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Plain bearings — Hydrodynamic plain journal bearings under steady-state conditions —

Part 1:
Calculation of multi-lobed and tilting pad journal bearings

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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).

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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*.

A list of all parts in the ISO/TS 31657 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.

Introduction

The aim of this document is the operationally-safe design of plain journal bearings for medium or high journal circumferential velocities, U_{J}, up to approximately 90 m/s by applying a calculation method for oil-lubricated hydrodynamic plain bearings with complete separation of journal and bearing sliding surfaces by a lubricating film.

For low circumferential velocities up to approximately 30 m/s usually circular cylindrical bearings are applied. For these bearings a similar calculation method is given in ISO 7902-1, ISO 7902-2 and ISO 7902-3.

Based on practical experience the calculation procedure is usable for application cases where specific bearing load times circumferential speed, $\overline{p} \cdot U_{\mathrm{J}}$, does not exceed approximately 200 MPa·m/s.

This document discusses multi-lobed journal bearings with two, three and four equal, symmetrical sliding surfaces, which are separated by laterally-closed lubrication pockets, and symmetrically-loaded tilting-pad journal bearings with four and five pads. Here, the curvature radii, R_B, of the sliding surfaces are usually chosen larger than half the bearing diameter, D, so that an increased bearing clearance results at the pad ends.

The calculation method described here can also be used for other gap forms, for example asymmetrical multi-lobed journal bearings like offset-halves bearings, pressure-dam bearings or other tilting-pad journal bearing designs, if the numerical solutions of the basic formulas are available for these designs.

Plain bearings — Hydrodynamic plain journal bearings under steady-state conditions —

Part 1: Calculation of multi-lobed and tilting pad journal bearings

1 Scope

This document specifies the general principles, assumptions and preconditions for the calculation of multi-lobed and tilting-pad journal bearings by means of an easy-to-use calculation procedure based on numerous simplifying assumptions. For a reliable evaluation of the results of this calculation method, it is indispensable to consider the physical implications of these assumptions as well as practical experiences for instance from temperature measurements carried out on real machinery under typical operating conditions. Applied in this sense, this document presents a simple way to predict the approximate performance of plain journal bearings for those unable to access more complex and accurate calculation techniques.

The calculation method serves for the design and optimisation of plain bearings, for example in turbines, compressors, generators, electric motors, gears and pumps. It is restricted to steady-state operation, i.e. in continuous operating states the load according to size and direction and the angular velocity of the rotor are constant.

Unsteady operating states are not recorded. The stiffness and damping coefficients of the plain journal bearings required for the linear vibration and stability investigations are indicated in ISO/TS 31657-2 and ISO/TS 31657-3.

2 Normative references

There are no normative references in this document.

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

Table 1 contains the symbols used in the ISO 31657 series.

Table 1 — Symbols and units

Symbol	Description	Unit
B	Bearing width	m
B^*	Relative bearing width, width ratio as given by: $B^* = \frac{B}{D}$	1

Table 1 *(continued)*

Symbol	Description	Unit
b_P	Width of lubricant pocket	m
b_P^*	Relative width of lubricant pocket, as given by: $b_P^* = \dfrac{b_P}{B}$	1
C_R	Bearing radial clearance, as given by: $C_R = R - R_J$	m
$C_{R,eff}$	Effective radial bearing clearance	m
c_{ik}	Stiffness coefficient of lubricant film (i,k = 1,2)	N/m
c_{ik}^*	Non-dimensional stiffness coefficient of lubricant film, as given by: $c_{ik}^* = \dfrac{\psi_{eff}^3}{2 \cdot B \cdot \eta_{eff} \cdot \omega} \cdot c_{ik}\ (i,k=1,2)$	1
c_p	Specific heat capacity (p = constant)	J/(kg K)
D	Nominal bearing diameter (inside diameter of journal bearing)	m
D_{max}	Maximum value of D	m
D_{min}	Minimum value of D	m
D_J	Journal diameter (diameter of the shaft section located inside of a journal bearing)	m
$D_{J,max}$	Maximum value of D_J	m
$D_{J,min}$	Minimum value of D_J	m
d_{ik}	Damping coefficient of lubricant film (i,k = 1,2)	N s/m
d_{ik}^*	Non-dimensional damping coefficient of lubricant film, as given by: $d_{ik}^* = \dfrac{\psi_{eff}^3}{2 \cdot B \cdot \eta_{eff} \cdot \omega} \cdot \omega \cdot d_{ik}\ (i,k=1,2)$	1
e	Eccentricity (distance between journal and bearing axis)	m
e_B	Eccentricity of the bearing sliding surfaces (pads) of a multi-lobed or tilting-pad journal bearing	m
f	Bearing force, bearing load, nominal bearing load, load-carrying capacity	N
ΔF_x	Component of additional dynamic force in x-direction	N
ΔF_y	Component of additional dynamic force in y-direction	N
ΔF_x^*	Component of additional dynamic force parameter in x-direction, as given by: $\Delta F_x^* = \dfrac{\Delta F_x \cdot \psi_{eff}^2}{B \cdot D \cdot \eta_{eff} \cdot \omega}$	1
ΔF_y^*	Component of additional dynamic force parameter in y-direction, as given by: $\Delta F_y^* = \dfrac{\Delta F_y \cdot \psi_{eff}^2}{B \cdot D \cdot \eta_{eff} \cdot \omega}$	1
F_f	Friction force, as given by: $F_f = f \cdot F$	N
F_f^*	Friction force parameter, as given by: $F_f^* = \dfrac{f}{\psi_{eff}} \cdot So$	1
F_{tr}	Bearing force at transition to mixed friction	N
F	Coefficient of friction	1
f_J	Journal deflection	m
$h(\varphi)$	Local lubricant film thickness	m

Table 1 *(continued)*

Symbol	Description	Unit
$h^*(\varphi)$	Relative local lubricant film thickness, as given by: $h^*(\varphi)=\frac{h(\varphi)}{C_R}$	1
$h_{\text{lim,tr}}$	Minimum admissible lubricant film thickness at transition to mixed friction	m
$h^*_{\text{lim,tr}}$	Minimum admissible relative lubricant film thickness at transition to mixed friction, as given by: $h^*_{\text{lim,tr}}=\frac{h_{\text{lim,tr}}}{C_{\text{R,eff}}}$	1
h_{min}	Minimum lubricant film thickness, minimum gap	m
h^*_{min}	Minimum relative lubricant film thickness, minimum relative gap, as given by: $h^*_{\text{min}}=\frac{h_{\text{min}}}{C_{\text{R,eff}}}$	1
$h_{\text{min,tr}}$	Minimum lubricant film thickness at transition to mixed friction	m
$h^*_{\text{min,tr}}$	Minimum relative lubricant film thickness at transition to mixed friction, as given by: $h^*_{\text{min,tr}}=\frac{h_{\text{min,tr}}}{C_{\text{R,eff}}}$	1
$h_0(\varphi)$	Local gap at $\varepsilon=0$, gap function	m
$h^*_0(\varphi)$	Relative local gap at $\varepsilon=0$, profile function, as given by: $h^*_0(\varphi)=\frac{h_0(\varphi)}{C_R}$	1
$h_{0,\text{max}}$	Maximum gap at $\varepsilon=0$	m
$h^*_{0,\text{max}}$	Maximum relative gap at $\varepsilon=0$, gap ratio, as given by: $h^*_{0,\text{max}}=\frac{h_{0,\text{max}}}{C_R}$	1
K_P	Profile factor (relative difference between lobe or pad bore radius and journal radius), as given by: $K_P=\frac{\Delta R_B}{C_R}=\frac{1}{1-m}$	1
$K_{\text{P,eff}}$	Effective profile factor	1
$K_{\text{P,20}}$	Profile factor at 20 °C	1
M	Mixing factor	1
m	Preload factor, preload of bearing or pad sliding surface	1
N	Rotational speed (rotational frequency) of the rotor (revolutions per time unit)	s^{-1}
N_{cr}	Critical speed (critical rotational frequency)	s^{-1}
N_{lim}	Rotational speed (rotational frequency) at the stability speed limit of the rotor supported by plain bearings	s^{-1}
N_{rsn}	Resonance speed (resonance rotational frequency) of the rotor supported by plain bearings	s^{-1}
N_{tr}	Rotational speed (rotational frequency) at transition to mixed friction, transition rotational speed, transition rotational frequency	s^{-1}
O_B	Centreline of plain bearing	1
O_i	Centreline of sliding surface No. i	1
O_J	Centreline of journal	1
P_f	Frictional power, as given by: $P_f=F_f\cdot U_J$	W
$P_{\text{th},L}$	Heat flow via the lubricant	W
p	Lubricant film pressure, local lubricant film pressure	Pa

Table 1 *(continued)*

Symbol	Description	Unit
$\bar{p}$	Specific bearing load, as given by: $\bar{p}=\frac{F}{B \cdot D}$	Pa
p_{en}	Lubricant supply pressure	Pa
p_{en}^*	Lubricant supply pressure parameter, as given by: $p_{en}^*=\frac{p_{en} \cdot \psi_{eff}^2}{\eta_{eff} \cdot \omega}$	1
p_{lim}	Maximum admissible lubricant film pressure	Pa
$\bar{p}_{lim,tr}$	Maximum admissible specific bearing load at transition to mixed friction	Pa
p_{max}	Maximum lubricant film pressure	Pa
p_{max}^*	Maximum lubricant film pressure parameter, as given by: $p_{max}^*=\frac{p_{max}}{\bar{p}}$	1
$\bar{p}_{tr}$	Specific bearing load at transition to mixed friction, as given by: $\bar{p}_{tr}=\frac{F_{tr}}{B \cdot D}$	Pa
Q	Lubricant flow rate, as given by: $Q=Q_3+Q_p$	m^3/s
Q_{lim}	Minimum admissible lubricant flow rate	m^3/s
Q_p	Lubricant flow rate due to supply pressure	m^3/s
Q_p^*	Lubricant flow rate parameter due to supply pressure, as given by: $Q_p^*=\frac{Q_p}{p_{en}^* \cdot Q_0}$	1
Q_0	Reference value of Q, as given by: $Q_0=R^3 \cdot \omega \cdot \psi_{eff}$	m^3/s
Q_1	Lubricant flow rate at the entrance into the lubrication gap (circumferential direction)	m^3/s
Q_2	Lubricant flow rate at the exit of the lubrication gap (circumferential direction), as given by: $Q_2=Q_1-Q_3$	m^3/s
Q_2^*	Lubricant flow rate parameter at the exit of the lubrication gap (circumferential direction), as given by: $Q_2^*=\frac{Q_2}{Q_0}$	1
Q_3	Lubricant flow rate due to hydrodynamic pressure build-up (side flow rate)	m^3/s
Q_3^*	Lubricant flow rate parameter due to hydrodynamic pressure build-up (side flow parameter), as given by: $Q_3^*=\frac{Q_3}{Q_0}$	1
R	Journal bearing inside radius, as given by: $R=\frac{D}{2}$	m
R_B	Lobe or pad bore radius of a multi-lobed or tilting-pad journal bearing	m
ΔR_B	Difference between lobe or pad bore radius and journal radius, as given by: $\Delta R_B=R_B-R_J$	m
R_J	Journal radius (radius of the shaft section located inside of a journal bearing), as given by: $R_J=\frac{D_J}{2}$	m
$R_{z,B}$	Surface finish ten-point average of bearing sliding surface	m
$R_{z,J}$	Surface finish ten-point average of journal sliding surface	m
Re	Reynolds number, as given by: $Re=\frac{\rho \cdot \omega \cdot R \cdot C_{R,eff}}{\eta_{eff}}$	1
Re_{cr}	Critical Reynolds number	1

Table 1 *(continued)*

Symbol	Description	Unit
So	Sommerfeld number, as given by: $So=\dfrac{F\cdot\psi_{eff}^2}{B\cdot D\cdot\eta_{eff}\cdot\omega}$	1
So_{tr}	Sommerfeld number at transition to mixed friction	1
S	Displacement amplitude of the rotor (mechanical oscillation)	m
T	Temperature	°C
ΔT	Heating of lubricant between bearing entrance and exit, as given by: $\Delta T=T_{ex}-T_{en}$	K
ΔT_{lim}	Maximum admissible heating of lubricant between bearing entrance and exit	K
T_B	Bearing temperature	°C
T_{eff}	Effective temperature of lubricant film	°C
T_{en}	Lubricant temperature at the bearing entrance	°C
T_{ex}	Lubricant temperature at the bearing exit	°C
T_J	Journal temperature	°C
T_{lim}	Maximum admissible bearing temperature	°C
T_{max}	Maximum temperature of lubricant film	°C
ΔT_{max}	Difference between maximum temperature of lubricant film and lubricant temperature in the lubricant pocket, as given by: $\Delta T_{max}=T_{max}-T_1$	K
ΔT^*_{max}	Non-dimensional difference between maximum temperature of lubricant film and lubricant temperature in the lubricant pocket, as given by: $\Delta T^*_{max}=\dfrac{\rho\cdot c_p\cdot\Psi_{eff}}{\bar{p}\cdot f}\cdot\Delta T_{max}$	1
T_1	Lubricant temperature at the entrance into the lubrication gap (circumferential direction)	°C
ΔT_1	Difference between lubricant temperature at the entrance into the lubrication gap and lubricant temperature at the bearing entrance, as given by: $\Delta T_1=T_1-T_{en}$	K
T_2	Lubricant temperature at pressure profile trailing edge (circumferential direction)	°C
ΔT_2	Difference between lubricant temperature at pressure profile trailing edge and lubricant temperature at the entrance into the lubrication gap, as given by: $\Delta T_2=T_2-T_1$	K
t	Time	s
U_J	Circumferential speed of the journal, sliding velocity $U_J=\omega\cdot R_J$	m/s
U_{tr}	Circumferential speed at transition to mixed friction	m/s
$U_{lim,\,tr}$	Minimum admissible circumferential speed at transition to mixed friction	m/s
u	Velocity component in the φ-direction	m/s
$\bar{u}$	Average velocity component in the φ-direction	m/s
w	Velocity component in the z- direction	m/s
$\bar{w}$	Average velocity component in the z-direction	m/s
x	Coordinate of journal radial motion, normal to direction of load	m
x^*	Relative coordinate of journal radial motion, normal to direction of load, as given by: $x^*=\dfrac{x}{C_R}$	1
y	Coordinate normal to sliding surface (across the lubricant film, in the radial direction); coordinate of journal radial motion, in direction of load	m
y^*	Relative coordinate of journal radial motion, in direction of load, as given by: $y^*=\dfrac{y}{C_R}$	1

Table 1 *(continued)*

Symbol	Description	Unit
y_{h}	Coordinate normal to sliding surface (across the lubricant film)	m
Z	Number of sliding surfaces (pads), number of pockets per bearing	1
z	Coordinate parallel to the sliding surface, normal to direction of motion (normal to circumferential direction, in the axial direction)	m
$\alpha_{l,B}$	Linear thermal expansion coefficient of bearing material	K^{-1}
$\alpha_{l,\mathrm{J}}$	Linear thermal expansion coefficient of journal material	K^{-1}
β	Attitude angle (angular position of journal eccentricity related to the direction of load)	°
$\beta_{h,\mathrm{min}}$	Angle between direction of load and position of minimum lubricant film thickness	°
δ_{J}	Journal misalignment angle (angular deviation of journal)	°
ε	Relative eccentricity: $\varepsilon = \dfrac{e}{C_{R,\mathrm{eff}}}$	1
η	Dynamic viscosity of the lubricant	Pa s
η_{eff}	Effective dynamic viscosity in the lubricant film	Pa s
ρ	Density of the lubricant	$\mathrm{kg/m^3}$
φ	Angular coordinate in circumferential direction	°
φ_F	Angular coordinate of pivot position of pad (tilting-pad bearing)	°
φ_P	Angular coordinate of lubricant pocket centreline	°
φ_0	Angular coordinate of bearing sliding surface (segment or pad) centreline at multi-lobed or tilting-pad journal bearings (with non-tilted pads), see Figure 1, a)	°
φ_1	Angular coordinate at the entrance into the gap	°
φ_2	Angular coordinate at the end of the hydrodynamic pressure build-up	°
φ_3	Angular coordinate at the exit of the gap	°
ψ	Relative bearing clearance, as given by: $\psi = \dfrac{C_{\mathrm{R}}}{R}$	‰
$\Delta\psi$	Tolerance of ψ, as given by: $\Delta\psi = \psi_{\mathrm{max}} - \psi_{\mathrm{min}}$	‰
ψ_{eff}	Effective relative bearing clearance	‰
ψ_{max}	Maximum value of ψ	‰
ψ_{min}	Minimum value of ψ	‰
$\Delta\psi_{\mathrm{th}}$	Thermal change of ψ	‰
ψ_{20}	Relative bearing clearance at 20 °C	‰
Ω	Angular span of bearing sliding surface (segment or pad), as given by: $\Omega = \varphi_3 - \varphi_1$	°
Ω_F	Angular distance between leading edge and pivot position of pad (tilting-pad bearing), as given by: $\Omega_F = \varphi_F - \varphi_1$	°
Ω_F^*	Relative angular distance between leading edge and pivot position of pad (tilting-pad bearing), as given by: $\Omega_F^* = \Omega_F / \Omega$	1
Ω_P	Angular span of lubricant pocket, as given by: $\Omega_P = \dfrac{360°}{Z} - \Omega$	°
ω	Angular speed of the rotor, as given by: $\omega = 2 \cdot \pi \cdot N$	s^{-1}
ω_{tr}	Angular speed at transition to mixed friction	s^{-1}

5 General principles, assumptions and preconditions

The bearing bore form of multi-lobed journal bearings [see Figure 1, a)] and tilting-pad journal bearings [with non-tilted pads according to Figure 1, b)] is described by the profile function $h_0^*(\varphi)=\frac{h_0(\varphi)}{C_R}$ in the case of a centric journal position $\varepsilon=\frac{e}{C_R}=0$. The angle φ is counted, starting from the load direction, in the journal rotational direction.

Formula (1) applies to the shell segment or pad i with the angular length $\Omega_i=\varphi_{3,i}-\varphi_{1,i}$:

$$h_{0,i}^*(\varphi)=\frac{\Delta R_B}{C_R}+\left(\frac{\Delta R_B}{C_R}-1\right)\cdot\cos(\varphi-\varphi_{0,i}),i=1,...,Z \tag{1}$$

with the profile factor

$$K_P=\frac{\Delta R_B}{C_R}=\frac{R_B-R_J}{C_R}=1+\frac{e_B}{C_R},$$

minimum clearance

$$C_R=R-R_J=\frac{D-D_J}{2}$$

and the lubricant film thickness ratio as given by Formula (2):

$$h_0^*(\varphi_{P,i})=h_{0,\max}^*=\frac{h_{0,\max}}{C_R} \tag{2}$$

Here the position of the sliding surface (segment or pad) axis (curvature centre "point") of the shell segment or pad i is uniquely described by the sliding surface eccentricity e_B and the associated angle coordinate $\varphi_{0,i}$.

In the case of cylindrical bearings, K_P = 1 and $h_0^*(\varphi)=1$.

NOTE Instead of the profile factor, K_P, the "preload factor", m, is frequently used internationally; the following relation exists between both variables:

$$K_P=\frac{1}{1-m}$$

In the case of an eccentric position of the journal (ε, β), Formula (3) applies to the lubricant film thickness, $h(\varphi)$, of the multi-lobed journal bearings [(see Figure 1, c)]:

$$h(\varphi)=C_R\cdot h^*(\varphi)=C_R\cdot[h_0^*(\varphi)-\varepsilon\cdot\cos(\varphi-\beta)] \tag{3}$$

In the case of tilting-pad journal bearings [see Figure 1, d)], the individual pads automatically adjust themselves (optimally) so that the lubricant film force F_i passes through the supporting pad pivot, respectively[9]. For a more precise calculation of tilting-pad journal bearings, the elasticities in the pad support and the elastic and thermal deformations of the pads shall be considered.