▪ Pre-loading

A certain degree of clearance is required under normal conditions to allow a shaft to make linear and rotational motion in a sliding bearing. Guide bearings, composed of rolling elements such as balls or rollers, however, allow slight moves of a shaft even in the presence of negative clearance, or interference, referred to as "pre-loaded" which constitutes one of the most remarkable characteristics of rolling guide bearings. Proper pre-loads, eliminate jolts and provide increased rigidity. Thus, an applied external force will cause only an extraordinarily small deformation. In this way, the center of a shaft, even under an external force which fluctuates in its intensity, is kept intact, maintaining high-precision in the host machine. The reason for this in the case of balls, is that a correlation between an external force and its resultant deformation does not give a linear form, but instead a non-linear spring characteristic, as shown in Fig. 1, in which an initially soft spring changes into a hard one following a certain amount of external force being applied. Rollers on the other hand, assume a linear spling-like characteristic and show a less effective rigidity increase from the application of a apply a proper pre-load. Excessive pre-loading along with the rise in temperature caused by the increased friction will sharply reduce the life of the bearing. The recommended pre-loading value in common use, an empirical value, is about one third of the external force applied on a bearing.

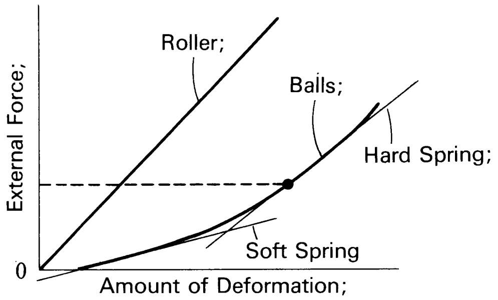


Fig. 1

▪ Deformation and Hertz's Stress caused by Application of Loads on Linear Bearings

The following formulas give values for both deformation and Hertz's stress, with a ball confined between an outer sleeve and a shaft under external force P, as shown in Fig. 2.

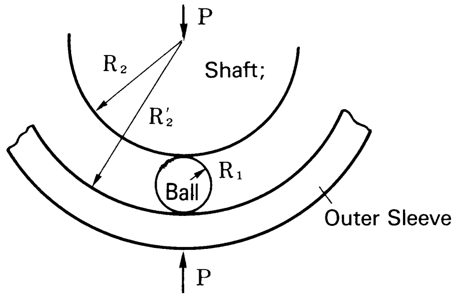


Fig. 2

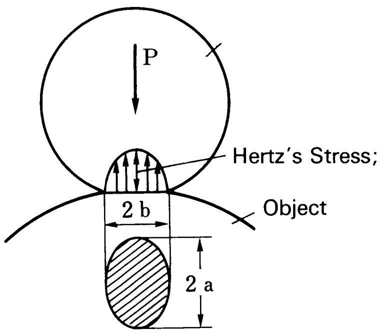


Fig. 3

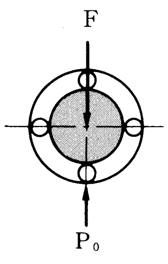
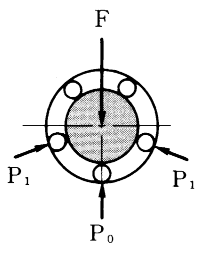
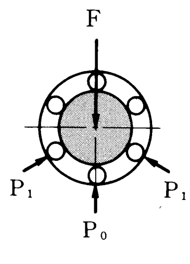
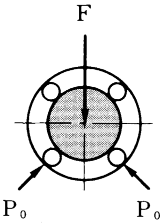
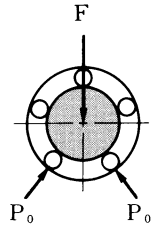
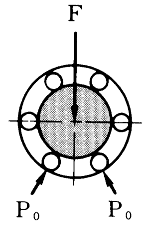
Ball Row Alignment vs. External Load	Number of Rows of Balls		
	Four Rows	Five Rows	Six Rows
Ball Row Alignment Case (A)	 $F = P_0$	 $F = 1.106P_0$	 $F = 1.354P_0$
Ball Row Alignment Case (B)	 $F = 1.414P_0$	 $F = 1.618P_0$	 $F = 1.732P_0$
Load Ratio	1.414	1.463	1.280

Fig. 4

Aggregate Total Deformation:

$$\delta=0.0014P^{\frac{2}{3}}\left[\left(\frac{2}{R_1}+\frac{1}{R_2}\right)^{\frac{1}{3}}+\left(\frac{2}{R_1}-\frac{1}{R_2'}\right)^{\frac{1}{3}}\right]\ (\text{mm})\cdots\cdots(1)$$

Hertz's Stress:

Stress on the Shaft Side ··· $$\sigma_i=178\left(\frac{2}{R_1}+\frac{1}{R_2}\right)^{\frac{2}{3}}P^{\frac{1}{3}}\ (\text{kgf/mm}^2)\cdots\cdots(2)$$

Stress on the Outer Sleeve Side ··· $$\sigma_o=178\left(\frac{2}{R_1}-\frac{1}{R_2'}\right)^{\frac{2}{3}}P^{\frac{1}{3}}\ (\text{kgf/mm}^2)\cdots\cdots(3)$$

A ball, in contact with an object under external force P, provides an extremely small contact surface in an elliptical shape, as shown in Fig. 3 in which the contact surface has expanded markedly. The stress, originating on the surface, supports the force P. The maximum value, or the center value of the resultant pressure on the contact surface is referred to as "Hertz's Stress". The maximum design value of Hertz' Stress ranges from 280 to 300 kgf/mm^2 under normal conditions, and 150 to 200 kgf/mm^2 for non-stage transmissions such as a ring-cone type.

Fig. 4 illustrates correlations between the load P, working on each row of balls, and the external force F, acting on linear bearings which comprise in normal cases four to six rows of balls. The values of the basic dynamic load rating C, shown in our catalogues are all for use in case (A), in which an external force F is applied right above a row of balls. The case (B) in which uniformly distributed rows of balls support an external force however, require a larger C value obtained by multiplying the value C, given in our catalogues, by a corresponding value of the load ratios in Fig. 4.

■ Linear Bearings with Moment Loads Applied

A load whose point of application deviates from the central part of an outer sleeve on the shaft of a linear bearing causes moment which has extremely adverse effects on the performance of a linear ball bearing, such as marked reduction in its operating life and increased friction. Avoid to the utmost, possible moment application on linear ball bearings. If not, prior calculations for service life and frictional force of linear ball bearings, based on the maximum value of load distribution in a linear bearing, must be done under acting moment.

An example of a correlation between load buildup ratio, k, and load eccentricity, e, is shown in Fig. 6 with a bearing clearance factor, a, as parameter, and a bearing comprising six rows of balls with each row composed of nine balls, Z=9, and an external force acting right above one of the row of balls. The example shows that the value k increases in response to a growing value e, or to an increased distance of an external force F, from the center of an outer sleeve. Also, it illustrates that the larger the bearing clearance factor a, the more sharply the value k, rises. It also shows, however that the value k starts to increase gradually in and around the area given by e=1 in all cases.

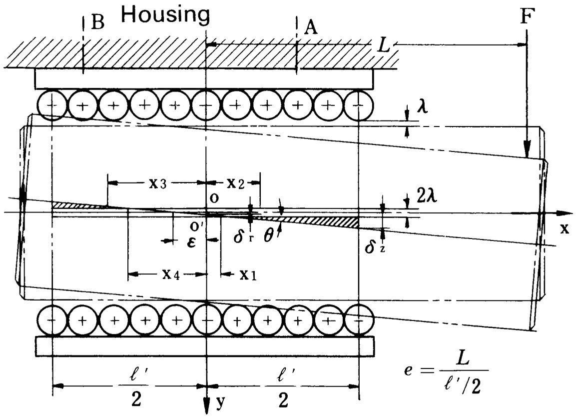
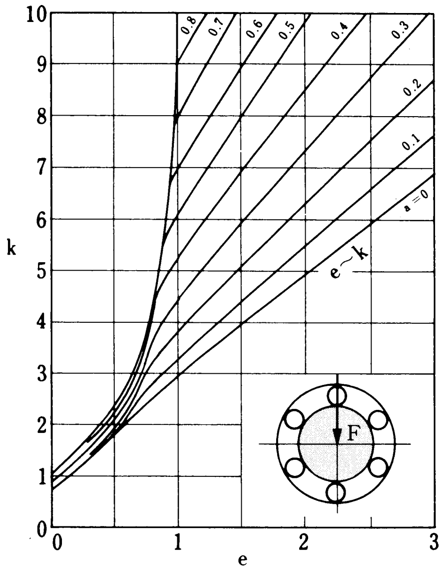


Fig. 5; Load Status When Moment Load Applied;



A case with positive bearing clearance and six rows of balls

Fig. 6; Moment Load Factor k

$$e=\frac{L}{\ell'/2}\qquad a=\frac{\lambda}{\delta}$$

λ : Radial clearance of a linear bearing (μm)

δ : Eccentricity of shaft center (μm); See Fig. 5.

$$K=\frac{P_{max}}{F/Z}$$

P_{max} : Maximum value of load distribution on balls in a linear bearing (kgf)

F : External force (kgf)

Z : Number of balls in a row

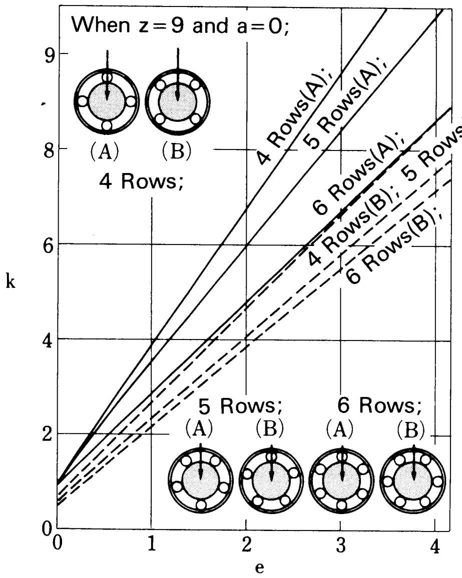
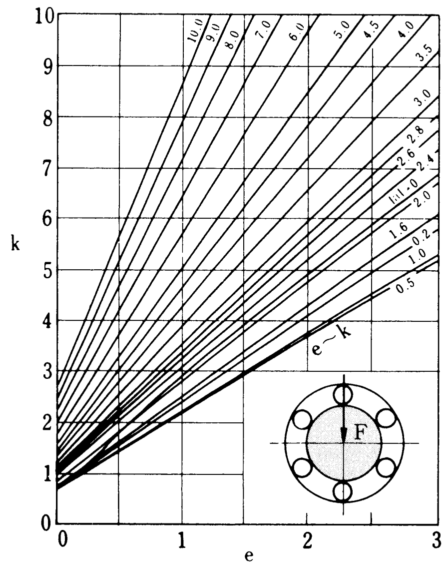


Fig. 7



A case with negative bearing clearance (pre-loading) and six rows of balls.

Fig. 8. Moment Load Factor k

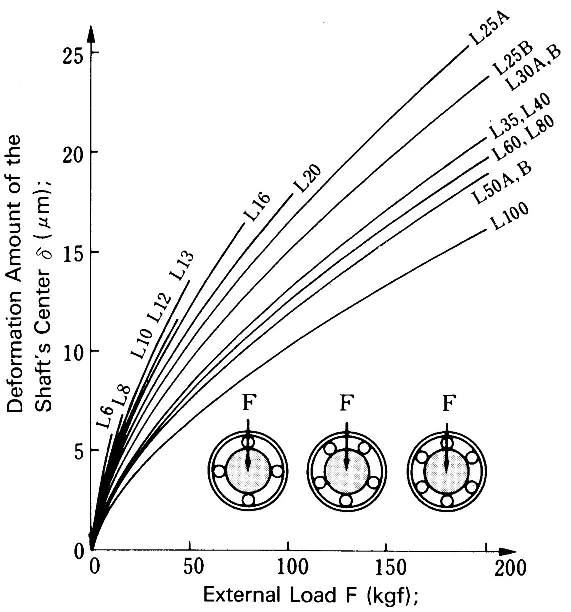


Fig. 9(A)

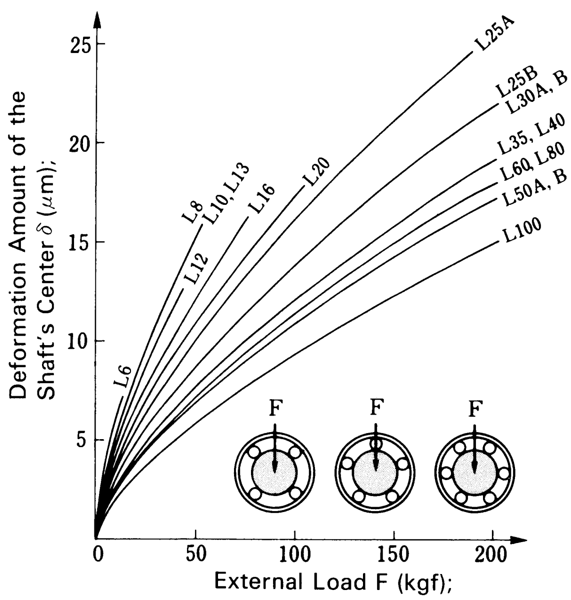
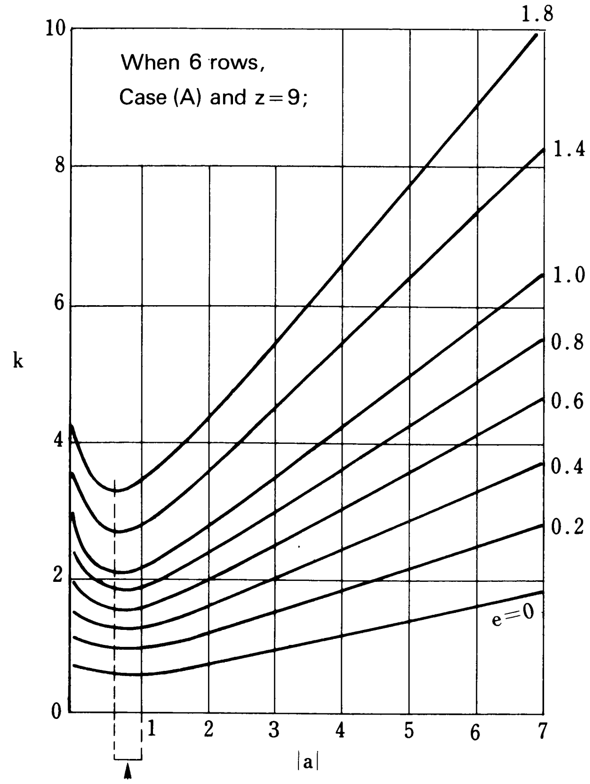


Fig. 9(B)

Fig. 7 shows correlations between k and e with four, five and six rows of balls, in both case (A), where an external force is applied right above a row of balls, and case (B), where an external force is in the center of two rows of balls. A corresponding value δ to an external force F can be obtained from Fig. 9 for various OZAK linear bearings. Fig. 8 illustrates k~e correlations in the case (A) with six rows of balls under the pre-loaded conditions which were explained earlier. It shows, when compared with Fig. 6, that the load buildup ratio with moderate pre-loading, or a≦2, stands at a much smaller value than in the absence of bearing clearance, or a=0. Thus, pre-loading not only eliminates jolts on the shaft but lowers the maximum value of load distribution on balls, leading to a favorable increase in the linear bearing life.



Domain giving Minimum k Values

Fig. 10

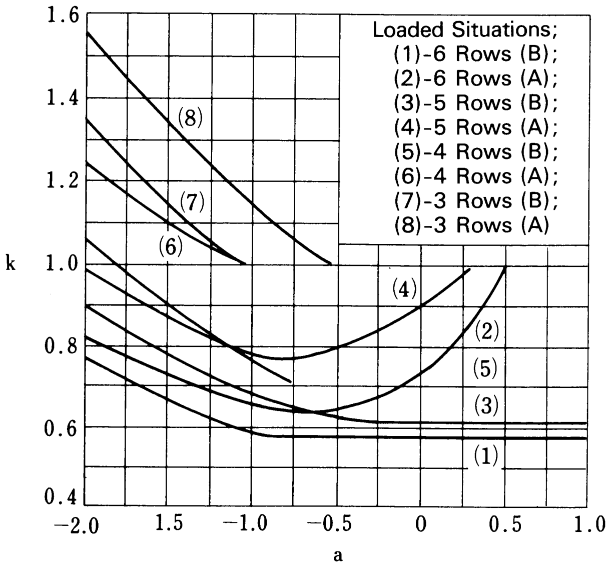


Fig. 11

Furthermore, Fig. 10 clearly shows the beneficial features of pre-loading and indicates its recommended values, in this case |a|= 0.6~0.8. Also, correlations between k and a in both different numbers of rows of balls and various loaded conditions are shown in Fig. 11, which suggests a ball bearing be used in a condition given by $-1 < a < 0$.

Relative Location of Balls against External Load

Case (A):

$$\delta=1.4\left(\frac{F}{\bigcirc Z}\right)^{\frac{2}{3}}\left[\left(\frac{2}{R_1}+\frac{1}{R_2}\right)^{\frac{1}{3}}+\left(\frac{2}{R_1}-\frac{1}{R_2'}\right)^{\frac{1}{3}}\right] \quad (\mu m)$$

Case (B):

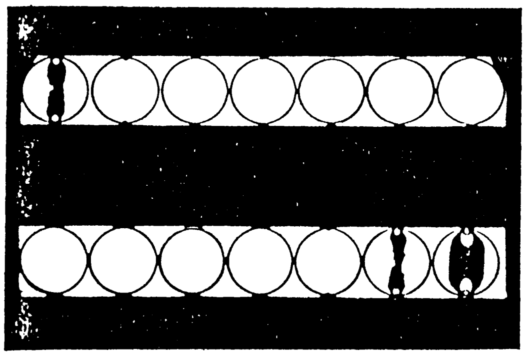
$$\delta=\frac{1}{\cos\left(\frac{180^\circ}{Z'}\right)}1.4\left(\frac{F}{\square Z}\right)^{\frac{2}{3}}\left[\left(\frac{2}{R_1}+\frac{1}{R_2}\right)^{\frac{1}{3}}+\left(\frac{2}{R_1}-\frac{1}{R_2'}\right)^{\frac{1}{3}}\right] \quad (\mu m)$$

Z = Number of balls in a row

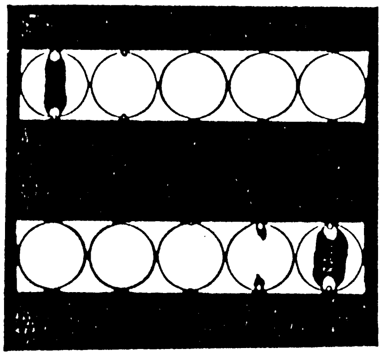
Z' = Number of rows of balls

$\bigcirc$, $\square = F/P_0$

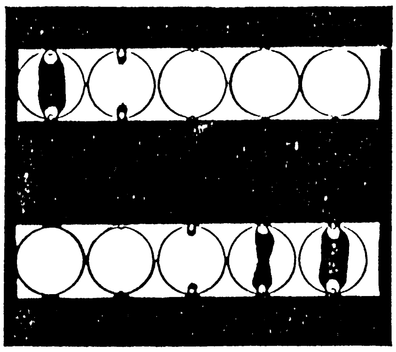
■ Photoelastic pictures taken during moment load application



$\lambda = 15\mu m$



$\lambda = 0$



$\lambda = -5\mu m$

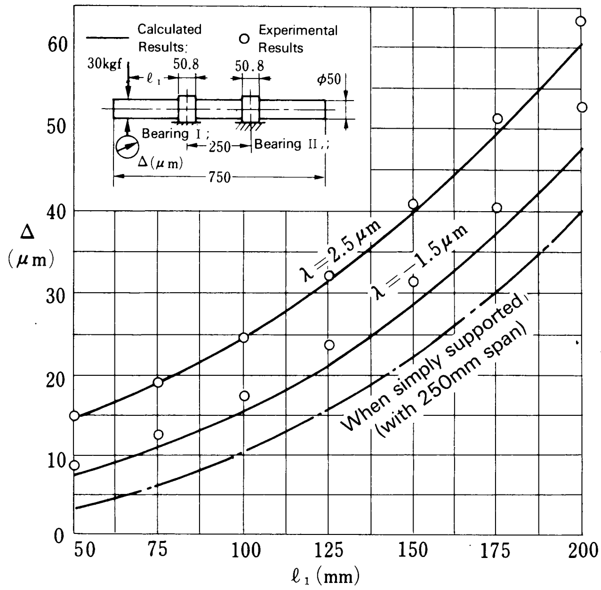
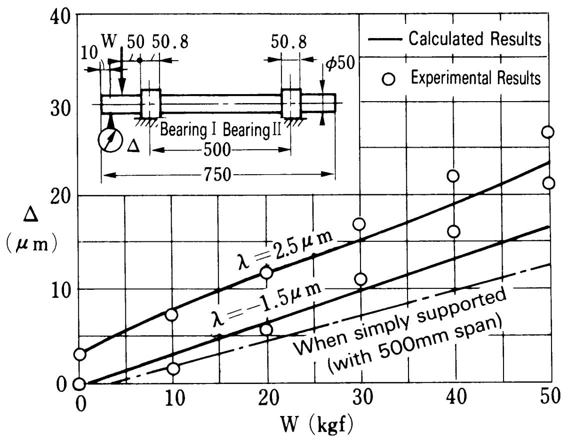


Fig. 12

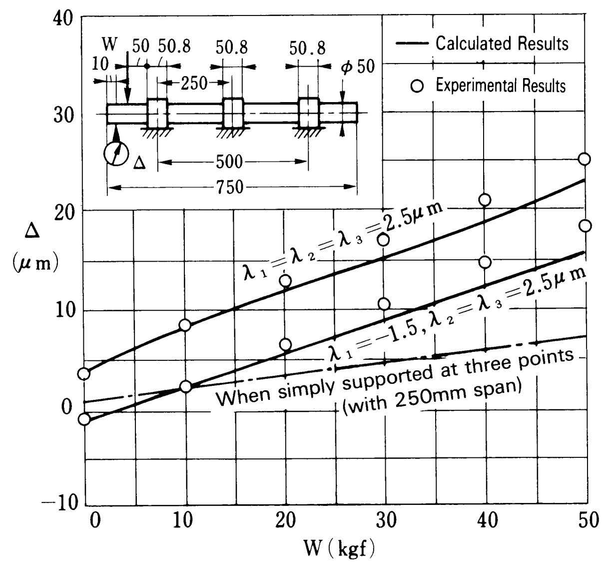
■ Shaft's rigidity supported by linear bearings

Reflecting recent demands for high-precision machinery, factors like the rigidity of a shaft, or the amount of deformation brought about by acting loads, have become urgent issues. Higher rigidity, in fact, is indispensable for high-precision machines. Responding to this need, some special devices have been unveiled, such as granite bedding and a guide shaft made of fine ceramics. Fig. 12 shows the strain on a shaft caused by a load, 30 kgf, which is applied on one end of the shaft supported by two linear bearings. It indicates that the amount of strain, Δ, increases corresponding to the expansion of distance, ℓ_1, to the loaded point. Also, it shows that a sharp rise in rigidity can be expected following the pre-load application, when compared a situation under −1.5μm pre-loading with a ball bearing maintained in the 2.5μm radial clearance. Meanwhile, the shaft supported by linear bearings bears a larger deformation with the additional strain caused by loaded balls than when compared with that of a simply supported shaft, as shown by a dashed line in Fig. 12. It can't be totally eliminated, however, in order to minimize the unfavorable stress, either increase the number of rows of balls, or install grooves to allow balls to roll. Fig. 13 shows a comparison of rigidity between two cases — Case (a) with 500mm of a shaft's span supported by two linear bearings; and Case (B) with an additional linear bearing installed at the centerspan of the shaft in Case (A). Stress in both cases expands along with an increase in acting loads, however, there is only a slight difference between the two cases. Thus, two linear bearings prove to be adequate to support a shaft. Also, the comparison indicates that pre-loading is extremely effective in both cases.



(a) When Two Bearings Used;

Fig. 13(a)



(b) When Three Bearings Used;

Fig. 13(b)

■ Linear Bearing Noise During Operation

(1) Outlines

In general, the machines with rolling elements tend to generate noises and vibrations. Most of the noises are of 1kHz or more and are said to make most human beings experience unpleasant sensations.

- Linear Bearing Noise
  - Essential Noises
    - Structural Noises
      - Raceway Track Noises
      - Jarring Noises
      - Impact Noises of Balls
      - Noises at the Return Section
    - Noises caused by Manufacturing Processes — Chattering Noises
    - Noises caused by Handling
      - Noises caused by Flaws
      - Noises caused by Foreign Objects
  - External Noises
    - Beating Noises of Motor
    - Vibration Noises from Bearing Shaft

(2) Raceway Track Noises

The noises are caused by the rolling balls on the raceway tracks. The sound pressure level increases along with the increase in the rotational speed. The smaller the more the sound pressure level increase. However, shaft vibration caused by excessive clearance makes the sound pressure level even larger. The sound pressure level goes down when lubricants with a high viscosity are in use. When grease is used, the level decreases corresponding to the higher viscosity of the base oil. However, the grease consistency and the shape and size of the soap fibers in the grease affect the pressure level. The level goes down with an increase in housing rigidity. It is a proven fact that the raceway track noises are caused by the minor surface variations of the raceway tracks and balls.

(3) Jarring Noises

Many cases of jarring have been reported when roller bearing are used. The jarring noises are caused when a radial load is applied on the bearings with excessive radial clearance. Also, the use of grease with a poor lubrication ability causes the noise, especially in winter. Furthermore, the noise can be caused by the skidding of the suddenly-accelerated rolling elements when they enter the loaded area, following the slow-down of the rotating speed in the non-loaded area caused by grease resistance, etc. However, the noise can be reduced to some degree by means of proper oil films and additional work in the design state to reduce ball skidding.

(4) Impact Noises and Noises at the Return Section

Impact noises are caused by the collision of the balls when they are either accelerated or decelerated at the circuit gates, and by the differences in the ball speed caused by fluctuations of the contact points between the balls and the raceway surface. Also, there are noises caused by the collision between the balls and the return section and the jarring noises of the balls confined in the return section. Those noises can be reduced by means of the use of non-metal spacer balls or using the proper oil film.

(5) Chattering Noises

Chattering noises are considerably harsh to the ear with their constant frequency of noise at the same rotating speed. The noises are said to be caused by the waviness of both the raceway and ball surface with a relatively big height in their circumferential direction.

(6) Noises caused by Flaws

These noises are caused by the flaws, indentations or rust on the surface of the raceway tracks and balls. The continuous or periodic noises are caused by the flaws on the raceway tracks, the intermittent noises are caused by those on the ball surface.

(7) Noises caused by Foreign Objects

The non-periodic noises, accompanied by vibrations, are caused by the intrusion of foreign objects whose locations change from time to time. Neither the sound pressure level of the noises nor the effect on the noises by the rotating speed is said to be stable. The noises can be removed by removal of foreign objects from the lubricant and an additional installation of seals.

■ Service Life Calculation of Linear Bearings under Shaft's Deflection

The following formulas give the service life calculation of the linear bearings under the large shaft's deflection.

$$L=\left(\frac{C}{P}\cdot f_{\alpha}\right)^3\times 50\text{km}$$

f_a: Influence factor caused by shaft's deflection

f_a can be obtained from Fig. 15, based on the shaft's supported condition illustrated in Fig. 14.

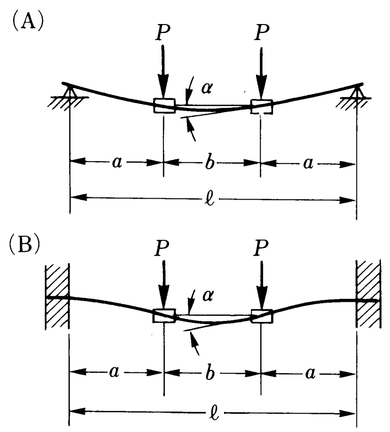


Fig. 14. Shaft's Supported Conditions

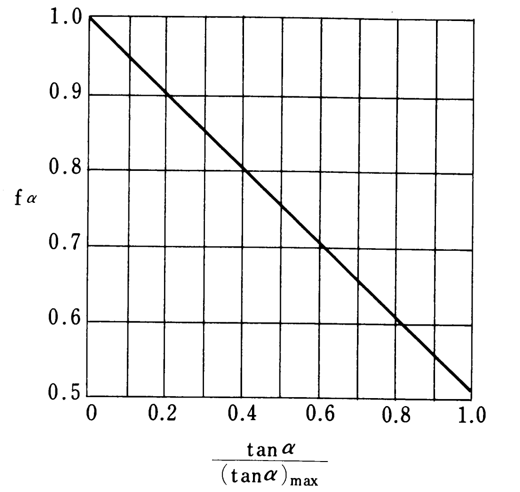


Fig. 15. Correlation between Shaft Deflection and Influence factor fa:

(A) When Just Supported on Both Ends:

$$\tan\alpha=\frac{P\cdot a\cdot b}{2\cdot E\cdot I}=4.853\times10^{-6}\,\frac{P\cdot a\cdot b}{d^4}$$

(B) When Fixedly Supported on Both Ends:

$$\tan\alpha=u_f\cdot\frac{P\cdot a\cdot b}{2\cdot E\cdot I}=u_f\cdot 4.853\times10^{-6}\,\frac{P\cdot a\cdot b}{d^4}$$

d : Shaft diameter (mm); P: Acting Load (kgf)
E : Modulus of longitudinal elasticity 2.1×10^4 (kgf/mm²);
u_f : means (b/ℓ) values of fixed support shaft, See Fig. 16.; Also, the geometrical moment of inertia, $I=\pi d^4/64$, and $(\tan\alpha)_{max}$ can be obtained from Table. 1 which is classified by the shaft's diamters.

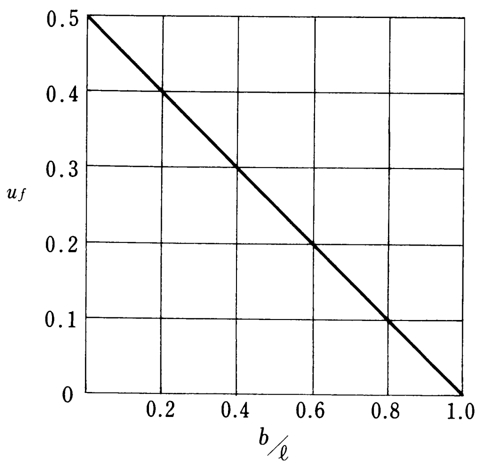


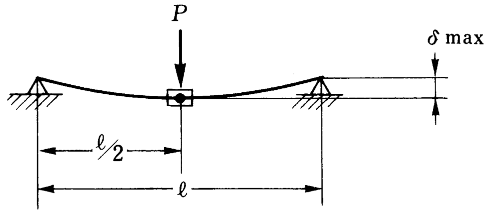
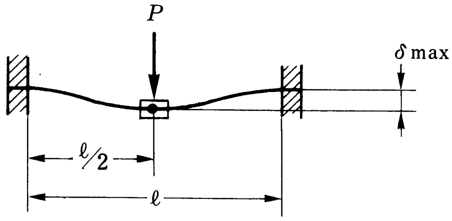
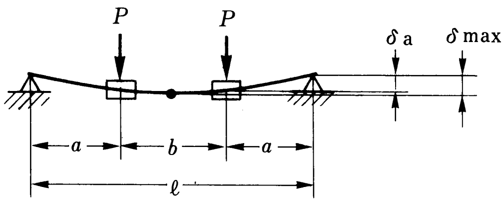
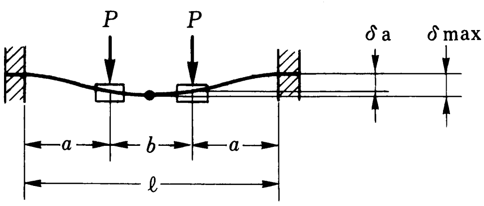
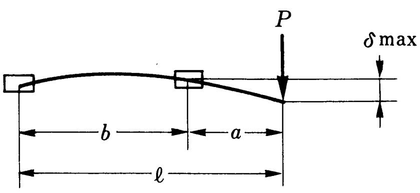
Fig. 16. Factor u_f Value

Table 1.

Shaft Diameter	$(\tan\alpha)_{max}$
3	8.3×10^{-4}
4	7.0
5	7.1
6	8.0
8	7.1
10	7.9
12	7.0
13	10.7
16	7.2

Shaft Diameter	$(\tan\alpha)_{max}$
20	7.4×10^{-4}
25	6.6
30	5.5
35	4.5
40	6.6
50	5.3
60	5.7
80	8.3
100	5.5

■ Shaft Deflection Amount Calculation Formulas:

Supported & Loaded Variations;	Shaft's Deflection Amount (mm)
	$\delta_{max}=\frac{P\cdot\ell^3}{48\cdot E\cdot I}=2.021\times10^{-5}\frac{P\cdot\ell^3}{d^4}$
	$\delta_{max}=\frac{P\cdot\ell^3}{192\cdot E\cdot I}=5.053\times10^{-6}\frac{P\cdot\ell^3}{d^4}$
	$\delta_a=\frac{P\cdot a^2}{6\cdot E\cdot I}(2a+3b)=1.617\times10^{-4}\frac{P\cdot a^2\cdot(2a+3b)}{d^4}$ $\delta_{max}=\frac{P\cdot a}{24\cdot E\cdot I}(3\ell^2-4a^2)=4.042\times10^{-5}\frac{P\cdot a\cdot(3\ell^2-4a^2)}{d^4}$
	$\delta_a=\frac{P\cdot a^3}{6\cdot E\cdot I}(2-\frac{3a}{\ell})=1.617\times10^{-4}\frac{P\cdot a^3}{d^4}(2-\frac{3a}{\ell})$ $\delta_{max}=\frac{P\cdot a^2}{24\cdot E\cdot I}(2a+3b)=4.042\times10^{-5}\frac{P\cdot a^2\cdot(2a+3b)}{d^4}$
	$\delta_{max}=\frac{Pa^2\ell}{3\cdot E\cdot I}=3.234\times10^{-4}\frac{Pa^2\ell}{d^4}$

E : Modulus of longitudinal elasticity 2.1×10^4 (kgf/mm²);
P : Acting Load (kgf);
I : Geometrical moment of inertia, $I=\pi d^4/64$ for solid shafts and $I=\pi(d^4-d_0^{\,4})/64$ (mm⁴) for hollow shafts;
d_0: Pipe's inside diameter (mm)