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Second edition
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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
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Foreword

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The committee responsible for this document is ISO/TC 123, *Plain bearings*, Subcommittee SC 4, *Methods of calculation of plain bearings*.

This second edition cancels and replaces the first edition (ISO 7902-1:1998), which has been technically revised.

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*

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 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 can result from vibration effects and instabilities of rapid-running rotors, are not taken into account.

2 Normative references

The following documents, in whole or in part, are normatively referenced in this document and are indispensable for its application. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies.

ISO 3448, *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, *Hydrodynamic plain journal bearings under steady-state conditions — Circular cylindrical bearings — Part 3: Permissible operational parameters*

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:

$$\frac{\partial}{\partial x}\left(h^3 \frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial x}\left(h^3 \frac{\partial p}{\partial z}\right) = 6\eta\left(u_J + u_B\right)\frac{\partial h}{\partial x} \quad (1)$$

The symbols are given in Clause 5.

See References [1] to [3] and References [11] to [14] for the derivation of Reynolds' differential equation and References [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: $p\left[\varphi_2(z), z\right] = 0$;

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

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

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

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

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

$$p(\varphi = \varphi_2, z) = 0 \tag{3}$$

3.4 The numerical integration of the Reynolds' differential equation is carried out (possibly by applying transformation of pressure as suggested in References [3], [11], and [12]) 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 [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, and relative bearing length. The application of magnitudes of similarity reduces the number of numerical solutions required of Reynolds' differential 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.

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 Calculation is understood to mean 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 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.

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 not only 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.);

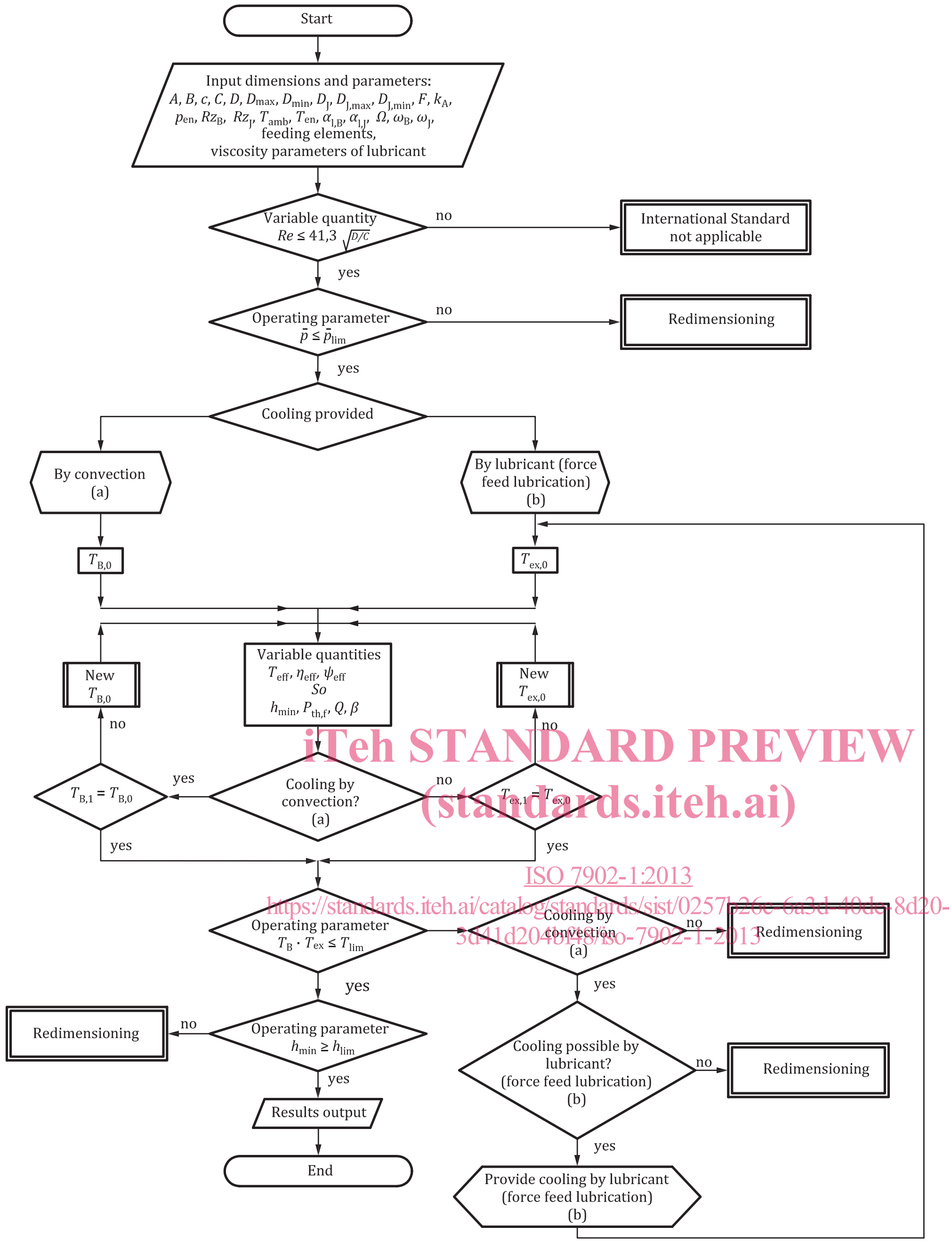

Figure 1 — Outline of calculation

— 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 The 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:

$$Re = \frac{\rho U_{\mathrm{J}} \frac{C_{\mathrm{R,eff}}}{2}}{\eta} = \frac{\pi D N_{\mathrm{J}} \frac{C_{\mathrm{R,eff}}}{2}}{\nu} \leq 41{,}3 \sqrt{\frac{D}{C_{\mathrm{R,eff}}}} \qquad (4)$$

In the case of plain bearings with $Re > 41{,}3\sqrt{D/C_{\mathrm{R,eff}}}$ (for example as a result of high peripheral speed), higher loss coefficients and bearing temperatures shall 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}:

$$h_{\mathrm{min}} = \frac{D - D_{\mathrm{J}}}{2} - e = 0{,}5 D\psi\left(1 - \varepsilon\right) \qquad (5)$$

where the relative eccentricity, ε, is given by

$$\varepsilon = \frac{e}{\frac{D - D_{\mathrm{J}}}{2}} \qquad (6)$$

If

$$\varphi_2 - (\pi - \beta) < \frac{\pi}{2} \qquad (7)$$

then

$$h_{\mathrm{min}} = 0{,}5 D\psi(1 + \varepsilon \cos\varphi_2) \qquad (8)$$

6 Definition of symbols

6.1 Load-carrying capacity

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

$$So = \frac{F\psi^2_{\mathrm{eff}}}{DB\eta_{\mathrm{eff}}\omega_{\mathrm{h}}} = So\left(\varepsilon, \frac{B}{D}, \Omega\right) \tag{9}$$

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 ϕ_{eff} take into account the thermal effects and the angular velocities of shaft, bearing, and bearing force (see 6.4 and 6.7).

The relative eccentricity, ε, together with the attitude angle, β (see ISO 7902-2), describes the magnitude and position of the minimum thickness of lubricant film. For a full bearing (Ω = 360°), 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.

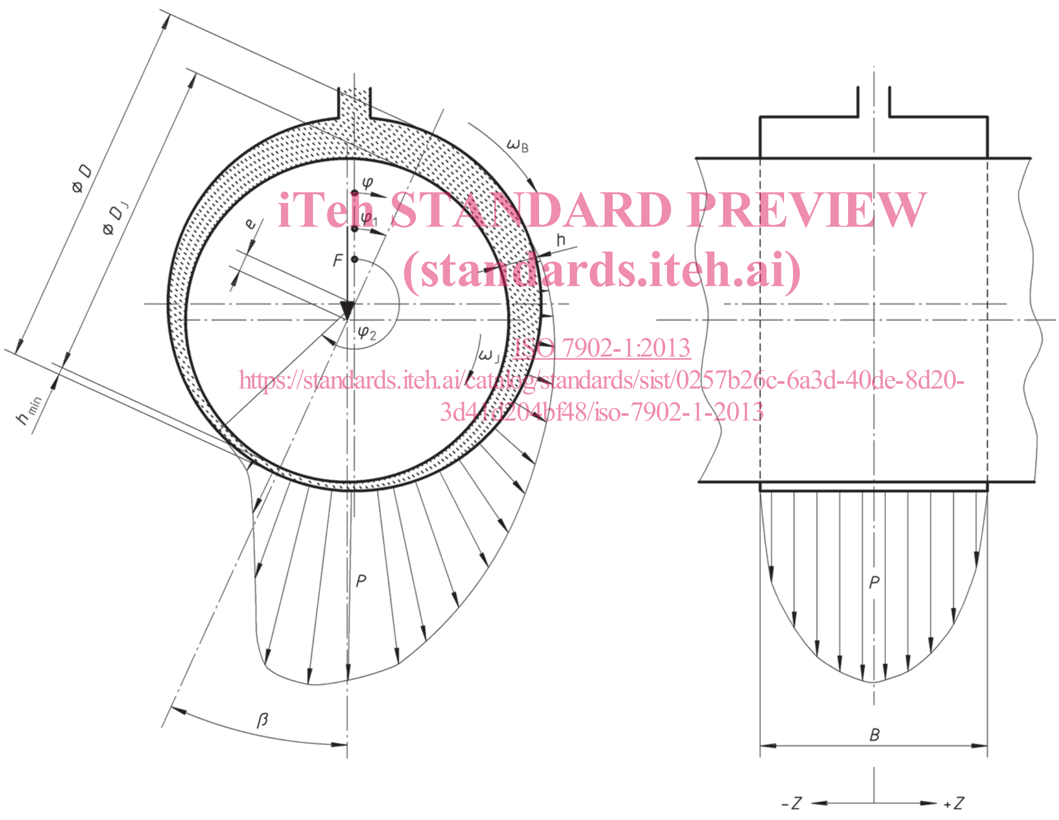

Figure 2 — Illustration of symbols

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

Table 1 *(continued)*

Symbol	Designation	Unit
$C_{R,eff}$	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/(m2·K)
l_G	Length of oil groove	m
l_P	Length of oil pocket	m
L_H	Length of bearing housing Rotational	m
N_B	Frequency of the bearing Rotational	s^{-1}
N_F	Frequency of the bearing force Rotational	s^{-1}
N_J	Frequency of the shaft	s^{-1}
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

Table 1 *(continued)*

Symbol	Designation	Unit
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
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
$T_\mathrm{B,0}$	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
$T_\mathrm{ex,0}$	Assumed initial lubricant temperature at bearing exit	°C
$T_\mathrm{ex,1}$	Calculated lubricant temperature at bearing exit	°C
T_J	Shaft temperature	°C
T_lim	Maximum permissible bearing temperature	°C
$\bar{T}_\mathrm{L}$	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
$\alpha_{l,\mathrm{B}}$	Linear heat expansion coefficient of the bearing	K^{-1}
$\alpha_{l,\mathrm{J}}$	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