INTERNATIONAL STANDARD

ISO 12130-2

Third edition
2020-02

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

Part 2: Functions for calculation of tilting pad thrust bearings

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

Partie 2: Fonctions pour le calcul des butées à patins oscillants

Reference number
ISO 12130-2:2020(E)

ISO copyright office
CP 401 • Ch. de Blandonnet 8
CH-1214 Vernier, Geneva
Phone: +41 22 749 01 11
Fax: +41 22 749 09 47
Email: copyright@iso.org
Website: www.iso.org

Published in Switzerland

Contents

Page

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 third edition cancels and replaces the second edition (ISO 12130-2:2013), of which it constitutes a minor revision. The changes compared to the previous edition are as follows:

— adjustment to ISO/IEC Directives, Part 2:2018;

— correction of typographical errors.

A list of all parts in the ISO 12130 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.

Introduction

The functions of the following type are necessary for the calculation of oil-lubricated tilting pad thrust bearings in accordance with ISO 12130-1, assuming hydrodynamic conditions with full lubrication. They are based on the premises and boundary conditions specified therein. The values necessary for the calculation can be determined by means of the given formulae as well as from diagrams and tables. The formulae are approximations of the numerically determined values traced as curves in accordance with Reference [2]. The explanation of the symbols and examples for the calculation are included in ISO 12130-1.

On account of the premises laid down in ISO 12130-1:2001, Clause 3, g) and k), the following definitions are not applicable to the calculation of thrust bearings with centrally supported tilting pads ($a_F^* = 0{,}5$), which, under the premises indicated therein, have no hydrodynamic load-carrying capacity. For the determination of the characteristic values of such bearings, it is necessary to consider at least the deformations of the tilting pads which occur during operation. For example, References [3] and [4] can be compared.

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

Part 2: Functions for calculation of tilting pad thrust bearings

1 Scope

This document specifies the derivation of mathematical functions to be applied when calculating tilting pad thrust bearings.

This document is not applicable to heavily loaded tilting pad thrust bearings.

2 Normative references

There are no normative references in this document.

3 Terms and definitions

No terms and definitions are listed in this document.

ISO and IEC maintain terminological databases for use in standardization at the following addresses:

— ISO Online browsing platform: available at https://www.iso.org/obp

— IEC Electropedia: available at http://www.electropedia.org/

4 Functions for the tilting pad thrust bearing

4.1 General

An explanation of the symbols is given in ISO 12130-1.

4.2 Characteristic value of load-carrying capacity

The characteristic value of load-carrying capacity, F^* is defined as a function of the relative bearing width, B/L, and the relative minimum lubricant film thickness, h_{min}/C_{wed}.

By substituting Formulae (2) to (5) into Formula (1), the approximate curves in Figure 1 are obtained (range of application: $0{,}2 \leq \frac{h_{min}}{C_{wed}} \leq 2$).

$$F^* = 5\left(\frac{h_{min}}{C_{wed}}\right)^2 \times \left[\ln\frac{1+(h_{min}/C_{wed})}{h_{min}/C_{wed}} - \frac{2}{1+2\times\frac{h_{min}}{C_{wed}}}\right] \times \frac{A^* + B^*\left[1-\frac{1}{h_{min}/C_{wed}}\right] + C^*\left[1-\frac{1}{h_{min}/C_{wed}}\right]^2}{1+a\left[\frac{1}{B/L}\right]^2} \quad (1)$$

$$a=\frac{10}{\left(1+2\frac{h_{\min}}{C_{\text{wed}}}\right)^2}\times\left\{\left[\frac{h_{\min}}{C_{\text{wed}}}+\left(\frac{h_{\min}}{C_{\text{wed}}}\right)^2\right]^2+\frac{1-2\left[\frac{h_{\min}}{C_{\text{wed}}}+\left(\frac{h_{\min}}{C_{\text{wed}}}\right)^2\right]}{12\left[\left(1+2\frac{h_{\min}}{C_{\text{wed}}}\right)\times\ln\frac{1+\left(h_{\min}/C_{\text{wed}}\right)}{h_{\min}/C_{\text{wed}}}-2\right]}\right\} \tag{2}$$

$$A^*=1{,}168\,6-0{,}329\,45\times\left(\frac{B}{L}\right)+0{,}222\,67\times\left(\frac{B}{L}\right)^2-0{,}046\,51\times\left(\frac{B}{L}\right)^3 \tag{3}$$

$$B^*=-0{,}100\,95+0{,}197\,43\times\left(\frac{B}{L}\right)-0{,}131\,36\times\left(\frac{B}{L}\right)^2+0{,}028\,703\times\left(\frac{B}{L}\right)^3 \tag{4}$$

$$C^*=-0{,}004\,879\,1+0{,}008\,601\times\left(\frac{B}{L}\right)-0{,}005\,401\,5\times\left(\frac{B}{L}\right)^2+0{,}001\,127\,8\times\left(\frac{B}{L}\right)^3 \tag{5}$$

The values of F^* for $h_{\min}/C_{\text{wed}}$ are shown in Table 1.

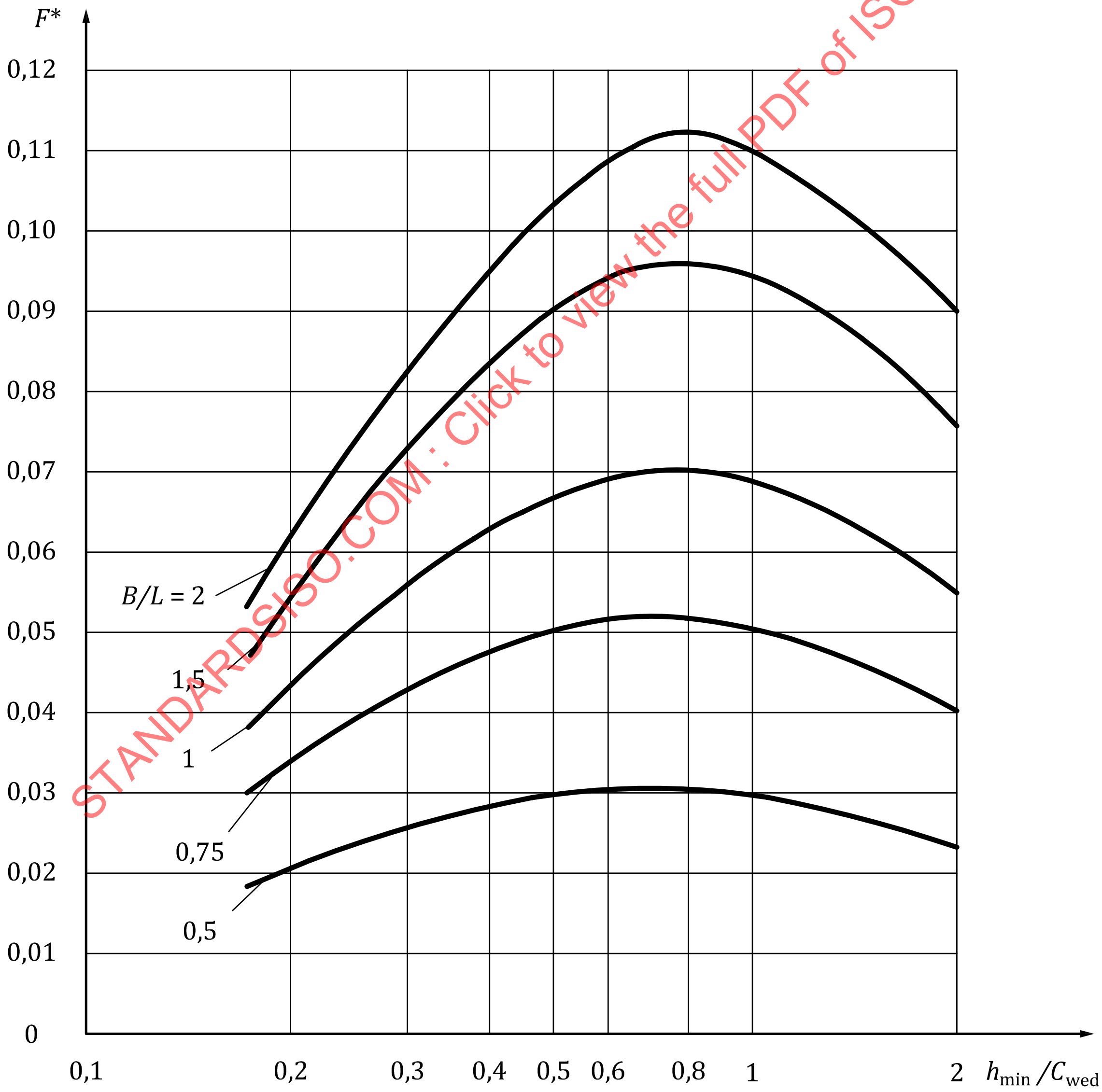

Figure 1 — Characteristic value of load-carrying capacity, F^*, as a function of the relative bearing width, B/L, and the relative minimum lubricant film thickness, $h_{\min}/C_{\text{wed}}$

Table 1 — Values to Figure 1 [$F^* = f(B/L, h_{min}/C_{wed})$]

h_{min}/C_{wed}	B/L				
	2	1,5	1	0,75	0,5
2,000	0,089 95	0,077 21	0,055 75	0,040 39	0,022 88
1,000	0,109 6	0,094 57	0,068 94	0,050 37	0,028 92
0,667	0,109 5	0,094 97	0,069 97	0,051 58	0,030 05
0,500	0,103 2	0,090 01	0,067 01	0,049 83	0,029 45
0,333	0,087 19	0,076 88	0,058 36	0,044 09	0,026 76
0,250	0,072 85	0,064 87	0,050 11	0,038 37	0,023 82
0,200	0,061 27	0,055 05	0,043 20	0,033 45	0,021 17

4.3 Characteristic value of friction

The characteristic value of friction, f^* is defined as a function of the relative bearing width, B/L, and the relative minimum lubricant film thickness, h_{min}/C_{wed}.

By substituting Formulae (7) and (8) into Formula (6), the approximate curves in Figure 2 are obtained (range of application: $0{,}2 \le \frac{h_{min}}{C_{wed}} \le 2$).

$$f^* = \frac{6}{5}\left\{4\times\frac{h_{min}}{C_{wed}}\times\ln\frac{1+(h_{min}/C_{wed})}{h_{min}/C_{wed}} - \frac{6\times\frac{h_{min}}{C_{wed}}}{1+2\times\frac{h_{min}}{C_{wed}}}\right\}\times\left\{1+\alpha\left[\frac{1}{B/L}\right]^2\right\}A^* \quad (6)$$

$$\alpha = \frac{10}{\left[1+2\times\frac{h_{min}}{C_{wed}}\right]^2}\left\{\left[\frac{h_{min}}{C_{wed}}+\left(\frac{h_{min}}{C_{wed}}\right)^2\right]^2 + \frac{1-2\left[\frac{h_{min}}{C_{wed}}+\left(\frac{h_{min}}{C_{wed}}\right)^2\right]}{12\left[\left(1+2\times\frac{h_{min}}{C_{wed}}\right)\times\ln\frac{1+(h_{min}/C_{wed})}{h_{min}/C_{wed}}-2\right]}\right\} \quad (7)$$

$$A^* = -0{,}214\ 59 + 0{,}880\ 71\left(\frac{B}{L}\right) - 0{,}297\ 60\left(\frac{B}{L}\right)^2 + 0{,}037\ 91\left(\frac{B}{L}\right)^3 \quad (8)$$

The values of f^* for h_{min}/C_{wed} are shown in Table 2.

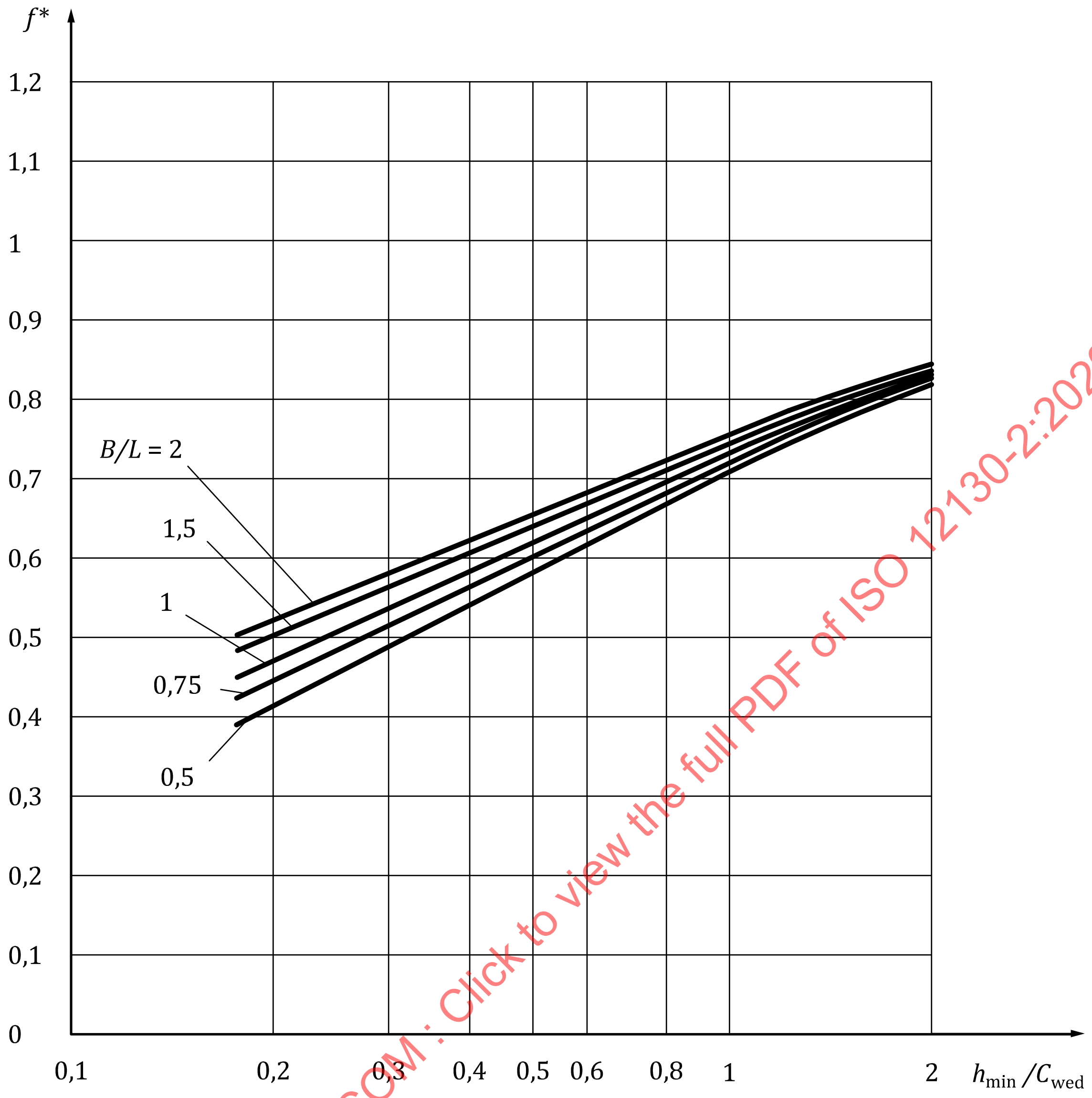

Figure 2 — Characteristic value of friction, f^*, as a function of the relative bearing width, B/L, and the relative minimum lubricant film thickness, h_{min}/C_{wed}

Table 2 — Values to Figure 2 [$f^* = f(B/L, h_{min}/C_{wed})$]

h_{min}/C_{wed}	B/L				
	2	1,5	1	0,75	0,5
2,000	0,833 4	0,830 2	0,824 9	0,821 0	0,816 7
1,000	0,748 0	0,740 4	0,727 6	0,718 3	0,707 6
0,667	0,693 0	0,682 1	0,663 3	0,649 5	0,633 4
0,500	0,652 5	0,639 3	0,616 3	0,599 1	0,678 8
0,333	0,592 9	0,577 4	0,549 6	0,528 2	0,502 2
0,250	0,548 1	0,532 1	0,502 6	0,479 1	0,450 0
0,200	0,511 5	0,496 0	0,466 3	0,442 0	0,411 3

4.4 Relative lubricant flow rate

The relative lubricant flow rates Q_1* and Q_3* are defined as a function of the relative bearing width, B/L, and the relative minimum lubricant film thickness, h_{min}/C_{wed}.

By substituting Formulae (10) to (13) into Formula (9), the approximate curves in Figures 3 and 4 are obtained (range of application: $0{,}2 \leq \frac{h_{min}}{C_{wed}} \leq 2$).

$$Q_i^* = \frac{1 + (h_{min}/C_{wed})}{1 + 2 \times \frac{h_{min}}{C_{wed}}} \times \left\{ A_i + B_i \times \left[1 - \frac{1}{h_{min}/C_{wed}} \right] \right\} \quad (9)$$

with constants A_i and B_i

— for i = 1:

$$A_i = 1{,}549\,4 - 0{,}344\,48 \left(\frac{B}{L} \right) + 0{,}072\,457 \left(\frac{B}{L} \right)^2 \quad (10)$$

$$B_i = -0{,}572\,08 + 0{,}370\,91 \left(\frac{B}{L} \right) - 0{,}079\,18 \left(\frac{B}{L} \right)^2 \quad (11)$$

— for i = 3:

$$A_i = 2 \left[0{,}358\,6 - 0{,}240\,57 \left(\frac{B}{L} \right) + 0{,}052\,129 \left(\frac{B}{L} \right)^2 \right] \quad (12)$$

$$B_i = 2 \left[-0{,}276\,82 + 0{,}186\,07 \left(\frac{B}{L} \right) - 0{,}040\,081 \left(\frac{B}{L} \right)^2 \right] \quad (13)$$

The values of Q_1* and Q_3* for h_{min}/C_{wed} are shown in Tables 3 and 4, respectively.

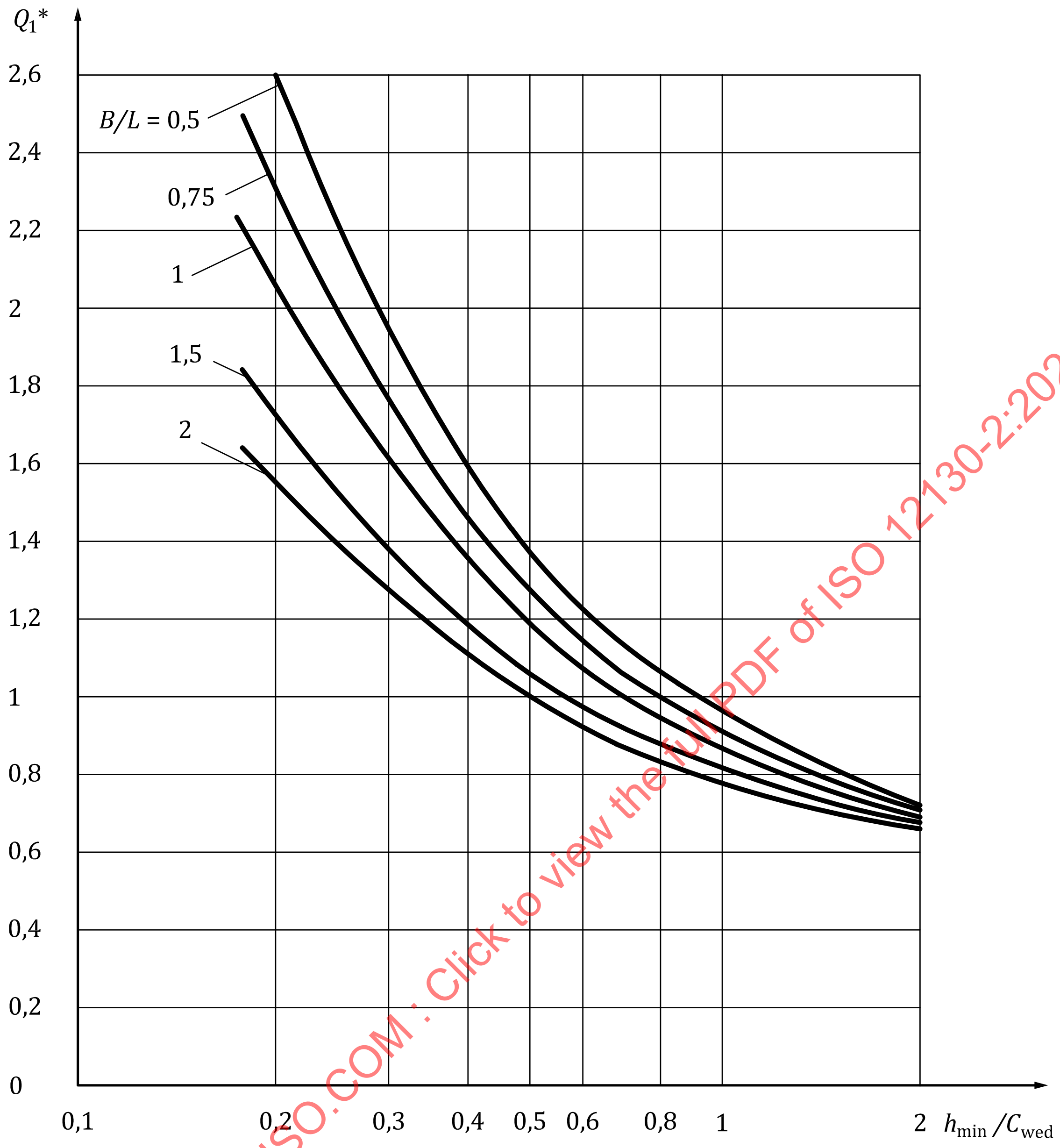

Figure 3 — Relative lubricant flow rate Q_1^* as a function of the relative bearing width, B/L, and the relative minimum lubricant film thickness, h_{min}/C_{wed}

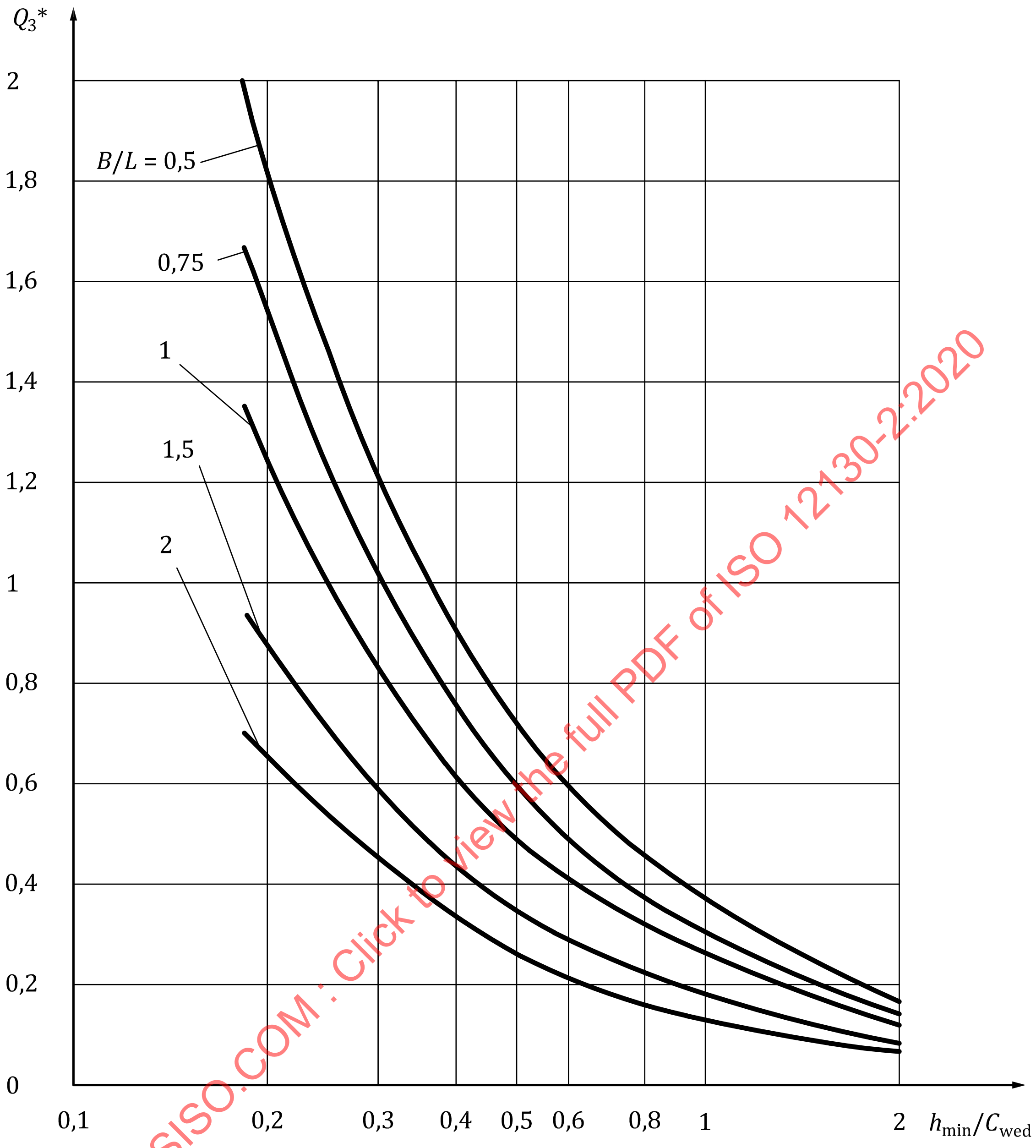

Figure 4 — Relative lubricant flow rate Q_3^* as a function of the relative bearing width, B/L, and the relative minimum lubricant film thickness, h_{min}/C_{wed}

Table 3 — Values to Figure 3 [$Q_1^* = f(B/L, h_{min}/C_{wed})$]

h_{min}/C_{wed}	B/L				
	2	1,5	1	0,75	0,5
2,000	0,643 5	0,657 1	0,678 6	0,696 0	0,715 2
1,000	0,765 5	0,796 0	0,847 3	0,884 3	0,927 0
0,667	0,876 0	0,925 3	1,008	1,068	1,137
0,500	0,979 5	1,049	1,165	1,249	1,345
0,333	1,173	1,283	1,470	1,607	1,761
0,250	1,362	1,510	1,769	1,960	2,174
0,200	1,544	1,731	2,063	2,311	2,588

Table 4 — Values to Figure 4 [$Q_3^* = f(B/L, h_{min}/C_{wed})$]

h_{min}/C_{wed}	B/L				
	2	1,5	1	0,75	0,5
2,000	0,064 75	0,086 07	0,122 3	0,147 7	0,178 1
1,000	0,129 4	0,172 4	0,246 2	0,297 2	0,357 6
0,667	0,194 6	0,259 5	0,370 7	0,447 9	0,539 0
0,500	0,259 8	0,347 1	0,496 4	0,600 1	0,721 4
0,333	0,390 8	0,522 8	0,750 0	0,907 7	1,091
0,250	0,522 0	0,699 3	1,006	1,219	1,463
0,200	0,653 5	0,876 0	1,263	1,531	1,838

4.5 Relative pressure centre coordinate or tilting pad supporting point

The relative pressure centre coordinate or tilting pad supporting point a_F^* is defined as a function of the relative bearing width, B/L, and the relative minimum lubricant film thickness, h_{min}/C_{wed}.

Formula (14) shows the approximate curves in Figure 5.

$$a_F^* = f(h_{min}/C_{wed}; B/L):$$

$$a_F^* = 0{,}5 + \left[a + \frac{b}{B/L}\right] \times \tanh\left\{\left[c + \frac{d}{B/L}\right] \times \frac{1}{h_{min}/C_{wed}}\right\} \quad (14)$$

Permissible input values: $0{,}2 \leq \frac{h_{min}}{C_{wed}} < 2$

$a = 0{,}138\ 107\ 909$

$b = 0{,}035\ 120\ 970\ 9$

$c = 0{,}476\ 542\ 662$

$d = 0,010\ 956\ 802\ 1$

The values of a_F* for h_{min}/C_{wed} are shown in Table 5.

$$\frac{h_{min}}{C_{wed}} = f\left(a_F^*; B/L\right): \frac{h_{min}}{C_{wed}} = 2 \times \frac{c + \frac{d}{B/L}}{\ln\left[\frac{a + \frac{b}{B/L} + a_F^* - 0,5}{a + \frac{b}{B/L} - a_F^* + 0,5}\right]}$$

Permissible values for calculation:

$$0,333 \le \frac{h_{min}}{C_{wed}} \le 1$$

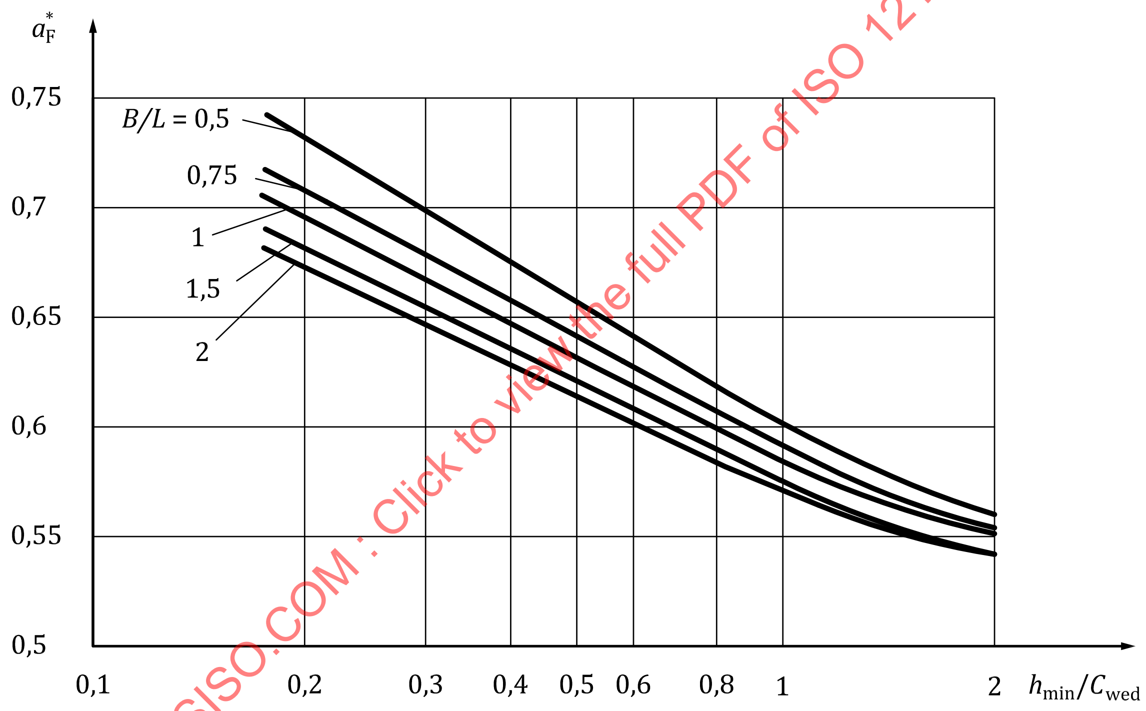

Figure 5 — Relative coordinate of the pressure centre or supporting point in the direction of motion (circumferential direction), a_F^*, as a function of the relative bearing width, B/L, and the relative minimum lubricant film thickness, h_{min}/C_{wed}

Table 5 — Values to Figure 5 [$a_F^* = f(B/L; h_{min}/C_{wed})$]

h_{min}/C_{wed}	B/L				
	2	1,5	1	0,75	0,5
2,000	0,543 1	0,544 6	0,548 3	0,552 2	0,559 7
1,000	0,573 0	0,575 6	0,581 8	0,588 3	0,600 5
0,667	0,595 5	0,599 0	0,606 9	0,615 2	0,630 7
0,500	0,613 2	0,617 4	0,626 8	0,636 4	0,654 1
0,333	0,639 7	0,645 1	0,656 7	0,667 9	0,688 5
0,250	0,658 6	0,665 2	0,678 3	0,690 6	0,712 7
0,200	0,672 9	0,680 4	0,695 0	0,707 8	0,730 9

5 Effective dynamic viscosity of the lubricant

The effective dynamic viscosity of the lubricant, η_{eff} is defined as a function of the effective lubricant film temperature, T_{eff}.

For liquid lubricants, Formula (15) given in Reference [5] is generally applicable:

$$\eta = K_1 \times \exp\left(\frac{K_2}{T + K_3}\right) \tag{15}$$

For mineral oils, this formula can be completed with sufficient accuracy by the constant K_3 = 95 °C according to Reference [6].

Reference [7] shows that the operational viscosity, η, for mineral oils can also be calculated directly from the ISO viscosity grade (VG).

With density, ρ, in kg/m^3, it results in Formula (16):

$$\ln\frac{\eta}{\eta_x} = \left(\frac{159{,}56}{T + 95\ °\text{C}} - 0{,}181\,913\right) \times \ln\frac{\rho \times \text{VG}}{10^6 \times \eta_x} \tag{16}$$

In Formula (16), $\eta_x = 0{,}18 \times 10^{-3}$ Pa·s is a constant coefficient.

The viscosity of ISO standard oils is given for a mean density ρ = 900 kg/m^3 in Figure 6.

Engine and gear box oils for road vehicles are standardized in accordance with SAE international viscosity classes.

The SAE classification of these lubricants can only be incompletely compared with the ISO VG classification. The SAE classification is so inaccurate that, for especially precise calculations, viscosity data should be requested from the supplier.

As compared to pure mineral oils, multigrade oils have a more even viscosity-temperature behaviour.

Synthetic oils very often reach such conditions without intrinsically viscous additives as required for mineral oils.