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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

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Contents

Page

Foreword

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This document was prepared by Technical Committee ISO/TC 123, *Plain bearings*, Subcommittee SC 8, *Calculation methods for plain bearings and their applications*.

This fourth edition cancels and replaces the third edition (ISO 7902-1:2020), of which it constitutes a minor revision.

The changes are as follows:

— the bibliography has been updated and cross references have been renumbered.

A list of all parts in the ISO 7902 series can be found on the ISO website.

Any feedback or questions on this document should be directed to the user’s national standards body. A complete listing of these bodies can be found at www.iso.org/members.html.

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

Part 1:
Calculation procedure

1 Scope

This document 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 provide dimensions 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 the 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 those that can result from vibration effects and instabilities of rapid-running rotors, are not taken into account.

NOTE Equivalent calculation procedures exist that enable operating conditions to be estimated and checked against acceptable conditions. The use of them is equally admissible.

2 Normative references

The following documents are referred to in the text in such a way that some or all of their content constitutes requirements of this document. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies.

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

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

3 Terms and definitions

No terms and definitions are listed in this document.

ISO and IEC maintain terminology databases for use in standardization at the following addresses:

— ISO Online browsing platform: available at https://www.iso.org/obp

— IEC Electropedia: available at https://www.electropedia.org/

4 Symbols and units

Symbols and units are defined in Figure 1 and Table 1.

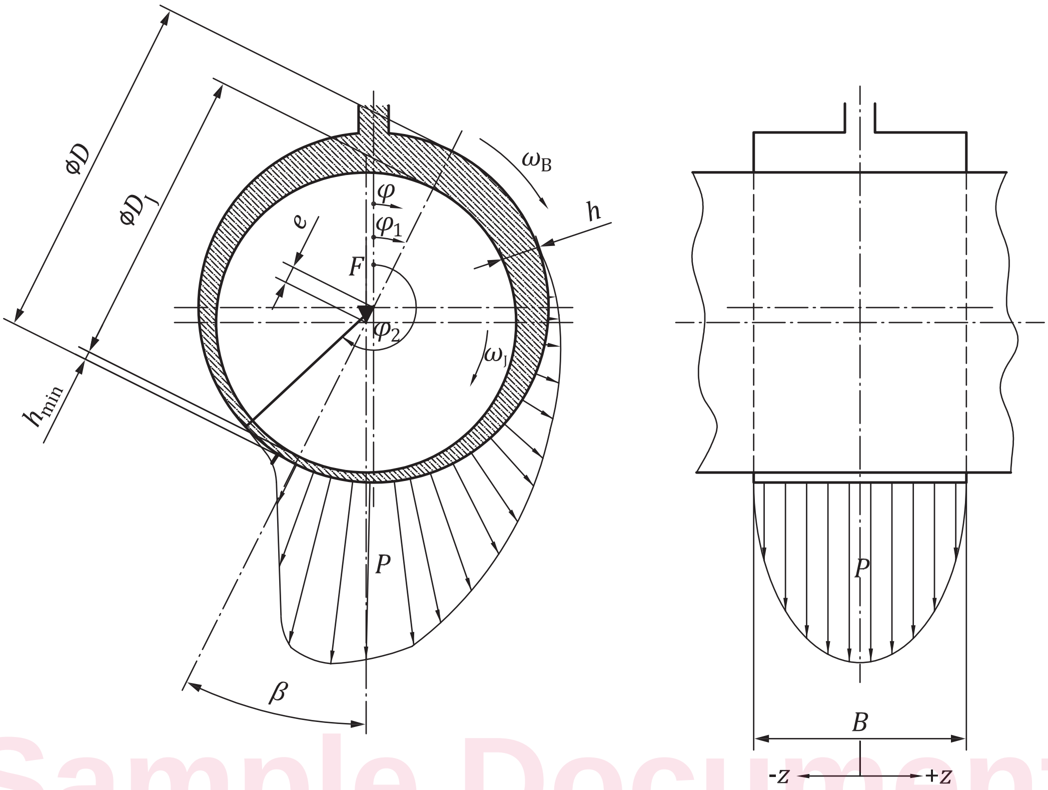

Figure 1 — Illustration of symbols

Table 1 — Symbols and their designations

Symbol	Designation	Unit
A	Area of heat-emitting surface (bearing housing)	m^2
B	Nominal bearing width	m
B_H	Length of the axial housing	m
b_G	Width of lubrication groove	m
b_P	Width of lubrication pocket	m
C	Nominal bearing clearance	m
$C_{R,eff}$	Effective bearing radial clearance	m
c_p	Specific heat capacity of the lubricant	J/(kg·K)
D	Nominal bearing diameter (inside diameter)	m
D_H	Length of the outside diameter of the housing	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
d_L	Lubrication hole diameter	m
e	Eccentricity between the axis of the shaft and the bearing axis	m
F	Bearing force (nominal load)	N

Table 1 *(continued)*

Symbol	Designation	Unit
F_f	Friction force in the loaded area of the lubricant film	N
F_f'	Frictional force in both the loaded and the unloaded area of the lubricant film	N
f	Coefficient of friction in the loaded area of the lubricant film ($f = F_f/F$)	1
f'	Coefficient of friction in both the loaded and unloaded area of the lubricant film	1
H	Length of the total height of the pedestal bearing	m
h	Local lubricant film thickness	m
h_{eff}	Effective lubricant film thickness	m
h_G	Depth of lubrication groove	m
h_{lim}	Minimum permissible lubricant film thickness	m
h_{min}	Minimum lubricant film thickness	m
h_P	Depth of lubrication pocket	m
k_A	Heat transfer coefficient	W/(m²·K)
N_B	Rotational frequency of the bearing	s^{-1}
N_J	Rotational frequency of the shaft	s^{-1}
P_f	Frictional power	W
P_f'	Frictional power in both the loaded and the unloaded area of the lubricant film	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
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
Q	Lubricant flow rate	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
Q_p	Lubricant flow rate due to feed pressure	m^3/s
Q_p^*	Lubricant flow rate parameter due to feed pressure	1
q_L	Coefficient related to lubricant flow rate due to feed pressure	1
q_P	Coefficient related to lubricant flow rate from pocket	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
So_u	Transition 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_{eff}	Effective lubricant temperature	°C
T_{en}	Lubricant temperature at bearing entrance	°C

Table 1 *(continued)*

Symbol	Designation	Unit
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
$\overline{T}_L$	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 the circumferential direction	m
y	Coordinate perpendicular to the sliding surface	m
z	Coordinate parallel to the sliding surface in the 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)	°
ε	Relative eccentricity [$\varepsilon = 2e/(D - D_J)$]	1
ε_u	Transition 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	m^2/s
ξ	Coefficient of resistance to rotation in the loaded area of the lubricant film	1
ξ'	Coefficient of resistance to rotation in both the loaded and 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 the 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
$\overline{\psi}$	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_F	Angular velocity of rotating force	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	°

5 Basis of calculation, assumptions, and preconditions

5.1 Reynolds equation

The basis of calculation is the numerical solution to Reynolds equation for a finite bearing length, taking into account the physically correct boundary conditions for the generation of pressure. Reynolds equation is defined as Formula (1).

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

See References [2] to [4] and References [9] to [12] for the derivation of Reynolds equation and References [5] to [7], [10] and [11] for its numerical solution.

5.2 Assumptions and preconditions

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.

5.3 Boundary conditions

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

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

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