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- Aero and Vibroacoustics of Automotive Turbochargers. Springer Berlin-Heidelberg (2013)
- Tensor Analysis and Elementary Differential Geometry for Physicists and Engineers. Springer Berlin-Heidelberg (2014)
- Rotordynamics of Automotive Turbochargers, Second Edition. Springer Berlin-Heidelberg (2015).

Preface

Rolling element bearings play a key role for working activity and mobility in our everyday life. They are mostly applied to computer and semiconductor industries, electronics industry, chemical and pharmaceutical industries, wind turbines, airplanes, steel industry, automotive industry, and household appliances, for which over 50 billion rolling bearings have been produced across the world. Without rolling bearings, our everyday activities stop working and the world mobility stops moving forever.

At first, rolling bearings look very simple and easy things. They only contain a few balls or rollers lubricated by grease or pure oil that rotate on the inner and outer raceways. The legitimate question is why the research and development of rolling bearings have been still done for many decades worldwide. In fact, bearing development encompasses vast, difficult interdisciplinary fields, such as elastohydrodynamics (EHD) for oil-film thickness and contact stress at the Hertzian contact zone, tribology of surface textures, bearing failure mechanisms, fatigue lifetimes of bearings and greases based on the Weibull distribution, rotor balancing, induced bearing airborne noises (NVH), low wear, and less-noise operating conditions at high rotor speeds.

This book deals with the computational design of rolling bearings that is based on the above working fields in nine concentrated chapters. The readers will learn and understand how the interdisciplinary working fields mutually work in the design of rolling bearings for automotive industry and many other industries. Furthermore, bearing design is partially based on DIN ISO standards, in which some semi-empirical formulas are taken into account in the computations of rolling bearings using MATLAB®.

I would appreciate the managing directors of EM-motive GmbH Mr. Volker Hansen and Dr. Axel Humpert for giving me the opportunity of writing this book for practicing engineers. Furthermore, I would like to thank my colleagues Dr. Zhenhuan Wu at Robert Bosch GmbH and Mr. Andreas Poy at EM-motive for their helpful discussions about rolling bearings and NVH, respectively.

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Finally, my special thanks go to my wife for her understanding, patience, and endless support as I wrote this book in my leisure time during weekends and vacations.

Ludwigsburg, Germany Hung Nguyen-Schäfer

Contents

1 Fundamentals of Rolling Element Bearings 1
- 1.1 Bearing Types 1
- 1.2 Applications of Bearings 2
- 1.3 Bearing Geometry 3
- 1.4 Curvatures of Bearings 10
- 1.5 Bearing Speeds 16
- References 16

2 Design of Rolling Bearings 19
- 2.1 Design Rule of Rolling Bearings 19
- 2.2 Computing Loads Acting upon Bearings 21
  - 2.2.1 Two-Bearing Rigid Rotors 21
  - 2.2.2 Three-Bearing Flexible Rotors 22
- 2.3 Basic Static Load Rating 24
- 2.4 Basic Dynamic Load Rating 27
- 2.5 Dynamic Equivalent Radial Load on Bearings 28
- 2.6 Load Distribution on Balls Under Dynamic Equivalent Load 30
- 2.7 Operating Contact Angle Under Thrust Load 36
- 2.8 Load Distribution on Balls Under Combined Loads 39
- References 46

3 Contact Stresses in Rolling Bearings 47
- 3.1 Hertzian Contact Zone 47
- 3.2 Procedure of Computing the Hertzian Pressure 48
- 3.3 Case Study of Computing the Hertzian Pressure 54
- 3.4 Subsurface Stress in the Hertzian Contact Zone 55
- 3.5 Influenced Parameters on the Hertzian Pressure 58
- References 61

4 Oil-Film Thickness in Rolling Bearings 63
4.1 Introduction 63
4.2 Hydrodynamic and Elastohydrodynamic Lubrications 63
4.3 Oil-Film Pressures in the Hertzian Contact Area 67
4.4 Computing the Oil-Film Thickness 71
4.4.1 Governing Equations of the Oil-Film Thickness 75
4.4.2 The Oil-Film Thickness in Ball Bearings 76
4.4.3 The Oil-Film Thickness in Roller Bearings 77
4.4.4 The Oil-Film Pressure Spike in Roller Bearings 78
4.5 Influence of the Oil-Film Thickness on Wear Mechanism 78
4.6 Case Study of Computing the Oil-Film Thickness 80
References 82

5 Tribology of Rolling Bearings 83
5.1 Introduction 83
5.2 Characteristics of Lubricating Oils 83
5.3 Grease Lubrication in Rolling Element Bearings 84
5.4 HTHS Viscosity of Lubricating Oils 87
5.5 Viscosity Index of Lubricating Oils 91
5.6 Stribeck Curve 93
5.7 Surface Texture Parameters 96
5.7.1 Surface Height Profile 96
5.7.2 Surface Tribological Parameters 98
5.8 Elastic and Plastic Deformations in the Bearings 106
5.8.1 Normal Stress 107
5.8.2 Shear Stress 108
5.8.3 Friction Force in the Bearings 109
5.8.4 Friction Power in the Bearings 111
5.8.5 Mohr's Circle Diagram 113
References 115

6 Lifetimes of Rolling Bearings 117
6.1 Introduction 117
6.2 Fatigue Lifetime of Rolling Bearings 117
6.2.1 Extended Fatigue Lifetime 117
6.2.2 Fatigue Lifetime at Point Contact of Rolling Elements 129
6.2.3 Fatigue Lifetime at Line Contact of Rolling Elements 130
6.3 Lifetime Factor 132
6.4 Grease Lifetime in Bearings 134
6.5 Bleeding Time of Grease in Bearings 135
6.6 Case Study of Computing Lifetimes of Bearings 136
References 139

7 Reliability Using the Weibull Distribution 141
7.1 Introduction 141
7.2 Weibull Distribution 141
7.3 Probability of Survival Samples 144
7.4 Probability Density Function 146
7.5 Time Interval Between Two Failures 148
7.6 Mean Lifetime, Variance, and Median Value 150
7.7 Percentile Lifetime 154
7.8 Estimating the Parameters of β and η 157
7.8.1 Weibull Plot (WP) 157
7.8.2 Computational Method of Maximum Likelihood (ML) . . . 160
7.9 Prediction of the System Lifetime 161
7.9.1 Proof of Eq. 7.33 162
7.9.2 A Computational Example 163
7.10 Hazard Rate Functions 164
7.11 Weibull Regression 166
7.12 The Monte Carlo Simulation Method 168
References 170

8 Bearing Friction and Failure Mechanisms 171
8.1 Friction in Rolling Bearings 171
8.2 Failure Mechanisms in Rolling Bearings 173
8.2.1 Initiated Surface Microcracks 177
8.2.2 Initiated Subsurface Microcracks 179
8.2.3 False Brinelling 181
8.2.4 Surface Distress 183
References 184

9 Rotor Balancing and NVH in Rolling Bearings 185
9.1 Reasons for Rotor Balancing 185
9.2 Kinds of Rotor Balancing 185
9.3 Two-Plane Low-Speed Balancing of a Rigid Rotor 186
9.4 Bearing Noises 193
9.4.1 Excitation Frequencies 193
9.4.2 Induced Noises in Bearing Components 197
9.4.3 Bearing Defect-Related Frequencies 201
9.5 Structure-Borne and Airborne Noise 203
References 205

Appendix A: Normal Probability Density Function and Cumulative Distribution Function 207

Appendix B: Maximum Likelihood Method 211

Appendix C: Simpson's Rule 215

Appendix D: Kinematics of Rolling Bearings 223

Appendix E: Least Squares Regression 227

Index 233

Chapter 1
Fundamentals of Rolling Element Bearings

The main purpose of this book is not to deal with the rolling element bearings in general as a normal textbook of bearings but to focus on the computational design of bearings, especially ball and roller bearings that are used in automotive industries. Only essential things about the rolling bearings are briefly handled with the motto "the shorter the better." Therefore, some issues of technical constructions for all types of bearings are intentionally not discussed in this book. The readers can find them in other literature, e.g., [1–3]. However, some fundamental characteristics of the rolling bearings are recapitulated and discussed. They are essential for the computational design of bearings.

1.1 Bearing Types

Rolling element bearings can be used as the radial and thrust bearings with single or double row. Radial bearings keep the rotor in balance with external forces acting on it in radial direction. Similarly, thrust bearings keep the rotor stable with thrust forces acting on it in axial direction. Ball and roller bearings are the usual rolling element bearings with different setups, such as deep-groove ball bearings (BB), angular contact BB, cylindrical roller bearings (RB), spherical RB, needle RB, and tapered RB.

In the case of using balls for the rolling elements (RE), the rolling bearing is called the *ball bearing* (BB); in the case of cylindrical rollers, the *roller bearing* (RB). This book mainly deals with the deep-groove ball and cylindrical roller bearings under the combined radial and thrust loads acting upon the bearing (s. Fig. 1.1).

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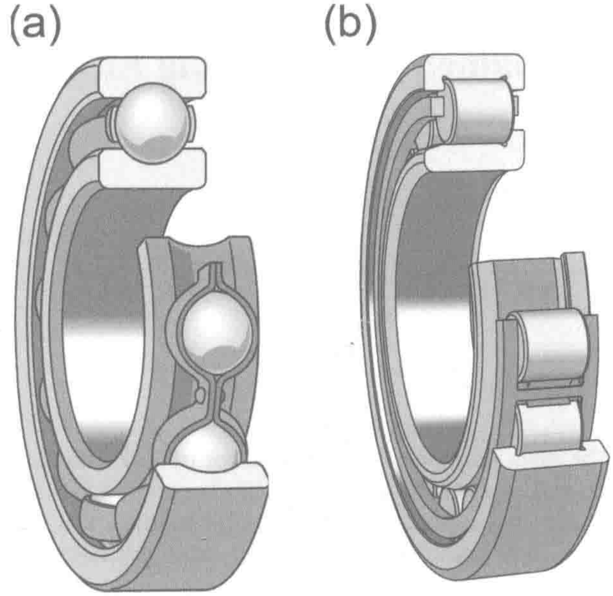


Fig. 1.1 Deep-groove ball and cylindrical roller bearings (Courtesy SKF)

1.2 Applications of Bearings

Ball and roller bearings have some positive characteristics, such as low friction, maintenance-free with greases, oil-free applications, and working under high loads and high-temperature environments. Therefore, they are mostly applied to various industries:

- Computer and semiconductor industries for hard drives, DVD sputtering, and microprocessor producing equipments
- Electronics industry for liquid crystal panel bonding and LC sealing furnace
- Chemical industry for etching equipments and centrifuges
- Wind turbine generator
- Automotive industry for electric motors, turbochargers, and gearbox
- Aeroplane industry for jet engines
- Steel industry for manufacturing machines, furnace cars, etc.
- Household appliances

Note that billion ball and roller bearings have been worldwide produced every year for such applications.

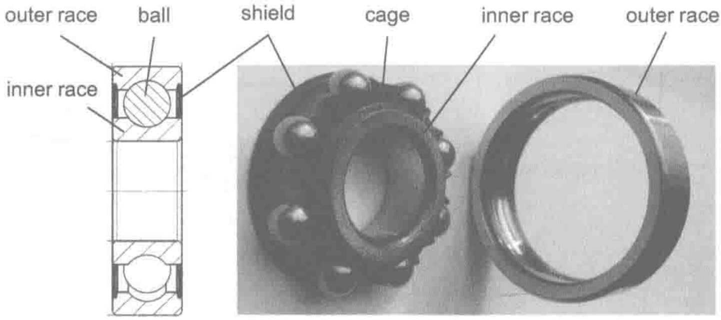


Fig. 1.2 Components of a deep-groove ball bearing

1.3 Bearing Geometry

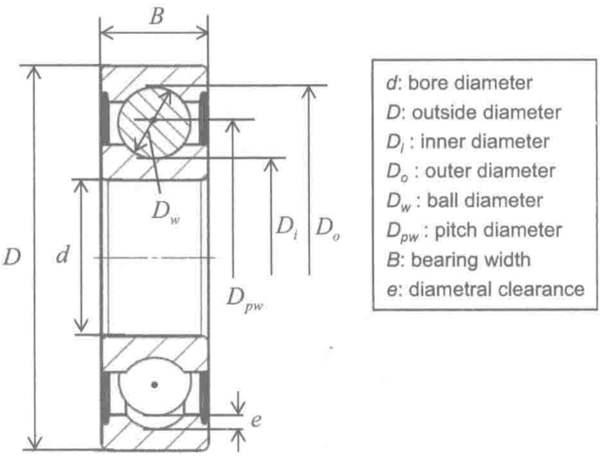
Deep-groove ball bearings consist of many components, as shown in Fig. 1.2. The balls are held in a polyamide cage and are supported by the inner and outer raceways. To keep lubricating grease inside the bearing and to protect the bearing from hard particles and contaminants from outside during the operation, two lip seals and shields are installed at both sides of the bearings.

The bearing is lubricated using grease that is filled between the balls in the inner and outer raceways. Due to rotation of the balls, in-grease dissolved oil is separated from grease in the oil film between the balls and raceways. The EHL pressure (elastohydrodynamic lubrication) of the oil film is created to support the rotor against external forces acting upon it. The oil-film thickness depends on the external forces, oil viscosity, oil temperature, and rotor speed. In the case of EHL, the minimum oil-film thickness in the bearing is in the order of a few hundred nanometers.

Some essential geometries of a ball/roller bearing for the computing design are defined in Fig. 1.3.

- The bore diameter d is defined as the inside diameter of the bearing at which the bearing is mounted in the rotor shaft.
- The outside diameter D is defined as the outside diameter of the bearing at which the bearing is mounted in the bearing housing.
- The inner diameter D_i of the inner raceway is defined as the diameter at which the balls contact the inner raceway.
- The outer diameter D_o of the outer raceway is defined as the diameter at which the balls contact the outer raceway.
- The bearing diametral clearance e is defined as the total gap in diametral direction between the balls and raceways before assembly of the bearing.
- The pitch diameter D_{pw} is defined as the diameter in which the ball centers locate.

Fig. 1.3 Geometry of a deep-groove ball bearing



Using the bearing nomenclature (e.g., type 6305), the bore diameter d (in mm) results from multiplying the last double digit DD by a factor 5 as

$$d = DD \times 5 = 05 \times 5 = 25mm$$

The outside diameter D (in mm) is approximately calculated as [4]

$$D \approx d + f_D d^{0.9}$$

where f_D is the factor that depends on the diameter series of the bearing nomenclature.

For the bearing type 6305, the first digit shows the *bearing type* (ball or roller bearings); the second digit is the *diameter series DS*; i.e., $DS = 3$.

DS:	**7**	**8**	**9**	**0**	**1**	**2**	**3**	**4**
f_D :	0.34	0.45	1.62	1.84	1.12	1.48	1.92	2.56

The bearing width B (in mm) is approximately calculated using the approximate outside diameter D as [4]

$$B = 0.5 f_B (D - d) = 0.5 f_B f_D d^{0.9}$$

where f_B is the factor that depends on the width series of the bearing nomenclature, d (in mm). Ball and roller bearings without seals and plates have the *width series* $WS = 0$:

WS:	**0**	**1**	**2**	**3**	**4**	**5**	**6**	**7**
f_B :	0.64	0.88	1.15	1.5	2.0	2.7	3.6	4.8

However, the outside diameter D and width B of the bearing could be little changed from bearing manufacturers.

The pitch diameter can be approximately calculated from the inner and outer diameters of the raceways as

$$D_{pw} \approx 0.5(D_i + D_o) \tag{1.1}$$

The diametral clearance obviously results from the inner, outer diameters, and ball diameter as

$$e = D_o - D_i - 2D_w. \tag{1.2a}$$

Therefore, the ball diameter results as

$$D_w = 0.5(D_o - D_i - e) \tag{1.2b}$$

In fact, the inner and outer diameters of the raceways are generally not given in the supplier catalogues. However, they can be approximately calculated from the bore and outside diameters that can be found in the catalogues as:

- The approximate diameter of the outer raceway for ball bearings:

$$\overline{D}_o \approx \frac{4}{5}D + \frac{1}{5}d = 0.80D + 0.20d \tag{1.3a}$$

- The approximate diameter of the outer raceway for roller bearings:

$$\overline{D}_o \approx \frac{3}{4}D + \frac{1}{4}d = 0.75D + 0.25d \tag{1.3b}$$

Using Eq. 1.1, the diameter of the inner raceway is calculated as

$$D_i = 2D_{pw} - D_o \tag{1.4}$$

According to NSK, the pitch diameter can be approximately calculated as

$$\overline{D}_{pw} = 1.025 \times \left(\frac{D + d}{2}\right) \approx D_{pw} \tag{1.5}$$

Substituting Eq. 1.5 into Eq. 1.4, one obtains the approximate diameter of the inner raceway:

$$\overline{D}_i \approx 1.025(D + d) - \overline{D}_o \tag{1.6}$$

Equations 1.3a, 1.3b and 1.6 show how to calculate the inner and outer raceway diameters with an acceptable geometrical tolerance of ± 3 %.

As an example of a ball bearing type 6305 with $d = 25$ mm and $D = 62$ mm (cf. bearing catalogue of SKF), the diameter of the outer raceway results from Eq. 1.3a as

$$\overline{D}_o \approx \frac{4D + d}{5} = 54.6\ mm$$

Using Eq. 1.6, the diameter of the inner raceway is given as

$$\overline{D}_i \approx 1.025(D + d) - \overline{D}_o = 34.57\ mm$$

Thus, the approximate pitch diameter results using Eq. 1.1 as

$$\overline{D}_{pw} \approx \frac{\overline{D}_i + \overline{D}_o}{2} = 44.58\ mm$$

The real nominal pitch diameter of 44.60 mm is nearly equal to the calculated one.

Furthermore, the approximate ball diameter is given from Eq. 1.2b with a zero diametral clearance as

$$\begin{aligned} \overline{D}_w &= 0.5\left(\overline{D}_o - \overline{D}_i - e\right) \\ &\approx 0.5 \times (54.6 - 34.57 - 0) \approx 10\ mm \end{aligned}$$

Compared to the real nominal ball diameter of 10.32 mm, the calculated ball diameter is a little smaller with an acceptable tolerance of about 3 %. Note that the ball diameter is never given in the bearing catalogues; and it is not easy to measure the ball diameter in the assembled bearing.

The bearing axial clearance G_a is calculated as [1, 2]

$$\begin{aligned} G_a &= \sqrt{4D_w e(\kappa_i + \kappa_o - 1) - e^2} \\ &= \sqrt{e(4\rho_0 - e)} \end{aligned} \tag{1.7}$$

where κ_i and κ_o are the osculations of the inner and outer raceways, respectively.

The distance ρ_0 between two raceway centers of M_I and M_O in Eq. 1.7 is defined as

$$\rho_0 = (\kappa_i + \kappa_o - 1) \cdot D_w \tag{1.8}$$

According to DIN-ISO 281 [5], the inner and outer osculations of ball bearings are defined as the ratio of the inner and outer raceway radii to the ball diameter (s. Fig. 1.4):

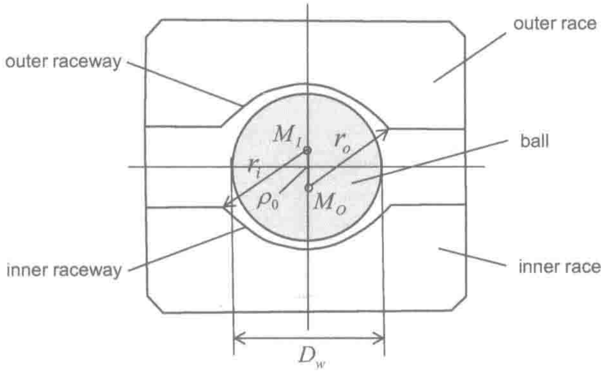


Fig. 1.4 Radii of ball bearings

$$\kappa_i \equiv \frac{r_i}{D_w}; \quad \kappa_o \equiv \frac{r_o}{D_w} \tag{1.9}$$

In fact, the bearing osculation describes the spatial clearance between the balls and raceways. Note that the value of the bearing osculation must be larger than 50 %; otherwise, the balls contact the raceways. Normally in the ball bearings, the inner osculation varies from about 50.6–52 %; the outer osculation varies from 52.7 % to 53 %. The larger the bearing osculation, the more the room is between the balls and raceways or vice versa. The influences of the bearing osculation on the bearing behavior will be discussed in the following section (cf. Table 1.1).

The selection of the bearing osculation depends on the strategy of the bearing suppliers and the manufacturing processes. On the one hand, a large osculation leads to less noise, small bearing friction, and less wear induced in the bearing. However, it increases the Hertzian pressure at the contact area, reduces the oil-film thickness, and decreases both static and dynamic load ratings. These effects cause a reduction of the bearing lifetime. On the other hand, a small osculation increases both static and dynamic load ratings and reduces the Hertzian pressure at the contact area. Both lead to the increase of the bearing lifetime. However, it increases induced noise in the bearing and causes much more bearing friction and more wear at the contact area.

Note that the larger the axial clearance, the more the *false brinelling* (fatigue damage) is caused by high-frequency vibrations at the standstill during the transport. As a result, we have to take a compromise between the effects that depend on customer requirements.

At the assembled bearing, the diametral clearance should be by experience between −20 μm and +5 μm for the operating temperature range, as shown in Fig. 1.5. The lifetime of the bearing increases at a small negative bearing clearance between −10 and −20 μm. However, the lifetime of bearings strongly reduces at a clearance less than −20 μm due to mixed or boundary lubrication of the oil film.