PRINCIPLES OF FLUID FILM BEARING DESIGN AND APPLICATION

by

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ABSTRACT

After a summary of the basic hydrodynamic principles of fluid film bearings, this paper considers their application to currently used types of bearing, both thrust and journal.

The emphasis will be on providing information in a form which may help both designers of new machinery and also machinery users.

Currently available techniques for selecting fluid film thrust and journal bearings will be outlined and their application to actual design problems will be described. Consideration will be given to both mechanical design options and material options. While the main emphasis will be on oil lubrication, lubrication by other fluids will also be considered.

In conclusion some comments will be made on lubrication systems for fluid film bearings with particular reference to arrangements for feeding combined thrust and journal bearings.

INTRODUCTION

The object of this paper is to review the principles of fluid film bearing design and application from the point of view of the machine designer who is more interested in making a correct bearing selection than working from first principles. The types of bearings considered are those depending on self-generated fluid films under essentially steady load conditions; these are the types of bearings (other than rolling element or anti-friction types) which are used in the majority of turbomachinery applications.

Before considering in some detail the basic design parameters of first journal (radial) then thrust (axial) bearings, it is useful to consider the fluid film or hydrodynamic principle upon which the successful operation of both depends. This principle depends on the formation of a converging wedge profile between the two elements of the bearing surface; into this wedge the fluid is drawn by the movement of one of the bearing surfaces, and a pressure is generated in the fluid which tends to separate the two bearing surfaces. In a journal bearing, this converging wedge profile is formed automatically by a circular shaft running eccentrically in a cylindrical bush as shown in Figure 1 (a). In the case of thrust bearings, it is necessary that the converging wedge profile is either machined into one surface (as shown in Figure 1 (b)) or allowed to form itself due to mechanical and thermal distortions or deflections.

The basic behaviour of the fluid in the converging film is defined by the Reynolds equation [1]; this relates the generated pressures and flows in the oil film to the fluid viscosity, sliding speed and film profile.

Subsequent work has and still is concentrated on enabling accurate predictions to be made of these parameters for finite bearing surfaces of various types.

In the next two sections the design of journal and thrust bearings will be considered, using oil as the lubricating fluid; this is followed by a section on other lubricating fluids, a section on bearing materials and a section on lubrication of bearing assemblies.

JOURNAL BEARINGS

For practical reasons, journal bearings normally have their design features on the static element through which a plain cylindrical shaft rotates; for convenience therefore when referring to a journal bearing type, it will be assumed that the reference is to the bush or shell surrounding the shaft.

In its simplest form the journal bearing may be no more than a plain ungrooved bore lined with a suitable bearing material. While this construction is still occasionally used, it is normally necessary to machine oil entry grooves into the bearing

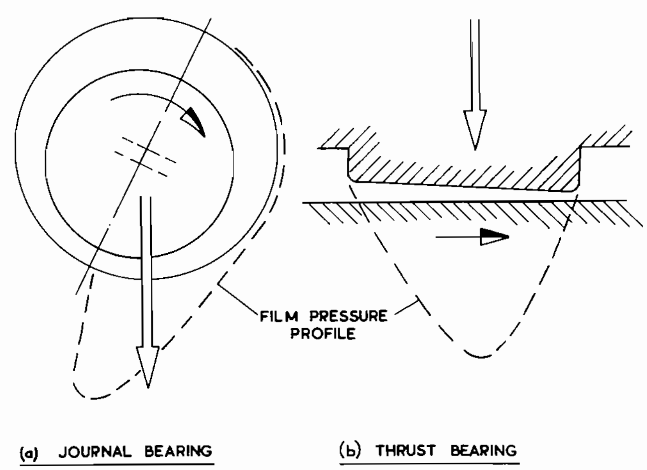


Figure 1. Hydrodynamic Converging Wedge Principle on Journal and Thrust Bearing.

surface and often non-cylindrical profiles are also used. In Figure 2 some common bearing profiles are comparatively rated as a preliminary guide to the merits of each type. It is beyond the scope of this paper to consider every possible type of journal bearing profile, but it is the author's belief that the types shown in Figure 2 cover the requirements of the majority of turbomachinery applications. Reference [2] can be used to evaluate in further detail some of the other factors which have to be taken into account such as load direction, acceptance of rotating load, relative cost and size.

Before considering in more detail some factors which may affect the choice of bearing type, it is proposed below to study the basic steps which have to be executed in order to establish the viability of a journal bearing design.

This study is for a two axial groove, cylindrical bore bearing with steady load and is a condensation of the design procedure described in full in [3]. Computer solutions are available for this and most other bearing profiles, but it is suggested that it is useful to understand the operation of a manual method such as is described below, as it emphasizes the limits of acceptability which have to be included in all solutions. It is unfortunately true that many rigorous and excellent solutions of fluid film bearing problems are in the final analysis dependent on somewhat arbitrary limits of acceptable film thickness and bearing temperature.

Journal Bearing Design Procedure

The machine designer is faced with the problem of providing a bearing capable of operating satisfactorily within given

BEARING TYPE	LOAD CAPACITY	SUITABLE DIRECTION OF ROTATION	RESISTANCE TO HALFSPEED WHIRL	STIFFNESS AND DAMPING
CYLINDRICAL BORE	GOOD	→ ←	WORST	MODERATE
CYLINDRICAL BORE WITH DAMMED GROOVE	GOOD	→		MODERATE
LEMON BORE	GOOD	→ ←		MODERATE
THREE LOBE	MODERATE	→ ←	INCREASING	GOOD
OFFSET HALVES	GOOD	→		EXCELLENT
TILTING PAD	MODERATE	→ ←	BEST	GOOD

Figure 2. Comparison of Journal Bearing Types.

load, speed and size constraints. He will wish to establish the bearing length, the diametral clearance, the required oil flow and the power absorbed in the bearing. He will also need to establish the suitability of the oil grade he is proposing to use and whether the supply temperature and pressure are acceptable.

A fluid film bearing under running conditions can only be damaged (or fail) from the point of view of fluid film conditions from two causes:

1. Too thin a film which leads to a breakdown of the film and damage to one or both bearing surfaces.
2. Too hot an oil film which causes damage to one or both bearing surfaces.

These limits in terms of bearing load and speed are shown diagrammatically in Figure 3; a unique envelope curve of this type can be drawn for any size of journal bearing operating with a given grade of oil at a given entry temperature to the bearing. Therefore the designer's problem is to establish whether the chosen bearing size will fall inside or outside the limiting envelope for the particular operating conditions envisaged.

If it is found that the initial choice is unsatisfactory, changes can be made to such parameters as bearing length, bearing diameter, diametral clearance and oil grade until a satisfactory choice is achieved.

The design procedure is as follows:

1. Fix values of load, shaft diameter, oil grade, shaft speed and bearing length from the overall design requirements of the application.
2. Select appropriate minimum diametral clearance from Figure 4.
3. Check from Figure 5 (a) and Figure 5 (c) that the design speed/load point gives an adequate minimum oil film.
4. Check from Figure 5 (b) and Figure 5 (c) that the design speed/load point gives an acceptable maximum film temperature. (The charts assume a white metal bearing surface and a limiting temperature of 120°C (248°F).)
5. If necessary, repeat steps 3 and 4 with different values of oil grade and bearing length to obtain an acceptable

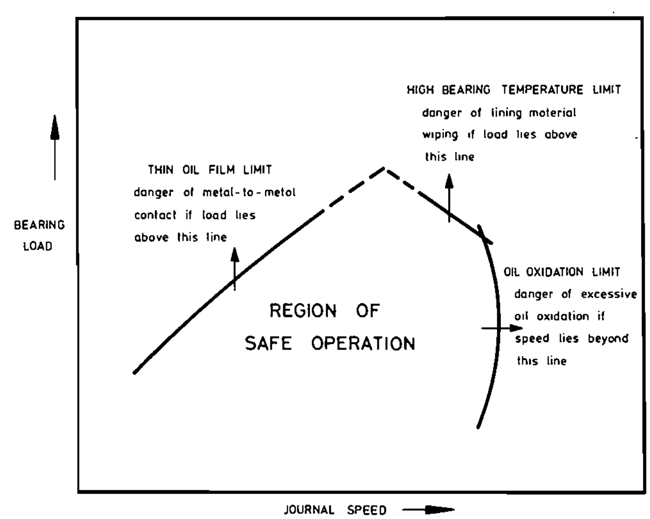


Figure 3. Limits of Safe Operation for Hydrodynamic Journal Bearings.

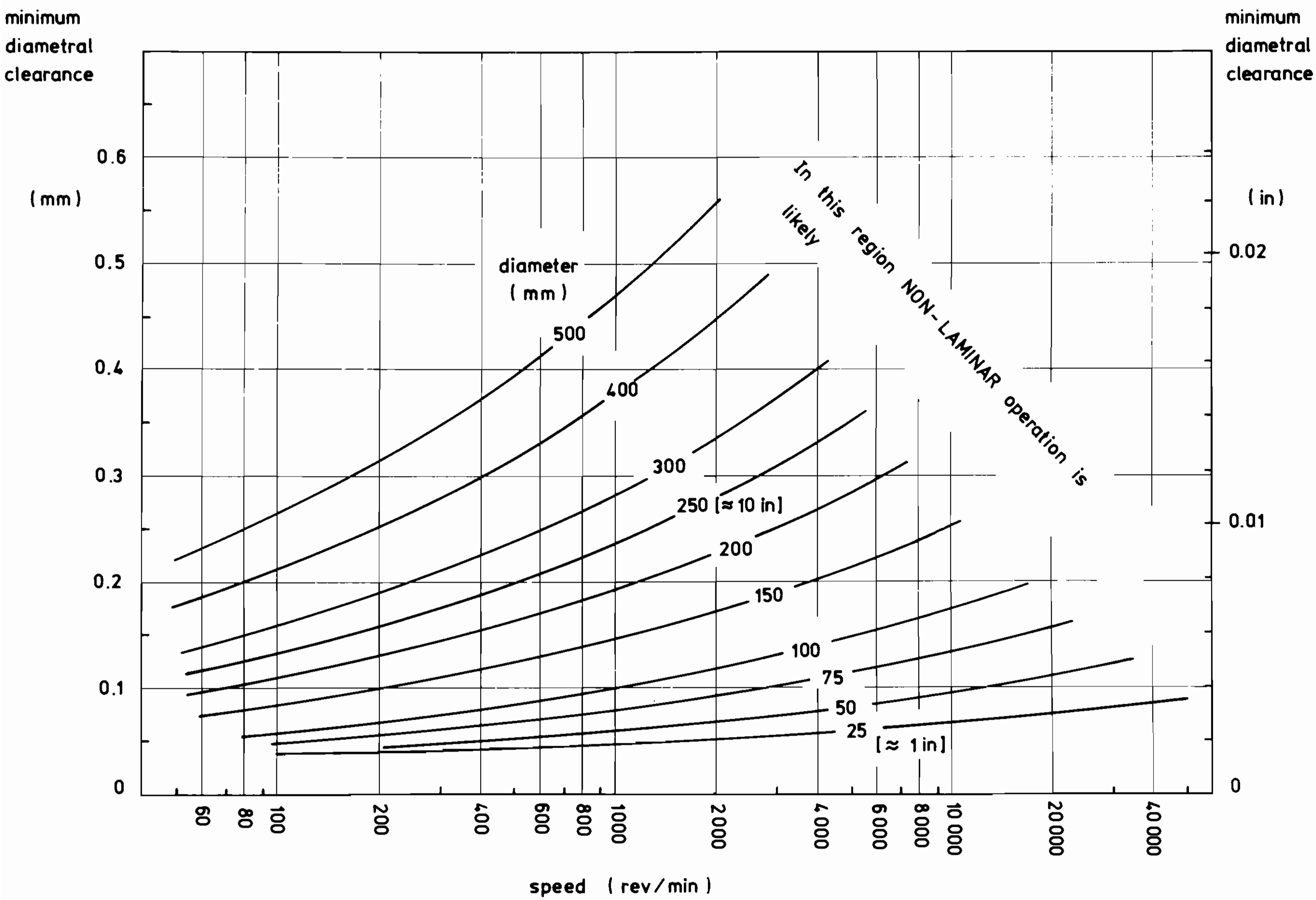


Figure 4. Recommended Minimum Clearances for Journal Bearings (at Extreme of Manufacturing Tolerance Range).

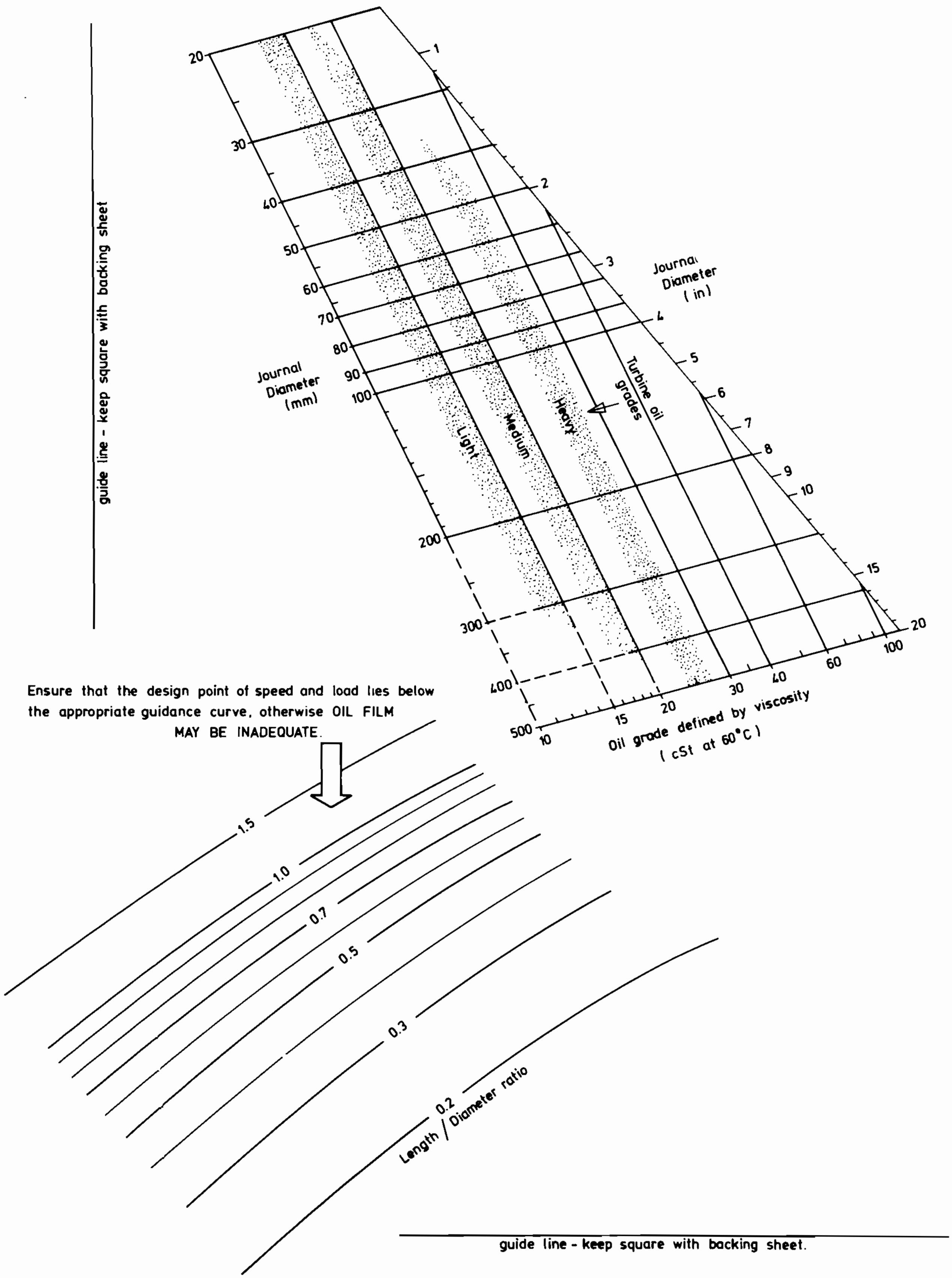


Figure 5 (a). Journal Bearing Load Capacity Slide Chart: Small Oil Film Limit (Transparency).

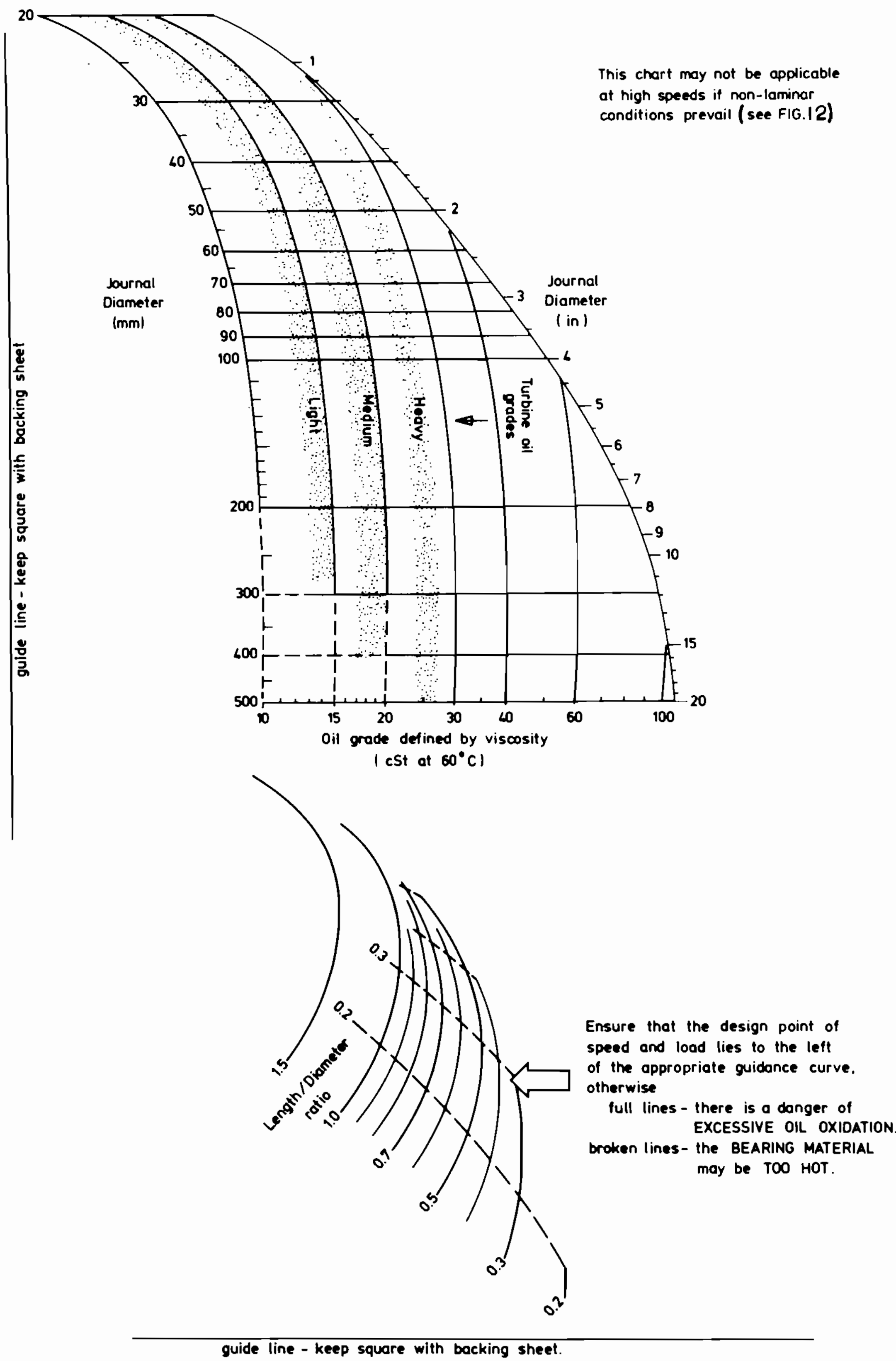


Figure 5 (b). Journal Bearing Load Capacity Slide Chart: High Temperature Limits (Transparency).

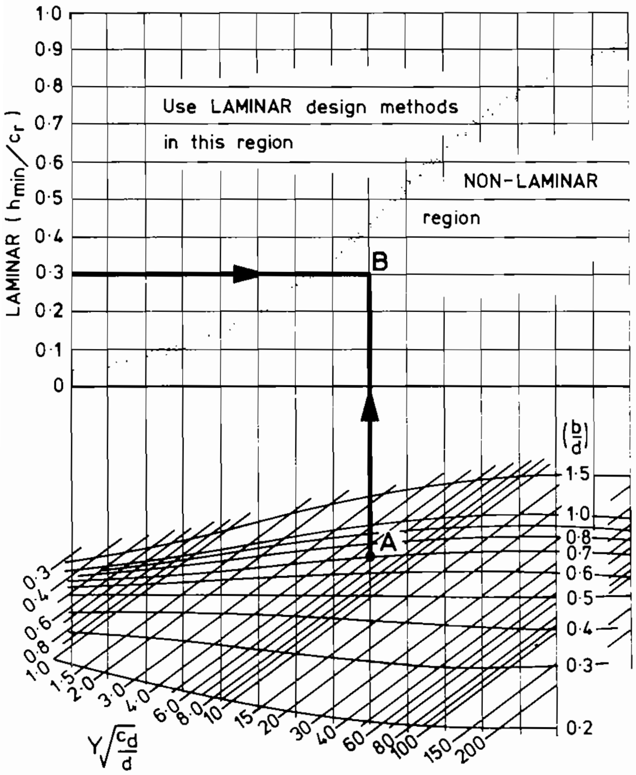


Figure 12. Prediction of Non-Laminar Conditions in Journal Bearing.

3) Misalignment

All bearings suffer to a greater or lesser extent from misalignment of the shaft relative to the bearing surface. The design procedures described so far assume that alignment is perfect but it is common practice to derate bearings to allow for possible misalignment. It is possible to quantify the derating effect of misalignment by adjusting the minimum film thickness, as shown in Figure 13, reproduced from [3].

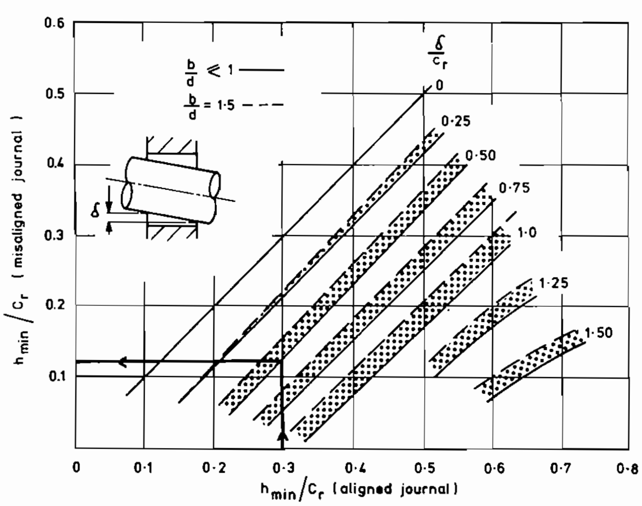


Figure 13. Derating Effect of Misalignment on Journal Bearing Film Thickness.

From Figure 13 the h_{min} for the aligned case obtained from Figure 6 may be reduced for the misaligned condition and checked against the allowable minimum film in Figure 4. As an approximation, the power loss, oil flow and maximum temperature for the misaligned case may be calculated by using the charts for the aligned case with the h_{min}/C_r value for the aligned case.

4) Oil Supply Temperature and Pressure.

In a bearing which is suffering from too high a temperature, it is not uncommon to try and reduce its temperature either by reducing the oil inlet temperature or increasing the supply pressure. The effect of these two changes cannot be estimated from the procedures described, and therefore, it is appropriate to note that they usually have little effect on the maximum temperature in the bearing [5]. In these circumstances it is usually much more beneficial to consider a change in oil grade or bearing geometry; the effect of these changes can be estimated from the design procedures.

THRUST BEARINGS

Figure 14 shows a comparison between the principal types of fluid film thrust bearings in terms of their technical characteristics.

Plain grooved thrust washers are rarely used when any continuous load has to be taken and their use tends to be confined to cases where the thrust load is of very short duration (seconds only) or possibly occurs at standstill or low speed only. Occasionally this type of bearing is used for light loads (less than 3.5 bar (50 lb/in²)), and in these circumstances the operation is probably hydrodynamic due to small distortions present in the nominally flat bearing surface [5].

When significant continuous loads have to be taken on a thrust washer, it is necessary to machine into the bearing surface a profile in order to generate a fluid film. This profile can be either a tapered wedge or occasionally a small step (Raleigh type); see Figure 15.

The former is more common in practice due to the small step height the latter requires for optimum performance. As has been shown in [5], from a theoretical viewpoint the taper land bearing has good load capacity but has had relatively limited use in practice due to manufacturing and operational disadvantages. Taper land bearings tend to be used at specific pressures below about 15 bar (210 lb/in²) and are particularly suitable for small,

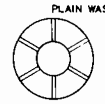
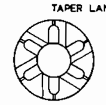
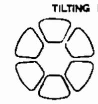
BEARING TYPE	LOAD CAPACITY	SUITABLE DIRECTION OF ROTATION	TOLERANCE OF CHANGING LOAD / SPEED	TOLERANCE OF MISALIGNMENT	SPACE REQUIRMENT
PLAIN WASHER 	POOR		GOOD	MODERATE	COMPACT
TAPER LAND  BIDIRECTIONAL	MODERATE		POOR	POOR	COMPACT
UNIDIRECTIONAL	GOOD		POOR	POOR	COMPACT
TILTING PAD  BIDIRECTIONAL	GOOD		GOOD	GOOD	GREATER
UNIDIRECTIONAL	GOOD		GOOD	GOOD	GREATER

Figure 14. Comparison of Thrust Bearing Types.

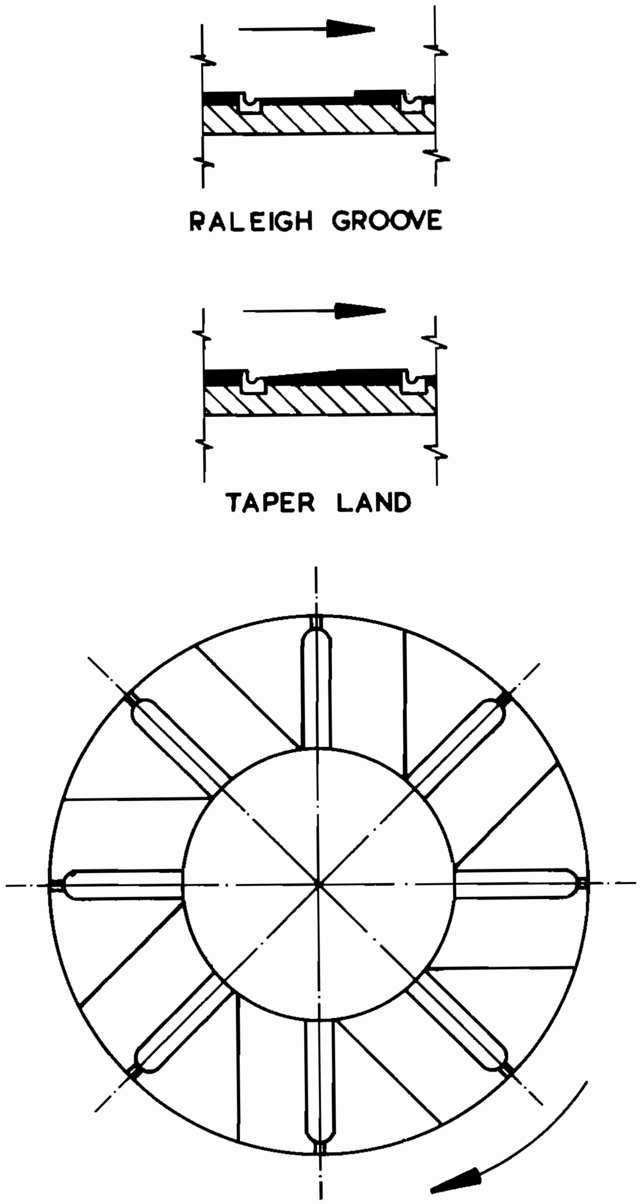


Figure 15. Fixed Land Type Thrust Bearings.

very high speed machines where large film thicknesses are generated.

The most popular type of thrust bearing for taking turbomachinery thrust loads is the tilting pad type due to its versatility in adjusting to various load/speed conditions and its better ability to cope with operating conditions such as poor alignment and dirt. In the rest of this section, therefore, the factors affecting the selection and design of this type of bearing will be considered.

Tilting Pad Thrust Bearing Design Procedure

The machine designer is faced with a similar problem with a thrust bearing as with a journal bearing. Again the two failure limits of minimum film thickness and maximum allowable surface temperature apply, and a modified form of the journal bearing envelope (Figure 3) can be drawn for the thrust bearing, as shown in Figure 16.

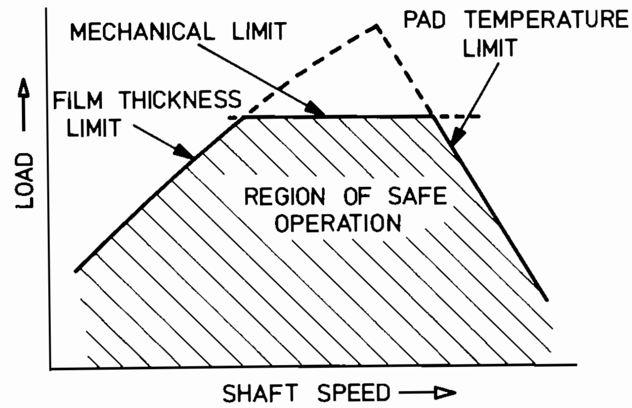


Figure 16. Limits of Safe Operation for Tilting Pad Thrust Bearings.

It will be noticed in Figure 16 that an additional limit has been imposed — a specific load or mechanical limit. It is necessary in a practical design to limit the load on the thrust pad pivot and prevent excessive deflections in the pad. Also the limit to high speed operation is more likely to be the thrust collar bursting stress rather than an oil oxidation limit.

In [6] a procedure is described for establishing the viability of a tilting pad bearing of a given size for given operating conditions and estimating the power absorbed and oil requirements.

This procedure is strictly applicable only to one particular geometry of bearing, as described in [6] but can be used for other thrust bearing geometries with a fair degree of accuracy. The basic geometry of the type of bearing considered is shown in Figure 17 and it will be seen that the pads are of approximately square shape with centerline pivots.

The design steps using this procedure are as follows:

1. Fix values of load, shaft diameter, oil grade and shaft speed from the overall design requirements of the application.
2. Select suitable size thrust bearing from Figure 18 in terms of the load to be carried and the shaft diameter. Note that it is necessary to select a whole number of pads; and to simplify subsequent steps, the number should correspond to one of those shown in the figure. Allowance has been made for a clearance between the shaft and the thrust pad inside diameter.
3. Use Figure 19 (a) and (b) as shown in Figure 20 to check that the design speed/load point falls below the limit lines for the appropriate number of pads. The upper mechanical limit line is formed from the projection of the oil grade/pad size line, as shown in Figure 20.
4. If necessary, step 3 may be repeated using a different oil grade, or steps 2 and 3 using a different bearing size in order to obtain an acceptable design.

At this stage the viability of the bearing size has been established and the next stage is to determine the power loss and oil requirement of the bearing as follows:

5. Establish the specific load on the bearing surface.

$$\text{Specific load (MN/m}^2\text{)} = \frac{\text{Bearing Load (N)}}{(\text{Pad Size (mm)})^2 \times \text{No. of pads}}$$

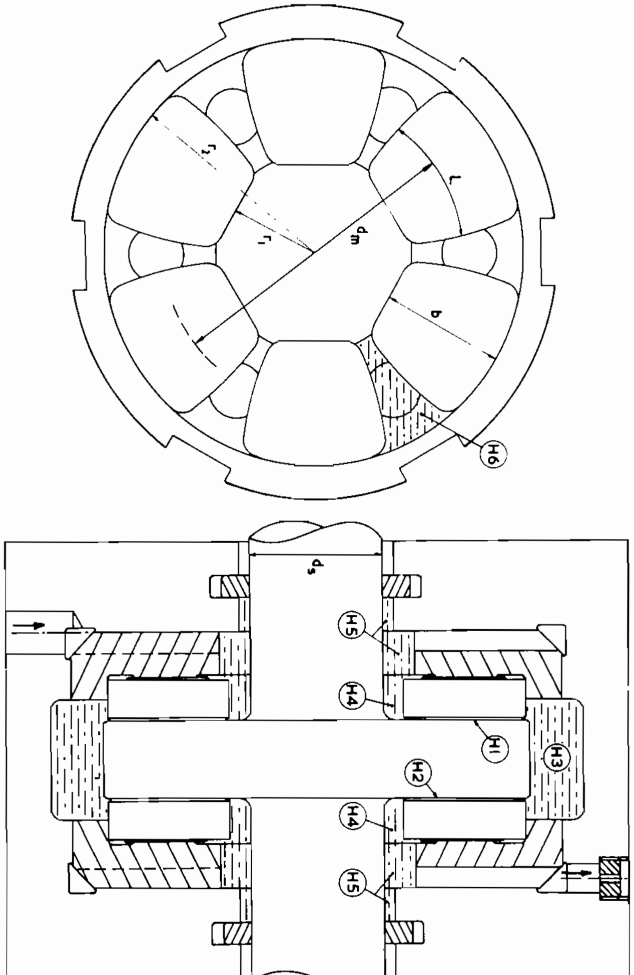


Figure 17. Tilting Pad Thrust Bearing Showing Sources of Power Loss (for Power Loss Relationships see Figure 25).

6. Use Figure 21 (a) and (b) as shown in Figure 22 to determine the value of the power absorbed in the double bearing shown in Figure 17 for the bearing size, specific load, oil grade and shaft speed being considered.
7. Calculate the oil flow requirement from the power loss as follows:

   Oil flow (1/sec) = 0.035 × power loss (kw)

   Oil flow (Imp.gal/min) = 0.35 × power loss (hp)

   Oil flow (US gal/min) = 0.42 × power loss (hp)

In using this procedure, the following points should be noted.

a. The oil inlet temperature is fixed at 50°C (122°F) and the temperature rise through the housing at 17°C (30°F).
b. In using Figures 19 and 21, it is assumed that the (a) sheets are available in transparent form for use with the respective (b) backing sheets.
c. The power loss calculated is for a double thrust bearing installed as shown in Figure 17; allowance is made for turbulent conditions in each part of the bearing as appropriate.

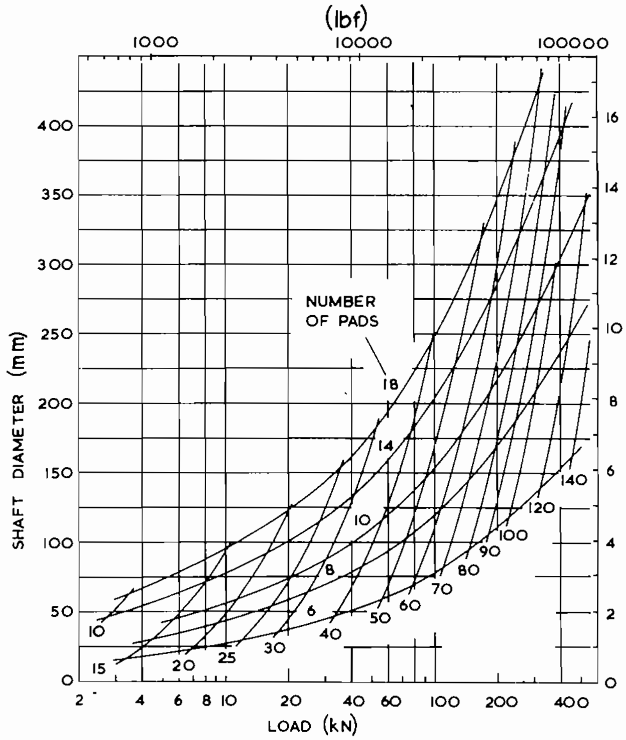


Figure 18. Selection of Thrust Pad Size and Number for Use in Figures 19 & 21.

d. A full explanation of the procedure and the assumptions made are given in [6].

Factors Affecting Bearing Selection

Under this heading, comments are given on some of the more important factors which may affect the choice of a tilting pad thrust bearing once the decision has been made to use a bearing of this type and the size has been chosen as outlined above. As will be seen, these factors all tend to modify the basic hydrodynamic limits (film thickness and pad temperature) and, therefore, in theory they should be taken into account in the initial bearing size selection. In practice it is more usual to select a bearing size, and then consider whether the selection needs modifying to take account of any of the special factors indicated below.

1) Misalignment

The ratings for thrust bearings obtained from the design procedure above make allowances for manufacturing tolerances between thrust pad heights and for some degree of misalignment between thrust collar face and the bearing face.

The basis of these allowances is described in [6], and in [7] bearing performance data is given for a thrust bearing running under conditions of misalignment. In addition there is much practical experience available which confirms that a non-equalized thrust bearing of the type considered can run satisfactorily at these ratings with design misalignments of the order of 0.0005 to 0.001 slope across the thrust collar. When load is applied to the bearing these misalignments tend to diminish due to component deflections [7].

When larger misalignments are anticipated in a machine, a device for aligning the thrust face to the misaligned collar is

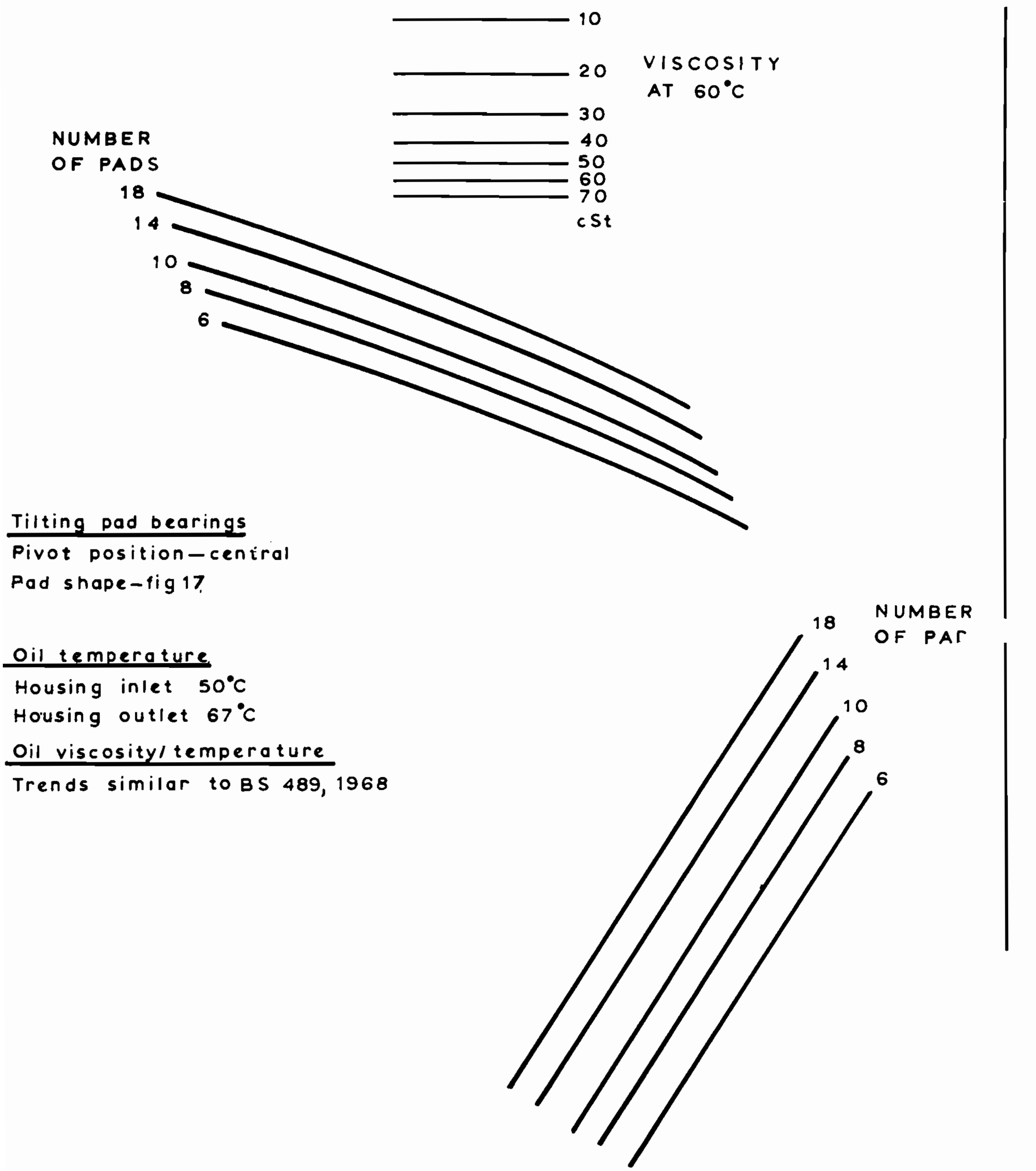


Figure 19 (a). Thrust Bearing Load Capacity Slide Chart (Transparency).

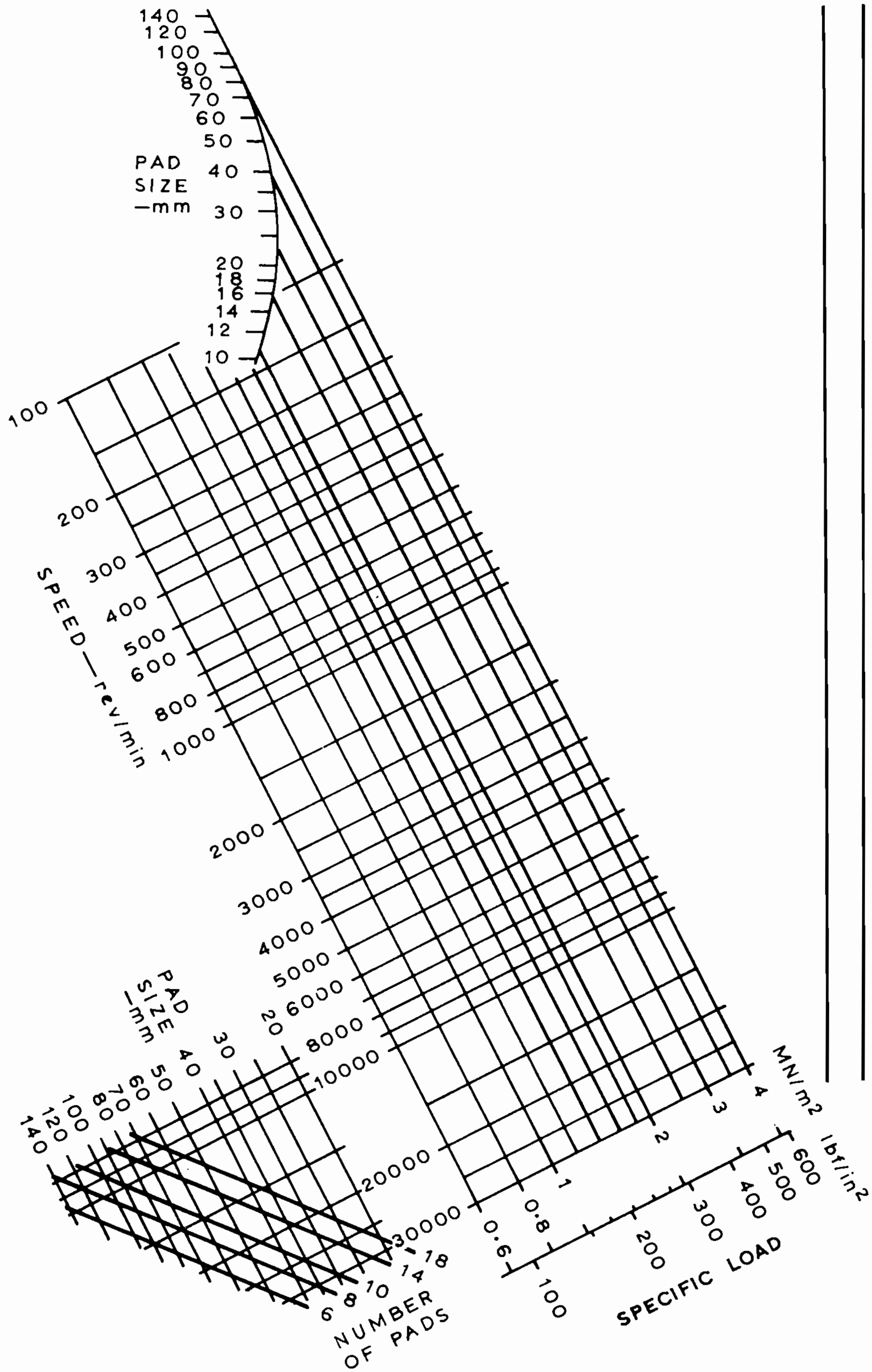


Figure 19 (b). Thrust Bearing Load Capacity Slide Chart (Backing Sheet).

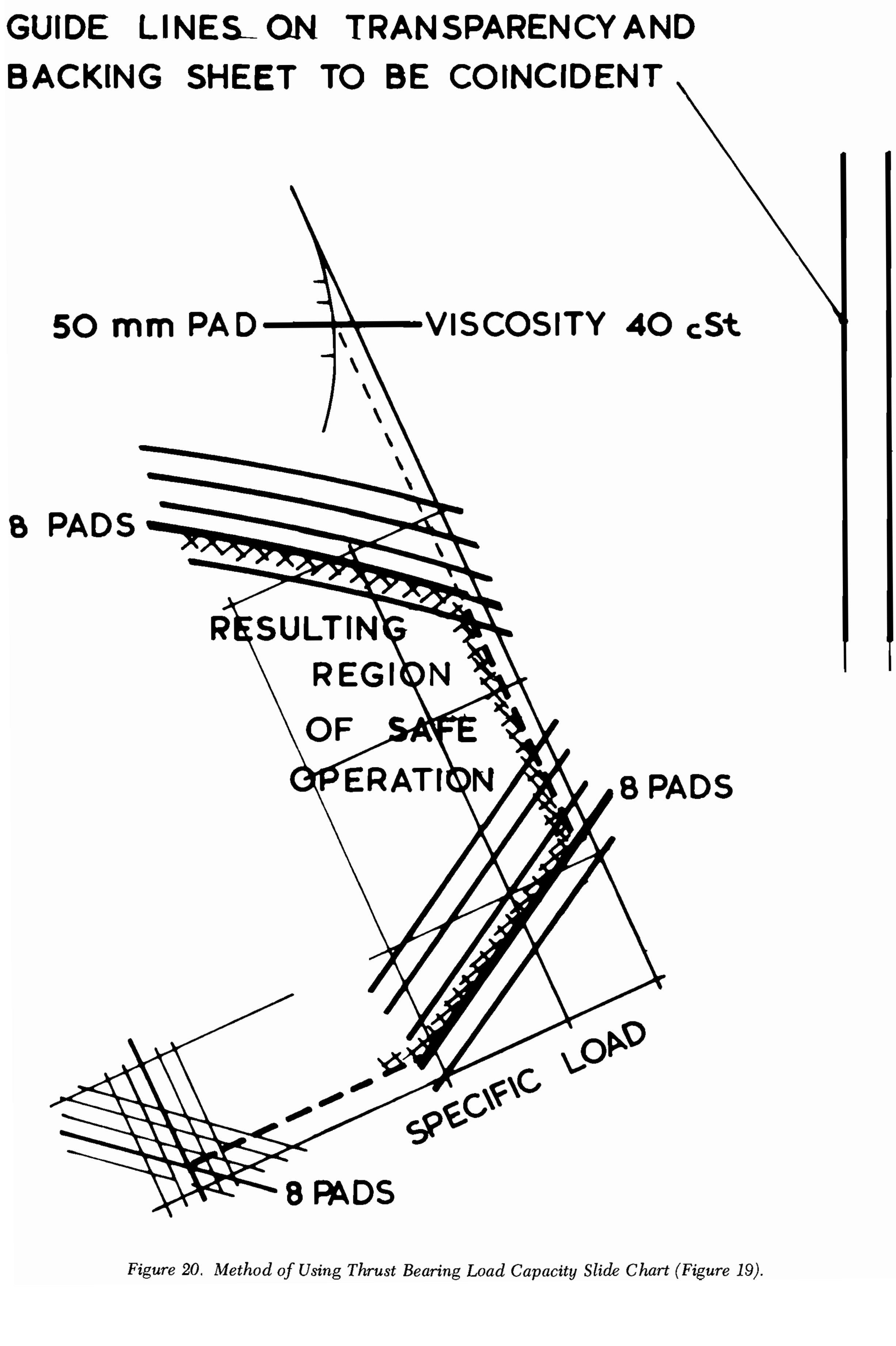


Figure 20. Method of Using Thrust Bearing Load Capacity Slide Chart (Figure 19).

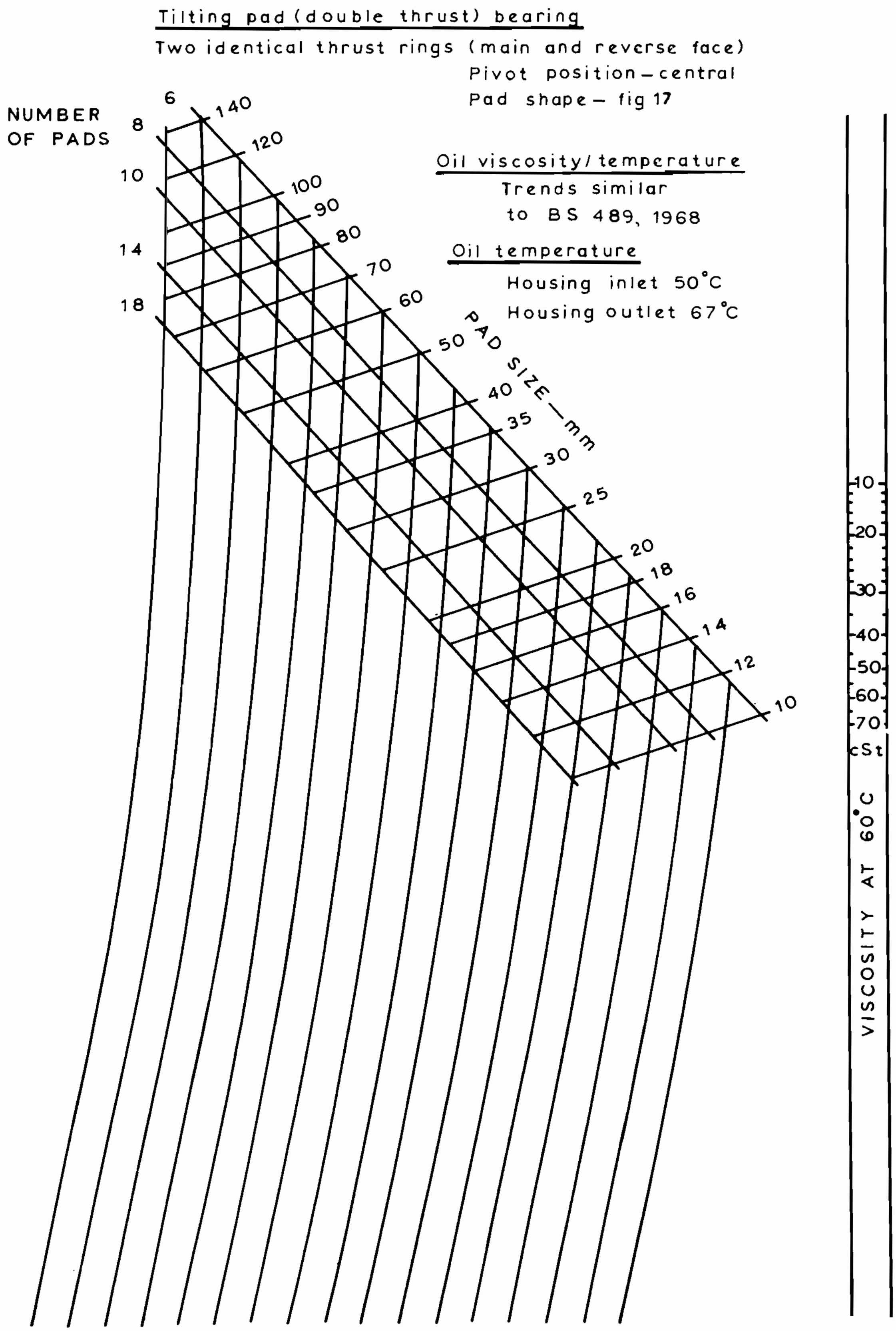


Figure 21 (a). Double Thrust Bearing Power Loss Slide Chart (Transparency).

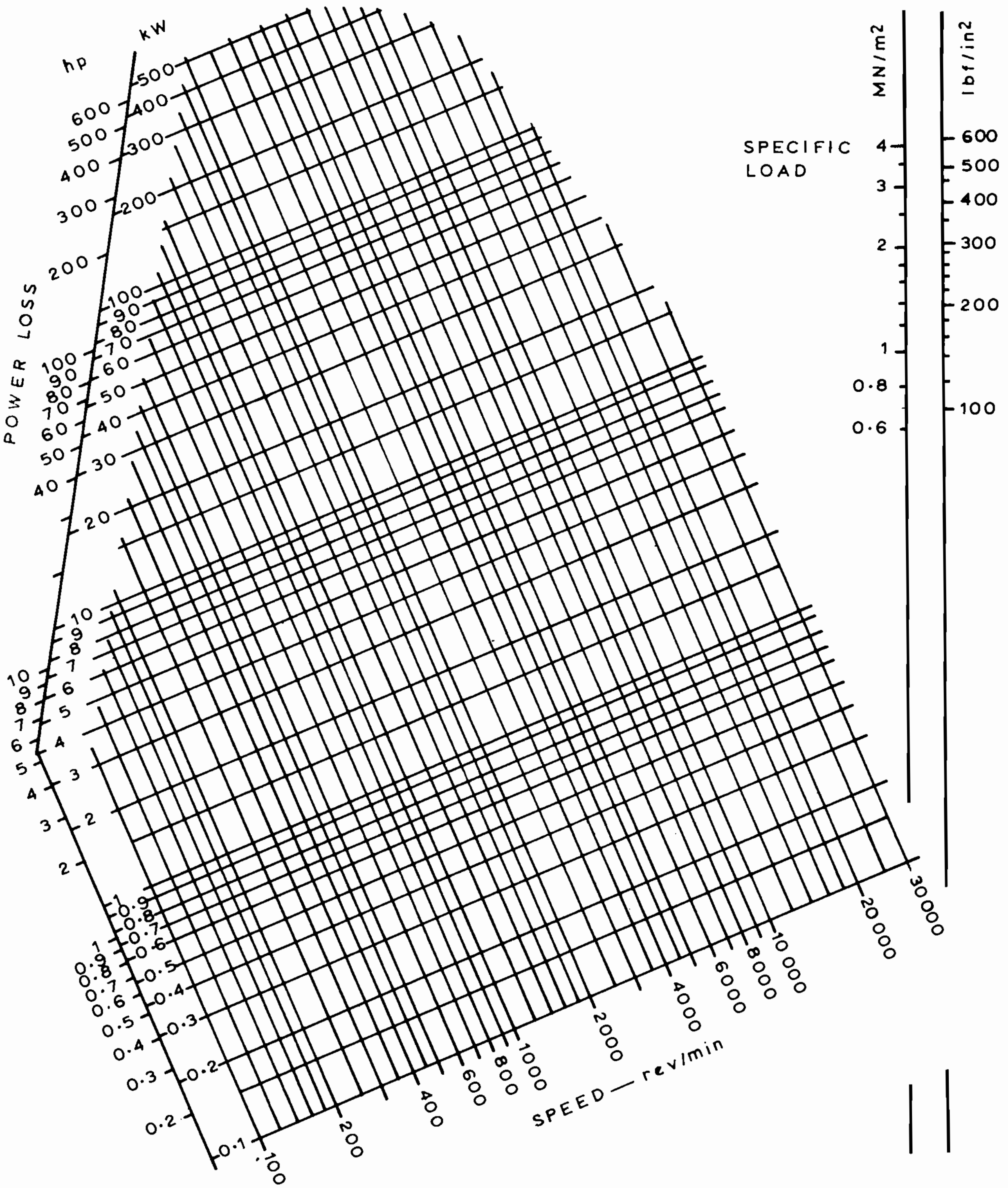


Figure 21 (b). Double Thrust Bearing Power Loss Slide Chart (Backing Sheet).

often used: three methods of achieving this are shown in Figure 23. All of these devices have some disadvantages and none compensate completely for misalignment. The operating of the equalizing level type, which has wide usage, has been reported in [8] and [9]; the spring pack system has similar characteristics to the lever, while the spherical seat can only be used for initial alignment at machine erection after which it should be locked solid to prevent fretting at the interface.

Before leaving the subject of alignment, it should be emphasized that the above comments apply to misalignment in a static plane and do not apply to a thrust collar with misalignment in a moving plane (swash plate action). For this latter type of misalignment, total indicator readings, measured at the collar outside diameter, should not exceed 0.00004 mm/mm (in/in) of collar diameter whether or not an alignment device is used with the bearing.

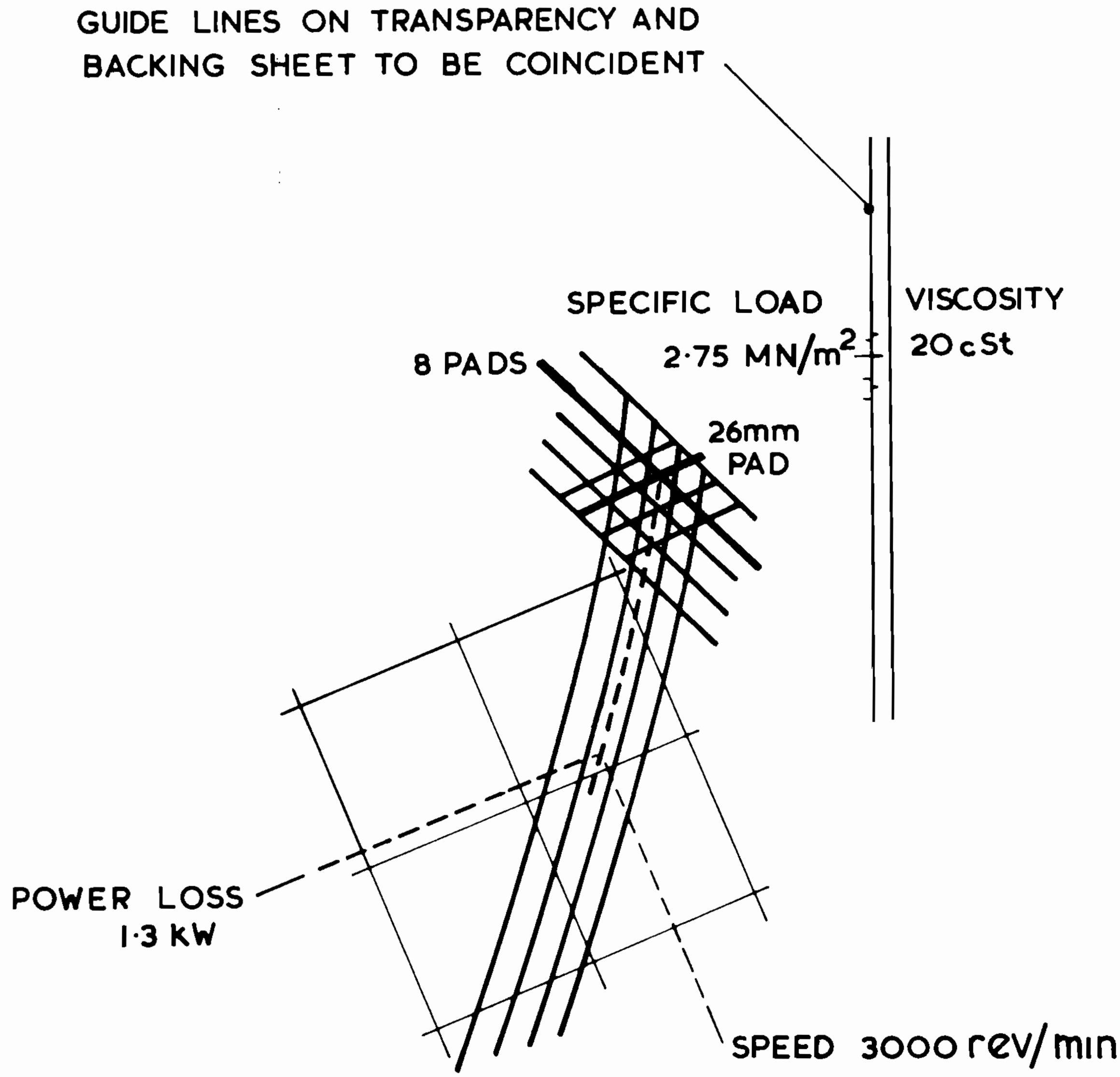


Figure 22. Method of Using Thrust Bearing Power Loss Slide Chart (Figure 21).

2) Lubrication

The general subject of lubrication of bearings is dealt with later, but it is appropriate to consider now some factors which are unique to tilting pad thrust bearings. The three main methods used to lubricate tilting pad thrust bearings are those shown in Figure 17 and Figure 24 and these have been compared in [10] and [11]. The pressurized casing method (Figure 17) is normally satisfactory for low and medium speed applications but at higher speeds, power loss and hence oil requirement increase steeply due to a progressive change from laminar to turbulent conditions in various parts of the bearing. This change commences at a velocity, measured at the mean diameter of the thrust pads, of between 50 - 70 m/s (164-230ft/sec); and beyond this point, both the throttled inlet system and various forms of directed lubrication have been used. These systems seek to avoid the churning losses inherent in the pressurized casing system and it will be seen from Figure 25 that elimination of these losses can reduce power losses by 50% or more. The throttled inlet system, Figure 24 (a), can only achieve significant power loss savings at the expense of higher bearing temperatures [10] and [11], and recent work has been concentrated on directed lubrication systems where the oil is directed in a controlled way to the inlet edges of the thrust pads. When this is done correctly, not only is power loss saved, but bearing surface temperature is reduced [7]. Figure 24 (b) shows the oil flow system in a directed lubrication bearing, while Figure 26 shows the appearance of a bearing of this type.

3) Pad Pivot Position

Most tilting pad thrust bearings in operation today use central pivots, either of the line type or point type, due to the practical advantages of having a bearing suitable for either direction of rotation. It has been shown in [9] that at higher sliding speeds, however, significant advantage in terms of bearing surface temperature can be achieved by using offset pivots; this is illustrated in Figure 27.

4) Thrust Pad Backing Material

Many years ago it was common to manufacture thrust pads from bronze with white metal surfaces but economic, as well as technical factors, led most manufacturers to standardize on steel

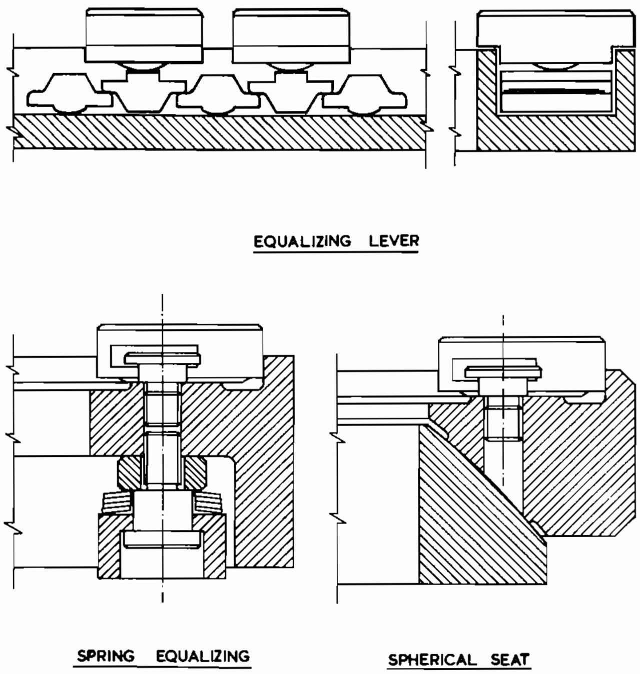


Figure 23. Misalignment Compensation Devices for Tilting Pad Thrust Bearings.

backed pads. Recently, however, it has been shown that using a copper backed pad can significantly reduce bearing surface temperature [11]. These pads have normal white metal surfaces with a 98 or 99% copper alloy backing; the alloying element (which is commonly chromium) is added to give better mechanical strength than pure copper. The benefits of using this material with a thrust bearing of the type shown in Figure 17 are shown in Figure 28.

The main disadvantage of this material is that of cost and this has tended to restrict its use to specialized applications where cost is not of prime importance.

BEARING SURFACE MATERIALS

The vast majority of turbomachinery thrust and journal bearings have a bearing surface lined with white metal and it is only recently that any significant number of bearings have been used with other materials. Alternative surface materials always involve sacrifice of one or more beneficial properties of white metal which may be summarized as: (1) easy to apply and bond to the bearing surface, (2) sufficiently soft to allow for embedability of small dirt particles, (3) good properties under boundary lubrication conditions, (4) relatively high melting point. Figure 29 shows the properties of a number of bearing surface materials of which the 40% tin, 60% aluminium alloy is of interest to turbomachinery designers who require a superior material to white metal. This alloy offers an attractive alternative to white metal as it is only marginally inferior in embedability and boundary

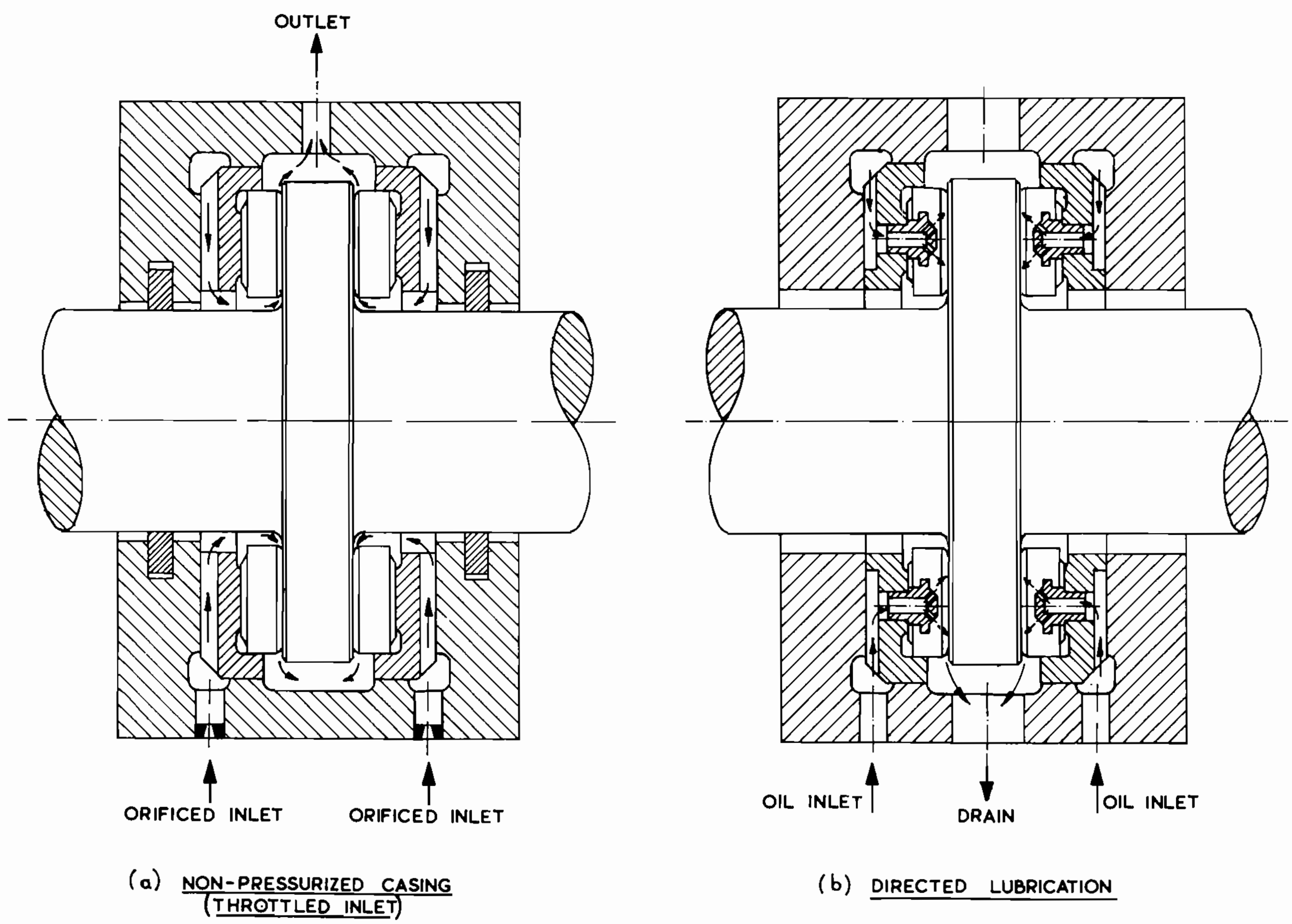


Figure 24. Lubrication Systems for Tilting Pad Thrust Bearings.

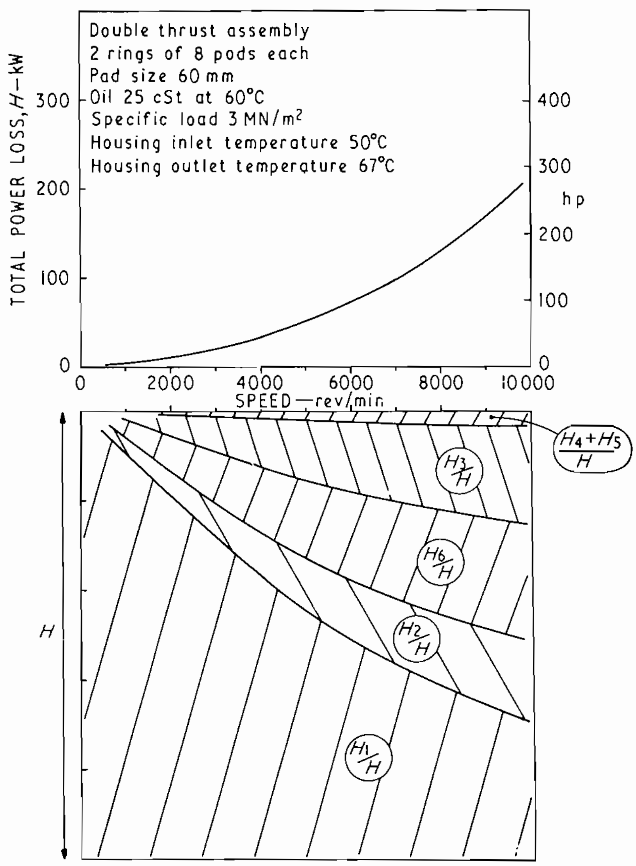


Figure 25. *Distribution of Power Loss in a Double Thrust Bearing.*

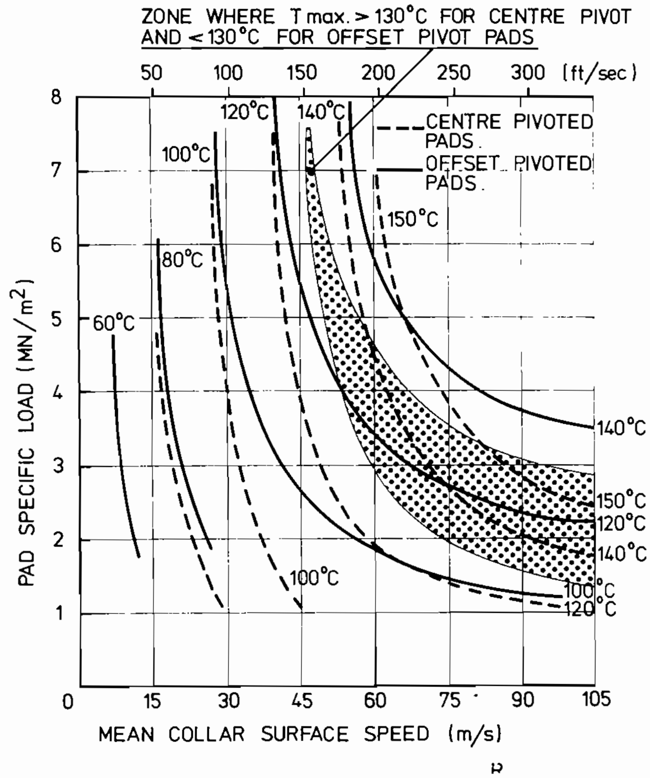


Figure 27. Comparison of Offset & Center Pivoted Thrust Pad Temperatures Versus Load and Speed. Pad Size: 40mm (1.59 in). Area 1635mm²(2.54 in²). Oil Viscosity 41 cSt at 50°C (340 SSU at 100°F).

lubrication properties, but can tolerate a significantly higher surface temperature, and this enables higher loads or higher speeds to be taken on a given bearing size [9]. The principal disadvantage is that its manufacture is more specialized as it cannot be cast directly onto the backing material.

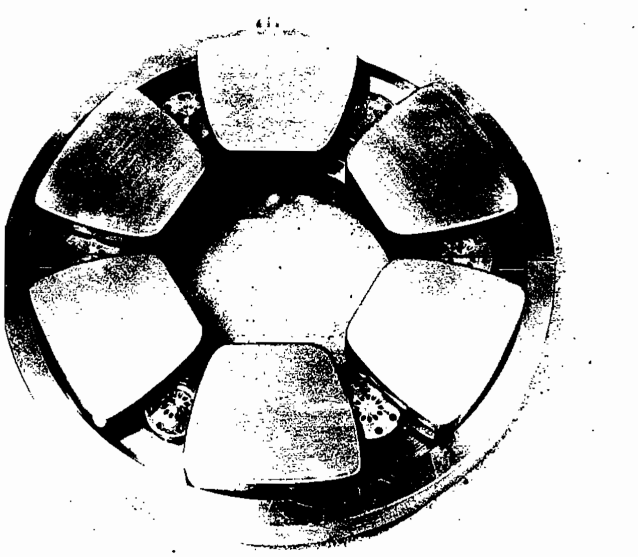


Figure 26. Tilting Pad Thrust Bearing Fitted with Directed Lubrication.

LUBRICANTS OTHER THAN OIL

The comments given so far in this paper are formulated around oil as the medium in the fluid film though basic fluid film theory holds true for any Newtonian fluid. After oil the most common lubricant used for a fluid film bearing is water, with or without additives.

In principle, a fluid film bearing with water as the lubricant is designed in the same manner as described earlier for oil lubricated bearings; the much lower viscosity of the lubricant however leads to physically large bearings, if similar criteria of film thickness are used.

It is of interest to note however that this problem disappears at higher speeds where temperature is the limit as the lower viscosity and better conductivity of water is now a positive benefit. Figure 30 illustrates this point comparing the load capacity envelope for the same size of thrust bearing with oil and water. This advantage however has been unexploited commercially and most examples of successful water lubricated fluid film bearings in industrial applications are confined to low or medium speed applications (i.e., below 30 m/s (98ft/sec) mean sliding velocity).

Providing clean water is used, acceptable life is obtained with most water lubricated bearings at film thicknesses which would not be acceptable in oil lubricated bearings. This allows the bearing dimensions to be minimized but does require a bearing surface material which is tolerant of boundary lubrica-

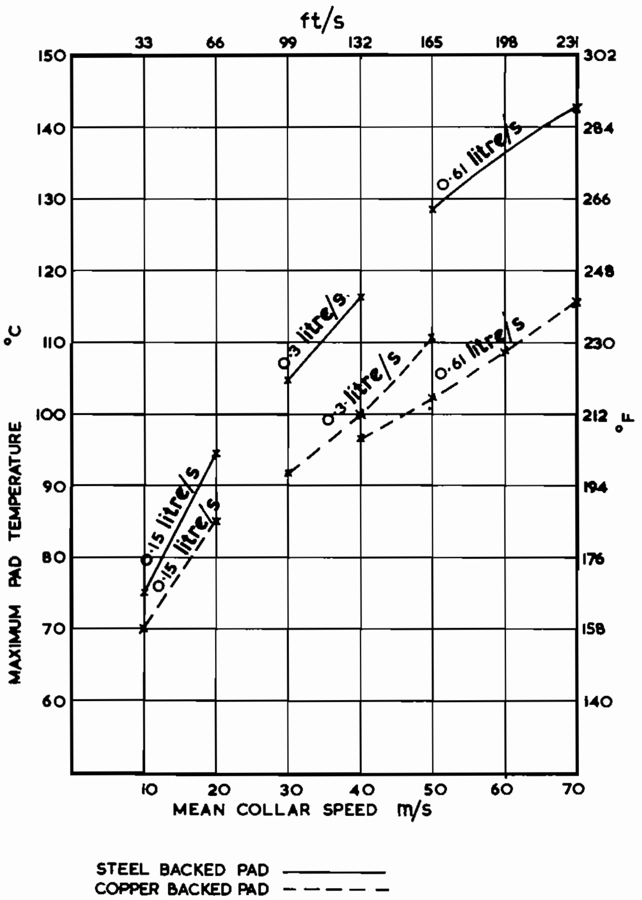


Figure 28. Comparison of Thrust Pad Surface Temperatures with Steel and Copper Backed Thrust Pads. 8 Pad Thrust Bearing 124mm (5 in) Outside Dia. Specific Load 4 MN/m², (571 lb/in²). Oil Flows as Stated. Oil Viscosity 42 cSt at 50°C (340 SSU at 100°F).

tion conditions in the presence of water. While bronze and white metal can be used as surface materials for water lubricated bearings, better life is obtained if materials such as graphite filled epoxys or impregnated bronze sinters are used. With these materials, satisfactory life can be obtained at specific pressures of 1.5 MN/m^2 (214 lb/in^2) at sliding speeds of 15 m/s (50 ft/sec); under test conditions at the same speed, negligible wear has been recorded on thrust bearings operating for some hundreds of hours at 5 MN/m^2 (714 lb/in^2). The main limitation to the more widespread use of water lubricated bearings is undoubtedly the difficulty of providing really clean lubricant water: 5 micron or better filtration is required for many applications. If clean water cannot be provided, the best solution is to use extremely hard bearing surface materials so that wear from dirt is minimized [12]. Again, however, this approach has not been used to any significant extent in industrial applications.

LUBRICATION OF BEARING ASSEMBLIES

When more than one plain bearing is used in an assembly, ideally each bearing should be provided with a separate oil inlet and outlet; this ensures that there is minimum interaction between each bearing.

In practice combined oil inlets and/or outlets are often used as this enables a more compact and simple assembly to be designed with little loss in bearing safety.

In Figure 31 the oil flow paths in four types of bearing assembly are shown. These designs represent the four main categories of assembly and each of these pose different problems for the designer. Forced lubrication must always be preferred strictly from the viewpoint of optimum bearing performance: a horizontal assembly of this type is shown in Figure 31 (a) while a vertical assembly is shown in Figure 31 (c). Apart from possible space constraints, the main problem with this type of assembly concerns the seals required where the shaft passes through the assembly housing; in particular the lower seal on a vertical assembly can be troublesome at higher sliding speeds.

Self-contained assemblies present greater design problems, as in addition to sealing, a pumping system has to be provided together with a cooler of sufficient capacity to deal with the heat generated by the bearings. These problems tend to limit self-contained assemblies to sliding speeds below about 30 m/s (99 ft/sec) above which force lubricated assemblies are normally used. Figure 31 (b) shows a water coil cooled horizontal self-contained assembly while Figure 31 (d) shows an air cooled vertical seal contained unit. It will be seen that the horizontal assembly poses the greater design problems, as in order to lubricate the thrust bearing, the oil has to be raised to the top of the unit, whereas in a vertical assembly it is usually possible to immerse the bearings in the oil reservoir with pumping only being required to circulate the oil. A final point to watch in self-contained assemblies is the tendency for the oil to foam at the free surfaces; this can only be avoided by careful design of the oil passages and, in particular, keeping oil discharge holes below the running oil level wherever possible.

CONCLUSION

In a paper of this length it is not possible to cover more than the salient points of plain bearing design and particularly those which are applicable to a wide range of applications. Each type of application tends to have its own special requirements, but these usually need not be considered until the basic bearing design has been established according to the methods described in this paper. For those interested in further studying of recent developments in plain bearing design practice, it is suggested that, among others, the reference lists in [3] and [13] for journal bearings and thrust bearings, respectively, would provide a good starting point.

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MATERIAL TYPE	SAE EQUIVALENT	NOMINAL COMPOSITION (%)						NOMINAL HARDNESS (HV5)	MAX DESIGN SURFACE TEMP (SEE NOTE)	
		Al	Cu	Pb	Sb	Sn	Ni		°C	°F
TIN BASE WHITEMETAL	SAE 12	—	3·5	—	—	7·5	89	31	130	266
LEAD BASE WHITEMETAL	SAE 13	—	0·5	83·5	10	6	—	16	130	266
ALUMINIUM BASE ALLOY	—	60	—	—	—	40	—	27	155	311
ALUMINIUM BASE ALLOY	SAE 770	92	1	—	—	6	1	45	155	311
COPPER BASE ALLOY	SAE 48	—	70	30	—	—	—	30-45	170+	338+
COPPER BASE ALLOY	—	—	72	26	—	2	—	44-50	170+	338+

NOTE: THIS IS THE CALCULATED TEMPERATURE MEASURED AT THE HOTTEST POINT ON THE BEARING SURFACE WHICH IS SAFE FOR CONTINUOUS OPERATION. DAMAGE TO THE BEARING SURFACE WILL NORMALLY NOT OCCUR UNTIL TEMPERATURES AT LEAST 30°C (54°F) HIGHER THAN THESE VALUES ARE REACHED. IN THE CASE OF THE COPPER BASE ALLOYS THE LIMITATION IS BREAKDOWN OF THE LUBRICANT OIL WHICH MAY OCCUR AT TEMPERATURES IN EXCESS OF 170°C (338°F) RATHER THAN A MATERIAL LIMITATION.

Figure 29. Properties of Typical Turbomachinery Bearing Alloys.

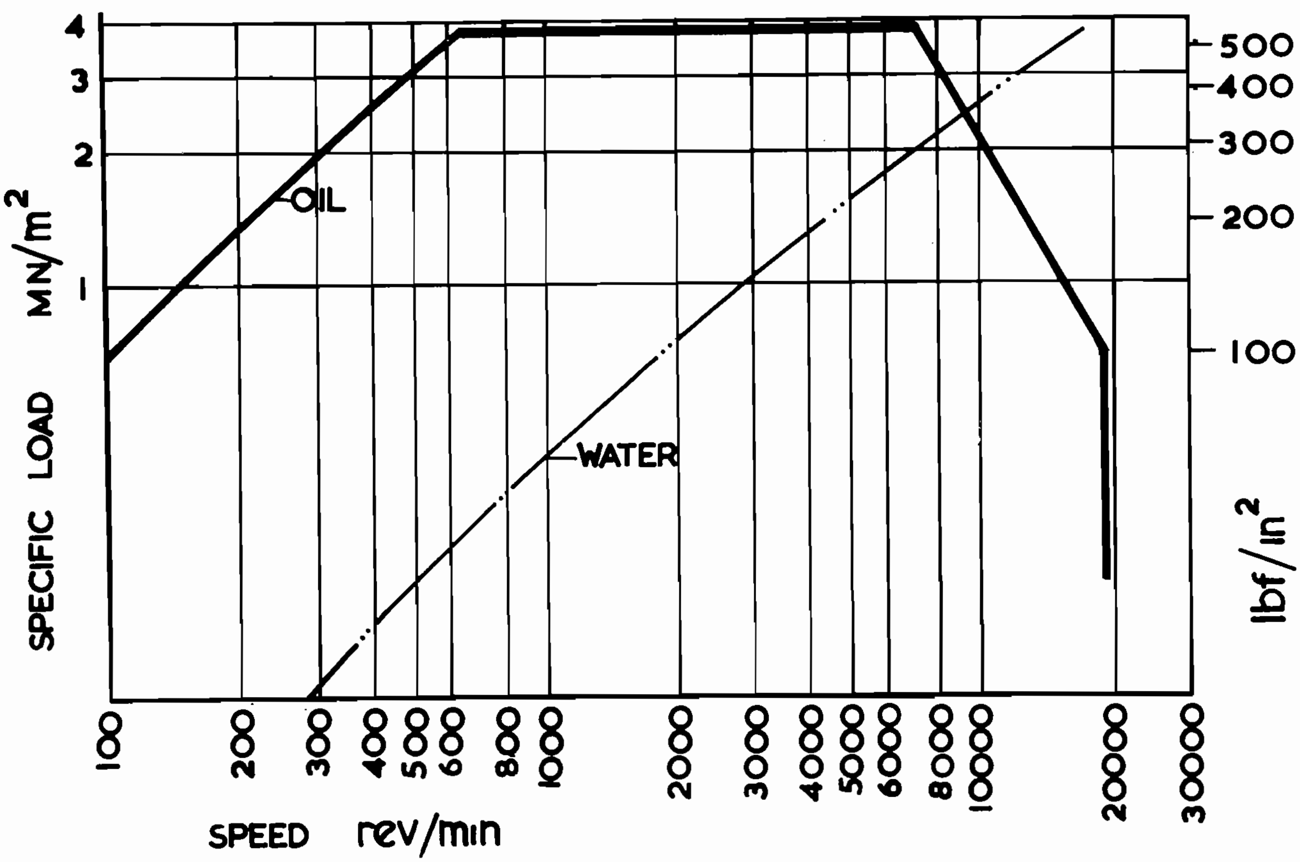

Figure 30. Comparison of Load Capacity of Tilting Pad Thrust Bearing with Oil and Water (Ref. Figure 19). 8 Pad Thrust Bearing 251mm (9.875 in) Outside Diameter. Oil Viscosity 29 cSt at 50°C (240 SSU at 100°F). Lubricant Inlet Temperature 50°C (122°F).

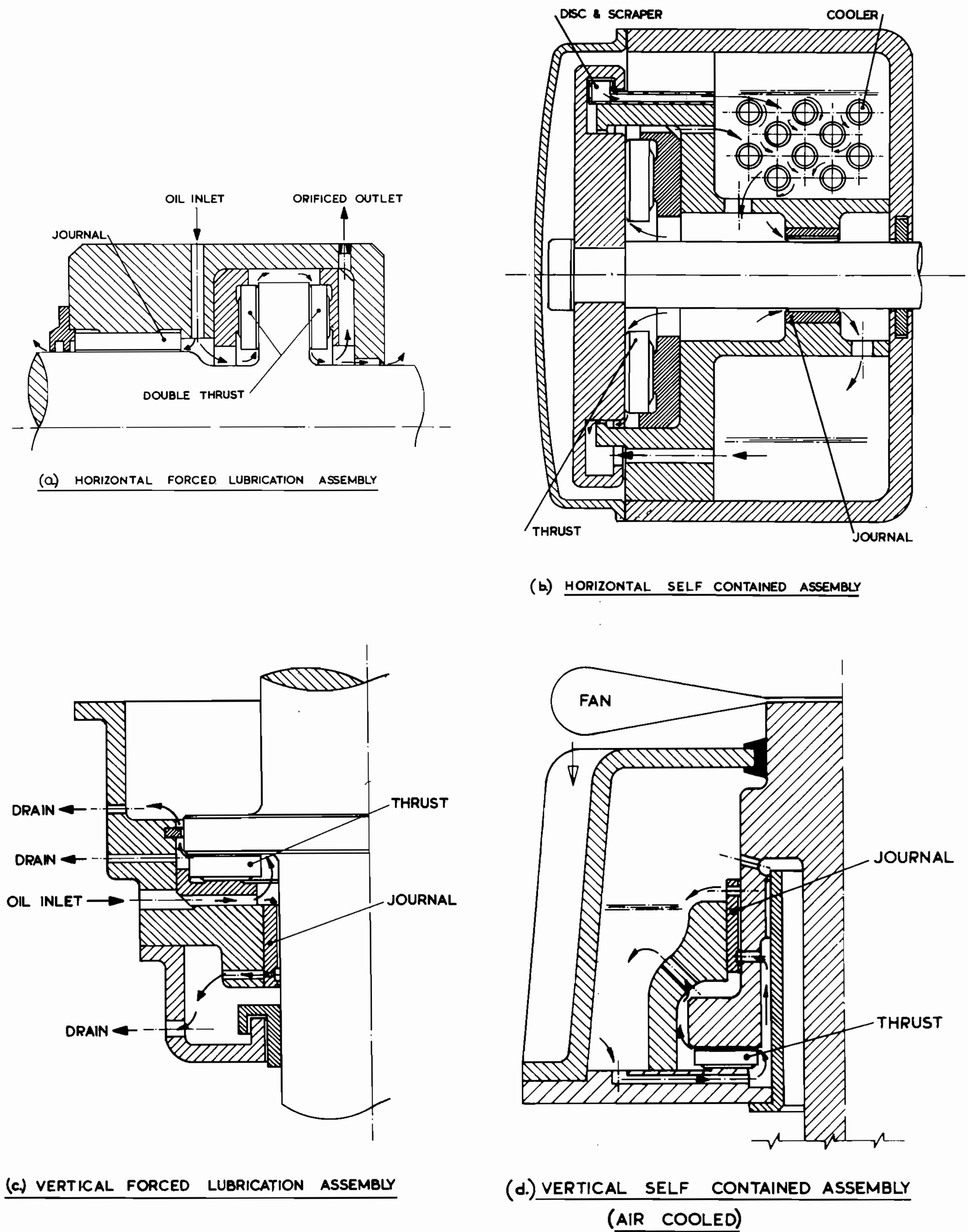

Figure 31 (a) (b) (c) (d). Typical Thrust & Journal Assemblies Showing Oil Circulation Paths.

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