11. DESIGN OF SHAFTS AND HOUSINGS

11.1 Accuracy and Surface Finish of Shafts and Housings

If the accuracy of a shaft or housing does not meet the specification, the performance of the bearings will be affected and they will not provide their full capability. For example, inaccuracy in the squareness of the shaft shoulder may cause misalignment of the bearing inner and outer rings, which may reduce the bearing fatigue life by adding an edge load in addition to the normal load. Cage fracture and seizure sometimes occur for this same reason. Housings should be rigid in order to provide firm bearing support. High rigidity housings are advantageous also from he standpoint of noise, load distribution, etc.

For normal operating conditions, a turned finish or smooth bored finish is sufficient for the fitting surface; however, a ground finish is necessary for applications where vibration and noise must be low or where heavy loads are applied.

In cases where two or more bearings are mounted in one single-piece housing, the fitting surfaces of the housing bore should be designed so both bearing seats may be finished together with one operation such as in -line boring. In the case of split housings, care must be taken in the fabrication of the housing so the outer ring will not become deformed during installation. The accuracy and surface finish of shafts and housings are listed in Table 11.1 for normal operating conditions.

Table 11. 1 Accuracy and Roughness of Shaft and Housing

Item	Class of Bearings	Shaft	Housing Bore
Tolerance for Out-of-roundness	Normal, Class 6	$\frac{IT3}{2}$ to $\frac{IT4}{2}$	$\frac{IT4}{2}$ to $\frac{IT5}{2}$
	Class 5, Class 4	$\frac{IT2}{2}$ to $\frac{IT3}{2}$	$\frac{IT2}{2}$ to $\frac{IT3}{2}$
Tolerance for Cylindricality	Normal, Class 6	$\frac{IT3}{2}$ to $\frac{IT4}{2}$	$\frac{IT4}{2}$ to $\frac{IT5}{2}$
	Class 5, Class 4	$\frac{IT2}{2}$ to $\frac{IT3}{2}$	$\frac{IT2}{2}$ to $\frac{IT3}{2}$
Tolerance for Shoulder Runout	Normal, Class 6	IT3	IT3 to IT4
	Class 5, Class 4	IT3	IT3
Roughness of Fitting Surfaces R_a	Small Bearings	0.8	1.6
	Large Bearings	1.6	3.2

Remarks This table is for general recommendation using radius measuring method, the basic tolerance (IT) class should be selected in accordance with the bearing precision class. Regarding the figures of IT, please refer to the Appendix Table 11 (page C22).
In cases that the outer ring is mounted in the housing bore with interference or that a thin cross-section bearing is mounted on a shaft and housing, the accuracy of the shaft and housing should be higher since this affects the bearing raceway directly.

11.2 Shoulder and Fillet Dimensions

The shoulders of the shaft or housing in contact with the face of a bearing must be perpendicular to the shaft center line. (Refer to Table 11.1) The front face side shoulder bore of the housing for a tapered roller bearing should be parallel with the bearing axis in order to avoid interference with the cage.

The fillets of the shaft and housing should not come in contact with the bearing chamfer; therefore, the fillet radius r_a must be smaller than the minimum bearing chamfer dimension r or r_1.

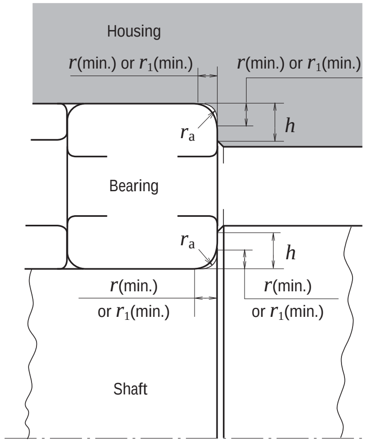

Fig. 11.1 Chamfer Dimensions, Fillet Radius of Shaft and Housing, and Shoulder Height

The shoulder heights for both shafts and housings for radial bearings should be sufficient to provide good support over the face of the bearings, but enough face should extend beyond the shoulder to permit use of special dismounting tools. The recommended minimum shoulder heights for metric series radial bearings are listed in Table 11.2

Nominal dimensions associated with bearing mounting are listed in the bearing tables including the proper shoulder diameters. Sufficient shoulder height is particularly important for supporting the side ribs of tapered roller bearings and cylindrical roller bearings subjected to high axial loads.

The values of h and r_a in Table 11.2 should be adopted in those cases where the fillet radius of the shaft or housing is as shown in Fig. 11.2 (a), while the values in Table 11.3 are generally used with an undercut fillet radius produced when grinding the shaft as shown in Fig. 11.2 (b).

Table 11. 2 Recommended Minimum Shoulder Heights for Use with Metric Series Radial Bearings

Units : mm

Nominal Chamfer Dimensions r (min.) or r_1 (min.)	Shaft or Housing		
	Fillet Radius r_a (max.)	Minimun Shoulder Heights h (min.)	
		Deep Groove Ball Bearings, Self-Aligning Ball Bearings, Cylindrical Roller Bearings, Solid Needle Roller Bearings	Angular Contact Ball Bearings, Tapered Roller Bearings, Spherical Roller Bearings
0.05	0.05	0.2	—
0.08	0.08	0.3	—
0.1	0.1	0.4	—
0.15	0.15	0.6	—
0.2	0.2	0.8	—
0.3	0.3	1	1.25
0.6	0.6	2	2.5
1	1	2.5	3
1.1	1	3.25	3.5
1.5	1.5	4	4.5
2	2	4.5	5
2.1	2	5.5	6
2.5	2	—	6
3	2.5	6.5	7
4	3	8	9
5	4	10	11
6	5	13	14
7.5	6	16	18
9.5	8	20	22
12	10	24	27
15	12	29	32
19	15	38	42

Remarks
1. When heavy axial loads are applied, the shoulder height must be sufficiently higher than the values listed.
2. The fillet radius of the corner is also applicable to thrust bearings.
3. The shoulder diameter is listed instead of shoulder height in the bearing tables.

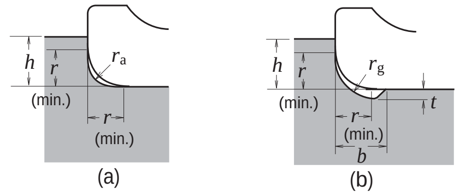

Fig. 11. 2 Chamfer Dimensions, Fillet Radius, and Shoulder Height

Table 11. 3 Shaft Undercut

Units : mm

Chamfer Dimensions of Inner and Outer Rings r (min.) or r_1 (min.)	Undercut Dimensions		
	t	r_g	b
1	0.2	1.3	2
1.1	0.3	1.5	2.4
1.5	0.4	2	3.2
2	0.5	2.5	4
2.1	0.5	2.5	4
2.5	0.5	2.5	4
3	0.5	3	4.7
4	0.5	4	5.9
5	0.6	5	7.4
6	0.6	6	8.6
7.5	0.6	7	10

For thrust bearings, the squareness and contact area of the supporting face for the bearing rings must be adequate. In the case of thrust ball bearings, the housing shoulder diameter D_a should be less than the pitch circle diameter of the balls, and the shaft shoulder diameter d_a should be greater than the pitch circle diameter of the balls (Fig. 11.3).
For thrust roller bearings, it is advisable for the full contact length between rollers and rings to be supported by the shaft and housing shoulder (Fig. 11.4).
These diameters d_a and D_a are listed in the bearing tables.

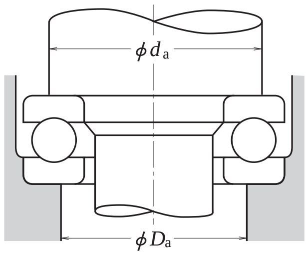

Fig. 11.3 Face Supporting Diameters for Thrust Ball Bearings

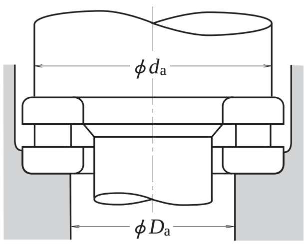

Fig. 11.4 Face Supporting Diameters for Thrust Roller Bearings

11.3 Bearing Seals

To insure the longest possible life of a bearing, it may be necessary to provide seals to prevent leakage of lubricant and entry of dust, water and other harmful material like metallic particles. The seals must be free from excessive running friction and the probability of seizure. They should also be easy to assemble and disassemble. It is necessary to select a suitable seal for each application considering the lubricating method.

11.3.1 Non-Contact Type Seals

Various sealing devices that do not contact the shaft, such as oil grooves, flingers, and labyrinths, are available. Satisfactory sealing can usually be obtained with such seals because of their close running clearance. Centrifugal force may also assist in preventing internal contamination and leakage of the lubricant.

(1) Oil Groove Seals

The effectiveness of oil groove seals is obtained by means of the small gap between the shaft and housing bore and by multiple grooves on either or both of the housing bore and shaft surface (Fig. 11.5 (a), (b)).
Since the use of oil grooves alone is not completely effective, except at low speeds, a flinger or labyrinth type seal is often combined with an oil groove seal (Fig. 11.5 (c)). The entry of dust is impeded by packing grease with a consistency of about 200 into the grooves.
The smaller the gap between the shaft and housing, the greater the sealing effect; however, the shaft and housing must not come in contact while running. The recommended gaps are given in Table 11.4.
The recommended groove width is approximately 3 to 5mm, with a depth of about 4 to 5mm. In the case of sealing methods using grooves only, there should be three or more grooves.

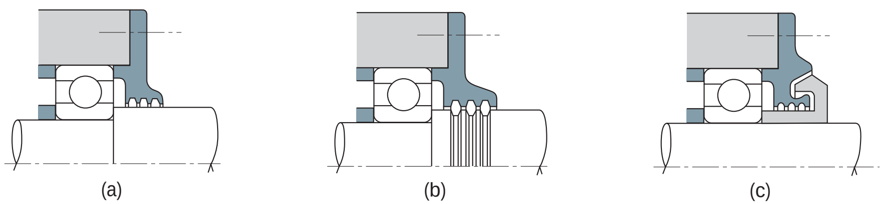

Fig. 11.5 Examples of Oil Grooves

(2) Flinger (Slinger) Type Seals

A flinger is designed to force water and dust away by means of the centrifugal force acting on any contaminants on the shaft. Sealing mechanisms with flingers inside the housing as shown in Fig. 11.6 (a), (b) are mainly intended to prevent oil leakage, and are used in environments with relatively little dust. Dust and moisture are prevented from entering by the centrifugal force of flingers shown in Figs 11.6 (c), (d).

(3) Labyrinth Seals

Labyrinth seals are formed by interdigitated segments attached to the shaft and housing that are separated by a very small gap. They are particularly suitable for preventing oil leakage from the shaft at high speeds.
The type shown in Fig. 11.7 (a) is widely used because of its ease of assembly, but those shown in Fig. 11.7 (b), (c) have better seal effectiveness.

Table 11. 4 Gaps between Shafts and Housings for Oil-Groove Type Seals

Units : mm

Nominal Shaft Diameter	Radial Gap
Under 50	0.25 to 0.4
50-200	0.5 to 1.5

Table 11. 5 Labyrinth Seal Gaps

Units : mm

Nominal Shaft Diameter	Labyrinth Gaps	
	Radial Gap	Axiall Gap
Under 50	0.25 to 0.4	1 to 2
50-200	0.5 to 1.5	2 to 5

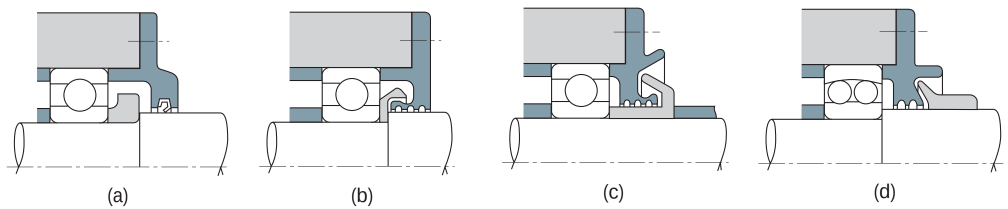

Fig. 11.6 Examples of Flinger Configurations

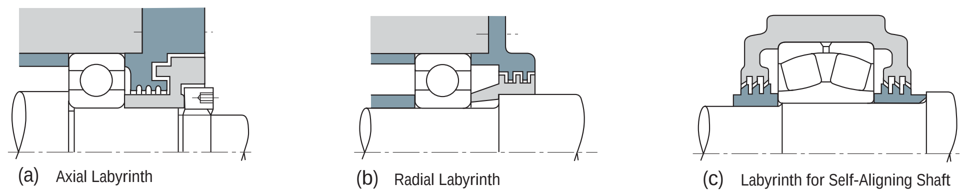

Fig. 11.7 Examples of Labyrinth Designs

11.3.2 Contact Type Seals

The effectiveness of contact seals is achieved by the physical contact between the shaft and seal, which may be made of synthetic rubber, synthetic resin, felt, etc. Oil seals with synthetic rubber lips are most frequently used.

(1) Oil Seals

Many types of oil seals are used to prevent lubricant from leaking out as well as to prevent dust, water, and other foreign matter from entering (Figs. 11.8 and 11.9)
In Japan, such oil seals are standardized (Refer to JIS B 2402) on the basis of type and size. Since many oil seals are equipped with circumferential springs to maintain adequate contact force, oil seals can follow the non-uniform rotational movement of a shaft to some degree.
Seal lip materials are usually synthetic rubber including nitrile, acrylate, silicone, and fluorine. Tetrafluoride ethylene is also used. The maximum allowable operating temperature for each material increases in this same order.
Synthetic rubber oil seals may cause trouble such as overheating, wear, and seizure, unless there is an oil film between the seal lip and shaft. Therefore, some lubricant should be applied to the seal lip when the seals are installed. It is also desirable for the lubricant inside the housing to spread a little between the sliding surfaces. However, please be aware that ester-based grease will cause acrylic rubber material to swell. Also, low aniline point mineral oil, silicone-based grease,and silicon-based oil will cause silicone-based material to swell. Moreover, urea-based grease will cause fluorine-based material to deteriorate.
The permissible circumferential speed for oil seals varies depending on the type, the finish of the shaft surface, liquid to be sealed, temperature, shaft eccentricity, etc. The temperature range for oil seals is restricted by the lip material. Approximate circumferential surface speeds and temperature permitted under favorable conditions are listed in Table 11.6.
When oil seals are used at high circumferential surface speed or under high internal pressure, the contact surface of the shaft must be smoothly finished and the shaft eccentricity should be less than 0.02 to 0.05 mm.
The hardness of the shaft's contact surface should be made higher than HRC40 by means of heat treatment or hard chrome plating in order to gain abrasion resistance. If possible, a hardness of more than HRC 55 is recommended.
The approximate level of contact surface finish required for several shaft circumferential surface speeds is given in Table 11.7.

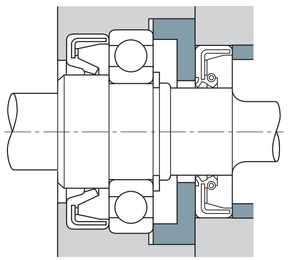

Fig. 11.8 Example of Application of Oil Seal (1)

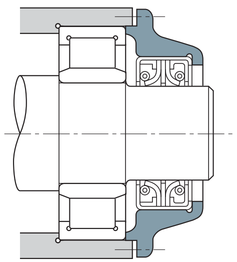

Fig. 11.9 Example of Application of Oil Seal (2)

Table 11. 6 Permissible Circumferential Surface Speeds and Temperature Range for Oil Seals

Seal Materials		Permissible Circumferential Speeds(m/sec)	Operating Temperature Range(°C)(1)
Synthetic Rubber	Nitrile Rubber	Under 16	−25 to +100
	Acrylic Rubber	Under 25	−15 to +130
	Silicone Rubber	Under 32	−70 to +200
	Fluorine-containes Rubber	Under 32	−30 to +200
Tetrafluoride Ethylene Resin		Under 15	−50 to +220

Note (1) The upper limit of the temperature range may be raised about 20 °C for operation for short intervals.

Table 11. 7 Shaft Circumferential Surface Speeds and Finish of Contact Surfaces

Circumferential Surface Speeds(m/s)	Surface Finish R_a (μm)
Under 5	0.8
5 to 10	0.4
Over 10	0.2

(2) Felt Seals

Felt seals are one of the simplest and most common seals being used for transmission shafts, etc.
However, since oil permeation and leakage are unavoidable if oil is used, this type of seal is used only for grease lubrication, primarily to prevent dust and other foreign matter from entering. Felt seals are not suitable for circumferential surface speeds exceeding 4 m/sec; therefore, it is preferable to replace them with synthetic rubber seals depending on the application.

12. LUBRICATION

12.1 Purposes of Lubrication

The main purposes of lubrication are to reduce friction and wear inside the bearings that may cause premature failure. The effects of lubrication may be briefly explained as follows:

(1) Reduction of Friction and Wear
Direct metallic contact between the bearing rings, rolling elements and cage, which are the basic components of a bearing, is prevented by an oil film which reduces the friction and wear in the contact areas.
(2) Extension of Fatigue Life
The rolling fatigue life of bearings depends greatly upon the viscosity and film thickness between the rolling contact surfaces. A heavy film thickness prolongs the fatigue life, but it is shortened if the viscosity of the oil is too low so the film thickness is insufficient.
(3) Dissipation of Frictional Heat and Cooling
Circulation lubrication may be used to carry away frictional heat or heat transferred from the outside to prevent the bearing from overheating and the oil from deteriorating.
(4) Others
Adequate lubrication also helps to prevent foreign material from entering the bearings and guards against corrosion or rusting.

12.2 Lubricating Methods

The various lubricating methods are first divided into either grease or oil lubrication. Satisfactory bearing performance can be achieved by adopting the lubricating method which is most suitable for the particular application and operating condition.
In general, oil offers superior lubrication; however, grease lubrication allows a simpler structure around the bearings. A comparison of grease and oil lubrication is given in Table 12.1.

Table 12. 1 Comparison of Grease and Oil Lubrication

Item	Grease Lubrication	Oil Lubrication
Housing Structure and Sealing Method	Simple	May be complex, Careful maintenance required.
Speed	Limiting speed is 65% to 80% of that with oil lubrication.	Higher limiting speed.
Cooling Effect	Poor	Heat transter is possible using forced oil circulation.
Fluidity	Poor	Good
Full Lubricant Replacement	Sometimes difficult	Easy
Removal of Foreign Matter	Removal of particles from grese is impossible.	Easy
External Contamination due to Leakage	Surroundings seldom contaminated by leakage.	Often leaks without proper countermeasures. Not suitable if external contamination must be avoided.

12.2.1 Grease Lubrication

(1) Grease Quantity

The quantity of grease to be packed in a housing depends on the housing design and free space, grease characteristics, and ambient temperature. For example, the bearings for the main shafts of machine tools, where the accuracy may be impaired by a small temperature rise, require only a small amount of grease. The quantity of grease for ordinary bearings is determined as follows.
Sufficient grease must be packed inside the bearing including the cage guide face. The available space inside the housing to be packed with grease depends on the speed as follows:
1/2 to 2/3 of the space … When the speed is less than 50% of the limiting speed.
1/3 to 1/2 of the space … When the speed is more than 50% of the limiting speed.