(5) Effects of heavy load and short stroke

If the ball screw is used under heavy load and short strokes, such as for drive of plastic injection molding machine and of press machines, the fatigue life may become significantly shorter than the rated fatigue life which is calculated in B-2-5.2.

This decreased life occurs because the heavy load generates large stress (surface pressure) in the contact point of balls and ball grooves of the screw shaft and the nut, adversely affecting the life. In such case, the life calculation should take into account the size of the surface pressure as well as the size of the stroke.

The axial load F_{amax}[*1] during operation and the size of strokes, which affect fatigue life, can be obtained by the following formula.

In such case, the life calculation should take into account the size of the surface pressure as well as the size of the stroke. Please consult with NSK.

$$F_{amax} \geq 0.10C_{0a} \qquad \cdots (\text{II-16})$$

$$S \leq 4$$

In this formula:

F_{amax} : Maximum load to axial direction during drive (N)

C_{0a} : Basic static load rating (N)

S : Stroke (rev)

$$S = \frac{L_s}{l}$$

L_s : Stroke distance (mm)

l : Lead (mm)

* Axial load : The load is applied to the axial direction when screw shaft and the nut of ball screw are rotating relatively each other. The rotational speed is irrelevant.

B-2-5.3 Ball screw and Hardness

Table 5.4 indicates NSK standard ball screw and their hardness.

Table 5.4 Ball screw materials and their hardness

Component	Heat treatment method	Hardness (HRC)
Screw shaft	Carburizing	58 or over
	Induction hardening	58 or over
Nut	Carburizing	58 or over

* NSK manufactures special material ball screws for special environments (stainless steel: SUS440C, SUS630). NSK also furnishes surface treatment (Refer to Page D5). Please consult NSK for such request.

B-2-5.4 Wear Life

Wear of materials, as is the case for other mechanical components, is significantly affected by use conditions, lubrication conditions and other factors. It is difficult to estimate its volume, and measuring requires various tests and field data.

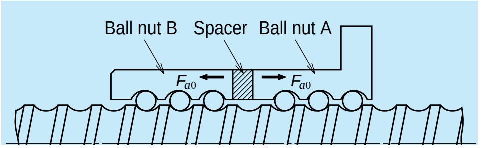
NSK has data of wear accumulated through abundant experience. Please contact NSK for inquiry pertaining to the wear.

B-2-6 Preload and Rigidity

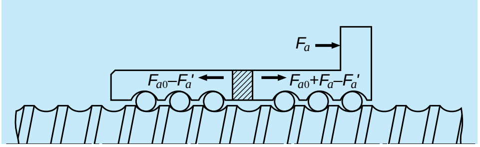
B-2-6.1 Elastic Deformation of the Preloaded Ball Screw

(1) Position preload (D, Z, P preloads)

Double nut preload ball screw shown in Fig. 6.1.



(a) Extemal load : 0



(b) Extemal load : F_a

Fig. 6.1 Position preload (double-nut)

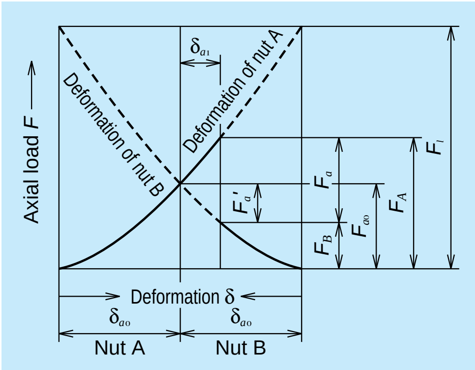


Fig. 6.2 Deformation of A and B nut (position preload)

Elastic deformation of Nut A and B is already given at time of assembly by the amount of δao by preload F_{ao}. When the external load F_a is added to Nut A, the elastic deformation δ_a and δ_b of each Nut A and B change as shown in Fig. 6.2,

$$\delta_a = \delta_{ao} + \delta_{a1} \qquad \delta_b = \delta_{ao} - \delta_{a1}$$

At this time, the load to each Nut A and B are:

$$F_A = F_{ao} + F_a - F_a'$$

$$F_B = F_{ao} - F_a'$$

It shows that the load applied to Nut A is affected by Nut B and reduced by the amount of F_a'. Thereby, the elastic deformation of Nut A becomes smaller. This effect continues until the elastic deformation by the external load becomes δ_{ao}, and the preload by Nut B disappears.

Assuming that the load when the preload is absorbed is F_l, the relationship between the axial load and the elastic deformation is as follows. (Fig. 6.2)

$$\delta_{ao} = K \bullet F_{ao}^{2/3} \qquad 2\delta_{ao} = K \bullet F_l^{2/3}$$

(K : Invariable number)

$$\left[\frac{F_l}{F_{ao}}\right]^{2/3} = \frac{2\delta_{ao}}{\delta_{ao}} = 2$$

$$F_l = 2^{3/2} \times F_{ao} \fallingdotseq 3F_{ao}$$

For this reason, the preload should be about 1/3 of the maximum axial load. Please note that the preload of about 1/3 of the maximum axial load increases heat, and shortens life if it exceeds 10% of C_a. The criterion for the maximum preload is 0.1 C_a.

Fig. 6.3 shows two types of elastic deformation curves: one is by the ball screw with preload, the other without preload. When an axial load which is about three times as large as the preload is applied, the deformation of the preloaded ball screw is 1/2 of the deformation of the ball screw without preload.

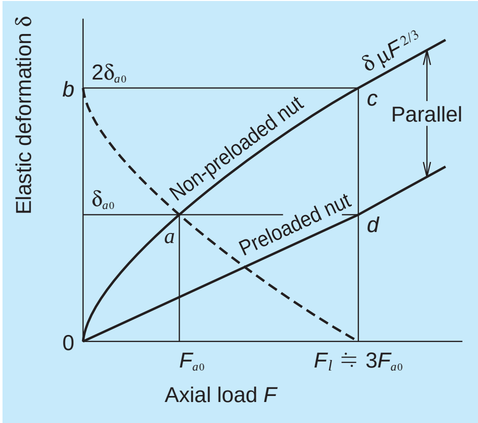


Fig. 6.3 Deformation of preloaded ball nut (position preload)

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60

(2) Constant pressure preload (J preload: preloaded by spring)

Fig. 6.5 shows an elastic deformation of the ball screw which is preloaded with "constant pressure." The rigidity of the preload spring is sufficiently smaller than the nut rigidity. Therefore, the deformation of the spring becomes nearly parallel to the axis of abscissa. For this reason, the elastic deformation by the preload with constant pressure changes along the deformation curve by Nut A.

In order to take advantage of the characteristics of the preload with constant pressure, the major external load should be applied in the directions shown by arrows (Fig. 6.4.).

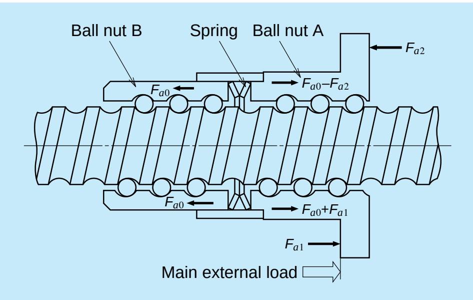

Fig. 6.4 Constant pressure preload (double nut)

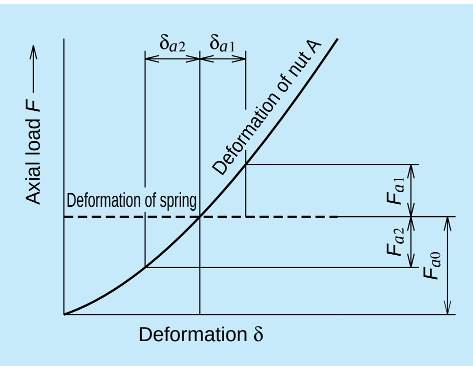

Fig. 6.5 Deformation curve of constant pressure preloaded nut

B-2-6.2 Rigidity of the Feed Screw System

A low rigidity around the feed screw mounting area causes lost motion. To improve the positioning accuracy of precision machines such as NC machine tools, it requires a good balance in axial rigidities of composing parts of the feed screw system.

Also should examine torsional rigidities of the feed screw system.

(1) Axial rigidity of the feed screw system K_T

Elastic deformation and rigidity of the feed screw system can be obtained by the following formula.

$$\delta = \frac{F_a}{K_T} \quad \text{(II-17)}$$

$$\frac{1}{K_T} = \frac{1}{K_S} + \frac{1}{K_N} + \frac{1}{K_B} + \frac{1}{K_H} \quad \text{(II-18)}$$

In this formula:

- δ : Volume of axial elastic deformation of the feed screw system (μm)
- F_a : Axial load to the feed screw system (N)
- K_T : Axial rigidity of the feed system (N/μm)
- K_S : Axial rigidity of the screw shaft (N/μm)
- K_N : Axial rigidity of the nut (N/μm)
- K_B : Axial rigidity of the support bearing (N/μm)
- K_H : Axial rigidity of the nut and bearing mounting section (N/μm)

(2) Axial rigidity of the screw shaft: K_S

(a) In case of: Fixed support - Free (axial direction)

$$K_S = \frac{A \bullet E}{x} \times 10^{-3} \quad \text{(II-19)}$$

In this formula:

- K_S : Axial rigidity of the screw shaft (N/μm)
- A : Cross section area of the screw shaft (mm^2)

$$A = \frac{\pi}{4}\, dr^2$$

- dr : Screw shaft root diameter (mm)
- E : Elastic modulus ($E = 2.06 \times 10^5$ MPa)
- x : Distance between points of load application (mm)

(b) In case of: Fixed – Fixed support (axial direction)

$$K_S = \frac{A \bullet E \bullet L}{x\,(L - x)} \times 10^{-3} \quad \text{(II-20)}$$

In this formula:

- K_S : Axial rigidity of the screw shaft (N/μm)
- L : Unsupported length (mm)
- x : Axial deformation is maximum at position $x = L/2$.

Axial rigidity of the screw shaft can be obtained by the following formula.

$$K_S = \frac{4A \bullet E}{L} \times 10^{-3} \quad \text{(II-21)}$$

<<Axial rigidity example of calculation (1)>>

Obtain axial rigidity of the screw shaft under the condition in Fig. 6.6.

<Use conditions>

Nut model: DFT 4010-5

From Fig. 6.6: Supporting condition ;

Fixed support --Free (axial direction)

Distance between points of load application

x = 1200 mm

Screw shaft root diameter (From the dimension table)

d_r = 34.4 mm

<Calculation>

By Formula II-19, axial rigidity K_S is :

$$A = \frac{\pi}{4}\, d_r^2 = \frac{3.14}{4} \times 34.4^2 = 929.4 \ (\text{mm}^2)$$

$$K_S = \frac{A \bullet E}{x} \times 10^{-3} = \frac{929.4 \times 2.06 \times 10^5}{1200} \times 10^{-3} = 159 \ (\text{N/μm})$$

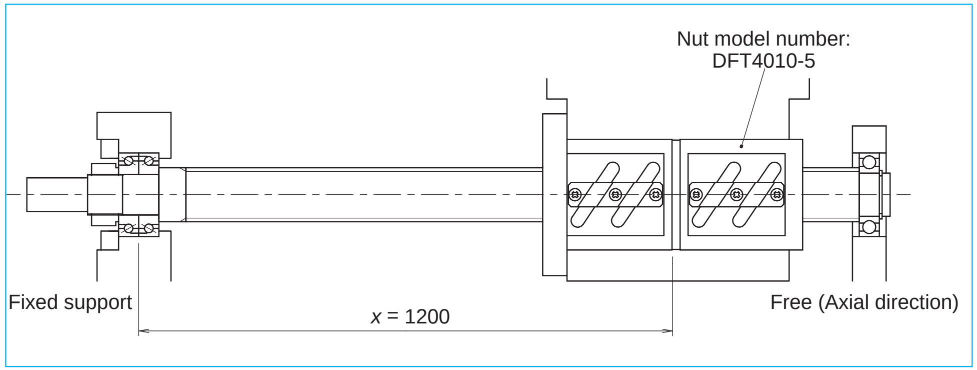

Fig. 6.6 Axial rigidity of the screw shaft calculation example (1)

B 62

<<Axial rigidity example of calculation (2)>>

Obtain axial rigidity of the screw shaft under the conditions in Fig. 6.7.

<Use conditions>

Nut model: DFT 4010-5
From Fig. 6.7: Supporting condition:
Fixed - Fixed support (axial direction)
L = 1200 mm
Distance between points of load application:
Screw shaft root diameter (From the dimension table)
dr = 34.4 mm

<Calculation>

By Formula Ⅱ-21, axial rigidity K_S is :

$$A = \frac{\pi}{4} dr^2 = \frac{3.14}{4} \times 34.4^2 = 929.4\ (\text{mm}^2)$$

$$K_S = \frac{4A \cdot E}{L} \times 10^{-3} = \frac{4 \times 929.4 \times 2.06 \times 10^5}{1200} \times 10^{-3} = 638\ (\text{N}/\mu\text{m})$$

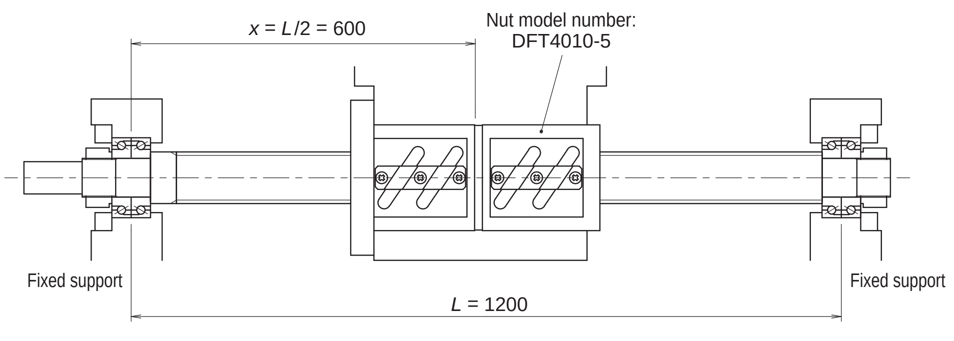

Fig. 6.7 Axial rigidity of the screw shaft calculation example (2)

(3) Axial rigidity of the ball nut : K_N

(a) Rigidity of the nut with axial play

Theoretical rigidity value K is shown in the dimension table. K is obtained from the elastic deformation between screw grooves and balls when an axial load which is equivalent to 30% of the basic dynamic load rating C_a is applied. The criterion for calculation of ball nut rigidity is 80% of the value listed in the table taking into consideration of deformation of the ball nut, etc. Rigidity value K_N is obtained by the following formula when the axial load "F_a" is not 30% of "C_a."

$$K_N = 0.8 \times K \left(\frac{F_a}{0.3\, C_a} \right)^{1/3} (\text{N}/\mu\text{m}) \qquad (\text{Ⅱ-22})$$

In this formula:

K : Rigidity value in dimension tables (N/μm)
F_a : Axial load (N)
C_a : Basic dynamic load rating (N)

<<Axial rigidity example of calculation (3)>>

Obtain axial rigidity of the nut under the following conditions.

<Use conditions>

Nut model: SFT 4010-5
Axial load: F_a = 6000 N
F_a = Rigidity at 0.3 C_a K = 706 N/μm
(From the dimension table)

<Calculation>

By Formula Ⅱ-22, axial rigidity K_N is :

$$K_N = 0.8 \times K \left(\frac{F_a}{0.3 \cdot C_a} \right)^{1/3}$$

$$= 0.8 \times 706 \times \left(\frac{6000}{0.3 \times 52000} \right)^{1/3}$$

$$= 410\ (\text{N}/\mu\text{m})$$

(b) Rigidity of preloaded ball nut

Theoretical rigidity K is shown in each dimension table. K is obtained from the elastic deformation of the ball rolling surface and the balls when: a preload which is equivalent to 10% of the basic dynamic load rating C_a (P Preload. 5% for single-nut oversize ball pre-load system) is applied, followed by an axial load. The criterion for calculation of nut rigidity is 80% of the value listed in the table taking into consideration of deformation of the ball nut, etc. Rigidity K_N is obtained by the following formula when preload "F_{a0}" is not 10% (or 5%) of "C_a".

$$K_N = 0.8 \times K \left(\frac{F_{a0}}{\varepsilon \cdot C_a} \right)^{1/3} (\text{N}/\mu\text{m}) \qquad (\text{Ⅱ-23})$$

In this formula:

K : Rigidity in the dimension tables (N/μm)
F_{a0} : Preload (N)
ε : Basic factor to calculate rigidity (ε = 0.1. Use 0.05 for P Preload)

<<Axial rigidity of the screw shaft calculation example (4)>>

Obtain axial rigidity of the nut under the following conditions.

<Use conditions>

Nut model : DFT 4010-5
Preload : F_{a0} = 4000 N
F_{a0} = Rigidity when ε C_a : K = 1388 N/μm
(From the dimension table)
Basic factor to calculate rigidity when D Preload: ε = 0.1

<Calculation>

By Formula Ⅱ-23

$$K_N = 0.8 \times K \left(\frac{F_{a0}}{\varepsilon \cdot C_a} \right)^{1/3}$$

$$= 0.8 \times 1388 \times \left(\frac{4000}{0.1 \times 52000} \right)^{1/3}$$

$$= 1017\ (\text{N}/\mu\text{m})$$

The criterion of the preload to ball screw

Nut rigidity increases by a larger preload volume. But excessive preload shortens life, and generates heat. Set the maximum preload about at 0.1 C_a (0.05 for *P* Preload). Table 6.1 shows the criteria for preload for different application.

Table 6.1 Criteria of preload

Ball screw application	Preload (relative to dynamic load rating C_a)
Robots,material handling systems, etc.	Axial play or under 0.01 C_a
Semiconductor manufacturing systems, etc. That require highly accurate positioning	0.01 C_a – 0.04 C_a
Medium- high-speed machine tools for cutting	0.03 C_a – 0.07 C_a
Low to medium-speed systems that require especially high rigidity	0.07 C_a – 0.1 C_a

(4) Axial rigidity of support bearing: K_B

Rigidity of the combined thrust angular contact ball bearings which is widely used as a support bearing of the ball screw for high-precision equipment can be obtained by the following formula.

$$K_B \fallingdotseq \frac{3F_{a0}}{\delta_{a0}} \text{ (N/µm)} \qquad (\text{II-24})$$

In this formula:

K_B : Rigidity of the combined thrust angular contact ball bearings (N/µm)
F_{ao} : Preload of the bearings (N)
δ_{ao} : Axial elastic deformation by preload (µm)

$$\delta_{a0} \fallingdotseq \frac{0.44}{\sin\alpha}\left(\frac{Q^2}{D_w}\right)^{1/3} \text{ (µm)} \qquad (\text{II-25})$$

$$Q = \frac{F_{a0}}{Z} \cdot \sin\alpha$$

α : Contact angle
D_w : Ball diameter (mm)
Z : Number of balls

Refer to Page B457 for data regarding thrust angular contact ball bearings which support high-precision ball screws (TAC Series).

(5) Axial rigidity of the ball nut and bearing mounting section: K_H

The effect of rigidity of mounting section on positioning accuracy is big, we recommend incorporating high rigidity of the mounting sections of ball nut and support bearings into the design at the early stage of designing the machine.

(a) Torsional rigidity of the feed screw system

Major torsion factors in the rotating system that bring about error in positioning accuracy are given three points below.

- Torsional deformation of the screw shaft
- Torsional deformation of the joint section
- Torsional deformation of the motor

The value of the effect of torsional strain to positioning accuracy is smaller than axial deformation. However, check the effect when designing equipment that requires high positioning accuracy.

(b) Suppress thermal error

It is necessary to minimize the thermal error for ever increasing demand for positioning accuracy give three points below.

- Suppress heat
- Forced cooling
- Avoid effect of temperature rise

Refer to "Measures against thermal expansion" on Page B44.

B-2-7 Friction Torque and Drive Torque

Operations that use ball screw drives require a motor torque which is equivalent to the total of two:

- Friction torque, i.e. the friction of the ball screw itself
- Drive torque which is required for operation

B-2-7.1 Friction Torque

(1) Starting friction torque (Break away torque)

A large torque is necessary to start ball screw. This is called "starting friction torque" or "brakeaway torque." This torque is 2 to 2.5 times larger than preloaded dynamic (friction) torque which is described below. Starting friction torque quickly diminishes once the ball screw begins to move.

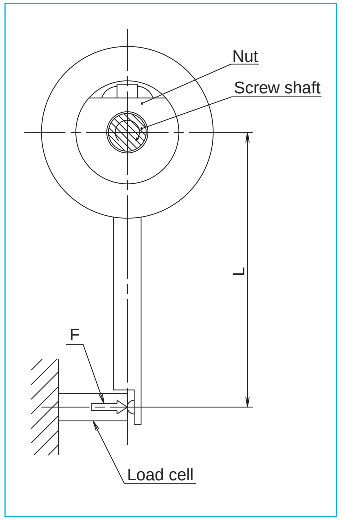

Fig. 7.2 Preload dynamic torque measuring method

(2) Dynamic preloaded drag torque (preloaded dynamic friction torque)

When the ball screw is moving, two types of torque generate: 1. Dynamic friction torque by preload; 2. Friction torque associated with ball recirculation. JIS B1192 sets standard of dynamic preloaded torque, which is the total of these two torque types. They are defined in Fig. 7.1.

The preload dynamic friction torque is calculated by following formula. When screw shaft is rotated as Fig. 7.2 in following measure condition, measuring the nut stop power *F* and the distance from action line and right angle direction to the measured screw shaft multiple by it's power value *F*.

$$T_p = F \cdot L \qquad (\text{II-26})$$

- Measuring rotational speed 100 min^{-1}
- Viscosity of lubrication is prescribed in JIS K 2001 ISO VG 68
- Without measurement Seal

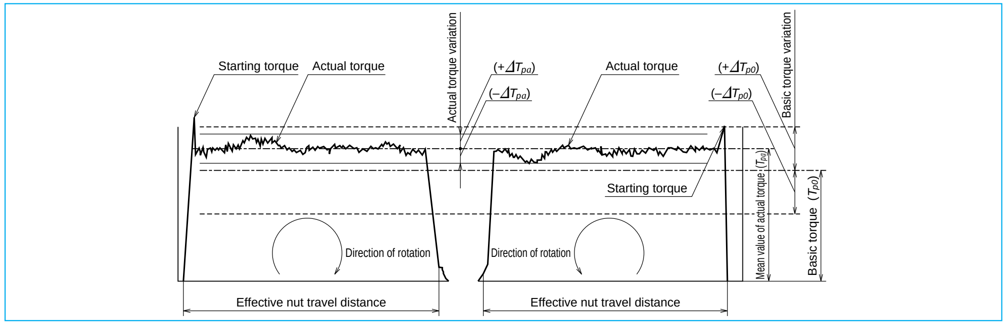

Fig. 7.1 Definitions of dynamic preloaded drag torque