted on a plate that forms a spherical seat with the machine housing. This is usually effective for initial alignment only, since, under load, the friction at the seat may be so high that the bearing will stay in a fixed position during machine operation. In addition, this arrangement does not equalize the height of the surfaces of the pads. Equalizers normally take more axial space; however, with high strength materials and innovative design, the writer and others have been able to fit an equalized bearing in places where a non-equalized or fixed type had been used.

2) Fixed Type

The two basic types are the tapered land and the step bearing, Figure 6 (A) and (B). Normally, the taper covers 80% of the segment, with the balance flat for the runner to rest on at stop. A compound taper can be added to improve film flow. A shrouded step reduces side flow, which increases load capacity.

The taper acts like a fixed tilting pad, and the step acts as a dam to restrict flow and develop pressure, which is maximum at the step. The taper inclination or step depth is usually the same magnitude as the film thickness, and is determined by one specific set of operating conditions. Machining, coining, and etching have all been used as methods of manufacture. Although these designs are in wide use, the manufacturing tolerances on the surface profile, distortions, and the sensitivity to a deviation from the design operating conditions make it necessary to use them with care.

The parallel surface grooved bearing, Figure 6 (C), does carry load, although it has no wedge surface profile. This will be discussed later. It is a low cost bearing, and is generally used as a positioning device for relatively light loads.

Journal Bearings

The types that will be discussed are those where a positive supply of lubricant is fed to the bearing at all times. Ring and wick-oiled bearings will not be considered here. A number of types are shown in Figure 7.

The circumferential-grooved bearing, (a), normally has the oil groove at half the bearing length. This does reduce load capacity by dividing the bearing into two parts, but it is an aid to better cooling. It is normally used in main and connecting rod bearings.

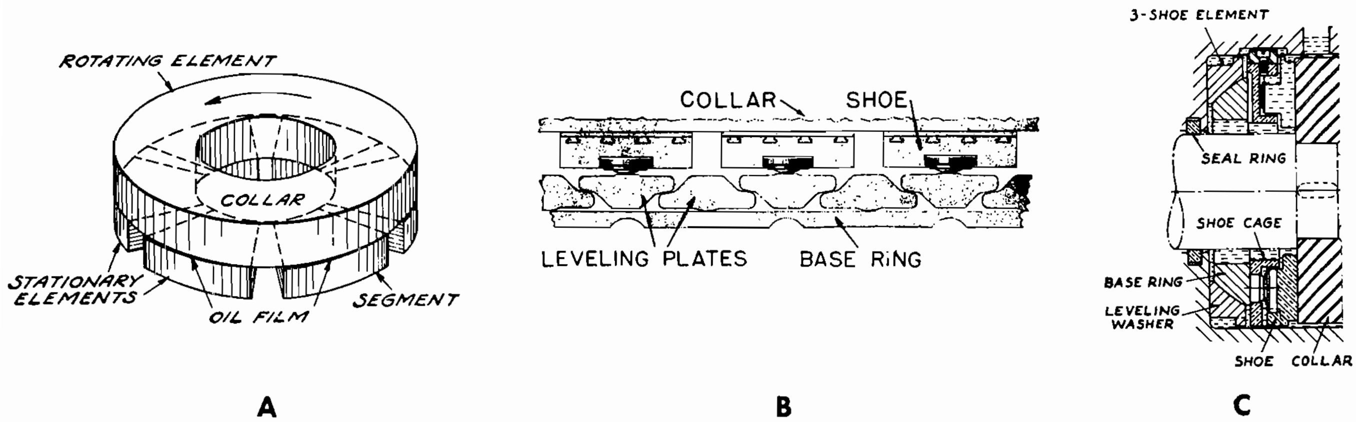

Figure 5. (a) Schematic Thrust Bearing, (b) Equalizer System, (c) Spherical Seat Arrangement. (Courtesy of Kingsbury Machine Works, Inc.)

The cylindrical bearing, (b), is one of the most commonly used turbine bearings. It has a split construction with two axial oil feed grooves at the split. With unidirectional loads, it is analyzed as a 150° arc partial bearing (k). The cylindrical overshot bearing, (c), has a wide circumferential groove in the top unloaded half to provide cooling and reduce power loss at high speed.

The remaining bearing types in Figure 7 are used where machine conditions may require bearing stability. The tilting-pad and pressure dam, (i) and (d), are in common use for this purpose. The tilting-pad type has the advantage that it can be self-aligning if the pads use spherical pivots.

Special Types

The spiral groove bearing, Figure 8 (a), pumps and develops fluid pressure in the spiral steps. The foil bearing, Figure 8 (b), is really a fitted bearing where the thin flexible foil is

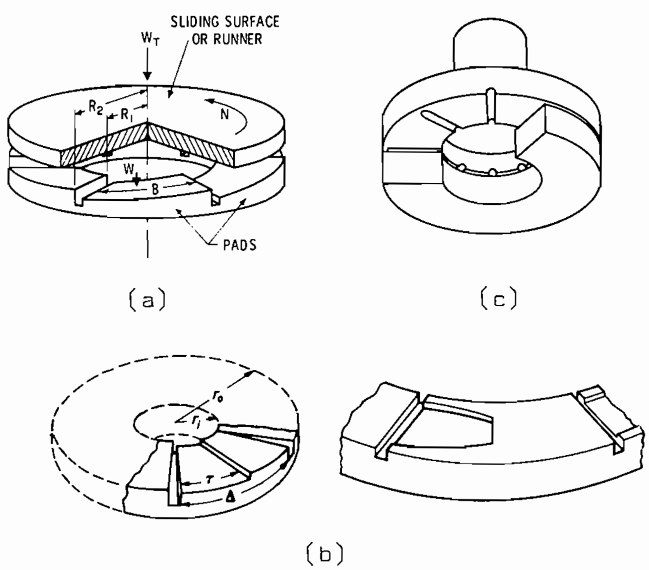

Figure 6. Fixed Type Thrust Bearings; (a) Tapered Land, (b) Step, (c) Parallel Face.

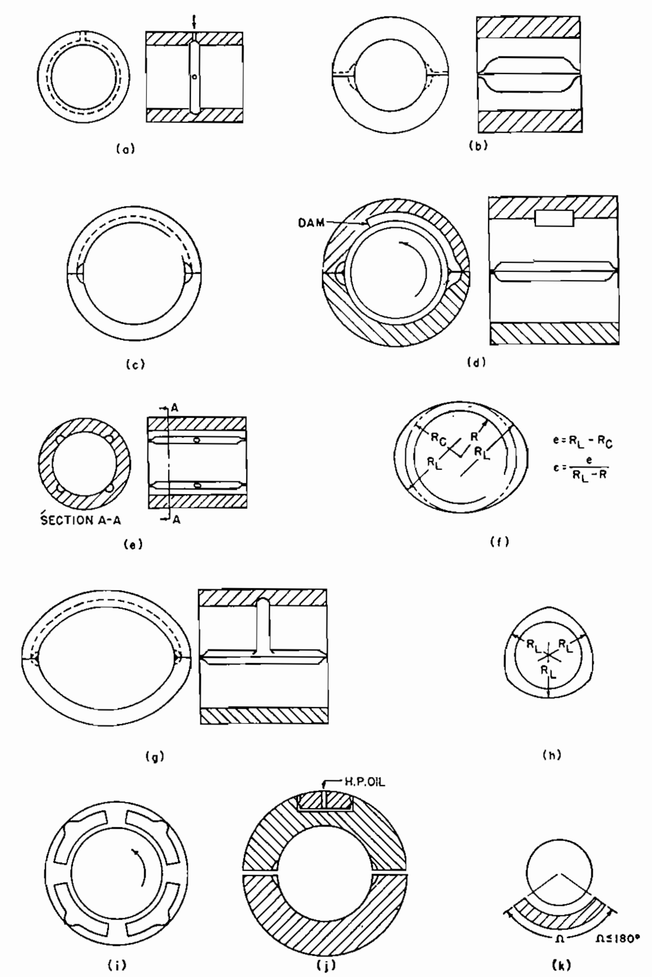

Figure 7. Sketches Showing Shapes of Several Types of Pressure-fed Bearings; (a) Circumferential-Groove, (b) Cylindrical, (c) Cylindrical Overshot, (d) Pressure, (e) Multiple-Groove, (f) Elliptical, (g) Elliptical Overshot, (h) Three-Lobe, (i) Pivoted-Shoe, (j) Nutcracker, (k) Partial (from reference 8 — Courtesy McGraw Hill Book Co.).

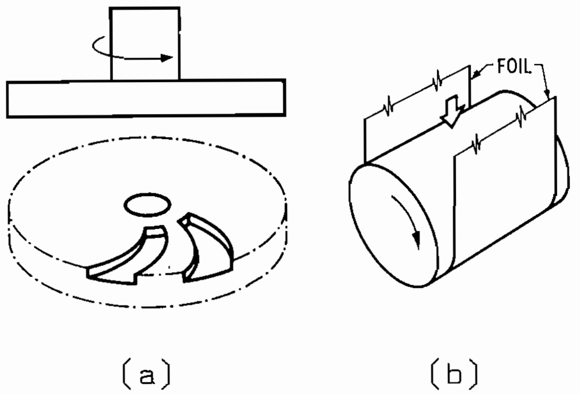

Figure 8. (a) Spiral Grooved Thrust Bearing, (b) Foil Bearing.

able to distort and conform to the required film shape. It is usually used in machines for supporting or guiding tapes.

The compliant, or non-rigid, surface bearing is a significant new aspect of fluid film bearings [16]. It is shown in Figure 9 as a thrust bearing. It does accept misalignment and has a high tolerance to dirt. In addition, a kind of pressure puddle is formed and retards the escape of pressurized lubricant from the bearing, thereby increasing load capacity.

Figure 10 is a composite photograph of a few of the bearings that have been discussed. Shown are a parallel face thrust bearing, a tilting-pad journal bearing, and a combination partial journal and tilting-pad thrust bearing. A photograph of a tilting-pad equalized thrust bearing is in Figure 10A.

BEARING DESIGN

Today, there is a wealth of published theoretical and experimental information on hydrodynamic fluid film bearings. Some of the theoretical works are valuable fundamental contributions such as inertia effects, etc., but others are directed

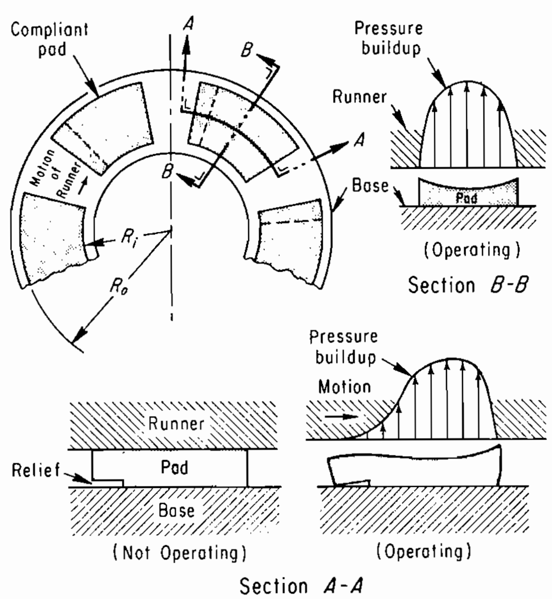

Figure 9. Compliant-Surface Thrust Bearing (from reference 30).

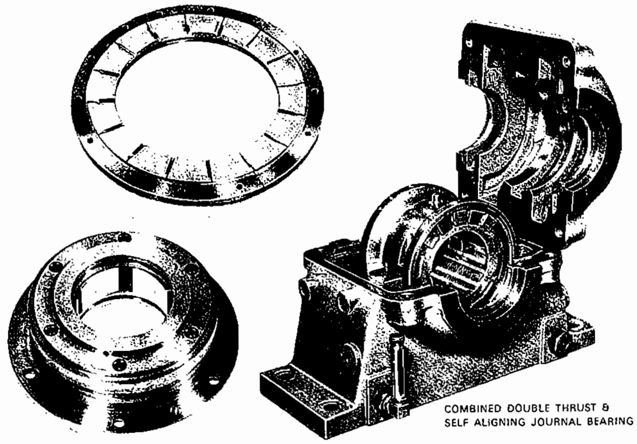

Figure 10. Bearing Hardware. (Courtesy of Waukesha Bearings Corp. and Michell Bearings Ltd.)

to the designer and are quite usable as a design tool. The experimental data are specific for particular bearings and operating conditions, and care must be exercised in extrapolating to other conditions. Aside from the references mentioned in the introduction, there are publications such as [17 and 18] that discuss design limits. Here one must keep in mind that they represent the author's own approach and experience.

Many organizations have excellent computer programs for bearing analysis that are invaluable for handling complex problems. For a large number of applications, a design can be satisfactorily analyzed by hand, providing all variables are considered and proven equations and coefficients are used. Computer generated design curves and coefficients are available for a wide range of conditions in the principal categories of bearing design. They are normally presented in a clear and easily understood form.

Load Capacity

The majority of bearing designs aim for a minimum oil film thickness in order to determine the maximum load that a bearing will support. The value of this film thickness is dependent on such factors as bearing size and type, expected misalignment, possible load and speed variations, and the designer's experience with the particular machine. In addition, how and where the machine is used will dictate how conservative the design should be. It is therefore not surprising that, for a specific application, this prescribed minimum film value will vary, even among experienced designers.

In high speed applications, the large shear forces in the fluid film may sometimes produce film temperatures high enough to cause a creep flow of the bearing surface material. Since there could be a satisfactory fluid film, temperature would then be another load capacity criteria for a special set of extreme conditions.

To intelligently use even the most routine design curves, one should first be aware of the variables involved and the relationship between them. Referring to Figure 11, the equation for a geometric wedge is

$$h_2 = \sqrt{\frac{\mu U B^2 L}{W}} \times K = \sqrt{\frac{\mu U B}{P}} \times K \qquad (1)$$

where: h_2 is the minimum film thickness; μ is the viscosity of the lubricant in the film; U, the velocity of the moving element;

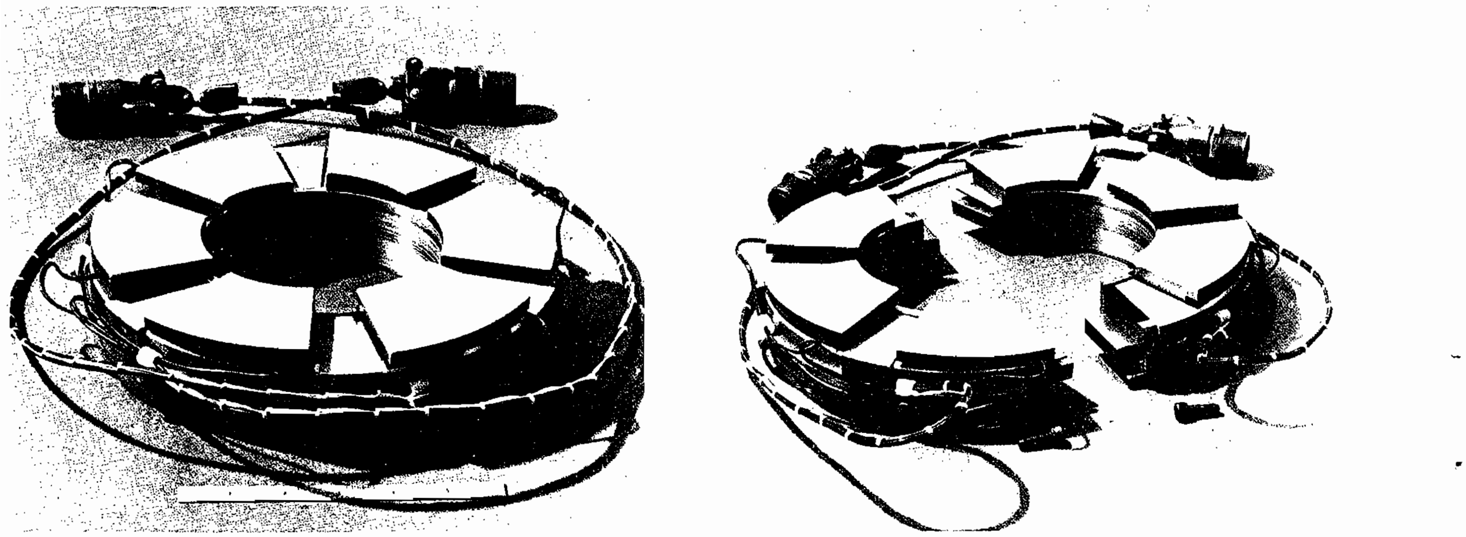

Figure 10A. Oil Lubricated Tilting Pad Equalized Thrust Bearing for Steam Turbine Operating at 10,000 rpm.

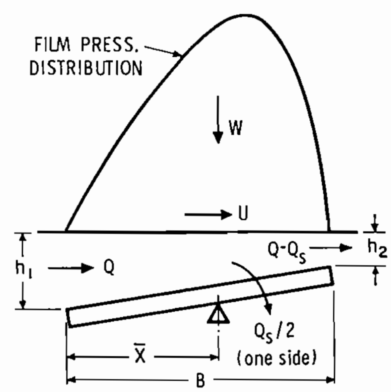

Figure 11. Geometric Fluid Wedge.

B, the length of the pad in the direction of motion; L, the width of the pad; W, the load (force) on the pad; P, the unit loading in psi; and K is a constant that is a function of the pad inclination angle and the pad L/B ratio.

If the calculated film thickness is the principal basis for bearing operation, much, of course, depends on the reliability of the analysis. Proven values of the constant, K, are available for a practical range of pad characteristics. However, assuming that the load is accurately known, the most tentative variable in the equation is the film viscosity. This depends on the temperature of the oil entering the pad and the rise in temperature as it passes through the pad. A rigorous computer analysis will attempt to take all factors into account, although the actual transmission of heat, churning in the housing, and mixing in the grooves between pads, limit the accuracy of any analytical treatment.

The geometric wedge that is basic to all hydrodynamic bearings applies most directly to thrust bearing analysis. The journal bearing analysis is not as direct, since it must take into account the diametrical clearance between the journal and the bearing bore, and the location of the shaft in the bearing. Here we make use of the "Sommerfeld Number," S, defined as:

$$S = \frac{\mu N}{P}\left(\frac{R}{C}\right)^2 \qquad (2)$$

where: μ is the absolute viscosity of the lubricant in the film; N, the journal speed in rps; P, unit loading, psi, based on projected area (bearing length × diameter); R, journal radius; C, bearing radial clearance.

This dimensionless number, S, is used as the basis for determining the position and value of the minimum film thickness, lubricant flow through the film, film absorbed power and temperature rise, and dynamic coefficients. Data is available for just about every common type of bearing configuration.

Thrust Bearing

It is very common to talk about load capacity in terms of the load per square inch of bearing area, or "Unit Loading." It is easy to determine and comfortable to use in evaluation. However, it is misleading. Unless one compares different size bearings for a particular machine, the variables in equation (1) all play essential roles in determining load capacity.

In an effort to present a realistic evaluation guide, the writer prepared a family of calculated curves that he felt would cover a range of conditions normally found in a large number of turbomachines. The curves are shown in Figure 12, and are for an 8 pad, tilting pad equalized thrust bearing with offset pivoted pads of optimum shape. They interrelate unit loading, bearing O.D., and thrust runner speed. To do this, the writer had to fix certain variables. The bearing O.D. is twice the I.D., and the pad sector angle is 38.2 degrees. The lubricant is SAE 10 oil at 120°F housing inlet temperature; a common condition in many turbines. The minimum operating film thickness was the most difficult to establish since so many practical factors are involved. For the purpose of evaluation and to determine the feasibility of a design, the film should be satisfactory as a minimum value, but not overly conservative. In this light, the minimum film thicknesses used in the curves were varied continuously from 0.0008" at 5" O.D. to 0.0014" at 20" O.D. Changes in film viscosity and pad curvature were included over the range of the curves.

As an example of the use of the curves, we can start with a 4" diameter shaft. This would allow a 5" I.D. thrust bearing, and therefore a 10" O.D. bearing would be used in the curves. If, say, the shaft speed is 6000 rpm, the maximum unit loading comes out to be about 450 psi. A higher load would result in a

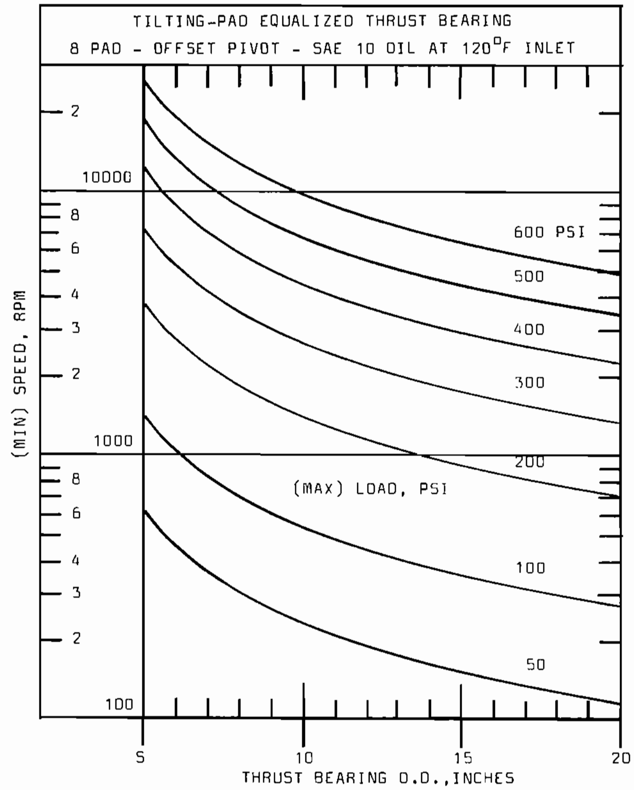

Figure 12. Calculated Evaluation Curves for a Tilting Pad, Equalized Thrust Bearing.

smaller film, and a lower load would give a greater margin of safety with a thicker film.

Although one could argue about the selected film thickness values, the results could be approximately altered using one's own values, since the bases for the curves are known. It must be emphasized that this is only an evaluation tool and any design must be based on a thorough analysis.

Fixed pad bearing theory can predict the same load capacity for a step or tapered land design as that for a tilting pad type. This assumes that the incline or step is the optimum value for the specific set of operating conditions. What with machining tolerances and bearing distortions, this is never the case, unless by accident. So, although this type can be designed to carry high loads, from a practical standpoint we can expect a lower load capacity as compared to a well designed tilting pad type. How much lower depends on the physical factors.

Since these fixed types are relatively simple bearings, there has been effort to improve their load capacity. For the step bearing, reducing side flow by shrouding can increase load capacity by about 50%. Many tapered land bearings have a compound taper at the inside diameter to equalize oil flow across the pad segment and thus help to equalize oil temperatures and minimize distortions. Reference [22] describes load tests on a tapered land bearing using a babbitt face on a copper back material instead of steel. As much as a 75% increase in load capacity is shown, and is attributed to the reduced distortions and increased heat flow due to the properties of the copper.

The tapered land and step bearings can be analyzed using such references as [8, 9, 24, and 30]. It is essential that the analysis be done for the upper and lower limits of step height and taper drop, based on manufacturing tolerances. A mean value will almost never exist.

The parallel surface bearing is the least capable of carrying load. There is still speculation on how it develops a pressurized film. There are the "Thermal Wedge" and "Viscosity Wedge" theories. Others talk of a pressure wedge due to runner misalignment and a wedge shape due to thermal distortions. All probably contribute. A number of technical papers exist, and can be found referenced in the text books.

Centrally Pivoted Thrust Pads

The curves of Figure 12 were prepared for a bearing with pad pivots offset in the direction of runner rotation, rendering it unidirectional. The amount of this circumferential offset determines the inclination of the pad, and is optimum at about 58% of the pad length. When the pivot is moved to a central position, (50%), theory predicts that if the pad surface is perfectly flat it will have no load capacity. However, they do carry substantial loads, and are in common use because the central pivot allows operation for both directions of runner rotation. Their load capacity is derived from a convex pad surface profile which develops during operation. This convex surface results from mechanical deflection due to load, and thermal deflection due to the thermal gradient across the thickness of the pad. The effect of the convex surface is presented theoretically in [19, 20, and 21]. It is shown that loads comparable to an offset pivot can be predicted for a central pivot if the crown height is at or near its optimum value. The optimum crown height is small and is on the order of the film thickness.

An exact theoretical prediction of the height and shape of the crown is difficult to achieve, although a reasonably useful value can be obtained. Thermal crowning is a function of the film and surrounding oil temperatures, and the many real factors that have an influence have been discussed. The force that deflects the pad elastically is the non-uniform film pressure over the pad area. In addition, the shoe shape is not uniform, the type of support must be considered, and the pad thickness varies due to buttons and recesses. Although it is complex, it is workable using hydrodynamic and elasticity analyses with, possibly, simplifying assumptions. Because of the practical influences on the crowning, a considerable amount of experimental work has been done to determine the actual performance of the centrally pivoted pad thrust bearing.

References [22 and 23] are experimental papers on large (25″ O.D.), and small (6″ O.D.) centrally pivoted, tilting pad thrust bearings. Both describe the use of pad materials with high thermal conductivity to significantly increase load capacity. The first paper was written in 1959, so this concept has been around for some time.

Figure 13 is taken directly from [23], and is an experimental plot of pad temperature versus loading for conventional babbitt faced, steel backed pads with center and offset pivot locations. The advantage of using an offset pivot is shown by the significantly lower "offset" curve (c). Since the centrally pivoted pad requires a small amount of convexity for optimum performance, the problem appears to be to limit distortion in order to approach the required amount. The use of babbitt faced, copper backed pads produced dramatic results. This is shown in the failure test curves in Figure 14, which were replotted from [23], comparing the offset pivot steel backed to the center pivot copper backed pads. The offset steel backed pad shows a lower temperature at low loads, but at the higher loads the temperatures are about the same, and failure occurs at the same load point for both. Reference [22], qualitatively,

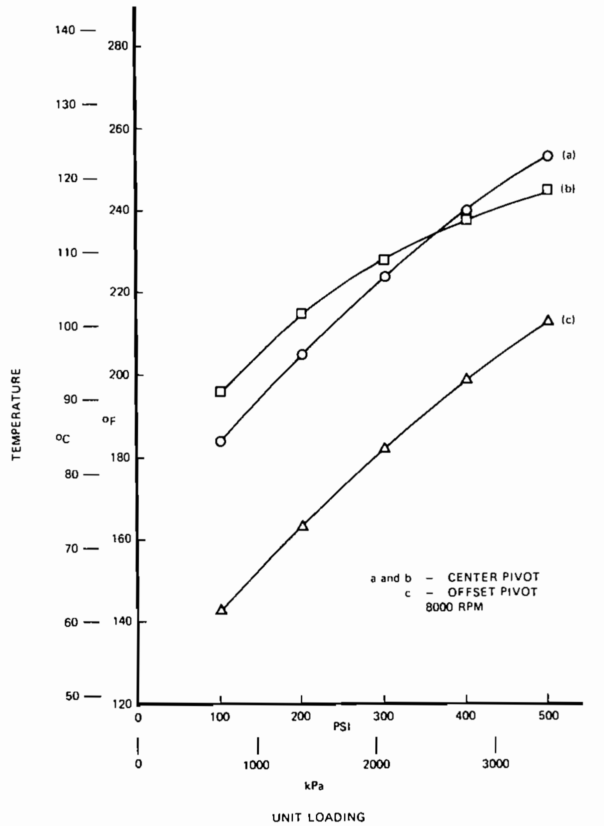

Figure 13. Influence of Pivot Position on Pad Temperature for Babbitted, Steel-backed Pads; from Test. For a Six Inch Diameter Tilting Pad Thrust Bearing Using Light Turbine Oil (from reference 23).

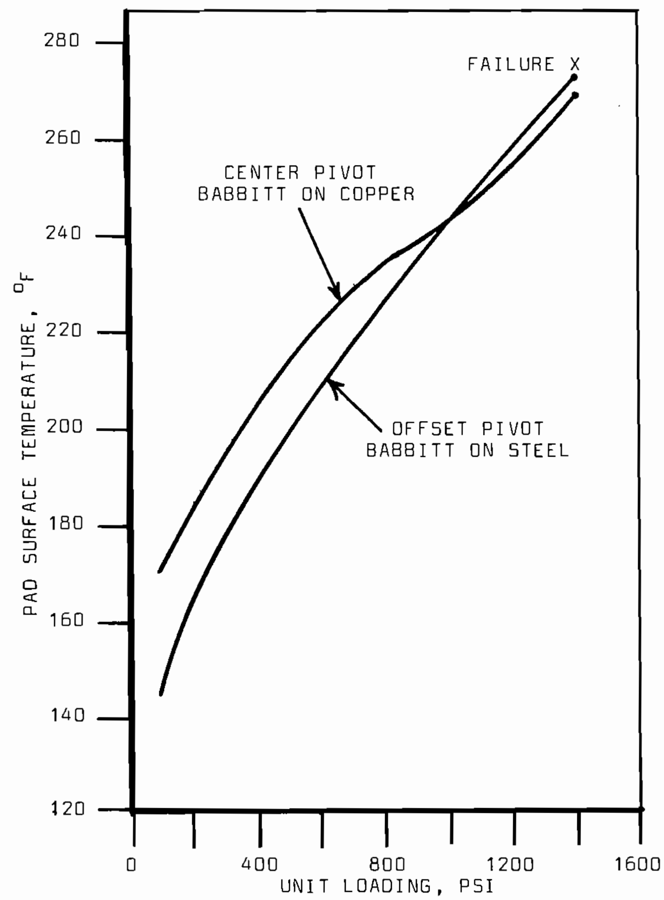

Figure 14. Failure Test Showing Influence of Pivot Position and Babbitt Backing. For a Six Inch Diameter Tilting Pad Thrust Bearing Using Light Turbine Oil (from reference 23).

shows the same results. Even though we can expect greater elastic crowning due to the lower modulus of elasticity of copper, the increased heat flow and reduced thermal crowning appears to produce a net effect that is beneficial.

Journal Bearing

A family of calculated evaluation curves were also prepared for journal bearings, Figure 15. These are for a split construction cylindrical bearing with two 30 degree width axial feed grooves at the split. It is one of the most widely used turbine bearings.

The psi unit loading is based on the bearing projected area. The oil is SAE 10 at 120°F inlet temperature. The diametral clearance was established for each diameter by using a constant clearance modulus of 0.0015; where the modulus is the ratio of the diametral clearance to the diameter. In reality, this would vary somewhat with speed and other factors, but using a constant was convenient and felt to be satisfactory for these evaluation curves. The minimum film thickness was varied continuously from 0.0008" for a 2" diameter journal to 0.0014" for an 8" journal. Also, the L/D ratio was varied continuously from 1 for the 2" journal to ⅝ for the 8" journal. The curves are for "central loading," where the load line bisects the bearing arc. "Offset loading," where the load line divides the arc unequally, can be analyzed using [25].

Although one can refer to the text books for the analysis of the many other types of journal bearings, the tilting pad type does warrant discussion. It has become popular, particularly for machines where rotor instability may be encountered. There are a number of design papers, and one that presents design curves for the 3 and 5 pad bearing is [26].

In discussing the tilting pad journal bearing, "geometric preload" must be introduced. It can be compared to the crowning that was described for the thrust pad. There are two radial clearances in the bearing. The "ground," or "pad," clearance is where the pad ring, before splitting, is machined with a bearing surface radius equal to the journal radius plus a clearance value. The "assembled," or "bearing," clearance is the clearance between the journal surface and the point on the pad surface over the pivot, and can be done by machining methods or adjustable pivots. The amount of preload is the ratio of the assemble to the ground clearance. It usually varies between 0.5 and 1 (for no preload). Preloading produces a stiffer bearing.

Many designers will not recommend this type bearing without some preload, because without it there is the possibility of a condition called "spragging" [27]. In Figure 16 (b), if an unloaded pad happens to become tipped so that its leading edge touches or gets very close to the journal, the divergent film created causes the film pressure between pad and journal to fall. This sucks the leading edge tighter against the journal, thereby aggravating and perpetuating the condition. In addition, in the case where a journal is lightly loaded, this decrease in pressure at one pad can cause a shift in journal position, cause spragging of the other pads, and produce an audible type

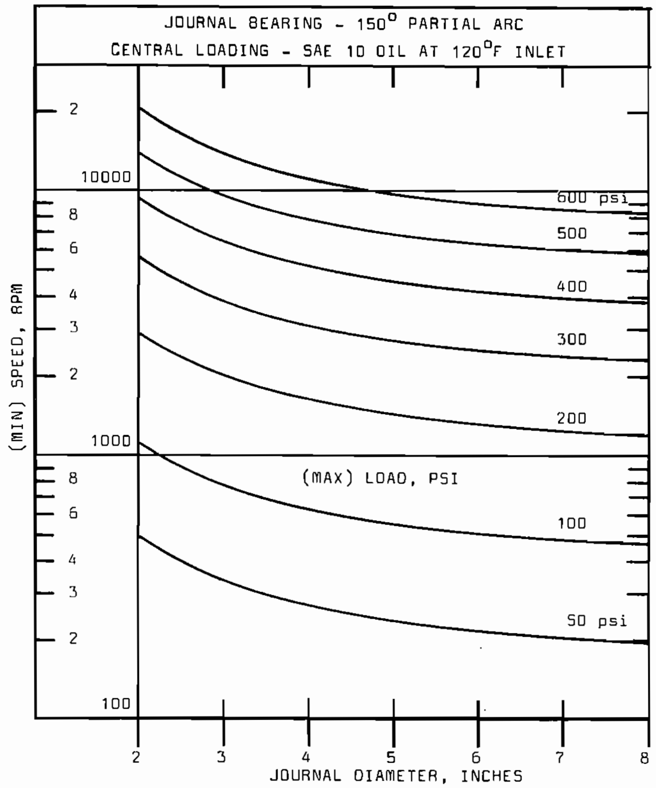

Figure 15. Calculated Evaluation Curves for a Partial Journal Bearing.

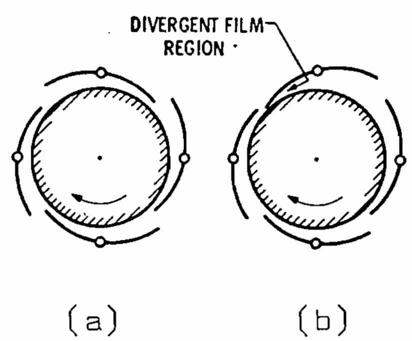

Figure 16. Tilting Pad Journal Bearing; (a) Normal Operation, (b) "Spragged" Condition.

of instability. A generous amount of preload will prevent this possibility by always maintaining a converging film at the leading edge of the pad.

Power and Temperature

Sometimes bearings that are properly designed from the hydrodynamic-oil-film point of view can fail badly from overheating. The heat generated in the bearing from fluid friction must be removed in order that some steady state operating temperature is reached. Bearing surface temperature, when monitored in the field, is a direct indicator of the state of health of a bearing, if interpreted correctly.

Referring to Figure 17, the fundamental equation for laminar fluid friction is

$$F = \mu \frac{UA}{h} \tag{3}$$

where: F is the friction force, and A is the surface area where the fluid shear takes place.

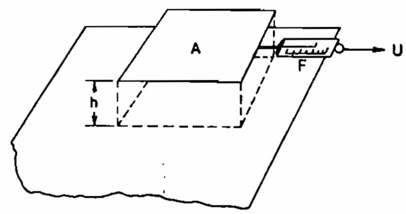

Figure 17. Shear Force for a Viscous Fluid.

After converting the force to power, the film temperature rise is determined by the flow through the film and the density and specific heat of the lubricant. A simplifying assumption normally used is that the heat rejection to the lubricant is the primary flow of heat from the film, due to the film flow.

Although equation (3) is fundamental and shows the relationship between variables, more is involved in bearing design. Turbulent flow can exist, and will be discussed in the next section. The film shape is a wedge, and not parallel. The film temperature varies along the pad or bearing arc in a manner depending on the film profile and local pressure gradients, and will result in a variable fluid viscosity. Because of end and side flows, a maximum temperature, higher than the discharge value, will exist at some point in the pad or arc, and for severe applications will be critical to the bearing material. Bulk flow in the neighborhood of the bearing surface, and, in some cases, convection and radiation are other factors involved. These considerations are treated in the text books, and such references as [28] and [29] go into the detailed theoretical analysis.

Actually, a fluid film bearing is quite a mixer, and a rigorous heat balance can be complex. A clear and usable method of analysis, with an example, can be found in [30], where temperatures are calculated at various points in the bearing and housing. An additional factor called "hot oil carry over" is treated in [31] with equations and coefficients. It applies to a thrust bearing, and deals with the hot oil that is carried over from the discharge of one pad to the inlet of the adjacent pad.

So far, the discussion has been devoted to the power and temperature rise in the oil film. However, at high speeds the power absorbed due to the churning of the oil in the bearing casing can sometimes be greater than that due to the film friction power. This is most pronounced in thrust bearings where you have a rotating disk of a relatively large diameter. Reference [32] investigates this experimentally for a tilting pad thrust bearing. Three methods of supplying oil to the bearing were used: (1) Pressurized Casing — where the discharge from the casing is restricted, so that it is full and the bearing is flooded; (2) Throttled Inlet — where the oil entering the casing is restricted and a critical amount of flow is allowed; (3) Directed Lubrication — where the oil is supplied to the inlet of each pad through suitable nozzles mounted between the pads. Figure 18 is reproduced directly from the paper, and shows the effects on absorbed power.

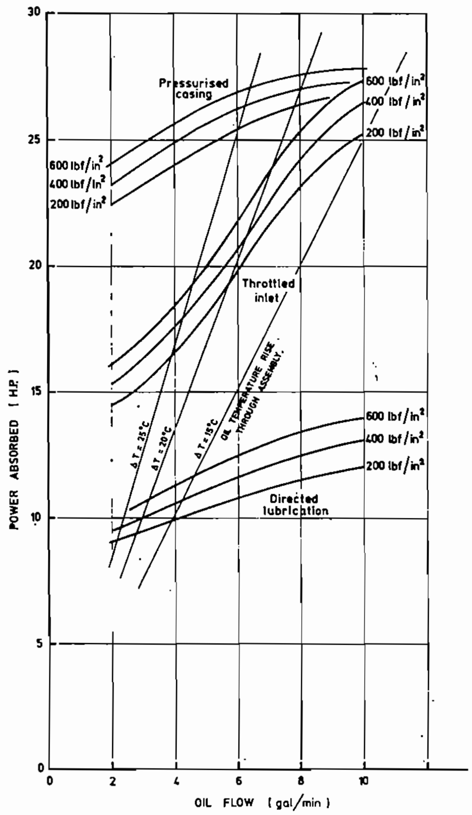

Figure 18. Influence of Various Lubricating Systems on Bearing Absorbed Power. For a Five Inch Diameter, Tilting Pad, Center Pivot, Thrust Bearing Operating at 12,000 rpm (from reference 32).

The directed lubrication gave a substantial reduction in power for the 12,000 rpm speed. The throttled inlet is quite sensitive to flow, where too much flow approaches a flooded condition, and, more important, too little flow could starve the bearing. In all cases, there is an optimum amount of flow, beyond which there is little or no effect on power; a fact many have seen in the field as related to temperature. The decision on a design approach would depend on speed, and other factors such as complexity and margin of safety.

Even with all of the considerations involved, adequate results can be obtained for most bearing designs. With proper assumptions, and using an iterative process, equations and coefficients are available for the analysis. Here, as for load capacity, the real inlet temperature to the film is questionable, and the viscosity in the equations will be the least accurate variable.

Turbulence

There is an upper limit in speed beyond which the bearing performance based on laminar flow in the oil film is inaccurate and can result in large errors. Beyond this limiting speed, turbulence occurs in the oil film. This need not be feared or avoided, since turbulence, as a phenomenon, has no intrinsic harmful effects on bearing behavior. In effect, it can be treated as equivalent to operating with an increased lubricant viscosity. This then results in higher power loss, load capacity, and temperature rise than that predicted for laminar flow conditions.

The first reports of turbulence in fluid film bearings were experimental, [33, 34, 35], and provided a starting point for the large amount of theoretical work that has been done since. Reference [36] is a good compilation of recent papers that also provides an extensive bibliography within them.

The Reynolds number is defined as

$$R_e = \frac{Uh}{\nu} = \frac{Uh}{\mu}\rho \qquad (4)$$

Where: U is the moving surface velocity, in/sec.; h is a film thickness in inches, which is the developed film in a thrust bearing, and the radial clearance in a journal bearing; the lubricant characteristics are: ν, the kinematic viscosity, in²/sec.; μ, the absolute viscosity, lb-sec/in²; ρ, the mass density, lb-sec²/in⁴.

It is first necessary to determine if turbulence exists in the bearing film. The critical Reynolds number that denotes the onset of turbulence does not appear to be one fixed value, and will vary somewhat with the bearing characteristics. A typical value used in both journal and thrust bearings is R_{ec} = 1000. This is based on a mean film thickness value, which for a thrust bearing is about 1.4 times the minimum film thickness, and for a full journal bearing is equal to the radial clearance.

A convenient method for determining bearing conditions in the turbulent regime is to first analyze the bearing assuming laminar conditions, and then correct the results using a multiplication factor, K. Figure 19 is a plot of Reynolds number versus the factor K, for load capacity and power, and for journal and thrust bearings. The thrust bearing, an optimum offset pivot tilting pad type, uses the minimum film thickness for calculating Reynolds number. The journal bearing, full cylindrical to 100° arc, uses the radial clearance in Reynolds number. Both are for aspect ratios, (L/D, L/B), of one. They are

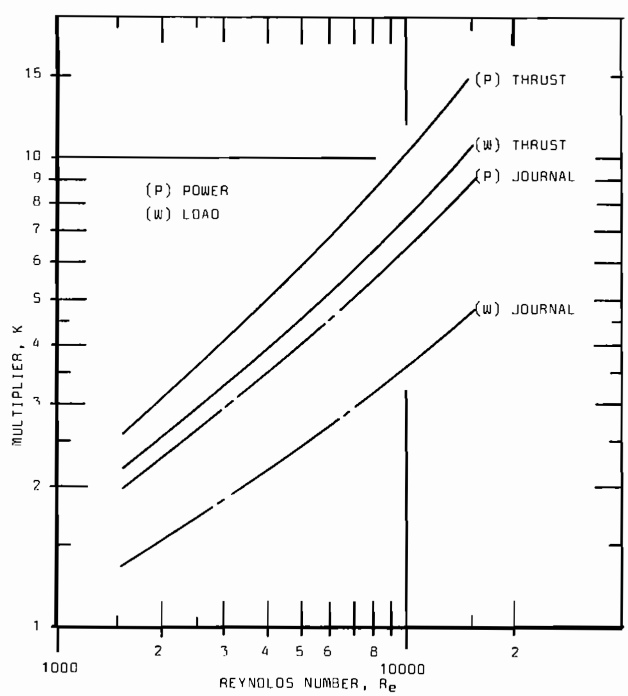

Figure 19. Multiplication Factor to Correct Calculated Bearing Laminar Power and Load for Turbulent Film Conditions. Use Thrust Minimum Film Thickness and Journal Radial Clearance for Calculating R_e.

average values, and were prepared by the writer from data obtained from a number of sources, including [37 and 38]. The tilting pad journal bearing was not included because it involved too many variables and average values would have no real meaning. Reference [39] gives this data.

As a general note, the higher load capacity in the turbulent regime would suggest that a reduction in power could be obtained by reducing the surface area of the bearing. If this is possible, one would design for an optimum bearing size.

Vibrations and Stability

In recent years there has been a marked increase in the average operating speed of rotating machinery. Because of this, rotor-bearing system dynamics problems are more prevalent. They can include prediction and control of rotor response, balancing, and rotor-bearing stability.

The primary causes of machine vibrations are unbalance and misalignment. Balancing techniques are more complex since rigid rotor balancing methods are not adequate for balancing high speed flexible rotors. In properly balanced commercial rotating machinery, vibration problems are normally due to various forms of journal bearing instability, as related to the rotor.

Rotor-bearing system dynamics are extremely complex when treated in a thorough manner. This is displayed in Figure 20, where all properties of the rotor are involved and the bearings and their pedestals are treated as nonlinear springs and dampers. An accurate solution requires a correctly programmed computer analysis.

Apart from unbalance and non-fluid vibrations, there are three basic varieties of instability associated with journal bearings.

1. A "half-frequency whirl" that may occur with a rigid shaft at any running speed, having a frequency just under half the speed of the shaft, and which is self generated in the bearing.
2. A "resonant whip," occurring at running speeds not less than twice the natural frequency of the rotor due to shaft flexibility; the whip frequency being equal to the rotor natural frequency.
3. "Oil-film criticals" as a function of the spring constant of the film and the natural frequency of the rotor.

Half frequency whirl is well reported in the literature. Resonant whip, although reported in a number of references, is difficult to predict reliably. Actually this holds true for all rotor-bearing systems involving a flexible shaft. They can only be analyzed properly using a computer program involving such disciplines as hydrodynamics, mechanics, and elasticity.

The effect of the bearing spring constant on the rotor can be understood by observing the mode shapes for the first three natural frequencies of a simple uniform shaft, as shown in Figure 21. The infinitely stiff bearing would be similar to that of a simply supported shaft. There is sufficient published data to get a good approximation of the natural frequency of a rotor-bearing system; except for flexible rotors. A critical speed map, Figure 22, of bearing stiffness versus system critical is plotted for a particular rotor. Where the horizontal and vertical bearing stiffnesses intersect the curve is the system critical, and can be avoided in design. Note that as the bearing stiffness increases, the system critical becomes asymptotic to a constant value, which is the natural frequency of the rotor on simple supports. Some dynamic bearing coefficients can be found in [38 and 39]. Their use must be considered in the light of such effects as elastic housing distortions, misalignment, etc. Other more general references are [40, 41, and 42].

Rotor response is also dependent on the excitation and damping in the system. Although certain bearing types can eliminate some forms of instability, whirl can still occur with severe excitation forces. Some frequently occurring sources of excitation are: mass unbalance, coupling misalignment, frictional vibrations such as seal rub, gears, universal joints, liquid pulsations, fans, etc. Bearing damping is included in the analysis, and, in general, is a small factor in increasing the rotor critical speed. It does play a significant role in the amplitude of vibration.

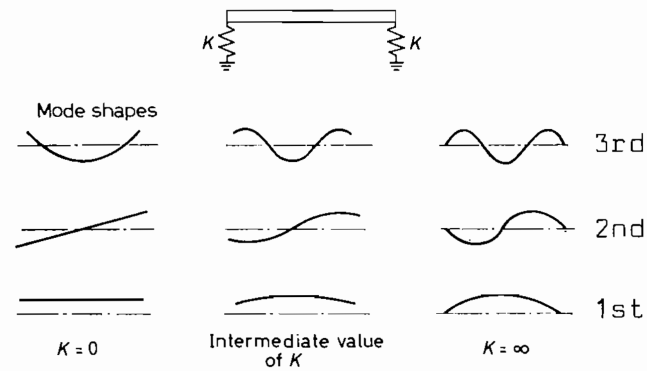

Figure 21. The Effect of Bearing Stiffness on the Mode Shapes of a Simple Beam.

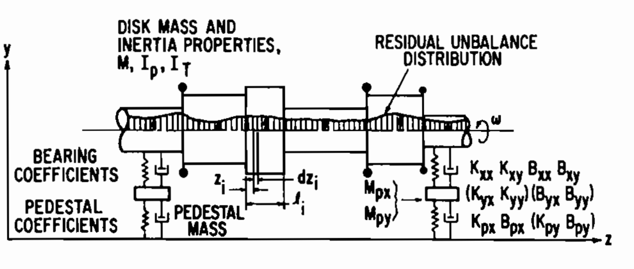

Figure 20. Factors Involved in Rotor-Bearing System Analysis (from reference 41).

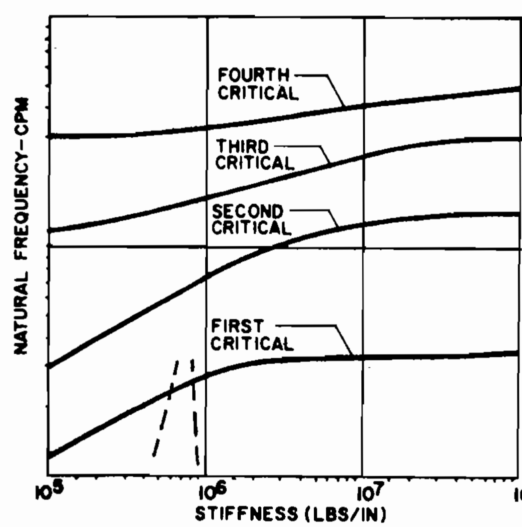

Figure 22. Influence of the Bearing Stiffness on the Natural Frequency of the System (from reference 42).

There are a number of ways in which instability can be eliminated or its effect minimized. Journal bearing types such as the pressure dam, lobe, and tilting pad, can eliminate some forms of instability, if properly designed. Heavier bearing loads, stiffer shafts, and a reduction in rotor mass will improve the stability characteristics. For certain applications, a thin film of oil can be designed around and to support the bearing shell, to increase damping. This "squeeze film damper" must be accurately tuned to be effective.

In some cases, poor quality control such as bearing misalignment, ellipticity, and surface irregularities have been known to improve stability. Capillary seals are susceptible to half frequency whirl, since they are really an unloaded journal bearing with poor stability geometry. Vibrations in thrust bearings are rarely troublesome, but with tilting pad types, pad flutter must be analyzed and avoided in design. This also applies to the tilting pad journal bearing.

Bearing Materials

In all the time since Isaac Babbitt patented his special alloy in 1839, nothing has come along that encompasses all of its excellent properties as an oil lubricated bearing surface material. The babbitts have excellent compatibility and nonscoring characteristics, and are outstanding in embedding dirt and conforming to geometric errors in machine construction and operation. They are, however, relatively weak in fatigue strength, especially at elevated temperatures, and where the babbitt is over about 0.015" thick. In general, the selection of a bearing material is always a compromise, and no single composition can include all desirable properties. Babbitts can tolerate momentary rupture of the oil film, and may well minimize shaft or runner damage in the event of a complete failure. Tin babbitts are more desirable than the lead based materials since they have better corrosion resistance, less tendency to pick up on the shaft or runner, and are easier to bond to a steel shell.

An excellent investigation of the limiting temperatures for babbitt is described in [43]. It takes into account the temperature and local pressure in the oil film. Application practices normally suggest a maximum design temperature of about 300°F for babbitt, and designers will set a limit of about 50 degrees less. As temperatures increase, there is a tendency for the metal to creep under the softening influence of the rising temperature. Reference [43] relates oil film pressure and temperature to conditions where babbitt creep is initially encountered. Creep can occur with generous film thicknesses, and can be observed as ripples on the bearing surface where flow took place. With tin babbitts it was observed that creep temperature ranges from 375°F for bearing loads below 200 psi to about 260°-270°F for steady loads of 1000 psi. It is suggested that this may be improved by using very thin layers of babbitt, such as in automotive bearings.

Of course, some other materials are being tested or are in use for special applications or to improve performance. The use of copper (1% Cr.) backing to improve thermal conductivity has been discussed. Solid Aluminum (6% Tin) and Copper-Lead facing material were tested in [23]. Where lubricants other than oils are used, special well tested materials are available and will be discussed later.

Misalignment

Some amount of shaft and bearing misalignment will exist in every machine. The amount and how it manifests itself determines the effect on the bearings. With bearing film thicknesses on the order of 0.001", it is hard to believe that any mechanical alignment device can completely take care of misalignments due to manufacturing tolerances, and thermal and elastic distortions. If the bearing works, it is probably supporting a higher load, or it has deformed, or both. A classic story goes like this. A machine manufacturer was asked, "What are your bearing loads?" "That we don't know," came the surprising reply. "But surely you must have some idea of the rotor weight and impeller loads," the questioner persisted. "Yes, indeed," came the voice of experience, "but those are not the important loads. The real loading is due to misalignment." This is somewhat exaggerated for most applications, but it does make the point.

Self aligning bearings do minimize an added load effect, and can save a bearing that has a small margin of safety. The two sketches in Figure 23 attempt to cover the various forms of misalignment that bearings will feel. The case where the runner or shaft does not "run out" during rotation is shown in (a), where just the bearings are misaligned. This is also the case in (b) if the shaft is misaligned because the journal bearings are not aligned along their centerline. Here, self aligning bearings having spherical seats or equalizers can do a good job by "setting" themselves at assembly. A bearing that aligns due to deformation, whether by design or not, can also work. However, the deformation forces could be high enough to be a problem, and for tilting pads, pivot points could become sluggish so that the pads may be less sensitive to changes in operating conditions. As to journal alignment, line boring and facing in the same setup should be done wherever possible.

The problem is entirely different where there is a run-out, or a "swash" effect, Figure 23 (b). This would occur when the rotating shaft is bent or deflected, or when the thrust runner is not perpendicular to the shaft. Here, even the most sensitive self aligning bearing will not follow the run-out because of the friction and inertia of the bearing components. However, sensitive alignment designs with hardened surfaces will do some work and reduce the effect of this type of misalignment. There have been some theoretical treatments for the "swash" effect and for misaligned shafts with small amounts of tilt.

From the writer's experience, there is sufficient evidence to support the importance of self aligning bearings in the majority of turbomachines. In most critical and expensive machines the relatively small added cost is warranted.

Water Lubricated Bearings

Bearings that use water as the lubricant can be designed as accurately and reliably as those using oil. This was not always true. In the early 1950's a number of groups conducted an extensive water lubricated bearing development program [44]. The application was the primary water circulating pump in a water cooled nuclear reactor. Since essentially zero leakage was permitted, the pump was hermetically sealed, necessitating the use of the water process fluid as the lubricant for the journal and thrust bearings. The nature of this application placed the prime emphasis on reliability, with no maintenance over years of service where there would be many stops and

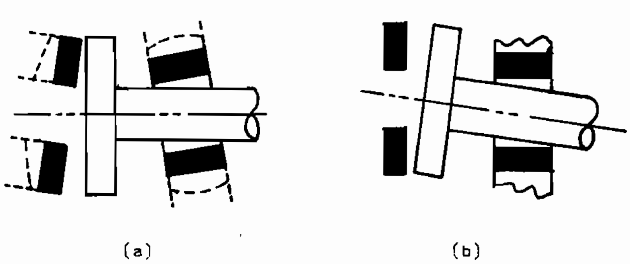

Figure 23. Various Forms of Misalignment.

starts and possible shock loading. The results of that successful program provided the design criteria for future less sophisticated but high performance applications, where low viscosity corrosive liquids could be used as the bearing lubricant.

Water has a number of advantages as compared to oil. It is not flammable, not contaminating, is expendable, can operate at high temperatures, and has a high specific heat. As the process fluid, equipment is simplified due to the elimination of the auxiliary lube oil system, and possibly seals. However, water has three important weaknesses that must be allowed for in bearing design.

1. *Low Viscosity* — Because the viscosity of a conventional oil is approximately 100 times that of water, the bearing will support less load. Therefore, in order to have a useful load capacity, the minimum film thickness must generally be rated lower than that for oil bearings. This usually means that a better quality bearing is needed, and that some form of self alignment is essential. In addition, the low kinematic viscosity does make it more susceptible to turbulence.
2. *Corrosion* — Corrosion resistant materials are required for the bearing surfaces and the structural components.
3. *Boundary Lubrication* — Water has absolutely no boundary lubrication properties, and the bearing is completely dependent on materials for this help.

A journal bearing design is shown in Figure 24. It has a self aligning spherical surface making line contact with its straight cylindrical housing. A thrust bearing, Figure 25, is a tilting pad, equalized type where all articulating points are sphere against flat. The structural materials are 300 series Stainless Steel, and all contact pins and spherical surfaces are Stainless Steel 17-4 PH, hardened to Rc-40. As will be discussed, the bearing surface materials are usually brittle and weak in bending, and special methods of retention are provided, as shown in the figures. Other type materials may be bonded to a backing material. Two general references are [45] and [13].

The writer conducted the thrust bearing portion of the water bearing program that was mentioned. The bearing was a center pivot, tilting pad equalized type, and the investigation was primarily concerned with bearing surface materials. Using a wide variety of noncorrosive materials, just about all of them failed at the start. Through a series of experimental happenings a crowned surface was indicated, and was imposed on the surface of each pad. All of the materials that failed before operated successfully. The theory in [19] was derived to explain it.

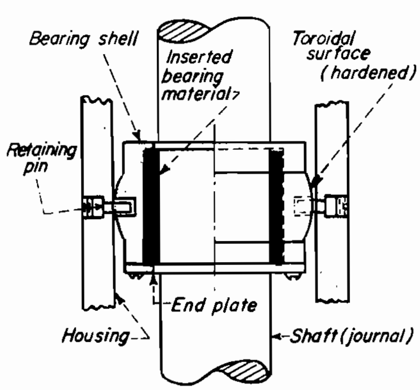

Figure 24. Water Lubricated Self Aligning Journal Bearing (from reference 45).

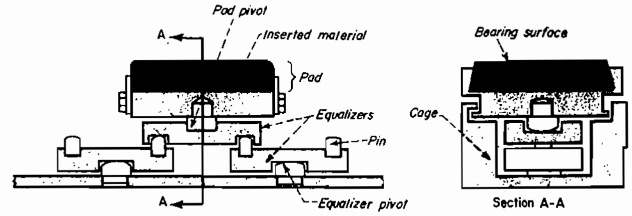

Figure 25. Water Lubricated Tilting Pad Equalized Thrust Bearing (from reference 45).

Since the specific heat of water is high and bearing loads are relatively low, the temperature rise in a water bearing film is usually quite low. This low film temperature, coupled with the low coefficient of expansion of most of the recommended materials, results in a thermal crown that is either extremely small or nonexistent. The same is true for the elastic crowning, because of the low load. Therefore, a convex surface must be physically imposed on the surfaces of pads that are centrally pivoted. The crown height may be on the order of 0.0003", and can be done by lapping techniques, and checked with an optical flat. This imposed crowning may also be beneficial for starting where there is no boundary lubrication help from the water. During that program it was found that, with crowning, most materials could easily accomplish a prescribed 500 stop-start test with no observable wear. At the time, the writer speculated that the crown developed a type of squeeze film that aided in starting [46].

As to bearing surface materials, corrosion resistance eliminates most of the common materials used for oil. The very hard materials such as Aluminum Oxide and Tungsten Carbide can operate well but have poor friction and wear properties in the event of an overload, or for marginal operating conditions. Impregnated laminated phenolics are used successfully, and have the advantage of not being brittle, and can be bonded to a backing. An excellent material that has a long history is Carbon-graphite. It is largely self-lubricating, has a low coefficient of friction, has little tendency to score soft steel, wears very slowly, will take a surface profile, and can be used at high temperature. However, it is brittle and must be adequately mounted and supported. It will operate well against many journal and runner materials, although a well tested material is Nitrided 300 series Stainless Steel. In contrast, there are many less sophisticated applications, as old as lubrication itself, that use Bronze. Also, the very successful Cutless bearing, Figure 26, uses rubber for contaminated water.

The design analysis is the same as that for oil, or any other incompressible fluid. Load capacity may be 1/10 that of oil, and film thickness ratings can be as low as 0.0005. These are the low values, and generally one can design for more. The absorbed power may be 1/5 of that using oil, but turbulence could reduce that margin. This is only intended as a feel for what can be expected. Dynamic characteristics are especially critical because the bearing stiffness and damping are much lower than would be for oil. Reference [47] pursues this.

Whereas in the past, water lubrication may have been considered esoteric for high performance applications, it now can be considered a conservative design approach.

DESIGN PITFALLS

Everything presented to this point has been based on documented and accepted information, with some personal comment. The following is, to a large extent, subjective. Al-

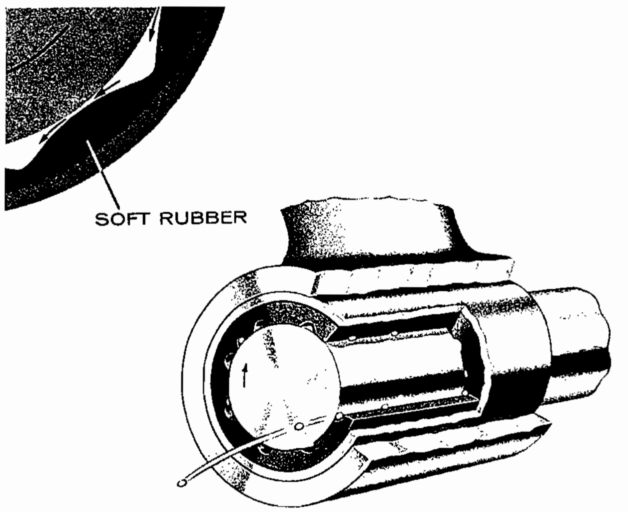

Figure 26. Cutless Bearing for Lubrication with Contaminated Water. As Shaft Rotates, Grit is Rolled into Water Grooves and Flushed Away. (Courtesy of B. F. Goodrich Co.)

though designers will probably agree on the overall content, the degree of importance and emphasis in different areas depends on one's experience. This then represents the writer's experience in having dealt with bearing problems over the years.

It is obvious that reliability is of paramount importance in a working machine. Bearing failures do occur, even when the proper analysis is performed and proven practices are followed. Some of the reasons why this may happen is our subject.

The problem areas are:

(a) Conditions the bearing will see.

(b) Type of bearing — selection and use.

(c) Physical design and hardware.

(d) Theoretical design analysis.

(e) Installation and maintenance.

They are listed in the order as the most prevalent reason for bearing trouble, although (a) and (b) can be considered about equal in importance. Because of space, the comments and number of examples are limited.

(a) *THE CONDITIONS THE BEARING WILL SEE* in the machine are not always what the bearing design includes. They may be overlooked, not given, unknown, or just machine characteristics that are difficult to analyze accurately. They can include: the effects of couplings and seals; thermal and mechanical distortions and machine tolerances that produce misalignment; accurate bearing loads, from all sources; load/speed conditions; rotor unbalance and natural frequencies; installation fits, alignment and piping flexibility; and the lube oil supply system.

Extracting bearing conditions is really the bearing designer's responsibility. He knows what he needs. Where there are unknowns he must try to deduce conditions, and in doing so, be conservative. It usually follows that a good bearing designer is also a good general engineer, since all machine disciplines are involved in his work.

Here is an example involving an external load. A steam turbine driving a compressor through a gear type coupling is shown schematically in Figure 27. The active tilting pad thrust bearing in the turbine had occasional high shoe temperatures, and inspections showed signs of pad surface distress. The coupling was the source of a significant added thrust load, adding about 5000 pounds to the 10,000 pound turbine thrust load. When cold, the thrust bearing supported the turbine load. As the turbine heated up, the shaft expanded. The friction at the gear teeth in the coupling prevented slip until the axial expansion force was high enough to overcome the axial coupling friction force. That extra load had to be supported by the active thrust bearing, and was displayed on test, through load cells. This reaction would occur during every thermal change of the system, and although the coupling load would eventually drop, it never completely relieved itself. The magnitude of the axial coupling force is a function of the shaft torque, coupling size, and the coefficient of friction at the teeth as influenced by materials, lubrication, finish, hardness, and design. Since this was a "fix," space was limited so that the thrust bearing size could only be increased a small amount. A new optimum design used hardened spherical offset pad pivots, and a sensitive equalizer system. A photograph of the replacement bearing is shown in Figure 28. After many years of operation in a number of identical installations, pad temperatures have been low and normal, and when inspected, the pad surfaces are in a "like new" condition. Coupling forces, based on a coefficient of friction, are now being spelled out in bearing inquiries for this type of application.

Journal bearing examples also involve unspecified loads. They may be off-centered, reversal, vibratory, or due to shock. Misalignment loads can be high and are exaggerated in large machines.

(b) *THE SELECTION OF THE BEARING TYPE* requires theoretical analysis and experienced judgment. It is easy to be safe and use the most sophisticated bearing, but space, complexity, and cost are important factors. In large expensive machines, a conservative approach may be justified. In other machines, it is not. There is usually more than one good

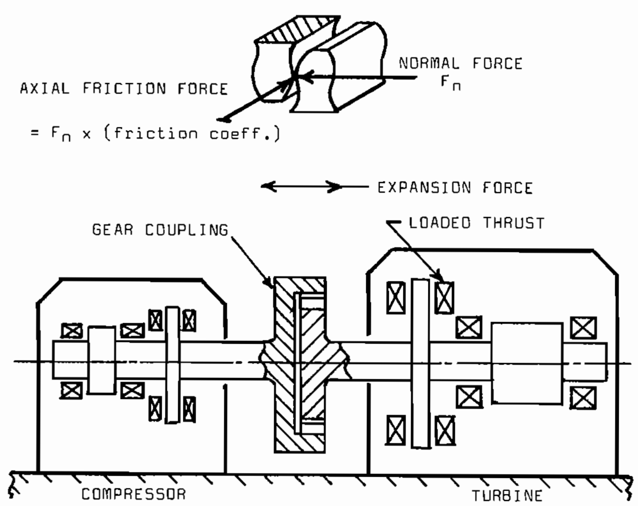

Figure 27. Schematic Sketch of a Steam Turbine Driving a Compressor Using a Gear Type Coupling.

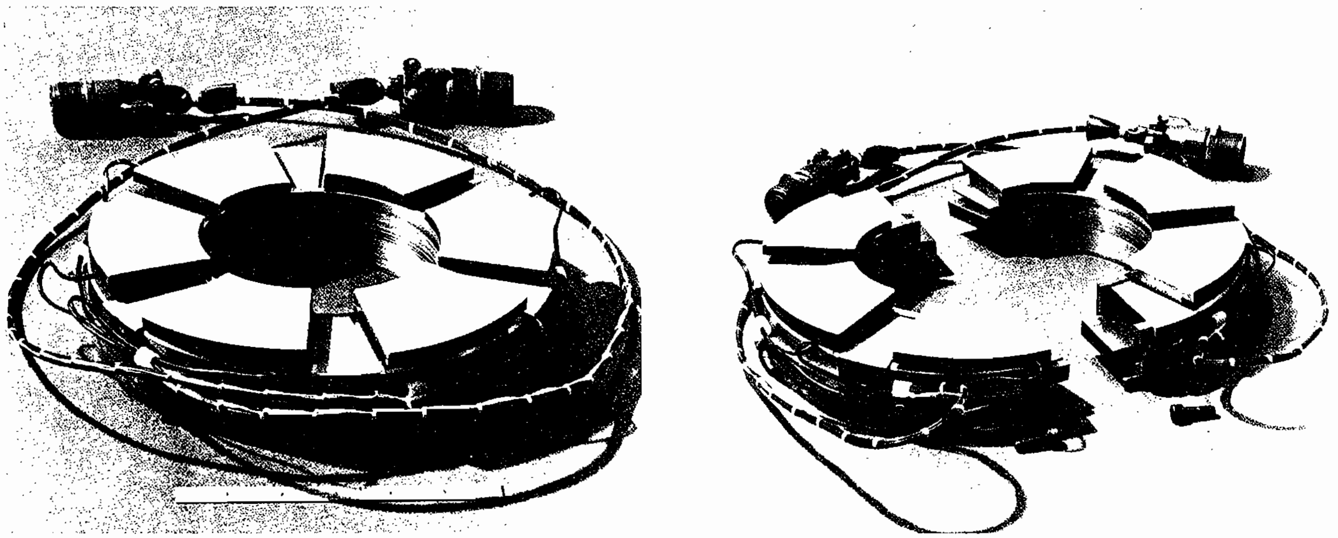

Figure 28. Oil Lubricated Tilting Pad Equalized Thrust Bearing for Steam Turbine Operating at 10,000 rpm.

choice, and the designer's job is to try to make the most balanced selection based upon his particular needs.

As a thrust bearing example, a small compressor, having a 1 inch shaft operating at 20,000 rpm, used a tapered-land thrust bearing. Space was limited, and the lubricant required was a very light Mil oil. For the relatively high load, the machine builder had calculated a 0.0003" bearing film thickness. Bearings would constantly wipe and fail. A tilting pad equalized bearing with offset pivots was designed into the exact same space and operated well, with no problems. The point here is that the tilting pad bearing gave the same 0.0003" calculated film thickness, but because of the bearing type, it was there. In the tapered-land bearing, manufacturing tolerances resulted in significant deviations from the very small taper angle required for that film thickness. The result was a sharp reduction in load capacity. The builder's analysis apparently did not take this into account. Even if it had been considered, it is doubtful that the tapers could have been made and remain right for this high load condition.

The choices for journal bearings are even larger than for thrust bearings. A high speed air compressor had oil lubricated partial type journal bearings. The conditions were such that hydrodynamic instability could be anticipated, and the bearings had to be designed with that as the primary consideration. However, in doing so, the bearing load capacity was reduced, and any unusual increase in radial load would result in bearing distress. A tilting pad journal bearing gave a reliable alternative. Since it was inherently stable for this application, it could be freely designed for a conservative load capacity. In addition, the spherical pad pivots provided self alignment for added safety.

These examples show an upgrading in bearing type because they are about the solution of bearing problems. There are many large parallel face thrust bearings that are properly used in turbines. Of course, grooved and ungrooved bushings are common for a wide variety of applications, and all factors considered, they are usually the right bearing for the job.

(c) *THE PHYSICAL DESIGN* of the bearing is normally spelled out in rather complete form before it gets to the drawing board. Manufacturing tolerances on journal bores and built up bearing components should be established at the analysis stage. Many load capacity failures and instabilities have occurred because the tolerance range was too large, and was not analyzed for the extremes. If there is a tolerance, one should be able to use it.

The choice of materials should not only be specified for the bearing surfaces, but also for the bearing structural components, and thrust runner and shaft, if possible. Hardening may be required for some surfaces. Inadequate, or omitted stress relieving has been the source of failures where relaxation distortions have occurred over a period of time. The coefficient of expansion of the bearing material and its housing must be analyzed so that the design bearing clearances and prescribed housing fits exist at the operating temperature.

Bearings have critical working edges at grooves and pads. This is where the lubricant is introduced for film formation, and a sharp edge can sometimes cause failure at the start. These edges should have a blended radius or a bevel, and not just a "break sharp edge" notation. Standards for the finish and flatness of bearing surfaces are available.

Since the machine housing is an extension of the bearing, it requires similar concern. A concentricity or perpendicularity spec for the bearing also demands similar specs for the housing, or a cap that has a rabbet fit in the housing. Line boring and facing in the same setup should be done wherever possible.

Rarely is there a problem in designing a lube oil system for the required flow, with the proper controls and auxiliary components. However, problems have occurred when the proper quantity of oil gets to the bearing too late. With pumps connected to the machine rotor, the flow is a function of rotor speed. If the pump is external, and not tied in to the machine start, it depends on the machine operator. Some maximum bearing loads exist at zero rpm, and others are a function of speed. Boundary lubrication can help a starved bearing, but not for long.

(d) *THEORETICAL BEARING ANALYSIS* procedures and data have been discussed. In addition to the references here, the yearly production of technical papers provide new experimental data, and offer advanced and new methods for analysis. Therefore, one can perform an analysis for a wide variety of applications, keeping in mind that the answers must be viewed with an eye towards the bearing type and machine conditions.

As previously discussed, the stability analysis of rotor-bearing systems is complex, especially with flexible rotors. Computer programs are available. Here the problem is that the bearing designer either overlooks this area, or treats it too simply instead of getting outside help.

(e) *INSTALLATION AND MAINTENANCE* seems to be the least troublesome as far as the direct cause of a bearing problem. It is the writer's experience that the men doing the bearing installation know their job. They usually have long experience with their specific class of machine, and the variety of fluid film bearing types used. It's not unusual for an experienced installation man to point out a harmful condition that had been overlooked by the bearing designer.

Where machines are critical and must provide a continuous output, maintenance procedures are well conceived and schedules are adhered to. Warranties, insurance, and production dependence dictate this, and problems are not too common. However, problems can occur due to dirty oil or a disrupted oil supply. Also, although rare, a unidirectional bearing can be installed wrong, and a new but damaged bearing can be overlooked and installed.

DISCUSSION

It is a risky job to meaningfully cover a relatively large field in a short presentation. If too much information is presented, it can be misleading and mistakenly used as a solution for a particular problem. It must be emphasized that nothing presented here takes the place of a thorough bearing analysis.

There are many excellent references that are available, but were not listed. They can be found referenced in the publications that are listed in the bibliography.

It is hoped that this discussion has given the reader an understanding of the operation of fluid film bearings, and has provided design information that can be used as an evaluation and feasibility guide, and as a prelude for further study.

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