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Calculation methods for steadily loaded fixed-inclined-pad thrust bearings

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(Continued on inside back cover)

CALCULATION METHODS FOR STEADILY LOADED FIXED-INCLINED-PAD THRUST BEARINGS

CONTENTS

	Page
1. NOTATION AND UNITS	1
2. INTRODUCTION	3
3. DESCRIPTION	3
4. SPECIFICATION OF DESIGN REQUIREMENTS	5
4.1 Load and Speed	5
4.2 Dimensional Constraints	5
4.3 The Lubricant	5
5. PRELIMINARY BEARING DESIGN	6
5.1 Required Total Pad Area	6
5.1.1 Running requirements and mean pad pressure	6
5.1.2 Start-up requirements and land pressure	6
5.2 Preliminary Bearing Dimensions	7
5.2.1 Full complement bearings	7
5.2.2 Partial complement bearings	8
6. PERFORMANCE ANALYSIS	8
6.1 Initial Assumptions for Performance Analysis	8
6.1.1 Effective film temperature and viscosity	8
6.1.2 Pad rise and pad proportions	9
6.2 Performance Calculations	10
6.2.1 Film thickness	10
6.2.2 Film power loss	11
6.2.3 Lubricant flow rates	12
6.2.4 Temperatures within bearing	12
7. LIMITING CRITERIA	15
7.1 Minimum Film Thickness Criterion	15
7.2 Maximum Temperature Criteria	16
8. FURTHER DESIGN CONSIDERATIONS	17
8.1 Lubricant Supply	17
8.2 Churning Losses	18
8.3 Bearings for Bi-directional Operation	19

9.	DERIVATION AND REFERENCES	20
9.1	Derivation	20
9.2	Reference	20
10.	EXAMPLES	21
10.1	Example 1	21
10.1.1	Specification of design requirements (see Section 4)	21
10.1.2	Preliminary bearing design (see Section 5)	21
10.1.3	Performance analysis (see Section 6)	22
10.1.4	Calculation of temperatures (see Section 6.2.4)	23
10.1.5	Power loss and lubricant supply (see Section 8)	24
10.2	Example 2	25
10.2.1	Specification of design requirements (see Section 4)	25
10.2.2	Preliminary bearing design (see Section 5)	25
10.2.3	Performance analysis (see Section 6)	26
10.2.4	Calculation of temperatures (see Section 6.2.4)	27
10.2.5	Power loss and lubricant supply (see Section 8)	28
10.3	Example 3	29
10.3.1	Specification of design requirements (see Section 4)	29
10.3.2	Preliminary bearing design (see Section 5)	30
10.3.3	Performance analysis (see Section 6)	30
10.3.4	Calculation of temperatures (see Section 6.2.4)	31
10.3.5	Power loss and lubricant supply (see Section 8)	33
10.4	Example 4	34
10.4.1	Specification of design requirements (see Section 4)	34
10.4.2	Preliminary bearing design (see Section 5)	35
10.4.3	Performance analysis (see Section 4)	35
10.4.4	Calculation of temperature (see Section 6.2.4)	36
10.4.5	Power loss and lubricant supply (see Section 8)	36
11.	TABLES	38
	FIGURES 1 TO 6	41-46

CALCULATION METHODS FOR STEADILY LOADED FIXED-INCLINED-PAD THRUST BEARINGS

1. NOTATION AND UNITS

The International System of Units (SI) is used in this Item. This is a coherent system. It should be noted that any other coherent system of units (using revolutions as the unit of shaft rotation) can be used in place of SI where the data are given in non-dimensional form.

		SI Units
b	pad radial width (see Sketch 3.3)	m
C	specific heat capacity of lubricant	J/kg K
D	outside diameter of thrust pads (see Sketch 3.3)	m
D_R	diameter of runner (see Sketch 3.1)	m
d	inside diameter of thrust pads (see Sketch 3.3)	m
d_m	mean diameter of thrust pads, $(D + d)/2$	m
d_s	shaft diameter (see Sketch 3.1)	m
H	total film power loss, nH_p	W
H_{ch}	churning power loss	W
H_p	film power loss for single thrust pad	W
H_t	total power loss for whole bearing, $H + H_{ch}$	W
h	pad outlet film thickness (see Sketch 3.2)	m
h_a	minimum acceptable film thickness where misalignment is known to be minimal (see Section 7.1)	m
h_{min}	minimum allowable pad film thickness	m
h_r	representative film thickness for Peclet number (see Section 6.2.4)	m
K	factor for calculation of churning losses (see Section 8.2)	–
L	length of pad at mean diameter (see Sketch 3.3)	m
N	frequency of rotation	rev/s
n	number of thrust pads	–

Issued November 1982 – 42 pages
 With Amendments A and B, September 2002 – 46 pages
 Re-issued November 2008 – 46 pages
 This page Amendment B

Pe	Peclet number = $\frac{\pi N d_m^2}{\kappa L} h_r^2$ (see Section 6.2.4)	–
P_l	land pressure (at start-up)	N/m ²
p_{mean}	mean pad pressure	N/m ²
Q	total lubricant feed flow rate to bearing	m ³ /s
Q_f	lubricant feed flow rate to single groove	m ³ /s
$(Q_f)_{min}$	minimum lubricant feed flow rate to single groove	m ³ /s
Q_i	pad entry lubricant flow rate	m ³ /s
Q_s	lubricant flow rate from inner and outer sides of single thrust pad	m ³ /s
R_a	surface roughness value	μm
Re	runner Reynolds number = $\frac{\rho N D_R^2}{\eta}$	–
s_1	empirical factor for calculation of temperatures (see Section 6.2.4)	–
s_2	empirical factor for calculation of temperatures (see Section 6.2.4)	–
T_e	effective film temperature of lubricant for design calculations	°C
T_f	pad feed temperature	°C
T_{max}	pad maximum temperature	°C
T_{in}	lubricant temperature at inlet to housing	°C
T_{out}	lubricant temperature at outlet from housing	°C
t	runner thickness (see Sketch 3.1)	m
W	total load, $n W_p$ (see Sketch 3.1)	N
W_p	load for single thrust pad (see Sketch 3.2)	N
W_s	total start-up load	N
α	factor for calculation of temperatures (see Section 6.2.4)	–
β	taper rise of thrust pad (see Sketch 3.2)	m
η	effective dynamic viscosity of lubricant in housing	N s/m ²
η_e	effective dynamic viscosity of lubricant in film	N s/m ²

κ	thermal diffusivity of lubricant	m^2/s
ν	kinematic viscosity of lubricant in housing	m^2/s
ρ	density of lubricant	kg/m^3

2. INTRODUCTION

This Item provides charts and calculation methods for the design of fixed-inclined-pad thrust bearings. The Item may also be used to analyse the performance of existing bearing designs in failure analysis or to investigate the feasibility of uprating their performance.

An alternative form of thrust bearing incorporates tilting pads (see Reference 1). A number of factors may influence the choice between the two forms but discussion of these factors is beyond the scope of this Item.

The charts and calculation methods presented in this Item have been derived largely from the established data available for rectangular pad bearings with a land length of 20 per cent (measured at the mean diameter). However, no significant errors should arise for land lengths within the range 10 to 30 per cent. The use of rectangular pad data for the design and analysis of bearings with sector shaped pads does not lead to errors of practical significance with sector angles up to 45° . The bearings considered are steadily loaded with the load shared equally between the pads but applications with loads and speeds varying slowly may be treated by the method of this Item.

The text of this Item is divided into two major parts. The first part of the text (Sections 3, 4 and 5) describes the operation of a fixed-inclined-pad thrust bearing, gives guidance on the specification of the design parameters and provides a method for estimating the required dimensions of the bearing. The second part of the text (Sections 6, 7 and 8) provides a method for analysing the performance of a bearing design and gives guidance on modifying the performance and ensuring that the bearing operates with an adequate margin of safety.

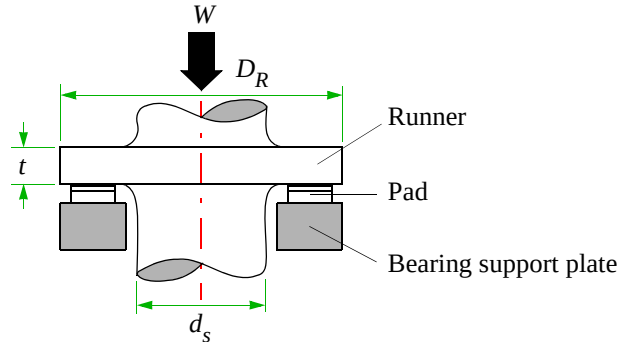
3. DESCRIPTION

The operation of a hydrodynamic thrust bearing depends, as it does for a journal bearing, upon a viscous lubricant being drawn into a wedge-shaped space. A pressure is generated in the tapering lubricant film so balancing the applied load.

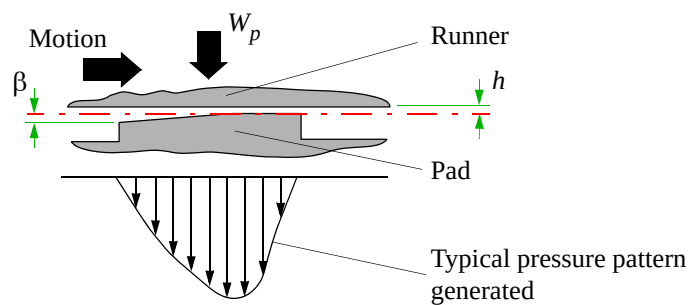
The main components of a fixed-inclined-pad thrust bearing are shown in Sketch 3.1. In such a thrust bearing, for constant direction of rotation, the wedge-shaped space is obtained by the use of a suitably machined pad (see Sketch 3.2). A number of such pads divides the bearing support plate as shown in Sketch 3.3. Each pad is profiled so as to provide a convergent lubricant film and the motion of the runner in the direction of decreasing film thickness draws the lubricant into the converging taper, thus establishing the load carrying film. The pads are usually made with near "square" ($b = L$) proportions (see Section 6.1.2).

A land is normally left after machining the taper to provide starting load capacity (see Section 5.1.2). The run out of the tapered portion of the pad surface onto the land is often arranged to be parallel to the leading edge of the pad as illustrated. The pads are usually surfaced with a bearing metal such as whitmetal and mate with a plain carbon steel runner. Chromium alloy steel runners are prone to wire wool failure and are, therefore, not recommended.

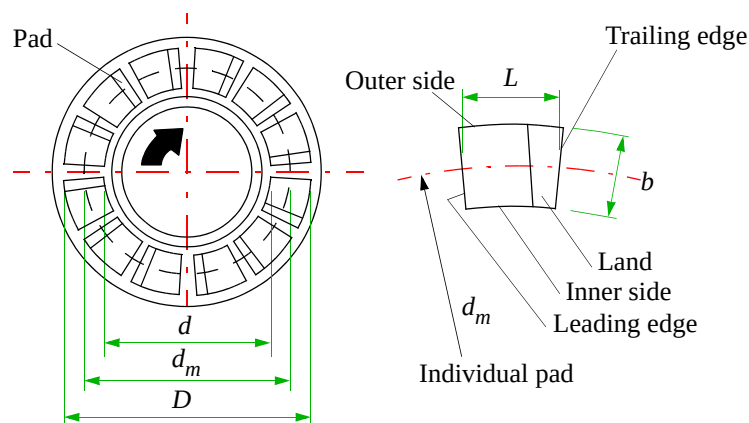
The bearing is commonly enclosed in a housing and is provided with a lubricant supply and a lubricant drain. It is assumed that the bearing receives sufficient lubricant by either supplying the lubricant directly between the pads (directed lubrication) or by arranging that the bearing operates with the housing full of lubricant (flood feed).



Sketch 3.1 View showing relative position of bearing components



Sketch 3.2 Section through a thrust pad



Sketch 3.3 View showing layout of pads on bearing support plate

4. SPECIFICATION OF DESIGN REQUIREMENTS

Certain basic information regarding the intended application is required in order to facilitate the design and analysis of the thrust bearing.

4.1 Load and Speed

The load, W , and frequency of rotation, N , should be specified assuming steady loading and a constant rotational speed of the bearing. In specifying the rotational speed of the bearing, it should be noted that short overspeeds can usually be tolerated. However, sometimes “off design” loads and speeds occur that can give rise to more severe operating conditions. It is, therefore, advisable to investigate the operation of the bearing under all potentially critical conditions.

High loads combined with low speeds are normally critical with regard to film thickness (see Section 6.2.1) whilst high loads combined with moderate to high speeds can be critical with regard to temperatures (see Section 6.2.4).

The starting load is also an important factor (see Section 5.1.2).

4.2 Dimensional Constraints

The shaft diameter, d_s , is usually predetermined by strength or stiffness requirements.

In choosing a value for the inner diameter, d , of the bearing pads allowance should be made for blending radii, oilways and running tolerances. The inner diameter is usually made 10 to 20 per cent greater than the shaft diameter.

Sometimes the overall bearing size is restricted. However, a machine design should not impose space restrictions unless absolutely necessary and for this reason early consideration of bearing requirements is desirable.

4.3 The Lubricant

The lubricant is often specified to suit the design requirements of an associated journal bearing or another component. The designer should have available the temperature-viscosity characteristics of the lubricant. Typical temperature-viscosity characteristics are shown in Sketch 6.1 and will serve as a guide where the designer has a free choice of lubricant.

For the purposes of analysing the performance of the bearing (see Section 6) it is necessary to assume a value for the temperature of the lubricant available to the pads, T_f . In flooded or oil-bath lubrication systems (see Section 8.1) this is not likely to be significantly different from the housing drain temperature, T_{out} , the value of which would be controlled by adjustment of the lubricant flow rate through the housing. Typically for oils, T_f would be between 50 and 70°C. With directed lubrication systems (see Section 8.1) T_f may be assumed to be equal to the supply temperature, T_{in} .

5. PRELIMINARY BEARING DESIGN

When designing a bearing it is convenient to start with an approximate design arrived at by the use of broad assumptions in order to:

- (i) provide a starting point for a more accurate iterative procedure (which rapidly converges) from which the final bearing dimensions and performance may be calculated,
- (ii) see whether a lower than ideal margin of safety may have to be accepted in order to obtain a bearing design which satisfies specified space limitations.

This section gives guidance on estimating the required dimensions of the bearing. The procedure to be followed is shown in the flow chart of Table 11.1.

The designer should always strive to design bearings that are not unnecessarily large since they are wasteful of materials and energy. However, the bearing must be made large enough to provide sufficient total pad area to support the running load safely (see Section 5.1.1). Sometimes the bearing size is dictated by the necessity to provide sufficient land area to support the start-up load (see Section 5.1.2).

Particular design requirements and constraints, such as space restrictions, may require that the initial bearing dimensions be revised. Points at which the designer may have to revise the values are indicated in the flow chart. Subsequent analysis of the performance of the preliminary bearing design (see Section 6) may necessitate further modifications to the bearing dimensions.

5.1 Required Total Pad Area

The required total pad area will be dictated by either running requirements or start-up requirements.

5.1.1 Running requirements and mean pad pressure

The mean pad pressure, p_{mean} , is an important parameter in the operation of a thrust bearing. It is possible to operate thrust bearings up to very high mean pressures of about 7 MN/m² but very careful design is then required in order to ensure that an adequate lubricant film thickness is maintained between the pads and the runner and to avoid high bearing temperatures. In practice the loads applied to the bearing are often not known accurately and may increase throughout the life of the machine. Also the ability of the bearing to accommodate the load safely can be degraded by, for example, wear of the pads. As a consequence it is unusual for bearings to be designed for mean pad pressures exceeding about 3 MN/m². In the majority of applications it is considered advisable to design initially for a mean pad pressure of typically 2 MN/m². Some applications, where low viscosity lubricants such as water are used, may require that the mean pressure be considerably lower than this value.

Having chosen the initial value of the mean pad pressure the total pad area required is equal to W/p_{mean} . Where the overall bearing size is restricted it may be necessary to reduce the total pad area so allowing a more compact bearing to be designed. In such cases the design mean pad pressure would be increased.

5.1.2 Start-up requirements and land pressure

At start-up of the bearing the lubricant film separating the pads and the runner has not developed. The load on the bearing is, therefore, supported on the land of each pad. As a general guide a value of 0.7 MN/m² for the land pressure, p_l , may be used for whitemetal faced pads to determine the allowable thrust under start-up conditions. Bronze pads, however, have poor start-up characteristics and for this reason their use is not recommended. When start-up requirements dictate the bearing design consideration should be given to increasing the land length to 30 per cent. Where only infrequent stop/starts are anticipated higher land pressures can be allowed at start-up. Still greater loads can be carried if hydrostatic jacking is used.

This page Amendment B

Control over the land length when machining the pads is very difficult due to the small taper rise, β , of the pad surface. The length of the land should, ideally, be within the range 10 to 30 per cent of the pad length in the circumferential direction. The charts and calculation methods given in this Item relate to a land length of 20 per cent but the results are not greatly affected by moderate variations from this value.

Having chosen the value of the land pressure the total pad area required, assuming a 20 per cent land, is

$$\frac{100}{20} \times \frac{W_s}{p_l}.$$

5.2 Preliminary Bearing Dimensions

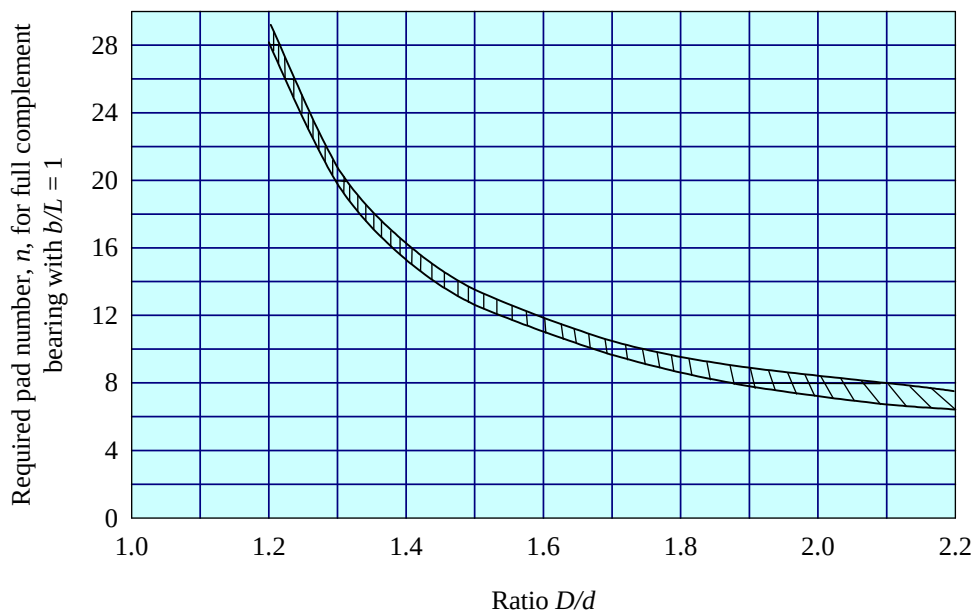
5.2.1 Full complement bearings

Initially the designer should consider a full complement bearing. Full complement bearings have the minimum practical overall size for a required total pad area and it is normal practice for the pads to occupy no more than approximately 5/6 of the area enclosed between D and d . The minimum practical outside diameter for the required total pad area may be found using Equation (5.1).

$$\text{Total pad area} = \frac{5}{6} \times \frac{\pi}{4} (D^2 - d^2). \quad (5.1)$$

In a full complement bearing the required number of pads, n , depends on the ratios D/d and b/L . It is usual to design for square pads (see Section 6.1.2) in which case guidance on the required number of pads is as indicated in Sketch 5.1.

Ratios of D/d exceeding those covered by Sketch 5.1 are not normally to be recommended. Where larger D/d ratios have been calculated the designer is advised to increase the inside diameter of the bearing, d , and to recalculate the bearing geometry.



Sketch 5.1 Required pad number

5.2.2 Partial complement bearings

Where the specified inside diameter of the bearing, d , is comparatively large for the design load, only a small D/d ratio would be required to accommodate the required total pad area in a full complement bearing and a correspondingly large number of small pads would be called for. Such a bearing design might be regarded as unduly complex and expensive and it is often advisable in such cases to employ a partial complement of larger pads in a bearing with an increased overall diameter. The required square pad size for a specified pad number is then given by Equation (5.2).

$$bL = b^2 = L^2 = \frac{\text{Required total pad area}}{n} \quad (5.2)$$

The increased overall diameter may be found using Equation (5.3).

$$D = d + 2b \quad (5.3)$$

6. PERFORMANCE ANALYSIS

This section gives guidance on analysing the performance of a proposed or existing design. Charts and calculation methods are provided to allow the film thickness, the film power loss, the lubricant flow rates, and the pad maximum temperature to be determined.

The procedure to be followed is shown in the flow chart of Table 11.2. The flow chart has alternative entry points depending on whether the performance of a proposed design or an existing design is to be analysed. When analysing the performance of a proposed design the designer should check that the predicted performance is acceptable and satisfies the film thickness and temperature limit criteria described in Section 7. Where the performance is inadequate the designer may have to modify the dimensions of the bearing or, in extreme cases, revise the design requirements. Points at which this may be necessary are indicated in the flow chart.

When designing a bearing, the first requirements to be met are that it should operate with an adequate minimum film thickness and an acceptable maximum pad temperature. Once this has been achieved the design may then be further refined to reduce the power loss, possibly by the use of directed lubrication (see Section 8.1).

For the purposes of design there is freedom to adjust the parameters of the bearing in order to obtain a satisfactory performance. However, practical considerations limit the range of values available to the designer. For example, the pads should have near square proportions and problems of machining the pads may limit the practical values of β/h (see Section 6.1.2).

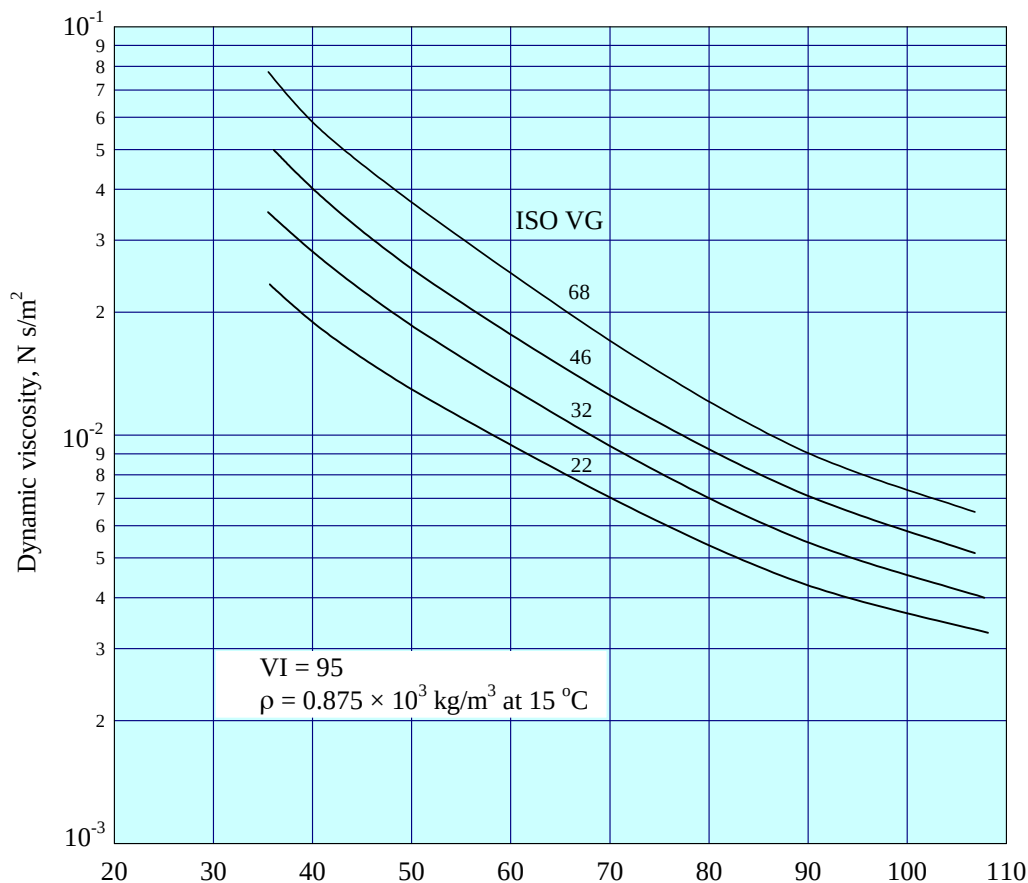
6.1 Initial Assumptions for Performance Analysis

6.1.1 Effective film temperature and viscosity

To determine the performance of the bearing the effective viscosity of the lubricant in the film, η_e , must be known. However, the effective viscosity of the lubricant depends on its temperature which in turn depends on the dimensions of the bearing and the operating conditions. It is, therefore, necessary to assume initial values for the effective temperature, T_e , and the effective viscosity, η_e , of the lubricant within the film and then to refine these values using an iterative procedure.

The effective temperature of the lubricant in the film, T_e , will always be higher than the pad feed temperature, T_f . Typically, T_e would be 10 to 30 K above T_f .

In analysing the performance of the bearing the designer should have available the temperature-viscosity characteristics of the lubricant. The effective dynamic viscosity of the lubricant, η_e , corresponding to the assumed value of T_e may then be determined. Sketch 6.1 shows the temperature-viscosity characteristics of typical lubricants (ISO 3448).



Sketch 6.1 Temperature-viscosity characteristics of mineral oils

6.1.2 Pad rise and pad proportions

The parameter β/h is important for the satisfactory operation of the bearing. Small values of β/h ($\beta/h \leq 2.0$) can give rise to high operating temperatures (see Section 6.2.4) and high power losses (see Section 6.2.2) and should therefore be avoided. A value of $\beta/h \approx 3.0$ is usually considered optimal for the operation of the bearing.

The value of pad rise, β , corresponding to a β/h value of 3 is $3h$. Where the film thickness is of necessity small, such as in small bearings or water lubricated bearings, the value of β corresponding to $\beta/h = 3.0$ can be so small as to present machining difficulties. In such applications the designer may have to specify a value of β that can be effectively machined and design for a β/h value correspondingly greater than 3.0.

It is recommended that b/L ratios should be within the range 0.8 to 1.2. The available pad area is best used as approximately square pads. However, bearings with non-square pads (rectangular pads) do exist and the designer may wish to investigate such designs.

This page Amendment B

The performance analysis procedure for a proposed design described in this Item assumes initial values of $\beta/h = 3$ and $b/L = 1$. Figures 1 to 4 allow investigation of the performance of bearings with different values of β/h (0 to 19) and b/L (0.6 to 1.5).

6.2 Performance Calculations

To proceed with the performance calculations the mean pad pressure, p_{mean} , should be calculated according to Equation (6.1).

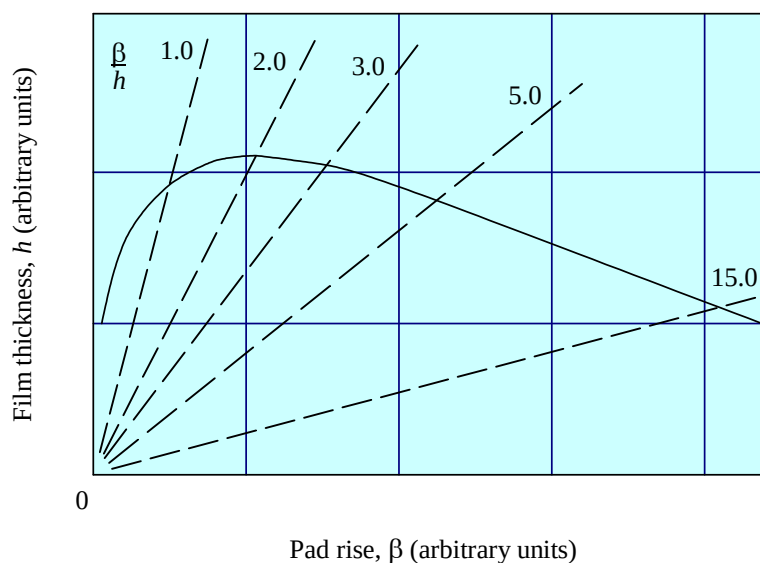
$$p_{mean} = \frac{W}{nbL} \quad (6.1)$$

6.2.1 Film thickness

The value of the film thickness, h , is required in order that the safe operation of the bearing can be checked (see Section 7.1) and so that the value of the pad rise, β , corresponding to an assumed value of β/h (see Section 6.1.2) can be determined.

The film thickness depends on the dimensions of the pads, the pad rise, the surface speed of the runner, the mean pad pressure and the effective viscosity of the lubricant in the film. In general, increasing the size of the bearing for a specified duty increases the film thickness. However, larger bearings will have a greater power loss than compact bearings (see Sections 6.2.2 and 8.2).

A typical variation of the film thickness with the pad rise for a bearing lubricated with a mineral oil is shown in Sketch 6.2. A similar variation of film thickness with pad rise applies to bearings with alternative lubricants. As shown, small values of β , corresponding to values of the parameter β/h less than about 1.0, give rise to seriously reduced film thickness. Pad wear, which could be accelerated by the higher pad temperatures that occur with small values of β/h (see Section 6.2.4), will reduce the pad rise and further reduce the film thickness.



Sketch 6.2 Typical variation of film thickness with pad rise

From Figure 1 the value of the non-dimensional load per pad corresponding to $\beta/h = 3.0$ and $b/L = 1.0$ is 1.50. Hence, where values of $\beta/h = 3.0$ and $b/L = 1.0$ have been assumed as a starting point for the performance analysis (see Section 6.1.2) the film thickness is given by Equation (6.2),

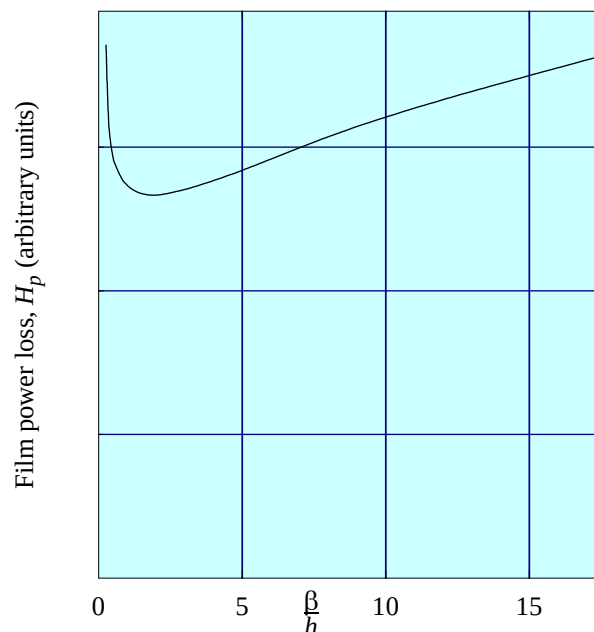
$$h = 0.5 \sqrt{\frac{\eta_e N d_m b^3}{W_p}} = 0.5 \sqrt{\frac{\eta_e N d_m b}{p_{mean}}} \quad (6.2)$$

Where the value of β/h is not known but the pad rise, β , is either known or has been specified (see Section 6.1.2), Figure 1 may be used to find the value of the parameter β/h and hence h . Figure 1 may also be used to find β/h and h where b/L ratios other than unity have been used.

The calculated value of h should be compared with the minimum film thickness criterion (see Section 7.1). Where the calculated value of h is inadequate the bearing design will have to be modified in order to increase the film thickness. It is usually impractical to increase the viscosity of the lubricant (see Section 4.3) in order to increase the film thickness. In the majority of applications the designer should decrease the mean pad pressure in order to increase the film thickness.

6.2.2 Film power loss

The value of the film power loss is required in order to calculate the bearing temperatures[†] (see Section 6.2.4). The total power loss of the bearing arises from friction losses within the lubricant film and churning losses. Guidance on the estimation of churning losses is given in Section 8.2. The film power loss of an individual pad depends on the dimensions of the pad, the pad rise, the sliding speed of the pads relative to the runner, the mean pad pressure and the effective viscosity of the lubricant in the film. A typical variation of the film power loss with the parameter β/h for a bearing lubricated with a mineral oil is shown in Sketch 6.3. As shown reducing β/h below about 1.0 gives rise to a rapid increase in film power loss.



Sketch 6.3 Typical variation of power loss with $\frac{\beta}{h}$

[†] Where a bearing design with $\beta/h = 3$ and $b/L = 1$ is being analysed the bearing temperatures can be determined without first calculating the film power loss (see Table 11.2).

Figure 2 may be used to find the film power loss for a range of values of β/h and b/L . The power loss is expressed on a per pad basis and should, therefore, be multiplied by the number of pads, n , in order to obtain the total film power loss, H .

From Figure 2 the value of the non-dimensional film power loss per pad corresponding to $\beta/h = 3$ and $b/L = 1$ is 20.4. Hence, where a bearing with $\beta/h = 3.0$ and $b/L = 1.0$ is being designed (see Section 6.1.2) the film power loss per pad, H_p , is given by Equation (6.3),

$$H_p = 9.1 \frac{\beta}{b} N d_m W_p. \quad (6.3)$$

In some applications the power loss of the bearing can be critical. In general compact bearings have the lowest power loss. However, compact bearings may mean that high values of p_{mean} and small h values are inevitable.

6.2.3 Lubricant flow rates

The lubricant flow rates, Q_i and Q_s , are required in order to establish the heat balance[†] (see Section 6.2.4) and, with directed lubrication (see Section 8.1) to estimate the lubrication requirements.

Figures 3 and 4 may be used to find the lubricant flow rates for a range of values of β/h and b/L .

From Figure 3 the value of the non-dimensional lubricant inlet flow rate per pad corresponding to $\beta/h = 3$ and $b/L = 1$ is 1.38. From Figure 4 the value of the non-dimensional lubricant side leakage flow rate per pad corresponding to $\beta/h = 3$ and $b/L = 1$ is 0.77. Hence, where a bearing with $\beta/h = 3.0$ and $b/L = 1.0$ is being designed (see Section 6.1.2) the lubricant flow rates are given by Equations (6.4) and (6.5),

$$Q_i = 1.38 \beta N d_m b \quad (6.4)$$

$$\text{and} \quad Q_s = 0.77 \beta N d_m b. \quad (6.5)$$

6.2.4 Temperatures within bearing

The pad maximum temperature is required in order that the safe operation of the bearing can be checked (see Section 7.2). The procedure to be followed when calculating bearing temperatures in an existing or proposed design is shown in the flow chart of Table 11.3.

The pad maximum temperature is influenced by the pad rise, β . Sketch 6.4 shows a typical variation of the pad maximum temperature, T_{max} , with the parameter β/h . As shown, reducing β/h below about 2.0 gives rise to a rapid increase in pad temperature.

[†] Where a bearing design with $\beta/h = 3$ and $b/L = 1$ is being analysed the heat balance can be established without first calculating the lubricant flow rates (see Table 11.2).