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Plain bearings — Hydrodynamic plain thrust pad bearings under steady-state conditions —

Part 1: Calculation of thrust pad bearings

Paliers lisses — Butées hydrodynamiques à patins géométrie fixe fonctionnant en régime stationnaire —

Partie 1: Calcul des butées à segments

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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).

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*.

This second edition cancels and replaces the first edition (ISO 12131-1:2001), which has been technically revised.

The main changes compared to the previous edition are the correction of typographical errors.

A list of all parts in the ISO 12131 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.

Plain bearings — Hydrodynamic plain thrust pad bearings under steady-state conditions —

Part 1: Calculation of thrust pad bearings

1 Scope

The aim of this document is to achieve designs of plain bearings that are reliable in operation, by the application of a calculation method for oil-lubricated hydrodynamic plain bearings with complete separation of the thrust collar and plain bearing surfaces by a film of lubricant[1].

This document applies to plain thrust bearings with incorporated wedge and supporting surfaces having any ratio of wedge surface length l_{wed} to length of one pad L. It deals with the value $l_{wed}/L = 0{,}75$ as this value represents the optimum ratio[2]. The ratio of width to length of one pad can be varied in the range $B/L = 0{,}5$ to 2.

The calculation method described in this document can be used for other incorporated gap shapes, e.g. plain thrust bearings with integrated baffle, when for these types the numerical solutions of Reynolds equation are known.

The calculation method serves for designing and optimizing plain thrust bearings e.g. for fans, gear units, pumps, turbines, electrical machines, compressors and machine tools. It is limited to steady-state conditions, i.e. load and angular speed of all rotating parts are constant under continuous operating conditions. Dynamic operating conditions are not included.

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 3448, *Industrial liquid lubricants — ISO viscosity classification*

ISO 12131-2:2016, *Plain bearings — Hydrodynamic plain thrust pad bearings under steady-state conditions — Part 2: Functions for the calculation of thrust pad bearings*

ISO 12131-3, *Plain bearings — Hydrodynamic plain thrust pad bearings under steady-state conditions — Part 3: Guide values for the calculation of thrust pad bearings*

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

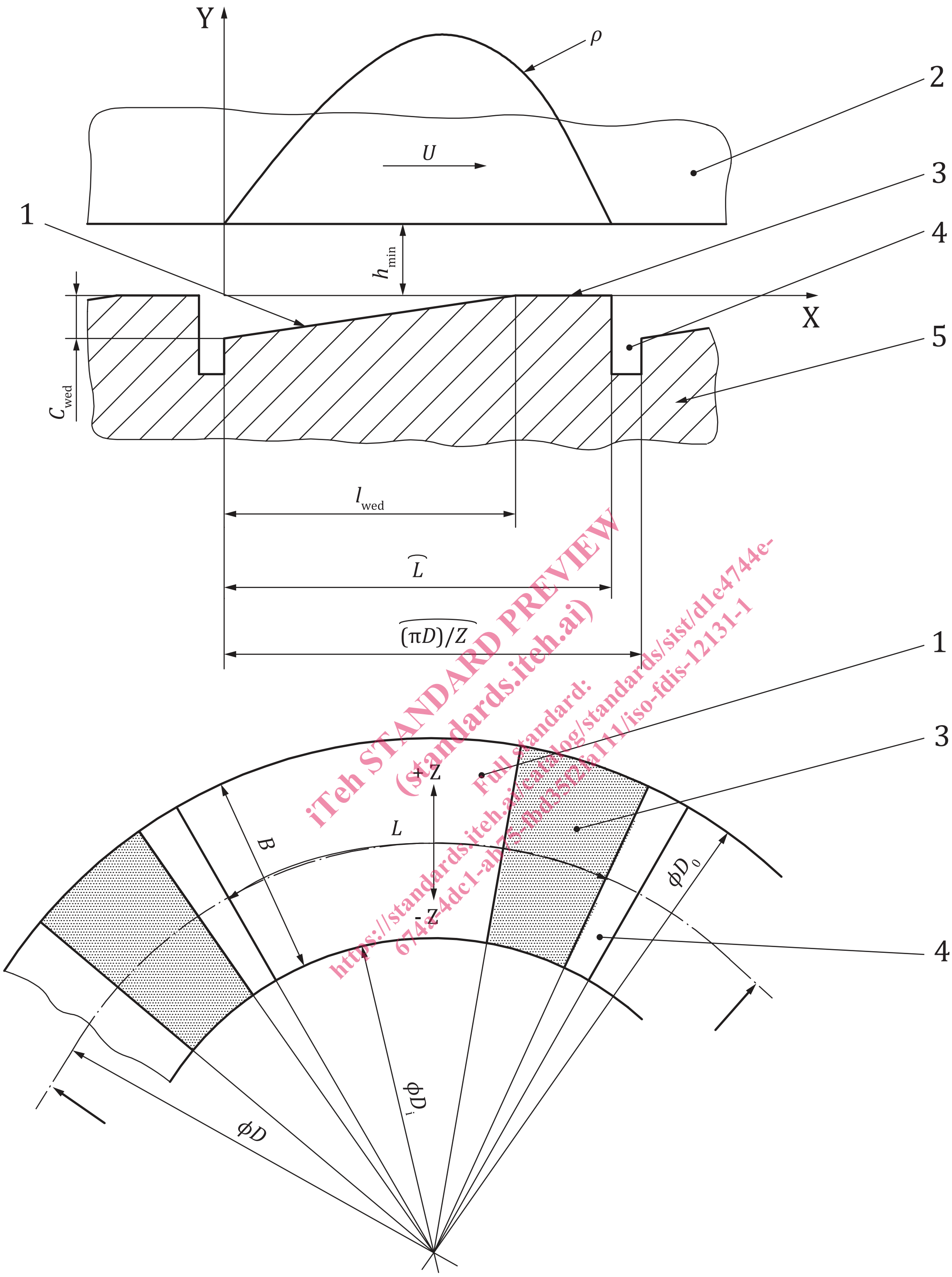
See Table 1 and Figure 1.

Table 1 — Symbols and units

Symbol	Designation	Unit
A	Heat emitting surface of the bearing housing	m^2
B	Width of one pad	m
B_H	Axial housing width	m
Cp	Specific heat capacity of the lubricant (p = constant)	J/(kg·K)
C_{wed}	Wedge depth	m
D	Mean sliding diameter (diameter of thrust bearing ring)	m
D_H	Housing outside diameter	m
D_i	Inside diameter of thrust bearing ring	m
D_o	Outside diameter of thrust bearing ring	m
f^*	Characteristic value of friction	1
$f_B{}^*$	Characteristic value of friction for thrust pad bearing	1
F	Bearing force (nominal load)	N
F^*	Characteristic value of load carrying capacity	1
$F_B{}^*$	Characteristic value of load carrying capacity for thrust pad bearing	1
F_{st}	Bearing force (load) under stationary conditions	N
h	Local lubricant film thickness (clearance gap height)	m
h_{lim}	Minimum permissible lubricant film thickness during operation	m
$h_{lim,\,tr}$	Minimum permissible lubricant film thickness in the transition into mixed lubrication	m
h_{min}	Minimum lubricant film thickness (minimum clearance gap height)	m
k	Heat transfer coefficient related to the product $B \times L \times Z$	$W/(m^2{\cdot}K)$
k_A	External heat transfer coefficient (reference surface A)	$W/(m^2{\cdot}K)$
l_{wed}	Wedge length	m
L	Length of one pad in circumferential direction	m
M	Mixing factor	1
N	Rotational frequency (speed) of thrust collar	s^{-1}
p	Local lubricant film pressure	Pa
$\overline{p}$	Specific bearing load $\overline{p} = F/(B \times L \times Z)$	Pa
P_f	Frictional power in the bearing or heat flow rate generated by it	W
$\overline{p}_{lim}$	Maximum permissible specific bearing load	Pa
$P_{th,\,amb}$	Heat flow rate to the environment	W
$P_{th,\,f}$	Heat flow rate arising from the frictional power	W
$P_{th,\,L}$	Heat flow rate in the lubricant	W
Q	Lubricant flow rate	m^3/s
Q^*	Characteristic value of lubricant flow rate	1
Q_0	Relative lubricant flow rate $Q_0 = B \times h_{min} \times U \times Z$	m^3/s
Q_1	Lubricant flow rate at the inlet of the clearance gap (circumferential direction)	m^3/s
Q^*_1	Characteristic value of lubricant flow rate at the inlet of the clearance gap	1
Q_2	Lubricant flow rate at the outlet of the clearance gap (circumferential direction)	m^3/s
Q^*_2	Characteristic value of lubricant flow rate $Q^*_1 - Q^*_3$ at the outlet of the clearance gap	1

Table 1 *(continued)*

Symbol	Designation	Unit
Q_3	Lubricant flow rate at the sides (perpendicular to circumferential direction)	m^3/s
Q^*_3	Characteristic value of lubricant flow rate at the sides	1
Re	Reynolds number	1
Re_{cr}	Critical Reynolds' number	1
T_{amb}	Ambient temperature	°C
T_B	Bearing temperature	°C
T_{eff}	Effective lubricant film temperature	°C
T_{en}	Lubricant temperature at the inlet of the bearing	°C
T_{ex}	Lubricant temperature at the outlet of the bearing	°C
T_{lim}	Maximum permissible bearing temperature	°C
T_1	Lubricant temperature at the inlet of the clearance gap	°C
T_2	Lubricant temperature at the outlet of the clearance gap	°C
U	Sliding velocity relative to mean diameter of bearing ring	m/s
w_{amb}	Velocity of air surrounding the bearing housing	m/s
x	Coordinate in direction of motion (circumferential direction)	m
y	Coordinate in direction of lubrication clearance gap (axial)	m
z	Coordinate perpendicular to the direction of motion (radial)	m
Z	Number of pads	1
η	Dynamic viscosity of the lubricant	Pa·s
η_{eff}	Effective dynamic viscosity of the lubricant	Pa·s
ρ	Density of the lubricant	kg/m^3



Key

1 wedge surface

2 thrust collar

3 supporting surface

4 lubrication groove

5 thrust bearing ring

Figure 1 — Schematic view of a thrust pad bearing (bearing with incorporated wedge and supporting surfaces)

5 Fundamentals, assumptions and premises

The calculation is always carried out with the numerical solutions of Reynolds equation for sliding surfaces with finite width, taking into account the physically correct boundary conditions for the generation of pressure.

$$\frac{\partial}{\partial x}\left(h^3 \frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial z}\left(h^3 \frac{\partial p}{\partial z}\right)=6\times\eta\times U\times\frac{\partial h}{\partial x} \tag{1}$$

See [1] for the derivation of Reynolds equation and [2] for the numerical solution.

For the solution of Formula (1), the following idealizing assumptions and premises are used, the reliability of which has been sufficiently confirmed by experiment and in practice[3].

a) The lubricant corresponds to a Newtonian fluid.

b) All lubricant flows are laminar.

c) The lubricant adheres completely to the sliding surfaces.

d) The lubricant is incompressible.

e) The lubrication clearance gap is completely filled with lubricant.

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

g) The components forming the lubrication clearance gap are rigid or their deformation is negligible; their surfaces are completely even.

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

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

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

k) The lubricant is isoviscous over the entire lubrication clearance gap.

l) The lubricant is fed in at the widest lubrication clearance gap; the magnitude of the lubricant feed pressure is negligible as compared to the lubricant film pressures themselves.

m) The pad shape of the sliding surfaces is replaced by rectangles.

The boundary conditions for the solution of Reynolds equation are the following.

1) The gauge pressure of the lubricant at the feeding point is $p\ (x = 0, z\ ) = 0$.

2) The feeding of the lubricant is arranged in such a way that it does not interfere with the generation of pressure in the lubrication clearance gap.

3) The gauge pressure of the lubricant at the lateral edges of the plain bearing is $p\ (x, z = 0{,}5\ B) = 0$.

4) The gauge pressure of the lubricant is $p\ (x = L, z) = 0$ at the end of the pressure field.

The application of the principle of similarity in hydrodynamic plain bearing theory results in dimensionless parameters of similarity for such characteristics as load carrying capacity, friction behaviour and lubricant flow rate.

The use of parameters of similarity reduces the number of necessary numerical solutions of Reynolds equation which are compiled in ISO 12131-2. In principle, other solutions are also permitted provided they satisfy the conditions given in this document and have the corresponding numerical accuracy.

ISO 12131-3, contains guide values according to which the calculation result is to be oriented in order to ensure the functioning of the plain bearings.

In special cases, guide values deviating from ISO 12131-3, may be agreed for specific applications.

6 Calculation procedure

6.1 Loading operations

6.1.1 General

Calculation means the mathematical determination of the correct functioning using operational parameters (see Figure 2) which can be compared with guide values. Thereby, the operational parameters determined under varying operation conditions shall be permissible as compared to the guide values. For this purpose, all continuous operating conditions shall be investigated.

6.1.2 Wear

Safety against wear is given if complete separation of the mating bearing parts is achieved by the lubricant. Continuous operation in the mixed lubrication range results in premature loss of functioning. Short-time operation in the mixed lubrication range such as starting up and running down machines with plain bearings, is unavoidable and can result in bearing damage after frequent occurrence. When subjected to heavy load, an auxiliary hydrostatic arrangement may be necessary for starting up or running down at a low speed. Running-in and adaptive wear to compensate for surface geometry deviations from the ideal geometry are permissible as long as these are limited in time and locality and occur without overload effects. In certain cases, a specific running-in procedure may be beneficial. This can also be influenced by the selection of the material. Attention is drawn to the fact that in the case of this bearing design, wear can lead to a rapid decrease in the load carrying capacity.

6.1.3 Mechanical loading

The limits of mechanical loading are given by the strength of the bearing material. Slight permanent deformation is permissible as long as it does not impair correct functioning of the plain bearing.

6.1.4 Thermal loading

The limits of thermal loading result not only from the thermal stability of the bearing material but also from the viscosity-temperature relationship and the ageing tendency of the lubricant.

6.1.5 Outside influences

Calculation of correct functioning of plain bearings presupposes that the operating conditions are known for all cases of continuous operation. In practice, however, additional disturbing influences frequently occur which are unknown at the design stage and cannot always be computed. Therefore, the application of an appropriate safety margin between the operational parameters and the permissible guide values is recommended. Disturbing influences are, e.g.:

— spurious forces (out-of-balance, vibrations, etc.);

— deviations from the ideal geometry (machining tolerances, deviations during assembly, etc.);

— lubricants contaminated by solid, liquid and gaseous foreign matters;

— corrosion, electric erosion, etc.

Information as to further influence factors is given in 6.8.

The applicability of this document for which laminar flow in the lubrication clearance gap is a necessary condition, is to be checked by the Reynolds' number:

$$Re = \frac{\rho \times U \times h_{min}}{\eta_{eff}} \leq Re_{cr} \tag{2}$$

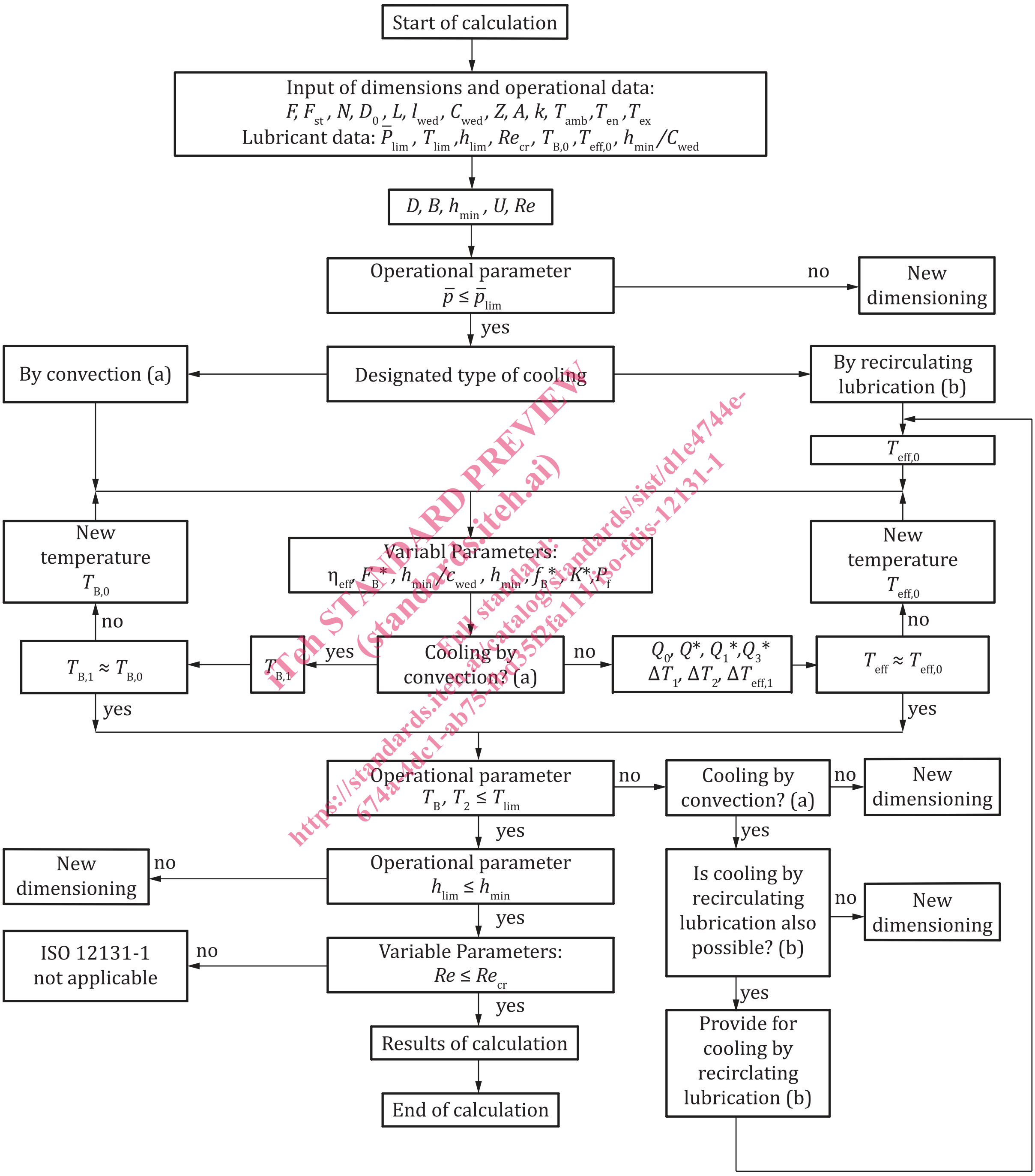


Figure 2 — Scheme of calculation (flow chart)

For wedge-shaped gaps with h_{min}/C_{wed} = 0,8 a critical Reynolds' number of Re_{cr} = 600 can be assumed as guide value according to [4].