INTERNATIONAL STANDARD

ISO 12130-1

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

Part 1: **Calculation of tilting pad thrust bearings**

Paliers lisses — Butées hydrodynamiques à patins oscillants fonctionnant en régime stationnaire —

Partie 1: Calcul des butées à patins oscillants

Reference number
ISO 12130-1:2001(E)

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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.

International Standards are drafted in accordance with the rules given in the ISO/IEC Directives, Part 3.

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 12130 may be the subject of patent rights. ISO shall not be held responsible for identifying any or all such patent rights.

International Standard ISO 12130-1 was prepared by Technical Committee ISO/TC 123, *Plain bearings*, Subcommittee SC 4, *Methods of calculation of plain bearings*.

ISO 12130 consists of the following parts, under the general title *Plain bearings — Hydrodynamic plain tilting pad thrust bearings under steady-state conditions*:

— *Part 1: Calculation of tilting pad thrust bearings*

— *Part 2: Functions for calculation of tilting pad thrust bearings*

— *Part 3: Guide values for the calculation of tilting pad thrust bearings*

Annex A forms a normative part of this part of ISO 12130.

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

Part 1: Calculation of tilting pad thrust bearings

1 Scope

The aim of ISO 12130 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.

This part of ISO 12130 applies to plain thrust bearings with tilting-type sliding blocks (tilting pads), where a wedge-shaped lubrication clearance gap is automatically formed during operation. 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 part of ISO 12130 can be used for other gap shapes, e.g. parabolic lubrication clearance gaps, as well as for other types of sliding blocks, e.g. circular sliding blocks, when for these types the numerical solutions of Reynolds' differential equation are present. ISO 12130-2 gives only the characteristic values for the plane wedge-shaped gap; the values are therefore not applicable to tilting pads with axial support.

The calculation method serves for designing and optimizing plain thrust bearings e.g. for fans, gear units, pumps, turbines, electric 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.

This part of ISO 12130 is not applicable to heavily loaded tilting pad thrust bearings.

2 Normative references

The following normative documents contain provisions which, through reference in this text, constitute provisions of this part of ISO 12130. 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 12130 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 12130-2, *Plain bearings — Hydrodynamic plain tilting pad thrust bearings under steady-state conditions — Part 2: Functions for calculation of tilting pad thrust bearings*

ISO 12130-3, *Plain bearings — Hydrodynamic plain tilting pad thrust bearings under steady-state conditions — Part 3: Guide values for the calculation of tilting pad thrust bearings*

3 Fundamentals, assumptions and premises

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

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

Reference is made, e.g., to [1] for the derivation of Reynolds' differential equation and to [2] for the numerical solution.

For the solution to equation (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, 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;

m) the magnitude of the lubricant feed pressure is negligible as compared to the lubricant film pressures themselves;

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

The boundary conditions for the solution of Reynolds' differential 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 $p\left(x, z = \pm 0{,}5B\right) = 0$

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

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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' differential equation which are compiled in ISO 12130-2. In principle, other solutions are also permitted if they satisfy the conditions given in this part of ISO 12130 and have the corresponding numerical accuracy.

ISO 12130-3, 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 12130-3, may be agreed for specific applications.

4 Symbols, terms and units

See Table 1 and Figure 1.

Table 1 — Symbols, terms and units

Symbol	Term	Unit
a_F	Distance between supporting point and inlet of the clearance gap in the direction of motion (circumferential direction)	m
a^*_F	Relative distance between supporting point and inlet of the clearance gap in the direction of motion (circumferential direction)	1
A	Heat-emitting surface of the bearing housing	m^2
B	Width of one pad	m
c_p	Specific heat capacity of the lubricant (p = constant)	J/(kg·K)
C_{wed}	Wedge depth	m
D	Mean sliding diameter	m
D_i	Inside diameter over tilting pads	m
D_o	Outside diameter over tilting pads	m
f^*	Characteristic value of friction	1
F	Bearing force (load) at nominal rotational frequency	N
F^*	Characteristic value of load carrying capacity	1
F_{st}	Bearing force (load) at standstill	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 on 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	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

Table 1 (*continued*)

Symbol	Term	Unit
$\overline{p}$	Specific bearing load $\overline{p} = F/(B \times L \times Z)$	Pa
$\overline{p}_{\text{lim}}$	Maximum permissible specific bearing load	Pa
P_{f}	Frictional power in the bearing or power generated heat flow rate	W
$P_{\text{th,amb}}$	Heat flow rate to the environment	W
$P_{\text{th,L}}$	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 $Q_0 = B \times h_{\text{min}} \times U \times Z$	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
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
Rz	Average peak-to-valley height of thrust collar	µm
Re	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 tilting-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

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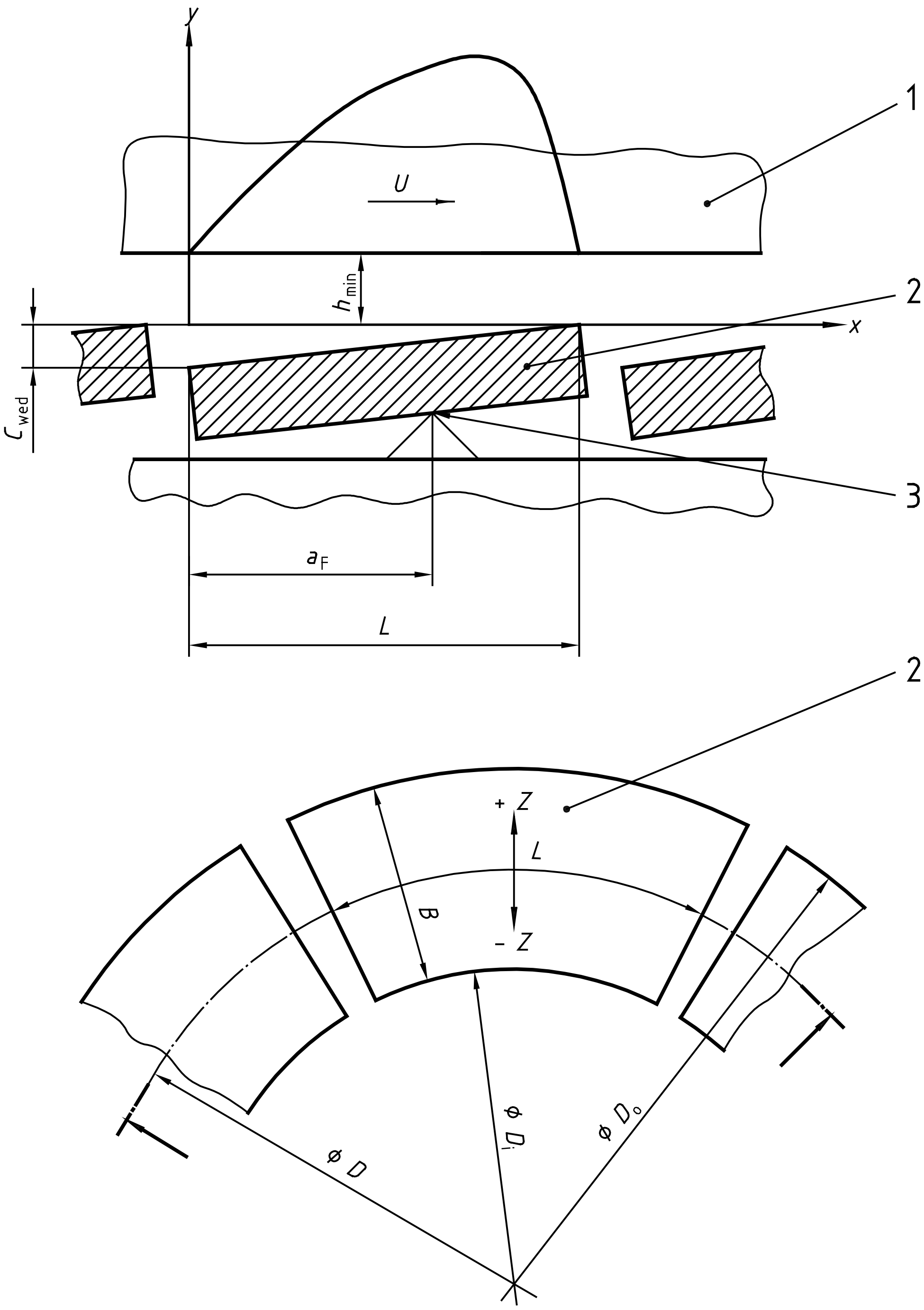

Key

1 Thrust collar
2 Tilting-pad
3 Centre of pressure (supporting surface)

Figure 1 — Schematic view of a tilting-pad thrust bearing

5 Calculation procedure

5.1 Loading operations

5.1.1 General

Calculation means the mathematical determination of the correct functioning using operational parameters (see Figure 2) which has to 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.

5.1.2 Wear

Safety against wear is assured if complete separation of the mating bearing parts is achieved by the lubricant. Continuous operation in the mixed lubrication range results in early 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 occurence. When subjected to heavy loads, an auxiliary hydrostatic arrangement may be necessary for starting up or running down at low speed. Running-in and adaptive wear to compensate for surface geometry deviations from ideal geometry are permissible as long as these are limited in time and locality and occur without overloading effects. In certain cases, a specific running-in procedure may be beneficial. This can also be influenced by the selection of the material.

5.1.3 Mechanical loading

The limits of mechanical loading are given by the strength of the bearing material. Slight permanent deformations are permissible as long as these do not impair correct functioning of the plain bearing.

5.1.4 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.

5.1.5 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 ideal geometry (machining tolerances, deviations during assembly, etc.);
- lubricants contaminated by solid, liquid and gaseous foreign materials;
- corrosion, electric erosion, etc.

Information as to further influence factors is given in 5.9.

The applicability of this part of ISO 12130, for which laminar flow in the lubrication clearance gap is a necessary condition, shall be checked using the Reynolds' number:

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

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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].

The plain bearing calculation comprises, starting from the known bearing dimensions and operating data:

— the relationship between load-carrying capacity and lubricant film thickness;

— the frictional power;

— the lubricant flow rate;

— the heat balance;

these all being interdependent. The solution is obtained using an iterative method, the sequence of which is summarized in the calculation flow chart in Figure 2.

For optimization of individual parameters, parameter variation can be performed; and modification of the calculation sequence is possible.

5.2 Coordinate of centre of pressure

In the case of tilting pads, the x-coordinate of the centre of pressure a_F corresponds with the x-coordinate of the axis of tilt. The x-coordinate of the centre of pressure $a^*_F = a_F / L$ related to the length of the sliding block is a function of the relative minimum lubricant film thickness h_{min}/C_{wed} and the relative width of sliding block B/L. ISO 12130-2 represents $a^*_F = f\,(h_{min} / C_{wed};\ \ B / L)$. An approximate function is also given there.

It is essential for the calculation that the relative minimum lubricant film thickness h_{min}/C_{wed} as well as the characteristic values of load-carrying capacity, frictional power and lubricant flow rate are specified by the selection of the supporting point a_F and that these values remain unchanged even under alternating operating conditions.

5.3 Load-carrying capacity

The parameter for the load-carrying capacity is the dimensionless characteristic value of load-carrying capacity F^*:

$$F^* = \frac{F \times h_{min}^2}{U \times \eta_{eff} \times L^2 \times B \times Z} \qquad (3)$$

The function $F^* = f(h_{min}/C_{wed};\ B/L)$ is presented in ISO 12130-2 on the basis of [5]. An approximate function is also given there.

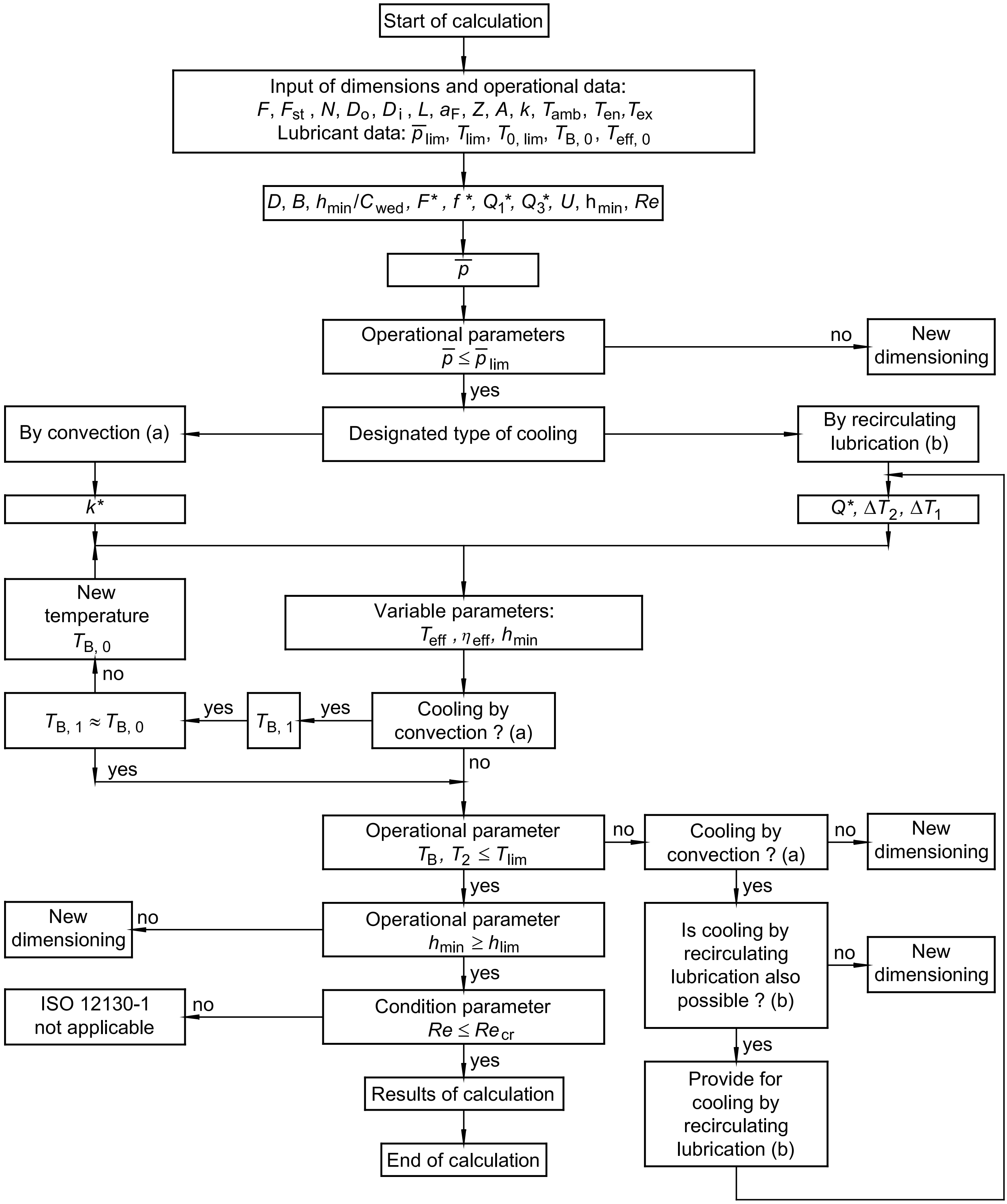


Figure 2 — Scheme of calculation (flow chart)

5.4 Frictional power

The losses due to friction in a hydrodynamic plain thrust bearing are given by the characteristic value of friction f^* which is defined as follows:

$$f^* = P_{\text{f}} \times \frac{h_{\text{min}}}{U^2 \times \eta_{\text{eff}} \times B \times L \times Z} \tag{4}$$

Thus the frictional power is calculated as follows:

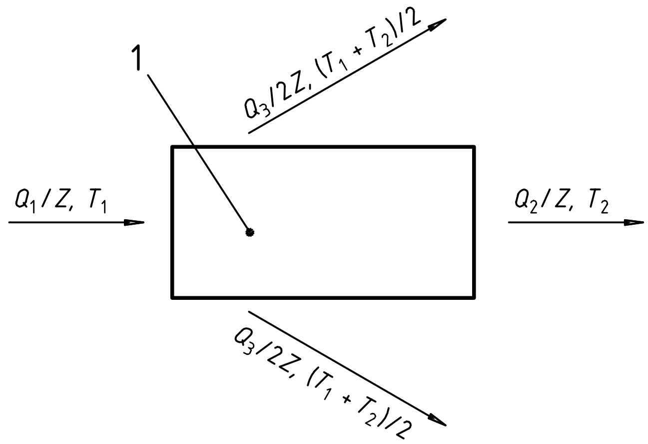
$$P_{\text{f}} = f^* \times \frac{U^2 \times \eta_{\text{eff}} \times B \times L \times Z}{h_{\text{min}}} \tag{5}$$

The function $f^* = f(h_{\text{min}}/C_{\text{wed}}\,;\, B/L)$ is presented in ISO 12130-2 on the basis of [5]. An approximate function is also given there.

5.5 Lubricant flow rate

The lubricant fed to the bearing forms a solid lubricant film separating the sliding surfaces. At the same time, the lubricant has the task of dissipating the frictional heat which develops in the bearing.

Due to the rotational motion of the thrust collar, the lubricant is carried, with increasing pressure, in the direction of the converging clearance gap. Thereby part of the lubricant is forced out at the sides of each pad. It is assumed that the lateral portions have approximately the same size. See Figure 3.



Key

1 Tilting-pad

Figure 3 — Schematic view of the lubricant balance and heat balance of one tilting pad

In Figure 3:

$$Q_1 = Q_2 + Q_3 \tag{6}$$

where

$$Q_1 = Q_1^* \times Q_0 \tag{7}$$

$$Q_3 = Q_3^* \times Q_0 \tag{8}$$

$$Q_2 = Q_1 - Q_3 \tag{9}$$

$$Q_0 = B \times h_{min} \times U \times Z \tag{10}$$

The relative values of $Q_1^* = Q_1/Q_0$ and $Q_3^* = Q_3/Q_0$ can be taken from ISO 12130-2 as a function of the geometry (B/L) and the arising relative lubricant film thickness h_{min}/C_{wed}. Approximate functions are also given there.

It is assumed that the lubricant forced out at the sides of the pads, at Q_3, has the temperature $(T_1 + T_2)/2$ and the lubricant forced out at the ends, at Q_2, has the temperature T_2.

5.6 Heat balance

5.6.1 General

The thermal condition of the plain bearing results from the heat balance.

The heat flow $P_{th,f}$ arising from the frictional power P_f in the bearing is dissipated via the bearing housing to the environment and via the lubricant emerging from the bearing. With practical applications, one of the two kinds of heat dissipation is predominant. Additional safety is given for the design by neglecting the other kind of heat dissipation. The following assumptions can be made:

a) With pressureless lubricated bearings (self-lubrication, natural cooling) heat dissipation to the environment takes place mostly by convection:

$$P_f = P_{th,amb}$$

b) With pressure-lubricated bearings (recirculating lubrication) heat dissipation takes place mostly via the lubricant (recooling):

$$P_f = P_{th,L}$$

5.6.2 Heat dissipation by convection

Heat dissipation by convection [5.6.1 a)] takes place by thermal conduction and lubricant recirculation in the bearing housing and subsequently by radiation and convection from the surface of the housing to the environment. According to [6] the complex processes during the heat dissipation can be summarized as follows:

$$P_{th,amb} = k_A \times A \times (T_B - T_{amb}) \tag{11}$$

where

$$k_A = 15\ \text{W/(m}^2\cdot\text{K) to } 20\ \text{W/(m}^2\cdot\text{K)}$$

or when the bearing housing is subjected to an air-flow at a velocity of $w_{amb} > 1{,}2$ m/s

$$k_A = 7 + 12\sqrt{w_{amb}} \tag{12}$$

where w_{amb} is expressed in m/s and k_A in W/(m^2·K).

NOTE Thereby, the factor k_A accounts for the thermal conduction in the bearing housing as well as for the convection and radiation from the bearing housing to the environment. That part of the frictional heat arising in the bearing, which is dissipated via the shaft, is neglected here due to its very small amount in most cases.

By equating P_f from equation (5) and $P_{th,amb}$ from equation (11) and with

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$$k = \frac{k_A \times A}{B \times L \times Z} \tag{13}$$

the effective bearing temperature is obtained

$$T_{\text{eff}} = f^* \times \frac{U^2 \times \eta_{\text{eff}}}{k \times h_{\text{min}}} + T_{\text{amb}} \tag{14}$$

In this case, the bearing temperature is

$$T_B = T_{\text{eff}} \tag{15}$$

If the heat-emitting surface A of the bearing housing is not known exactly, the following can be substituted as an approximation:

for cylindrical housings

$$A = 2 \times \frac{\pi}{4} \times D_H^2 + \pi D_H B_H \tag{16}$$

for bearings in the machine structure

$$A = (15 \text{ to } 20) \times B \times L \times Z \tag{17}$$

where

B_H is the axial housing width in metres;

D_H is the housing outside diameter in metres.

5.6.3 Heat dissipation by recirculating lubrication

In case of recirculating lubrication, heat dissipation takes place via the lubricant [5.6.1 b)]

$$P_{\text{th,L}} = \rho \times c_p \times Q(T_{\text{ex}} - T_{\text{en}}) \tag{18}$$

For mineral lubricants, the volume specific heat capacity amounts to

$$\rho \times c_p = 1{,}8 \times 10^6 \text{ J/(m}^3\text{·K)}$$

5.6.4 Mixing processes in the lubrication recess

As a tilting-pad thrust bearing consists of a certain number of separate tilting-pads it is necessary to consider not only the lubricant flow rate of one single tilting-pad but also the lubricant flow rate of the complete bearing and thus the mutual influence of the lubricant flow rate. The lubricant forced out at the end of the pads at Q_2 (according to Figure 3) is mixed with newly-fed lubricant in the gap between the next tilting-pad, i.e. the lubricant temperature T_1 at the inlet of the lubrication clearance gap is higher by ΔT_1 than that of the newly-fed lubricant with temperature T_{en} (see Figure 4).

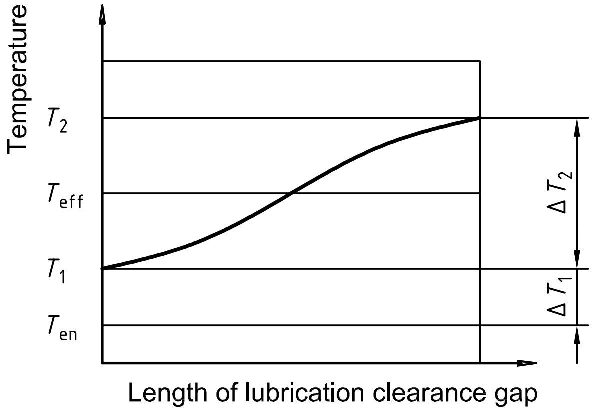

Figure 4 — Schematic view of the temperature distribution in the lubricant film

When determining the temperature difference

$$\Delta T_1 = T_1 - T_{en} \tag{19}$$

An empirical factor shall be introduced as a purely theoretical consideration of this mixing problem has not yet led to satisfying results.

A mixing factor M can be introduced as follows in order to achieve conformity with experience gathered (see [7]):

$$\Delta T_1 = \frac{Q_2}{M \times Q + (1 - M) \times Q_3} \times \Delta T_2 = \frac{Q_2^*}{M \times Q^* + (1 - M) \times Q_3^*} \times \Delta T_2 \tag{20}$$

for $Q \geqslant Q_3$ and $Q^* \geqslant Q$ respectively.

To explain the mixing factor we take a look at the limiting values. A mixing factor $M = 0$ means that there is no mixing in the gaps between the tilting pads, i.e. the lubricant flow rate Q_2 forced out of the lubrication clearance gaps completely enters the following lubrication clearance gap. On this assumption, a high lubricant flow rate Q would be ineffective as the largest part of this newly-fed lubricant would flow out of the gaps between the tilting pads in a radial direction without influencing the operational parameters. A mixing factor $M = 1$ means "complete" mixing in the gaps between the tilting pads. $M = 0{,}4$ up to 0,6 can be introduced as an empirical value. It is a function of the design and cannot be definitely indicated.

The total amount of lubricant to be fed to the thrust bearing can be determined from a given amount of heating

$$\Delta T = T_{ex} - T_{en} \tag{21}$$

$$Q = \frac{P_f}{c_p \times \rho \times \Delta T} = Q^* \times Q_0 \tag{22}$$

It results for the relative lubricant flow rate of the bearing:

$$Q^* = \frac{f^*}{F^*} \times \frac{F}{B \times L \times Z \times c_p \times \rho \times \Delta T} \tag{23}$$

By experience the value for ΔT is chosen in the range of 10 up to 30 K.

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With

$$\Delta T_2 = T_2 - T_1 \tag{24}$$

it can be written:

$$P_{th,L} = c_p \times \rho \times (Q_2 + 0{,}5 \times Q_3) \times \Delta T_2 \tag{25}$$

The following relationship results from equations (22) and (25) for the temperature rise in the lubrication clearance gap:

$$\Delta T_2 = \frac{\Delta T \times Q^*}{Q_2^* + 0{,}5 \times Q_3^*} = \frac{\Delta T \times Q^*}{Q_1^* - 0{,}5 \times Q_3^*} \tag{26}$$

with

$$\Delta T^* = \frac{\Delta T_1}{\Delta T_2}$$

the effective bearing temperature can be determined as follows:

$$T_{eff} = T_{en} + \Delta T_1 + 0{,}5 \times \Delta T_2 = T_{en} + (\Delta T^* + 0{,}5)\ \Delta T_2 \tag{27}$$

The bearing temperature is in this case

$$T_B = T_2 = T_{en} + \Delta T_1 + \Delta T_2 = T_{en} + (\Delta T^* + 1)\ \Delta T_2 \tag{28}$$

The permissibility of the values calculated for T_B and T_2 according to 5.6.2 and 5.6.3 is to be checked by comparison with the guide values T_{lim} according to ISO 12130-3.

5.7 Minimum lubricant film thickness and specific bearing load

After the calculation of the thermal steady-state condition, the minimum lubricant film thickness h_{min} can be calculated using the characteristic value of load-carrying capacity F^*.

The permissibility of this value for h_{min} is to be checked by comparison with the guide value h_{lim} in accordance with ISO 12130-3.

The permissibility of the specific bearing load

$$\overline{p} = \frac{F}{B \times L \times Z} \tag{29}$$

is to be checked by comparison with the guide values $\overline{p}_{lim}$ according to ISO 12130-3.

5.8 Operating conditions

If the plain bearing is to be operated under varying of operating conditions over a long period of time, then those operating conditions shall be checked under which $\overline{p}$, h_{min} and T_B are least favourable. First it shall be decided whether the bearing can be lubricated without pressure and whether heat dissipation by convection only is sufficient. For this purpose, the least favourable thermal case shall be investigated that, as a rule, corresponds to an operating condition at high rotational frequency and simultaneous high load. If, at pure convection, excessive bearing temperatures arise which even by increasing the dimensions of the bearing or of the surface area of the housing within the given range cannot be lowered to permissible values, then recirculating lubrication and oil recooling are necessary.

If an operating condition with high thermal loading (low dynamic lubricant viscosity) is followed directly by one with high specific bearing load and low rotational frequency, then this new operating condition shall be investigated while maintaining the thermal condition of the preceding operating point.

The transition into mixed lubrication takes place when the roughness peaks of thrust collar and bearing are in contact according to the criterion for $h_{\mathrm{lim,tr}}$ given in ISO 12130-3. Possible deformations have not been taken into account.

5.9 Further influence factors

The dynamic viscosity is strongly dependent on temperature. It is thus necessary to know the temperature dependence of the lubricant and its specification. See ISO 3448.

The effective dynamic viscosity η_{eff} is determined at the effective lubricant film temperature T_{eff}; i.e., η_{eff} results from averaging the temperatures T_1 and T_2 and not from averaging the dynamic viscosities $\eta(T_1)$ and $\eta(T_2)$.

The dynamic viscosity is also pressure-dependent but to a smaller degree. For bearings under steady-state conditions and the usual specific bearing loads $\bar{p}$, the pressure dependence can, however, be neglected. This neglect represents an additional factor of safety for the design.

In case of non Newtonian lubricants (intrinsically viscous oils, multi-range oils), reversible and irreversible fluctuations of viscosity occur as a function of the shearing stress in the lubrication clearance gap and of the service life. In [8], these effects are investigated for a few lubricants only and are not considered in this part of ISO 12130.

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Annex A
(normative)

Examples of calculation

A.1 Example

To be checked is a tilting pad thrust bearing with the dimensions D_i = 0,28 m, D_o = 0,34 m and B = 0,03 m operating under a constant load F = 25 000 N at a rotational frequency of 10 s^{-1}.

It is assumed that these operating conditions are the critical condition for the heat balance.

The bearing housing surface area is A = 1,25 m^2.

The oil is supplied via the inside diameter D_i. The lubricant used is an oil of viscosity ISO VG 68. Whether it is sufficient to effect heat dissipation by convection only is to be checked.

The ambient temperature shall be T_{amb} = 20 °C, the maximum permissible bearing temperature T_{lim} = 90 °C.

If T_{lim} is exceeded, recirculating lubrication with external recooling of oil shall be provided. It is assumed in this case that the lubricant is fed to the bearing with an oil inlet temperature T_{en} = 40 °C.

Dimensions and operational data

Bearing force at nominal rotational frequency	F = 25 000 N = constant
Bearing force under stationary conditions	F_{st} = 0
Thrust collar rotational frequency	N = 10 s^{-1}
Outside diameter over tilting-pads	$D_o = 340 \times 10^{-3}$ m
Inside diameter over tilting-pads	$D_i = 280 \times 10^{-3}$ m
Length of one tilting-pad	$L = 30 \times 10^{-3}$ m
Relative width of bearing	B/L = 1
Relative coordinate of supporting point in the direction of motion	a_F^* = 0,6
Number of tilting-pads	Z = 24
Heat emitting surface of the bearing housing	A = 1,25 m^2
Heat transfer coefficient	k = 20 W/($m^2 \cdot$K)
Ambient temperature	T_{amb} = 20 °C

Lubricant inlet temperature
with recirculating lubrication T_{en} = 40 °C

Lubricant outlet temperature
with recirculating lubrication T_{ex} = 50 °C

Limiting values according to ISO 12130-3:

Maximum permissible specific bearing load $\bar{p}_{lim} = 5 \times 10^6$ Pa

Maximum permissible bearing temperature T_{lim} = 90 °C

Minimum permissible lubricant film thickness $h_{lim} = 15 \times 10^{-6}$ m

Lubricant Oil ISO VG 68

Density of the lubricant $\rho = 900$ kg/m^3

Volume specific heat capacity of the lubricant $c_p \times \rho = 1{,}8 \times 10^6$ J/(m^3·K)

Critical Reynolds' number Re_{cr} = 600

Table A.1

T_{eff} °C	$\eta_{eff}(T_{eff})$ Pa·s
40	0,061
50	0,038
60	0,025
70	0,017
80	0,013
90	0,009 5

With the relative coordinate of the supporting point $a_F^* = 0{,}6$ and relative bearing width B/L = 1 in accordance with ISO 12130-2 the following values result (Figures 1, 2, 3, 4 and 5 referred to are in ISO 12130-2):

Relative minimum lubricant film thickness according to Figure 5 h_{min}/C_{wed} = 0,78

For h_{min}/C_{wed} = 0,78:

characteristic value of load carrying capacity according to Figure 1 F^* = 0,07

characteristic value of friction according to Figure 2 f^* = 0,69

relative lubricant flow rate at the inlet of the lubrication clearance gap
according to Figure 3 Q_1^* = 0,94

relative lubricant flow rate at the sides of the lubrication clearance gap
according to Figure 4 Q_3^* = 0,29

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Preliminary assumptions

Bearing temperature $T_{B,0} = 80\ °C$

Effective lubricant film temperature $T_{eff,0} = 80\ °C$

Calculation by means of the flow chart according to Figure 2

The sliding diameter D and the tilting-pad width B are calculated as follows:

$$D = \frac{D_o + D_i}{2} = 310 \times 10^{-3}\,\mathrm{m}$$

$$B = \frac{D_o - D_i}{2} = 30 \times 10^{-3}\,\mathrm{m}$$

where $U = \pi \times D \times N = \pi \times 310 \times 10^{-3} \times 10 = 9{,}74\ \mathrm{m/s}$

and $\eta_{eff} = 0{,}013$ from equation (3):

$$h_{min} = \sqrt{F^* \times \frac{U \times \eta_{eff} \times L^2 \times B \times Z}{F}}$$

$$h_{min} = \sqrt{0{,}07 \times \frac{9{,}74 \times 0{,}013 \times (30 \times 10^{-3})^2 \times 30 \times 10^{-3} \times 24}{25\,000}} = 15{,}2 \times 10^{-6}\ \mathrm{m}$$

Check for permissible specific bearing load according to equation (29).

$$\bar{p} = \frac{25\ 000}{30 \times 10^{-3} \times 30 \times 10^{-3} \times 24} = 1{,}16 \times 10^{6}\,\mathrm{Pa}$$

The specific bearing load is permissible as

$\bar{p} = 1{,}16 \times 10^{6}\ \mathrm{Pa} < 5 \times 10^{6}\ \mathrm{Pa}$ (see ISO 12130-3).

Heat dissipation by convection

First step:

Assumed bearing temperature $T_{B,0} = T_{eff} = 80\ °C$.

Effective dynamic viscosity of the lubricant at $T_{eff} = 80\ °C$ from the input data:

$\eta_{eff} = 0{,}013\ \mathrm{Pa{\cdot}s}$

where $\eta_{eff} = 0{,}013\ \mathrm{Pa{\cdot}s}$ it follows from equation (3)

$h_{min} = 15{,}2 \times 10^{-6}\ \mathrm{m}$

To check the assumed bearing temperature $T_{B,0}$, $T_{B,1}$ is determined according to equation (14) using equation (13):

$$k = \frac{20 \times 1{,}25}{30 \times 10^{-3} \times 30 \times 10^{-3} \times 24} = 1157{,}4\ \mathrm{W/(m^2{\cdot}K)}$$

$$T_{B,1} = 0{,}69 \times \frac{9{,}74^2 \times 0{,}013}{1157{,}4 \times 15{,}2 \times 10^{-6}} + 20 = 68{,}4\ °C.$$

As $T_{B,1} < T_{B,0}$ and the difference between $T_{B,0}$ and $T_{B,1}$ is not yet sufficiently small (if e.g., it is required that $\left| T_{B,0} - T_{B,1} \right| \leqslant 2K$), the assumed bearing temperature $T_{B,0}$ = 80 °C shall be corrected.

Second step:

Improved assumption of the bearing temperature:

$T_{B,0}$ = 0,5(80 + 68,4) = 74,2 °C.

η_{eff} = 0,015 2 Pa·s

h_{min} from equation (3) is:

$$h_{min} = \sqrt{0{,}07 \times \frac{9{,}74 \times 0{,}015\,2 \times (30 \times 10^{-3})^2 \times 30 \times 10^{-3} \times 24}{25\,000}} = 16{,}4 \times 10^{-6}\ \text{m}$$

$$T_{B,1} = 0{,}69 \times \frac{9{,}74^2 \times 0{,}015\,2}{1157{,}4 \times 16{,}4 \times 10^{-6}} + 20 = 72{,}4\ °C$$

As the difference between $T_{B,0}$ and $T_{B,1}$ is sufficiently small (if, e.g., it is required that $\left| T_{B,0} - T_{B,1} \right| \leqslant 2K$) then T_B = 72 °C can be accepted as the calculated bearing temperature.

In this case of heat dissipation by convection according to equation (15), the effective bearing temperature is equal to the bearing temperature.

Therefore the calculated value of T_B is compared with the limiting value T_{lim}:

$T_B = T_{eff} < T_{lim}$

As $T_B < T_{lim}$, the bearing temperature is permissible.

As $h_{min} > h_{lim}$, the minimum lubricant film thickness is permissible.

Check for laminar flow according to equation (2) with the effective viscosity and determined minimum lubricant film thickness h_{min}:

$$Re = \frac{900 \times 9{,}74 \times 16{,}4 \times 10^{-6}}{0{,}015\,2} = 9{,}5$$

$Re = 9{,}5 < Re_{cr} = 600$

For the calculated condition, the flow is laminar. That means that this part of ISO 12130 is applicable to this case.

According to equation (5), the frictional power is

$$P_f = 0{,}69 \times \frac{9{,}74^2 \times 0{,}015\,2 \times 30 \times 10^{-3} \times 30 \times 10^{-3} \times 24}{16{,}4 \times 10^{-6}} = 1{,}31 \times 10^3\ \text{W}$$

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A.2 Example

To be checked is a tilting-pad thrust bearing for speed-dependent load with the dimensions: D_i = 0,21 m, D_o = 0,33 m and B = 0,06 m, operating at a nominal rotational frequency of 50 s^{-1} and a load F = 40 000 N. At standstill the bearing is not loaded (F_{st} = 0).

It is assumed that this operating condition at nominal speed is the critical condition for the heat balance.

The bearing housing surface area is A = 1,2 m^2. The oil is supplied via the inside diameter D_i. The lubricant used shall be an oil ISO VG 46. Whether It is sufficient to effect heat dissipation by convection only shall be checked.

The ambient temperature shall be T_{amb} = 20 °C and the permissible bearing temperature T_{lim} = 90 °C.

If T_{lim} is exceeded, recirculating lubrication with external recooling of oil shall be provided. In this case it is assumed that the lubricant is fed to the bearing with an oil inlet temperature T_{en} = 40 °C.

Dimensions and operating data

Bearing force at nominal rotational frequency	F = 40 000 N
Bearing force under stationary conditions	F_{st} = 0
Thrust collar rotational frequency	N = 50 s^{-1}
Outside diameter over tilting pads	$D_o = 330 \times 10^{-3}$ m
Inside diameter over tilting pads	$D_i = 210 \times 10^{-3}$ m
Length of one tilting-pad	$L = 40 \times 10^{-3}$ m
Relative width of bearing	B/L = 1,5
Relative coordinate of supporting point in direction of motion	a_F^* = 0,6
Number of tilting-pads	Z = 12
Heat emitting surface of the bearing housing	A = 1,2 m^2
Heat transfer coefficient	k = 20 W/(m^2·K)
Ambient temperature	T_{amb} = 20 °C
Lubricant inlet temperature with recirculating lubrication	T_{en} = 40 °C
Lubricant outlet temperature with recirculating lubrication	T_{ex} = 50 °C

Limiting values according to ISO 12130-3:

Maximum permissible specific bearing load	$\bar{p}_{lim} = 5 \times 10^6$ Pa
Maximum permissible bearing temperature	T_{lim} = 90 °C

Minimum permissible lubricant film thickness	$h_{lim} = 10 \times 10^{-6}$ m
Lubricant	Oil ISO VG 46
Density of the lubricant	$\rho = 900$ kg/m^3
Volume specific heat capacity of the lubricant	$c_p \times \rho = 1{,}8 \times 10^6$ J/(m^3·K)
Critical Reynolds' number	$Re_{cr} = 600$

Table A.2

T_{eff} °C	$\eta_{eff}(T_{eff})$ Pa·s
40	0,041
50	0,027
60	0,018
70	0,013
80	0,009 5
90	0,007 3
100	0,005 7

With the relative coordinate of the supporting point $a_F^* = 0{,}6$ and the relative bearing width $B/L = 1{,}5$ in accordance with ISO 12130-2 the following values result (Figures 1, 2, 3, 4 and 5 referred to are in ISO 12130-2):

Relative minimum lubricant film thickness according to Figure 5 $h_{min}/C_{wed} = 0{,}68$

For $h_{min}/C_{wed} = 0{,}68$:

characteristic value of load carrying capacity according to Figure 1	$F^* = 0{,}095$
characteristic value of friction according to Figure 2	$f^* = 0{,}69$
relative lubricant flow rate at the inlet to the lubrication clearance gap according to Figure 3	$Q_1^* = 0{,}92$
relative lubricant flow rate at the sides of the lubrication clearance gap according to Figure 4	$Q_3^* = 0{,}25$

Preliminary assumptions:

Bearing temperature	$T_{B,0} = 90$ °C
Effective lubricant film temperature	$T_{eff,0} = 90$ °C

Calculation by means of the flow chart according to Figure 2

The sliding diameter D and the tilting-pad width B are calculated as follows:

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$$D = \frac{D_o + D_i}{2} = 270 \times 10^{-3} \text{ m}$$

$$D = \frac{D_o - D_i}{2} = 60 \times 10^{-3} \text{ m}$$

where $U = \pi \times D \times N = \pi \times 270 \times 10^{-3} \times 50 = 42{,}4$ m/s

and η_{eff} = 0,007 3 from equation (3):

$$h_{min} = \sqrt{F^* \times \frac{U \times \eta_{eff} \times L^2 \times B \times Z}{F}}$$

$$h_{min} = \sqrt{0{,}095 \times \frac{42{,}4 \times 0{,}007\,3 \times (40 \times 10^{-3})^2 \times 60 \times 10^{-3} \times 12}{40\,000}} = 29{,}1 \times 10^{-6} \text{ m}$$

Check for permissible specific bearing load according to equation (29)

$$\bar{p} = \frac{40\,000}{40 \times 10^{-3} \times 60 \times 10^{-3} \times 12} = 1{,}39 \times 10^{6} \text{ Pa}$$

The specific bearing load is permissible as $\bar{p} = 1{,}39 \times 10^6$ Pa $< 5 \times 10^6$ Pa (see ISO 12130-3).

Heat dissipation by convection

First step:

Assumed bearing temperature $T_{B,0} = T_{eff} = 90$ °C

Effective dynamic viscosity of the lubricant at $T_{eff} = 90$ °C from the input data:

η_{eff} = 0,007 3 Pa·s

where η_{eff} = 0,007 3 Pa·s it follows from equation (3):

$h_{min} = 29{,}1 \times 10^{-6}$ m

To check the assumed bearing temperature $T_{B,0}$, $T_{B,1}$ is calculated according to equation (14) using equation (13):

$$k = \frac{20 \times 1{,}2}{60 \times 10^{-3} \times 40 \times 10^{-3} \times 12} = 833{,}3 \text{ W/(m}^2\text{·K)}$$

$$T_{B,1} = 0{,}69 \times \frac{42{,}4^2 \times 0{,}007\,3}{833{,}3 \times 29{,}1 \times 10^{-6}} + 20 = 393{,}1 \text{ °C}$$

As $T_{B,1} > T_{B,0}$, the assumed bearing temperature $T_{B,0}$ = 90 °C is to be corrected.

Second step:

Improved assumption of the bearing temperature:

$T_{B,0} = 0{,}5(90 + 393{,}1) = 241{,}6$ °C

Further steps of iteration are given in Table A.3

In the fourth calculation step $|T_{B,1} - T_{B,0}| < 2$ K, i.e. the bearing temperature, $T_B = 186$ °C, has been calculated with sufficient accuracy.

Comparison with the limiting value T_{lim}:

$T_B > T_{lim}$, thus the bearing temperature is not permissible. Heat dissipation by convection is therefore not sufficient and the bearing is to be cooled by the lubricant (recirculating lubrication).

Table A.3

Variable parameter	Unit	Steps of iteration			
		1	2	3	4
$T_{B,0} = T_{eff}$	°C	90	241,6	195,7	187,4
η_{eff}	Pa·s	0,007 3	0,000 88	0,001 3	0,001 4
h_{min}	m	$29,1 \times 10^{-6}$	$10,1 \times 10^{-6}$	$12,4 \times 10^{-6}$	$13,0 \times 10^{-6}$
$T_{B,1}$	°C	393,1	149,8	179,1	186,2
$\lvert T_{B,1} - T_{B,0} \rvert$	K	303,1	91,8	16,6	1,2

Heat dissipation by recirculating lubrication

Relative lubricant flow rate of the bearing from equation (23) with the assumed lubricant temperature difference according to equation (21) $\Delta T = 12$ K:

$$Q^* = \frac{0{,}69}{0{,}095} \times \frac{40\,000}{12 \times 40 \times 10^{-3} \times 60 \times 10^{-3} \times 1{,}8 \times 10^{6} \times 12} = 0{,}467$$

Temperature rise in the lubrication clearance gap ΔT_2 according to equation (26):

$$\Delta T_2 = \frac{12 \times 0{,}467}{0{,}92 - (0{,}5 \times 0{,}25)} = 7{,}0 \text{ K}$$

Temperature rise of the lubricant after the mixing process ΔT_1 according to equation (20) with relative lubricant flow rates according to equations (7), (8), (9) and (22) with $M = 0{,}5$:

$$\Delta T_1 = \frac{0{,}92 \times 0{,}25}{0{,}5 \times 0{,}467 + (1 - 0{,}5) \times 0{,}25} = 13{,}1 \text{ K}$$

The effective bearing temperature T_{eff} results from equation (27):

$$T_{eff} = 40 + 13{,}1 + (0{,}5 \times 7{,}0) = 56{,}6 \text{ °C}$$

The bearing temperature T_B, which is equal to the lubricant outlet temperature T_2, according to equation (28):

$$T_B = T_2 = 40 + 13{,}1 + 7{,}0 = 60{,}1 \text{ °C}$$

The effective dynamic viscosity of the lubricant at $T_{eff} = 56{,}6$ °C results from the input data:

$$\eta_{eff} = 0{,}020 \text{ Pa·s}$$

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where η_{eff} = 0,020 Pa·s, it follows from equation (3):

$$h_{min} = \sqrt{0{,}095 \times \frac{42{,}4 \times 0{,}020 \times (40 \times 10^{-3})^2 \times 60 \times 10^{-3} \times 12}{40\ 000}} = 48{,}7 \times 10^{-6} \text{ m}$$

As $T_B = T_2 < T_{lim}$, the lubricant outlet temperature is permissible.

As $h_{min} > h_{lim}$, the minimum lubricant film thickness is permissible.

Check for laminar flow according to equation (2) with the effective viscosity and the determined minimum lubricant film thickness h_{min}:

$$Re = \frac{900 \times 42{,}4 \times 48{,}7 \times 10^{-6}}{0{,}020} = 92{,}9$$

$$Re = 92{,}9 < Re_{cr} = 600$$

For the calculated condition, the flow is laminar. That means that this part of ISO 12130 is applicable to this case.

Frictional power according to equation (5):

$$P_f = 0{,}69 \times \frac{42{,}4^2 \times 0{,}020 \times 60 \times 10^{-3} \times 40 \times 10^{-3} \times 12}{48{,}7 \times 10^{-6}} = 14{,}7 \times 10^3 \text{ W}$$

Relative lubricant flow rate of the bearing Q_0 according to equation (10):

$$Q_0 = 60 \times 10^{-3} \times 48{,}7 \times 10^{-6} \times 42{,}4 \times 12 = 1{,}49 \times 10^{-3} \text{ m}^3\text{/s}$$

Lubricant flow rate of the bearing Q according to equation (22):

$$Q = \frac{14{,}7 \times 10^3}{1{,}8 \times 10^6 \times 12} = 6{,}81 \times 10^{-4} \text{ m}^3\text{/s}$$

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