INTERNATIONAL STANDARD

ISO 12168-1

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Plain bearings — Hydrostatic plain journal bearings without drainage grooves under steady-state conditions —

Part 1: Calculation of oil-lubricated plain journal bearings without drainage grooves

Paliers lisses — Paliers lisses radiaux hydrostatiques sans rainure d'écoulement fonctionnant en régime stationnaire —

Partie 1: Calcul pour la lubrification des paliers lisses radiaux sans rainure d'écoulement

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Foreword

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The main task of technical committees is to prepare International Standards. Draft International Standards adopted by the technical committees are circulated to the member bodies for voting. Publication as an International Standard requires approval by at least 75 % of the member bodies casting a vote.

Attention is drawn to the possibility that some of the elements of this part of ISO 12168 may be the subject of patent rights. ISO shall not be held responsible for identifying any or all such patent rights.

ISO 12168-1 was prepared by Technical Committee ISO/TC 123, *Plain bearings*, Subcommittee SC 4, *Methods of calculation of plain bearings*.

ISO 12168 consists of the following parts, under the general title *Plain bearings — Hydrostatic plain journal bearings without drainage grooves under steady-state conditions*:

— *Part 1: Calculation of oil-lubricated plain journal bearings without drainage grooves*

— *Part 2: Characteristic values for the calculation of oil-lubricated plain journal bearings without drainage grooves*

Annexes A and B form a normative part of this part of ISO 12168.

Introduction

The functioning of hydrostatic bearings is characterized by the fact that the supporting pressure of the bearing is generated by external lubrication. The special advantages of hydrostatic bearings are lack of wear, quiet running, wide useable speed range as well as high stiffness and damping capacity. These properties are also the reason for the special importance of hydrostatic bearing units in different fields of application such as e.g. machine tools.

The bases of calculation described in this part of ISO 12168 apply to bearings with different numbers of recesses and different width/diameter ratios for identical recess geometry. In this part of ISO 12168 only bearings without oil drainage grooves between the recesses are taken into account. As compared to bearings with oil drainage grooves, this type needs less power with the same stiffness behaviour.

The oil is fed to each bearing recess by means of a common pump with constant pump pressure (system p_{en} = constant) and via preceding linear restrictors (e.g. in the form of capillaries).

The calculation procedures listed in this part of ISO 12168 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 12168 contains the design of the required lubrication system including the calculation of the restrictor data.

Plain bearings — Hydrostatic plain journal bearings without drainage grooves under steady-state conditions —

Part 1: Calculation of oil-lubricated plain journal bearings without drainage grooves

1 Scope

This part of ISO 12168 applies to hydrostatic plain journal bearings under steady-state conditions.

In this part of ISO 12168 only bearings without oil drainage grooves between the recesses are taken into account.

2 Normative references

The following normative documents contain provisions which, through reference in this text, constitute provisions of this part of ISO 12168. For dated references, subsequent amendments to, or revisions of, any of these publications do not apply. However, parties to agreements based on this part of ISO 12168 are encouraged to investigate the possibility of applying the most recent editions of the normative documents indicated below. For undated references, the latest edition of the normative document referred to applies. Members of ISO and IEC maintain registers of currently valid International Standards.

ISO 3448:1992, *Industrial liquid lubricants — ISO viscosity classification*

ISO 12168-2:2001, *Plain bearings — Hydrostatic plain journal bearings without drainage grooves under steady-state conditions — Part 2: Characteristic values for the calculation of oil-lubricated plain journal bearings without drainage grooves*

3 Bases of calculation and boundary conditions

Calculation within the meaning of this part of ISO 12168 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 equation furnishes the theoretical bases 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 12168 is based on two basic equations for describing the flow via the bearing lands, which can be derived from Reynolds' differential equation when special boundary conditions are observed. The Hagen-Poiseuille law describes the pressure flow in a parallel clearance gap and the Couette equation 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 apply to the calculation procedures described in this part of ISO 12168:

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 for the frictional behaviour;

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

g) half 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.

With the aid of the above-mentioned approximation equations, 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 12168 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 12168 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 the factor 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, difference is made between the two extreme cases, load in the direction of recess centre and load in the direction of land centre.

Apart from the afore-mentioned 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 assured 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.

As 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 assured 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

See Table 1.

Table 1 — Symbols, terms and units

Symbol	Term	Unit
a	Inertia factor	1
A_{lan}	Land area	m^2
A^*_{lan}	Relative land area $\left(A^*_{\text{lan}} = \dfrac{A_{\text{lan}}}{\pi \times B \times D}\right)$	1
A_{p}	Recess area	m^2
b	Width perpendicular to the direction of flow	m
b_{ax}	Width of axial outlet $\left[b_{\text{ax}} = \dfrac{\pi \times D}{Z}\right]$	m
b_{c}	Width of circumferential outlet $\left(b_{\text{c}} = B - l_{\text{ax}}\right)$	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 $\left[C_{\text{R}} = \left(D_{\text{B}} - D_{\text{J}}\right)/2\right]$	m
d_{cp}	Diameter of capillaries	m
D	Bearing diameter (D_{J}: shaft; D_{B}: bearing; $D \approx D_{\text{J}} \approx D_{\text{B}}$)	m
e	Eccentricity (shaft displacement)	m
F	Load-carrying capacity (load)	N
F^*	Characteristic value of load-carrying capacity [$F^* = F/(B \times D \times p_{\text{en}})$]	1
F^*_{eff}	Characteristic value of effective load-carrying capacity	1
$F^*_{\text{eff,0}}$	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

Table 1 — (*continued*)

Symbol	Term	Unit
l_{ax}	Axial land length	m
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 $\left[\bar{p} = F/(B \times D)\right]$	Pa
p_{en}	Feed pressure (pump pressure)	Pa
p_i	Pressure in recess i	Pa
$p_{i,0}$	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
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 $\varepsilon = 0$, $\left(R_{P,0} = 0{,}5 R_{lan,ax}\right)$	$Pa \cdot s/m^3$
Re	Reynolds number	1
So	Sommerfeld number	1
T	Temperature	°C
ΔT	Temperature difference	K
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 1st recess related to recess centre	rad
β	Attitude angle of shaft	°
γ	Exponent in viscosity formula	1
ε	Relative eccentricity ($\varepsilon = e/C_R$)	1

Table 1 — (*continued*)

Symbol	Term	Unit
η	Dynamic viscosity	Pa·s
κ	Resistance ratio $\left(\kappa = \frac{R_{\text{lan,ax}}}{R_{\text{lan,c}}} = \frac{l_{\text{ax}} \times b_{\text{c}}}{l_{\text{c}} \times b_{\text{ax}}}\right)$	1
ξ	Restrictor ratio $\left(\xi = \frac{R_{\text{cp}}}{R_{\text{P,0}}}\right)$	1
π_{f}	Relative frictional pressure $\left(\pi_{\text{f}} = \frac{\eta_{\text{B}} \times \omega}{p_{\text{en}} \times \psi^2}\right)$	1
ρ	Density	kg/m^3
τ	Shearing stress	N/m^2
φ	Angular coordinate	rad
ψ	Relative bearing clearence $\left(\psi = \frac{2 \times C_{\text{R}}}{D}\right)$	1
ω	Angular velocity $(\omega = 2 \times \pi \times N)$	s^{-1}

5 Method of calculation

5.1 General

This part of ISO 12168 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 equations, 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 consideration of 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 shall be checked in addition 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.3.2.2).

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 equation.

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, 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 above mentioned premises, the characteristic value of load carrying capacity

$$F^* = \frac{F}{B \times D \times p_{\mathrm{en}}} = \frac{\bar{p}}{p_{\mathrm{en}}} \tag{1}$$

still depends on the following parameters:

— the number of recesses Z;

— the width/diameter ratio B/D;

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

— the relative land width in circumferential direction l_{c}/B;

— the relative journal eccentricity ε;

the relative frictional pressure $\pi_{\mathrm{f}} = \dfrac{\eta_{\mathrm{B}} \times \omega}{p_{\mathrm{en}} \times \psi^2}$ (2)

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

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

In Figures 1 and 2 of ISO 12168-2:2001, the functions $F^*(\varepsilon, \pi_{\mathrm{f}})$ and $\beta\ (\varepsilon, \pi_{\mathrm{f}})$ are represented for $Z = 4$, $\xi = 1$, $B/D = 1$, $l_{\mathrm{ax}}/B = 0{,}16$, $l_{\mathrm{c}}/B = 0{,}26$, i.e. restriction by means of capillaries, load in direction of centre of bearing recess.

These figures represent a comparison between the approximation and the more precise solution by means of Reynolds equation. Further, the influence of rotation on the characteristic value of the load-carrying capacity and on the attitude angle can be realized.

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 equation (1) and π_{f} according to equation (2). For this geometry, the relevant values for ε and β can be taken from Figures 1 and 2 in ISO 12168-2:2001 and thus $h_{\mathrm{min}} = C_{\mathrm{R}}(1 - \varepsilon)$.

According to the approximation method described in annex A, this results in a dependence of the characteristic value of effective load-carrying capacity formed with the so-called “effective bearing width” $B - l_{\mathrm{ax}}$