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Calculation methods for steadily loaded axial groove hydrodynamic journal bearings. Superlaminar operation (To be used in conjunction with Item No. 84031)

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THE PREPARATION OF THIS DATA ITEM

The work on this particular Item, which supersedes the method given in Item No. 77034, was monitored and guided by the Working Party on Journal Bearings which has the following membership:

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and which was appointed by the Tribology Steering Group. The Tribology Steering Group, which also approved the final draft, first met in 1973 and now has the following membership:

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(Continued on inside Back Cover)

CALCULATION METHODS FOR STEADILY LOADED AXIAL GROOVE HYDRODYNAMIC JOURNAL BEARINGS. SUPERLAMINAR OPERATION

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CALCULATION METHODS FOR STEADILY LOADED AXIAL GROOVE HYDRODYNAMIC JOURNAL BEARINGS. SUPERLAMINAR OPERATION

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.

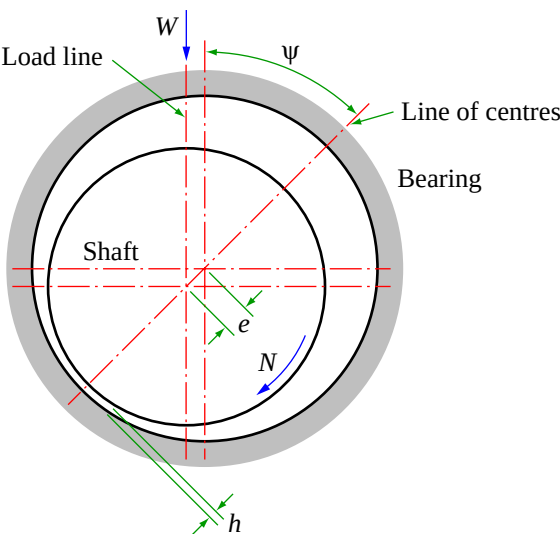
a	axial length of lubricant groove	m
b	axial length of bearing	m
C	specific heat capacity of lubricant	J/kg K
c_d	diametral clearance under running conditions	m
d	diameter of journal	m
e	eccentricity for an aligned bearing (see Sketch 1.1)	m
F	groove function for calculation of Q_p	—
F_H	Reynolds number correction factor for power loss = $(H')_{turb}/(H')_{lam}$	—
F_Q	Reynolds number correction factor for side flow = $(Q'_s)_{turb}/(Q'_s)_{lam}$	—
F_ε	Reynolds number correction factor for eccentricity ratio = $(\varepsilon)_{turb}/(\varepsilon)_{lam}$	—
H	power loss	W
H'	non-dimensional power loss = $\frac{H}{\eta_e N^2 d^2 b} \left(\frac{c_d}{d} \right)$	—
h	film thickness at closest approach for aligned bearing	m
h_s	safe allowable film thickness	m
N	frequency of shaft rotation	rev/s
Pe	Peclet number = $\frac{N c_d^2}{\kappa}$, used in calculation of temperature rise	—
p_f	lubricant feed pressure	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

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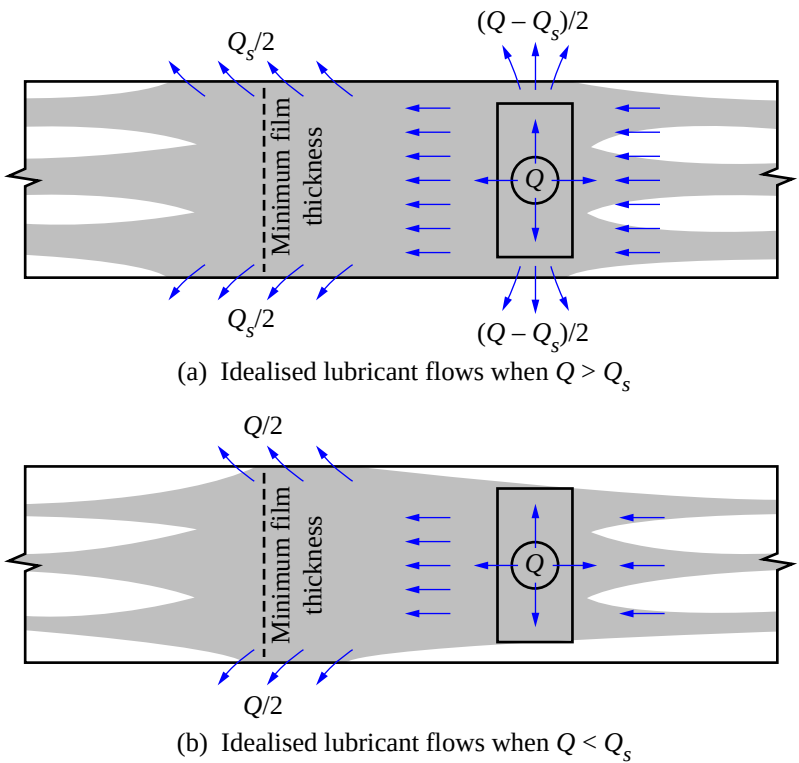
Q'_p	non-dimensional feed pressure flow rate of lubricant from groove(s) = $\frac{Q_p \eta_f}{c_d^3 p_f}$	–
Q_s	idealised side flow rate (see Sketch 1.2)	m ³ /s
Q'_s	non-dimensional idealised side flow rate = $\frac{Q_s}{c_d N b d}$	–
Re	bearing Reynolds number = $\frac{\pi d N c_d \rho}{2 \eta_s}$	–
T_e	effective temperature of lubricant in loaded part of bearing	°C
T_f	lubricant feed temperature	°C
T_{max}	maximum bearing surface temperature	°C
T_{out}	bulk outlet temperature from bearing	°C
T_s	effective side leakage temperature	°C
W	load	N
W'	non-dimensional load = $\frac{W}{\eta_e N b d} \left(\frac{c_d}{d} \right)^2$	–
$\Delta\theta$	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_f	dynamic viscosity of lubricant feed	N s/m ²
η_s	effective dynamic viscosity of side leakage	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 of ESDU 84031, Derivation 4)	–
ψ	attitude angle (angle between load line and line of centres, see Sketch 1.1)	deg

Subscripts

<i>lam</i>	assuming laminar flow conditions
<i>turb</i>	assuming turbulent flow conditions
<i>crit</i>	critical condition beyond which laminar flow theory becomes increasingly inaccurate



Sketch 1.1 Illustration of notation



Sketch 1.2 Idealised lubricant flows

2. INTRODUCTION

This Item extends the scope of ESDU 84031 (Derivation 4) to the analysis of bearings under conditions for which the flow in the lubricating film is not laminar. Such superlaminar conditions can arise at high surface speed or if the lubricant is of very low viscosity.

The flow regime is largely determined by the Reynolds number but the onset of turbulence does not occur at a precisely predictable value of the Reynolds number and the transition is likely to feature a regular vortex pattern before random turbulence is encountered.

The presence of vortices and/or turbulence results in increased power loss and, generally, higher operating temperature than that predicted by laminar flow theory. For design purposes the influence of both regular vortices and random turbulence is similar and the treatment in this Item may be used for superlaminar flow in general.

3. APPLICABILITY

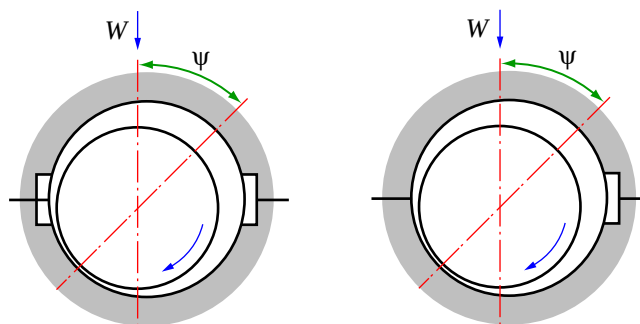
Bearing behaviour in the superlaminar regime is not sufficiently understood to permit performance prediction to be made with as much confidence as in the laminar regime. However, the data presented in this Item have been compared with available experimental results and are considered adequate for design purposes. In general better accuracy may be expected in the fully turbulent regime than in the transition regime.

Available evidence suggests that laminar flow theory may be applied up to the point of appearance of vortices, given by

$$(Re)_{crit} \left(\frac{c_d}{d} \right)^{1/2} = 41 - 38\varepsilon + 64\varepsilon^2 \quad (3.1)$$

and that a transition region extends to twice this value. Beyond this a fully turbulent condition can be assumed. In the transition region, it is to be expected that power loss and operating temperature will lie between values predicted by laminar flow theory (see ESDU 84031, Derivation 4) and by the turbulent flow theory of this Item (see Example 2).

The data of this Item apply to a bearing of circular cylindrical form having two axial grooves located at 90° to the load line. The data may be applied also to a bearing having a single groove located at 90° upstream of the load line (see Sketch 3.1).



Sketch 3.1 Groove arrangements

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4. PERFORMANCE PREDICTION

A procedure similar to that given in ESDU 84031 (see Derivation 4) may be followed for performance prediction so a step-by-step description is not repeated in this Item. ESDU 84031 may also be consulted for more detailed information concerning the principles of hydrodynamic journal bearing design and recommendations concerning allowable film thickness and operating temperature.

It is assumed that the bearing dimensions are known or that a choice of dimensions has been made from an initial selection procedure such as that in ESDU 84031. These values are used to obtain the performance characteristics and to check the design for the required duty. The procedure to be followed is shown in the flowchart of Figure 1 and differs from the procedure of ESDU 84031 only in the Reynolds number correction to be applied to the eccentricity, power loss and the idealised side flow.

4.1 Reynolds Number, Effective Film Temperature and Effective Side Leakage Temperature

For bearing design purposes the relevant Reynolds number is based on the lubricant viscosity at the effective side leakage temperature, T_s . The effective side leakage temperature differs from the effective film temperature, T_e , the latter being used to define the effective film viscosity for the purposes of bearing performance analysis. In order to apply the superlaminar data of this Item it is, therefore, necessary to assume an initial value, not only for T_e , but also for T_s and then refine these values using an iterative procedure.

It is recommended that an initial value for the effective temperature, T_e , be obtained by first applying the laminar analysis, unmodified, of ESDU 84031. This same value may then be used as the initial value for the effective side leakage temperature, T_s .

4.2 Eccentricity Ratio

For a given value of the non-dimensional load, W' , the eccentricity ratio under turbulent conditions is less than for laminar conditions. The eccentricity ratio under laminar conditions, $(\varepsilon)_{lam}$, may be found from Figure 6 of ESDU 84031. This is then multiplied by the factor F_ε obtained by interpolation either from Figure 2 or from Table A.1. That is

$$(\varepsilon)_{turb} = (\varepsilon)_{lam} F_\varepsilon. \quad (4.1)$$

4.3 Power Loss

Depending on the groove arrangement, Figure 7 or 8 of ESDU 84031 may be used to find the non-dimensional power loss for laminar conditions, $(H')_{lam}$. The corresponding value for turbulent conditions may then be calculated using the data of Figure 3 or Table A.2. That is

$$(H')_{turb} = (H')_{lam} F_H. \quad (4.2)$$

4.4 Lubricant Flow Rate

The feed pressure flow rate, Q_p , may be obtained using the method presented in Section 6.4 of ESDU 84031. The accuracy of determination of Q_p decreases at high Reynolds number but so also does the significance of Q_p .

The non-dimensional idealised side flow rate for laminar flow, $(Q'_s)_{lam}$, is obtained from Figure 10 of ESDU 84031. The corresponding value for turbulent flow is given by

$$(Q'_s)_{turb} = (Q'_s)_{lam} F_Q, \quad (4.3)$$

where
$$F_Q = 1 + 0.0095 \varepsilon \left(\frac{b}{d}\right) (Re)^{0.424}.$$

4.5 Temperatures Within Bearing

The maximum bearing temperature, T_{max} , and the effective film temperature, T_e , may be obtained using the method presented in Section 6.5 of ESDU 84031. As recommended in that section the calculations should be repeated with a value of T_e equal to the arithmetic mean of the calculated and the assumed values until the change in T_e is less than 2 K. The value of T_s to use in this recalculation should be based on the new assumed value of T_e and is given by

$$T_s - T_f = \frac{5 - \Phi}{4} (T_e - T_f), \quad (4.4)$$

where the value of Φ will have been obtained during the computation of T_e using ESDU 84031.

4.6 Film Thickness

Guidance on determining the minimum film thickness is given in Section 6.6 of ESDU 84031.

4.7 Shaft Position

The position of the shaft centre may be determined from a knowledge of the eccentricity ratio, ε , and the attitude angle, ψ . The attitude angle for turbulent conditions, $(\psi)_{turb}$, is larger than for laminar conditions and is given, approximately, by

$$(\psi)_{turb} = (\psi)_{lam} + 1.5 \left(\frac{Re}{1000}\right)^{1/2} \quad (4.5)$$

where ψ is in deg and $(\psi)_{lam}$ is obtained from Figure 13 of ESDU 84031.

5. DERIVATION

The derivation lists selected sources that have assisted in the preparation of this Item.

1. JONES, G.J.
MARTIN, F.A. Turbulent journal bearings: dimensionless steady-state performance charts for the two-axial groove case. *Transactions of the ASLE*, Vol 22, No 2, pp. 163–174, April 1980.
2. NEAL, P.B. Unpublished work at Sheffield University, 1985.
3. WILCOCK, D.F. Turbulence in high-speed journal bearings. *Transactions of the ASME*, paper No. 49, A–37, pp. 825–834, August 1950.
4. ESDU Calculation methods for steadily loaded axial groove hydrodynamic journal bearings. ESDU 84031, ESDU, London, November 1984.