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Plain bearings — Thermo-hydrodynamic lubrication design charts for circular cylindrical bearings under steady-state conditions

Paliers lisses — Diagrammes de conception de la lubrification thermo-hydrodynamique des paliers cylindriques circulaires dans des conditions de régime permanent

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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*.

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Introduction

The calculation procedure specified in ISO 7902-1:2020 is useful to calculate the performance of hydrodynamic plain journal bearings with a circular cylindrical shape. This procedure, however, does not specify the maximum bearing temperature, which is one of the most important bearing characteristics. The reason for this is that ISO 7902-1:2020 is based on the Reynolds equation which assumes a constant lubricant film temperature. Therefore, the calculation procedure requires some numerical iteration before the effective dynamic viscosity in the lubricant film is converged.

This document provides a calculation procedure for the maximum bearing temperature and the effective dynamic viscosity in the lubricant film of oil-lubricated and statically loaded hydrodynamic plain journal bearings with a circular cylindrical shape, without any complicated numerical analysis and iterative calculation. The basic formulae contain the energy equation and the formula of temperature-viscosity of the lubricant to obtain the maximum bearing temperature. Since the results already satisfy the energy balance, no iterative calculation is required.

For the reason given above, the effective dynamic viscosity in the lubricant film obtained by the procedure in this document can also be a good input data for ISO 7902-1:2020. Annex A shows an example of how the calculated results serve to provide input data for ISO 7902-1:2020.

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Plain bearings — Thermo-hydrodynamic lubrication design charts for circular cylindrical bearings under steady-state conditions

1 Scope

This document specifies a calculation procedure for the maximum bearing temperature and effective dynamic viscosity in the lubricant film of oil-lubricated and statically loaded hydrodynamic plain journal bearings with a circular cylindrical shape, angular span Ω of 360° and width ratio B^* of 0,5 to 1,5 under fluid lubrication regime. The bearing characteristics are obtained by design charts from four dimensionless numbers which are calculated from bearing dimensions, operating conditions and viscosity characteristics of the lubricant.

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 4378-1, *Plain bearings — Terms, definitions, classification and symbols — Part 1: Design, bearing materials and their properties*

3 Terms and definitions

For the purposes of this document, the terms and definitions given in ISO 4378-1 and the following apply.

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/

3.1
streamlet
axially separated stream of lubricant flow at bearing clearance where the gap increases in the rotational direction

4 Symbols, units and abbreviated terms

Symbols and units are defined in Figure 1 and Table 1. Abbreviated terms are defined in Table 2.

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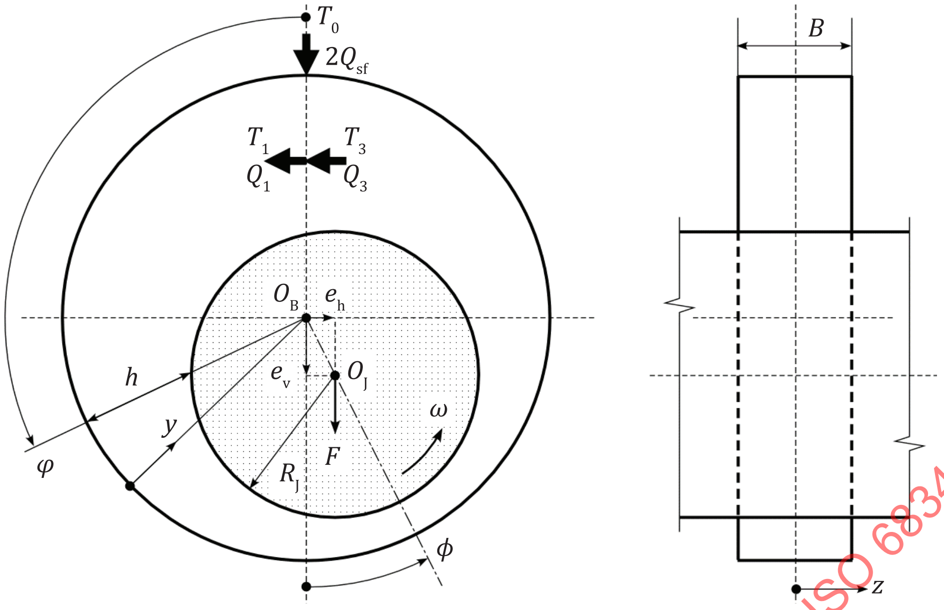

Figure 1 — Illustration of symbols

Table 1 — Symbols and their designations

Symbol	Designation	Unit
B	Bearing width	m
B^*	Width ratio ($B^* = B/D$)	1
C_R	Bearing radial clearance ($C_R = R - R_J$)	m
c_p	Specific heat of the lubricant	J/(kg·K)
D	Inside diameter of journal bearing	m
D_J	Journal diameter	m
E_0	Function of relative dynamic viscosity of the lubricant	1
E_1	Function of relative dynamic viscosity of the lubricant	1
e	Eccentricity between journal and bearing axis	m
e_h	Eccentricity in the horizontal direction between journal and bearing axis	m
e_v	Eccentricity in the vertical direction between journal and bearing axis	m
e_3^*	Relative heat energy of lubricant at the exit of the gap	1
F	Bearing load	N
F_0	Function of relative dynamic viscosity of the lubricant	1
F_1	Function of relative dynamic viscosity of the lubricant	1
F_2	Function of relative dynamic viscosity of the lubricant	1
h	Local lubricant film thickness	m
h^*	Relative local lubricant film thickness ($h^* = h/C_R$)	1
N	Rotational frequency of the rotor	s^{-1}
O_B	Centerline of circular cylindrical bearing	1
O_J	Centerline of journal	1

NOTE 1 S number is frequently referred to as Sommerfeld number. See References [4] to [6].

NOTE 2 In ISO 4378-5,[1] β is defined as the attitude angle or the temperature viscosity coefficient. In ISO 7902-1:2020, β is defined as the attitude angle. However, in this document, the attitude angle and the temperature viscosity coefficient are represented by ϕ and β, respectively, to avoid confusion due to duplication.

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Table 1 *(continued)*

Symbol	Designation	Unit
Pe	Peclet number ($Pe = C_R^2 \rho c_p \omega / \lambda$)	1
p	Local lubricant film pressure	Pa
p^*	Relative local lubricant film pressure [$p^* = p\psi^2/(\eta_0\omega)$]	1
Q_{sf}	Lubricant side flow rate leaking out of one of the bearing side ends due to hydrodynamic pressure	m^3/s
Q_{sf}^*	Lubricant side flow rate parameter leaking out of one of the bearing side ends due to hydrodynamic pressure [$Q_{sf}^* = Q_{sf} / (RBC_R\omega)$]	1
Q_1	Lubricant flow rate at the entrance into the gap ($\varphi = \varphi_1$)	m^3/s
Q_3	Lubricant flow rate at the exit of the gap ($\varphi = \varphi_3$)	m^3/s
Q_3^*	Lubricant flow rate parameter at the exit of the gap [$Q_3^* = Q_3 / (RBC_R\omega)$]	1
R	Journal bearing inside radius ($R = D/2$)	m
R_J	Journal radius ($R_J = D_J/2$)	m
S_0	S number formed with η_0 and ω [$S_0 = BD\eta_0\omega/(F\psi^2)$]	1
T	Lubricant temperature	°C
T_a	Average temperature of lubricant film	°C
$\Delta T_{a,r}$	Relative difference between average temperature of lubricant film and lubricant supplying temperature [$\Delta T_{a,r} = \psi^2 \rho c_p (T_a - T_0)/(\eta_0\omega)$]	1
T_B	Bearing temperature	°C
$T_{B,max}$	Maximum bearing temperature	°C
$\Delta T_{B,r}$	Relative difference between bearing temperature and lubricant supplying temperature [$\Delta T_{B,r} = \psi^2 \rho c_p (T_B - T_0)/(\eta_0\omega)$]	1
$\Delta T_{B,max,r}$	Relative difference between maximum bearing temperature and lubricant supplying temperature [$\Delta T_{B,max,r} = \psi^2 \rho c_p (T_{B,max} - T_0)/(\eta_0\omega)$]	1
$\widetilde{\Delta T_{B,max,r}}$	Logarithmic modified relative difference between maximum bearing temperature and lubricant supplying temperature [$\widetilde{\Delta T_{B,max,r}} = \log_{10}\left(\Delta T_{B,max,r}\ B^{*-0,154}\ Pe^{1,75}\ \beta^{*8}\right)$]	1
T_{eff}	Effective lubricant film temperature	°C
$T_{ex,0}$	Assumed initial lubricant temperature at bearing exit	°C
$T_{ex,1}$	Calculated lubricant temperature at bearing exit	°C
T_J	Journal surface temperature	°C
$\Delta T_{J,r}$	Relative difference between journal surface temperature and lubricant supplying temperature [$\Delta T_{J,r} = \psi^2 \rho c_p (T_J - T_0)/(\eta_0\omega)$]	1
ΔT_r	Relative difference between lubricant temperature and lubricant supplying temperature [$\Delta T_r = \psi^2 \rho c_p (T - T_0)/(\eta_0\omega)$]	1
T_0	Lubricant supplying temperature	°C
T_1	Lubricant temperature at the entrance into the gap ($\varphi = \varphi_1$)	°C
$\Delta T_{1,r}$	Relative difference between lubricant temperature at the entrance into the gap and lubricant supplying temperature [$\Delta T_{1,r} = \psi^2 \rho c_p (T_1 - T_0)/(\eta_0\omega)$]	1
T_3	Lubricant temperature at the exit of the gap ($\varphi = \varphi_3$)	°C
U_J	Circumferential speed of the journal ($U_J = R_J\omega$)	m/s
u	Velocity component in the circumferential direction	m/s
u^*	Relative velocity component in the circumferential direction ($u^* = u/U_J$)	1

NOTE 1 S number is frequently referred to as Sommerfeld number. See References [4] to [6].

NOTE 2 In ISO 4378-5,[1] β is defined as the attitude angle or the temperature viscosity coefficient. In ISO 7902-1:2020, β is defined as the attitude angle. However, in this document, the attitude angle and the temperature viscosity coefficient are represented by ϕ and β, respectively, to avoid confusion due to duplication.

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Table 1 *(continued)*

Symbol	Designation	Unit
v	Velocity component in the cross-film direction	m/s
v^*	Relative velocity component in the cross-film direction [$v^* = v/(C_\mathrm{R}\omega)$]	1
w	Velocity component in the axial direction	m/s
w^*	Relative velocity component in the axial direction [$w^* = w/(B\omega)$]	1
y	Coordinate across the lubricating film	m
y^*	Relative coordinate across the lubricating film ($y^* = y/h$)	1
z	Coordinate in the axial direction	m
z^*	Relative coordinate in the axial direction ($z^* = z/B$)	1
α	Contraction ratio of the lubricant streamlet	1
β	Temperature viscosity coefficient of the lubricant [$\beta = \ln(\eta_{40}/\eta_{100})/60$]	1/°C
β^*	Relative temperature viscosity coefficient of the lubricant [$\beta^* = \eta_0\omega\beta/(\rho c_p\psi^2)$]	1
$\gamma_{\Delta T_\mathrm{B,max}}$	The first logarithmic relative coordinate variable in design chart for difference between maximum bearing temperature and lubricant supplying temperature [$\gamma_{\Delta T_\mathrm{B,max}} = \log_{10}\left(S_0\ B^{*1,54}\right)$]	1
$\gamma_{\eta_\mathrm{eff,r}}$	The first logarithmic relative coordinate variable in design chart for effective relative dynamic viscosity in lubricant film [$\gamma_{\eta_\mathrm{eff,r}} = \log_{10}\left(S_0\ B^{*1,90}\right)$]	1
ε	Relative eccentricity ($\varepsilon = e/C_\mathrm{R}$)	1
ε_h	Relative eccentricity in the horizontal direction ($\varepsilon_\mathrm{h} = e_\mathrm{h}/C_\mathrm{R}$)	1
ε_v	Vertical eccentricity in the vertical direction ($\varepsilon_\mathrm{v} = e_\mathrm{v}/C_\mathrm{R}$)	1
η	Dynamic viscosity of the lubricant	Pa·s
η_a	Average dynamic viscosity in lubricant film corresponding to difference ΔT_a between average temperature of lubricant film and lubricant supplying temperature	Pa·s
$\eta_\mathrm{a,r}$	Relative average dynamic viscosity in lubricant film corresponding to relative difference $\Delta T_\mathrm{a,r}$ between average temperature of lubricant film and lubricant supplying temperature ($\eta_\mathrm{a,r} = \eta_\mathrm{a}/\eta_0$)	1
η_eff	Effective dynamic viscosity in lubricant film	Pa·s
$\eta_\mathrm{eff,r}$	Effective relative dynamic viscosity in lubricant film ($\eta_\mathrm{eff,r} = \eta_\mathrm{eff}/\eta_0$)	1
η_rel	Relative dynamic viscosity of the lubricant ($\eta_\mathrm{rel} = \eta/\eta_0$)	1
η_0	Dynamic viscosity at the lubricant supplying temperature	Pa·s
η_{40}	Dynamic viscosity of the lubricant at 40 °C	Pa·s
η_{100}	Dynamic viscosity of the lubricant at 100 °C	Pa·s
$\widetilde{\eta_\mathrm{eff,r}}$	Logarithmic modified effective relative dynamic viscosity in lubricant film [$\widetilde{\eta_\mathrm{eff,r}} = \log_{10}\left(\eta_\mathrm{eff,r}\ B^{*-0,01}\ Pe^{1,10}\ \beta^{*2,35}\right)$]	1
$\kappa_{\Delta T_\mathrm{B,max}}$	The second logarithmic relative coordinate variable in design chart for difference between maximum bearing temperature and lubricant supplying temperature [$\kappa_{\Delta T_\mathrm{B,max}} = \log_{10}\left(Pe\ \beta^{*4}\right)$]	1
$\kappa_{\eta_\mathrm{eff,r}}$	The second logarithmic relative coordinate variable in design chart for effective relative dynamic viscosity of lubricant film [$\kappa_{\eta_\mathrm{eff,r}} = \log_{10}\left(Pe\ \beta^{*1,95}\right)$]	1
λ	Thermal conductivity of the lubricant	W/(m·K)

NOTE 1 S number is frequently referred to as Sommerfeld number. See References [4] to [6].

NOTE 2 In ISO 4378-5,[1] β is defined as the attitude angle or the temperature viscosity coefficient. In ISO 7902-1:2020, β is defined as the attitude angle. However, in this document, the attitude angle and the temperature viscosity coefficient are represented by ϕ and β, respectively, to avoid confusion due to duplication.

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Table 1 *(continued)*

Symbol	Designation	Unit
π	Circular constant (π = 3,141 592 ...)	1
ρ	Density of the lubricant	kg/m^3
φ	Angular coordinate in the circumferential direction	rad
φ_1	Angular coordinate at the entrance into the gap ($\varphi_1 = 0$)	rad
φ_2	Angular coordinate at the end of the hydrodynamic pressure build-up	rad
φ_3	Angular coordinate at the end of the gap ($\varphi_3 = 2\pi$)	rad
ϕ	Attitude angle (angular position of the shaft eccentricity related to the direction of load)	°
ψ	Relative bearing clearance ($\psi = C_R/R$)	1
Ω	Angular span of bearing segment ($\Omega = 360°$)	°
Ω_G	Angular span of lubricant pocket ($\Omega_G = 0°$)	°
ω	Angular speed of the rotor ($\omega = 2\pi N$)	rad/s
NOTE 1 *S* number is frequently referred to as Sommerfeld number. See References [4] to [6]. NOTE 2 In ISO 4378-5,[1] β is defined as the attitude angle or the temperature viscosity coefficient. In ISO 7902-1:2020, β is defined as the attitude angle. However, in this document, the attitude angle and the temperature viscosity coefficient are represented by ϕ and β, respectively, to avoid confusion due to duplication.		

Table 2 — Abbreviated terms and their designations

Abbreviated term	Designation
ISOADI	isothermal condition at the journal surface (lubricant film-journal interface) and adiabatic condition at the bearing surface (lubricant film-bearing interface)
THL	thermo-hydrodynamic lubrication

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5 Basis of calculation, assumptions, and preconditions

5.1 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 is incompressible.

d) Inertia effects, gravitational and magnetic force of the lubricant are negligible.

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

f) The lubricant adheres completely to the sliding surfaces.

g) The lubrication clearance gap in the convergent clearance is fully filled with the lubricant. The lubrication film in the divergent clearance ruptures and divides into multiple streamlets from $\varphi = \varphi_2$ where $p^* = \partial p^*/\partial\varphi = 0$. The streamlet shrinks its width as the lubrication clearance gap increases along the circumferential direction (φ-coordinate). The circumferential centerline of each streamlet coincides with each of the circumferential computational grid lines. The density and the thermal properties such as specific heat and thermal conductivity of the lubricant are uniform in the streamlets. There exist stationary air cavities between the streamlets.

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

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

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

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

l) The viscous force due to the cross-film gradient of the circumferential and the axial velocity components (φ- and z-coordinate) is significantly larger than the other viscous forces.

m) The temperature and viscosity of the lubricant film are two-dimensionally distributed within the lubrication clearance gap (φy-plane). They vary little in the axial direction (z-coordinate) and can be represented by the distribution in the midplane of the bearing width.

n) The heat flow due to heat conduction within the lubrication clearance gap is significantly larger in the cross-film direction (y-coordinate) than in the other directions (φ- and z-coordinate).

o) The viscous energy of the lubricant film dissipated by the cross-film gradient of the circumferential velocity component (φ-coordinate) is significantly larger than those due to the other velocity gradients.

p) The surface temperature of the rotating journal is uniform in the circumferential and the axial directions (φ- and z-coordinate).

q) The stationary bearing surface is adiabatic and does not allow heat flux to pass through.

r) The viscosity of the lubricant changes exponentially with the temperature.

s) Lubricant at a constant temperature is fed from the single pocket located at the apex of bearing. The angular span of the lubricant pocket Ω_G is negligibly small (Ω_G = 0°). The magnitude of the lubricant feed pressure is negligible in comparison with the lubricant film pressure. The lubricant feeding rate is the same as the lubricant leakage rate from the bearing side ends. The fed lubricant and the recirculating lubricant are fully mixed and form a uniform mixing temperature. It flows into the lubrication clearance gap.

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5.2 ISOADI THL model

5.2.1 General

ISOADI THL model, which is classified as one of simplified THL models in References [5] and [6], is adopted to prepare the design charts to read the maximum bearing temperature and the effective viscosity in lubricant film for the THL bearing design in the dimensionless form. The model assumes an isothermal journal surface (T_J constant) and an adiabatic bearing surface. The model uses four dimensionless bearing design variables (S number S_0, width ratio B^*, Peclet number Pe and relative temperature viscosity coefficient of the lubricant β^*) as input data. The model solves the generalized Reynolds equation[7] and the energy equation simultaneously, coupled with some related formulae such as lubricant film thickness, lubricant viscosity, etc. The maximum bearing temperature and the effective dynamic viscosity in the lubricant film are calculated for a wide range of bearing specifications and operating conditions in the dimensionless form.

5.2.2 Generalized Reynolds equation

The generalized Reynolds equation for a journal bearing with finite width is defined as Formula (1) in the dimensionless form:

$$\frac{\partial}{\partial\varphi}\left[h^{*3}F_2\frac{\partial p^*}{\partial\varphi}\right]+\frac{1}{4B^{*2}}\frac{\partial}{\partial z^*}\left[h^{*3}F_2\frac{\partial p^*}{\partial z^*}\right]=\frac{\partial}{\partial\varphi}\left[h^*\left(1-\frac{F_1}{F_0}\right)\right] \quad (1)$$

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where

$$F_0 = E_0(\varphi, 1),$$

$$F_1 = E_1(\varphi, 1),$$

$$F_2 = \int_0^1 \frac{y^{*2}}{\eta_{\mathrm{rel}}} dy^* - \frac{F_1^{\,2}}{F_0},$$

as

$$E_0(\varphi, y^*) = \int_0^{y^*} \frac{1}{\eta_{\mathrm{rel}}} dy^*, E_1(\varphi, y^*) = \int_0^{y^*} \frac{y^*}{\eta_{\mathrm{rel}}} dy^*$$

Formula (1) is numerically solved, taking into account the boundary conditions for the generation of pressure. See Reference [7] for the derivation of the generalized Reynolds equation.

5.2.3 Energy equation for lubricant film temperature distribution

The energy equation for the lubricant film temperature distribution along the midplane of the bearing width is defined as Formula (2) in the dimensionless form:

$$u^* \frac{\partial \Delta T_{\mathrm{r}}}{\partial \varphi} + \frac{1}{h^*}\left(v^* - y^* \frac{\partial h^*}{\partial \varphi} u^*\right)\frac{\partial \Delta T_{\mathrm{r}}}{\partial y^*} = \frac{\partial}{\partial y^*}\left(\frac{1}{Peh^{*2}} \frac{\partial \Delta T_{\mathrm{r}}}{\partial y^*}\right) + \frac{\eta_{\mathrm{rel}}}{h^{*2}}\left(\frac{\partial u^*}{\partial y^*}\right)^2 \quad (2)$$

where u^* and v^* are defined as shown by Formulae (3) and (4):

$$u^* = h^{*2}\left[E_1(\varphi, y^*) - \frac{F_1}{F_0} E_0(\varphi, y^*)\right]\frac{\partial p^*}{\partial \varphi} + \frac{E_0(\varphi, y^*)}{F_0} \quad (3)$$

$$v^* = -h^* \int_0^{y^*}\left(\frac{\partial u^*}{\partial \varphi} - \frac{y^*}{h^*}\frac{\partial h^*}{\partial \varphi}\frac{\partial u^*}{\partial y^*}\right) dy^* \quad (4)$$

Formula (2) is numerically solved, taking into account the simplified boundary conditions for the lubricant film temperature distribution. See Reference [4] for the derivation of the energy equation.

5.2.4 Formula for lubricant film thickness

The relative local lubricant film thickness of a circular cylindrical bearing is given in Formula (5):

$$h^* = 1 + \varepsilon_{\mathrm{v}} \cos\varphi + \varepsilon_{\mathrm{h}} \sin\varphi \quad (5)$$

where

$$\varepsilon_{\mathrm{v}} = \varepsilon \cos\phi,$$

$$\varepsilon_{\mathrm{h}} = \varepsilon \sin\phi.$$

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5.2.5 Formula for axial contraction ratio of lubricant streamlet

The axial contraction ratio α of the lubricant streamlet is calculated based on the law of conservation of mass and the assumption of uniform width in the cross-film direction (y-coordinate) as shown by Formula (6):

$$\alpha = \frac{h^{*}(\varphi_2)\left(1 - \dfrac{F_1(\varphi_2)}{F_0(\varphi_2)}\right)}{h^{*}(\varphi)\left(1 - \dfrac{F_1(\varphi)}{F_0(\varphi)}\right)} \tag{6}$$

In the case of $\alpha > 1$, α is corrected to 1.

5.2.6 Temperature-viscosity relationship

The relative dynamic viscosity of the lubricant is given as Formula (7) following the exponential model:

$$\eta_{\mathrm{rel}} = \mathrm{e}^{-\beta^{*} \Delta T_{\mathrm{r}}} \tag{7}$$

5.2.7 Zero net heat flow method for journal surface temperature

A uniform journal surface temperature $\Delta T_{\mathrm{J,r}}$ is determined by applying the zero net heat flow method in which the total heat flow exchanged between the journal and the surrounding lubricant film is equal to zero as Formula (8) in the dimensionless form:

$$\int_0^{2\pi} \alpha h^{*} \left(\frac{\partial \Delta T_{\mathrm{r}}}{\partial y^{*}}\right)_{y^{*}=1} d\varphi = 0 \tag{8}$$

5.2.8 Formula for mixing temperature

The uniform mixing temperature, which corresponds to the lubricant temperature at the entrance into the gap, is given as Formula (9) in the dimensionless form:

$$\Delta T_{1,\mathrm{r}} = \frac{e_3^{*}}{Q_3^{*} + 2Q_{\mathrm{sf}}^{*}} \tag{9}$$

where Q_{sf}^{*}, Q_3^{*} and e_3^{*} are defined as shown by Formulae (10) to (12):

$$Q_{\mathrm{sf}}^{*} = \int_0^{2\pi} \int_0^1 \left(h^{*} w^{*}\right)_{z^{*}=1/2} dy^{*} d\varphi \tag{10}$$

$$Q_3^{*} = \int_{-1/2}^{1/2} \int_0^1 \left(\alpha h^{*} u^{*}\right)_{\varphi=2\pi} dy^{*} dz^{*} \tag{11}$$

$$e_3^{*} = \int_{-1/2}^{1/2} \int_0^1 \left(\alpha h^{*} u^{*} \Delta T_{\mathrm{r}}\right)_{\varphi=2\pi} dy^{*} dz^{*} \tag{12}$$

As Formula (13):

$$w^{*} = \frac{h^{*2}}{4B^{*2}} \left[E_1(\varphi, y^{*}) - \frac{F_1}{F_0} E_0(\varphi, y^{*})\right] \frac{\partial p^{*}}{\partial z^{*}} \tag{13}$$

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5.2.9 Balance of bearing load and lubricant film reaction force

The relative eccentricity ε and the attitude angle ϕ are calculated so that the bearing load and the lubricant film reaction force are balanced. The equations of the balance in the horizontal and vertical direction are shown in Formulae (14) and (15), respectively.

$$-\frac{1}{2}\int_{-1/2}^{1/2}\int_{0}^{2\pi} p^{*}\sin\varphi d\varphi dz^{*}=0 \tag{14}$$

$$-\frac{1}{2}\int_{-1/2}^{1/2}\int_{0}^{2\pi} p^{*}\cos\varphi d\varphi dz^{*}=\frac{1}{S_0} \tag{15}$$

5.3 Boundary conditions

5.3.1 Pressure distribution of lubricant film

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

— at the entrance into the lubrication clearance gap: $p^{*}\left(\varphi_1, z^{*}\right)=0$;

— at the end of the lubrication clearance gap: $p^{*}\left(\varphi_3, z^{*}\right)=0$;

— at the bearing side ends: $p^{*}\left(\varphi,\pm 1/2\right)=0$;

— at the end of the hydrodynamic pressure build-up: $p^{*}\left(\varphi_2, z^{*}\right)=0$ and $\partial p^{*}/\partial\varphi\left(\varphi_2, z^{*}\right)=0$;

— in the ruptured region of the lubricant film from $\varphi = \varphi_2$ to $\varphi = \varphi_3$: $p^{*}\left(\varphi, z^{*}\right)=0$.

5.3.2 Temperature distribution of lubricant film

The boundary conditions for the lubricant film temperature distribution along the midplane of the bearing width fulfil the following thermally simplified conditions in the ISOADI THL model:

— at the entrance into the lubrication clearance gap: $\Delta T_{\mathrm{r}}\left(\varphi_1, y^{*}\right)=\Delta T_{1,\mathrm{r}}$;

— at the interface of the lubricant film and the rotating journal surface: $\Delta T_{\mathrm{r}}\left(\varphi,1\right)=\Delta T_{\mathrm{J,r}}$;

— at the interface of the lubricant film and the stationary bearing surface: $\partial\Delta T_{\mathrm{r}}/\partial y^{*}\left(\varphi,0\right)=0$.

NOTE The boundary condition at the end of the lubrication clearance gap ($\varphi = \varphi_3$) is not required because there exists no reverse flow of the lubricant from the lubricant pocket into the lubrication clearance gap, and therefore the lubricant pocket is always located on the downstream side.

5.4 Basis of calculation

By solving Formulae (1) to (15) under the related boundary conditions simultaneously and numerically, the maximum bearing temperature and the effective dynamic viscosity of the lubricant are obtained as ISOADI THL solutions.

The procedure is as follows:

a) After the values of four dimensionless bearing design variables S_0, B^*, Pe and β^* are given, the uniform distributions of p^*, ΔT_{r} and η_{rel} are initially set to be 0, 0 and 1, respectively, and the initial distribution of h^* is calculated from Formula (5) for the assumed values of ε and ϕ.

b) The generalized Reynolds equation [see Formula (1)] is solved for p^*.

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c) Then, relative velocity components in the circumferential, axial and cross-film directions and also contraction ratio of the lubricant streamlet are calculated from Formulae (3), (13), (4) and (6), respectively.

d) The energy equation (see Formula (2)) is solved, coupled with Formulae (8) and (9), for the simultaneous solution of ΔT_{r}, $\Delta T_{\mathrm{J,r}}$ and $\Delta T_{\mathrm{1,r}}$.

e) Next, η_{rel} is updated, utilizing ΔT_{r} (see Formula (7)).

f) The calculation steps from the Reynolds equation to the update of the dynamic viscosity of the lubricant are repeated until the relative error resulting from the update is sufficiently small.

g) When Formulae (14) and (15) are not satisfied, repeat the procedure from b) to f) with corrected values of ε and ϕ until Formulae (14) and (15) are satisfied.

h) The value of $\Delta T_{\mathrm{a,r}}$ is calculated as the arithmetic mean value of all temperature values calculated at all numerical grid points in the lubricant film.

i) The maximum value of ΔT_{r} and the value of $\eta_{\mathrm{a,r}}$ corresponding to $\Delta T_{\mathrm{a,r}}$ are saved as the values of $\Delta T_{\mathrm{B,max,r}}$ and $\eta_{\mathrm{eff,r}}$, respectively.

6 Design charts

6.1 General

A set of design charts shown in Figures 2 and 3 are given as design tools for obtaining the dimensionless maximum bearing temperature $\Delta T_{\mathrm{B,max,r}}$ and the effective relative dynamic viscosity $\eta_{\mathrm{eff,r}}$ in the lubricant film, respectively, from the given values of four dimensionless bearing design variables (S_0, B^*, Pe and β^*) that govern the steady-state characteristics of circular cylindrical bearings.

The axes of the design charts are the four dimensionless coordinate variables ($\gamma_{\Delta T_{\mathrm{B,max}}}$, $\kappa_{\Delta T_{\mathrm{B,max}}}$, $\gamma_{\eta_{\mathrm{eff,r}}}$ and $\kappa_{\eta_{\mathrm{eff,r}}}$), which are defined using S_0, B^*, Pe and β^*. Isotherms of $\widetilde{\Delta T_{\mathrm{B,max,r}}}$ and isoviscous lines of $\widetilde{\eta_{\mathrm{eff,r}}}$ are shown in Figures 2 and 3, respectively. A value of $\widetilde{\Delta T_{\mathrm{B,max,r}}}$ is obtained corresponding to the cross-point specified by the horizontal coordinate $\gamma_{\Delta T_{\mathrm{B,max}}}$ and the vertical one $\kappa_{\Delta T_{\mathrm{B,max}}}$ in Figure 2. A value of $\widetilde{\eta_{\mathrm{eff,r}}}$ is obtained corresponding to the cross-point specified by both coordinates $\gamma_{\eta_{\mathrm{eff,r}}}$ and $\kappa_{\eta_{\mathrm{eff,r}}}$ in Figure 3.

NOTE 1 The procedure for drawing the design charts based on a database is shown in Reference [6].

NOTE 2 The data set covers a wide range of calculated relative eccentricity ε from 0,03 to 0,93. Two design charts can be applied in the design of small to large circular cylindrical bearings operated in the hydrodynamic lubrication mode.

NOTE 3 The determination method of the formulae $\widetilde{\Delta T_{\mathrm{B,max,r}}} = \log_{10}\left(\Delta T_{\mathrm{B,max,r}}\ B^{*-0{,}154}\ Pe^{1{,}75}\ \beta^{*8}\right)$ and $\widetilde{\eta_{\mathrm{eff,r}}} = \log_{10}\left(\eta_{\mathrm{eff,r}}\ B^{*-0{,}01}\ Pe^{1{,}10}\ \beta^{*2{,}35}\right)$ from $\Delta T_{\mathrm{B,max,r}}$ and $\eta_{\mathrm{eff,r}}$ which are calculated by ISOADI THL with consideration of the dimensionless bearing design variables (S_0, B^*, Pe and β^*) is shown in Reference [6].

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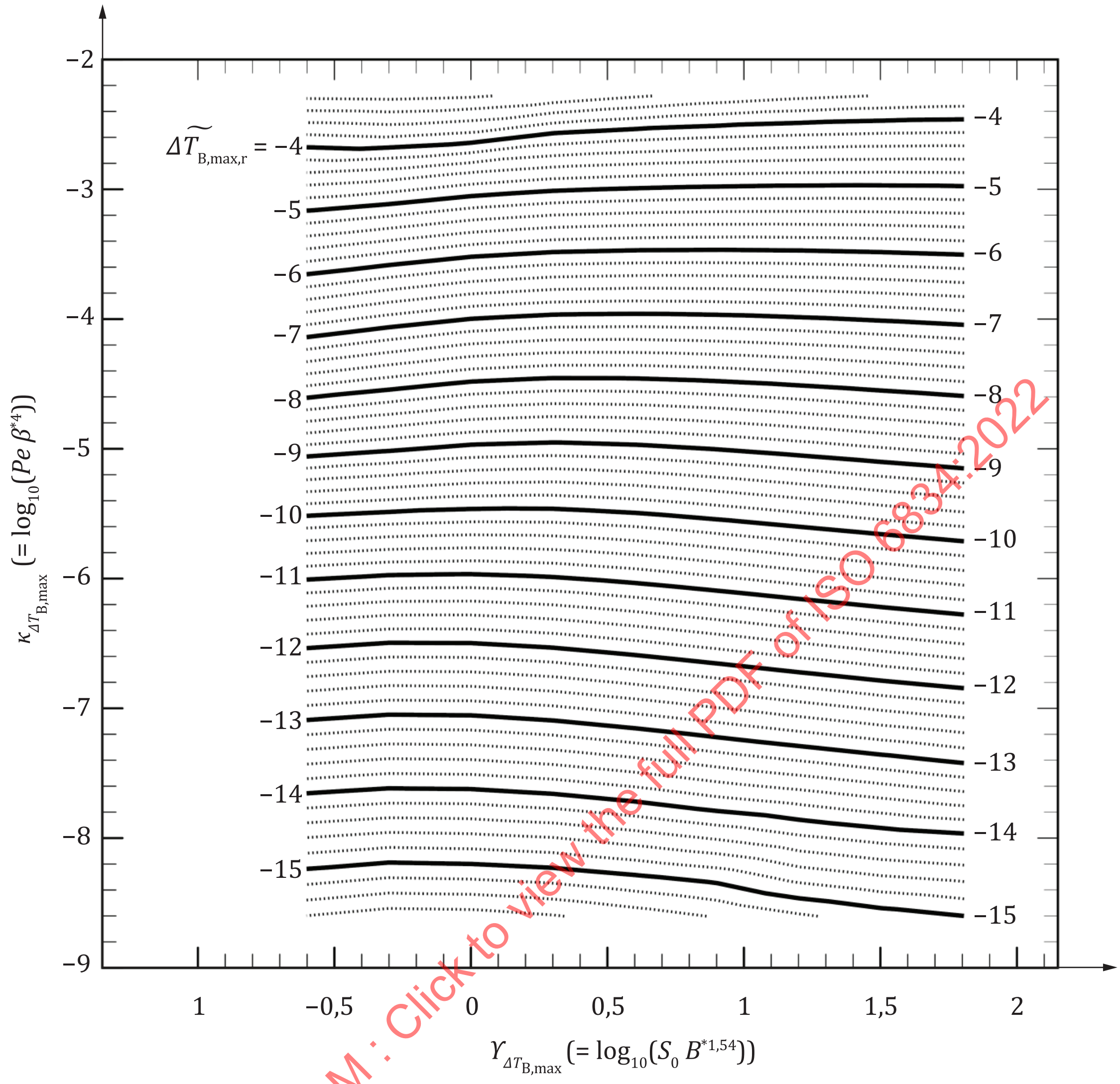

Figure 2 — Design chart for dimensionless maximum bearing temperature $\widetilde{\Delta T_{B,max,r}}$ ($= \log_{10}\left(\Delta T_{B,max,r}\ B^{*-0,154}\ Pe^{1,75}\ \beta^{*8}\right)$)

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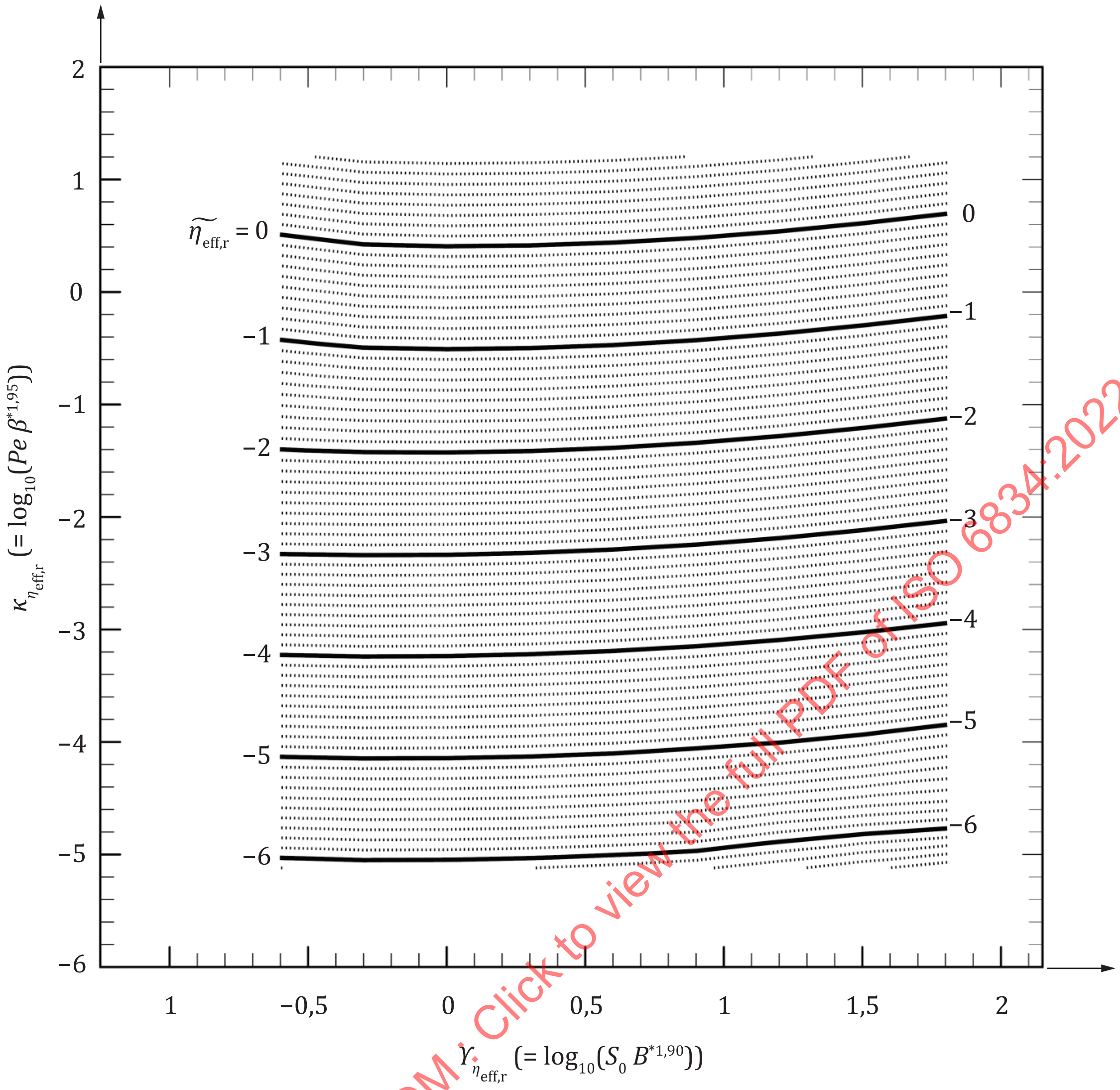

Figure 3 — Design chart for dimensionless effective lubricant film viscosity $\widetilde{\eta_{\mathrm{eff,r}}}$ ($= \log_{10}\left(\eta_{\mathrm{eff,r}}\, B^{*-0,01}\, Pe^{1,10}\, \beta^{*2,35}\right)$)

6.2 Input of design charts

The four dimensionless bearing design variables S_0, B^*, Pe and β^* specified to the set of the two design charts are defined as shown by Formulae (16) to (19):

$$S_0 = \frac{BD\eta_0\omega}{F\psi^2} \tag{16}$$

$$B^* = \frac{B}{D} \tag{17}$$

$$Pe = \frac{C_R{}^2 \rho c_p \omega}{\lambda} \tag{18}$$

$$\beta^* = \frac{\eta_0\omega\beta}{\rho c_p \psi^2} \tag{19}$$

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6.3 Axes of design charts

The four dimensionless coordinate variables shown in the design charts ($\gamma_{\Delta T_{\text{B,max}}}$, $\kappa_{\Delta T_{\text{B,max}}}$, $\gamma_{\eta_{\text{eff,r}}}$ and $\kappa_{\eta_{\text{eff,r}}}$) are defined using the four dimensionless bearing design variables S_0, B^*, Pe and β^* specified as shown by Formula (20) to (23):

$$\gamma_{\Delta T_{\text{B,max}}} = \log_{10}\left(S_0\ B^{*1,54}\right) \tag{20}$$

$$\kappa_{\Delta T_{\text{B,max}}} = \log_{10}\left(Pe\ \beta^{*4}\right) \tag{21}$$

$$\gamma_{\eta_{\text{eff,r}}} = \log_{10}\left(S_0\ B^{*1,90}\right) \tag{22}$$

$$\kappa_{\eta_{\text{eff,r}}} = \log_{10}\left(Pe\ \beta^{*1,95}\right) \tag{23}$$

NOTE The manner in which the four dimensionless coordinate variables are determined from the four dimensionless bearing design variables is shown in Reference [6].

6.4 Read of design charts

Figure 2 of double logarithmic coordinates shows the isotherms of logarithmic modified dimensionless maximum bearing temperature $\widetilde{\Delta T_{\text{B,max,r}}}$ $\left(=\log_{10}\left(\Delta T_{\text{B,max,r}}\ B^{*-0,154}\ Pe^{1,75}\ \beta^{*8}\right)\right)$ and reads $\widetilde{\Delta T_{\text{B,max,r}}}$ corresponding to the cross-point specified by the horizontal and vertical coordinates, $\gamma_{\Delta T_{\text{B,max}}}$ $\left(=\log_{10}\left(S_0\ B^{*1,54}\right)\right)$ and $\kappa_{\Delta T_{\text{B,max}}}$ $\left(=\log_{10}\left(Pe\ \beta^{*4}\right)\right)$.

Similarly, Figure 3 of double logarithmic coordinates shows the isoviscous lines of logarithmic modified relative effective dynamic viscosity in lubricant film $\widetilde{\eta_{\text{eff,r}}}$ $\left(=\log_{10}\left(\eta_{\text{eff,r}}\ B^{*-0,01}\ Pe^{1,10}\ \beta^{*2,35}\right)\right)$ and reads $\widetilde{\eta_{\text{eff,r}}}$ corresponding to the cross-point specified by the horizontal and vertical coordinates, $\gamma_{\eta_{\text{eff,r}}}$ $\left(=\log_{10}\left(S_0\ B^{*1,90}\right)\right)$ and $\kappa_{\eta_{\text{eff,r}}}$ $\left(=\log_{10}\left(Pe\ \beta^{*1,95}\right)\right)$.

6.5 Conversion of modified dimensionless values from design charts to dimensional ones

The values of dimensional maximum bearing temperature $T_{\text{B,max}}$ and dimensional effective relative dynamic viscosity in lubricant film η_{eff} are both obtained by a simple algebraic calculation. First, the values of $\widetilde{\Delta T_{\text{B,max,r}}}$ and $\widetilde{\eta_{\text{eff,r}}}$ are converted into the relative values of $\Delta T_{\text{B,max,r}}$ and $\eta_{\text{eff,r}}$ respectively as shown by Formulae (24) and (25):

$$\Delta T_{\text{B,max,r}} = B^{*0,154}\ Pe^{-1,75}\ \beta^{*-8} \times 10^{\widetilde{\Delta T_{\text{B,max,r}}}} \tag{24}$$

$$\eta_{\text{eff,r}} = B^{*0,01}\ Pe^{-1,10}\ \beta^{*-2,35} \times 10^{\widetilde{\eta_{\text{eff,r}}}} \tag{25}$$

Then, the relative values are converted into the dimensional values of $T_{\text{B,max}}$ and η_{eff} respectively as shown by Formulae (26) and (27):

$$T_{\text{B,max}} = \frac{\eta_0 \omega \Delta T_{\text{B,max,r}}}{\psi^2 \rho c_p} + T_0 \tag{26}$$

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$$\eta_{\mathrm{eff}} = \eta_0 \eta_{\mathrm{eff,r}} \tag{27}$$

By applying the effective dynamic viscosity in lubricant film η_{eff} to the isoviscous hydrodynamic lubrication model in accordance with ISO 7902-1:2020, the other operating parameters excluding the maximum bearing temperature $T_{\mathrm{B,max}}$ are calculated.

7 Calculation procedure

The value of maximum bearing temperature $T_{\mathrm{B,max}}$ and the value of the effective dynamic viscosity in the lubricant film η_{eff} shall be obtained from the set of the two design charts (see Figures 2 and 3) according to the following steps a) to e).

a) The values of four dimensionless bearing design variables S_0, B^*, Pe and β^* shall be calculated from the given bearing specifications and operating conditions.

b) The values of two dimensionless coordinate variables shown in the design chart of maximum bearing temperature ($\gamma_{\Delta T_{\mathrm{B,max}}}$ and $\kappa_{\Delta T_{\mathrm{B,max}}}$) shall be calculated.

c) The logarithmic modified dimensionless value of $\widetilde{\Delta T_{\mathrm{B,max,r}}}$ shall be obtained from the design chart given in Figure 2. The value shall be converted into the dimensional value of $T_{\mathrm{B,max}}$ via the relative value of $\Delta T_{\mathrm{B,max,r}}$.

d) Then the values of two dimensionless coordinate variables shown in the design chart of effective dynamic viscosity in lubricant film ($\gamma_{\eta_{\mathrm{eff,r}}}$ and $\kappa_{\eta_{\mathrm{eff,r}}}$) shall be calculated.

e) The logarithmic modified relative effective dynamic viscosity in lubricant film $\widetilde{\eta_{\mathrm{eff,r}}}$ shall be obtained from the design chart given in Figure 3. The value shall be converted into the dimensional value of η_{eff} via the relative value of $\eta_{\mathrm{eff,r}}$.

f) If the operating parameters excluding $T_{\mathrm{B,max}}$ are required, η_{eff} should be applied to the effective viscosity of the isoviscous hydrodynamic lubrication analysis model (e.g. ISO 7902-1:2020).

The sequence is outlined in the flow chart given in Figure 4. For optimization of design parameters, the above calculation sequence should be repeated.

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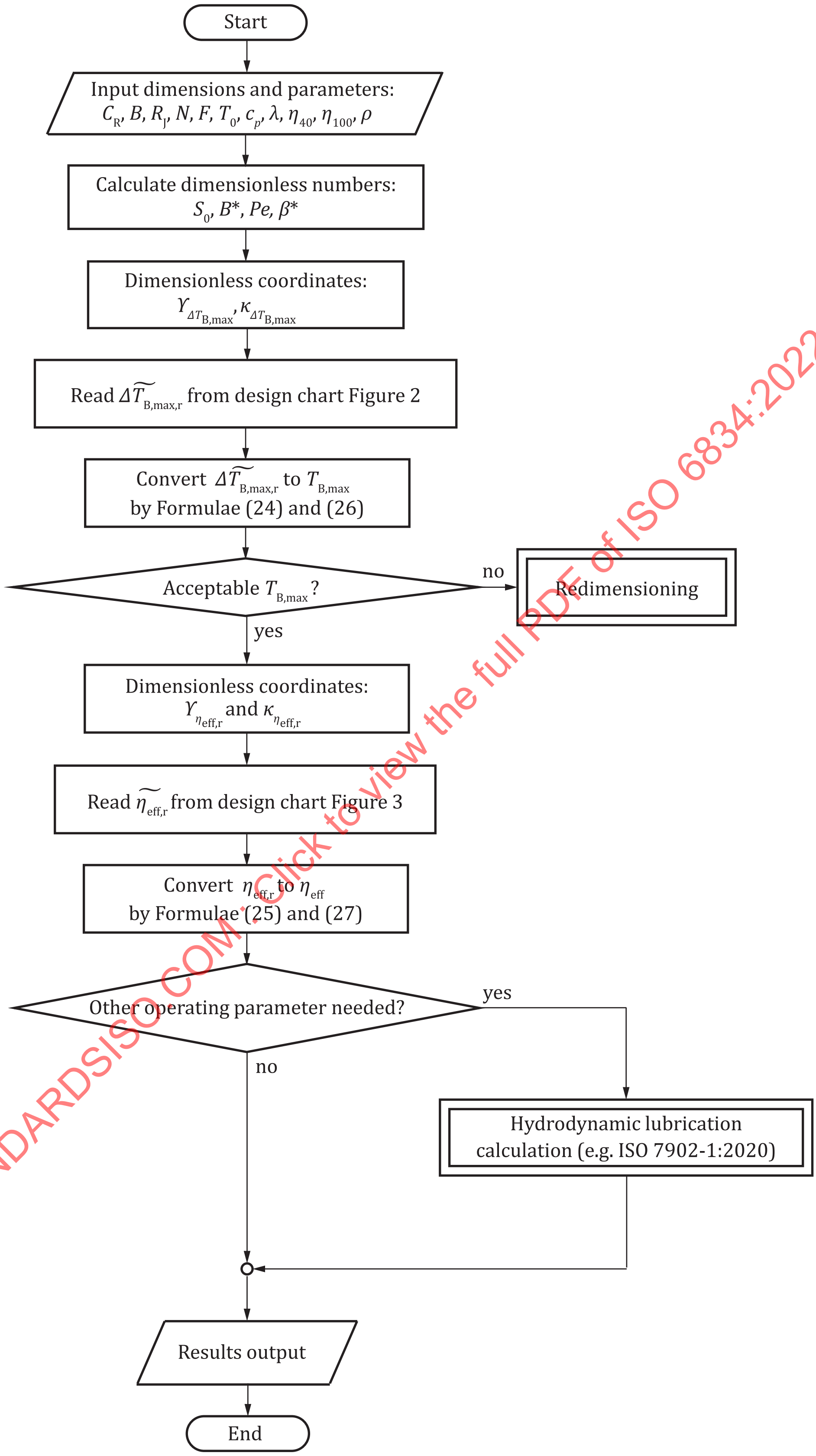

Figure 4 — Flow chart

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Annex A
(informative)

Calculation examples

A.1 Calculation based on the design charts

A.1.1 General

In A.1, a series of calculations based on the flow chart (see Figure 4) is to be made for the static characteristics of bearings.

First values of the maximum bearing temperature $T_{B,max}$, the effective dynamic viscosity in lubricant film η_{eff} and the effective lubricant film temperature T_{eff} are calculated based on the design charts.

Next, static characteristics of bearing are calculated based on ISO 7902-1:2020. In the calculation procedure of ISO 7902-1:2020, values of the η_{eff} and the T_{eff} are converged by iterative calculation. The calculated η_{eff} and T_{eff} by the design charts are adopted as initial values for iterative calculations. As a result, this calculation converges with fewer iterations than the recommended calculation procedure by ISO 7902-1:2020 (see A.2).

A.1.2 Input data for calculation

A statically loaded full circular cylindrical bearing is to be investigated. Dimensions and operational data are given in Table A.1. An employed lubricant (an oil of viscosity grade ISO VG 32) has the temperature dependence shown in Table A.2. The lubricant is supplied via a single pocket located at the apex of bearing.

Table A.1 — Dimensions and operational data

Designation	Unit	Value
Bearing load F	N	3 000
Rotational frequency of the rotor N	s^{-1}	33,33
Angular span of bearing segment Ω	°	360
Maximum diameter of the journal	m	$100,040 \times 10^{-3}$
Minimum diameter of the journal	m	$99,960 \times 10^{-3}$
Maximum inside diameter of the bearing	m	$100,240 \times 10^{-3}$
Minimum inside diameter of the bearing	m	$100,160 \times 10^{-3}$
Bearing width B	m	$75,0 \times 10^{-3}$
Lubricant supplying temperature T_0	°C	40
Width of lubrication pocket (for force-feed lubrication)	m	50×10^{-3}
Lubrication feed pressure (for force-feed lubrication)	Pa	$0,2 \times 10^{6}$
Dynamic viscosity of the lubricant at T_0 (η_0)	Pa·s	$27,0 \times 10^{-3}$
Dynamic viscosity of the lubricant at 100 °C (η_{100})	Pa·s	$4,37 \times 10^{-3}$
Density of the lubricant ρ	kg/m^3	$0,9 \times 10^{3}$
Specific heat of the lubricant c_p	J/(kg·K)	2×10^{3}
Thermal conductivity of the lubricant λ	W/(m·K)	0,13
Linear heat expansion coefficient of the bearing	K^{-1}	23×10^{-6}
Linear heat expansion coefficient of the journal	K^{-1}	11×10^{-6}

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Table A.2 — Temperature dependence of ISO VG 32

Lubricant temperature T °C	Dynamic viscosity of the lubricant η Pa·s
40	$27,0 \times 10^{-3}$
60	$14,7 \times 10^{-3}$
80	$8,02 \times 10^{-3}$
100	$4,37 \times 10^{-3}$

A.1.3 Calculation of $T_{B,max}$, η_{eff} and T_{eff} based on the design charts

The four dimensionless constant numbers [see Formulae (16), (17), (18) and (19)]:

$$S_0 = \frac{75{,}0\times10^{-3}\times100{,}2\times10^{-3}\times27{,}0\times10^{-3}\times209{,}4}{3\,000\times\left(1{,}996\times10^{-3}\right)^2} = 3{,}55$$

$$B^* = \frac{75{,}0\times10^{-3}}{100{,}2\times10^{-3}} = 0{,}749$$

$$Pe = \frac{\left(100\times10^{-6}\right)^2\times0{,}9\times10^3\times2\times10^3\times209{,}4}{0{,}13} = 29{,}0$$

$$\beta^* = \frac{27{,}0\times10^{-3}\times209{,}4\times30{,}4\times10^{-3}}{0{,}9\times10^3\times2\times10^3\times\left(1{,}996\times10^{-3}\right)^2} = 0{,}024\,0$$

where

$$\omega = 2\times\pi\times33{,}33 = 209{,}4\ \mathrm{rad/s}$$

$$D = \frac{(100{,}240+100{,}160)\times10^{-3}}{2} = 100{,}2\times10^{-3}\ \mathrm{m}$$

$$R = \frac{100{,}2\times10^{-3}}{2} = 50{,}1\times10^{-3}\ \mathrm{m}$$

$$D_J = \frac{(100{,}040+99{,}960)\times10^{-3}}{2} = 100{,}0\times10^{-3}\ \mathrm{m}$$

$$R_J = \frac{100{,}0\times10^{-3}}{2} = 50{,}0\times10^{-3}\ \mathrm{m}$$

$$C_R = (50{,}1-50{,}0)\times10^{-3} = 100\times10^{-6}\ \mathrm{m}$$

$$\psi = \frac{100\times10^{-6}}{50{,}10\times10^{-3}} = 1{,}996\times10^{-3}$$

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$$\beta = \ln\left(\frac{27{,}0\times10^{-3}}{4{,}37\times10^{-3}}\right)\times\frac{1}{60} = 30{,}4\times10^{-3}\ 1/°\mathrm{C}$$

Values of the horizontal coordinate $\gamma_{\Delta T_{\mathrm{B,max}}}$ and the vertical coordinate $\kappa_{\Delta T_{\mathrm{B,max}}}$ in the design chart for $\widetilde{\Delta T_{\mathrm{B,max,r}}}$ [see Formulae (20) and (21)]:

$$\gamma_{\Delta T_{\mathrm{B,max}}} = \log_{10}\left(3{,}55\times0{,}749^{1{,}54}\right) = 0{,}357$$

$$\kappa_{\Delta T_{\mathrm{B,max}}} = \log_{10}\left(29{,}0\times0{,}024\,0^{4}\right) = -5{,}02$$

Read a value of the $\widetilde{\Delta T_{\mathrm{B,max,r}}}$ in the design chart (see Figure 2):

$$\widetilde{\Delta T_{\mathrm{B,max,r}}} = -9{,}10$$

$\Delta T_{\mathrm{B,max,r}}$ [see Formula (24)]:

$$\Delta T_{\mathrm{B,max,r}} = 0{,}749^{0{,}154}\times29{,}0^{-1{,}75}\times0{,}024\,0^{-8}\times10^{-9{,}10} = 19{,}0$$

Maximum bearing temperature $T_{\mathrm{B,max}}$ [see Formula (26)]:

$$T_{\mathrm{B,max}} = \frac{27{,}0\times10^{-3}\times209{,}4\times19{,}0}{\left(1{,}996\times10^{-3}\right)^{2}\times0{,}9\times10^{3}\times2\times10^{3}} + 40 = 55{,}0\ °\mathrm{C}$$

Values of the horizontal coordinate $\gamma_{\eta_{\mathrm{eff,r}}}$ and the vertical coordinate $\kappa_{\eta_{\mathrm{eff,r}}}$ in the design chart for $\widetilde{\eta_{\mathrm{eff,r}}}$ [see Formulae (22) and (23)]:

$$\gamma_{\eta_{\mathrm{eff,r}}} = \log_{10}\left(3{,}55\times0{,}749^{1{,}90}\right) = 0{,}312$$

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$$\kappa_{\eta_{\text{eff},r}} = \log_{10}\left(29{,}0 \times 0{,}024\,0^{1{,}95}\right) = -1{,}70$$

Read a value of the $\widetilde{\eta_{\text{eff},r}}$ in the design chart (see Figure 3):

$$\widetilde{\eta_{\text{eff},r}} = -2{,}30$$

$\eta_{\text{eff,r}}$ [see Formula (25)]:

$$\eta_{\text{eff},r} = 0{,}749^{0{,}01} \times 29{,}0^{-1{,}10} \times 0{,}024\,0^{-2{,}35} \times 10^{-2{,}30} = 0{,}788$$

Effective dynamic viscosity in lubricant film η_{eff} [see Formula (27)]:

$$\eta_{\text{eff}} = 27{,}0 \times 10^{-3} \times 0{,}788 = 0{,}021\,3\ \text{Pa}\cdot\text{s}$$

A value of the effective lubricant film temperature T_{eff} is obtained by modifying Formula (7):

$$\frac{\eta_{\text{eff}}}{\eta_0} = e^{-\beta^* \frac{\psi^2 \rho c_p (T_{\text{eff}} - T_0)}{\eta_0 \omega}}$$

therefore

$$T_{\text{eff}} = T_0 - \frac{\eta_0 \omega}{\beta^* \psi^2 \rho c_p} \ln\left(\frac{\eta_{\text{eff}}}{\eta_0}\right) = T_0 - \frac{\eta_0 \omega}{\beta^* \psi^2 \rho c_p} \ln\left(\eta_{\text{eff},r}\right) =$$

$$40 - \frac{27{,}0 \times 10^{-3} \times 209{,}4}{0{,}024\,0 \times \left(1{,}996 \times 10^{-3}\right)^2 \times 0{,}9 \times 10^3 \times 2 \times 10^3} \times \ln(0{,}788) = 47{,}83\ ^\circ\text{C}.$$

A.1.4 First step in the iterative calculation based on ISO 7902-1:2020

ISO 7902-1:2020 recommends the following procedure to calculate initial values of the η_{eff} and T_{eff} (see ISO 7902-1:2020, 7.4):

a) Calculating initial lubricant outlet temperature ($T_{\text{ex},0}$): $T_{\text{ex},0} = T_0 + 20$.

b) Calculating initial T_{eff}: $T_{\text{eff}} = 0{,}5 \times \left(T_{\text{ex},0} + T_0\right)$.

c) Calculating initial η_{eff} at initial T_{eff} from the given parameters.

On the other hand, in this calculation, calculation results based on the design charts are adopted as the initial values of η_{eff} and T_{eff}:

$$\eta_{\text{eff}} = 0{,}021\,3\ \text{Pa}\cdot\text{s}$$

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