U.PORTO
FEUP FACULDADE DE ENGENHARIA
UNIVERSIDADE DO PORTO

Rolling Bearings

Complements of Machine Elements

Carlos Fernandes

2023

Mestrado em Engenharia Mecânica

Since there is no model in nature for guiding wheels on axles or axle journals, man faced a great task in designing bearings – a task which has not lost its importance and attraction to this day.

Rolling Bearings and Their Contribution to the Progress of Technology

Recommended bibliography

- Robert C. Juvinall, Kurt M. Marshek; “Fundamentals of machine component design”, Wiley, 2017.

- Fernandes, C. M. C. G., Marques, P. M. T., Martins, R. C., Seabra, J. H. O. (2015). Gearbox power loss. Part I: Losses in rolling bearings. Tribology International, 88(0)

- Harris, T. A., & Kotzalas, M. N. (2006). Rolling Bearing Analysis Essential Concepts of Bearing Technology (5th ed.). Taylor & Francis Group.

Hyperlink

Lecture 1

Summary

Introduction

Since ancient times humankind has developed multiple ways of reducing friction.

An example of the use of friction reduction techniques is the building of the pyramids of Egypt as represented in Figure 1.

The simplest possible bearings are unlubricated plain or sliding bearings—like the wooden cart wheels mounted directly on wooden axles in ancient times [1, 2].

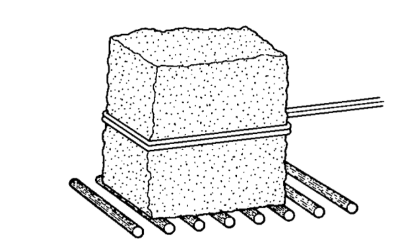


Figure 1: Heavy stone transportation.

Figure 2: Wooden cart.

Introduction

Ball and roller bearings, together called rolling bearings, are commonly used machine elements. They are used to permit motion of, or about, shafts in simple commercial devices such as bicycles, roller skates, and electric motors.

Complex mechanisms such as aircraft gas turbines, rolling mills, dental drills, gyroscopes, and power transmissions also use rolling bearings [1, 3].

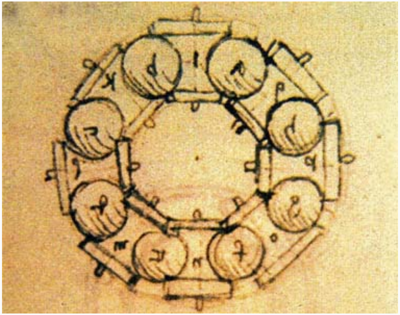


Figure 3: Ancient usage of bearings.

Rolling bearing principles

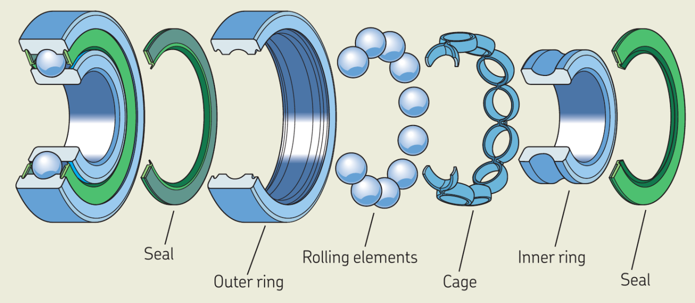


Figure 4: Rolling bearing components [4].

- inner ring (anel interno)
- outer ring (anel externo)
- rolling elements (elementos rolantes)
- cage (gaiola)
- seal (vedante)

Rolling elements

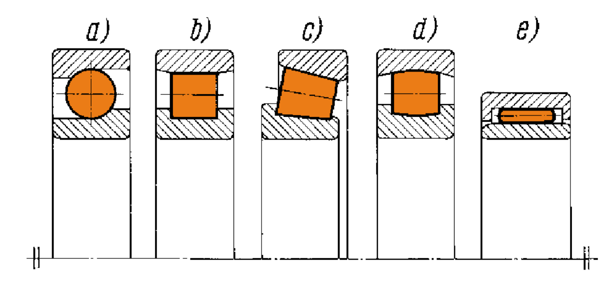


Figure 5: Rolling element shapes [5].

a) Ball (esferas)
b) Cylindrical roller (rolos cilíndricos)
c) Tapered roller (rolos cónicos)
d) Barrel roller (autocompensador de rolos)
e) Needle roller (agulhas)

Rolling bearing cage types

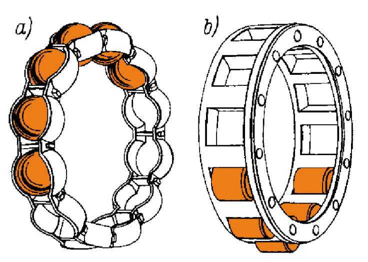


Figure 6: Rolling bearing cage types [5].

a) Sheet metal cage: steel

b) Solid section cage: brass, bronze, polyamide

Rolling bearing shields

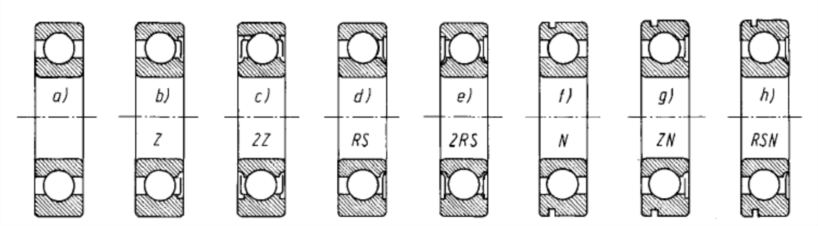


Figure 7: Types of deep groove ball bearings (DIN 625) [5].

a) normal design
b) with one shield disc (steel)
c) with two shield discs (steel)
d) with one seal
e) with two seals - life lubricated
f) with annular groove for snap ring
g) with annular groove and shield
h) with annular groove and one seal

Supported load

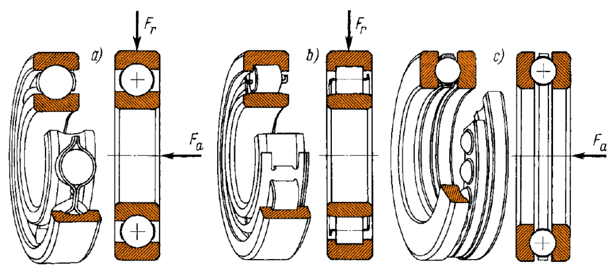
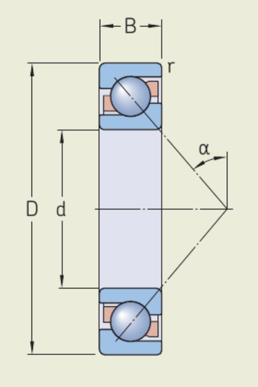


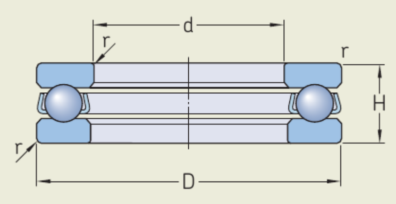
Figure 8: [5].

a) Axial (thrust) and radial
b) Radial
c) Axial (thrust)

Rolling bearing terminology



(a) Radial bearings



(b) Thrust bearings

Figure 9: Rolling bearing terminology [4].

d - Bore diameter; **D** - Outside diameter

B - Bearing width; **H** - Bearing height

r - Chamfer dimension;

α - Contact angle

Rolling bearing types - radial ball bearings

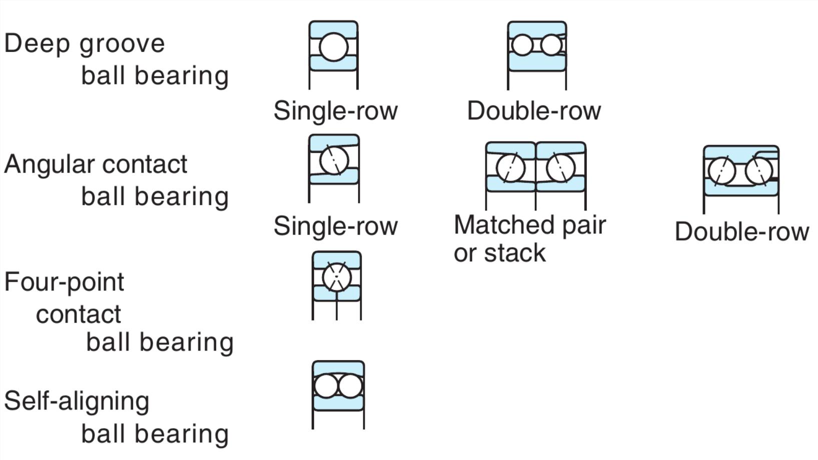


Figure 10: Radial ball bearing types [6].

Rolling bearing types - radial roller bearings

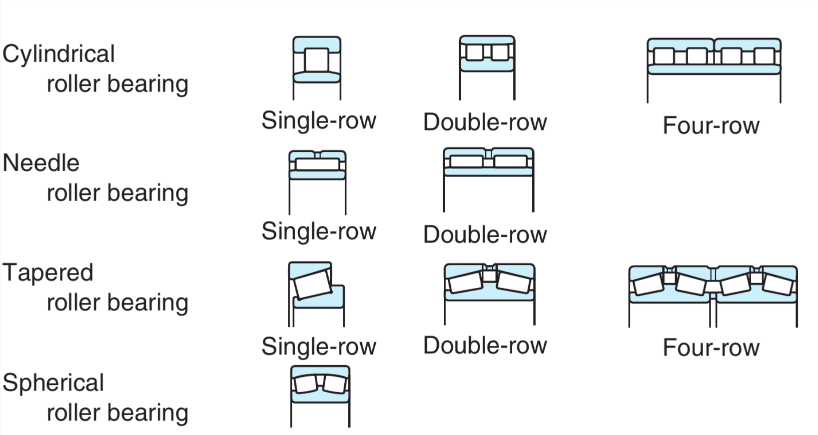


Figure 11: Radial roller bearing types [6].

Rolling bearing types - thrust ball bearings

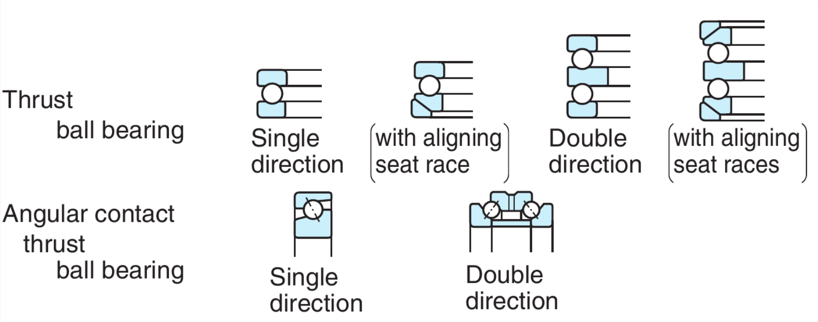


Figure 12: Thrust ball bearing types [6].

Rolling bearing types - thrust roller bearings

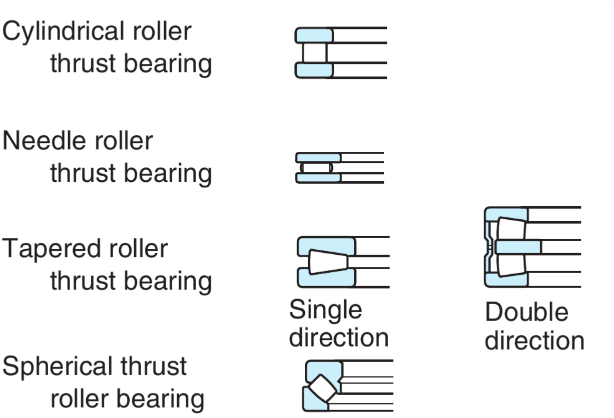


Figure 13: Thrust roller bearing types [6].

Static misalignment

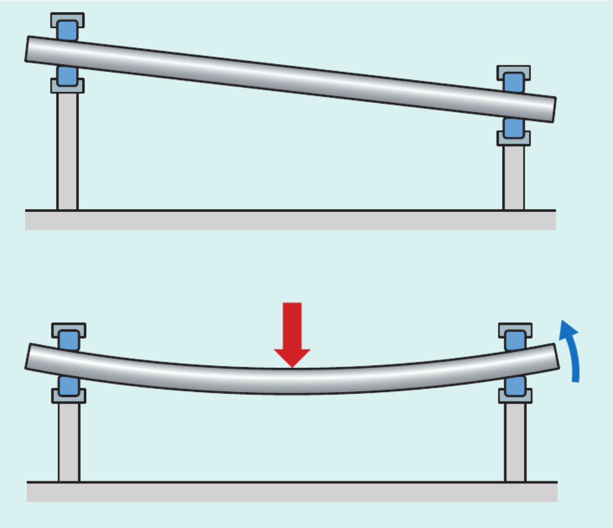


Figure 14: Static misalignment [4].

Dynamic misalignment

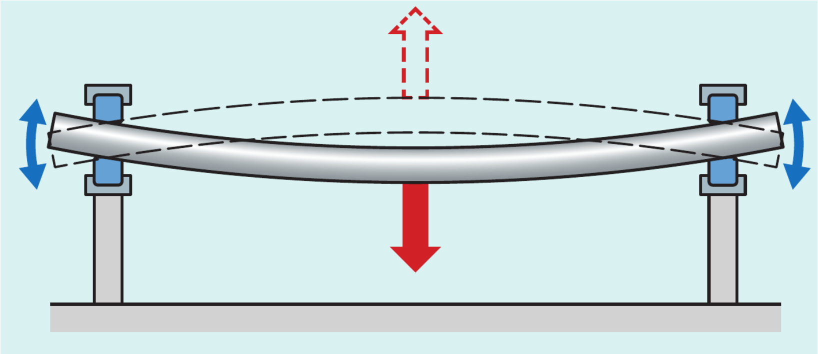


Figure 15: Dynamic misalignment [4].

Varying shaft deflection creates misalignment between bearing inner and outer rings that is continuously changing in magnitude or direction.

Misalignment

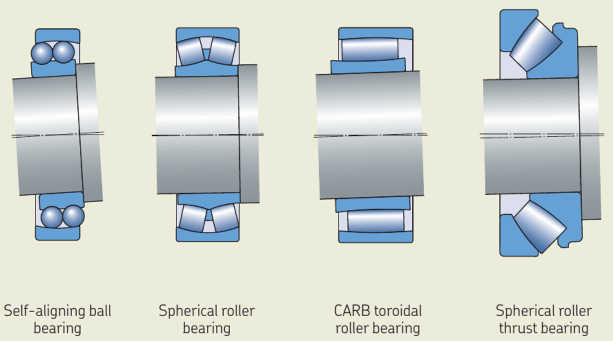


Figure 16: Self-aligning bearings [4].

The normal rolling bearings accept very small misalignment (< 10 minutes), but the self-aligning bearings accept several degrees.

Radial internal clearance

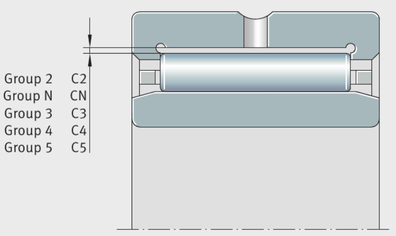


Figure 17: Radial clearance groups [7].

The radial internal clearance applies to bearings with an inner ring and is determined on the unmounted bearing.

The radial internal clearance of a bearing is dependent on the specific bore diameter and the type.

The groups are defined in DIN 620-4 or ISO 5753-1 respectively and are described in DIN 620-4 by means of symbols comprising the letter C and a number.

ISO 5753-1 designates the groups by means of “Group” and a number [7].

Radial internal clearance

Internal clearance group		Description
DIN 620-4	ISO 5753-1	
CN	Group N [1]	Normal radial internal clearance
C2	Group 2	Internal clearance $<$ Group N
C3	Group 3	Internal clearance $>$ Group N
C4	Group 4	Internal clearance $>$ Group 3
C5	Group 5	Internal clearance $>$ Group 4

- CN - normal operating conditions with shaft and housing tolerances;
- C2 - heavy alternating loads combined with swivel motion;
- C3, C4, C5 - bearing rings with press fits and large temperature differential between the inner and outer ring.

[1] Group N is not included in bearing designations

Lecture 2

Summary

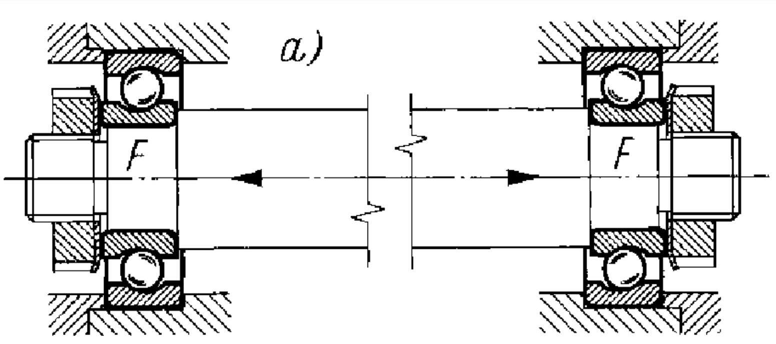


Figure 18: Locating – Locating (fixo – fixo) [5].

Arrangements

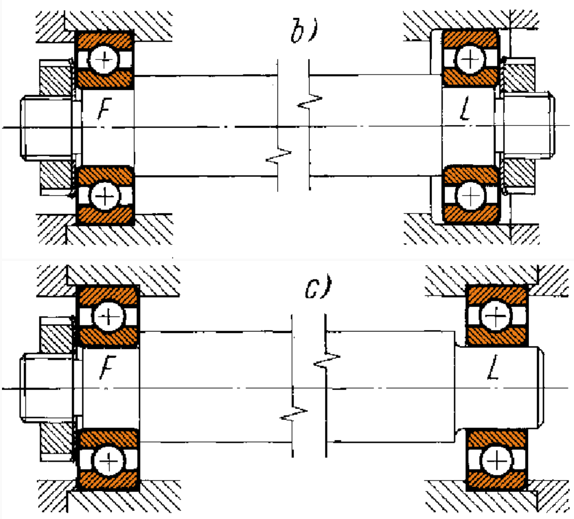


Figure 19: Locating – Non-locating (fixo – livre) [5].

Arrangements

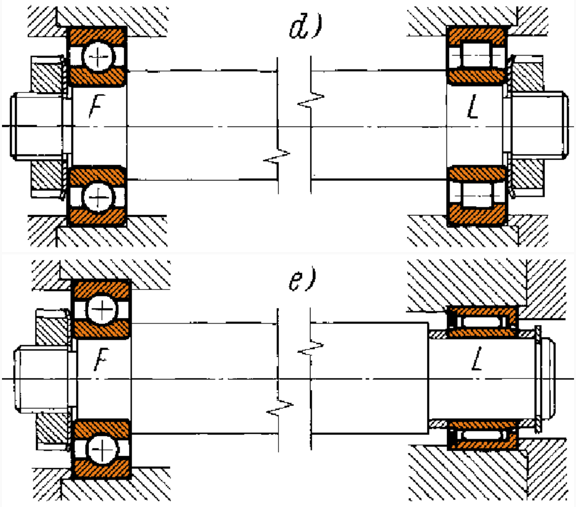


Figure 20: Locating – Non-locating (fixo – livre) [5].

Arrangements

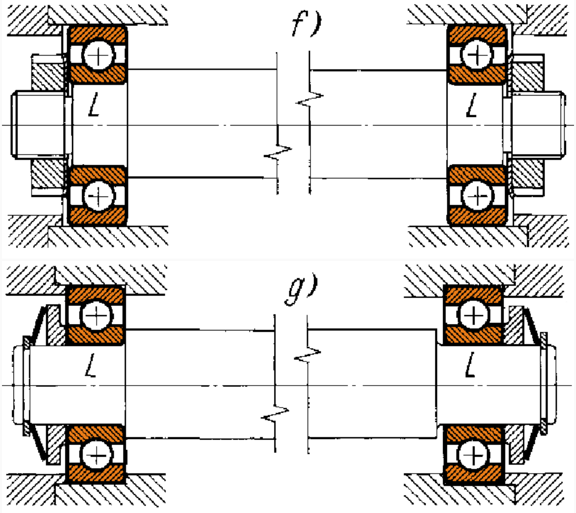


Figure 21: Non-locating – Non-locating (livre – livre) [5].

Arrangements

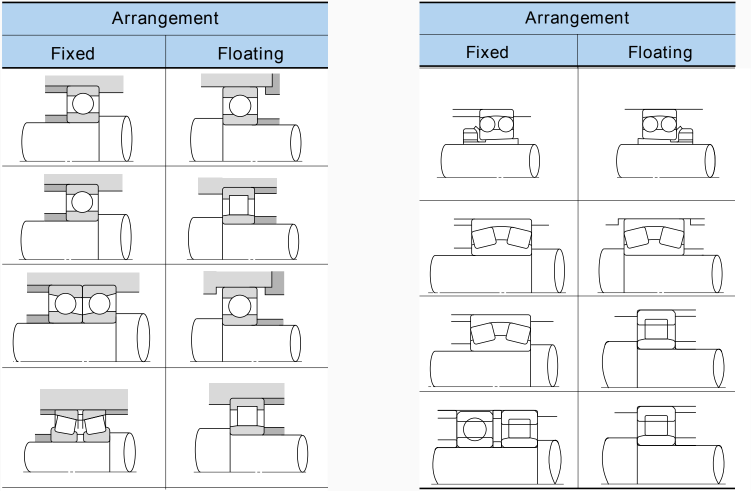
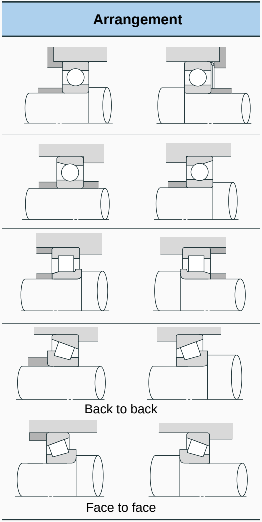


Figure 22: Bearing arrangements.

Arrangements



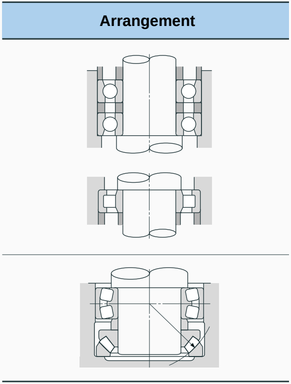


Figure 23: Bearing arrangements.

Recommended bearing fits

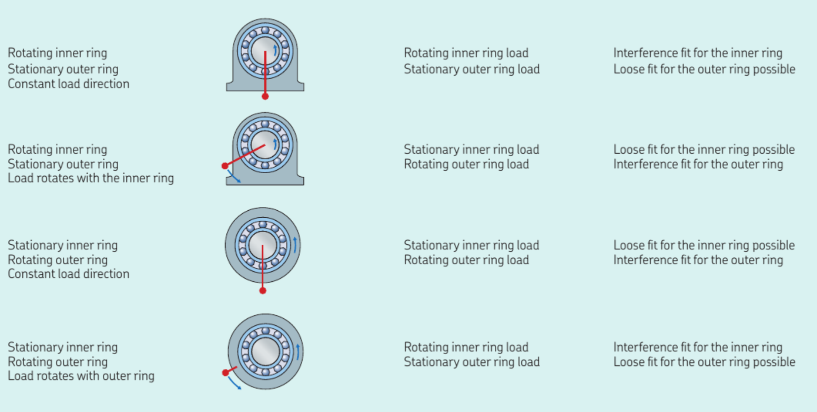


Figure 24: Recommended bearing fits [4].

Static load safety factor

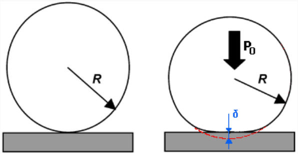


Figure 25: Displacement due to Hertzian contact: P_0 is a static load.

If a rolling bearing operates with infrequent or without rotary motion, its size is determined with the basic load rating C_0. The basic static load rating C_0 causes a permanent deformation at the contact point of about 0.01% of bearing diameter. Condition to avoid *brinelling*.

Hertz pressure under C_0 loading:

- roller bearings: 4000 MPa
- ball bearings: 4200 MPa

Static load safety factor

$$\boxed{S_o = \frac{C_o}{P_o}}$$

C_o - Basic static load ration (given in Tables) / N

P_o - Equivalent static load / N

The greatest value from: $P_o = X_o \cdot F_{or} + Y_o \cdot F_{oa}$ or $P_o = F_{or}$

F_{or} - radial static load / N
F_{oa} - axial static load / N
X_o - radial factor (tables)
Y_o - axial factor (tables)
See ISO 76: 1987, Rolling bearings – Static load ratings [8]

Static load safety factor

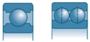
Type	Cross-section	Series	Contact angle	X_0	Y_0
Single- or double-row radial contact ball bearings		60-62-63-64 160-618-619-622 623 42-43		0.6	0.5
Single-row angular contact ball bearings		72 - 73 QJ2 - QJ3	40° 35°	0.5 0.5	0.26 0.29
Double-row angular contact ball bearing		32 - 33 32..A - 33..A 52 - 53 32B - 33B	35° 25° 32°	1.0 1.0 1.0	0.58 0.76 0.63
Double-row self-aligning ball bearings		12 - 13 22 - 23 112 - 113		0.5	
Tapered roller bearings		302 - 303 - 313 320 - 322 - 322..B 323 - 323..B - 330 331 - 332		1.0	See list of Standard Bearings
Double-row spherical roller bearings		213 - 222 - 223 230 - 231 - 232 240 - 241		1.0	
Cylindrical roller bearings		N..2 - N..3 - N..4 N..10 N..22 - N..23		1.0	0
Single-direction ball thrust bearings		511 - 512 - 513 514		0	1
Spherical roller thrust bearings		293 - 294		2.7 si Fr / Fa< 0.55	1

Figure 26: Static radial X_0 and static axial Y_0 factors.

Rigidity

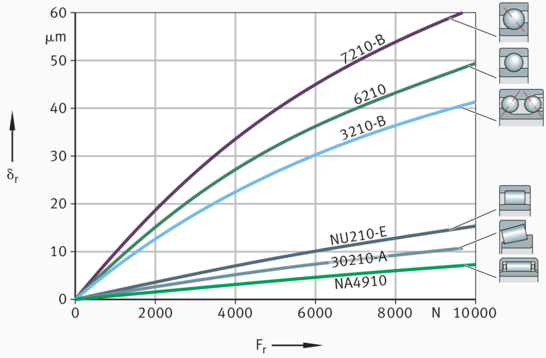


Figure 27: Radial deflection (δ_r) vs. radial load (F_r).

The rigidity is determined by the type, size and operating clearance of the bearing. The rigidity increases with the number of rolling elements supporting the load.

Equivalent dynamic load

The equivalent dynamic load is given by:

$$\boxed{P = X \cdot F_r + Y \cdot F_a}$$

F_r - radial dynamic load / N

F_a - axial dynamic load / N

X - dynamic radial factor (Tables)

Y - dynamic axial factor (Tables)

Equivalent dynamic load

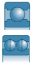
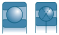
Type	Cross-section	Series	Contact angle	F_a/C_0	e	$F_a/F_r \le e$		$F_a/F_r > e$	
						X	Y	X	Y
Single- or double-row radial contact ball bearings		60-62-63-64 160-618-619 622-623 42-43		0.014 0.028 0.056 0.084 0.110 0.170 0.280 0.420 0.560	0.19 0.22 0.26 0.28 0.30 0.34 0.38 0.42 0.44	1	0	0.56	2.30 1.99 1.71 1.55 1.45 1.31 1.15 1.04 1.00
Single-row radial contact ball bearings, with higher than normal residual clearance		60-62-63-64 160-618-619 622-623		0.014 0.029 0.057 0.086 0.110 0.170 0.280 0.430 0.570	0.29 0.32 0.36 0.38 0.40 0.44 0.49 0.52 0.54	1	0	0.46	1.88 1.71 1.52 1.41 1.34 1.23 1.10 1.01 1.00
Single-row angular contact ball bearings		72-73 QJ2-QJ3	40° 30° 35°		1.14 0,80 0.95	1 1 1	0 0 0	0.35 0.39 0.37	0.57 0.76 0.66
Double-row angular contact ball bearings		32-33 32..A-33..A 52-53 32..B-33..B	35° 25° 32°		0.95 0.68 0.86	1 1 1	0.66 0.92 0.73	0.60 0.67 0.62	1.07 1.41 1.17
Double-row self-aligning ball bearings		12-13 22-23 112-113			see list of Standard bearings	1	see list of Standard bearings	0.65	see list of Standard bearings
Tapered roller bearings		302-303-313 320-322-322..B 323-323..B 330-331-332			see list of Standard bearings	1	0	0.40	see list of Standard bearings
Double-row spherical roller bearings		213-222-223 230-231-232 240-241			see list of Standard bearings	1	see list of Standard bearings	0.67	see list of Standard bearings
Cylindrical roller bearings		N..2-N..3-N..4 N..10 N..22-N..23			–	1	–	1.00	–
Single- or double-direction ball thrust bearing		511-512-513 514			–	–	–	–	1.00
Spherical roller thrust bearing		293-294			1.82	–	–	1.20	1.00

Figure 28: Dynamic radial *X* and dynamic axial *Y* factors.

Rating life

$$\boxed{L_{10} = \left(\frac{C}{P}\right)^{p} \cdot 10^{6}} \qquad \boxed{L_{10\ h} = \frac{10^{6}}{60 \cdot n} \left(\frac{C}{P}\right)^{p}}$$

L_{10} - basic rating life in millions of revolutions is the life reached or exceed by 90% of a largely group of apparently identical bearings before the first evidence of material fatigue

$L_{10\ h}$ - basic rating life in hours

C - Basic dynamic load rating (Tables) / N

P - Equivalent dynamic load / N

n - operating speed / rpm

Bearing type:

- roller bearings: $p = 10/3$
- ball bearings: $p = 3$

Rating life

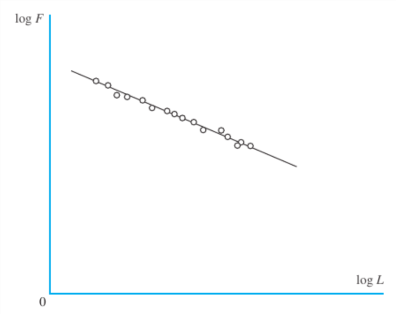


Figure 29: Typical bearing load-life curve [9].

The reliability associated with the points on the plot is 0.90. The load-life function have the following regression equation:

$$FL^{\frac{1}{p}} = C$$

After many experiments, the following exponents are found:

- roller bearings: $p = 10/3$
- ball bearings: $p = 3$

Adjusted rating life

Adjusted rating life obtained by adjustment of the basic rating life for a desired reliability level, special bearing properties and specific operating conditions.

$$L_{10} = \left(\frac{C}{P}\right)^p \cdot 10^6$$

$$L_{na} = a_1 \cdot a_2 \cdot a_3 \cdot L_{10}$$

- a_1 – Life adjustment factor for reliability
- a_2 – Life adjustment factor for special bearing properties
- a_3 – Life adjustment factor for operating conditions

See ISO 281: 1990, Rolling bearings – Dynamic load ratings and rating life [10]

Expanded adjusted rating life

$$\boxed{L_{nm} = a_1 \cdot a_{ISO} \cdot L_{10}}$$

- a_1 – Life adjustment factor for reliability
- a_{ISO} – Life adjustment factor for operating conditions.

The standardized method for calculating the life adjustment factor a_{ISO} adjustment factor essentially takes account of [7, 11]:

- the load on the bearing;
- the lubrication conditions;
- the fatigue limit of the material;
- the type of bearing;
- the residual stress in the material;
- the environmental conditions;
- contamination of the lubricant.

Life adjustment factor for reliability a_1

Reliability:

In the context of bearing life it corresponds to the percentage of bearings operating under the same conditions within a group of apparently identical bearings that are expected to attain or exceed a specified life.

The values of a_1 for a required reliability percentage is given in the following Table.

Reliability %	L_{nm}	a_1
90	$L_{10\ m}$	1
95	$L_{5\ m}$	0.64
96	$L_{4\ m}$	0.55
97	$L_{3\ m}$	0.47
98	$L_{2\ m}$	0.37
99	$L_{1\ m}$	0.25
99.2	$L_{0.8\ m}$	0.22
99.4	$L_{0.6\ m}$	0.19
99.6	$L_{0.4\ m}$	0.16
99.8	$L_{0.2\ m}$	0.12
99.9	$L_{0.1\ m}$	0.093
99.92	$L_{0.08\ m}$	0.087
99.94	$L_{0.06\ m}$	0.080
99.95	$L_{0.05\ m}$	0.077

Life adjustment factor for operating conditions a_{ISO}

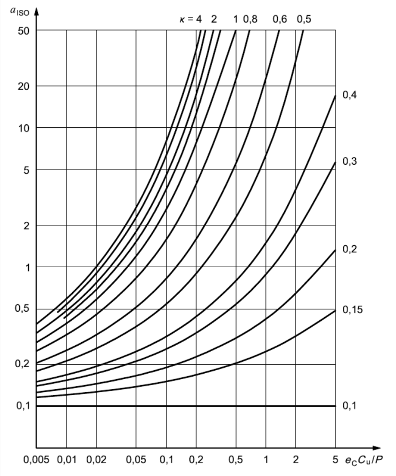


Figure 30: a_{ISO} life adjustment factor for radial ball bearings [11].

The a_{ISO} is function of the following factors:

$$a_{ISO} = f\left(\frac{e_C \cdot C_u}{P}, \kappa\right)$$

Where:

e_C - life adjustment factor for contamination

C_u - fatigue limit load

$\kappa = \frac{\nu}{\nu_1}$ is the viscosity ratio;

The a_{ISO} in the SKF bearing manufacturer catalog [4] is defined as $a_{SKF} = f\left(\frac{\eta_C \cdot P_u}{P}, \kappa\right)$

Contamination factor

Table 1: Contamination factor e_C ($dm = \frac{D+d}{2}$) [11].

Level of contamination	$d_m < 100\,\mathrm{mm}$	$d_m \geq 100\,\mathrm{mm}$
Extreme cleanliness	1	1
High cleanliness	0.8–0.6	0.9–0.8
Normal cleanliness	0.6–0.5	0.8–0.6
Slight contamination	0.5–0.3	0.6–0.4
Typical contamination	0.3–0.1	0.4–0.2
Severe contamination	0.1–0	0.1–0
Very severe contamination	0	0

Rated viscosity

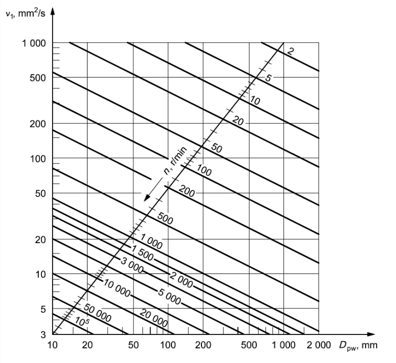


Figure 31: Rated viscosity ν_1 [11].

Viscosity ratio:

$$\kappa = \frac{\nu}{\nu_1}$$

κ is the viscosity ratio;

ν - actual operating viscosity of the lubricant [mm²/s]

ν_1 - rated viscosity, function of the mean bearing diameter and rotational speed [mm²/s]

Rolling bearing calculation tools

SKF

http://www.skf.com/pt/knowledge-centre/engineering-tools/index.html

Schaeffler

http://medias.schaeffler.com/

NTN

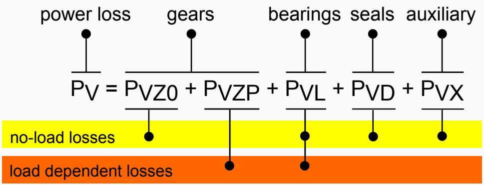
http://www.ntn.co.jp/tool/calc/index.php?lang=en_US

Kisssoft

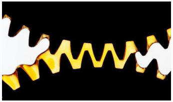
Lecture 3

Summary

Efficiency of gearboxes



Höhn *et al.* [12]

P_{VZ0}	P_{VZP}	P_{VL}	P_{VD}
			
Speed	Nominal power	Nominal power	Speed
Case geometry	Gear geometry	Bearing geometry	Seal diameter
Lubricant viscosity and density	Lubricant formulation	Lubricant formulation	Lubricant viscosity

Sources of bearing losses

The typical sources of power loss in a rolling bearing are:

- Rolling friction: deformation and hysteresis
- Sliding friction
- Drag losses
- Seal losses (if the rolling bearing has seals)

Rolling friction – deformation

The rolling friction losses in a rolling bearing are identified in literature by the following effects: **deformation and elastic hysteresis** [13, 14].

The normal load causes a **deformation** at each contact.

Due to the deformation and due to the rolling motion of the elements over the raceway the material is squeezed up to form a bulge in the forward portion.

A depression is subsequently formed in the rear of the contact area. Thus, an additional tangential load is required to overcome the resisting force of the bulge.

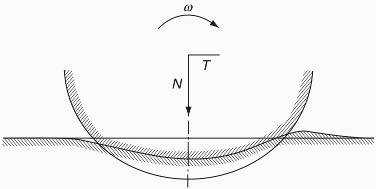


Figure 32: Roller-raceway contact showing bulge due to rolling deformation [14].

Rolling friction - hysteresis

A rolling element under compression travels over a raceway, the material in the forward portion of the contact in the direction of rolling undergoes compression while the material in the rear of the contact is relieved of stress.

It is possible to recognize that as load increases, a given stress corresponds to a smaller deflection than when load is decreasing.

The area between curves is called hysteresis loop, and it represents an energy loss [14].

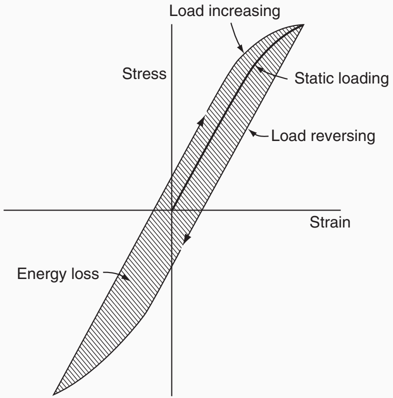


Figure 33: Hysteresis loop for elastic material subjected to reversing stresses [14].

Sliding friction

Sliding is the major source of friction in a rolling bearing, mainly at low speed. The sliding friction occurs due to microslip and sliding [13].

In a bearing without misalignment and moderate speed the total slip of one surface over another would not occur.

The elastic properties and the coefficient of friction between the contacting surfaces may cause microslip and energy is lost.

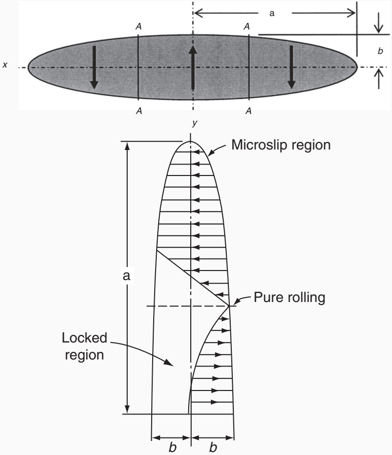


Figure 34: Ball-raceway contact area of a radial bearing with sliding directions presented [14].

As stated by the Elasto-Hydrodynamic Lubrication (EHL) theory [15], the lubricant builds up a film between the raceway and the rolling elements. Usually, from the oil provided to cool and lubricate the contact, only a small portion is used to build up the fluid film.

The excess of oil acts as a friction force contrary to the rotational speed of the rolling elements. The power loss due to drag friction is dependent on the viscosity of the lubricant, speed and oil level.

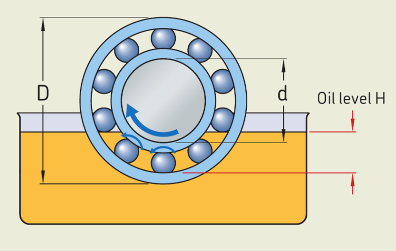


Figure 35: Oil level and drag losses [4].

Seals friction

Seals are used to prevent the bearing to become contaminated with moisture, corrosive media or any other material. Additionally the seal retains the lubricant in the housing.

The contact between the rubber of the seal and the shaft generates friction and must be considered as a source of power loss.

The loss due to seals friction is mainly dependent on the rotational speed.

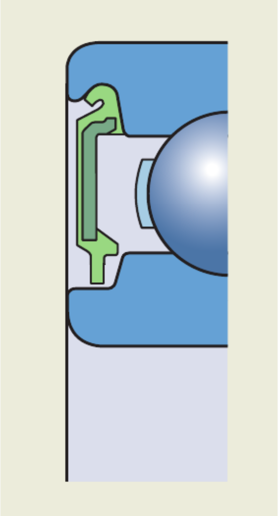


Figure 36: Bearing with a contact seal [4].

Coulomb model

$$\boxed{M_t = \mu \cdot F \cdot \frac{d}{2}}$$

- M_t – total friction torque / N mm
- μ – coefficient of friction
- $F = \sqrt{{F_r}^2 + {F_a}^2}$ – bearing load / N
- F_r – radial load / N
- F_a – axial load / N
- d – bearing bore diameter / mm

Eschmann performed a large number of rolling bearing tests and determined a reference coefficient of friction for the Coulomb model [16].

Manufacturers like NTN [17] and NSK [18] present in their catalogs a suggestion for the coefficient of friction (μ).

Arvrid Palmgren model (1959)

$$\boxed{M_t = M_0 + M_1}$$

- M_0 – no-load bearing friction torque / N mm
- M_1 – load-dependent bearing friction torque / N mm
- M_t – total friction torque / N mm

Arvrid Palmgren model (1959)

$$\boxed{M_0 = f_0 \cdot 10^{-7} \cdot (\nu \cdot n)^{2/3} \cdot d_m^3}$$

- ν – kinematic viscosity of the lubricant at the operating temperature / $\mathrm{mm^2/s}$
- d_m – bearing mean (or pitch) diameter / mm
- f_0 – coefficient taking into account the bearing type and lubrication method
- M_0 – no-load bearing friction torque / N mm
- n – bearing speed / rpm

The value of f_0 is suggested by Eschman [16], FAG [19] and some other rolling bearing manufacturers.

Arvrid Palmgren model (1959)

$$\boxed{M_1 = \mu_1 \cdot f_1 \cdot F \cdot \frac{d_m}{2}}$$

$$F = \sqrt{{F_r}^2 + {F_a}^2}$$

- M_1 – load-dependent bearing friction torque / N mm
- μ_1 – coefficient of friction depending on load and bearing type
- d_m – bearing mean (or pitch) diameter / mm
- f_1 – coefficient that considers the direction of load application
- F – bearing load / N
- F_r – radial force / N
- F_a – axial force / N

The values of f_1 and μ_1 are suggested by Eschman [16], FAG [19] and some other rolling bearing manufacturers.

SKF friction torque model (2004)

The SKF model considers the physical sources of power losses and presents equations to predict them.

$$\boxed{M_t = M_{rr}' + M_{sl} + M_{drag} + M_{seals}}$$

- M_{drag} – drag torque loss
- M_{rr}' – rolling torque loss
- M_{sl} – sliding torque loss
- M_{seals} – seals torque loss
- M_t – total friction torque

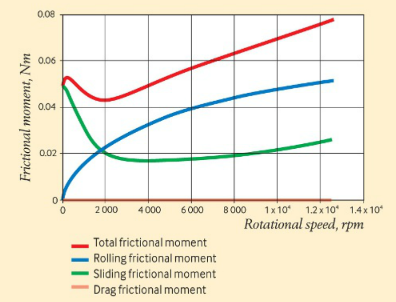


Figure 37: Different rolling bearing friction sources [4].

Rolling torque

$$\boxed{M'_{rr} = \phi_{ish} \cdot \phi_{rs} \cdot \left[G_{rr} \cdot (n \cdot \upsilon)^{0.6}\right]}$$

n is the bearing speed in rpm and ν is the lubricant kinematic viscosity in mm²/s

G_{rr} – Variable depending on the bearing type and load conditions

Inlet shear heating

$$\phi_{ish} = \frac{1}{1+1.84\times10^{-9}\cdot(n\cdot d_m)^{1.28}\cdot\nu^{0.64}}$$

Replenishment/starvation factor

$$\phi_{rs} = \frac{1}{e^{K_{rs}\cdot\nu\cdot n\cdot(d+D)\cdot\sqrt{\frac{K_Z}{2\cdot(D-d)}}}}$$

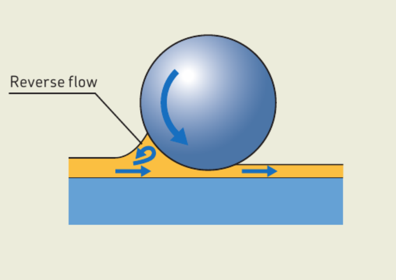


Figure 38: Reverse flow causing inlet shear heating [4].

Sliding torque

$$\boxed{M_{sl} = G_{sl} \cdot \mu_{sl}}$$

G_{sl} – Variable depending on the bearing type and load conditions

μ_{sl} **– Sliding coefficient of friction**

$$\mu_{sl} = \phi_{bl} \cdot \mu_{bl} + (1 - \phi_{bl}) \cdot \mu_{EHL}$$

μ_{bl} **– Boundary conditions coefficient:**

- 0.12 for $n \neq 0$
- 0.15 for $n = 0$

μ_{EHL} **– Full-film conditions coefficient:**

Typical values:

- 0.02 for cylindrical roller bearings
- 0.002 for tapered roller bearings

Other bearings:

- 0.05 for lubrication with mineral oils
- 0.04 for lubrication with synthetic oils

ϕ_{bl} **– Weighting lubrication factor**

Sliding torque

$$\phi_{bl} = \frac{1}{e^{2.6\times10^{-8}(n\cdot\nu)^{1.4}\cdot d_m}}$$

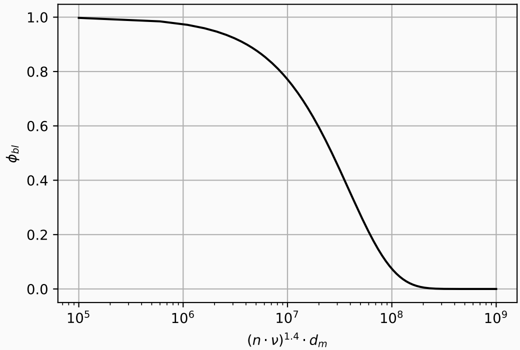


Figure 39: Weighting lubrication factor.

Lecture 4

Summary

When a bearing does fail prematurely, it usually is due to causes that could have been avoided.

It is important to identify the cause of a defect by means of studying its appearance. It allows to correct the causes and prevent future failures and the costs that follow.

Most bearing failures are usually attributed to a relatively small group of causes that are often interrelated and correctable.

These causes include lubrication, mounting, operational stress, bearing selection and environmental influence [20].

Figure 40: Rolling bearing failure.

How to avoid rolling bearing failures

- Use proper lubrication: type and amount of lubricant;
- Check misalignment or shaft deflection;
- Mantain the proper radial internal clearance: the press fit and thermal expansion under operation decreases the clearance;
- Assure a bearing temperature below 100 °C
- Avoid contamination and corrosive environments (sealing can be a solution) [20].

Causes of failure in rolling bearings

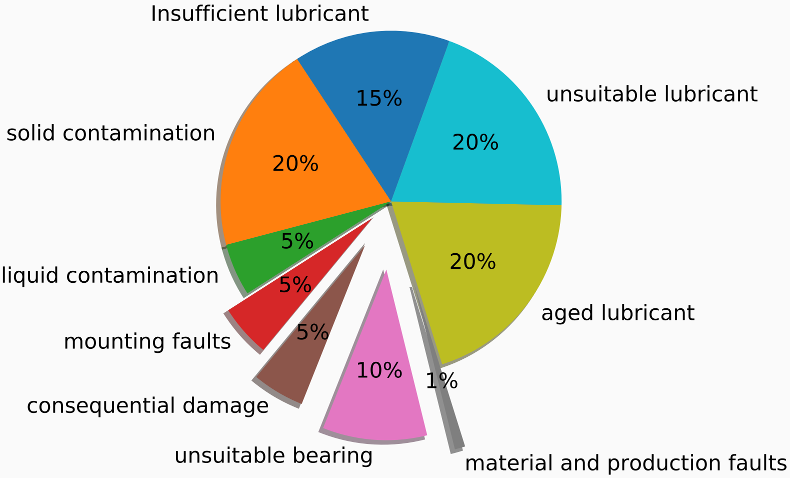


Figure 41: Causes of failure in rolling bearings [20].

Main failures in rolling bearings

Improper fits:

- Thight fits
- Overheating
- Fretting
- Outer ring fracture
- Misalignment

Improper lubrication:

- Slippage tracks
- Axial cracks
- Lubricant failure
- Seizure

Loading conditions

- Normal fatigue
- Excessive loads
- Lip fractures
- False Brinelling
- True Brinelling

Contamination

Corrosion

Electrical current:

- Fluting

Tight fits

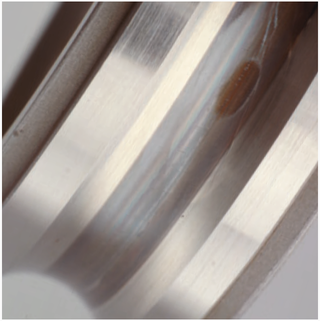


Figure 42: Tight fits [21].

Symptoms

A heavy rolling element wear path in the bottom of the raceway.

Overheating and inner ring axial crack can occur.

Causes

Excessive loading of the rolling elements when interference fits exceed the radial clearance at operating temperatures.

Remedies

Decrease total interference with better matching of bearings to shafts and housings, taking into consideration operating temperatures. Increased radial clearance in bearing selection.

Overheating

Figure 43: Overheating [21].

Symptoms

Discoloration of the rings, rolling elements and cages from gold to blue. Temperatures in excess of 200 °C can anneal ring and rolling element materials, reducing the bearing capacity and causing early failure.

Causes

Tight fits, insufficient clearance; heavy electrical heat loads and insufficient cooling or lubrication when loads and speed are excessive.

Remedies

Thermal or overload controls, adequate heat paths, and supplemental cooling.

Fretting

Figure 44: Fretting [21].

Symptoms

Fretting, the generation of fine metal particles which oxidize, leaving a distinctive brown color. Wear at the fitting surfaces causing noise and runout problems, possible fatigue fracture, and possible disturbance of floating bearing function.

Causes

Micromotion between fitted parts where the fits are too loose in relation to the acting forces.

Remedies

Follow bearing manufacturer mounting instructions for appropriate fit recommendations.

Outer ring fracture

Figure 45: Outer ring fracture [21].

Symptoms

Normally a crack spreads evenly in the circumferential direction, with several fractured pieces. With axial load, these fractures occur as a rule a little beyond the middle of the raceway. The outer ring outside surface normally shows an irregular load carrying pattern.

Causes

Poor support of the rings in the housing.

Remedies

Improvement in the bearing mounting.

Misalignment

Figure 46: Misalignment [21].

Symptoms

A wear path not parallel to raceway edges on the raceway of the non-rotating ring.

Causes

Bent shafts, burrs or dirt on shaft or housing shoulders, shaft threads that are not square with shaft seats, or locking nuts with faces that are not square to the thread axis.

Remedies

Inspect shafts and housings for runout of shoulders and bearing seats; use single point-turned or ground threads on non-hardened shafts and ground threads only on hardened shafts; use precision grade locknuts.

Slippage tracks

Figure 47: Slippage tracks [21].

Symptoms

Spotted smear marks, or roughening of rolling elements or raceways.

Causes

Rolling elements slide on the raceways when the load is low and lubrication is poor. Fast changes in speed.

Remedies

Select bearings with lower load carrying capacity; preload bearings; reduce bearing clearance; improve lubrication.

Axial cracks

Figure 48: Axial cracks [21].

Symptoms

Inner ring cracked in the axial direction. Slightly rounded fractured edges indicate that fracture originated during operation and was cyclic.

Causes

Bearing slippage; rotation of inner ring on shaft; inadequate lubrication; too tight of fit to shaft; grooved shaft; out-of-roundness; grazing against surrounding parts.

Remedies

Improve lubrication with additives or increased oil quantities, reduce water content in oil; select suitable fit; avoid grazing; provide for better seating conditions; consider special heat treatment for rings.

Lubricant Failure

Figure 49: Lubricant failure [21].

Symptoms

Discolored rolling elements (blue/brown) and rolling element tracks. Excessive wear of rolling elements, rings and cages resulting in overheating and catastrophic failure.

Causes

Restricted lubricant flow, or excessive temperatures that degrade the lubricant.

Remedies

Use of the appropriate and correct amount of lubricant, avoid grease loss, and follow appropriate relubrication intervals; ensure proper bearing fit and control preload to reduce bearing temperatures.

Seizure

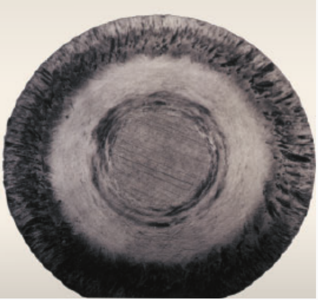


Figure 50: Seizure [21].

Symptoms

Partial or large-area welding and deep scratches in the lip and roller face areas. Also lubricant coking in this area.

Causes

Inadequate lubrication with high loads and high speeds (quantity or operating viscosity of lubricant too low); inadequate lubrication with high loads and low speeds (when there is no hydrodynamic lubricating film between the roller face and lip).

Remedies

Improve lubrication (increase viscosity, EP additives, increase quantity) and ensure correct adjustment of bearings.

Figure 51: Normal fatigue [21].

Symptoms

Known as spalling and indicated by the fracture of the running surfaces and subsequent removal of small discrete particles of material from the inner ring, outer ring or rolling elements.

Causes

Overloading; excessive preload; tight inner ring fits; bearing has remained in operation beyond its calculated fatigue life.

Remedies

Replace the bearing and/or consider redesigning to use a bearing with a greater calculated fatigue life.

Excessive Loads

Figure 52: Excessive loads [21].

Symptoms

Heavy rolling element wear paths, evidence of overheating, and widespread fatigue areas (spalling).

Causes

Excessive loading of the bearing.

Remedies

Reduce the load, or redesign using a bearing with greater capacity.

Lip Fractures

Figure 53: Lip fractures [21].

Symptoms

Supporting lips are partly or completely broken off or cracked.

Causes

Axial load unacceptably high; lip insufficiently supported; axial shock load; mounting damage.

Remedies

Ensure good lip support design, keep load within specified limits, and observe appropriate mounting instructions and procedures.

False brinelling

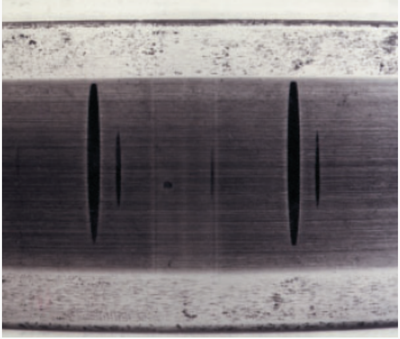


Figure 54: False brinelling [21].

Symptoms

Linear wear marks in axial direction at the rolling element pitch, no raised edges as opposed to marks due to incorrect mounting.

Causes

Vibrations in stationary machines leading to micromotion between rolling elements and raceways. Without motion an oil film cannot be formed to prevent raceway wear.

Remedies

Eliminate or absorb external vibration, and use lubricants containing anti-wear additives.

True Brinelling

Figure 55: True brinelling [21].

Symptoms

Brinell marks appear as indentations in the raceways, increasing bearing vibration (noise). Severe brinell marks can cause premature fatigue failure.

Causes

Static overload of the bearing or severe impact to the bearing, such as using a hammer to install the bearing.

Remedies

Observe static load ratings of the bearing and install bearings using appropriate equipment and by applying force only to the ring being press-fitted.

Contamination

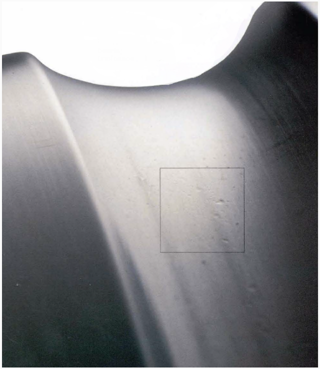


Figure 56: Contamination [21].

Symptoms

Denting of rolling elements and raceways, causing vibration.

Causes

Air-born dust, dirt or abrasive substances from contaminated work areas, dirty hands or tools, foreign matter in lubricants or cleaning solutions.

Remedies

Filtration of the lubricant and clean work areas. For contaminated operating environments, sealing arrangements should be considered.

Corrosion

Figure 57: Corrosion [21].

Symptoms

Red/brown stains or deposits on rolling elements, raceways or cages. Increased vibration followed by wear, increase in radial clearance or loss of preload.

Causes

Exposing bearings to corrosive fluids or atmospheres; formation of condensation caused by temperature changes.

Remedies

Divert corrosive fluids away from bearing areas. Use integrally sealed bearings, and consider external seals for particularly hostile environments.

Fluting

Figure 58: Fluting [21].

Symptoms

Brownish marks parallel to the axis on a large part of the raceway, or covering the entire raceway circumference.

Causes

Constant passage of alternating or direct current; even low currents.

Remedies

Prevent currents from flowing through the bearing by means of grounding or insulating, or use current insulated bearings.

Problems

Summary

Assignments

Assignments A7 and A8

Available on the Course Contents:

“Assignments” proposed for Complements of Machine Elements

Rating life of a rolling bearing

Consider a deep groove ball bearing (6206-C in Table below) with normal radial clearance.

The bearing has a radial load $F_r = 2000\,\mathrm{N}$ and an axial load $F_a = 1000\,\mathrm{N}$. The rolling bearing rotates at $n = 3000\,\mathrm{rpm}$ and the lubricant has a kinematic viscosity at the operating temperature $\nu = 20\,\mathrm{mm^2/s}$.

Determine, in order to assure a reliability of 90% considering a condition of high cleanliness:

1. Static load safety factor (assuming the bearing is not rotating);
2. The basic rating life L_{10};
3. The expanded adjusted rating life L_{nm}.

Rating life of a rolling bearing

Main dimensions			Basic load ratings		Fatigue limit load	Limiting speed	Speed rating	Factor	Mass	Designation
d	D	B	dyn. C_r	stat. C_{0r}	C_{ur}	n_G	$n_{\vartheta r}$	f_0	m	➤225 | 1.12 ➤226 | 1.13
			N	N	N	min^{-1}	min^{-1}		≈ kg	
30	42	7	4 500	2 950	149	24 500	11 500	14,2	0,027	**61806**
	42	7	4 500	2 950	149	8 100	–	14,2	0,027	**61806-2RSR**
	42	7	4 500	2 950	149	20 800	11 500	14,2	0,027	**61806-2Z**
	47	9	7 700	5 000	310	21 700	11 900	15,7	0,051	**61906**
	47	9	7 700	5 000	310	7 600	–	15,7	0,053	**61906-2RSR**
	47	9	7 700	5 000	310	18 400	11 900	15,7	0,053	**61906-2Z**
	55	13	13 500	8 000	390	18 500	13 100	14,8	0,122	**6006**
	55	9	11 900	7 300	370	18 900	10 500	15,1	0,082	**16006**
	55	13	13 500	8 000	390	7 000	–	14,8	0,126	**6006-2RSR**
	55	13	13 500	8 000	390	15 700	13 100	14,8	0,126	**6006-2Z**
	62	16	20 800	11 300	700	17 800	13 400	13,8	0,195	**6206-C**
	62	16	20 800	11 300	700	15 100	13 400	13,8	0,201	**6206-C-2BRS**
	62	16	20 800	11 300	700	10 600	–	13,8	0,201	**6206-C-2HRS**
	62	16	20 800	11 300	700	15 100	13 400	13,8	0,201	**6206-C-2Z**
	62	20	20 700	11 300	570	6 700	–	13,8	0,243	**62206-2RSR**
	72	27	30 000	15 800	1 060	6 000	–	13	0,486	**62306-2RSR**
	72	19	32 000	16 200	1 090	15 100	11 500	13	0,328	**6306-C**
	72	19	32 000	16 200	1 090	12 800	11 500	13	0,339	**6306-C-2BRS**
	72	19	32 000	16 200	1 090	8 900	–	13	0,34	**6306-C-2HRS**
	72	19	32 000	16 200	1 090	12 800	11 500	13	0,339	**6306-C-2Z**
	90	23	45 500	25 000	1 640	10 800	8 600	13	0,74	**6406**

Figure 59: Deep groove ball bearing data [7].

Friction torque of a rolling bearing

Consider a deep groove ball bearing (6206-C in Table below) with normal radial clearance.

The bearing has a radial load $F_r = 2000\,\text{N}$ and an axial load $F_a = 1000\,\text{N}$. The rolling bearing rotates at $n = 3000\,\text{rpm}$ and the mineral lubricant has a kinematic viscosity at the operating temperature $\nu = 20\,\text{mm}^2/\text{s}$.

Determine the total friction torque of the rolling bearing using the:

1. Coloumb model;
2. Arvrid Palmgren model considering for $f_0 = 1.75$, $f_1 = 1.45$ and $\mu_1 = 0.002\left(\frac{F}{C_0}\right)^{\frac{1}{2}}$;
3. SKF model (disregard drag losses).

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