INTERNATIONAL STANDARD

ISO 7902-1

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Hydrodynamic plain journal bearings under steady-state conditions — Circular cylindrical bearings —

Part 1:
Calculation procedure

Paliers lisses hydrodynamiques radiaux fonctionnant en régime stabilisé — Paliers circulaires cylindriques —

Partie 1: Méthode de calcul

Reference number
ISO 7902-1:1998(E)

Foreword

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Draft International Standards adopted by the technical committees are circulated to the member bodies for voting. Publication as an International Standard requires approval by at least 75 % of the member bodies casting a vote.

International Standard ISO 7902-1 was prepared by Technical Committee ISO/TC 123, *Plain bearings*, Subcommittee SC 4, *Methods of calculation of plain bearings*.

ISO 7902 consists of the following parts, under the general title *Hydrodynamic plain journal bearings under steady-state conditions — Circular cylindrical bearings*:

— *Part 1: Calculation procedure*

— *Part 2: Functions used in the calculation procedure*

— *Part 3: Permissible operational parameters*

Annexes A and B of this part of ISO 7902 are for information only.

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Hydrodynamic plain journal bearings under steady-state conditions — Circular cylindrical bearings —

Part 1:
Calculation procedure

1 Scope

This part of ISO 7902 specifies a calculation procedure for oil-lubricated hydrodynamic plain bearings, with complete separation of the shaft and bearing sliding surfaces by a film of lubricant, used for designing plain bearings that are reliable in operation.

It deals with circular cylindrical bearings having angular spans Ω, of 360°, 180°, 150°, 120° and 90°, the arc segment being loaded centrally. Their clearance geometry is constant except for negligible deformations resulting from lubricant film pressure and temperature.

The calculation procedure serves to dimension and optimize plain bearings, in turbines, generators, electric motors, gear units, rolling mills, pumps and other machines. It is limited to steady-state operation, i.e. under continuously driven operating conditions, with magnitude and direction of loading as well as the angular speeds of all rotating parts constant. It can also be applied if a full plain bearing is subjected to a constant force rotating at any speed. Dynamic loadings, i.e. those whose magnitude and direction vary with time, such as can result from vibration effects and instabilities of rapid-running rotors, are not taken into account.

2 Normative references

The following standards contain provisions which, through reference in this text, constitute provisions of this part of ISO 7902. At the time of publication, the editions indicated were valid. All standards are subject to revision, and parties to agreements based on this part of ISO 7902 are encouraged to investigate the possibility of applying the most recent editions of the standards indicated below. Members of IEC and ISO maintain registers of currently valid International Standards.

ISO 3448:1992, *Industrial liquid lubricants — ISO viscosity classification*.

ISO 7902-2:1998, *Hydrodynamic plain journal bearings under steady-state conditions — Circular cylindrical bearings — Part 2: Functions used in the calculation procedure*.

ISO 7902-3:1998, *Hydrodynamic plain journal bearings under steady-state conditions — Circular cylindrical bearings — Part 3: Permissible operational parameters*.

ISO 7904-2:1995, *Plain bearings — Symbols — Part 2: Applications*.

3 Basis of calculation, assumptions and preconditions

3.1 The basis of calculation is the numerical solution to Reynolds' differential equation for a finite bearing length, taking into account the physically correct boundary conditions for the generation of pressure:

$$\frac{\partial}{\partial x}\left(h^3 \frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial z}\left(h^3 \frac{\partial p}{\partial z}\right) = 6\eta\left(u_\mathrm{J} + u_\mathrm{B}\right)\frac{\partial h}{\partial x} \qquad \text{. . . (1)}$$

The symbols are given in clause 5.

See [1] to [3], and [11] to [14] in annex B, for the derivation of Reynolds' differential equation and [4] to [6], [12] and [13] for its numerical solution.

3.2 The following idealizing assumptions and preconditions are made, the permissibility of which has been sufficiently confirmed both experimentally and in practice.

a) The lubricant corresponds to a Newtonian fluid.

b) All lubricant flows are laminar.

c) The lubricant adheres completely to the sliding surfaces.

d) The lubricant is incompressible.

e) The lubricant clearance gap in the loaded area is completely filled with lubricant. Filling up of the unloaded area depends on the way the lubricant is supplied to the bearing.

f) Inertia effects, gravitational and magnetic forces of the lubricant are negligible.

g) The components forming the lubrication clearance gap are rigid or their deformation is negligible; their surfaces are ideal circular cylinders.

h) The radii of curvature of the surfaces in relative motion are large in comparison with the lubricant film thicknesses.

i) The lubricant film thickness in the axial direction (z-coordinate) is constant.

j) Fluctuations in pressure within the lubricant film normal to the bearing surfaces (y-coordinate) are negligible.

k) There is no motion normal to the bearing surfaces (y-coordinate).

l) The lubricant is isoviscous over the entire lubrication clearance gap.

m) The lubricant is fed in at the start of the bearing liner or where the lubrication clearance gap is widest; the magnitude of the lubricant feed pressure is negligible in comparison with the lubricant film pressures.

3.3 The boundary conditions for the generation of lubricant film pressure fulfil the following continuity conditions:

— at the leading edge of the pressure profile: $p(\varphi_1, z) = 0$

— at the bearing rim: $p(\varphi, z = \pm B/2) = 0$

— at the trailing edge of the pressure profile: $p[\varphi_2(z), z] = 0$

— and $\partial p/\partial\varphi[\varphi_2(z), z] = 0$

For some types and sizes of bearings, the boundary conditions may be specified.

In partial bearings, if the following expression is satisfied:

$$\varphi_2 - (\pi - \beta) < \frac{\pi}{2}$$

then the trailing edge of the pressure profile lies at the outlet end of the bearing:

$$p(\varphi = \varphi_2, z) = 0$$

3.4 The numerical integration of the Reynolds' differential equation is carried out — possibly by applying transformation of pressure as suggested in [3], [11] and [12] — by a transformation to a differential equation which is applied to a grid system of supporting points, and which results in a system of linear equations. The number of supporting points is significant to the accuracy of the numerical integration: the use of a non-equidistant grid as given in [6] and [13] is advantageous. After substituting the boundary conditions at the trailing edge of the pressure profile, integration yields the pressure distribution in the circumferential and axial directions.

The application of the similarity principle to hydrodynamic plain bearing theory results in dimensionless magnitudes of similarity for parameters of interest such as load-carrying capacity, frictional behaviour, lubricant flow rate, relative bearing length, etc. The application of magnitudes of similarity reduces the number of numerical solutions required of Reynolds' differential equation (see ISO 7902-2). Other solutions may also be applied, provided they fulfil the conditions laid down in ISO 7902-2 and are of a similar numerical accuracy.

3.5 ISO 7902-3 includes permissible operational parameters towards which the result of the calculation shall be oriented in order to ensure correct functioning of the plain bearings.

In special cases, operational parameters deviating from ISO 7902-3 may be agreed upon for specific applications.

4 Calculation procedure

4.1 By calculation is understood determination of correct operation by computation using actual operating parameters (see figure 1) which can be compared with operational parameters. The operating parameters determined under varying operating conditions must therefore lie within the range of permissibility as compared with the operational parameters. To this end, all operating conditions during continuous operation shall be investigated.

4.2 Freedom from wear is guaranteed only if complete separation of the mating bearing parts is achieved by the lubricant. Continuous operation in the mixed friction range results in failure. Short-time operation in the mixed friction range, for example starting up and running down machines with plain bearings, is unavoidable and does not generally result in bearing damage. When a bearing is subjected to heavy load, an auxiliary hydrostatic arrangement may be necessary for starting up and running down at a slow speed. Running-in and adaptive wear to compensate for deviations of the surface geometry from the ideal are permissible as long as they are limited in area and time and occur without overloading effects. In certain cases, a specific running-in procedure may be beneficial, depending on the choice of materials.

4.3 The limits of mechanical loading are a function of the strength of the bearing material. Slight permanent deformations are permissible as long as they do not impair correct functioning of the plain bearing.

4.4 The limits of thermal loading result from the thermal stability of the bearing material but also from the viscosity-temperature relationship and by degradation of the lubricant.

4.5 A correct calculation for plain bearings presupposes that the operating conditions are known for all cases of continuous operation. In practice, however, additional influences frequently occur, which are unknown at the design stage and cannot always be predicted. The application of an appropriate safety margin between the actual operating parameters and permissible operational parameters is recommended. Influences include, for example:

— spurious forces (out-of-balance, vibrations, etc.);

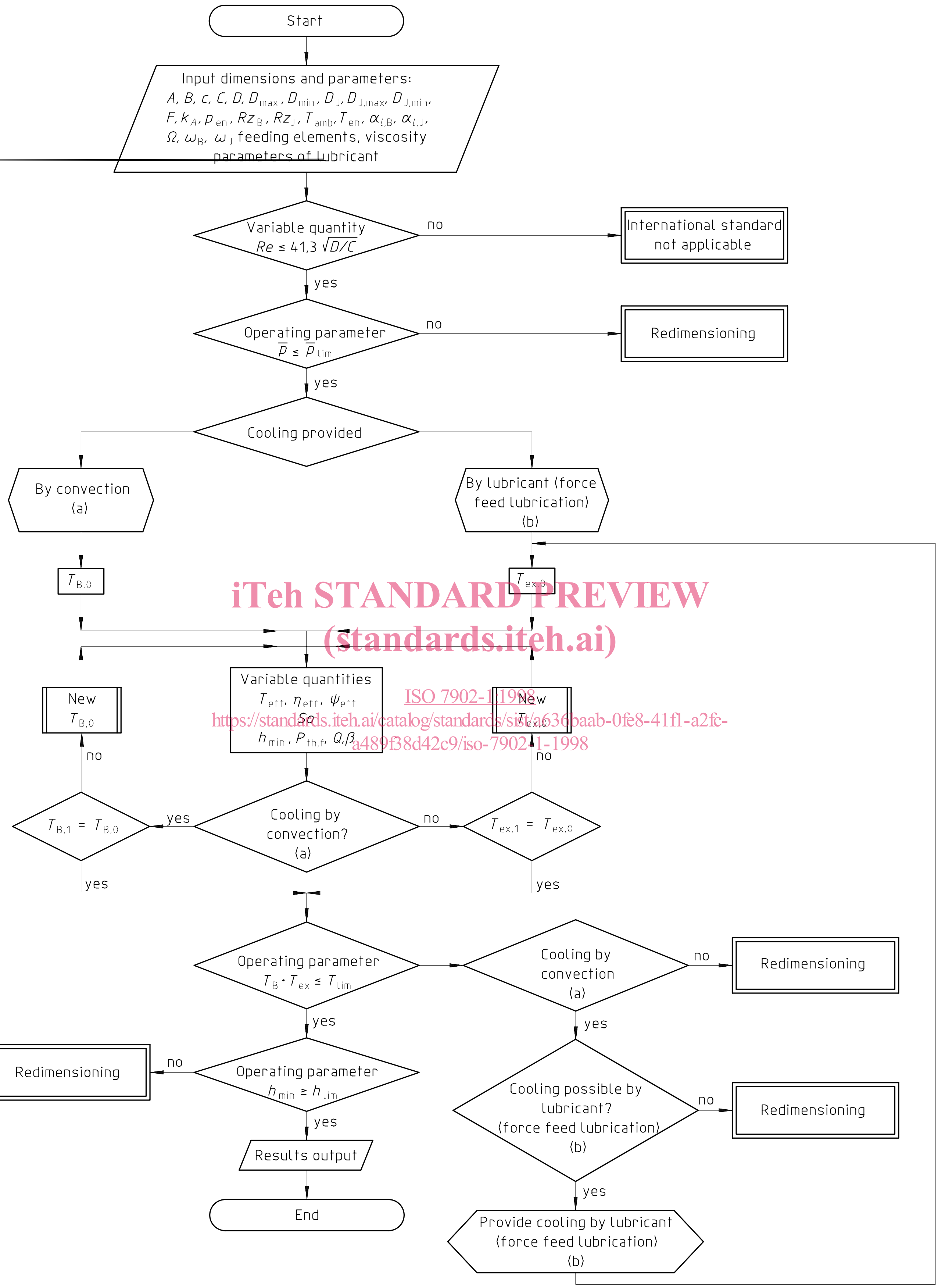


Figure 1 — Outline of calculation

— deviations from the ideal geometry (machining tolerances, deviations during assembly, etc.);

— lubricants contaminated by dirt, water, air, etc.;

— corrosion, electrical erosion, etc.;

Data on other influencing factors are given in 6.7.

4.6 Reynolds' number shall be used to verify that ISO 7902-2, for which laminar flow in the lubrication clearance gap is a necessary condition, can be applied:

$$Re = \frac{\rho U_J \frac{C_{R,\mathrm{eff}}}{2}}{\eta} = \frac{\pi D N_J \frac{C_{R,\mathrm{eff}}}{2}}{\nu} \leqslant 41{,}3\sqrt{\frac{D}{C_{R,\mathrm{eff}}}} \qquad \text{. . . (2)}$$

In the case of plain bearings with $Re > 41{,}3\sqrt{D / C_{R,\mathrm{eff}}}$ (for example as a result of high peripheral speed) higher loss coefficients and bearing temperatures must be expected. Calculations for bearings with turbulent flow cannot be carried out in accordance with this part of ISO 7902.

4.7 The plain bearing calculation takes into account the following factors (starting with the known bearing dimensions and operational data):

— the relationship between load-carrying capacity and lubricant film thickness;

— the frictional power rate;

— the lubricant flow rate;

— the heat balance.

All these factors are mutually dependent.

The solution is obtained using an iterative method; the sequence is outlined in the flow chart in figure 1.

For optimization of individual parameters, parameter variation can be applied: modification of the calculation sequence is possible.

5 Symbols and units

See figure 2 and table 1.

Minimum lubricant film thickness, h_{min}:

$$h_{\mathrm{min}} = \frac{D - D_J}{2} - e = 0{,}5 D \psi (1 - \varepsilon)$$

where the relative eccentricity, ε, is given by

$$\varepsilon = \frac{e}{\frac{D - D_J}{2}}$$

If

$$\varphi_2 - (\pi - \beta) < \frac{\pi}{2}$$

then

$$h_{\text{min}} = 0{,}5 D\psi(1 + \varepsilon \cos \varphi_2)$$

6 Definition of symbols

6.1 Load-carrying capacity

A characteristic parameter for the load-carrying capacity is the dimensionless Sommerfeld number, So:

$$So = \frac{F\psi_{\text{eff}}^2}{DB\eta_{\text{eff}}\omega_{\text{h}}} = So\left(\varepsilon, \frac{B}{D}, \Omega\right) \qquad \text{. . . (3)}$$

Values of So as a function of the relative eccentricity ε, the relative bearing length B/D and the angular span of bearing segment Ω are given in ISO 7902-2. The variables ω_{h}, η_{eff} and ψ_{eff} take into account thermal effects and the angular velocities of shaft, bearing and bearing force (see 6.4 and 6.7).

The relative eccentricity ε describes, together with the attitude angle β (see ISO 7902-2), the magnitude and position of the minimum thickness of lubricant film. For a full bearing (Ω = 360 °C), the oil should be introduced at the greatest lubricant clearance gap or, with respect to the direction of rotation, shortly before it. For this reason it is useful to know the attitude angle β.

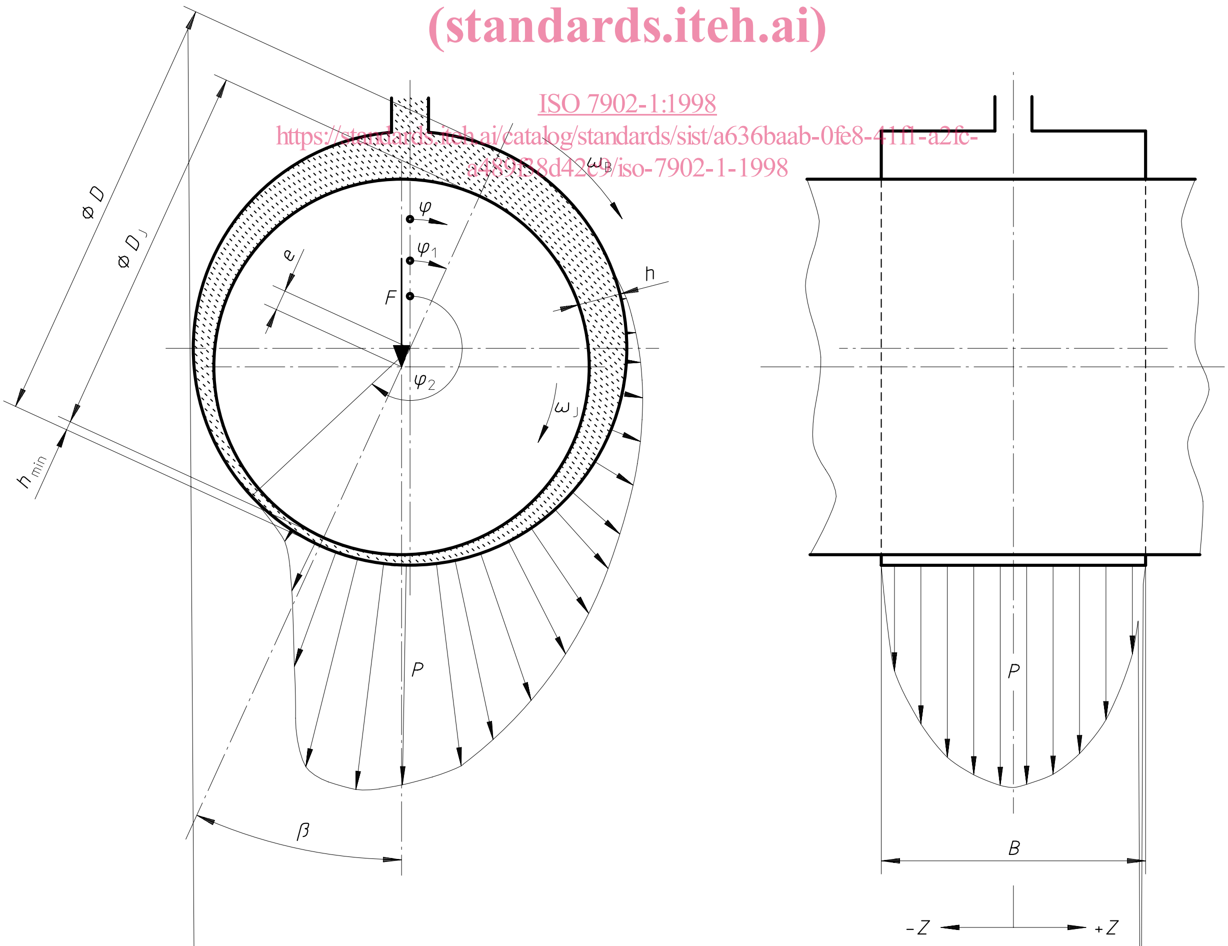


Figure 2 — Illustration of symbols

Table 1 — Symbols and their designations

Symbol	Designation	Unit
A	Area of heat-emitting surface (bearing housing)	m^2
b_G	Width of oil groove	m
B	Nominal bearing width	m
c	Specific heat capacity of the lubricant	J/(kg·K)
C	Nominal bearing clearance	m
$C_{R,eff}$	Effective bearing radial clearance	m
d_L	Oil hole diameter	m
D	Nominal bearing diameter (inside diameter)	m
D_J	Nominal shaft diameter	m
$D_{J,max}$	Maximum value of D_J	m
$D_{J,min}$	Minimum value of D_J	m
D_{max}	Maximum value of D	m
D_{min}	Minimum value of D	m
e	Eccentricity between the axis of the shaft and the bearing axis	m
E	Modulus of elasticity	1
f	Coefficient of friction	1
F	Bearing force (nominal load)	N
F_f	Friction force in the loaded area of the lubricant film	N
F_f'	Frictional force in the unloaded area of the lubricant film	N
G	Shear modulus	1
h	Local lubricant film thickness	m
h_{lim}	Minimum permissible lubricant film thickness	m
h_{min}	Minimum lubricant film thickness	m
h_{wav}	Waviness of sliding surface	m
$h_{wav,eff}$	Effective waviness of sliding surface	m
$h_{wav,eff,lim}$	Maximum permissible effective waviness	m
k_A	Outer heat transmission coefficient	W/(m^2·K)
l_G	Length of oil groove	m
l_P	Length of oil pocket	m
L_H	Length of bearing housing	m
N_B	Rotational frequency of the bearing	m/s
N_F	Rotational frequency of the bearing force	m/s
N_J	Rotational frequency of the shaft	m/s
p	Local lubricant film pressure	Pa
$\overline{p}$	Specific bearing load	Pa
p_{en}	Lubricant feed pressure	Pa
p_{lim}	Maximum permissible lubricant film pressure	Pa
$\overline{p}_{lim}$	Maximum permissible specific bearing load	Pa
P_f	Frictional power	W
P_{th}	Heat flow rate	W
$P_{th,amb}$	Heat flow rate to the ambient	W
$P_{th,f}$	Heat flow rate due to frictional power	W
$P_{th,L}$	Heat flow rate in the lubricant	W
Q	Lubricant flow rate	m^3/s
Q_1	Lubricant flow rate at the inlet to clearance gap	m^3/s
Q_2	Lubricant flow rate at the outlet to clearance gap	m^3/s
Q_3	Lubricant flow rate due to hydrodynamic pressure	m^3/s
Q_3^*	Lubricant flow rate parameter due to hydrodynamic pressure	1

Symbol	Designation	Unit
Q_p	Lubricant flow rate due to feed pressure	m^3/s
Q_p^*	Lubricant flow rate parameter due to feed pressure	1
Rz_B	Average peak-to-valley height of bearing sliding surface	m
Rz_J	Average peak-to-valley height of shaft mating surface	m
Re	Reynolds number	1
So	Sommerfeld number	1
T_{amb}	Ambient temperature	°C
T_B	Bearing temperature	°C
$T_{B,0}$	Assumed initial bearing temperature	°C
$T_{B,1}$	Calculated bearing temperature resulting from iteration procedure	°C
T_{en}	Lubricant temperature at bearing entrance	°C
T_{ex}	Lubricant temperature at bearing exit	°C
$T_{ex,0}$	Assumed initial lubricant temperature at bearing exit	°C
$T_{ex,1}$	Calculated lubricant temperature at bearing exit	°C
T_J	Shaft temperature	°C
T_{lim}	Maximum permissible bearing temperature	°C
$\bar{T}_L$	Mean lubricant temperature	°C
U_B	Linear velocity (peripheral speed) of bearing	m/s
U_J	Linear velocity (peripheral speed) of shaft	m/s
v_a	Air ventilating velocity	m/s
x	Coordinate parallel to the sliding surface in circumferential direction	m
y	Coordinate perpendicular to the sliding surface	m
z	Coordinate parallel to the sliding surface in axial direction	m
$\alpha_{l,B}$	Linear heat expansion coefficient of the bearing	K^{-1}
$\alpha_{l,J}$	Linear heat expansion coefficient of the shaft	K^{-1}
β	Attitude angle (angular position of the shaft eccentricity related to the direction of load)	°
δ_J	Angle of misalignment of the shaft	rad
ε	Relative eccentricity	1
η	Dynamic viscosity of the lubricant	Pa·s
η_{eff}	Effective dynamic viscosity of the lubricant	Pa·s
ν	Kinematic viscosity of the lubricant	Pa·s
ξ	Coefficient of resistance to rotation in the loaded area of the lubricant film	1
ξ'	Coefficient of resistance to rotation in the unloaded area of the lubricant film	1
ξ_G	Coefficient of resistance to rotation in the area of circumferential groove	1
ξ_P	Coefficient of resistance to rotation in the area of the pocket	1
ρ	Density of lubricant	kg/m^3
φ	Angular coordinate in circumferential direction	rad
φ_1	Angular coordinate of pressure leading edge	rad
φ_2	Angular coordinate of pressure trailing edge	rad
ψ	Relative bearing clearance	1
$\bar{\psi}$	Mean relative bearing clearance	1
ψ_{eff}	Effective relative bearing clearance	1
ψ_{max}	Maximum relative bearing clearance	1
ψ_{min}	Minimum relative bearing clearance	1
ω_B	Angular velocity of bearing	s^{-1}
ω_h	Hydrodynamic angular velocity	s^{-1}
ω_J	Angular velocity of shaft	s^{-1}
Ω	Angular span of bearing segment	°
Ω_G	Angular span of lubrication groove	°
Ω_P	Angular span of lubrication pocket	°

6.2 Frictional power loss

Friction in a hydrodynamic plain bearing due to viscous shear stress is given by the coefficient of friction $f = F_f/F$ and the derived non-dimensional characteristics of frictional power loss ξ and f/ψ_{eff}.

$$\xi = \frac{F_f \psi_{eff}}{DB\eta_{eff}\omega_h} \qquad \text{. . . (4)}$$

$$\frac{f}{\psi_{eff}} = \frac{\xi}{So} \qquad \text{. . . (5)}$$

They are applied if the frictional power loss is encountered only in the loaded area of the lubricant film.

It is still necessary to calculate frictional power loss in both the loaded and unloaded areas then the values

$$f, F_f, \xi, \frac{f}{\psi_{eff}}$$

are substituted by

$$f', F_f', \xi', \frac{f'}{\psi_{eff}}$$

in equations (4) and (5). This means that the whole of the clearance gap is filled with lubricant.

The values of f/ψ_{eff} and f'/ψ_{eff} for various values of ε, B/D and Ω are given in ISO 7902-2. It also gives the approximation equations, based on [15], which are used to determine frictional power loss values in the bearings taking account of the influence of lubricating pockets and grooves.

The frictional power in a bearing or the amount of heat generated is given by

$$P_f = P_{th,f} = fF \qquad \text{. . . (6)}$$

$$P_f' = f'F \qquad \text{. . . (7)}$$

6.3 Lubricant flow rate

The lubricant fed to the bearing forms a film of lubricant separating the sliding surfaces. The pressure build-up in this film forces lubricant out of the ends of the bearing. This is the proportion Q of the lubricant flow rate resulting from build-up of hydrodynamic pressure.

$$Q_3 = D^3 \psi_{eff} \omega_h Q_3^* \qquad \text{. . . (8)}$$

where $Q_3^* = Q_3^* (\varepsilon, B/D, \Omega_1)$ is given in ISO 7902-2.

There is also a flow of lubricant in the peripheral direction through the narrowest clearance gap into the diverging, pressure-free gap. For increased loading and with a small lubrication gap clearance, however, this proportion of the lubricant flow is negligible.

The lubricant feed pressure p_{en} forces additional lubricant out of the ends of the plain bearing. This is the amount Q_p of the lubricant flow rate resulting from feed pressure:

$$Q_p = \frac{D^3 \psi_{eff}^3 p_{en}}{\eta_{eff}} Q_p^* \qquad \text{. . . (9)}$$

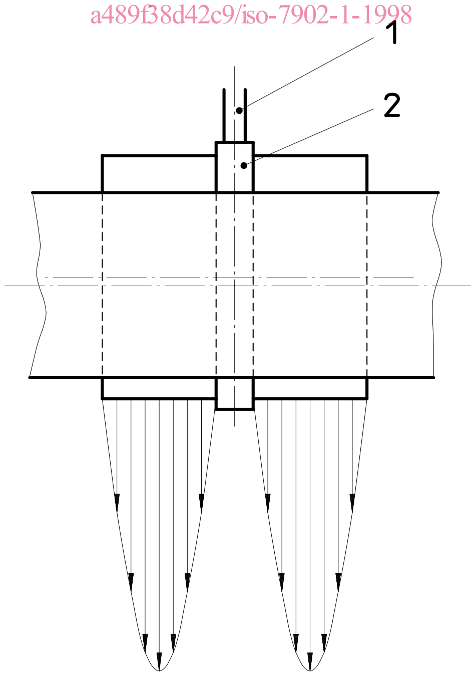
where $Q_p^* = Q_p^* (\varepsilon, B/D, \Omega)$ is given in ISO 7902-2.

6.3.1 Lubricant feed elements are lubrication holes, lubrication grooves and lubrication pockets. The lubricant feed pressure p_{en} should be markedly less than the specific bearing load $\overline{p}$, to avoid additional hydrostatic loads. Usually p_{en} lies between 0,05 MPa and 0,2 MPa. The depth of the lubrication grooves and lubrication pockets is considerably greater than the bearing clearance.

6.3.2 Lubrication grooves are elements designed to distribute lubricant in the circumferential direction. The recesses machined into the sliding surface run circumferentially and are kept narrow in the axial direction. If lubrication grooves are located in the vicinity of pressure rise, the pressure distribution is split into two independent pressure “hills” and the load-carrying capacity is markedly reduced (see figure 3). In this case, the calculation shall be carried out for half the load applied to each half bearing. However, because of the build-up of hydrodynamic pressure, Q_3, only half of the lubricant flow rate shall be taken into account when balancing heat losses (see 6.4), since the return into the lubrication groove plays no part in dissipating heat. It is more advantageous, for a full bearing, to arrange the lubrication groove in the unloaded part. The entire lubricant flow amount Q_p goes into the heat balance.

6.3.3 Lubrication pockets are elements for distributing the lubricant over the length of the bearing. The recesses machined into the sliding surface are oriented in the axial direction and should be as short as possible in the circumferential direction. Relative pocket lengths should be such shall $b_p/B < 0{,}7$. Although larger values increase the oil flow rate, the oil emerging over the narrow, restricting webs at the ends plays no part in dissipating heat. This is even more true if the end webs are penetrated axially. For full bearings ($\Omega = 360°$), a lubrication pocket opposite to the direction of load as well as two lubrication pockets normal to the direction of loading are machined in. Since the lubricant flow rate, even in the unloaded part of the bearing, provides for the dissipation of frictional heat arising from shearing, the lubricating pockets shall be fully taken into account in the heat balance. For shell segments ($\Omega < 360°$) the lubricant flow rate due to feed pressure through lubrication pockets at the inlet or outlet of the shell segment makes practically no contribution to heat dissipation, since the lubrication pockets are scarcely restricted at the segment ends and the greater proportion of this lubricant flow emerges directly.

If the lubricant fills the loaded area of the bearing and there is no lubricant in the unloaded part then the heat dissipation counts as lubricant flow rate in the loaded part only.



Key

1 Lubrication hole
2 Lubrication groove