TECHNICAL SPECIFICATION

# ISO/TS 31657-1

First edition
2020-06

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

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

Reference number
ISO/TS 31657-1:2020(E)

Documents
Document Sharing Hub

© ISO 2020

Full document details: ISO TS 31657-1-2020

**COPYRIGHT PROTECTED DOCUMENT**

© ISO 2020

All rights reserved. Unless otherwise specified, or required in the context of its implementation, no part of this publication may be reproduced or utilized otherwise in any form or by any means, electronic or mechanical, including photocopying, or posting on the internet or an intranet, without prior written permission. Permission can be requested from either ISO at the address below or ISO's member body in the country of the requester.

ISO copyright office
CP 401 • Ch. de Blandonnet 8
CH-1214 Vernier, Geneva
Phone: +41 22 749 01 11
Email: copyright@iso.org
Website: www.iso.org

Published in Switzerland



© ISO 2020 – All rights reserved

Documents
Document Sharing Hub

Full document details: ISO TS 31657-1-2020

# Contents

Page

**Foreword** ..... **iv**
**Introduction** ..... **v**
**1 Scope** ..... **1**
**2 Normative references** ..... **1**
**3 Terms and definitions** ..... **1**
**4 Symbols and units** ..... **1**
**5 General principles, assumptions and preconditions** ..... **7**
**6 Calculation method** ..... **9**
6.1 General ..... 9
6.2 Load carrying capacity ..... 11
6.3 Frictional power ..... 11
6.4 Lubricant flow rate ..... 12
6.5 Heat balance ..... 13
6.6 Maximum lubricant film temperature ..... 14
6.7 Maximum lubricant film pressure ..... 15
6.8 Operating states ..... 15
6.9 Further influencing parameters ..... 15
6.10 Stiffness and damping coefficients ..... 16
**7 Figures** ..... **18**
**Annex A** (informative) **Calculation examples** ..... **23**
**Bibliography** ..... **37**

© ISO 2020 – All rights reserved


Documents
Document Sharing Hub
Full document details: ISO TS 31657-1-2020

# 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).

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

Any trade name used in this document is information given for the convenience of users and does not constitute an endorsement.

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

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

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.


© ISO 2020 – All rights reserved
Documents
Document Sharing Hub
Full document details: ISO TS 31657-1-2020

# 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, $\overline{p} \cdot U_J$, 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.

© ISO 2020 – All rights reserved


Documents
Document Sharing Hub
Full document details: ISO TS 31657-1-2020

Documents
Document Sharing Hub

Full document details: ISO TS 31657-1-2020

# 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 |

© ISO 2020 – All rights reserved

Documents Document Sharing Hub
Full document details: ISO TS 31657-1-2020

**Table 1** *(continued)*

| Symbol | Description | Unit |
|---|---|---|
| $b_P$ | Width of lubricant pocket | m |
| $b_P^*$ | Relative width of lubricant pocket, as given by: $b_P^* = \frac{b_P}{B}$ | 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: $c_{ik}^* = \frac{\psi_{eff}^3}{2 \cdot B \cdot \eta_{eff} \cdot \omega} \cdot c_{ik}\ (i,k=1,2)$ | 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: $d_{ik}^* = \frac{\psi_{eff}^3}{2 \cdot B \cdot \eta_{eff} \cdot \omega} \cdot \omega \cdot d_{ik}\ (i,k=1,2)$ | 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 |
| $\Delta F_x$ | Component of additional dynamic force in $x$-direction | N |
| $\Delta F_y$ | Component of additional dynamic force in $y$-direction | N |
| $\Delta F_x^*$ | Component of additional dynamic force parameter in $x$-direction, as given by: $\Delta F_x^* = \frac{\Delta F_x \cdot \psi_{eff}^2}{B \cdot D \cdot \eta_{eff} \cdot \omega}$ | 1 |
| $\Delta F_y^*$ | Component of additional dynamic force parameter in $y$-direction, as given by: $\Delta F_y^* = \frac{\Delta F_y \cdot \psi_{eff}^2}{B \cdot D \cdot \eta_{eff} \cdot \omega}$ | 1 |
| $F_f$ | Friction force, as given by: $F_f = f \cdot F$ | N |
| $F_f^*$ | Friction force parameter, as given by: $F_f^* = \frac{f}{\psi_{eff}} \cdot So$ | 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 |


© ISO 2020 – All rights reserved
Documents
Document Sharing Hub
Full document details: ISO TS 31657-1-2020

**Table 1** *(continued)*

| Symbol | Description | Unit |
|---|---|---|
| $h^*(\varphi)$ | Relative local lubricant film thickness, as given by: $h^*(\varphi)=\frac{h(\varphi)}{C_R}$ | 1 |
| $h_{lim,tr}$ | Minimum admissible lubricant film thickness at transition to mixed friction | m |
| $h^*_{lim,tr}$ | Minimum admissible relative lubricant film thickness at transition to mixed friction, as given by: $h^*_{lim,tr}=\frac{h_{lim,tr}}{C_{R,eff}}$ | 1 |
| $h_{min}$ | Minimum lubricant film thickness, minimum gap | m |
| $h^*_{min}$ | Minimum relative lubricant film thickness, minimum relative gap, as given by: $h^*_{min}=\frac{h_{min}}{C_{R,eff}}$ | 1 |
| $h_{min,tr}$ | Minimum lubricant film thickness at transition to mixed friction | m |
| $h^*_{min,tr}$ | Minimum relative lubricant film thickness at transition to mixed friction, as given by: $h^*_{min,tr}=\frac{h_{min,tr}}{C_{R,eff}}$ | 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 |
| $h_{0,max}$ | Maximum gap at $\varepsilon=0$ | m |
| $h^*_{0,max}$ | Maximum relative gap at $\varepsilon=0$, gap ratio, as given by: $h^*_{0,max}=\frac{h_{0,max}}{C_R}$ | 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 |
| $K_{P,eff}$ | Effective profile factor | 1 |
| $K_{P,20}$ | 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 |
| $P_{th,L}$ | Heat flow via the lubricant | W |
| $p$ | Lubricant film pressure, local lubricant film pressure | Pa |

© ISO 2020 – All rights reserved

Documents
Document Sharing Hub
Full document details: ISO TS 31657-1-2020

**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: $p^*_{en}=\frac{p_{en}\cdot\psi^2_{eff}}{\eta_{eff}\cdot\omega}$ | 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 |
| $\Delta R_B$ | 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 |


© ISO 2020 – All rights reserved
Documents
Document Sharing Hub
Full document details: ISO TS 31657-1-2020

**Table 1** *(continued)*

| Symbol | Description | Unit |
|---|---|---|
| $So$ | Sommerfeld number, as given by: $So=\frac{F\cdot\psi_{\mathrm{eff}}^2}{B\cdot D\cdot\eta_{\mathrm{eff}}\cdot\omega}$ | 1 |
| $So_{\mathrm{tr}}$ | Sommerfeld number at transition to mixed friction | 1 |
| $S$ | Displacement amplitude of the rotor (mechanical oscillation) | m |
| $T$ | Temperature | °C |
| $\Delta T$ | Heating of lubricant between bearing entrance and exit, as given by: $\Delta T=T_{\mathrm{ex}}-T_{\mathrm{en}}$ | K |
| $\Delta T_{\mathrm{lim}}$ | Maximum admissible heating of lubricant between bearing entrance and exit | K |
| $T_{\mathrm{B}}$ | Bearing temperature | °C |
| $T_{\mathrm{eff}}$ | Effective temperature of lubricant film | °C |
| $T_{\mathrm{en}}$ | Lubricant temperature at the bearing entrance | °C |
| $T_{\mathrm{ex}}$ | Lubricant temperature at the bearing exit | °C |
| $T_{\mathrm{J}}$ | Journal temperature | °C |
| $T_{\mathrm{lim}}$ | Maximum admissible bearing temperature | °C |
| $T_{\mathrm{max}}$ | Maximum temperature of lubricant film | °C |
| $\Delta T_{\mathrm{max}}$ | Difference between maximum temperature of lubricant film and lubricant temperature in the lubricant pocket, as given by: $\Delta T_{\mathrm{max}}=T_{\mathrm{max}}-T_1$ | K |
| $\Delta T_{\mathrm{max}}^*$ | Non-dimensional difference between maximum temperature of lubricant film and lubricant temperature in the lubricant pocket, as given by: $\Delta T_{\mathrm{max}}^*=\frac{\rho\cdot c_p\cdot\psi_{\mathrm{eff}}}{\bar{p}\cdot f}\cdot\Delta T_{\mathrm{max}}$ | 1 |
| $T_1$ | Lubricant temperature at the entrance into the lubrication gap (circumferential direction) | °C |
| $\Delta T_1$ | Difference between lubricant temperature at the entrance into the lubrication gap and lubricant temperature at the bearing entrance, as given by: $\Delta T_1=T_1-T_{\mathrm{en}}$ | K |
| $T_2$ | Lubricant temperature at pressure profile trailing edge (circumferential direction) | °C |
| $\Delta T_2$ | 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 |
| $U_{\mathrm{J}}$ | Circumferential speed of the journal, sliding velocity<br>$U_{\mathrm{J}}=\omega\cdot R_{\mathrm{J}}$ | m/s |
| $U_{\mathrm{tr}}$ | Circumferential speed at transition to mixed friction | m/s |
| $U_{\mathrm{lim,\,tr}}$ | Minimum admissible circumferential speed at transition to mixed friction | m/s |
| $u$ | Velocity component in the $\varphi$-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: $x^*=\frac{x}{C_{\mathrm{R}}}$ | 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: $y^*=\frac{y}{C_{\mathrm{R}}}$ | 1 |

AVA Documents
Document Sharing Hub

© ISO 2020 – All rights reserved


Full document details: ISO TS 31657-1-2020

**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 | $K^{-1}$ |
| $\alpha_{l,J}$ | Linear thermal expansion coefficient of journal material | $K^{-1}$ |
| $\beta$ | Attitude angle (angular position of journal eccentricity related to the direction of load) | ° |
| $\beta_{h,\min}$ | Angle between direction of load and position of minimum lubricant film thickness | ° |
| $\delta_J$ | Journal misalignment angle (angular deviation of journal) | ° |
| $\varepsilon$ | Relative eccentricity: $\varepsilon = \frac{e}{C_{R,\mathrm{eff}}}$ | 1 |
| $\eta$ | Dynamic viscosity of the lubricant | Pa s |
| $\eta_{\mathrm{eff}}$ | Effective dynamic viscosity in the lubricant film | Pa s |
| $\rho$ | Density of the lubricant | $kg/m^3$ |
| $\varphi$ | Angular coordinate in circumferential direction | ° |
| $\varphi_F$ | Angular coordinate of pivot position of pad (tilting-pad bearing) | ° |
| $\varphi_P$ | Angular coordinate of lubricant pocket centreline | ° |
| $\varphi_0$ | 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) | ° |
| $\varphi_1$ | Angular coordinate at the entrance into the gap | ° |
| $\varphi_2$ | Angular coordinate at the end of the hydrodynamic pressure build-up | ° |
| $\varphi_3$ | Angular coordinate at the exit of the gap | ° |
| $\psi$ | Relative bearing clearance, as given by: $\psi = \frac{C_R}{R}$ | ‰ |
| $\Delta\psi$ | Tolerance of $\psi$, as given by: $\Delta\psi = \psi_{\max} - \psi_{\min}$ | ‰ |
| $\psi_{\mathrm{eff}}$ | Effective relative bearing clearance | ‰ |
| $\psi_{\max}$ | Maximum value of $\psi$ | ‰ |
| $\psi_{\min}$ | Minimum value of $\psi$ | ‰ |
| $\Delta\psi_{\mathrm{th}}$ | Thermal change of $\psi$ | ‰ |
| $\psi_{20}$ | Relative bearing clearance at 20 °C | ‰ |
| $\Omega$ | Angular span of bearing sliding surface (segment or pad), as given by: $\Omega = \varphi_3 - \varphi_1$ | ° |
| $\Omega_F$ | Angular distance between leading edge and pivot position of pad (tilting-pad bearing), as given by: $\Omega_F = \varphi_F - \varphi_1$ | ° |
| $\Omega_F^*$ | Relative angular distance between leading edge and pivot position of pad (tilting-pad bearing), as given by: $\Omega_F^* = \Omega_F / \Omega$ | 1 |
| $\Omega_P$ | Angular span of lubricant pocket, as given by: $\Omega_P = \frac{360°}{Z} - \Omega$ | ° |
| $\omega$ | Angular speed of the rotor, as given by: $\omega = 2 \cdot \pi \cdot N$ | $s^{-1}$ |
| $\omega_{\mathrm{tr}}$ | Angular speed at transition to mixed friction | $s^{-1}$ |


© ISO 2020 – All rights reserved
Documents
Document Sharing Hub
Full document details: ISO TS 31657-1-2020

## 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 $\varphi$ 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 $\Omega_i=\varphi_{3,i}-\varphi_{1,i}$:

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

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 ($\varepsilon$, $\beta$), 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.

© ISO 2020 – All rights reserved

Documents
Document Sharing Hub
Full document details: ISO TS 31657-1-2020

The pressure formation in the lubrication gaps is basically calculated with the numerical solutions of the Reynolds differential equation for a finite bearing width:

$$\frac{1}{R_J^2}\cdot\frac{\partial}{\partial\varphi}\left(h^3\cdot\frac{\partial p}{\partial\varphi}\right)+\frac{\partial}{\partial z}\left(h^3\cdot\frac{\partial p}{\partial z}\right)=6\cdot\eta\cdot\omega\cdot\frac{\partial h}{\partial\varphi} \tag{4}$$

with $\omega=2\cdot\pi\cdot N$ angular speed of the rotor.

For derivation of the Reynolds differential equation, reference is made to Reference [8], for the numerical solution to Reference [9].

When solving Formula (4), the following idealising assumptions and preconditions are made, whose permissibility shall be estimated according to Clause 6, if necessary[10].

a) The lubricant corresponds to a Newtonian fluid.

b) All flow processes of the lubricant are laminar.

c) The lubricant adheres fully to the sliding surfaces.

d) The lubricant is incompressible.

e) At the leading edge of the segment or pad, the lubrication gap is completely filled with lubricant.

f) Inertia effects, gravitation and magnetic forces of the lubricant are negligible.

g) The components forming the lubrication gap are rigid or their deformation is negligible; the surfaces of the journal and bearing bore are ideal circular cylinders or cylindrical segments.

h) The curvature radii of the surfaces moving relative to one another are large in comparison to the lubricant film thicknesses.

i) The lubricant film thickness in an axial direction ($z$ coordinate) is constant.

j) Pressure changes in the lubricant film normal to the sliding surfaces (in the lubricant film thickness direction) are negligible.

k) A movement normal to the sliding surfaces (in the lubricant film thickness direction) is not considered here, in contrast to 6.10.

l) The lubricant film is isoviscous in the entire lubrication gap.

m) The lubricant is supplied at the leading edge of the segments or pads respectively; the level of the supply pressure is negligible compared to the lubricant film pressures themselves.

The boundary conditions for the lubricant film pressure build-up satisfy the continuity condition.

The following applies respectively to the individual segments or pads (see Figures 2 and 4):

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

— in the lubrication pocket and on the sealing land $p(\varphi,z)=0$;

— at the pressure profile trailing edge $p[\varphi_2(z),z]=\frac{\partial p}{\partial\varphi}[\varphi_2(z),z]=0$;

— at the beginning of cavitation area $p[\varphi(z),z]=\frac{\partial p}{\partial\varphi}[\varphi(z),z]=\frac{\partial p}{\partial z}[\varphi(z),z]=0$;

— at the end of cavitation area $p[\varphi(z),z]=0$.

The cavitation theory according to Jakobsson, Floberg and Olsson[15][16] is used in the cavitation area and on its edge for fulfilment of the continuity condition.


© ISO 2020 – All rights reserved
Documents
Document Sharing Hub
Full document details: ISO TS 31657-1-2020