
**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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Published in Switzerland

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Foreword

ISO (the International Organization for Standardization) is a worldwide federation of national standards bodies (ISO member bodies). The work of preparing International Standards is normally carried out through ISO technical committees. Each member body interested in a subject for which a technical committee has been established has the right to be represented on that committee. International organizations, governmental and non-governmental, in liaison with ISO, also take part in the work. ISO collaborates closely with the International Electrotechnical Commission (IEC) on all matters of electrotechnical standardization.

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*.

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 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](#).

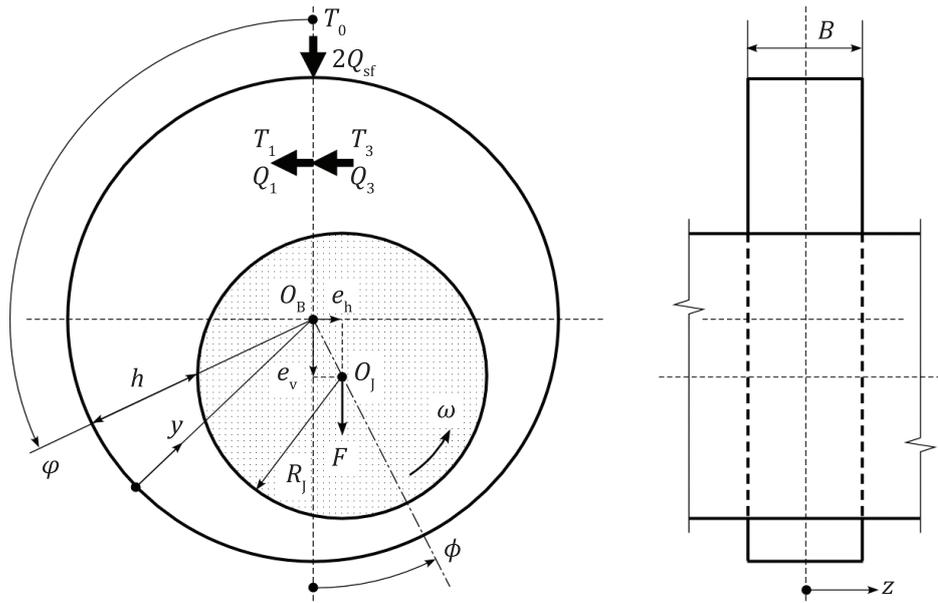


Figure 1 — Illustration of symbols

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, [4] 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 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 [$\widetilde{\Delta T_{B,max,r}} = \log_{10} (\Delta T_{B,max,r} B^{*-0,154} Pe^{1,75} \beta^{*8})$]	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.

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 [$v^* = v/(C_R\omega)$]	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
$\gamma_{\Delta T_{B,\max}}$	The first logarithmic relative coordinate variable in design chart for difference between maximum bearing temperature and lubricant supplying temperature [$\gamma_{\Delta T_{B,\max}} = \log_{10}(S_0 B^{*1,54})$]	1
$\gamma_{\eta_{\text{eff},r}}$	The first logarithmic relative coordinate variable in design chart for effective relative dynamic viscosity in lubricant film [$\gamma_{\eta_{\text{eff},r}} = \log_{10}(S_0 B^{*1,90})$]	1
ε	Relative eccentricity ($\varepsilon = e/C_R$)	1
ε_h	Relative eccentricity in the horizontal direction ($\varepsilon_h = e_h/C_R$)	1
ε_v	Vertical eccentricity in the vertical direction ($\varepsilon_v = e_v/C_R$)	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
$\eta_{a,r}$	Relative average dynamic viscosity in lubricant film corresponding to relative difference $\Delta T_{a,r}$ between average temperature of lubricant film and lubricant supplying temperature ($\eta_{a,r} = \eta_a/\eta_0$)	1
η_{eff}	Effective dynamic viscosity in lubricant film	Pa·s
$\eta_{\text{eff},r}$	Effective relative dynamic viscosity in lubricant film ($\eta_{\text{eff},r} = \eta_{\text{eff}}/\eta_0$)	1
η_{rel}	Relative dynamic viscosity of the lubricant ($\eta_{\text{rel}} = \eta/\eta_0$)	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
$\widetilde{\eta}_{\text{eff},r}$	Logarithmic modified effective relative dynamic viscosity in lubricant film [$\widetilde{\eta}_{\text{eff},r} = \log_{10}(\eta_{\text{eff},r} B^{*-0,01} Pe^{1,10} \beta^{*2,35})$]	1
$\kappa_{\Delta T_{B,\max}}$	The second logarithmic relative coordinate variable in design chart for difference between maximum bearing temperature and lubricant supplying temperature [$\kappa_{\Delta T_{B,\max}} = \log_{10}(Pe \beta^{*4})$]	1
$\kappa_{\eta_{\text{eff},r}}$	The second logarithmic relative coordinate variable in design chart for effective relative dynamic viscosity of lubricant film [$\kappa_{\eta_{\text{eff},r}} = \log_{10}(Pe \beta^{*1,95})$]	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.

Table 1 (continued)

Symbol	Designation	Unit
π	Circular constant ($\pi = 3,141\ 592 \dots$)	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 ($\Omega = 360^\circ$)	°
Ω_G	Angular span of lubricant pocket ($\Omega_G = 0^\circ$)	°
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, 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

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.

- The lubricant corresponds to a Newtonian fluid.
- All lubricant flows are laminar.
- The lubricant is incompressible.
- Inertia effects, gravitational and magnetic force of the lubricant are negligible.
- The components forming the lubrication clearance gap are rigid or their deformation is negligible; their surfaces are ideal circular cylinders.
- The lubricant adheres completely to the sliding surfaces.
- 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.
- 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 ($\Omega_G = 0^\circ$). 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.

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:

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