

OPTIMUM DESIGNING BALL SCREW

In order to work out an optimum design for machines, it is necessary to examine the rigidity of the feed screw system, the positioning accuracy and the driving torque in due consideration of the required function, performance and cost.

•RIGIDITY OF THE FEED SCREW SYSTEM

To improve the positioning accuracy and responsiveness of precision machines and equipment when they are controlled, it is necessary to take into consideration the rigidity of each component of the feed screw system. The rigidity (K) of the feed screw system is expressed by the following formula:

$$K = \frac{P}{\delta} \quad (\text{N}/\mu\text{m}) \dots \dots \dots (14)$$

Where,

- P: Axial load applied to feed screw system (N)
- δ: Axial elastic deformation of feed screw system (μm)

The relationship between the rigidity of feed screw system and that of each component is as follows:

$$\frac{1}{K} = \frac{1}{K_\ell} + \frac{1}{K_n} + \frac{1}{K_b} + \frac{1}{K_h} \quad \dots \dots \dots (15)$$

Where,

- K_ℓ: Rigidity of screw shaft to tension and compression
- K_n: Rigidity of nut
- K_b: Rigidity of support bearing
- K_h: Rigidity of nut mounting portion and bearing mounting portion

•Tension and compression strength of screw shaft “K_ℓ”

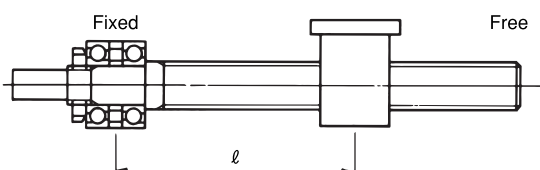
$$K_\ell = \frac{P}{\delta_\ell} \quad (\text{N}/\mu\text{m}) \dots \dots \dots (16)$$

Where,

- P: Axial load (N)
- δ_ℓ: Amount of expansion or contraction of screw shaft (μm)

When an external axial load is applied to the screw shaft, the axial expansion and contraction is expressed by the following formulae. The axial expansion and contraction comes out directly as the backlash of the ball screw.

1. Fixed-Free

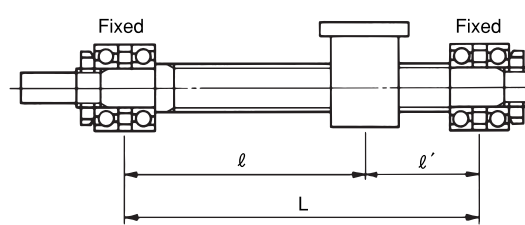


$$\delta_\ell = \frac{4P\ell}{E\pi d^2} \times 10^3 \quad (\mu\text{m}) \dots \dots \dots (17)$$

Where,

- P: Axial load (N)
- E: Young's modulus (2.06X10⁵N/mm²)
- d: Screw shaft root diameter (mm)
- ℓ: Distance between loading points (mm)

2. Fixed-Fixed



$$\delta_\ell = \frac{4P\ell\ell'}{E\pi d^2 L} \times 10^3 \quad (\mu\text{m}) \dots \dots \dots (18)$$

Where,

- P: Axial load (N)
- E: Young's modulus (2.06X10⁵N/mm²)
- d: Screw shaft root diameter (mm)
- ℓ, ℓ': Distance between loading points (mm)
- L: Distance between supports (mm)

The maximum value is obtained from formula 18 when ℓ = ℓ' = L/2

$$(\delta_\ell = \frac{PL}{E\pi d^2} \times 10^3)$$

Therefore, the maximum expansion and contraction of the screw shaft supported with the both ends fixed becomes 1/4 of the one with Fixed-Free.

•Rigidity of nut “K_n”

Rigidity of single nut (with axial clearance) “K_{ns}”

When a ball screw receives an axial load, the steel balls and the thread groove will be deformed.

The relationship between the axial load (P) and the axial elastic deformation (δ_{ns}) is expressed by the following formula:

$$\delta_{ns} = \frac{2.6}{\sin\alpha} \left(\frac{Q^2}{D_b} \right)^{\frac{1}{3}} K \quad (\mu\text{m}) \dots \dots \dots (19)$$

Where,

- α: Contact angle of steel ball with thread groove (45°)
- D_b: Steel ball diameter (mm)
- K: Coefficient corresponding to accuracy and construction (1.4-1.6)
- Q: Load per steel ball (N)

$$Q = \frac{P}{Z \sin\alpha}$$

- P: Axial load (N)
- Z: Number of steel balls

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Theoretical rigidity (K_{NS}) is obtained from the amount of elastic deformation when an axial load equivalent to 30% of the basic dynamic load rating (C) are shown in the dimension tables on and after Page 334. Rigidity (K_{NS}) corresponding to an axial load (P) is expressed by the following formula:

$$K_{nS} = K_{NS} \left(\frac{P}{0.3C} \right)^{\frac{1}{3}} \quad (\text{N}/\mu\text{m}) \dots \dots \dots \textcircled{20}$$

Where,
 C : Basic dynamic load rating (N)
 P : Axial load (N)

(δ_{nS}) in formula ⑱ is expressed by the following formula using the rigidity (K_{NS}) of the single nut and the basic dynamic load rating (C):

$$\delta_{nS} = \frac{(0.3C)^{\frac{1}{3}} P^{\frac{2}{3}}}{K_{NS}} \quad (\mu\text{m}) \dots \dots \dots \textcircled{21}$$

Where,
 K_{NS} : Theoretical rigidity of single nut (N/ μm)
 C : Basic dynamic load rating (N)
 P : Axial load (N)

Rigidity of preload nut " K_{nw} "

The dimension tables on and after Page 358 give theoretical rigidity (K_{NW}) obtained from the amount of elastic deformation that will occur when a preload of $1/15$ of the basic dynamic load rating (C) is applied to the nut and an axial load of about 3 times or less of the preload is applied.

These values are of practical use, because they were calculated on the basis of the results of the rigidity test including the nut rigidity test.

Rigidity (K_{nw}) corresponding to any preload can be obtained from the following formula:

$$K_{nw} = K_{NW} \left(\frac{P_L}{\frac{1}{15}C} \right)^{\frac{1}{3}} \quad (\text{N}/\mu\text{m}) \dots \dots \dots \textcircled{22}$$

Where,
 P_L : Preload (N)
 C : Basic dynamic load rating (N)

Backlash and preload

The backlash of a ball screw is the sum of the axial clearance and the elastic deformation caused by the axial load at the contact point of the steel balls with the thread groove.

The axial clearance can be completely eliminated by preloading and the axial elastic deformation can be reduced to a great extent by setting a proper preload and thus, the rigidity can be increased.

Preloading effect by double nut

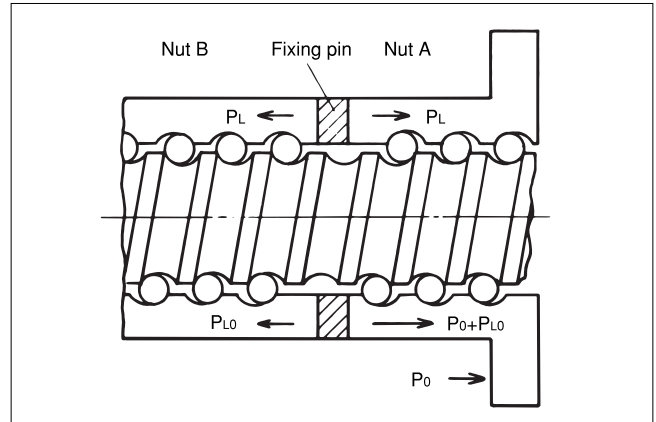


Fig. 9 Preloading effect by double nut

In Figs. 9 and 10, Nuts A and B undergo an elastic deformation of (δ_{nw0}) by preload (P_L) respectively.

When external load (P_0) is applied to Nut A, the elastic deformation of Nuts A and B is:

$$\delta_{nwA} = \delta_{nw0} + \delta_{nw1}$$

$$\delta_{nwB} = \delta_{nw0} - \delta_{nw1}$$

The load applied to Nuts A and B is:

$$P_A = P_L + P_0 - P_0' = P_0 + P_{L0}$$

$$P_B = P_L - P_0' = P_{L0}$$

Therefore, an amount of (P_0') of the external load (P_0) is offset because the deformation of Nut B decreases.

As a result, the elastic deformation of Nut A is reduced.

This effect will continue until (δ_{nwB}) is zeroed, namely until the elastic deformation caused by the external load becomes (δ_{nw0}) and the preload applied to Nut B is completely released.

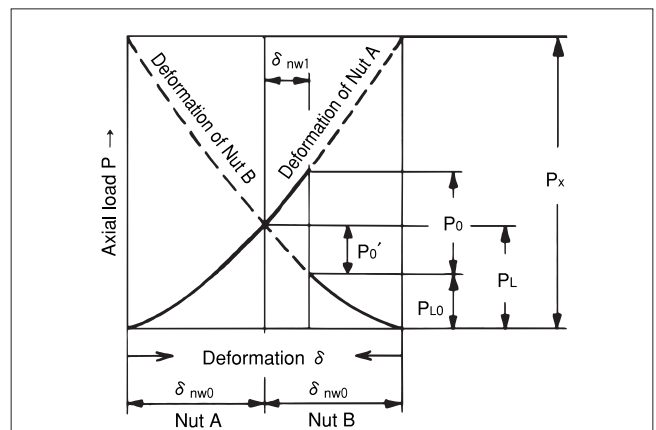


Fig. 10 Preload diagram

The guideline of the preload to ball screw

The axial elastic deformation (δ_{nw0}) is proportional to a value obtained by raising the axial load (P) to two-thirds power according to the Hertz's law of point contact.

Therefore, deformation caused by preloading is:

$$\delta_{nw0} = C \cdot P_L^{\frac{2}{3}}$$

Deformation caused by an external load when the preload applied to one nut is completely released is :

$$2\delta_{nw0} = C \cdot P_x^{\frac{2}{3}}$$

From the above two equations,

$$\left(\frac{P_x}{P_L}\right)^{\frac{2}{3}} = \frac{2\delta_{nw0}}{\delta_{nw0}} = 2$$

Therefore, the released preload is:

$$P_x = 2.8P_L \approx 3P_L \quad \text{.....(23)}$$

Where,

P_x : External axial load when preload applied to one nut returns to zero (N)

P_L : Preload (N)

As shown in equation (23), preloading effect is about 3 times as much as the preload amount. Generally, therefore, the preload amount is set at $\frac{1}{3}$ of the maximum axial load. On the other hand, taking into consideration the life and efficiency, the preload amount is usually set at $\frac{1}{20}$ to $\frac{1}{10}$ of the basic dynamic load rating.

Classification of preload

	Light preload	Normal preload	Medium preload	heavy preload
Preload	$\frac{1}{20} C$ or less	$\frac{1}{20} - \frac{1}{15} C$	$\frac{1}{15} - \frac{1}{10} C$	$\frac{1}{10} C$ or more

C: Basic dynamic load rating (N)

Elastic displacement curve of preloaded nut

Fig. 11 shows Elastic Displacement Curves of the single nut (non-preload) and the preloaded nut. Where an axial load (P_x), three times the preload (P_L), is applied, the elastic displacement of the preloaded nut is just one half of the elastic displacement of the non-preloaded single nut.

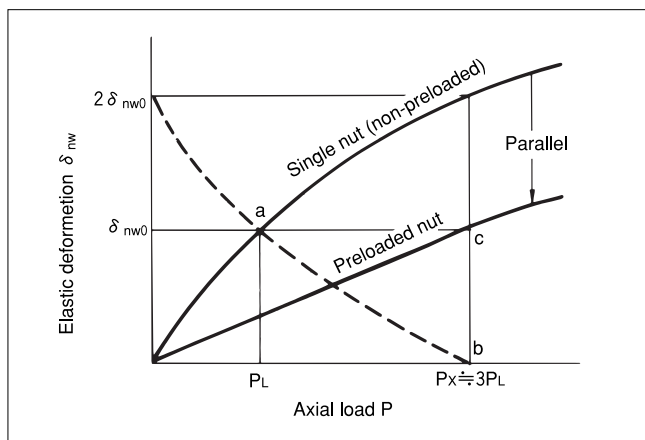


Fig. 11 Elastic deformation curve of nut

Preloading method of the double nut

Preloading is to be applied in general to two nuts in tension (tension preloading) or in compression (compression preloading).

KURODA's ball screws adopt the tension preloading unless otherwise specified.

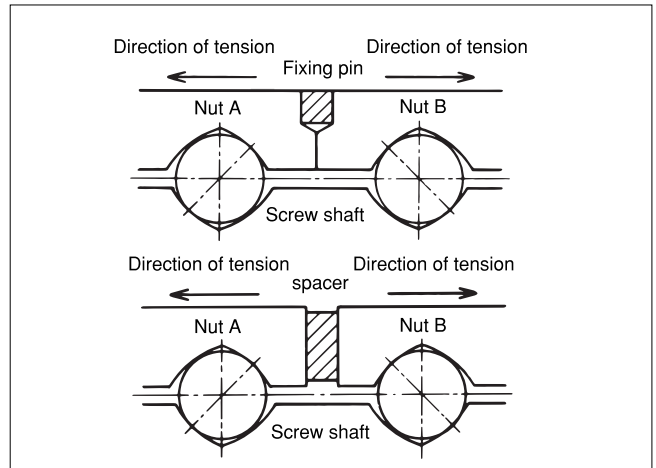


Fig. 12 Tension preloading

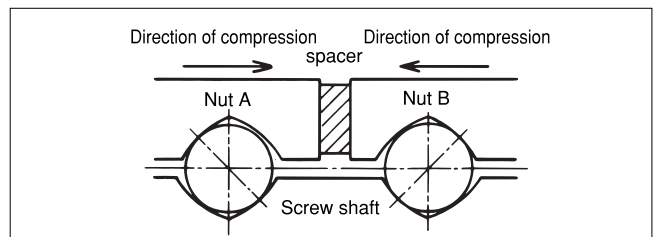


Fig. 13 Compression preloading

•Preloaded with pin

This is the simplest and most effective tension preloading method. The required preload is attained and kept by inserting a pin between the two nuts.

(Standard method of KURODA)

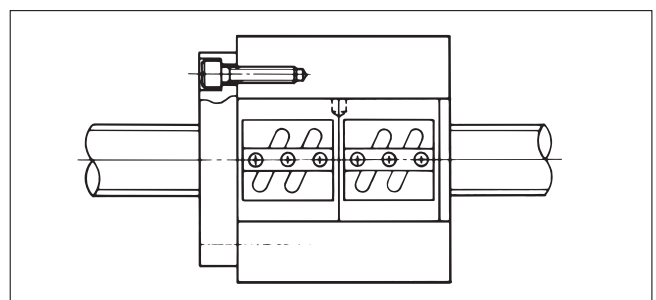


Fig. 14 Preloaded with pin (I)

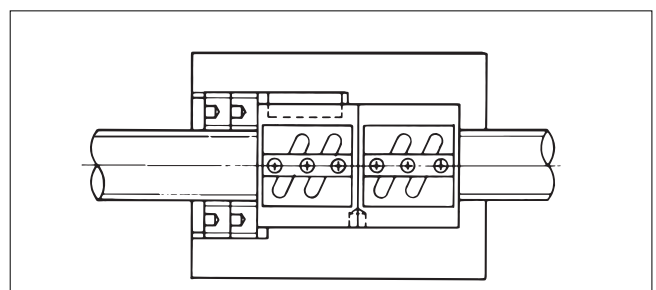


Fig. 15 Preloaded with pin (II)

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•Preloaded with spacer

In this method, preload is adjusted by the thickness of a spacer inserted between the two nuts. Both tension preloading and compression preloading are available.

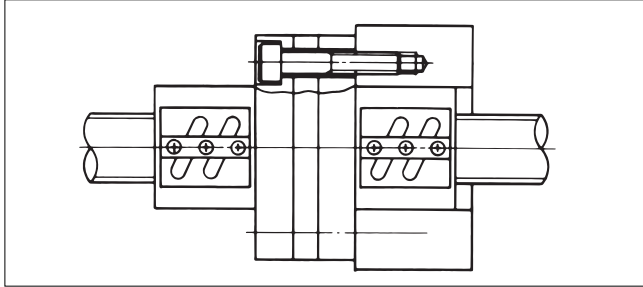


Fig. 16 Preloaded with spacer (I)

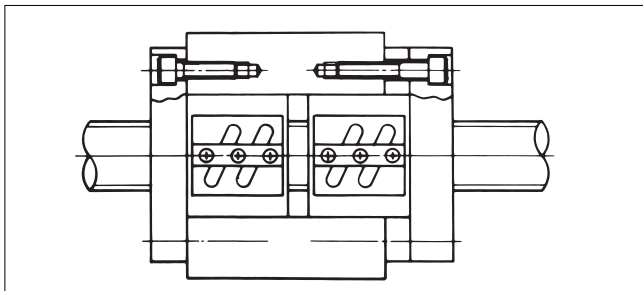


Fig. 17 Preloaded with spacer (II)

Preloading method of the integral nut

Slightly offset lead on the nut generates preload in a single nut.

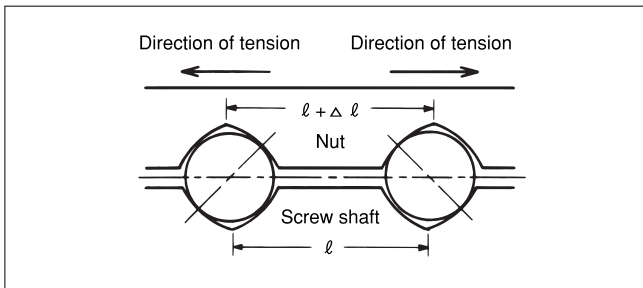


Fig. 18 Preloading method of the integral nut

Preloading method of the single nut

In this method, the single nut is preloaded by putting over-sized steel balls between the thread groove of the screw shaft and that of the nut.

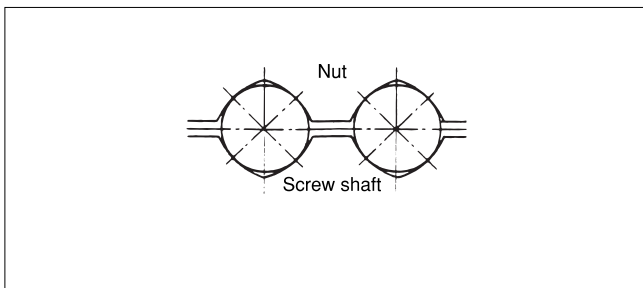


Fig. 19 Preloaded with over-sized ball

•Rigidity of support bearing“K_b”

The rigidity of a bearing to which preload (P_L) is applied is expressed by the following formula:

•Rigidity of ball bearing

$$K_b = \frac{2.83P_L}{\delta_b} \quad (\text{N}/\mu\text{m}) \dots\dots\dots 24$$

Where,

P_L:Preload (N)

δ_b:Axial elastic deformation to preload (μm)

Axial elastic deformation of angular ball bearing

$$\delta_b = \frac{2}{\sin\alpha} \left(\frac{Q^2}{D_b} \right)^{\frac{1}{3}} \quad (\mu\text{m}) \dots\dots\dots 25$$

$$Q = \frac{P}{Z \sin\alpha}$$

Axial elastic deformation of thrust ball bearing

$$\delta_b = 2.4 \left(\frac{Q^2}{D_b} \right)^{\frac{1}{3}} \quad (\mu\text{m}) \dots\dots\dots 26$$

$$Q = \frac{P}{Z}$$

Where,

δ_b:Axial elastic deformation (μm)

α:Contact angle

D_b:Steel ball diameter (mm)

Q:Load per steel ball (N)

Z:Number of steel balls

P:Axial load (N)

•Rigidity of roller bearing

$$K_b = \frac{2.16P_L}{\delta_b} \quad (\text{N}/\mu\text{m}) \dots\dots\dots 27$$

Where,

P_L:Preload (N)

δ_b:axial elastic deformation to preload (μm)

Axial elastic deformation of tapered roller bearing

$$\delta_b = \frac{0.6}{\sin\alpha} \cdot \frac{Q^{0.9}}{\ell^{0.8}} \quad (\mu\text{m}) \dots\dots\dots 28$$

$$Q = \frac{P}{Z \sin\alpha}$$

Where,

δ_b:Axial elastic deformation (μm)

α:Contact angle

Q:Load per roller (N)

Z:Number of rollers

P:Axial load (N)

ℓ :Effective contact length of roller (mm)

•Rigidity of nut mounting portion and bearing mounting portion“K_n”

In designing these portions, pay due regard to the thickness and the distance from the mounting surface to the ball screw shaft center, so that the rigidity of the nut bracket and the bearing box may be improved.

When the distortion due to tension of the set bolt is not negligible, use the mounting method shown in Fig. 17 on Page 404.

•Torsional rigidity of screw shaft

The screw shaft is twisted around the axial line by the torsional moment (driving torque), resulting in rotational strain. The torsional deformation can be expressed as the axial deformation of the ball screw by the following formula:

$$\delta_T = l \theta \frac{L}{2\pi} \dots\dots\dots(29)$$

$$\theta = \frac{32T}{\pi d^4 G} \dots\dots\dots(30)$$

Where,

- δ_T : Axial deformation caused by torsion (cm)
- l : Distance between working points (cm)
- θ : Angle of torsion (rad/cm)
- L : Lead of ball screw (cm)
- T : Torsional moment (N·cm)
- d : Root diameter of screw shaft (cm)
- G : Modulus of rigidity ($83 \times 10^5 \text{N/cm}^2$)

If the angle of torsion for the driving shaft is excessively large, the parts of the driving mechanism may be caused with torsional vibration of the shaft system.

For ordinary driving shafts, the angle of torsion due to the maximum operating torsional moment should be set within 4.36×10^{-5} (rad/cm).

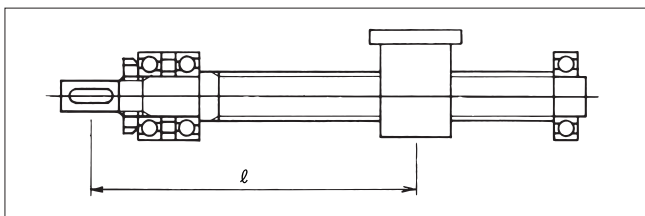


Fig. 20

TECHNICAL DATA OF BALL SCREWS

•NOTES FOR POSITIONING ACCURACY

This paragraph deals with how to select accuracy grade, how to determine specified travel, and how to take effective measures against thermal strain which will exert a great influence upon the positioning accuracy.

•Selection guide for the accuracy grade classified according to the type of machine

Select the accuracy grade of the ball screw meeting to the required positioning accuracy from Tables 2 and 3 on Page 389.

KURODA recommends to select the accuracy grade in accordance with the following table.

Table 21 Examples of accuracy grade of ball screw recommendable according to type of machine

Type of machine	NC machine tools								Industrial robots				
	Machining centers Milling machines		Lathes		Grinding machines		Electric discharge machines		Punching machines	wood working machines (NC routers)	Cartesian coordinates type (Assembly)	Vertical revolute type (Assembly)	SCARA type (Assembly)
	XY	Z	X	Z	X	Z	XY	Z					
Accuracy grade	C1 -C3	C2 -C5	C1 -C3	C3 -C5	C0 -C2	C1 -C3	C1 -C3	C2 -C5	C3-C5	C5-C7	C1-C5	C2-C5	C3-C5

Type of machine	Semiconductor manufacturing equipment				Printing equipment		Business equipment		
	Exposure system, Drawing system	Etching equipment, Ion implanting equipment	Wire bonders, Die bonders	Wafer probers	Electrical parts charging apparatus, Insert machines	Electronic color separating apparatus	Electronic composing machines	Color graphic printers	XY plotters, auto drawing machines
Accuracy grade	C0-C1	C3-C7	C1-C2	C0-C2	C2-C7	C0-C2	C1-C3	C1-C3	C1-C3

•Determining the specified travel

In general, the specified lead of a ball screw means the nominal lead. However, the specified lead of the screw shaft is sometimes set to the negative side or the positive side to correct the expansion due to temperature rise during operation or the expansion or contraction of the screw shaft due to the external load. In such a case, inform KURODA of the target value of the specified travel. The typical target values of the specified travel classified according to the type of machine are shown in Table 22 below. To correct the expansion or expansion, a tensile load may sometimes be applied to the screw shaft when mounting.

•Measures to be taken for thermal displacement

As the ball screw is so constructed that its rolling motion involves a slight sliding motion, it inevitably suffers a thermal strain due to temperature rise. The temperature rise is closely related to the operating conditions. The expansion of the thermal displacement can be expressed by the following formula:

$$\Delta l = \rho t l \quad (\text{mm}) \dots \dots \dots \textcircled{31}$$

Where,

- Δl : Axial thermal displacement (mm)
- ρ : Coefficient of thermal expansion ($11.7 \times 10^{-6} \text{ } ^\circ\text{C}^{-1}$)
- t : Temperature rise of screw shaft ($^\circ\text{C}$)
- l : Effective thread length (mm)

Table 22 Target values of specified travel classified according to type of machine (Unit:mm)

Machine type	Axis	Target value of specified travel (per meter)
NC lathes	X	-0.02~0.05
	Z	-0.02~0.03
Machining centers	X,Y	-0.03~0.04