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Plain bearings — Hydrostatic plain journal bearings with drainage grooves under steady-state conditions —

Part 1: Calculation of oil-lubricated plain journal bearings with drainage grooves

Paliers lisses — Paliers lisses radiaux hydrostatiques avec rainures d'écoulement fonctionnant en régime stationnaire —

Partie 1: Calcul pour la lubrification des paliers lisses radiaux avec rainures d'écoulement

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Foreword

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The committee responsible for this document is ISO/TC 123, *Plain bearings*.

This second edition cancels and replaces the first edition (ISO 12167-1:2001), of which it constitutes a minor revision.

ISO 12167 consists of the following parts, under the general title *Plain bearings — Hydrostatic plain journal bearings with drainage grooves under steady-state conditions*:

— *Part 1: Calculation of oil-lubricated plain journal bearings with drainage grooves*

— *Part 2: Characteristic values for the calculation of oil-lubricated plain journal bearings with drainage grooves*

Introduction

Hydrostatic bearings use external lubrication to support pressure on the bearings; thus, are less prone to wear and tear, run quietly, and have wide useable speed, as well as high stiffness and damping capacity. These properties also demonstrate the special importance of plain journal bearings in different fields of application such as in machine tools.

Basic calculations described in this part of ISO 12167 may be applied to bearings with different numbers of recesses and different width/diameter ratios for identical recess geometry.

Oil is fed to each bearing recess by means of a common pump with constant pumping pressure (system p_{en} = constant) and through preceding linear restrictors, e.g. capillaries.

The calculation procedures listed in this part of ISO 12167 enable the user to calculate and assess a given bearing design, as well as to design a bearing as a function of some optional parameters. Furthermore, this part of ISO 12167 contains the design of the required lubrication system including the calculation of the restrictor data.

Plain bearings — Hydrostatic plain journal bearings with drainage grooves under steady-state conditions —

Part 1: Calculation of oil-lubricated plain journal bearings with drainage grooves

1 Scope

This part of ISO 12167 applies to hydrostatic plain journal bearings under steady-state conditions.

In this part of ISO 12167, only bearings with oil drainage grooves between the recesses are taken into account. As compared to bearings without oil drainage grooves, this type needs higher power with the same stiffness behaviour.

2 Normative references

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

ISO 3448, *Industrial liquid lubricants — ISO viscosity classification*

ISO 12167-2:2001, *Plain bearings — Hydrostatic plain journal bearings with drainage grooves under steady-state conditions — Part 2: Characteristic values for the calculation of oil-lubricated plain journal bearings with drainage grooves*

3 Bases of calculation and boundary conditions

Calculation in accordance with this part of ISO 12167 is the mathematical determination of the operational parameters of hydrostatic plain journal bearings as a function of operating conditions, bearing geometry and lubrication data. This means the determination of eccentricities, load-carrying capacity, stiffness, required feed pressure, oil flow rate, frictional and pumping power, and temperature rise. Besides the hydrostatic pressure build up, the influence of hydrodynamic effects is also approximated.

Reynolds' differential formula furnishes the theoretical basis for the calculation of hydrostatic bearings. In most practical cases of application, it is, however, possible to arrive at sufficiently exact results by approximation.

The approximation used in this part of ISO 12167 is based on two basic formulae intended to describe the flow through the bearing lands, which can be derived from Reynolds' differential formula when special boundary conditions are observed. The Hagen-Poiseuille law describes the pressure flow in a parallel clearance gap and the Couette formula the drag flow in the bearing clearance gap caused by shaft rotation. A detailed presentation of the theoretical background of the calculation procedure is included in Annex A.

The following important premises are applicable to the calculation procedures described in this part of ISO 12167:

a) all lubricant flows in the lubrication clearance gap are laminar;

b) the lubricant adheres completely to the sliding surfaces;

c) the lubricant is an incompressible Newtonian fluid;

d) in the whole lubrication clearance gap, as well as in the preceding restrictors, the lubricant is partially isoviscous;

e) a lubrication clearance gap completely filled with lubricant is the basis of frictional behaviour;

f) fluctuations of pressure in the lubricant film normal to the sliding surfaces do not take place;

g) bearing and journal have completely rigid surfaces;

h) the radii of curvature of the surfaces in relative motion to each other are large in comparison to the lubricant film thickness;

i) the clearance gap height in the axial direction is constant (axial parallel clearance gap);

j) the pressure over the recess area is constant;

k) there is no motion normal to the sliding surfaces.

The bearing consists of Z cylindrical segments and rectangular recess of the same size and is supplied with oil through restrictors of the same flow characteristics. Each segment consists of a circumferential part between two centre lines of axial drainage grooves. With the aid of the above-mentioned approximation formulae, all parameters required for the design or calculation of bearings can be determined. The application of the similarity principle results in dimensionless similarity values for load-carrying capacity, stiffness, oil flow rate, friction, recess pressures, etc.

The results indicated in this part of ISO 12167 in the form of tables and diagrams are restricted to operating ranges common in practice for hydrostatic bearings. Thus, the range of the bearing eccentricity (displacement under load) is limited to $\varepsilon = 0$ to 0,5.

Limitation to this eccentricity range means a considerable simplification of the calculation procedure as the load-carrying capacity is a nearly linear function of the eccentricity. However, the applicability of this procedure is hardly restricted as in practice eccentricities $\varepsilon > 0,5$ are mostly undesirable for reasons of operational safety. A further assumption for the calculations is the approximated optimum restrictor ratio[1] $\xi = 1$ for the stiffness behaviour.

As for the outside dimensions of the bearing, this part of ISO 12167 is restricted to the range bearing width/bearing diameter $B/D = 0,3$ to 1, which is common in practical cases of application. The recess depth is larger than the clearance gap height by a factor of 10 to 100. When calculating the friction losses, the friction loss over the recess in relation to the friction over the bearing lands can generally be neglected on account of the above premises. However, this does not apply when the bearing shall be optimized with regard to its total power losses.

To take into account the load direction of a bearing, it is necessary to distinguish between the two extreme cases, load in the direction of recess centre and load in the direction of land centre.

Apart from the aforementioned boundary conditions, some other requirements are to be mentioned for the design of hydrostatic bearings in order to ensure their functioning under all operating conditions. In general, a bearing shall be designed in such a manner that a clearance gap height of at least 50 % to 60 % of the initial clearance gap height is ensured when the maximum possible load is applied. With this in mind, particular attention shall be paid to misalignments of the shaft in the bearing due to shaft deflection which may result in contact between shaft and bearing edge and thus in damage of the bearing. In addition, the parallel clearance gap required for the calculation is no longer present in such a case.

In the case where the shaft is in contact with the bearing lands when the hydrostatic pressure is switched off, it might be necessary to check the contact zones with regard to rising surface pressures.

It shall be ensured that the heat originating in the bearing does not lead to a non-permissible rise in the temperature of the oil.

If necessary, a means of cooling the oil shall be provided. Furthermore, the oil shall be filtered in order to avoid choking of the capillaries and damage to the sliding surfaces.

Low pressure in the relieved recess shall also be avoided, as this leads to air being drawn in from the environment and this would lead to a decrease in stiffness (see 5.7).

4 Symbols, terms and units

Table 1 — Symbols, terms and units

Symbol	Term	Unit
a	Inertia factor	1
A_{lan}	Land area	m^2
A^*_{lan}	Relative land area $\left(A^*_{lan} = \dfrac{A_{lan}}{\pi \times B \times D}\right)$	1
A_p	Recess area	m^2
b	Width perpendicular to the direction of flow	m
b_{ax}	Width of axial outlet $\left[b_{ax} = \dfrac{\pi \times D}{Z} - \left(l_c + b_G\right)\right]$	m
b_c	Width of circumferential outlet $\left(b_c = B - l_{ax}\right)$	m
b_G	Width of drainage groove	m
B	Bearing width	m
c	Stiffness coefficient	N/m
c_p	Specific heat capacity of the lubricant (p = *constant*)	J/kg·K
C_R	Radial clearance $\left[C_R = \left(D_B - D_J\right) / 2\right]$	m
d_{cp}	Diameter of capillaries	m
D	Bearing diameter (D_J: shaft; D_B: bearing; $D \approx D_J \approx D_B$)	m
e	Eccentricity (shaft displacement)	m
f	Relative film thickness [$f = h/C_R$]	1
$f_{en,i}$	Relative film thickness at $\varphi = \varphi'_{1,i}$	1
$f_{ex,i}$	Relative film thickness at $\varphi = \varphi'_{2,i}$	1
F	Load-carrying capacity (load)	N
F^*	Characteristic value of load-carrying capacity [$F^* = F/(B \times D \times p_{en})$]	1
F^*_{eff}	Characteristic value of effective load-carrying capacity	1
$F^*_{eff,0}$	Characteristic value of effective load-carrying capacity for $N = 0$	1
h	Local lubricant film thickness (clearance gap height)	m
h_{min}	Minimum lubricant film thickness (minimum clearance gap height)	m
h_p	Depth of recess	m
K_{rot}	Speed-dependent parameter	1
l	Length in the direction of flow	m
l_{ax}	Axial land length	m

Table 1 *(continued)*

Symbol	Term	Unit
l_c	Circumferential land length	m
l_{cp}	Length of capillaries	m
N	Rotational frequency (speed)	s^{-1}
p	Recess pressure, general	Pa
$\bar{p}$	Specific bearing load [$\bar{p} = F/(B \times D)$]	Pa
p_{en}	Feed pressure (pump pressure)	Pa
p_i	Pressure in recess i	Pa
P_i*	Pressure ratio [$P_i{}^* = P_i/P_{en}$]	1
$p_{i,\,0}$	Pressure in recess i, when $\varepsilon = 0$	Pa
P*	Power ratio ($P^* = P_f/P_p$)	1
P_f	Frictional power	W
P_p	Pumping power	W
P_{tot}	Total power ($P_{tot} = P_p + P_f$)	W
P_{tot}^*	Characteristic value of total power	1
Q	Lubricant flow rate (for complete bearing)	m^3/s
Q*	Lubricant flow rate parameter	1
$Q_{cp,\,i}$	Lubricant flow rate from capillary into recess 1	m^3/s
R_{cp}	Flow resistance of capillaries	$Pa{\cdot}s/m^3$
$R_{lan,\,ax}$	Flow resistance of one axial land $\left(R_{lan,ax} = \dfrac{12 \times \eta \times l_{ax}}{b_{ax} \times C_R^3}\right)$	$Pa{\cdot}s/m^3$
$R_{lan,\,c}$	Flow resistance of one circumferential land $\left(R_{lan,c} = \dfrac{12 \times \eta \times l_c}{b_c \times C_R^3}\right)$	$Pa{\cdot}s/m^3$
$R_{P,\,0}$	Flow resistance of one recess, when $\varepsilon = 0$, $\left(R_{p,0} = \dfrac{R_{lan,ax}}{2 \times (1 + \kappa)}\right)$	$Pa{\cdot}s/m^3$
Re	Reynolds number	1
So	Sommerfeld number	1
T	Temperature	°C
T_B	Mean temperature in the bearings; see Formula (15)	°C
ΔT	Temperature difference	°C
u	Flow velocity	m/s
U	Circumferential speed	m/s
$\bar{w}$	Average velocity in restrictor	m/s
Z	Number of recesses	1
α	Position of first recess related to recess centre measured from load direction; see Figure A.3	rad
β	Attitude angle of shaft	°
g	Exponent in viscosity formula	1
ε	Relative eccentricity ($\varepsilon = e/C_R$)	1

Table 1 *(continued)*

Symbol	Term	Unit
η	Dynamic viscosity	Pa·s
η_B	Dynamic viscosity for $T = T_B$	Pa·s
κ	Resistance ratio $\left(\kappa = \frac{R_{lan,\,ax}}{R_{lan,\,c}} = \frac{l_{ax} \times b_c}{l_c \times b_{ax}}\right)$	1
ξ	Restrictor ratio $\left(\xi = \frac{R_{cp}}{R_{P,\,0}}\right)$	1
π_f	Relative frictional pressure $\left(\pi_f = \frac{\eta_B \times \omega}{P_{en} \times \psi^2}\right)$	1
ρ	Density	kg/m^3
τ	Shearing stress	N/m^2
φ	Angular coordinate measured from radius opposite to eccentricity, e; see Figure A.3	rad
ψ	Relative bearing clearance $\left(\psi = \frac{2 \times C_R}{D}\right)$	1
ω	Angular velocity ($\omega = 2 \times \pi \times N$)	s^{-1}

5 Method of calculation

5.1 General

This part of ISO 12167 covers the calculation, as well as the design, of hydrostatic plain journal bearings. In this case, calculation is understood to be the verification of the operational parameters of a hydrostatic bearing with known geometrical and lubrication data. In the case of a design calculation, with the given methods of calculation, it is possible to determine the missing data for the required bearing geometry, the lubrication data and the operational parameters on the basis of a few initial data (e.g. required load-carrying capacity, stiffness, rotational frequency).

In both cases, the calculations are carried out according to an approximation method based on the Hagen-Poiseuille and the Couette formulae, mentioned in Clause 3. The bearing parameters calculated according to this method are given as relative values in the form of tables and diagrams as a function of different parameters. The procedure for the calculation or design of bearings is described in 5.2 to 5.7. This includes the determination of different bearing parameters with the aid of the given calculation formulae or the tables and diagrams. The following calculation items are explained in detail:

a) determination of load-carrying capacity with and without taking into account shaft rotation;

b) calculation of lubricant flow rate and pumping power;

c) determination of frictional power with and without consideration of losses in the bearing recesses;

d) procedure for bearing optimization with regard to minimum total power loss.

For all calculations, it is necessary to check whether the important premise of laminar flow in the bearing clearance gap, in the bearing recess and in the capillary is met. This is checked by determining the Reynolds numbers. Furthermore, the portion of the inertia factor in the pressure differences shall be kept low at the capillary (see A.3.1).

If the boundary conditions defined in Clause 3 are observed, this method of calculation yields results with deviations which can be neglected for the requirements of practice, in comparison with an exact calculation by solving the Reynolds differential formula.

5.2 Load-carrying capacity

Unless indicated otherwise, it is assumed in the following that capillaries with a linear characteristic are used as restrictors and that the restrictor ratio is $\xi = 1$. Furthermore, the difference is only made between the two cases, “load in direction of recess centre” and “load in direction of land centre”. For this reason, it is no longer mentioned in each individual case that the characteristic values are a function of the three parameters, “restrictor type”, “restrictor ratio” and “load direction relative to the bearing”.

Even under the abovementioned premises, the characteristic value of load carrying capacity [Formula (1)]

$$F^* = \frac{F}{B \times D \times p_{\text{en}}} = \frac{\bar{p}}{p_{\text{en}}} \qquad (1)$$

still depends on the following parameters:

— number of recesses, Z;

— width/diameter ratio, B/D;

— relative axial land width, l_{ax}/B;

— relative land width in circumferential direction, l_{c}/D;

— relative groove width, b_{G}/D;

— relative journal eccentricity, ε;

— relative frictional pressure when the difference is only made between the two cases, “load on recess centre” and “load on land centre”:

$$\pi_{\text{f}} = \frac{\eta_{\text{B}} \times \omega}{p_{\text{en}} \times \psi^2} \qquad (2)$$

NOTE The Sommerfeld number, So, common with hydrodynamic plain journal bearings can be set up as follows:

$$So = \frac{\bar{p} \times \psi^2}{\eta_{\text{B}} \times \omega} = \frac{F^*}{\pi_{\text{f}}}$$

In ISO 12167-2:2001, Figures 1 and 2, the functions $F^*(\varepsilon, \pi_{\text{f}})$ and $\beta(\varepsilon, \pi_{\text{f}})$ are represented for $Z = 4$, $\xi = 1$, $B/D = 1$, $l_{\text{ax}}/B = 0{,}1$, $l_{\text{c}}/D = 0{,}1$, $b_{\text{G}}/D = 0{,}05$, i.e. restriction by means of capillaries, load in direction of centre of bearing recess.

These figures show the influence of rotation on the characteristic value of load-carrying capacity and the attitude angle.

For the calculation of a geometrically similar bearing, it is possible to determine the minimum lubricant film thickness when values are given, e.g. for F, B, D, p_{en}, ω, ψ, and η_{B} (determination of η_{B} according to 5.6, if applicable).

All parameters are given for the determination of F^* according to Formula (1) and π_{f} according to Formula (2). For this geometry, the relevant values for ε and β can be taken from ISO 12167-2:2001, Figures 1 and 2 and thus, $h_{\text{min}} = C_{\text{R}}(1 - \varepsilon)$.

According to the approximation method described in Annex A, it transpires that the characteristic value of effective load-carrying capacity is no longer a function of the ratio B/D.

$$F_{\text{eff}}^{*} = \frac{F}{b_{\text{c}} \times Z \times b_{\text{ax}} \times P_{\text{en}}} = \frac{F^{*}}{\dfrac{b_{\text{c}}}{D} \times \dfrac{Z \times b_{\text{ax}}}{\pi \times B}} \quad (3)$$

If the resistance ratio

$$\kappa = \frac{R_{\text{lan, ax}}}{R_{\text{lan, c}}} = \frac{l_{\text{ax}} \times b_{\text{c}}}{l_{\text{c}} \times b_{\text{ax}}} \quad (4)$$

and the speed dependent parameter

$$K_{\text{rot}} = \frac{\xi \times \kappa \times \pi_{\text{f}} \times l_{\text{c}}}{D}$$
$$K_{\text{rot,nom}} = \frac{K_{\text{rot}}}{1+\kappa} \quad (5)$$

is introduced, there remains a dependence on the following parameters:

$$F_{\text{eff}}^{*}\left(Z, \varphi_{\text{G}}, \kappa, K_{\text{rot}}, \varepsilon\right)$$

If, in addition, advantage is taken of the fact that the function $F_{\text{eff}}^{*}(\varepsilon)$ is nearly linear for $\varepsilon \leq 0{,}5$, then it is practically sufficient to know that the function $F_{\text{eff}}^{*}\left(\varepsilon = 0{,}4\right) = f\left(Z, \varphi_{\text{G}}, \kappa, K_{\text{rot}}\right)$ for the calculation of the load carrying capacity.

For $K_{\text{rot}} = 0$, i.e. for the stationary shaft, the characteristic value of effective load-carrying capacity for $\varepsilon = 0{,}4$ only depends on three parameters:

$$F_{\text{eff}}^{*}\left(\varepsilon = 0{,}4\right) = f\left(Z, \varphi_{\text{G}}, \kappa\right)$$

Thus, in ISO 12167-2:2001, Figure 3, $F_{\text{eff},0}^{*}\left(\varepsilon = 0{,}4\right)$ for $Z = 4$ and 6 can be given via κ for different φ_{G} values.

The influence of the rotational movement on the characteristic value of load-carrying capacity is taken into account by the ratio $\dfrac{F_{\text{eff}}^{*}}{F_{\text{eff},0}^{*}} = f\left(Z, \varphi_{\text{G}}, \kappa, K_{\text{rot}}\right)$.

For $Z = 4$, the ratio $F_{\text{eff}}^{*} / F_{\text{eff},0}^{*}$ is shown in ISO 12167-2:2001, Figure 4. The hydrodynamically conditioned increase of the load-carrying capacity can be easily recognized when presented in such a manner.

If, e.g. Z and all parameters are given for the determination of F_{eff}^{*} according to Formula (3), κ according to Formula (4) and K_{rot} according to Formula (5), then the minimum lubricant film thickness developing during operation can be determined.