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Hidrodinamični radialni drsni ležaji za neprekinjeno obratovanje - Valjasti ležaji - 1. del: Postopek dimenzioniranja

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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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 —
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Foreword

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The procedures used to develop this document and those intended for its further maintenance are described in the ISO/IEC Directives, Part 1. In particular, the different approval criteria needed for the different types of ISO documents should be noted. This document was drafted in accordance with the editorial rules of the ISO/IEC Directives, Part 2 (see www.iso.org/directives).

Attention is drawn to the possibility that some of the elements of this document may be the subject of patent rights. ISO shall not be held responsible for identifying any or all such patent rights. Details of any patent rights identified during the development of the document will be in the Introduction and/or on the ISO list of patent declarations received (see www.iso.org/patents).

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For an explanation of the voluntary nature of standards, the meaning of ISO specific terms and expressions related to conformity assessment, as well as information about ISO's adherence to the World Trade Organization (WTO) principles in the Technical Barriers to Trade (TBT), see www.iso.org/iso/foreword.html.

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

This third edition cancels and replaces the second edition (ISO 7902-1:2013), which has been technically revised.

The main changes compared to the previous edition are as follows:

— subclause titles have been added;

— symbols have been corrected and added in Table 1;

— calculation values in Annex A have been corrected;

— adjustments have been made to ISO/IEC Directives, Part 2:2018;

— typographical errors have been corrected.

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.

INTERNATIONAL STANDARD ISO 7902-1:2020(E)

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 terminological databases for use in standardization at the following addresses:

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

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

4 Symbols and units

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

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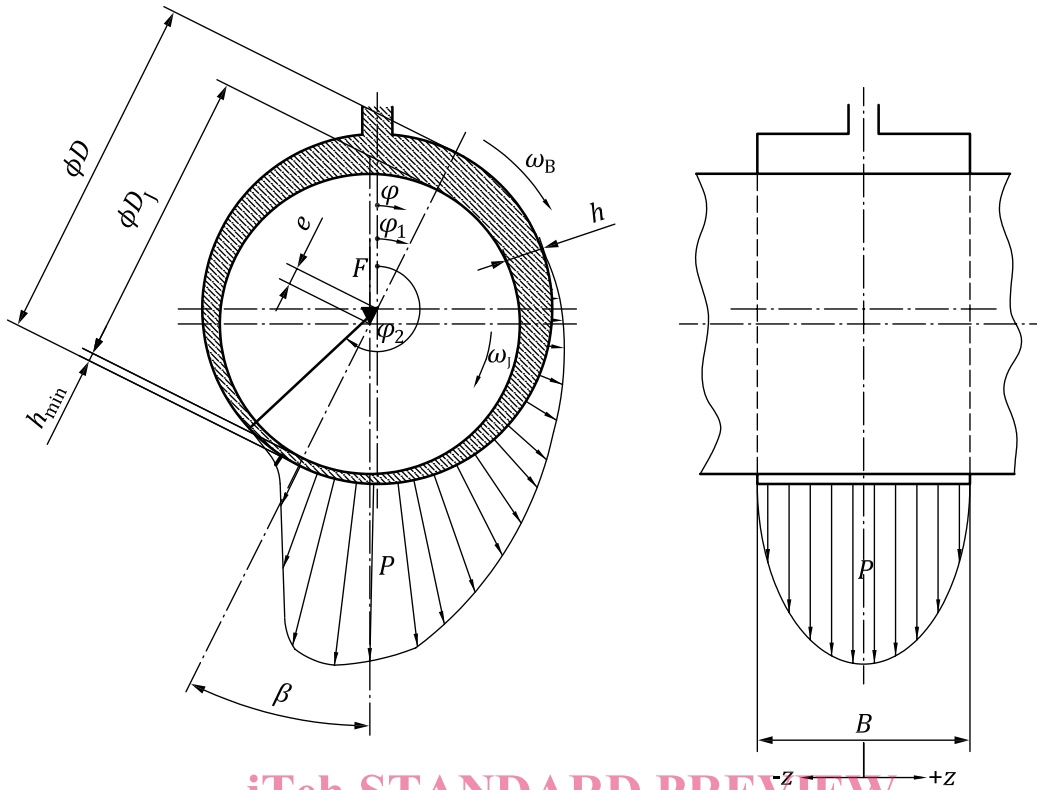
Figure 1 — Illustration of symbols

Table 1 — Symbols and their designations

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

Table 1 *(continued)*

Symbol	Designation	Unit
F_f'	Frictional force in both the loaded and the unloaded area of the lubricant film	N
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
H	Length of the total height of the pedestal bearing	m
k_A	Outer heat transmission coefficient	$W/(m^2 \cdot K)$
N_B	Rotational frequency of the bearing	s^{-1}
N_J	Rotational frequency of the shaft	s^{-1}
p	Local lubricant film pressure	Pa
$\bar{p}$	Specific bearing load	Pa
p_{en}	Lubricant feed pressure	Pa
p_{lim}	Maximum permissible lubricant film pressure	Pa
$\bar{p}_{lim}$	Maximum permissible specific bearing load	Pa
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
q_L	Coefficient related to lubricant flow rate due to feed pressure	1
q_P	Coefficient related to lubricant flow rate from pocket	1
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
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
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

Table 1 *(continued)*

Symbol	Designation	Unit
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 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
v	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
$\bar{\psi}$	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_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).

$$\frac{\partial}{\partial x}\left(h^3\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial z}\left(h^3\frac{\partial p}{\partial z}\right)=6\eta\left(U_J+U_B\right)\frac{\partial h}{\partial x} \tag{1}$$

See References [3] to [5] and References [13] to [16] for the derivation of Reynolds equation and References [6] to [8], [14] and [15] 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: $p(\varphi_1,z)=0$;

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

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

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

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

In partial bearings, if Formula (2) is satisfied:

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

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

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

5.4 Basis of calculation

The numerical integration of the Reynolds equation is carried out (possibly by applying transformation of pressure as suggested in References [5], [13] and [14]) by a transformation to a differential formula which is applied to a grid system of supporting points, and which results in a system of linear formulae. The number of supporting points is significant to the accuracy of the numerical integration; the use of a non-equidistant grid as given in References [8] and [15] 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.

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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 and relative bearing length. The application of magnitudes of similarity reduces the number of numerical solutions required of Reynolds equation specified in 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.

5.5 Permissible operational parameters

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.

6 Calculation procedure

6.1 General

Calculation is understood to mean determination of correct operation by computation using actual operating parameters (see Figure 2), which can be compared with operational parameters. The operating parameters determined under varying operating conditions shall 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.

6.2 Freedom from wear

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