



ESDU 84031

Issued November 1984
With Amendment A
September 1991
Re-issued November 2008

Calculation methods for steadily loaded axial groove hydrodynamic journal bearings

Endorsed by
The Institution of Mechanical Engineers

AVA Documents
Document Sharing Hub

ESDU DATA ITEMS

Data Items provide validated information in engineering design and analysis for use by, or under the supervision of, professionally qualified engineers. The data are founded on an evaluation of all the relevant information, both published and unpublished, and are invariably supported by original work of ESDU staff engineers or consultants. The whole process is subject to independent review for which crucial support is provided by industrial companies, government research laboratories, universities and others from around the world through the participation of some of their leading experts on ESDU Technical Committees. This process ensures that the results of much valuable work (theoretical, experimental and operational), which may not be widely available or in a readily usable form, can be communicated concisely and accurately to the engineering community.

We are constantly striving to develop new work and review data already issued. Any comments arising out of your use of our data, or any suggestions for new topics or information that might lead to improvements, will help us to provide a better service.

THE PREPARATION OF THIS DATA ITEM

The work on this particular Item was monitored and guided by the Journal Bearings Working Party:

Chairman	
Dr P.B. Neal	— University of Sheffield
Members	
Mr F.A. Martin	— Independent
Mr G.S. Ritchie	— GEC Power Engineering Ltd
Dr C.M. Taylor	— University of Leeds.

The Working Party was appointed by the Tribology Steering Group which approved the final draft and has the following constitution:

Chairman	
Mr F.A. Martin	— Independent
Vice-Chairman	
Dr P.B. Neal	— University of Sheffield
Members	
Prof. Dr-Ing.W.J. Bartz [†]	— Technische Akademie Esslingen, Federal Republic of Germany
Mr O. von Bertele	— Independent
Prof. H.S. Cheng [†]	— Northwestern University, Illinois, USA
Prof. D. Dowson [‡]	— University of Leeds
Mr T.I. Fowle	— Independent
Prof. J. Halling	— Independent
Prof. Y. Hori [†]	— University of Tokyo
Mr M.J. Neale [‡]	— Michael Neale & Associates
Dr D.W. Parkins	— Cranfield University
Dr S.Y. Poon	— I.T. Research Ltd
Mr G.S. Ritchie	— GEC Power Engineering Ltd
Dr W.H. Roberts	— National Centre of Tribology
Dr C.M. Taylor	— University of Leeds
Dr D.F. Wilcock [†]	— Tribolock Incorporated, NY, USA
Dr R.W. Wilson	— Independent.

[†] Corresponding Member

[‡] Member Emeritus

(Continued on inside back cover)

CALCULATION METHODS FOR STEADILY LOADED AXIAL GROOVE HYDRODYNAMIC JOURNAL BEARINGS

CONTENTS

	Page
1. NOTATION AND UNITS	1
2. INTRODUCTION	4
3. DESCRIPTION	4
4. SPECIFICATION OF DESIGN REQUIREMENTS	5
4.1 Load and Speed	6
4.2 Dimensional Constraints	6
4.3 The Lubricant	6
5. PRELIMINARY BEARING DESIGN	7
5.1 Required Overall Bearing Size	7
5.1.1 Running requirements and specific loading	7
5.1.2 Start-up requirements	8
5.1.3 Preliminary bearing dimensions	8
5.2 Other Bearing Dimensions	9
5.2.1 Bearing clearance	9
5.2.2 Groove dimensions	9
6. PERFORMANCE ANALYSIS	9
6.1 Effective Viscosity and Groove Viscosity	10
6.2 Eccentricity Ratio	10
6.3 Power Loss	10
6.4 Lubricant Flow Rates	10
6.5 Temperatures Within Bearing	11
6.6 Film Thickness	12
7. LIMITING CRITERIA	13
7.1 Minimum Film Thickness Criterion	13
7.2 Maximum Temperature Criterion	13
8. FURTHER DESIGN CONSIDERATIONS	14
8.1 Lubricant Supply	14
8.2 Alternative Feed Arrangements	14
8.3 Shaft Position	14
8.4 Non-laminar Operation	14
8.5 Bearing Induced Instability	15

9.	REFERENCES AND DERIVATION	15
9.1	References	15
9.2	Derivation	16
10.	EXAMPLES	16
10.1	Example 1	16
10.1.1	Specification of design requirements (see Section 4)	16
10.1.2	Preliminary bearing design (see Figure 2)	16
10.1.3	Performance analysis (see Figure 3)	17
10.1.4	Calculation of temperatures (see Figure 4)	19
10.1.5	Calculation of minimum film thickness (see Figure 4)	21
10.1.6	Further design considerations	21
10.1.7	Summary of final bearing design	22
10.2	Example 2	22
10.2.1	Specification of bearing	22
10.2.2	Performance analysis (see Figure 4)	23
10.2.3	Calculation of temperatures	24
10.2.4	Calculation of minimum film thickness	25
10.2.5	Further design considerations	26
10.2.6	Summary of analysis	26
10.3	Example 3	27
10.3.1	Specification of bearing	27
10.3.2	Performance analysis (see Figure 4)	27
10.3.3	Calculation of temperature (see Figure 4)	28
10.3.4	Calculation of minimum film thickness	29
10.4	Example 4	30
10.4.1	Specification of design requirements (see Section 4)	30
10.4.2	Preliminary bearing design (see Figure 2)	30
10.4.3	Performance analysis (see Figure 4)	30
10.4.4	Calculation of temperatures	31
10.4.5	Performance analysis (see Figure 4)	33
10.4.6	Summary	34

FIGURES 1 to 12	35 – 46
------------------------	----------------

CALCULATION METHODS FOR STEADILY LOADED AXIAL GROOVE HYDRODYNAMIC JOURNAL BEARINGS

1. NOTATION AND UNITS

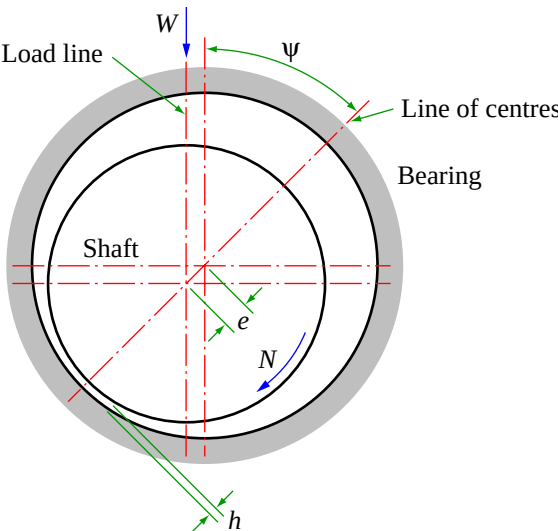
The International System of Units (SI) is used in this Item. This is a coherent system. It should be noted that any other coherent system of units (using revolutions as the unit of shaft rotation) can be used in place of SI where the data are given in non-dimensional form.

a	axial length of lubricant groove	m
b	axial length of bearing	m
C	specific heat capacity of lubricant	J/kg K
c_d	diametral clearance under running conditions	m
d	diameter of journal	m
e	eccentricity for an aligned bearing (see Sketch 1.1)	m
F	groove function for calculation of Q_p	–
g	acceleration due to gravity	m/s ²
H	power loss	W
H'	non-dimensional power loss = $\left(\frac{H}{\eta_e N^2 d^2 b} \right) \left(\frac{c_d}{d} \right)$	–
h	film thickness at closest approach for aligned bearing	m
h_{edge}	film thickness at closest approach for misaligned bearing	m
h_s	safe allowable film thickness	m
K	power loss factor (see Section 6.5)	–
K_s	empirical factor for calculation of temperature rise (see Section 6.5)	–
M	rotor mass associated with bearing	kg
N	frequency of shaft rotation	rev/s
Pe	Peclet number = $\frac{Nc_d^2}{\kappa}$ for calculation of temperature rise	–
p_f	lubricant feed pressure	N/m ²
p_{mean}	specific loading = $W/(bd)$	N/m ²
Q	lubricant flow rate into bearing (see Sketch 1.2)	m ³ /s
Q_p	feed pressure flow rate of lubricant from groove(s)	m ³ /s

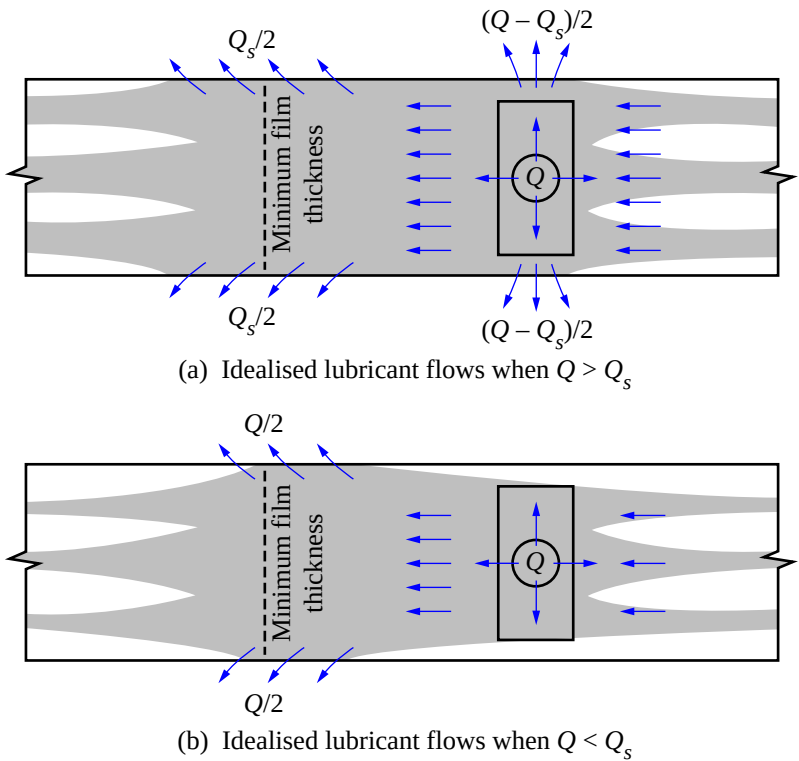
Issued November 1984 – 42 pages
 With Amendment A, September 1991 – 46 pages
 Re-issued November 2008 – 46 pages
 This page Amendment A

Q'_p	non-dimensional feed pressure flow rate of lubricant from groove(s) = $\frac{Q_p \eta_g}{c_d^3 p_f}$	–
Q_r	recirculating flow rate	m ³ /s
Q'_r	non-dimensional recirculating flow rate = $\frac{Q_r}{c_d N b d}$	–
Q_s	idealised side flow rate (see Sketch 1.2)	m ³ /s
Q'_s	non-dimensional idealised side flow rate = $\frac{Q_s}{c_d N b d}$	–
Ra	surface roughness value	m
Re	bearing Reynolds number = $\frac{\pi d N c_d \rho}{2 \eta_e}$	–
T_e	effective temperature of lubricant in loaded part of bearing	°C
T_f	lubricant feed temperature	°C
T_g	lubricant temperature in grooves	°C
T_{max}	maximum bearing surface temperature	°C
T_{out}	bulk outlet temperature from bearing	°C
W	load	N
W'	non-dimensional load = $\left(\frac{W}{\eta_e N b d}\right) \left(\frac{c_d}{d}\right)^2$	–
w	width of groove measured in circumferential direction	m
Z	factor used in calculation of temperature rise (see Section 6.5)	–
β	misalignment (see Figure 10)	–
γ	factor for oil grade (see Sketch 5.1)	–
δ	misalignment over bearing length (see Figure 10)	m
$\Delta\theta$	angular extent of lubricant groove	deg
ε	eccentricity ratio for an aligned bearing = $2e/c_d$	–
η_e	effective dynamic viscosity of lubricant in loaded part of bearing	N s/m ²
η_g	dynamic viscosity of lubricant in grooves	N s/m ²
κ	thermal diffusivity of lubricant	m ² /s

ρ	density of lubricant	kg/m^3
Φ	empirical factor for calculation of temperature rise (see Section 6.5)	–
ψ	attitude angle (angle between load line and line of centres, (see Sketch 1.1)	deg



Sketch 1.1 Illustration of notation



Sketch 1.2 Idealised lubricant flows

2. INTRODUCTION

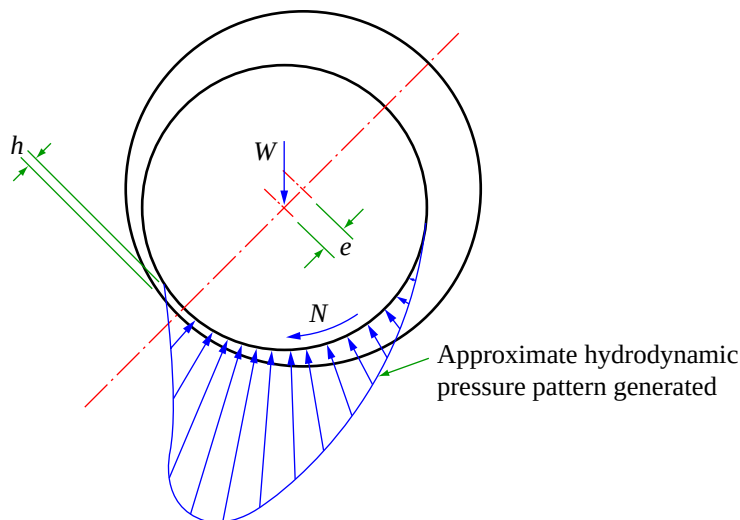
This Item provides charts and calculation methods for the design of forced lubricated hydrodynamic journal bearings. The Item may also be used to analyse the performance of existing bearing designs in failure analysis or to investigate the feasibility of uprating their performance.

The charts and calculation methods presented in this Item apply to bearings with one or two axial grooves at right angles to the load line (see Sketches 3.2 and 3.3). Alternative lubricant distribution arrangements are discussed. The bearings considered are steadily loaded but applications with loads and speeds varying slowly may also be treated by the method of this Item.

The text of this Item is divided into two main parts. The first part of the text (Sections 3, 4 and 5) describes the operation of a journal 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

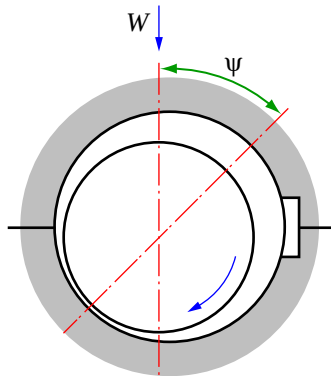
The operation of a hydrodynamic journal bearing depends upon a viscous lubricant being drawn into a wedge-shaped space. The journal takes up an eccentric position within the bearing clearance under the influence of the load which it carries. The eccentricity adjusts itself until the load is balanced by the pressures generated in the lubricant film as shown in Sketch 3.1.



Sketch 3.1 Pressure distribution in a journal bearing

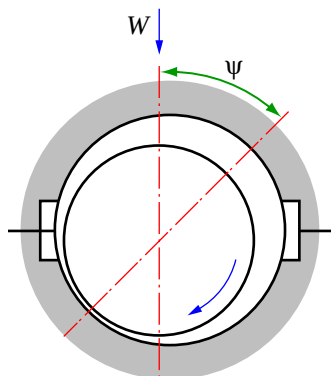
Oil lubricated bearings are usually lined with a material such as whitmetal and the journal is commonly manufactured from plain carbon steel. The bearing is often split along a diameter to facilitate assembly.

This Item deals with bearings in which the lubricant is supplied via a single axial groove at 90° to the load line or via two diametrically opposed axial grooves at 90° to the load line as shown in Sketches 3.2 and 3.3. Both lubricant distribution arrangements are suitable for split bearings.



Sketch 3.2 A single axial groove at 90° to the load line

With the single axial groove arrangement the groove would normally be located at the joint upstream of the minimum film thickness position as shown in Sketch 3.2. Shaft rotation in the opposite direction would generally involve reduced oil flow rate and higher outlet temperature.



Sketch 3.3 Two diametrically opposed axial grooves at 90° to the load line

The two axial groove arrangement, with grooves at both the joints (see Sketch 3.3), is equally suitable for shaft rotation in either direction.

Alternative arrangements for the lubricant feed are discussed in Section 8.2.

4. SPECIFICATION OF DESIGN REQUIREMENTS

Certain basic information regarding the intended application is required for the design and analysis of the journal bearing.

4.1 Load and Speed

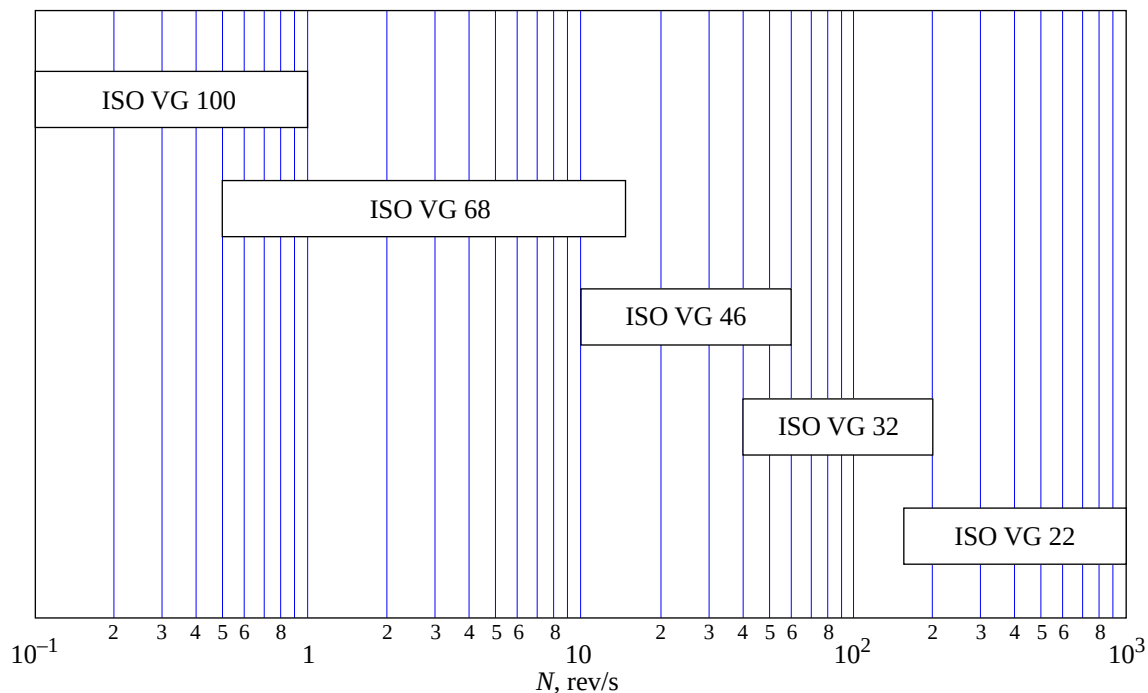
The load, W , and frequency of shaft rotation, N , must be specified assuming steady loading and a constant rotational speed. In specifying the rotational speed it should be noted that short overspeeds can usually be tolerated. Sometimes however, “off design” loads and speeds occur that can give rise 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 may be critical with regard to film thickness (see Section 6.6) whilst high loads combined with moderate to high speeds can be critical with regard to temperature (see Section 6.5). The starting load is also an important factor (see Section 5.1.2).

4.2 Dimensional Constraints

The journal diameter, d , is usually predetermined by strength or stiffness requirements. In choosing a value for the bearing length, b , consideration should be given to the recommendations on b/d ratio given in Section 5.1.3. Sometimes it may be necessary to increase the shaft diameter in order to allow a more favourable b/d ratio to be used.

4.3 The Lubricant

The lubricant is sometimes specified to suit the design requirements of another component. Sketch 4.1 will serve as a guide where a mineral oil is to be used and the designer has a free choice of lubricant grade. In some situations it may be best to use the process fluid as a lubricant (see Reference 1).



Sketch 4.1 Guide to choice of oil grade

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 supplied to the bearing, T_f , since this is the datum used in calculating temperature rise. Typically, T_f would be between 40 and 50 °C for oil lubricated bearings. In most circumstances a feed pressure, p_f , between 0.05 MN/m² and 0.35 MN/m² will be found to be adequate.

This page Amendment A

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 good starting point for the more comprehensive iterative design procedure 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 flowchart of Figure 2.

The designer should always strive to design bearings that are not unnecessarily large. However, the bearings must be made large enough to support the running load safely (see Section 5.1.1). Sometimes the bearing size is dictated by the necessity to support the start-up load (see Section 5.1.2).

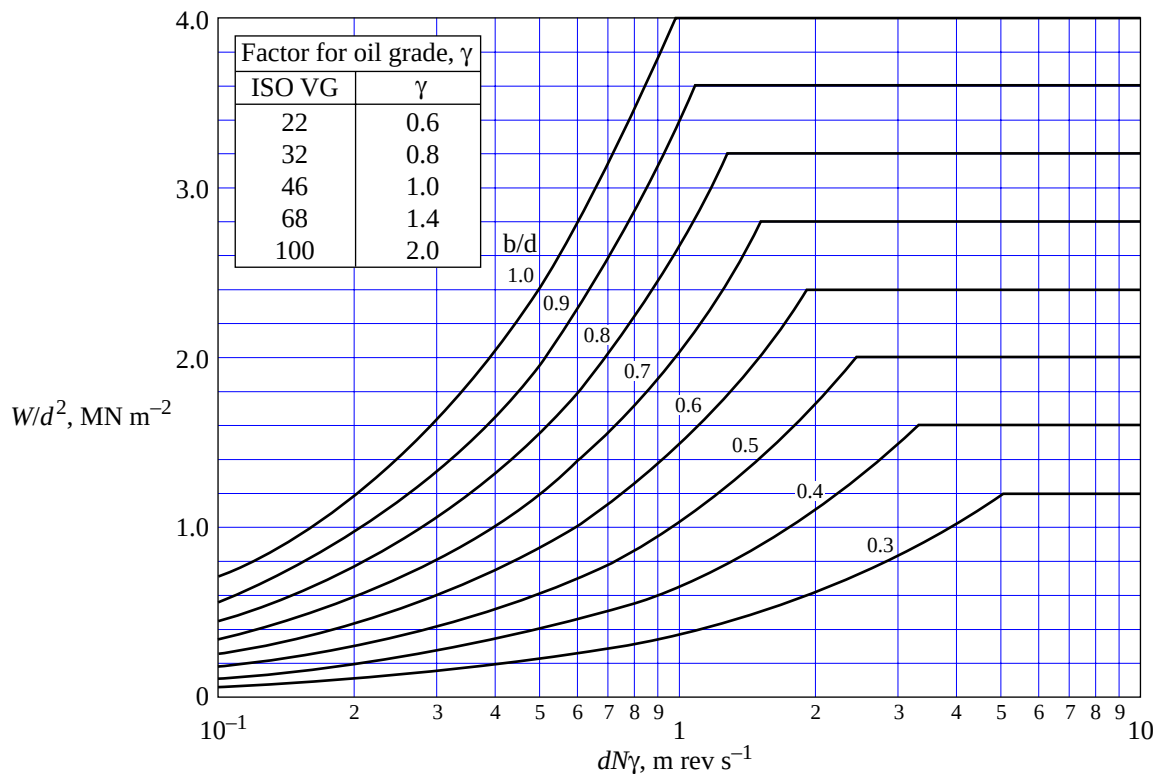
Particular design requirements and constraints, such as space restrictions, may require that the initial bearing dimensions be revised. Points at which the designer may have to revise the values are indicated in the flowchart. Subsequent analysis of the performance of the preliminary bearing design (see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

5.1 Required Overall Bearing Size

The required overall bearing size will be dictated by either running requirements or start-up requirements.

5.1.1 Running requirements and specific loading

The specific loading, p_{mean} , is an important parameter in the operation of a journal bearing. It is sometimes possible to operate journal bearings at very high specific loads of about 7 MN/m^2 but very careful workmanship is then required in order to ensure that an adequate lubricant film thickness is maintained between the journal and the bearing surfaces and to avoid high bearing temperatures. In practice the loads applied to the bearings 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 bearing. As a consequence it is unusual for whitmetal (babbit) bearings to be designed for specific loadings exceeding about 4 MN/m^2 . In many applications the specific loading will have to be less than this value in order to ensure an adequate lubricant film thickness. Where this is so, the allowable specific loading depends on the value of dN , as well as the viscosity of the lubricant as shown in Sketch 5.1. This sketch can be used to estimate the required bearing length, b , for initial design purposes when using mineral oil lubrication. Where low viscosity lubricants such as water are used (see Reference 1), the allowable specific loadings will be considerably lower than those incorporated in Sketch 5.1.



Sketch 5.1 Guide to required b/d ratio

5.1.2 Start-up requirements

At start-up of the bearing the lubricant film which will separate the journal and bearing surfaces has not developed. As a general guide a value of 1.3 MN/m^2 for the start-up specific loading may be used to determine the required size of whitmetal bearings with frequent stop/starts (several a day). Where only infrequent stop/starts (one a day or less) are anticipated specific loadings up to 2.5 MN/m^2 can be allowed. The required bearing length is given by the equation,

$$b = \frac{W}{dp_{\text{mean}}} \quad (5.1)$$

5.1.3 Preliminary bearing dimensions

Having estimated values of the bearing length, b , to satisfy both running requirements and start-up requirements, the larger value should be chosen.

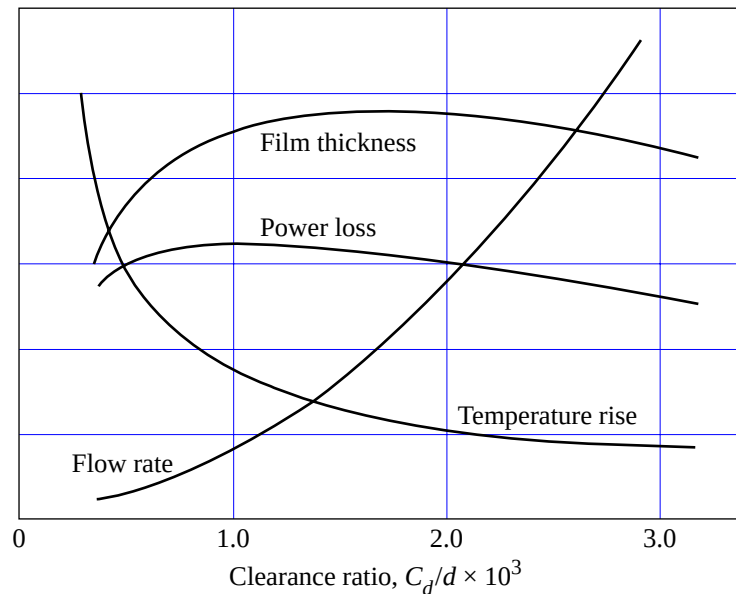
It is recommended that b/d should be less than 1.0. Large values of b/d may give rise to misalignment problems and where such values have been calculated the designer is advised to increase the journal diameter, d , and to recalculate the bearing geometry.

The procedure to be followed when choosing the preliminary bearing dimensions is shown in the flowchart of Figure 2.

5.2 Other Bearing Dimensions

5.2.1 Bearing clearance

The bearing clearance under running conditions is important for the satisfactory operation of the bearing. Both small and large values of the clearance ratio, c_d/d , can give rise to small lubricant film thickness. Small values of c_d/d can give rise to high bearing temperatures while large values of c_d/d can mean excessive lubricant flow rates as shown in Sketch 5.2.



Sketch 5.2 Typical variations of film thickness, power loss, flow rate and temperature rise with clearance ratio

Figure 1 shows values of clearance ratio which experience suggests are appropriate under running conditions. The diametral clearance must be held to close tolerance during manufacture.

5.2.2 Groove dimensions

In most applications adequate lubricant distribution will be ensured by lubricant grooves that extend 0.8 of the bearing length ($a/b = 0.8$) and have a circumferential width of about $0.25d$ (angular extent of about 30 deg). The depth of the grooves should not be less than $20c_d$ to avoid significant restriction of the inlet flow.

6. PERFORMANCE ANALYSIS

This section gives guidance on analysing the performance of a proposed or existing design. The performance of a hydrodynamic journal bearing depends on the dimensions of the bearing, the load, the effective viscosity of the lubricant, the rotational speed, the diametral clearance and any misalignment. Charts and calculation methods are provided to allow the film thickness, the power loss, the lubricant flow rates and the bearing temperatures to be determined.

The procedure to be followed is shown in the flowchart of Figure 3. 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 flowcharts.

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 temperature. Once this has been achieved the design may then be further refined.

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.

6.1 Effective Viscosity and Groove Viscosity

The performance prediction method of this Item requires values for the effective dynamic viscosity of the lubricant in the loaded part of the bearing, η_e , and the dynamic viscosity of the lubricant in the feed grooves, η_g . These viscosities correspond respectively to lubricant temperatures of T_e and T_g , values for which are assumed initially and then refined using an iterative procedure.

The effective temperature of the lubricant in the film, T_e , will always be higher than the bearing feed temperature, T_f . Typically, T_e would exceed T_f by 10 K at low speeds and 30 K at high speeds. Initially, T_g can be taken as equal to T_e .

In analysing the performance of the bearing the designer should have available the temperature-viscosity characteristics of the lubricant. The dynamic viscosities, η_e and η_g , corresponding to the lubricant temperatures, T_e and T_g , may then be determined. Figure 5 shows the temperature-viscosity characteristics of typical mineral oil lubricants.

6.2 Eccentricity Ratio

The eccentricity ratio, ϵ , is a fundamental parameter in the analysis of a hydrodynamic journal bearing. Its value, for an aligned bearing, may be found from Figure 6 after calculating the non-dimensional load, W' .

6.3 Power Loss

The power loss is of general design interest but is also required in order to calculate the bearing temperatures (see Section 6.5). In some applications the power loss of the bearing can be critical. Figure 7 may be used to find the power loss for the value of b/d selected.

6.4 Lubricant Flow Rates

The lubricant flow rates are required in order to establish the heat balance (see Section 6.5), and to estimate the lubricant supply requirements.

The bearing flow rate, Q , is given by

$$Q = Q_p + \frac{a}{b} Q_s. \quad (6.1)$$

Provided the angular extent of the groove(s) does not exceed 40 deg, the feed pressure flow rate, Q_p , can be calculated from

$$Q'_p = \frac{Q_p \eta_g}{c_d^3 p_f} = 0.026 \left(\frac{(1 - 0.2a/b)}{(b/a - 1)^{0.333}} + \frac{0.007 \Delta \theta}{(1 - a/b)(b/d)} \right) F, \quad (6.2)$$

where the groove function, F , may be found from Figure 8. When the ratio $a/b = 0.8$ (see Section 5.2.2), Q_p may be more simply calculated using

$$Q'_p = \frac{Q_p \eta_g}{c_d^3 p_f} = 0.035 \left(1 + 0.026 \frac{\Delta \theta}{b/d} \right) F, \quad (6.3)$$

or for $\Delta \theta = 30^\circ$

$$Q'_p = 0.035 \left(1 + \frac{0.78}{b/d} \right) F. \quad (6.4)$$

The non-dimensional idealised side flow rate, Q'_s , may be found using Figure 9.

The non-dimensional recirculating flow, Q'_r , which is required when calculating the bearing temperatures can be calculated from

$$Q'_r = \frac{Q_r}{c_d N d b} = 0.8(1 - \varepsilon). \quad (6.5)$$

6.5 Temperatures Within Bearing

The maximum temperature of the bearing is required to ensure that safe operation is achieved. The bulk outlet temperature is required in order that degradation of the lubricant can be checked (see Section 7). The procedure to be followed when calculating these temperatures in an existing or proposed design is shown in the flowchart of Figure 4.

The effective film temperature, T_e , the groove temperature, T_g , and the maximum bearing temperature, T_{max} , may be obtained using Equations (6.6), (6.7) and (6.8) together with predicted values of the power loss and the lubricant flow rates.

$$\text{For } Q \geq Q_s, T_e - T_f = \left(\frac{KH}{\rho C Q} \right) \left(\frac{\phi Q_r + Q}{\phi Q_r + K_s Q_s} \right), \quad (6.6)$$

$$\text{for } Q < Q_s, T_e - T_f = \left(\frac{KH}{\rho C Q} \right) \left(\frac{\phi Q_r + Q}{\phi Q_r + K_s Q} \right)$$

$$T_g - T_f = \left(\frac{\phi Q_r}{\phi Q_r + Q} \right) (T_e - T_f) \quad (6.7)$$

$$\text{and } T_{max} - T_g = \Phi (T_e - T_g). \quad (6.8)$$

The factors are given by the following equations.

$$\Phi = 1 + 2.75\epsilon^{1/2}Pe/(Pe + 10Z) \quad (\text{validated for } 1 < Pe < 250),$$

$$K = Pe/(Pe + Z) \text{ and } K_s = (5 - \Phi)/4$$

where the Peclet number, $Pe = \frac{Nc_d^2}{\kappa}$ and $Z = 6/(2 - \epsilon)$.

Typically, for mineral oils, $\rho C = 1.7 \times 10^6 \text{ J/m}^3 \text{ K}$ and $\kappa = 0.08 \times 10^{-6} \text{ m}^2/\text{s}$.

Having obtained a value for the effective film temperature, T_e , and the groove temperature, T_g , they should be compared with the values that were assumed initially (see Section 6.1). Where there is a significant difference between the values the calculations should be repeated with new values of T_e and T_g equal to the arithmetic means of the calculated and the old values[†]. It is recommended that this iteration be continued until the change in T_e and T_g is less than 2 K.

After termination of the iteration, the maximum bearing temperature, T_{max} , may be calculated. The calculated value of T_{max} should be compared with the maximum temperature criterion (see Section 7.2). Where the calculated value of T_{max} is too high an increased bearing clearance or bearing length may be required.

The lubricant temperature rise in passing through the bearing, $(T_{out} - T_f)$, is related to the lubricant flow rate, Q , according to the following equation

$$T_{out} - T_f = \frac{KH}{\rho C Q} \quad (6.9)$$

6.6 Film Thickness

The value of the minimum film thickness is required in order that the safe operation of the bearing can be checked (see Section 7). In general, increasing the size of bearing for a specified duty increases the film thickness. However, larger bearings will have a greater power loss than compact bearings.

The minimum film thickness for an aligned bearing, h , is given by,

$$h = (1 - \epsilon) \frac{c_d}{2} \quad (6.10)$$

In practice some misalignment is to be expected but its estimation is outside the scope of this Item. Misalignment can arise from several factors. One of these, the shaft slope due to deflection under load, is dealt with in Reference 2.

With misalignment, the minimum film thickness will be at the edge of the bearing and its value will be less than h . In all cases, especially where there is gross misalignment, this reduced value of film thickness, h_{edge} , should be estimated using Figure 10.

[†] An alternative strategy is to take a new value of T_e equal to the arithmetic mean of the calculated and old values but to base the new assumed value of T_g on Equation (6.7). That is, use

$$(T_g - T_f)_{assume} = \frac{T_g - T_f}{T_e - T_f} (T_e - T_f)_{assumed}$$