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Plain bearings — Bearing fatigue —

Part 1: Plain bearings in test rigs and in applications under conditions of hydrodynamic lubrication

Paliers lisses — Fatigue des paliers —

Partie 1: Paliers dans les machines d'essai et dans les applications en lubrification hydrodynamique

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Foreword

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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).

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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 bearing*, Subcommittee SC 2, *Material and lubricants, their properties, characteristics, test methods and testing conditions*.

This second edition cancels and replaces the first edition (ISO 7905-1:1995), which has been editorially revised.

The main changes compared to the previous edition are as follows:

— normative references have been updated;

— additional explanation for the test methods have been added.

A list of all parts in the ISO 7905 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 — Bearing fatigue —

Part 1:
Plain bearings in test rigs and in applications under conditions of hydrodynamic lubrication

1 Scope

This document specifies a method of improving test result comparability by evaluating the stresses in the bearing layers leading to fatigue (see Annex A). A similar evaluation is required in practical applications. Because the stresses are the result of pressure build-up in the hydrodynamic film, it is essential to fully state the conditions of operation and lubrication. In addition to dynamic loading, dimensional and running characteristics, the inclusion of the following adequately defines the fatigue system:

a) under conditions of dynamic loading the minimum bearing oil film thickness as a function of time and location to ensure no excessive local overheating or shearing as a result of mixed lubrication when running in;

b) the distribution of pressure circumferentially and axially with time under dynamic loading;

c) from this the resulting stresses in the bearing layers as a function of time and location, especially the maximum alternating stress.

Furthermore, bearing fatigue can be affected by mixed lubrication, wear, dirt, tribochemical reactions and other effects encountered in use thus complicating the fatigue problem. This document is therefore restricted to fatigue under full hydrodynamic separation of the bearing surfaces by a lubricant film.

This document applies to oil-lubricated plain cylindrical bearings, in test rigs and application running in conditions of full hydrodynamic lubrication. It comprises dynamic loading in bi-metal and multilayer bearings.

NOTE The number of practical applications with different requirements has led to the development of many bearing test rigs. If the conditions of lubrication employed on these test rigs are not defined in detail, test results from different rigs are generally neither comparable nor applicable in practice. Different test rigs can yield inconsistent ranking among equal materials.

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 4287, *Geometrical Product Specifications (GPS) — Surface texture: Profile method — Terms, definitions and surface texture parameters*

ISO 7902-1, *Hydrodynamic plain journal bearings under steady-state conditions — Circular cylindrical bearings — Part 1: Calculation procedure*

ISO 7902-2, *Hydrodynamic plain journal bearings under steady-state conditions — Circular cylindrical bearings — Part 2: Functions used in the calculation procedure*

ISO 7902-3, *Hydrodynamic plain journal bearings under steady-state conditions — Circular cylindrical bearings — Part 3: Permissible operational parameters*

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 Objective of testing

In this document the objective of testing with plain bearing test rigs, operating in conditions of full hydrodynamic lubrication, is to measure the dynamic load-carrying capacity e.g. the fatigue endurance limit of the bearing layer material in terms of amplitude of stress and number of cycles. This may be presented as a σ_{el}-N curve (endurance limit stress plotted against number of cycles), or as the endurance limit stress for a specified number of cycles. Endurance limit is reached when cracks (greater than 5 mm in length) appear in the bearing surface.

In terms of current understanding, the restriction to full hydrodynamic lubrication is a necessary simplification of the fatigue problem. This implies that the essential running-in of the bearing under test shall be carefully controlled to avoid significant predamage from excessive temperature and frictional shear stress which may cause surface microcracks.

It should be noted that fatigue testing of bearing materials may be conducted also by utilizing the more classic methods of testing. See ISO 7905-2 to ISO 7905-4.

5 Requirements

5.1 Test rigs

In order to define the operating and lubricating conditions, the test rig shall have the following characteristics:

a) simple and clear mechanical construction;

b) easy dismantling, preferably with an in situ bearing inspection capability;

c) bearing dimensional stability under test together with resistance to deformation of housing and shaft deflection;

d) adequate lubricant supply without impairing oil film pressure development;

e) be capable of exceeding the entire range of load/stress and temperature encountered in practice.

5.2 Test methods

The test methods shall have the following characteristics:

a) the ability to apply specialized measuring techniques for oil film thickness, lubricant temperature, pressure distribution and crack disintegration debris; such techniques for the latter aspect include continuous radio nuclide measurement of wear or X-ray fluorescent analysis of intermittently drained lubricant samples;

b) well-defined, experimentally verified hydrodynamic conditions (e.g the verification of viscosity and operating conditions indicative of hydrodynamic behaviour);

c) clear distinction between mixed lubrication during running-in and full hydrodynamic lubrication during fatigue testing;

d) the stress can traverse the bearing as uniformly as possible (rotating load) in order to detect irregularities in the bearing material;

e) simple, theoretically and experimentally reproducible hydrodynamic conditions (i.e. a rotating load produces a hydrodynamic film and pressure distribution equal to a static load);

f) repeatable and reliable assembly of components.

6 Test procedures

6.1 General

In order to assure the compatibility of test results from different test rigs and their putting into practice, all parameters controlling the hydrodynamic oil film shall be detailed, starting with test conditions, bearing dimensions, lubricant and other factors influencing hydrodynamic oil film. The following constitute the essential characteristic conditions and parameters for fatigue testing.

6.2 Characteristic conditions

6.2.1 Effective running-in procedure

This is designed in order to avoid excessive temperature and frictional shear stress due to heavy asperity contact. The progress of running-in may be monitored by measurements of temperature, electrical resistance, impedance or continuous radio nuclide measurement. For guidance h_0 should initially be greater than (Rz_b + Rz_s), where h_0 equals the minimum oil film thickness and shall be determined by measurement or calculation in accordance with ISO 7902-1 to ISO 7902-3, and Rz_b and Rz_s are the height of the profile irregularities in ten points of the bearing and the shaft face respectively, which shall be determined in accordance with ISO 4287. Polishing during running-in allows the value of h_0 to be reduced but during fatigue testing it should not be less than the initial value of Rz_s. The running-in procedure progressively reduces the minimum oil film thickness by a combination of reduced oil viscosity through increases in temperature, and by stepwise increases of load. The magnitude of load steps should be controlled by minimizing temperature spikes, excessive radio nuclide wear indication, or excessive duration of zero electrical contact resistance.

For electrical contact resistance control, the bearing is electrically isolated from the test rig. The electrical scheme should provide for monitoring a 10 mV difference of potential between the shaft and bearing at a supply point with 100 Ω. internal resistance, which drops to 0,01 mV during asperity contact. Load increments should be adjusted so as to minimize the duration of asperity contact.

WARNING — Radionuclides that find their way into the environment may cause harmful effects as radioactive contamination. They can also cause damage if they are excessively used during treatment or in other ways exposed to living beings, by radiation poisoning. Potential health damage from exposure to radionuclides depends on a number of factors and can damage the functions of healthy tissue/organs. Radiation exposure can produce effects ranging from skin redness and hair loss, to radiation burns and acute radiation syndrome. Prolonged exposure can lead to cells being damaged and in turn lead to cancer. Signs of cancerous cells might not show up until years, or even decades, after exposure[16].

6.2.2 Avoidance of deviation in the geometry of the structural elements of the plain bearing assembly

This is to avoid results being affected and their transferability reduced. Such geometrical discrepancies may include housing distortion, shaft deflection or misalignment and uneven hard rub marks in the plain bearing surface.

6.2.3 Effective temperature of the bearing and hydrodynamic film

These values calculated and verified in accordance with ISO 7902-1, ISO 7902-2, ISO 7902-3 represent the temperature distribution uniformity. Alternatively, the temperatures of oil inlet, outlet splash in the main loaded area and bearing surface/subsurface are to be measured.

6.2.4 Dynamic load amplitude and direction as a function of time

These form the basis of evaluation of peripheral/axial film pressure distribution as a function of time and position on the bearing surface. Alternatively, the measurement of pressure distribution may be used. Either method is to be used for evaluating the dynamic stresses in the individual bearing layers in order to find the surface location of maximum stress in terms of the mean and alternating stress at the endurance limit.

Pressure measurement not affecting hydrodynamic film development and stress by gauges may be carried out by evaporated thin metal film techniques. The measurement should be conducted beforehand under the same conditions, but not during the fatigue testing procedure.

6.2.5 Number of load cycles required to effect the first fatigue damage

This damage should be in the form of a crack or cracks (greater than 5 mm in length) or breakout of bearing lining material. Normally σ_{el}-N curve testing is terminated for practical considerations at 50×10^6 stress cycles. The endurance limit stress may be quoted at a specified number of cycles; e.g. 3×10^6, 10×10^6, 25×10^6 or 50×10^6. A specimen without failure during fatigue testing to a specified endurance should be identified in the report. Due to the scatter of test results normally experienced and the statistical nature of the fatigue limit, it is recommended that the results are evaluated on the basis of statistical methods.

6.3 Characteristic information

6.3.1 General

If the evaluation of the test results up to the endurance limit stress at fixed temperatures, controlled to ±2 °C, is not carried out by the investigator himself then it will be necessary to fully report the information below. If the bearing material undergoes change during test (e.g. diffusion or a similar process) this should be documented as additional information (e.g. a metallurgical report). The information is subdivided in such a way that the data requirements may be reduced depending on the degree of detailed evaluation of the end result – the endurance limit stresses.

6.3.2 Test rig description

This should comprise the designation, construction, load principles, design limits, lubricant supply including ancillary equipment and the measuring method and arrangements.

6.3.3 Test bearing description

This should consist of the following dimensions: bearing, including different layer thicknesses; housing in the diametral and axial directions; clearance, especially under test conditions; surface roughness parameters. Additionally, the material designation should be provided comprising chemical composition, manufacturing processes with thermophysical treatment and static strength data including Young's modulus and Poisson's ratio.

6.3.4 Test journal description

This should include dimensions, surface roughness parameter, hardness, material and, if evident, deflection and misalignment values.

6.3.5 Specific details of test load

This should include amplitude and direction, as a function of time; frequency and shaft speed, both during running-in and fatigue testing; the duration of the test.

6.3.6 Designation of lubricant and supply

This should include:

— type of lubricant;

— viscosity-temperature and density-temperature relationships;

— feed pressure;

— detailed dimensions and location of supply holes (or grooves);

— flow rate.

6.3.7 Test temperatures description

This should comprise the film temperature in bulk and inlet; outlet splash and representative bearing temperature near the damage zone as close as possible to the surface without disturbing the film pressure development.

All of the above descriptions are necessary for evaluating the hydrodynamic status of the bearing under test. If the hydrodynamic status is evaluated, then the information required is restricted to the following, together with data on bearing material temperature.

6.3.8 Test film thickness description

This should consist of film thickness variation with time and location in the bearing and minimum film thickness related to roughness data during running-in and fatigue testing.

6.3.9 Test film pressure description

This should contain lubricant film pressure distribution and variation with time and location relative to the bearing surface, in such detail that pressure gradients are indicated with sufficient precision.

6.3.10 Description of the dynamic stresses of the test

This should include the distribution with time and location relative to the bearing surface in order to determine the position of maximum fatigue stress by mean and alternating stress at the endurance limit.

The results may be compared with data from other mechanical test methods (see ISO 7905-2 to ISO 7905-4) by means of the Haigh diagram in which stress amplitude is plotted against mean stress.

6.3.11 Other test results

These should comprise a description of the damage; position and extent of cracking; absence or presence of wear or scoring; together with any findings resulting from a metallurgical examination. If measurable wear has occurred, i.e. more than light polishing, when no breakout of lining material has occurred, it shall be concluded that the oil film thickness is inadequate and the conditions of the test shall be changed in order to avoid wear.

7 Evaluation of stress in bearing materials

Evaluation of the stress relevant to fatigue is simpler if the hydrodynamic conditions are easily reproduced. The simplest dynamic load condition is one of pure rotation represented by a shaft loaded with out-of-balance masses to reduce shaft deflection. The hydrodynamic film condition is one of a purely rotating pressure which is most exactly describable by a calculation model. A predetermination at the representative mean and alternating stress at fixed Sommerfeld numbers and bearing width ratios is possible, if the following assumptions are applied: a cylindrical bearing; no significant misalignment or distortion; an optimum oil supply with film pressure development unimpaired; a fixed relationship of housing dimension, Young's modulus and Poisson's ratio (see Annex A).

In order to cause failure by fatigue in high strength material without wear or seizure it will be necessary to select the hydrodynamic characteristics (clearance, lubricant viscosity and very low surface roughness) to provide sufficient minimum oil film thickness to prevent metallic contact. It may also be possible to perform a similar determination for test rigs with a unidirectional pure sinusoidal load.

Annex A
(informative)

Evaluation of stress

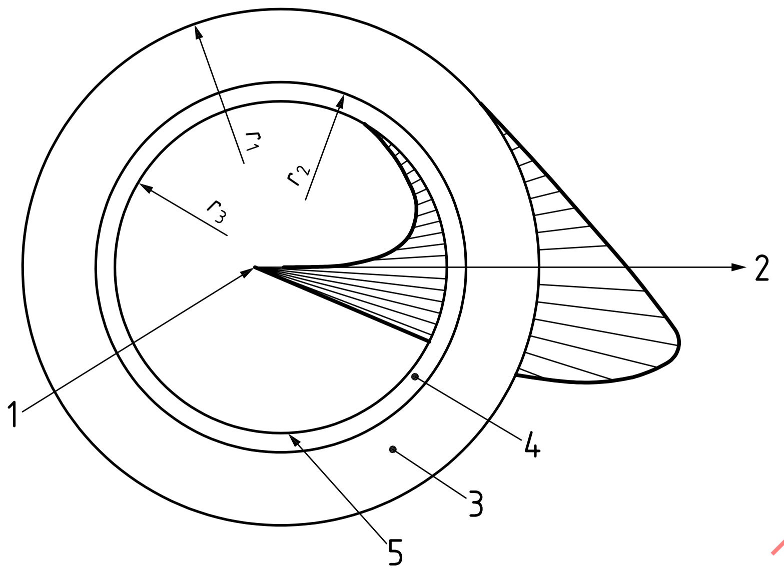
A.1 Evaluation of fatigue stresses

From practical experience and research, it is evident that fatigue starts with axial cracks in cylindrical bearings due to alternating tangential stresses. Whilst it is probable that the stresses will vary in the axial as well as the circumferential plane, in the absence of a full three-dimensional solution, evaluation may be made of the tangential stresses in the middle plane of the bearing, i.e. a two-dimensional solution.

Under dynamic load which varies not only with time, but also with position on the surface, the different time and location-dependent film pressures produce tangential stresses within the bearing layers. In order to evaluate the stress distribution resulting from momentary pressure distribution in the middle plane the bearing may be represented by a cylindrical ring including the bearing housing. Loading is by momentary film pressure at the inner running diameter balanced by outer diameter reaction pressures.

The ring model (see Figure A.1) may be treated as different material layers. On using such a system, the tangential stresses can be evaluated by several solutions. These are Airy's stress function [Formulae (A.1) or (A.2)] and analytical methods [Formulae (A.3), (A.4), (A.5), (A.6) and (A.7)] including a very exact simplification for very thin overlays. Others could be developed using stress analysis methods such as finite and boundary element techniques. The stress calculation must be applied in an adequate increment of the bearing circumference and load cycle to evaluate the mean and alternating stresses in a sufficient number of circumferential locations. Their maximum amplitudes will be responsible for fatigue.

It therefore becomes apparent that fatigue stress calculation under pure rotating load is simpler because an invariable film pressure distribution rotates round the bearing circumference and the resulting stresses likewise rotate in fixed distribution. Thus, only one pressure and resulting stress distribution has to be evaluated to determine the maximum compressive and tensile amplitudes at the same circumferential location to obtain mean and alternating stress amplitudes.



Key

1 peak oil film pressure
2 direction of load
3 ring 1 (housing and steel back) E_1, ν_1
4 ring 2 (lining/interlayer) E_2, ν_2
5 ring 3 (overlay) E_3, ν_3

Figure A.1 — Bearing ring model

A.2 Symbols

Symbol	Definition	Unit
b	bearing width	mm
d	diameter of running surface, $d=2r_3$	mm
d_H	housing diameter, $d_H=2r_1$	mm
d_H^*	dimensionless outer diameter of housing, $d_H^*=d_H/d$	—
	dimensionless outer diameter, valid for Figure A.3, $d_{H,0}^*=d_H/d=1{,}45$	—
E	Young's modulus	MPa
E^*	dimensionless Young's modulus, $E^* = E_2/E_{2,0}$	—
E_1	Young's modulus, housing and steel back	MPa
E_2	Young's modulus, lining	MPa
E_3	Young's modulus, overlay	MPa
$E_{2,0}$	Young's modulus for Figure A.3, $E_{2,0} = 63 \times 10^3$	MPa
$E_{3,0}$	Young's modulus, overlay, $E_{3,0} = 20 \times 10^3$	MPa
h_0	initial minimum lubricant film thickness	mm
K_H	correction factor for other housing dimension, d_H/d not equal to 1,45 (see Figure A.5).	—
K_2	correction factor for other lining thickness $s_{2,0}^*=s_2/d$ not equal to 0,004 7 (see Figure A.6).	—
N	Number of cycles	—
n	rotational speed	min^{-1}

Symbol	Definition	Unit
p	specific load	MPa
Rz	surface roughness (height of the profile irregularities in ten points)	µm
R^*	stress ratio, $R^* = \sigma_{min}/\sigma_{max}$	—
R_2^*	stress ratio, lining	—
R_3^*	stress ratio, overlay	—
r_1	outer radius of ring (housing and steel back)	mm
r_2	radius of interface between bearing back and lining	mm
r_3	radius of running surface (overlay thickness negligible)	mm
So	Sommerfeld number	—
s_2	thickness of lining	mm
s_2^*	dimensionless lining thickness, $s_2^* = s_2/d$	—
$s_{2,0}^*$	dimensionless lining thickness, valid for Figure A.3, $s_{2,0}^* = s_2/d = 0{,}0047$	—
η_{eff}	effective dynamic viscosity	Pa s
V	Poisson's ratio	—
v_1	Poisson's ratio, housing and steel back	—
v_2	Poisson's ratio, valid for Figure A.3 (all linings, $v_2 = 0{,}34$)	—
v_3	Poisson's ratio, valid for Figure A.4 (all overlays, $v_3 = 0{,}33$)	—
σ	stress	MPa
$\overline{\sigma}$	mean stress	MPa
σ^*	dimensionless stress, $\sigma^* = \sigma/p$	—
σ_A	alternating stress amplitude	MPa
σ^*_A	dimensionless alternating stress amplitude	—
σ_{el}	endurance limit stress	MPa
$\sigma^*_{A,2}$	dimensionless stress, lining	—
$\sigma^*_{A,3}$	dimensionless stress, overlay	—
ψ	relative bearing clearance	—
ω	angular velocity	s^{-1}

Subscripts:

A amplitude

b bearing

H housing

R^* stress ratio

s shaft

1 housing and steel back

2 Lining / interlayer

3 overlay

A.3 Stresses in bearing layers under rotating load

The range of tangential stress in bearing layers can be calculated for rotating load in dimensionless terms of stress $\sigma^* = \sigma/p$, i.e. related to specific load p as a function of Sommerfeld number:

$$So = \frac{p \times \psi^2}{\eta_{\mathrm{eff}} \times \omega}$$

Figures A.3 and A.4 present the alternating stress amplitude σ_{A}^* as a function of Sommerfeld number and bearing diameter/width ratio d/b for the bearing lining (interlayer) and overlay, namely for fixed values of the diameter/bearing width ratio d/b, the bearing housing parameter d_{H}^* , the lining thickness parameter s_2^* and for bearing lining material with Young's modulus $E_{2,0}$ = 63 × 10^3 MPa. Young's modulus E_3 for overlay material and Poisson's ratios for both layers are fixed as given in the list of symbols (see A.2).

For bearing lining material with Young's modulus E_2 not equal to $E_{2,0}$ the values of stress amplitude σ_{A}^* are obtained by:

For lining:

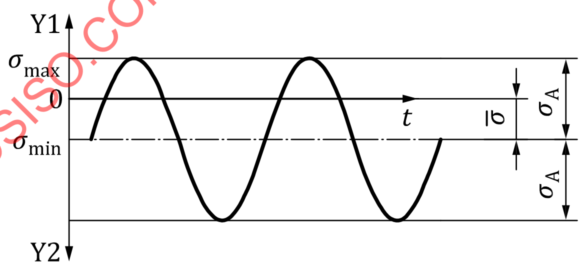
$$\sigma_{\mathrm{A},2}^* = \sigma_{\mathrm{A},2,0}^* \left(0{,}852 + 0{,}1430 \times E^*\right) \left(\frac{d}{b}\right)^{(-0{,}1034 + 0{,}1010 \times E^*)} \quad \text{(A.1)}$$

For overlay:

$$\sigma_{\mathrm{A},3}^* = \sigma_{\mathrm{A},3,0}^* \left(1{,}004 \times E^*\right)^{-0{,}0888} \quad \text{(A.2)}$$

Figures A.3 and A.4 include formulae for calculating the stress ratio $R^* = \sigma_{\mathrm{min.}}/\sigma_{\mathrm{max.}}$ (see Figure A.2). From this ratio, the mean stress $\overline{\sigma}$ can be obtained from the following formula:

$$\overline{\sigma} = \sigma_{\mathrm{A}} \times \frac{1 + R^*}{1 - R^*} \quad \text{(A.3)}$$



Key

Y1 tension

Y2 compression

t time

Figure A.2 — Sinusoidal stress curve

Mean stress is normally negative (compressive stress). The stress ratio R^* is nearly independent from d/b and only a function of Sommerfeld number. However, it has to be corrected for other lining material with modulus not equal to $E_{2,0}$ = 63 × 10^3 MPa:

For lining

$$R_2^* = -4{,}410 \times E^{*(-1{,}111)} + 0{,}023\,9 \times So \times E^{*(-2{,}542)} \tag{A.4}$$

For overlay

$$R_3^* = -3{,}200 \times E^{*(-0{,}6149)} + 0{,}0202 \times So \times E^{*(-0{,}4071)} \tag{A.5}$$

For other housing diameter or lining thickness parameters Figures A.5 and A.6 give correction factors K_H and K_2 in order to transfer the results from Figures A.3 and A.4 to other related bearing dimensions by simple multiplication. These factors are different for lining and overlay for both σ_A, and R^*.

$$\sigma_A = \sigma_A^* \times p \times K_{H,A} \times K_{2,A} \tag{A.6}$$

$$R^* = R_0^* \times K_{H,R^*} \times K_{2,R^*} \tag{A.7}$$

A.4 Worked example

Bearing fatigue of lead-based white metal lining PbSb14Sn1 under rotating specific load 14,7 MPa started after 1,8 × 10^6 load cycles. The bearing data were:

d = 61,4 mm

b = 24,6 mm

d/b = 2,5

Relative clearance (averaged value) ψ = 1/1 000

Housing outer diameter d_H = 170 mm d_H^* = 2,77

Lining thickness s_2 = 0,5 mm s_2^* = 0,008 1

Effective dynamic viscosity at 100 °C $\eta_{eff} = 1 \times 10^{-2}$ Pa s

Young's modulus of lining $E_2 = 29{,}5 \times 10^3$ MPa

Rotating speed n = 3 000 min^{-1} ω = 314,16 s^{-1}

Sommerfeld number

$$So = \frac{14{,}7 \times 10^6 \times 1 \times 10^{-6}}{1 \times 10^{-2} \times 314{,}16} = 4{,}68$$

From Figure A.3 for So = 4,68 and d/b = 2,5 find the dimensionless alternating stress in the lining:

$$\sigma^*_{A,2,0} = 0{,}95$$

To correct for the actual Young's modulus use Formula (A.1) with E^* = 29,5/63 = 0,468:

$$\sigma^*_{A,2} = 0{,}95 \times (0{,}852 + 0{,}1438 \times 0{,}468) \times 2{,}5^{(-0{,}1034 + 0{,}1010 \times 0{,}468)}$$
$$= 0{,}95 \times (0{,}852 + 0{,}0673) \times 2{,}5^{-0{,}056}$$
$$= 0{,}9 \times 0{,}9193 \times 0{,}9499 = 0{,}83$$

For stress ratio R^*_2, calculate from Formula (A.4) correction for Young's modulus:

$$R^* = -4{,}410 \times 0{,}468^{-1{,}111} + 0{,}0239 \times 4{,}68 \times 0{,}468^{-2{,}542}$$
$$= -4{,}410 \times 2{,}323 + 0{,}0239 \times 4{,}68 \times 6{,}881$$
$$= -10{,}24 + 0{,}77 = -9{,}47$$

Correction for housing diameter with $d^*_H / d^*_{H,0}$ = 2,77/1,45 = 1,91 and extrapolating Figure A.5 for lining gives:

$K_{H,A,2}$ = 1,30

and

$K_{H,R^*,2}$ = 0,90

Correction for lining thickness with $s^*_2 / s^*_{2,0}$ = 0,008 1/0,004 7 = 1,72 from Figure A.6 gives:

$K_{2,A,2}$ = 0,99

and

$K_{2,R^*,2}$ = 0,96

With specific load 14,7 MPa and the above calculated corrections the actual alternating stress amplitude is:

$$\sigma_A = \sigma^*_A \times p \times K_{H,A,2} \times K_{2,A,2} = 0{,}83 \times 14{,}7 \times 1{,}30 \times 0{,}99 = 15{,}7 \text{ in MPa}$$

The stress ratio is:

$$R^* = R^*_0 \times K_{H,R^*,2} \times K_{2,R^*,2} = -9{,}47 \times 0{,}90 \times 0{,}96 = -8{,}2$$

Finally the actual mean stress from Formula (A.3) is:

$$\bar{\sigma} = 15{,}7 \times \frac{-7{,}2}{9{,}2} = -12{,}3 \text{ in MPa}$$

