4. Internal clearance

4.1 Internal clearance

Internal clearance is one of the most important factors affecting bearing performance. The bearing "internal clearance" is the relative movement of the outer and inner rings when they are lightly pushed in opposite directions. Movement in the diametrical direction is called radial clearance and that in the shaft's direction is called axial clearance.

The reason why the internal clearance is so important for bearings is that it is directly related to their performance in the following respects. The amount of clearance influences the load distribution in a bearing and this can affect its life. It also influences the noise and vibration. In addition, it can influence whether the rolling elements move by rolling or sliding motion.

Normally, bearings are installed with interference for either the inner or outer ring and this leads to its expansion or contraction which causes a change in the clearance. Also, the bearing temperature reaches saturation during operation; however, the temperature of the inner ring, outer ring, and rolling elements are all different from each other, and this temperature difference changes the clearance (Fig. 1). Moreover, when a bearing operates under load, an elastic displacement of the inner ring, outer ring, and rolling elements also leads to a change in clearance. Because of these changes, bearing internal clearance is a very complex subject.

Therefore, what is the ideal clearance? Before considering this question, let us define the following different types of clearance. The symbol for each clearance amount is shown in parentheses.

Measured Internal Clearance (Δ_1)
This is the internal clearance measured under a specified measuring load and can be called "apparent clearance". This clearance includes the elastic deformation (δ_{FO}) caused by the measuring load.

$$\Delta_1=\Delta_0+\delta_{FO}$$

Theoretical Internal Clearance (Δ_0)
This is the radial internal clearance which is the measured clearance minus the elastic deformation caused by the measuring load.

$$\Delta_0=\Delta_1+\delta_{FO}$$

δ_{FO} is significant for ball bearings, but not for roller bearings where it is assumed to be equal to zero, and thus, $\Delta_0=\Delta_1$.

Residual Internal Clearance (Δ_f)
This is the clearance left in a bearing after mounting it on a shaft and in a housing. The elastic deformation caused by the mass of the shaft, etc. is neglected. Assuming the clearance decrease caused by the ring expansion or contraction is δ_f, then:

$$\Delta_f=\Delta_0+\delta_f$$

Effective Internal Clearance (Δ)
This is the bearing clearance that exists in a machine at its operating temperature except that the elastic deformation caused by load is not included. That is to say, this is the clearance when considering only the changes due to bearing fitting δ_f and temperature difference between the inner and outer rings δ_t. The basic load ratings of bearings apply only when the effective clearance Δ=0.

$$\Delta=\Delta_f-\delta_t=\Delta_0-(\delta_f+\delta_t)$$

Operating Clearance (Δ_F)
This is the actual clearance when a bearing is installed and running under load. Here, the effect of elastic deformation δ_F is included as well as fitting and temperature. Generally, the operating clearance is not used in the calculation.

$$\Delta_F=\Delta+\delta_F$$

The most important clearance of a bearing is the effective clearance as we have already explained. Theoretically speaking, the bearing whose effective clearance Δ is slightly negative has the longest life. (The slightly negative clearance means such effective clearance that the operating clearance turns to positive by the influence of bearing load. Strictly speaking, the amount of negative clearance varies with the magnitude of bearing load.) However, it is impossible to make the clearance of all bearings to the ideal effective clearance, and we have to consider the geometrical clearance Δ_0 in order to let the minimum value of effecive clearance be zero or a slightly negative value. To obtain this value, we should have both accurate reduction amount of clearance caused by the interference of the inner ring and outer ring δ_f and accurate amount of clearance change caused by the temperature difference between inner ring and outer ring δ_t. The methods of the calculation are discussed in the following sections.

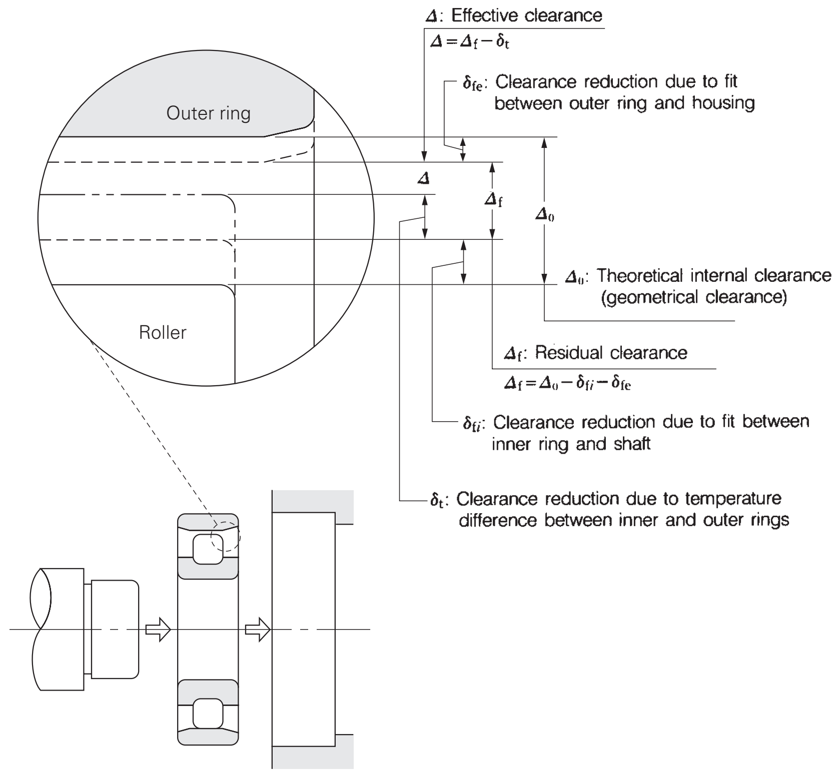

Fig. 1 Changes of radial internal clearance of roller bearing

4.2 Calculating residual internal clearance after mounting

The various types of internal bearing clearance were discussed in Section 4.1. This section will explain the step by step procedures for calculating residual internal clearance.

When the inner ring of a bearing is press fit onto a shaft, or when the outer ring is press fit into a housing, it stands to reason that radial internal clearance will decrease due to the resulting expansion or contraction of the bearing raceways. Generally, most bearing applications have a rotating shaft which requires a tight fit between the inner ring and shaft and a loose fit between the outer ring and housing. Generally, therefore, only the effect of the interference on the inner ring needs to be taken into account.

Below we have selected a 6310 single row deep groove ball bearing for our representative calculations. The shaft is set at k5, with the housing set at H7. An interference fit is applied only to the inner ring.

Shaft diameter, bore size and radial clearance are the standard bearing measurements. Assuming that 99.7% of the parts are within tolerance, the mean value ($m_{\Delta f}$) and standard deviation ($\sigma_{\Delta f}$) of the internal clearance after mounting (residual clearance) can be calculated. Measurements are given in units of millimeters (mm).

$$\sigma_s=\frac{R_s/2}{3}=0.0018$$

$$\sigma_i=\frac{R_i/2}{3}=0.0020$$

$$\sigma_{\Delta 0}=\frac{R_{\Delta 0}/2}{3}=0.0028$$

$$\sigma_f^2=\sigma_s^2+\sigma_i^2$$

$$m_{\Delta f}=m_{\Delta 0}-\lambda_i\ (m_s-m_i)=0.0035$$

$$\sigma_{\Delta f}=\sqrt{\sigma_{\Delta 0}^2+\lambda_i^2\ \sigma_f^2}=0.0035$$

where, σ_s: Standard deviation of shaft diameter
σ_i: Standard deviation of bore diameter
σ_f: Standard deviation of interference
$\sigma_{\Delta 0}$: Standard deviation of radial clearance (before mounting)
$\sigma_{\Delta f}$: Standard deviation of residual clearance (after mounting)
m_s: Mean value of shaft diameter (ϕ50+0.008)
m_i: Mean value of bore diameter (ϕ50–0.006)
$m_{\Delta 0}$: Mean value of radial clearance (before mounting) (0.014)
$m_{\Delta f}$: Mean value of residual clearance (after mounting)
R_s: Shaft tolerance (0.011)
R_i: Bearing bore tolerance (0.012)
$R_{\Delta 0}$: Range in radial clearance (before mounting) (0.017)
λ_i: Rate of raceway expansion from apparent interference (0.75 from Fig. 1)

The average amount of raceway expansion and contraction from apparent interference is calculated from $\lambda_i\ (m_m-m_i)$.

To determine, wihtin a 99.7% probability, the variation in internal clearance after mounting ($R_{\Delta f}$), we use the following equation.

$$R_{\Delta f}=m_{\Delta f}\pm 3\sigma_{\Delta f}=+0.014 \text{ to } -0.007$$

In other words, the mean value of residual clearance ($m_{\Delta f}$) is +0.0035, and the range is from −0.007 to +0.014 for a 6310 bearing.

We will discuss further in Section 4.5 the method used to calculate the amount of change in internal clearance when there is a variation in temperature between inner and outer rings.

Units: mm

Shaft diameter	ϕ50 $^{+0.013}_{+0.002}$
Bearing bore diameter, (d)	ϕ50 $^{0}_{-0.012}$
Radial internal clearance (Δ_0)	0.006 to 0.023([1])

Note ([1]) Standard internal clearance, unmounted

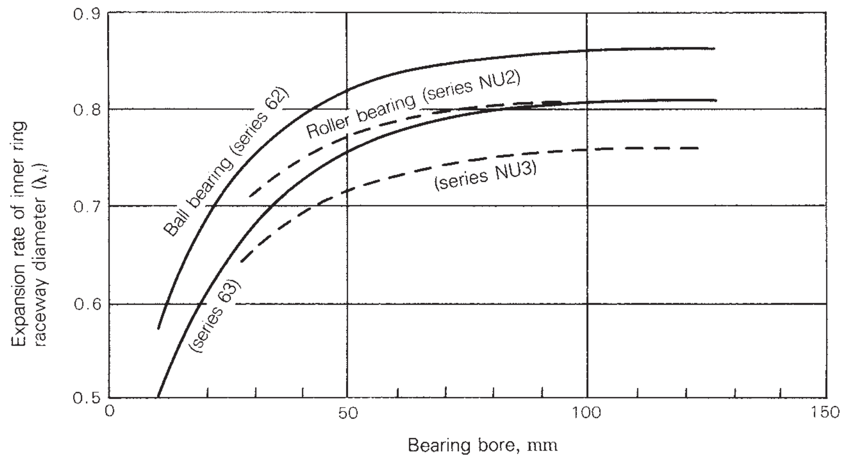

Fig. 1 Rate of inner ring raceway expansion (λ_i) from apparent interference

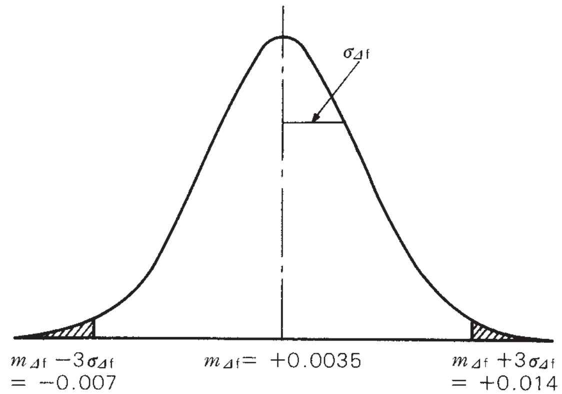

Fig. 2 Distribution of residual internal clearance

4.3 Effect of interference fit on bearing raceways (fit of inner ring)

One of the important factors that relates to radial clearance is the reduction in radial clearance resulting from the mounting fit. When inner ring is mounted on a shaft with an interference fit and the outer ring is secured in a housing with an interference fit, the inner ring will expand and the outer ring will contract.

The means of calculating the amount of ring expansion or contraction were previously noted in Section 3.4, however, the equation for establishing the amount of inner raceway expansion (ΔD_i) is given in Equation (1).

$$\Delta D_i = \Delta d \; k \, \frac{1-k_0^2}{1-k^2 k_0^2} \qquad \text{.................................. (1)}$$

where, Δd: Effective interference (mm)
k: Ratio of bore to inner raceway diameter; $k=d/D_i$
k_0: Ratio of inside to outside diameter of hollow shaft; $k_0=d_0/D_i$
d: Bore or shaft diameter (mm)
D_i: Inner raceway diameter (mm)
d_0: Inside diameter of hollow shaft (mm)

Equation (1) has been translated into a clearer graphical form in Fig. 1.

The vertical axis of Fig. 1 represents the inner raceway diameter expansion in relation to the amount of interference. The horizontal axis is the ratio of inside and outside diameter of the hollow shaft (k_0) and uses as its parameter the ratio of bore diameter and raceway diameter of the inner ring (k).

Generally, the decrease in radial clearance is calculated to be approximately 80% of the interference. However, this is for solid shaft mountings only. For hollow shaft mountings the decrease in radial clearance varies with the ratio of inside to outside diameter of the shaft. Since the general 80% rule is based on average bearing bore size to inner raceway diameter ratios, the change will vary with different bearing types, sizes, and series. Typical plots for Single Row Deep Groove Ball Bearings and for Cylindrical Roller Bearings are shown in Figs. 2 and 3. Values in Fig. 1 apply only for steel shafts.

Let's take as an example a 6220 ball bearing mounted on a hollow shaft (diameter d=100 mm, inside diameter d_0=65 mm) with a fit class of m5 and determine the decrease in radial clearance.

The ratio between bore diameter and raceway diameter, k is 0.87 as shown in Fig. 2. The ratio of inside to outside diameter for shaft, k_0, is $k_0=d_0/d$=0.65. Thus, reading from Fig. 1, the rate of raceway expansion is 73%.

Given that an interference of m5 has a mean value of 30 μm, the amount of raceway expansion, or, the amount of decrease in the radial clearance from the fit is 0.73×30=22 μm.

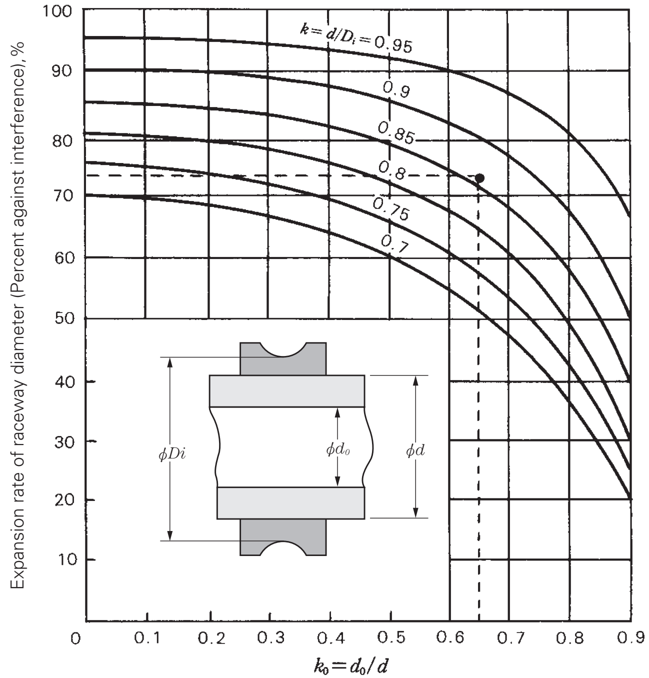

Fig. 1 Raceway expansion in relation to bearing fit (Inner ring fit upon steel shaft)

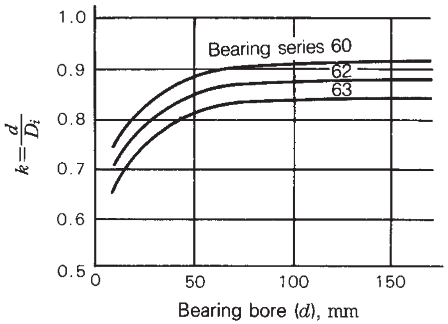

Fig. 2 Ratio of bore size to raceway diameter for single row deep groove ball bearings

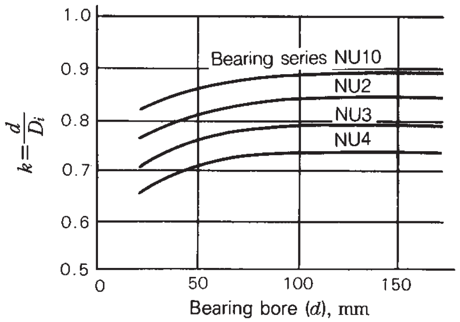

Fig. 3 Ratio of bore size to raceway diameter for cylindrical roller bearings

4.4 Effect of interference fit on bearing raceways (fit of outer ring)

We continue with the calculation of the raceway contraction of the outer ring after fitting.

When a bearing load is applied on a rotating inner ring (outer ring carrying a static load), an interference fit is adopted for the inner ring and the outer ring is mounted either with a transition fit or a clearance fit. However, when the bearing load is applied on a rotating outer ring (inner ring carrying a static load) or when there is an indeterminate load and the outer ring must be mounted with an interference fit, a decrease in radial internal clearance caused by the fit begins to contribute in the same way as when the inner ring is mounted with an interference fit.

Actually, because the amount of interference that can be applied to the outer ring is limited by stress, and because the constraints of most bearing applications make it difficult to apply a large amount of interference to the outer ring, and instances where there is an indeterminate load are quite rare compared to those where a rotating inner ring carries the load, there are few occasions where it is necessary to be cautious about the decrease in radial clearance caused by outer-ring interference.

The decrease in outer raceway diameter ΔD_e is calculated using Equation (1).

$$\Delta D_e=\Delta D\cdot h\,\frac{1-h_0^2}{1-h^2\,h_0^2}\quad\cdots\cdots\cdots\cdots (1)$$

where, ΔD: Effective interference (mm)
h: Ratio between raceway dia. and outside dia. of outer ring, $h=D_e/D$
h_0: Housing thickness ratio, $h_0=D/D_0$
D: Bearing outside diameter (housing bore diameter) (mm)
D_e: Raceway diameter of outer ring (mm)
D_0: Outside diameter of housing (mm)

Fig. 1 represents the above equation in graphic form.

The vertical axis show the outer-ring raceway contraction as a percentage of interference, and the horizontal axis is the housing thickness ratio h_0. The data are plotted for constant values of the outer-ring thickness ratio from 0.7 through 1.0 in increments of 0.05. The value of thickness ratio h will differ with bearing type, size, and diameter series. Representative values for single-row deep groove ball bearings and for cylindrical roller bearings are given in Figs. 2 and 3 respectively.

Loads applied on rotating outer rings occur in such applications as automotive front axles, tension pulleys, conveyor systems, and other pulley systems.

As an example, we estimate the amount of decrease in radial clearance assuming a 6207 ball bearing is mounted in a steel housing with an N7 fit. The outside diameter of the housing is assumed to be D_0=95, and the bearing outside diameter is D=72. From Fig. 2, the outer-ring thickness ratio, h, is 0.9. Because $h_0=D/D_0$=0.76, from Fig. 1, the amount of raceway contraction is 71%. Taking the mean value for N7 interference as 18 μm, the amount of contraction of the outer raceway, or the amount of decrease in radial clearance is 0.71×18=13 μm.

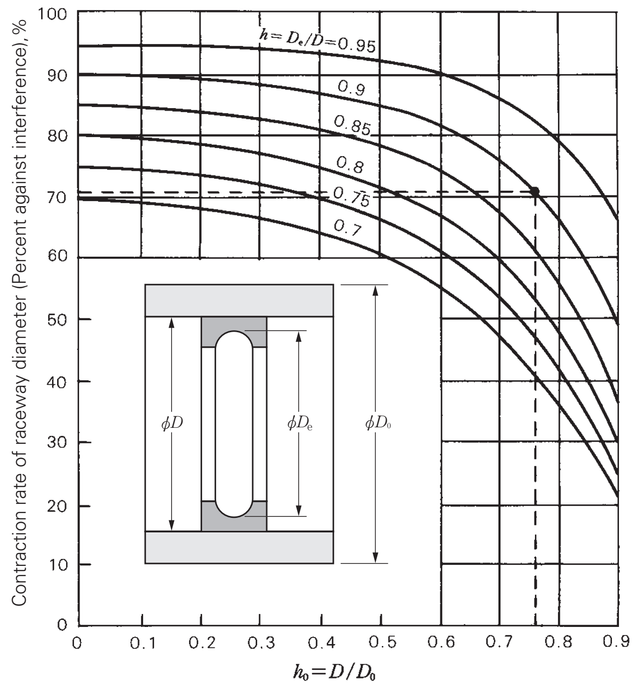

Fig. 1 Raceway contraction in relation to bearing fit (Outer ring fit in steel housing)

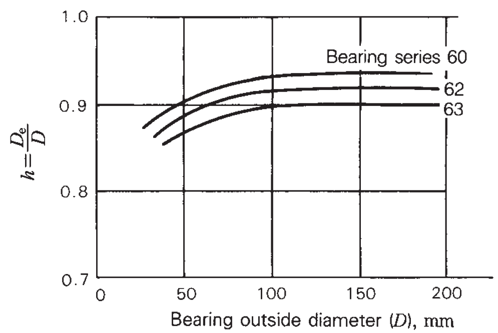

Fig. 2 Ratio of outside diameter to raceway diameter for single row deep groove ball bearings

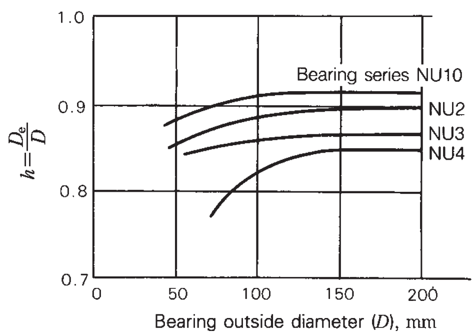

Fig. 3 Ratio of outside diameter to raceway diameter for cylindrical roller bearings

4.5 Reduction in radial internal clearance caused by a temperature difference between inner and outer rings

The internal clearance after mounting was explained in Section 4.2. We continue here by explaining the way to determine the reduction in radial internal clearance caused by inner and outer ring temperature differences and, finally, the method of estimating the effective internal clearance in a systematic fashion.

When a bearing runs under a load, the temperature of the entire bearing will rise. Of course, the rolling elements also undergo a temperature change, but, because the change is extremely difficult to measure or even estimate, the temperature of the rolling elements is generally assumed to be the same as the inner-ring temperature.

We will use the same bearing for our example as we did in Section 4.2, a 6310, and determine the reduction in clearance caused by a temperature difference of 5°C between the inner and outer rings using the equation below.

$$\delta_t=\alpha\Delta_t D_e \fallingdotseq \alpha\Delta_t \frac{4D+d}{5} \quad \cdots\cdots (1)$$

$$\fallingdotseq 12.5\times10^{-6}\times5\times\frac{4\times110+50}{5}$$

$$\fallingdotseq 6\times10^{-3}\ \text{(mm)}$$

where δ_t: Decrease in radial internal clearance caused by a temperature difference between the inner and outer rings (mm)

α: Linear thermal expansion coefficient for bearing steel, 12.5×10^{-6} (1/°C)

Δ_t: Difference in temperature between inner ring (or rolling elements) and outer ring (°C)

D: Outside diameter (6310 bearing, 110 mm)

d: Bore diameter (6310 bearing, 50 mm)

D_e: Outer-ring raceway diameter (mm)

The following equations are used to calculate the outer-ring raceway diameter:

Ball Bearings: $D_e=(4D+d)/5$

Roller Bearings: $D_e=(3D+d)/4$

Using the values for Δ_f, the residual clearance arrived at in Section 4.2, and for δ_t, the reduction in radial internal clearance caused by a temperature difference between the inner and outer rings just calculated, we can determine the effective internal clearance (Δ) using the following equation.

$$\Delta=\Delta_f-\delta_t=(+0.014 \text{ to } -0.007)\ -0.006$$
$$= +0.008 \text{ to } -0.013$$

Referring to Fig. 1 below (also see Section 2.8) we can see how the effective internal clearance influences bearing life (in this example with a radial load of 3 350 N {340 kgf}, or approximately 5% of the basic load rating). The longest bearing life occurs under conditions where the effective internal clearance is −13 μm. The lowest limit to the preferred effective internal clearance range is also −13 μm.

To summarize radial internal clearances briefly:

(1) Generally, the radial clearances given in tables and figures are theoretical internal clearances, Δ_0.

(2) The most important clearance for bearings is the effective radial internal clearance, Δ. It is a value determined by taking the theoretical clearance Δ_0 and subtracting δ_f, the reduction in clearance caused by the interference fit of one or both rings, and δ_t, the reduction in clearance caused by a temperature difference between the inner and outer rings. $\Delta=\Delta_0-(\delta_f+\delta_t)$.

(3) Theoretically, the optimum effective internal clearance Δ is a negative number close to zero which gives maximum bearing life. Therefore, it is desirable for a bearing to have an effective internal clearance greater than the ideal minimum value.

(4) To determine the relation between the effective internal clearance and bearing life (strictly speaking, the bearing load should also be considered), there is actually no need to give serious consideration to operating internal clearance Δ_F; the problem lies with the effective internal clearance Δ.

(5) The basic load rating C_r for a bearing is calculated for an effective internal clearance Δ=0.

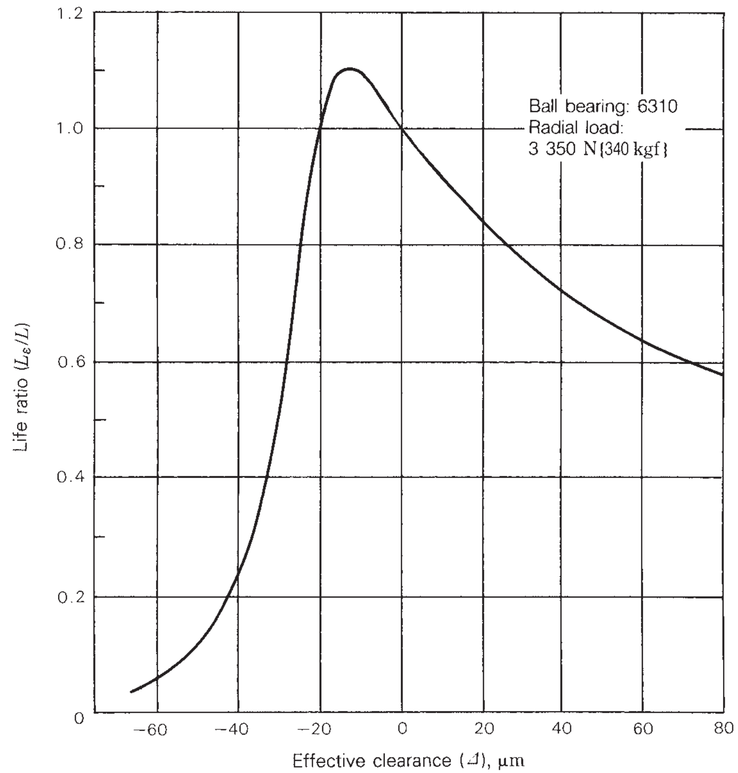

Fig. 1 Relation between effective clearance and bearing life for 6310 ball bearing

Remarks $L\varepsilon$: Life in case of effective clearance $\Delta=\varepsilon$
L: Life in case of effective clearance $\Delta=0$

4.6 Radial and axial internal clearances and contact angles for single row deep groove ball bearings

4.6.1 Radial and axial internal clearances

The internal clearance in single row bearings has been specified as the radial internal clearance. The bearing internal clearance is the amount of relative displacement possible between the bearing rings when one ring is fixed and the other ring does not bear a load. The amount of movement along the direction of the bearing radius is called the radial clearance, and the amount along the direction of the axis is called the axial clearance.

The geometric relation between the radial and axial clearance is shown in Fig. 1.

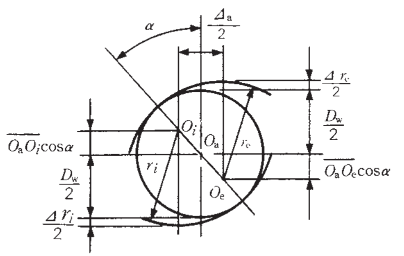

Fig. 1 Relationship Between Δ_r and Δ_a

Symbols used in Fig. 1

- O_a: Ball center
- O_e: Center of groove curvature, outer ring
- O_i: Center of groove curvature, inner ring
- D_w: Ball diameter (mm)
- r_e: Radius of outer ring groove (mm)
- r_i: Radius of inner ring groove (mm)
- α: Contact angle (°)
- Δ_r: Radial clearance (mm)
- Δ_a: Axial clearance (mm)

It is apparent from Fig. 1 that $\Delta_r = \Delta r_e + \Delta r_i$.

From geometric relationships, various equations for clearance, contact angle, etc. can be derived.

$$\Delta_r = 2\,(1-\cos\alpha)\,(r_e+r_i-D_w) \quad \cdots\cdots (1)$$

$$\Delta_a = 2\sin\alpha\,(r_e+r_i-D_w) \quad \cdots\cdots (2)$$

$$\frac{\Delta_a}{\Delta_r} = \cot\frac{\alpha}{2} \quad \cdots\cdots (3)$$

$$\Delta_a \doteqdot 2\,(r_e+r_i-D_w)^{1/2}\,\Delta_r^{1/2} \quad \cdots\cdots (4)$$

$$\alpha = \cos^{-1}\left(\frac{r_e+r_i-D_w-\frac{\Delta_r}{2}}{r_e+r_i-D_w}\right) \quad \cdots\cdots (5)$$

$$= \sin^{-1}\left(\frac{\Delta_a/2}{r_e+r_i-D_w}\right) \quad \cdots\cdots (6)$$

Because $(r_e+r_i-D_w)$ is a constant, it is apparent why fixed relationships between Δ_r, Δ_a and α exist for all the various bearing types.

As was previously mentioned, the clearances for deep groove ball bearings are given as radial clearances, but there are specific applications where it is desirable to have an axial clearance as well. The relationship between deep groove ball bearing radial clearance Δ_r and axial clearance Δ_a is given in Equation (4). To simplify,

$$\Delta_a \doteqdot K\,\Delta_r^{1/2} \quad \cdots\cdots (7)$$

where K: Constant depending on bearing design

$$K = 2\,(r_e+r_i-D_w)^{1/2}$$

Fig. 2 shows one example. The various values for K are presented by bearing size in Table 1 below.

Example

Assume a 6312 bearing, for a sample calculation, which has a radial clearance of 0.017 mm. From Table 1, K=2.09. Therefore, the axial clearance Δ_a is:

$\Delta = 2.09\times\sqrt{0.017} = 2.09\times0.13 = 0.27$ (mm)

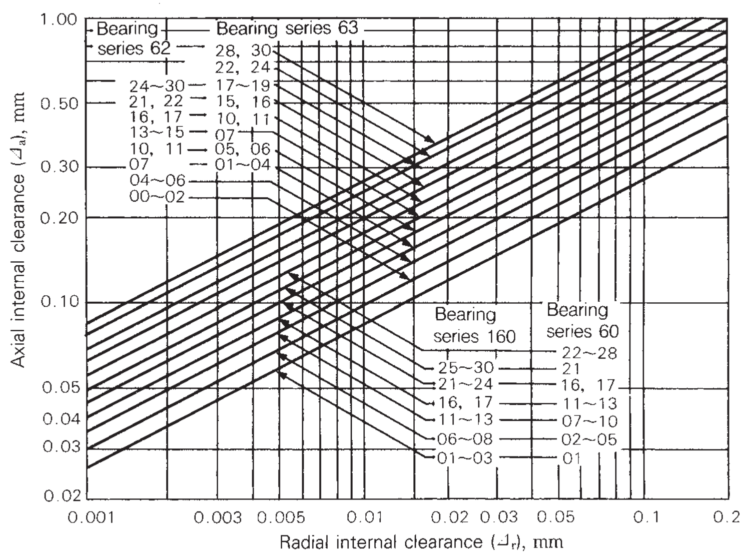

Fig. 2 Radial clearance, Δ_r and axial clearance, Δ_a of deep groove ball bearings

Table 1 Constant values of K for radial and axial clearance conversion

Bearing bore No.	K			
	Series 160	Series 60	Series 62	Series 63
00	—	—	0.93	1.14
01	0.80	0.80	0.93	1.06
02	0.80	0.93	0.93	1.06
03	0.80	0.93	0.99	1.11
04	0.90	0.96	1.06	1.07
05	0.90	0.96	1.06	1.20
06	0.96	1.01	1.07	1.19
07	0.96	1.06	1.25	1.37
08	0.96	1.06	1.29	1.45
09	1.01	1.11	1.29	1.57
10	1.01	1.11	1.33	1.64
11	1.06	1.20	1.40	1.70
12	1.06	1.20	1.50	2.09
13	1.06	1.20	1.54	1.82
14	1.16	1.29	1.57	1.88
15	1.16	1.29	1.57	1.95
16	1.20	1.37	1.64	2.01
17	1.20	1.37	1.70	2.06
18	1.29	1.44	1.76	2.11
19	1.29	1.44	1.82	2.16
20	1.29	1.44	1.88	2.25
21	1.37	1.54	1.95	2.32
22	1.40	1.64	2.01	2.40
24	1.40	1.64	2.06	2.40
26	1.54	1.70	2.11	2.49
28	1.54	1.70	2.11	2.59
30	1.57	1.76	2.11	2.59

4.6.2 Relation between radial clearance and contact angle

Single-row deep groove ball bearings are sometimes used as thrust bearings. In such applications, it is recommended to make the contact angle as large as possible.

The contact angle for ball bearings is determined by the geometric relationship between the radial clearance and the radii of the inner and outer grooves. Using Equations (1) to (6), Fig. 3 shows the particular relationship between the radial clearance and contact angle of 62 and 63 series bearings. The initial contact angle, α_0, is the initial contact angle when the axial load is zero. Application of any load to the bearing will change this contact angle.

If the initial contact angle α_0 exceeds 20°, it is necessary to check whether or not the contact area of the ball and raceway touch the edge of raceway shoulder. (Refer to Section 8.1.2)

For applications when an axial load alone is applied, the radial clearance for deep groove ball bearings is normally greater than the normal clearance in order to ensure that the contact anlgle is relatively large. The initial contact angles for C3 and C4 clearances are given for selected bearing sizes in Table 2 below.

Table 2 Initial contact angle, α_0, with C3 and C4 clearances

Bearing No.	α_0 with C3	α_0 with C4
6205	12.5° to 18°	16.5° to 22°
6210	11.5° to 16.5°	13.5° to 19.5°
6215	11.5° to 16°	15.5° to 19.5°
6220	10.5° to 14.5°	14° to 17.5°
6305	11° to 16°	14.5° to 19.5°
6310	9.5° to 13.5°	12° to 16°
6315	9.5° to 13.5°	12.5° to 15.5°
6320	9° to 12.5°	12° to 15°

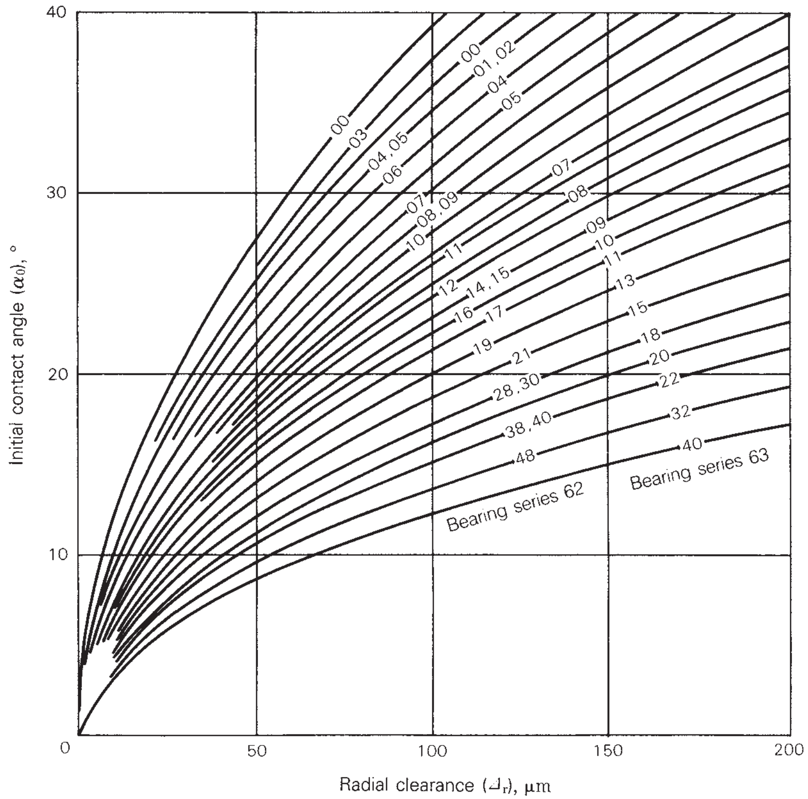

Fig. 3 Radial clearance and contact angle

4.7 Angular clearances in single-row deep groove ball bearings

When estimating bearing loads, the usual loads considered are radial loads, axial loads, or a combination of the two. Under such load, the movement of the inner and outer rings is usually assumed to be parallel.

Actually, there are many occasions when a bearing's inner and outer rings do not operate in true planar rotation because of housing or shaft misalignment, shaft deflection due to the applied load, or a mounting where the bearing is slightly skewed. In such cases, if the inner and outer ring misalignment angle is greater than a half of the bearing's angular clearance, it will create an unusual amount of stress, a rise in temperature, and premature flaking or other fatigue failure. There are more detailed reports available on such topics as how to determine the weight distribution and equivalent load for bearings which must handle moment loads. However, when considering the weight or load calculations, the amount of angular clearance in individual bearings is also of major concern in bearing selection. The angular clearance, which is clearly related to radial clearance, is the maximum angular displacement of the two ring axes when one of the bearing rings is fixed and the other is free and unloaded. An approximation of angular clearance can be determined from Equation (1) below.

$$\tan\frac{\theta_0}{2} \fallingdotseq \frac{2\,\{\mathit{\Delta}_r\,(r_e+r_i-D_w)\}^{1/2}}{D_{pw}}$$

$$= K_0 \cdot \mathit{\Delta}_r^{\,1/2} \quad \cdots\cdots\cdots (1)$$

where, $\mathit{\Delta}_r$: Radial clearance (mm)
r_e: Outer-ring groove radius (mm)
r_i: Inner-ring groove radius (mm)
D_w: Ball diameter (mm)
D_{pw}: Pitch diameter (mm)
K_0: Constant

$$K_0 = \frac{2\,(r_e+r_i-D_w)^{1/2}}{D_{pw}}$$

K_0 is a constant dependent on the individual bearing design. Table 1 gives values for K_0 for single-row deep groove bearing series 60, 62, and 63. Fig. 1 shows the relationship between the radial clearance $\mathit{\Delta}_r$ and angular clearance θ_0.

The deflection angle of the inner and outer rings is ± θ_0/2.

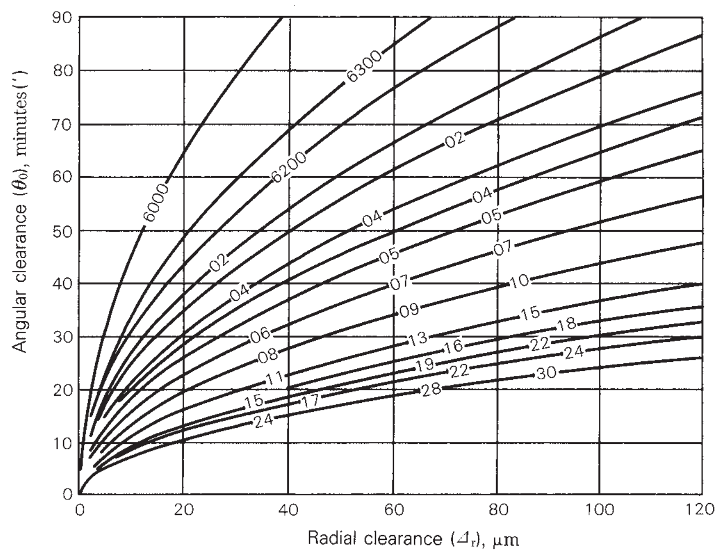

Fig. 1 Radial clearance and angular clearance

Table 1 Constant values of K_0 for radial and angular clearance conversion

Bearing bore No.	K_0 Series 60	Series 62	Series 63
	×10^{-3}	×10^{-3}	×10^{-3}
00	67.4	45.6	50.6
01	39.7	42.3	43.3
02	39.7	36.5	36.0
03	35.9	34.0	33.7
04	30.9	31.7	29.7
05	27.0	27.2	27.0
06	23.7	23.0	22.9
07	21.9	23.3	23.5
08	19.5	21.4	22.4
09	18.2	19.8	21.1
10	16.8	19.0	20.0
11	16.6	18.1	19.4
12	15.5	17.4	18.5
13	14.6	16.6	17.8
14	14.3	16.1	17.1
15	13.5	15.2	16.6
16	13.3	14.9	16.0
17	12.7	14.5	15.5
18	12.5	14.1	15.1
19	11.9	13.7	14.6
20	11.5	13.4	14.2
21	11.4	13.2	14.0
22	11.7	12.9	13.6
24	10.9	12.2	12.7
26	10.3	11.7	12.1
28	9.71	10.8	11.8
30	9.39	10.0	11.0

4.8 Relationship between radial and axial clearances in double-row angular contact ball bearings

The relationship between the radial and axial internal clearances in double-row angular contact ball bearings can be determined geometrically as shown in Fig. 1 below.

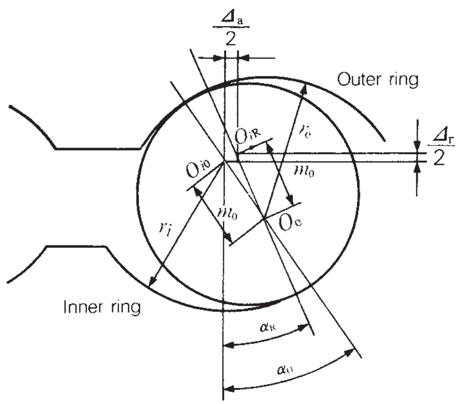

Fig. 1

where, Δ_r: Radial clearance (mm)
Δ_a: Axial clearance (mm)
α_0: Initial contact angle, inner or outer ring displaced axially
α_R: Initial contact angle, inner or outer ring displaced radially
O_e: Center of outer-ring groove curvature (outer ring fixed)
O_{i0}: Center of inner-ring groove curvature (inner ring displaced axially)
O_{iR}: Center of inner-ring groove curvature (inner ring displaced radially)
m_0: Distance between inner and outer ring groove-curvature centers, $m_0=r_i+r_e-D_w$
D_w: Ball diameter (mm)
r_i: Radius of inner-ring groove (mm)
r_e: Radius of outer-ring groove (mm)

The following relations can be derived from Fig. 1:

$$m_0 \sin \alpha_0 = m_0 \sin \alpha_R + \frac{\Delta_a}{2} \quad \cdots\cdots (1)$$

$$m_0 \cos \alpha_0 = m_0 \cos \alpha_R + \frac{\Delta_r}{2} \quad \cdots\cdots (2)$$

since $\sin^2\alpha_0 = 1-\cos^2\alpha_0$,

$$(m_0 \sin \alpha_0)^2 = m_0^2 - (m_0 \cos \alpha_0)^2 \quad \cdots\cdots (3)$$

Combined Equations (1), (2), and (3), we obtain:

$$\left(m_0 \sin \alpha_R + \frac{\Delta_a}{2}\right)^2 = m_0^2 - \left(m_0 \cos \alpha_R - \frac{\Delta_r}{2}\right)^2 \quad \cdots\cdots (4)$$

$$\therefore \Delta_a = 2\sqrt{m_0^2 - \left(m_0 \cos \alpha_R - \frac{\Delta_r}{2}\right)^2} - 2m_0 \sin \alpha_R \quad \cdots\cdots (5)$$

α_R is 25° for 52 and 53 series bearings and 32° for 32 and 33 series bearings. If we set α_R equal to 0°, Equation (5) becomes:

$$\Delta_a = 2\sqrt{m_0^2 - \left(m_0 - \frac{\Delta_r}{2}\right)^2}$$

$$= 2\sqrt{m_0 \Delta_r - \frac{\Delta_r^2}{4}}$$

However, $\frac{\Delta_r^2}{4}$ is negligible.

$$\therefore \Delta_a \fallingdotseq 2m_0^{1/2}\,\Delta_r^{1/2} \quad \cdots\cdots (6)$$

This is identical to the relationship between the radial and axial clearances in single-row deep groove ball bearings.

The value of m_0 is dependent on the inner and outer ring groove radii. The relation between Δ_r and Δ_a, as given by Equation (5), is shown in Figs. 2 and 3 for NSK 52, 53, 32, and 33 series double-row angular contact ball bearings. When the clearance range is small, the axial clearance is given approximately by

$$\Delta_a \fallingdotseq \Delta_r \cot \alpha_R \quad \cdots\cdots (7)$$

However, when the clearance is relatively large, (when $\Delta_r/D_w > 0.002$) the error in Equation (7) can be quite large.

The contact angle α_R is independent of the radial clearance; however, the initial contact angle α_0 varies with the radial clearance when the inner or outer ring is displaced axially. This relationship is given by Equation (2).

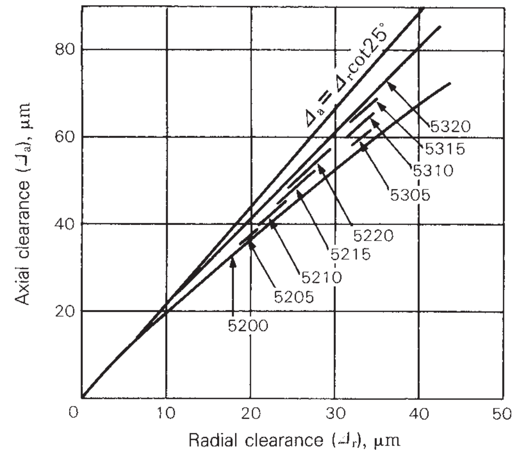

Fig. 2 Radial and axial clearances of bearing series 52 and 53

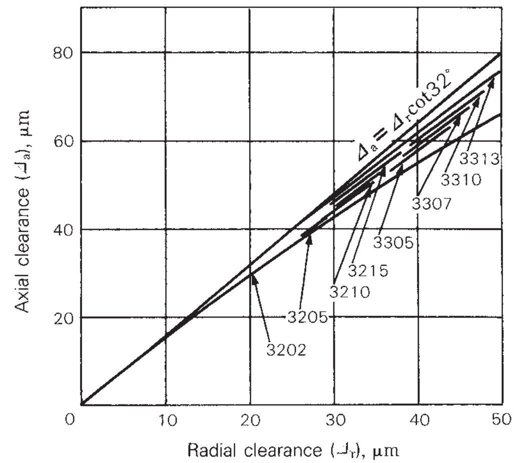

Fig. 3 Radial and axial clearances of bearing series 32 and 33

4.9 Angular clearances in double-row angular contact ball bearings

The angular clearance in double-row bearings is defined in exactly the same way as for single-row bearings; i.e., with one of the bearing rings fixed, the angular clearance is the greatest possible angular displacement of the axis of the other ring.

Since the angular clearance is the greatest total relative displacement of the two ring axes, it is twice the possible angle of inner and outer ring movement (the maximum angular displacement in one direction from the center without creating a moment).

The relationship between axial and angular clearance for double-row angular contact ball bearings is given by Equation (1) below.

$$\varDelta_a=2m_0\left\{\sin\alpha_0+\frac{\theta R_i}{2m_0}-\sqrt{1-\left(\cos\alpha_0+\frac{\theta l}{4m_0}\right)^2}\right\} \qquad \cdots\cdots (1)$$

where, $\varDelta_a$: Axial clearance (mm)
m_0: Distance between inner and outer ring groove curvature centers, $m_0=r_e+r_i-D_w$ (mm)
r_e: Outer-ring groove radius (mm)
r_i: Inner-ring groove radius (mm)
α_0: Initial contact angle (°)
θ: Angular clearance (rad)
R_i: Distance between shaft center and inner-ring groove curvature center (mm)
l: Distance between left and right groove centers of inner-ring (mm)

The above equation is shown plotted in Fig. 1 for NSK double-row angular contact ball bearings series 52, 53, 32, and 33.

The relationship between radial clearance $\varDelta_r$ and axial clearance $\varDelta_a$ for double-row angular contact ball bearings was explained in Section 4.8. Based on those equations, Fig. 2 shows the relationship between angular clearance θ and radial clearance $\varDelta_r$.

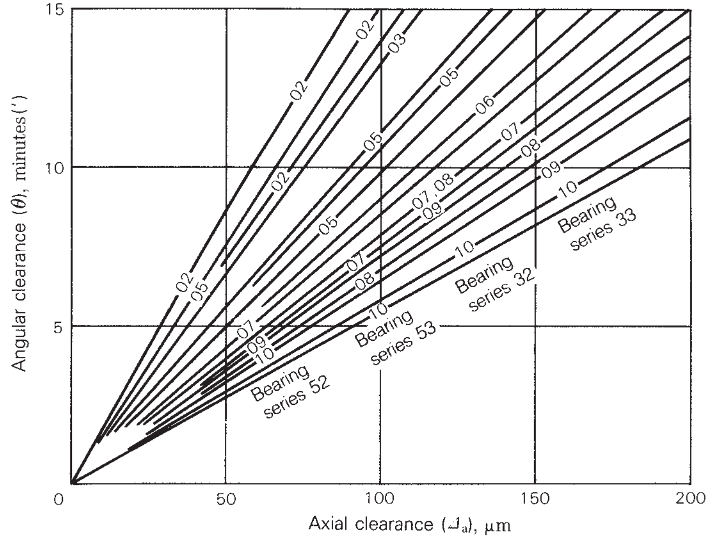

Fig. 1 Relationship between axial and angular clearances

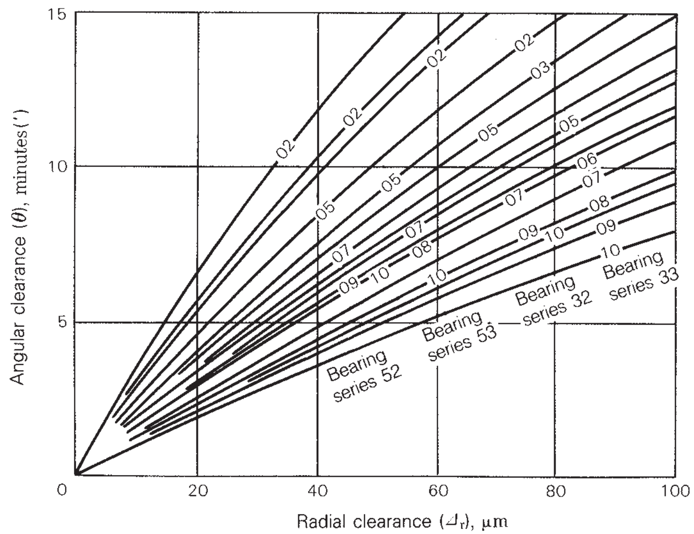

Fig. 2 Relationship between radial and angular clearances

4.10 Measuring method of internal clearance of combined tapered roller bearings (offset measuring method)

Combined tapered roller bearings are available in two types: a back-to-back combination (DB type) and a face-to-face combination (DF type) (see Fig. 1 and Fig. 2). The advantages of these combinations can be obtained by assembly as one set or combined with other bearings to be a fixed- or free-side bearing.

For the DB type of combined tapered roller bearing, as its cage protrudes from the back side of the outer ring, the outer ring spacer (K spacer in Fig. 1) is mounted to prevent mutual contact of cages. For the inner ring, the inner ring spacer (L spacer in Fig. 1), having an appropriate width, is provided to secure the clearance. For the DF type, as shown in Fig. 2, bearings are used with a K spacer.

In general, to use such a bearing arrangement either an appropriate clearance is required that takes into account the heat generated during operation or an applied preload is required that increases the rigidity of the bearings. The spacer width should be adjusted so as to provide an appropriate clearance or preload (minus clearance) after mounting.

Hereunder, we introduce you to a clearance measurement method for a DB arrangement.

(1) As shown in Fig. 3, put the bearing A on the surface plate and after stabilization of rollers by rotating the outer ring (more than 10 turns), measure the offset $f_A=T_A-B_A$.

(2) Next, as shown in Fig. 4, use the same procedure to measure the other bearing B for its offset $f_B=T_B-B_B$.

(3) Next, measure the width of the K and L spacers as shown in Fig. 5.

From the results of the above measurements, the axial clearance Δ_a of the combined tapered roller bearing can be obtained, with the use of symbols shown in Figs. 3 through 5 by Equation (1):

$$\Delta_a=(L-K)-(f_A+f_B) \quad \cdots\cdots (1)$$

As an example, for the combined tapered roller bearing HR32232JDB+KLR10AC3, confirm the clearance of the actual product conforms to the specifications. First, refer to NSK Rolling Bearing Catalog CAT. No. E1102 (Page A93) and notice that the C3 clearance range is Δ_r=110 to 140 μm.

To compare this specification with the offset measurement results, convert it into an axial clearance Δ_a by using Equation (2):

$$\Delta_a=\Delta_r \cot\alpha \fallingdotseq \Delta_r \frac{1.5}{e} \quad \cdots\cdots (2)$$

where, e: Constant determined for each bearing No. (Listed in the Bearing Tables of NSK Rolling Bearings Catalog)

refering to the said catalog (Page B127), with use of e=0.44, the following is obtained:

$$\Delta_a=(110 \text{ to } 140)\times\frac{1.5}{e}$$

$$\fallingdotseq 380 \text{ to } 480\ \mu m$$

It is possible to confirm that the bearing clearance is C3, by verifying that the axial clearance Δ_a of Equation (1) (obtained by the bearing offset measurement) is within the above mentioned range.

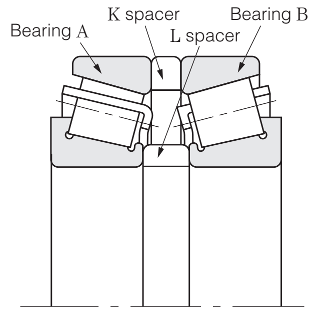

Fig. 1 DB arrangement

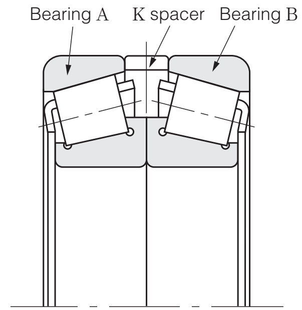

Fig. 2 DF arrangement

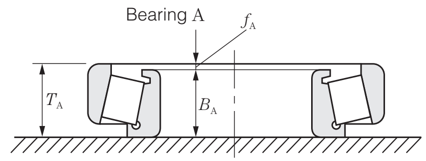

Fig. 3

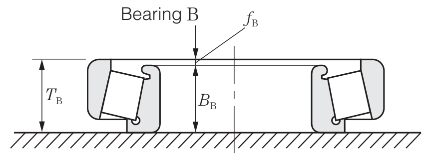

Fig. 4

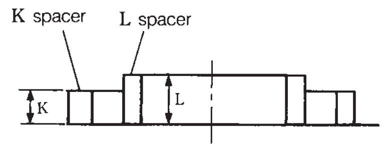

Fig. 5

4.11 Internal clearance adjustment method when mounting a tapered roller bearing

The two single row tapered roller bearings are usually arranged in a configuration opposite each other and the clearance is adjusted in the axial direction. There are two types of opposite placement methods: back-to-back arrangement (DB arrangement) and face-to-face arrangement (DF arrangement).

The clearance adjustment of the back-to-back arrangement is performed by tightening the inner ring by a shaft nut or a shaft end bolt. In Fig. 1, an example using a shaft end bolt is shown. In this case, it is necessary that the fit of the tightening side inner ring with the shaft be a loose fit to allow displacement of the inner ring in the axial direction.

For the face-to-face arrangement, a shim is inserted between the cover, which retains the outer ring in the axial direction, and the housing in order to allow adjustment to the specified axial clearance (Fig. 2). In this case, it is necessary to use a loose fit between the tightening side of the outer ring and the housing in order to allow appropriate displacement of the outer ring in the axial direction. When the structure is designed to install the outer ring into the retaining cover (Fig. 3), the above measure becomes unnecessary and both mounting and dismounting become easy.

Theoretically when the bearing clearance is slightly negative during operation, the fatigue life becomes the longest, but if the negative clearance becomes much bigger, then the fatigue life becomes very short and heat generation quickly increases. Thus, it is generally arranged that the clearance be slightly positive (a little bigger than zero) while operating. In consideration of the clearance reduction caused by temperature difference of inner and outer rings during operation and difference of thermal expansion of the shaft and housing in the axial direction, the bearing clearance after mounting should be decided.

In practice, the clearance C1 or C2 is frequently adopted which is listed in "Radial internal clearances in double-row and combined tapered roller bearing (cylindrical bore)" of NSK Rolling Bearing Catalog CAT. No. E1102, Page A93.

In addition, the relationship between the radial clearance Δ_r and axial clearance Δ_a is as follows:

$$\Delta_a = \Delta_r \cot \alpha \fallingdotseq \Delta_r \frac{1.5}{e}$$

where, α: Contact angle
e: Constant determined for each bearing No. (Listed in the Bearing Tables of NSK Rolling Bearing Catalog)

Tapered roller bearings, which are used for head spindles of machine tools, automotive final reduction gears, etc., are set to a negative clearance for the purpose of obtaining bearing rigidity. Such a method is called a preload method. There are two different modes of preloading: position preload and constant pressure preload. The position preload is used most often.

For the position preload, there are two methods: one method is to use an already adjusted arrangement of bearings and the other method is to apply the specified preload by tightening an adjustment nut or using an adjustment shim.

The constant pressure preload is a method to apply an appropriate preload to the bearing by means of spring or hydraulic pressure, etc. Next we introduce several examples that use these methods:

Fig. 4 shows the automotive final reduction gear. For pinion gears, the preload is adjusted by use of an inner ring spacer and shim. For large gears on the other hand, the preload is controlled by tightening the torque of the outer ring retaining screw.

Fig. 5 shows the rear wheel of a truck. This is an example of a preload application by tightening the inner ring in the axial direction with a shaft nut. In this case, the preload is controlled by measuring the starting friction moment of the bearing.

Fig. 6 shows an example of the head spindle of the lathe, the preload is adjusted by tightening the shaft nut.

Fig. 7 shows an example of a constant pressure preload for which the preload is adjusted by the displacement of the spring. In this case, first find a relationship between the spring's preload and displacement, then use this information to establish a constant pressure preload.

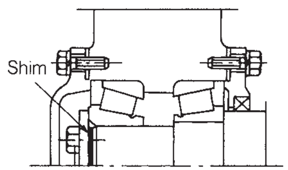

Fig. 1 DB arrangement whose clearance is adjusted by inner rings.

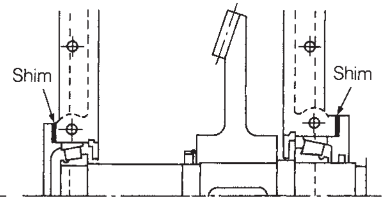

Fig. 2 DF arrangement whose clearance is adjusted by outer rings.

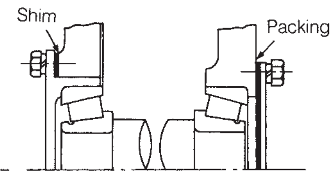

Fig. 3 Examples of clearance adjusted by shim thickness of outer ring cover

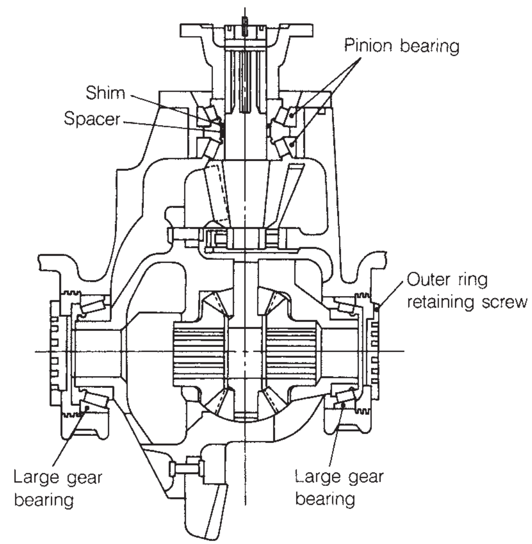

Fig. 4 Automotive final reduction gear

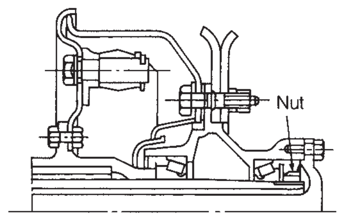

Fig. 5 Rear wheel of truck

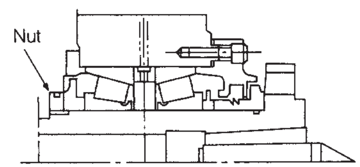

Fig. 6 Head spindle of lathe

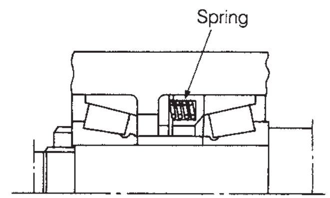

Fig. 7 Constant pressure preload applied by spring