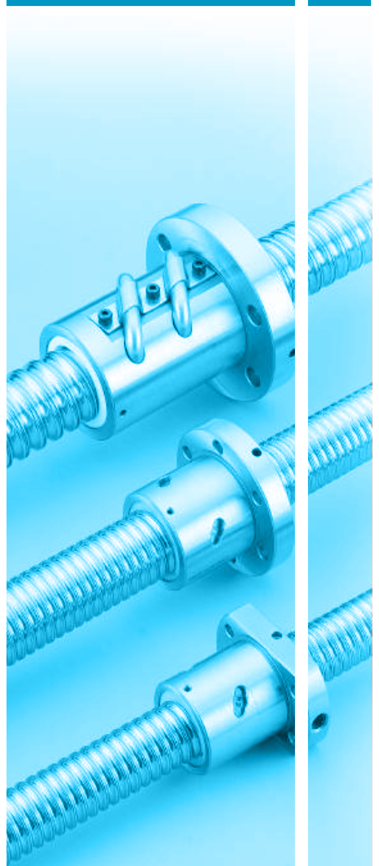
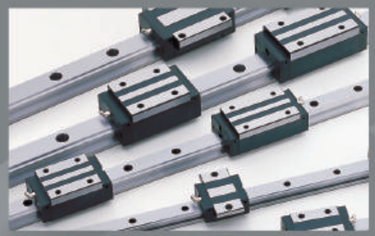
index

GTEN BALL SCREW TECHNOLOGY CO., LTD.





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BALL SCREW

GTEN is THE LEADERSHIP of BALL SCREW

GTEN BALL SCREW TECHNOLOGY CO., LTD.

1. Company Profile

GTEN Ball Screw Technology was established on December 2004.The manufacturing plant of GTEN is over 10,000 square feeter. The main goal of GTEN Ball Screw Technology is to develop high-precision rolled ball screw. Up to now, we can produce and reach C5 grade (JIS Standard). We already can keep pace with the same production level as main ball screw manufacture in the world. Due to the high production cost abroad, GTEN have a competition advantage at prices.

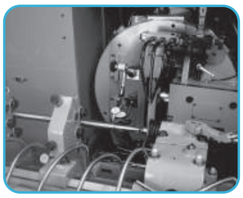
All of our products are designed by our own R&D team and sold with our own brand to the market. All products have already passed the multinational patent and authentication. We already sold our product abroad, including Europe, Asia and America countries. Based good public praise from our customer and company's high-quality management, GTEN has become a reliable business partner in the market.

Operation vision

The goal of personnel in GTEN

- Good Product Quality
- Good Skill training
- Good Service
- Good R&D
- Good Efficiency
- Good Productivity
- Good Collaboration
- Good Management
- Good Communication
- Good Team Work

Under complete experience and professional leadership of our general manager, Mr. Levite Lee, we promote machinery equipment and professional manufacturing capacity constantly. In the same time, we also work diligently to expand the popularity in the market in order to stand firmly on the leading position in Taiwan in the near future.



2. Technological Description of Ball Screws

2.1 Accuracy

2.1.1 Lead/Travel Accuracy

- Lead accuracy of **GTEN** ball screws (grade C0~C5) is specified in 4 basic terms (E,e,e_{300}, $e_{2\pi}$). There are defined in Fig. 2.1 Tolerance of deviation (±E) and variation (e) of accumulated reference travel are shown in Table 2.2 and 2.3.

- Accumulated travel deviations for grade C7 and C10 are specified only by the allowable value per 300mm measured within any portion of the thread length. They are 0.05mm for C7 and 0.21mm for C10.

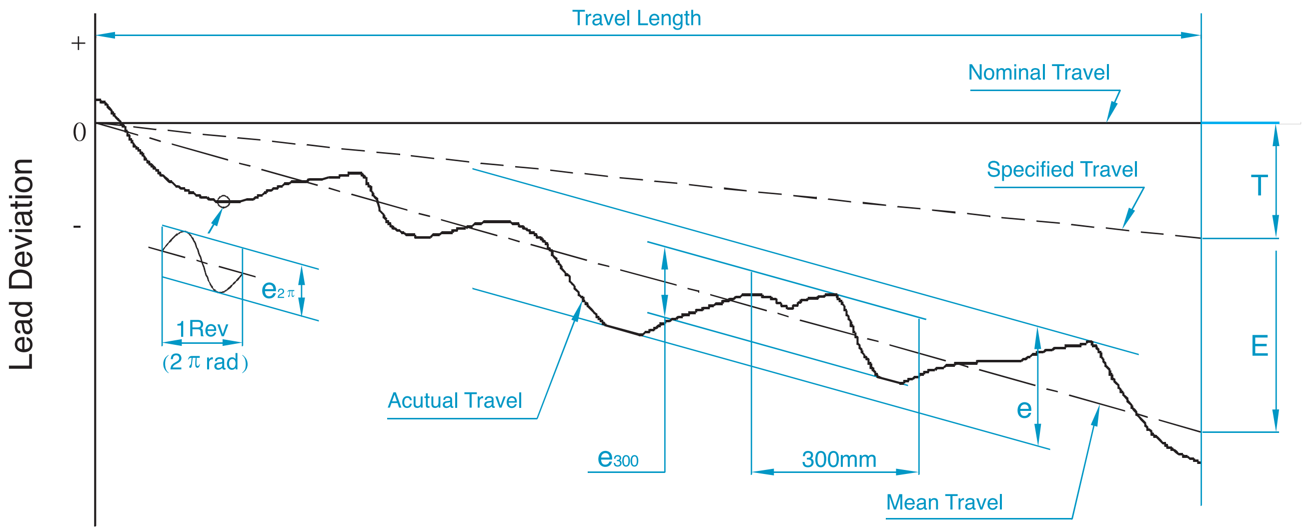


Fig. 2.1 Diagram of Lead Accuracy

Table 2.1 Definition of Terms for Lead Accuracy

Terms	Reference	Definition	Allowable
Travel Compensation	T	Travel compensation is the difference between specified and nominal travel within the useful travel. A slightly smaller value compared to the nominal travel is often selected by the customer to compensate for an expected elongation caused by temperature rise or external load. Therefore 〝T〞 is usually a negative value. Note : if no compensation is needed , specified travel is the same as nominal travel.	
Actual Travel		Actual travel is the axial displacement of the nut relative to the screw shaft.	
Mean Travel		Mean travel is the linear best fit line of actual. This could be obtained by the least quares method. This line represents the tendency of actual travel.	
Mean Travel Deviation	E	Mean travel deviation is the difference between mean travel and specified travel within travel length.	Table 2.2
Travel Variations	e e_{300} $e_{2\pi}$	Travel variations is the band of 2 lines drawn parallel to the mean travel , on the plus and minus side. Maximum width of variation over the travel length. Actual width of variation for the length of 300mm taken anywhere within the travel length. Wobble error , actual width of variation for one revolution (2π radian)	Table 2.2 Table 2.3 Table 2.3

Table 2.2 Mean Travel Deviation(±E)and Travel Variation(e) (JIS B 1192)

	Grade		C0		C1		C2		C3		C5		C7	C10
Travel Length(mm)	Over	Incl.	±E	e	±E	e	±E	e	±E	e	±E	e	e	e
		100	3	3	3.5	5	5	7	8	8	18	18		
	100	200	3.5	3	4.5	5	7	7	10	8	20	18		
	200	315	4	3.5	6	5	8	7	12	8	23	18		
	315	400	5	3.5	7	5	9	7	13	10	25	20		
	400	500	6	4	8	5	10	7	15	10	27	20		
	500	630	6	4	9	6	11	8	16	12	30	23		
	630	800	7	5	10	7	13	9	18	13	35	25		
	800	1000	8	6	11	8	15	10	21	15	40	27		
	1000	1250	9	6	13	9	18	11	24	16	46	30		
	1250	1600	11	7	15	10	21	13	29	18	54	35	±50/300mm	±210/300mm
	1600	2000			18	11	25	15	35	21	65	40		
	2000	2500			22	13	30	18	41	24	77	46		
	2500	3150			26	15	36	21	50	29	93	54		
	3150	4000			30	18	44	25	60	35	115	65		
	4000	5000					52	30	72	41	140	77		
	5000	6300					65	36	90	50	170	93		
	6300	8000							110	60	210	115		
	8000	10000									260	140		
	10000	12500									320	170		

Table 2.3 Variation per 300mm(e_{300})and Wobble Error($e_{2\pi}$) (JIS B 1192)

Unit：μm

Grade	C0	C1	C2	C3	C5	C7	C10
e_{300}	3.5	5	7	8	18	50	210
$e_{2\pi}$	3	4	5	6	8		

2.2 Axial Play

GTEN Axial Direction of Standard Backlash and Preload

Table2.4 Clearance in the Axial Direction of Ball Screw (P0)

Clearance in the Axial Direction of Ball Screw		Unit: mm
Screw Shaft OD	Rolled Ball Screw Clearance in the Axial Direction (max.)	Ground Ball Screw Clearance in the Axial Direction (max.)
4mm~14mm	0.05	0.015
15mm~50mm	0.08	0.025
50mm~80mm	0.12	0.05

Table2.5 Clearance in the Axial Direction (P1)

Clearance in the Axial Direction of Ball Screw		Unit: mm
Screw Shaft OD	Rolled Ball Screw Clearance in the Axial Direction (max.)	Ground Ball Screw Clearance in the Axial Direction (max.)
4mm~80mm	0	0

Table2.6 Spring Force of Internal Circulation

Spring Force of Internal Circulation (kgf.cm)						
Model No	P2		P3		P4	
	3%Spring Force	TP Reference Torque	8%Spring Force	TP Reference Torque	13%Spring Force	TP Reference Torque
1404-4	0.1	0.13	0.2	0.34	0.3	0.56
1604-3	0.1	0.17	0.3	0.45	0.5	0.73
1604-4	0.1	0.21	0.3	0.57	0.5	0.93
1605-3	0.2	0.29	0.4	0.79	0.7	1.28
1605-4	0.2	0.3	0.4	0.8	0.7	1.3
1610-3	0.2	0.39	0.5	1.04	0.9	1.69
2005-4	0.2	0.47	0.5	1.26	0.9	2.05
2504-4	0.1	0.33	0.3	0.88	0.6	1.43
2505-4	0.2	0.6	0.6	1.6	1.0	2.59
2510-3	0.4	1.11	1.2	2.95	1.9	4.79
2510-4	0.6	1.47	1.2	3.93	2.5	6.38
3205-4	0.2	0.76	0.6	2.02	1.0	3.28
3206-4	0.3	1.14	0.8	3.03	1.3	4.93
3210-3	0.6	2.02	1.7	5.37	2.7	8.73
3210-4	0.8	2.62	2.2	6.99	3.5	11.36
4005-4	0.2	0.95	0.6	2.53	1.1	4.11
4006-4	0.3	1.25	0.9	3.32	1.4	5.4
4010-3	0.8	2.59	2.2	6.91	3.6	11.23
4010-4	0.8	3.31	2.3	8.84	3.7	14.36
5010-3	0.9	3.29	2.3	8.77	3.8	14.26
5010-4	0.9	4.21	2.4	11.23	3.9	18.25
6310-4	1.0	5.42	2.7	14.46	4.4	23.49
6320-3	2.3	13.08	6.1	34.87	9.9	56.66
8010-4	1.1	6.68	2.9	17.82	4.6	28.96
8020-3	2.3	16.87	6.2	44.98	10.1	73.1

Table2.7 Spring Force of Plastic Circulation (kgf.cm)

Spring Force of Plastic Circulation (kgf.cm)						
Model No	P2		P3		P4	
	2%Spring Force	TP Reference Torque	5%Spring Force	TP Reference Torque	8%Spring Force	TP Reference Torque
1210-2	0.1	0.12	0.1	0.2	0.2	0.32
1605-4	0.2	0.32	0.4	0.81	0.7	1.29
1610-3	0.1	0.26	0.3	0.65	0.5	1.04
1610-4	0.1	0.33	0.4	0.83	0.6	1.33
1616-3	0.2	0.44	0.6	1.09	0.9	1.75
2005-4	0.2	0.42	0.4	1.04	0.7	1.67
2505-4	0.2	0.52	0.5	1.29	0.8	2.07
2510-4	0.3	0.84	0.8	2.09	1.3	3.34
3205-4	0.2	0.79	0.6	1.98	1.0	3.17
3220-3	0.4	1.45	1.1	3.62	1.8	5.8
4005-4	0.3	1.19	0.8	2.98	1.2	4.77
4020-3	0.8	3.14	2.0	7.85	3.2	12.55
5010-4	0.7	3.47	1.9	8.66	3.0	13.86
5020-5	1.5	6.98	3.8	17.46	6.0	27.93
1616-2	0.2	0.33	0.4	0.83	0.7	1.3
2020-2	0.2	0.45	0.4	1.12	0.7	1.79
2525-2	0.3	0.88	0.7	2.2	1.2	3.52
3232-2	0.4	1.61	1.1	4.04	1.7	6.46
4040-2	0.7	3.3	1.8	8.24	2.8	13.18
5050-2	1.3	7.35	3.3	18.38	5.3	29.41

Table2.8 Spring Force of External Circulation (kgf.cm)

Spring Force of External Circulation (kgf.cm)						
Model No	P2		P3		P4	
	3%Spring Force	TP Reference Torque	8%Spring Force	TP Reference Torque	15%Spring Force	TP Reference Torque
082.5-2.5	0.1	0.05	0.1	0.08	0.1	0.13
1003-2.5	0.1	0.06	0.1	0.15	0.2	0.24
1204-3.5	0.1	0.13	0.3	0.34	0.4	0.55
1205-3.5	0.2	0.22	0.5	0.59	0.7	0.95
1605-2.5	0.2	0.28	0.5	0.73	0.7	1.19
1520-1.5	1.5	3.41	4.0	9.08	6.6	14.76
2010-2.5	0.2	0.7	0.6	1.88	1.0	3.05

2.3 Definition of Mounting Accuracy and Tolerances on Ball Screw

To use a ball screw properly dimensional accuracy and tolerances are most important.

GTEN will help you determine the tolerance factors as they are subject to change according to accuracy grade.

(1)Periphery run-out of the supporting part of the screw shaft to the screw groove.

(2)Concentricity of a mounting portion of the shaft to the adjacent ground portion of the screw shaft.

(3)Perpendicularity of the shoulders to the adjacent ground portion of the screw shaft.

(4)Perpendicularity of the nut flange to the axis of the screw shaft.

(5)Concentricity of the ball nut diameter to the screw groove.

(6)Parallelism of the mounting surface of a ball nut to the screw groove.

(7)Total run-out of the screw shaft to the axis of the screw shaft.

All **GTEN** ball screws are manufactured, inspected and guaranteed to be within specifications.

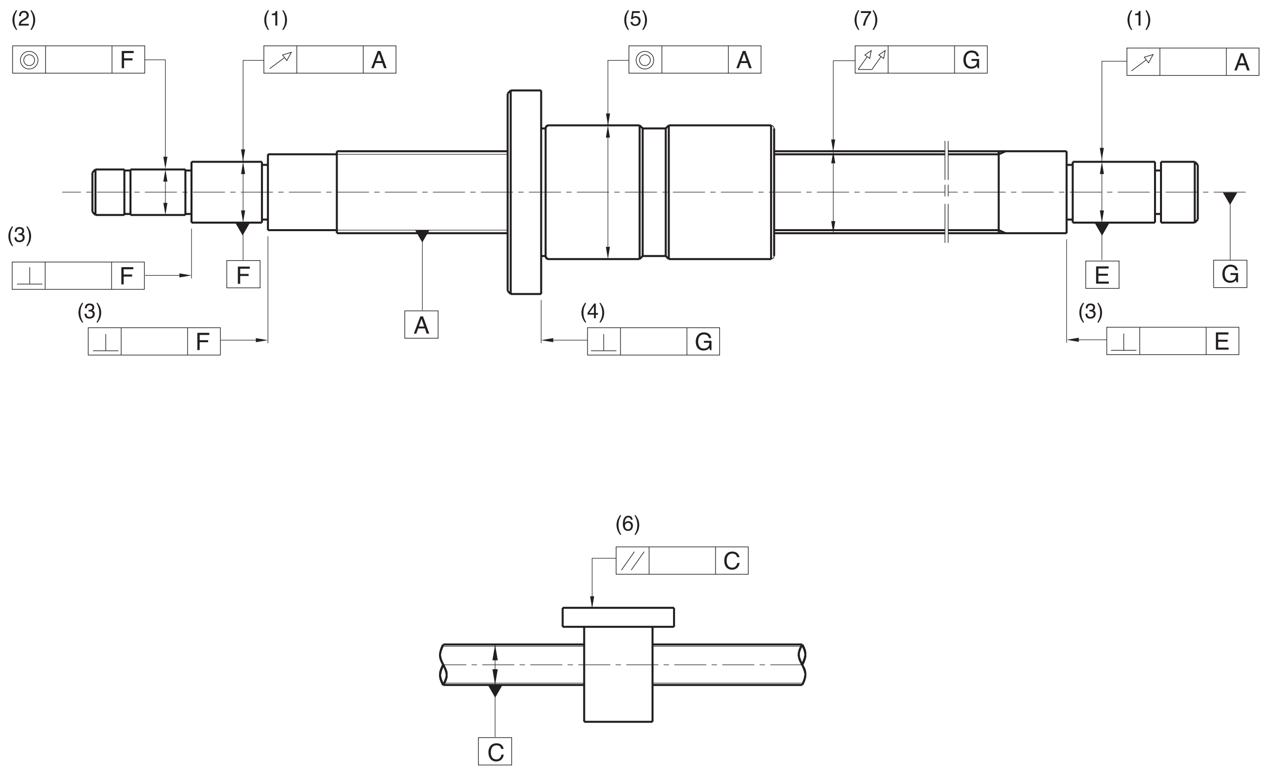


Fig. 2.2 Mounting Accuracy and Tolerances

2.4 Preload Torque

- Terms in relation to the preload torque generated during the rotation of the preload ball screws are shown in Fig. 2.3
- Permissible ranges of torque variation rates is shown in Table 2.6

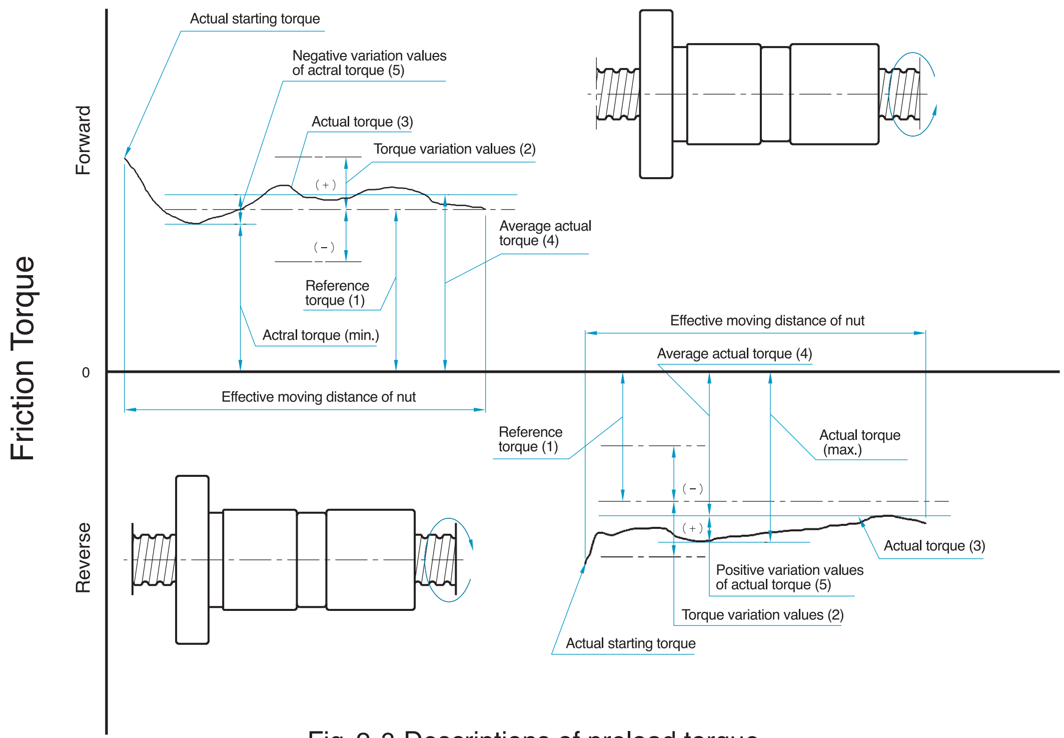


Fig. 2.3 Descriptions of preload torque

Glossary

(1) Preload

The stress generated inside the screws when inserting a set of steel balls of one gage (approximately 2 μ) larger into the nut or using them on the 2 nuts which exercise mutual displacements along the screws axis in order to eliminate the gaps of the screw or upgrade the rigidity of the screw.

(2) Preload dynamic torque

The dynamic torque required for continuously rotating the screws shaft or the nuts under unload condition after the specified preload has been applied upon the ball screws.

(3) Reference

The targeted preload dynamic torque【Fig.2.2-1】

(4) Torque variation values

The variation values of the targeted preload torque variation rates are specified generally based on JIS Standards as indicated in Table 3.5.

(5) Torque variation rate

The rate of variation values in relation to the reference torque.

(6) Actual torque

The actually measured preload dynamic torque of the ball screws.

(7) Average actual torque

The arithmetic average of the maximal and minimal actual torque values measured when the nuts are exercising reciprocating movements.

(8) Actual torque variation values

The maximal variation values measured within the effective length of the threads when the nuts are exercising reciprocating movements, the positive or negative values relative to the actual torque are adopted.

(9) Actual torque variation rate

The rate of actual torque variation values in relation to the average actual torque.

Table 2.9 Permissible ranges of torque variation rates

Reference torque kgf • cm		Effective threading length (mm)										
		Below 4000								4000～10000		
		Slenderness 1：below 40				Slenderness 1:40 ~ 1:60				–		
		Grade				Grade				Grade		
Over	Incl.	C0	C1	C2、C3	C5	C0	C1	C2、C3	C5	C1	C2、C3	C5
2	4	±35％	±40％	±45％	±55％	±45％	±45％	±55％	±65％	–	–	–
4	6	±25％	±30％	±35％	±45％	±38％	±38％	±45％	±50％	–	–	–
6	10	±20％	±25％	±30％	±35％	±30％	±30％	±35％	±40％	–	±40％	±45％
10	25	±15％	±20％	±25％	±30％	±25％	±25％	±30％	±35％	–	±35％	±40％
25	63	±10％	±15％	±20％	±25％	±20％	±20％	±25％	±30％	–	±30％	±35％
63	100	–	–	±15％	±20％	–	–	±20％	±25％	–	±25％	±30％

Remarks 1.Slenderness is the value of dividing the screws shaft outside diameter with the screws shaft threading length.
2.For reference torque less than 2 kgf • cm, **GTEN** specifications will apply.

Calculation of reference torque Tp

The formula for computing reference torque of the ball screws is given in following:

$$Tp = 0.05\,(\tan\beta)^{-0.5} \cdot \frac{Fao \cdot \ell}{2\pi}$$

Where，Fao：Preload (k f)

β ：Lead angle

ℓ ：Lead (cm)

Measurement conditions

The preload dynamic torque Tp is determined first by adopting the following measurement conditions together with the method illustrated in Diagram 3.4 for measuring the force F needed to rotate the screws shaft without bringing the nuts to rotate along with the shaft after the screws shaft has started rotating, then multiplying the measured value of F with the arm of force L, the product is Tp.

Tp=F • L

Measure conditions
(1) Measurement is executed under the condition of not attaching with scraper.
(2) The rotating speed during measurement maintains at 100 rpm.
(3) According to JSK 2001 (industrial lubrication oil viscosity classification standards), the lubrication oil used should be in compliance with ISO VG68.

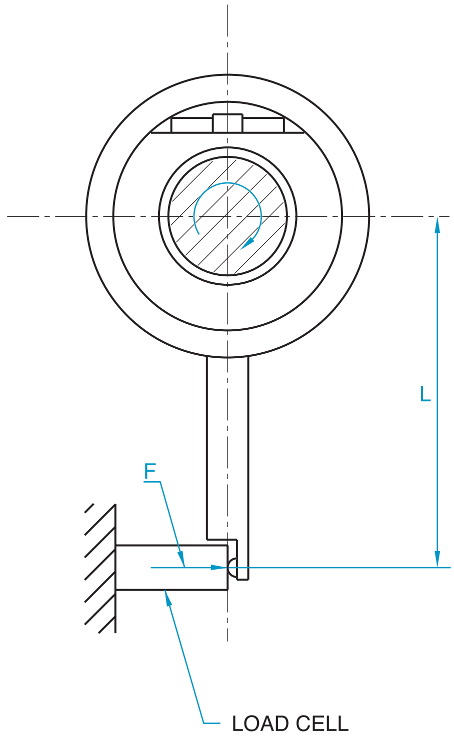


Fig. 2.4 Preload dynamic torque measuring method

3.1 Mounting Methods

- Both the critical speed and column bucking load depend upon the method of mounting and the unsupported length of the shaft,the most common mounting methods for ball screws are shown in Fig. 3.1~3.15.

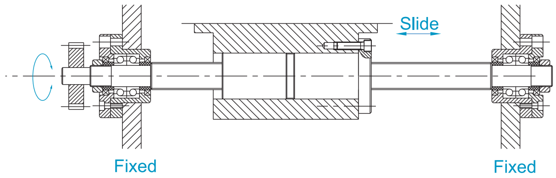


Fig. 3.1

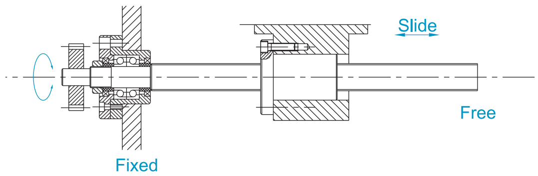


Fig. 3.2

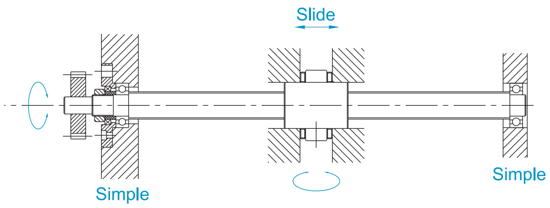


Fig. 3.3

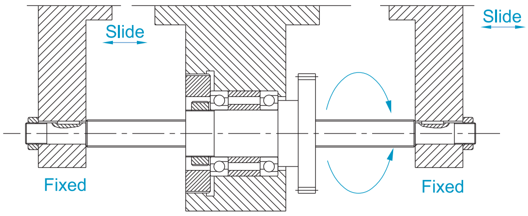


Fig. 3.4

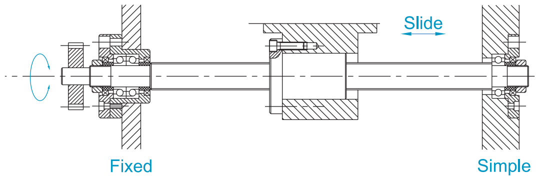


Fig. 3.5

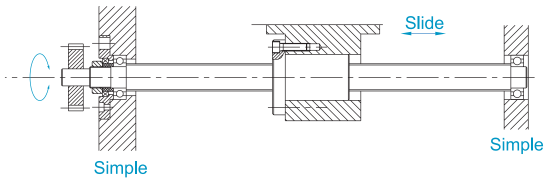


Fig. 3.6

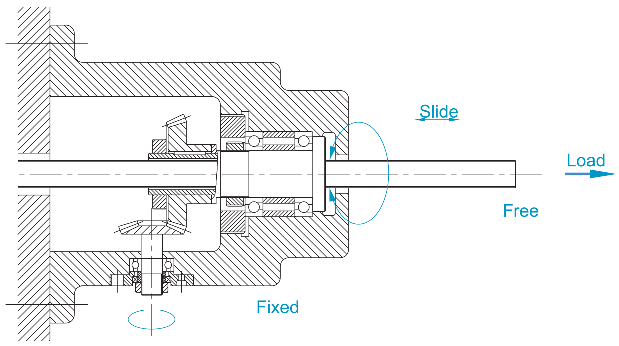


Fig. 3.7

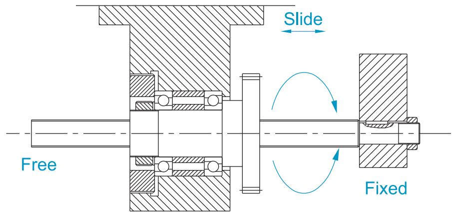


Fig. 3.8

Most Common Mounting Methods for Ball Screws

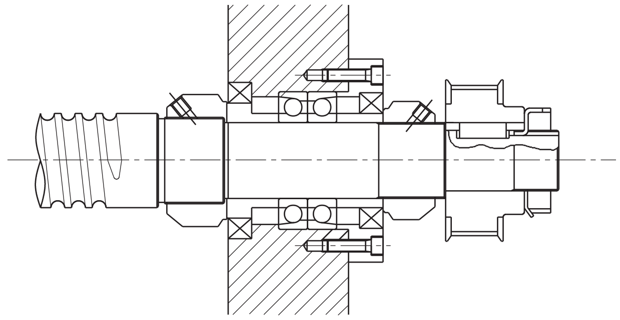


Fig. 3.9

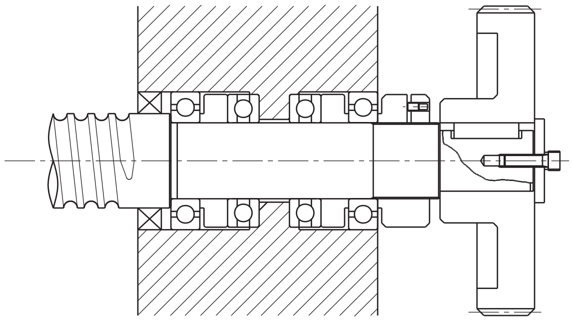


Fig. 3.11

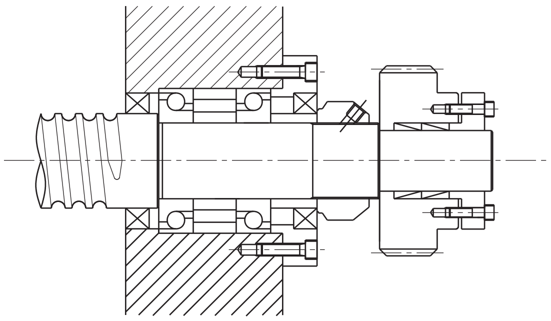


Fig. 3.10

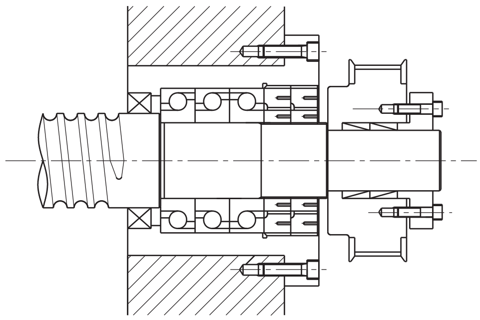


Fig. 3.12

Most Machines Mounting Methods for Ball Screws

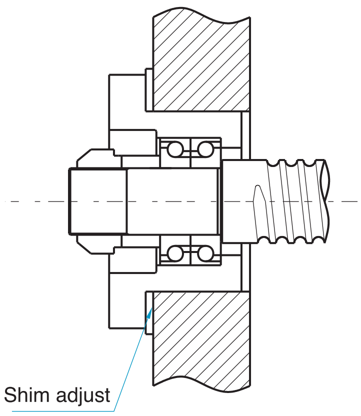


Fig. 3.17

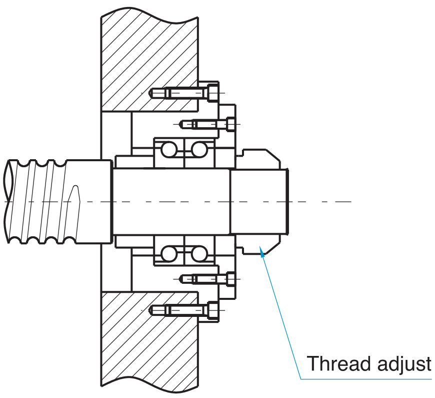


Fig. 3.18

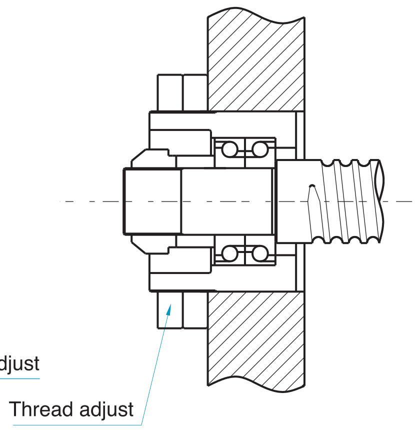


Fig. 3.19

Most Common Mounting Methods for Ball Screws

3.2 Buckling Load

The safety of the screw shaft against buckling needs to be checked when the shaft is expected to receive buckling loads. Fig. 3.16 shows a diagram which summarizes the allowable compressive load for buckling for each nominal outside diameter of screw shaft. (Calculate with the equation shown right when the nominal outside diameter of the screw shaft exceeds 125mm.)

Select the graduation of allowable axial load according to the method of ball screw support.

Remark: Allowable tensile / buckling load

Check the allowable tensile / buckling load (the formula shown below) and allowable load of the ball groove regardless of the mounting method when the mounting distance is short.

$$P=\sigma A=11.8dr^2 \ (kgf)$$

Where,

σ : Allowable tensile compressive stress (kgf / mm^2)

A : Sectional area (mm^2) of screw shaft root bottom diameter

dr : Screw shaft root diameter (mm)

$$P=\alpha\times\frac{N\pi^2 E}{L^2}=m\frac{dr^4}{L^2}\times 10^3$$

Where,

α : Safety Factor (0.5)

E : Vertical elastic modules ($E = 2.1\times10^4$ kgf/mm^2)

I : Min. secondary moment of screw shaft sectional area

$$I=\frac{\pi}{64}dr^4 \ (mm^4)$$

dr : Screw shaft root diameter (mm)

L : Mounting distance (mm)

m • N : Coefficient determined from mounting method of ball screw:

Simple - Simple m = 58.1 (N=1)

Fixed - Simple m = 10.2 (N=2)

Fixed - Fixed m = 20.3 (N=4)

Fixed - Free m = 1.3 (N=1/4)

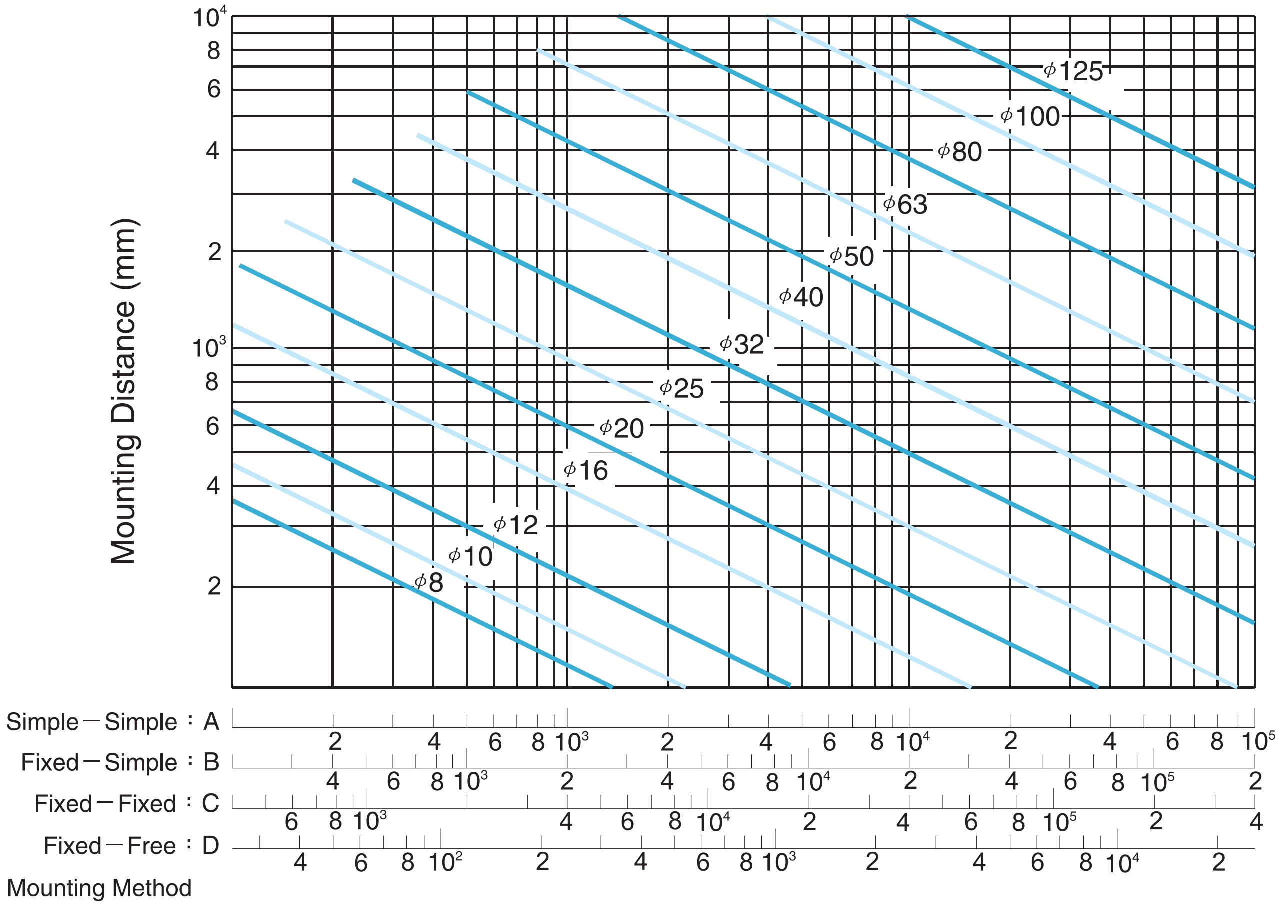


Fig. 3.16 Buckling Load vs. Shaft Dia. and Length

3.3 Critical Speed

It is necessary to check if the ball screw rotation speed is resonant with the natural frequency of the screw shaft.

GTEN has determined 80% or less of this critical speed as an allowable rotation speed. Fig. 3.17 shows a diagram which summarizes the allowable rotation speed for shaft nominal diameters up to outside diameter of the screw shaft exceeds 125mm.) Select the graduation of allowable rotation speed according to the method of supporting the ball screw.

Where the working rotation speed presents a problem in terms of critical speed, it would be best to provide an intermediate support to increase the natural frequency of the screw shaft.

dm.n value

The allowable rotation speed is regulated also by the dm・n value (dm:diameter of central circle of steel ball , n:Revolution speed , rpm) which expresses the peripheral speed.

Generally;

For precision (accuracy grade C7 to C0)

dm・n≦70,000

For general industry (C10)

dm・n≦50,000

Product exceeding the above limits can be produced, contact GTEN.

$$n=\alpha \times \frac{60\lambda^2}{2\pi L^2}\sqrt{\frac{E I g}{\gamma A}}=f\,\frac{dr}{L^2}\times 10^7 \text{ (rpm)}$$

Where,

α : Safety factor (α =0.8)

E : Vertical elastic modules (E=2.1×10^4 kgf / mm^2)

I : Min. secondary moment of screw shaft sectional area

$$I=\frac{\pi}{64}dr^4 (mm^4)$$

dr : Screw shaft root diameter (mm)

: Acceleration of gravity (=9.8×10^3 mm / s^2)

γ : Density (γ=7.8×10^{-6} kgf / mm^3)

A : Screw shaft sectional area (A= π dr^2/4 mm^2)

L : Mounting distance (mm)

f 、λ : Coefficient determined from the ball screw mountung method

Simple－Simple	f=9.7	(λ = π)
Fixed－Simple	f=15.1	(π =3.927)
Fixed－Fixed	f=21.9	(π =4.730)
Fixed－Free	f=3.4	(π =1.875)

(* Particular consideration is necessary for manufacturing when the screw length/shaft dia. Ratio is ε >70. In such an event, contact GTEN.)

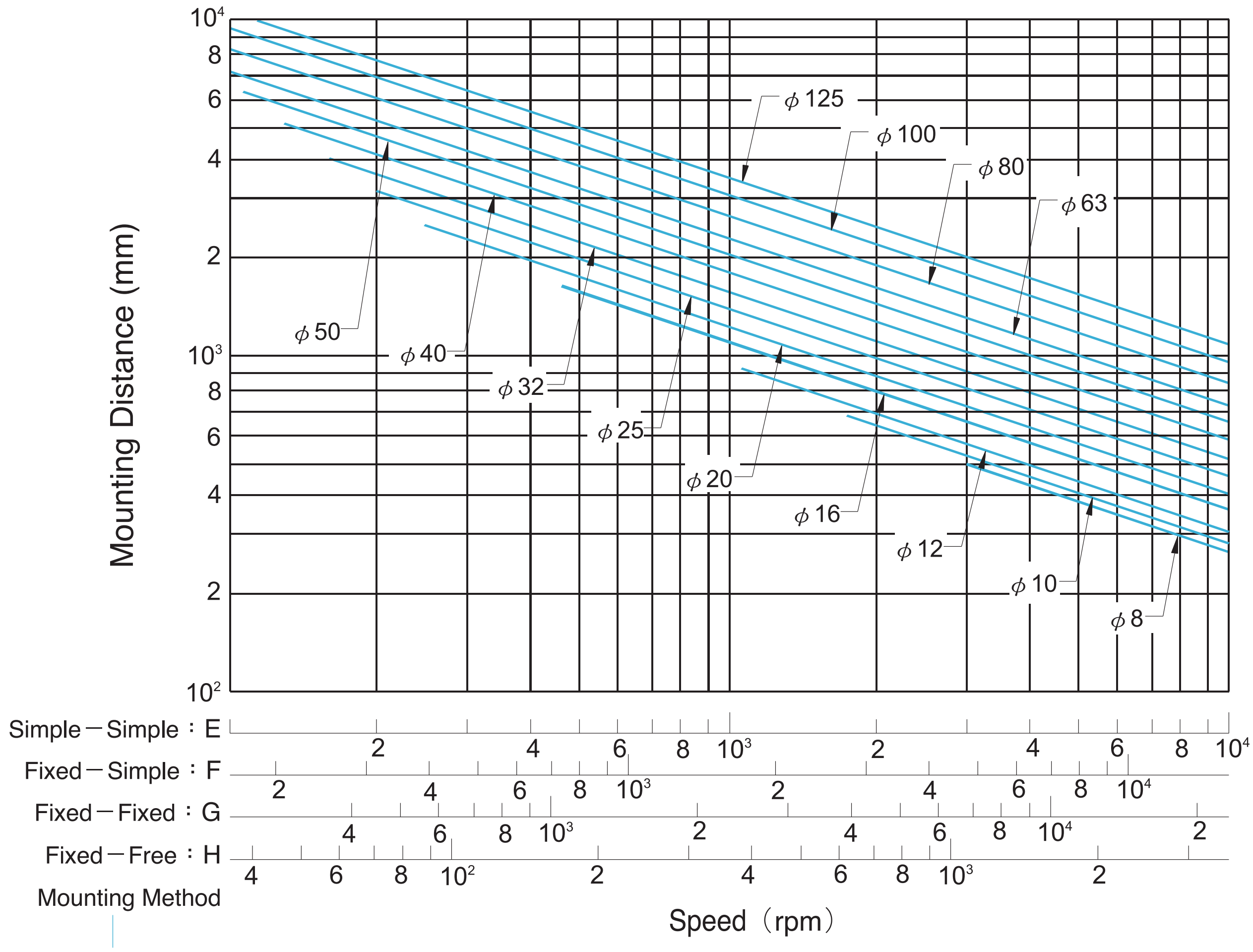


Fig. 3.17 Critical Speed vs. Shaft Dia

4. Nut Design

4.1 Selection of Nut

(1) Series

When making selection of series, please take into consideration of demanded accuracy, intended delivery time, dimensions(the outside diameter of the screw, ratio of lead / the outside diameter of the screw), preload load, etc.

(2) Circulation type

Selection of circulation type : Please focus on the economy of space for the nut installation portion.

(a) External circulation type

- Economy
- Suitable for mass production
- Applicable to those with larger lead / the outside diameter of the screw

(b) Internal circulation type

- With nuts of finely crafted outside diameter (occupying small space)
- Applicable to those with smaller lead / the outside diameter of the screw

(c) End-caps circulation type

- Suitable for high speed positioning

(3) Number of loop circuits

Performance and life of service should be considered when selecting number of loop circuits

(4) Shape of flanges

Please make selection based on the available space for the installation of nuts.

(5) Oil hole

Oil holes are provided for the precision ball screws, please use them during machine assembling and regular furnishing.

4.1.1 External Ball Circulation Nuts

Features:

- Offers smoother ball running.
- Offers better solution and quality for long lead or large diameter ballscrews.

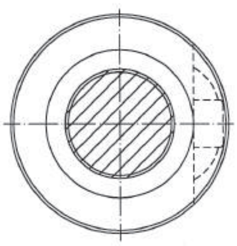


Fig.4.1 Immersion type

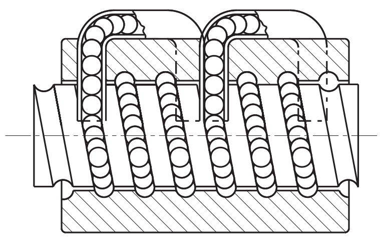


Fig. 4.2 Enternal ball circulation's nut

4.1.2 Internal Ball Circulation Nuts

Features:

The advantage of internal ball circulation nut is that the outer diameter is smaller than that of external ball circulation nut. Hence it is suitable for the machine with limit space for Ballscrew installation.

It is strictly required that there is at least one end of screw shaft with complete threads. Also the rest area next to this complete thread must be with smaller diameter than the nominal diameter of the screw shaft. Above are required for easy assembling the ballnut onto the screw shaft.

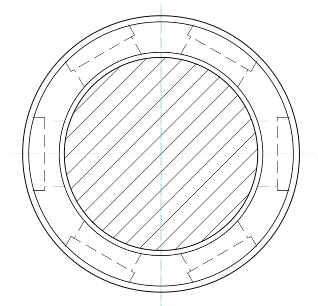


Fig. 4.3 Internal ball circulation's side view

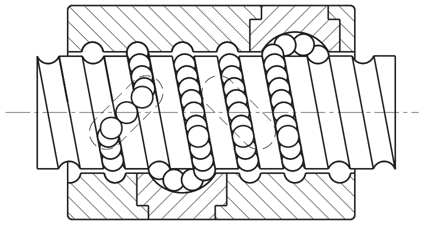


Fig. 4.4 Internal ball circulation's nut

4.1.3 High Lead Ballscrews

Features:

- It is important for a High-lead Ballscrew to be with characteristics of high rigidity, low noise and thermal control. GTEN designs and treatments are taken for following:

High DN Value

- The DN value can be 130,000 in normal case. For some special cases, for example in a fixed ends case, the DN value can be as high as 140,000. Please contact our engineers for this special application.

High Speed

- *GTEN* High-speed Ballscrews provide 100 m/min and even higher traverse speed for machine tools for high performance cutting.

High Rigidity

- Both the screw and ballnut are surface hardened to a specific hardness and case depth to maintain high rigidity and durability.
  Multiple thread starts are available to make more steel balls loaded in the ballnut for higher rigidity and durability.

Low Noise

- Special design of ball circulation tubes (patent pending) offer smooth ball circulation inside the ballnut. It also makes safe ball fast running into the tubes without damaging the tubes.
- Accurate ball circle diameter (BCD) through whole threads for consistent drag torque and low noise.

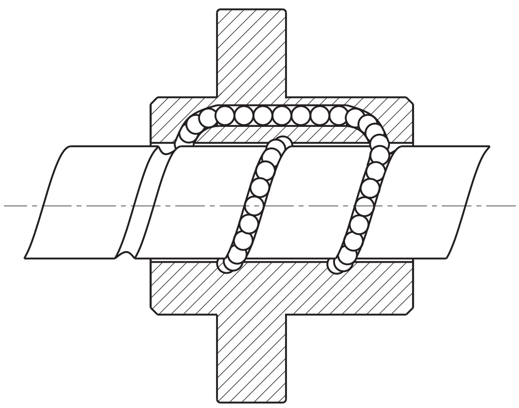


Fig. 4.5 Low noise circulation's nut

4.2 Axial Rigidity

Excessively weak rigidity of the screw's peripheral structure is one of the primary causes that result in lost motion. Therefore in order to achieve excellent positioning accuracy for the precision machines such as NC working machines, etc., axial rigidity balance as well as torsional rigidity for the parts at various portions of the transmission screw have to be taken into consideration at time of designing.

Static Rigidity K

The axial elastic deformation and rigidity of the transmission screw system can be determined from the formula below.

$$K = \frac{P}{e} \quad (kgf/mm)$$

P：Axial load (kgf) borne by the transmission screw system

e：Axial flexural displacement（mm）

$$\frac{1}{K} = \frac{1}{K_S} + \frac{1}{K_N} + \frac{1}{K_B} + \frac{1}{K_H} \quad (mm/kgf)$$

K_S：Axial rigidity of screw shaft（1）

K_N：Axial rigidity of nut（2）

K_B：Axial rigidity of bracing shaft（3）

K_H：Axial rigidity of installation portions of nuts and bearings（4）

（1） Axial rigidity K_S and displacement δ_S of screw shaft

$$K_S = \frac{P}{\delta_S} \quad (kgf/mm)$$

P：Axial load (kgf)

For places of Fixed – Fixed installation

$$\delta_{SF} = \frac{PL}{4AE} \quad (mm)$$

For places other than Fixed – Fixed installation

$$\delta_{SS} = \frac{PL_0}{4AE} \quad (mm)$$

$$\delta_{SS} = 4\,\delta_{SF}$$

δ_{SF}：Directional displacement at places of fixed-fixed installation

δ_{SS}：Directional displacement at places other than fixed-fixed installation

A：Cross-sectional area of the screw shaft tooth root diameter（mm^2）

E：Longitudinal elastic modulus（$2.1\times10^4 kgf/mm^2$）

L：Distance between installations（mm）

L_0：Distance between load applying points（mm）

（2） Axial rigidity K_N and displacement δ_N of nut

$$K_N = \frac{P}{\delta_S} \quad (kgf/mm)$$

(a) In case of single nut

$$\delta_{NS} = \frac{K}{\sin\beta}\left(\frac{Q^2}{d}\right)^{1/3} \times \frac{1}{\xi} \quad (mm)$$

$$Q = \frac{P}{n \cdot \sin\beta} \quad (kgf)$$

$$n = \frac{D_0 \pi m}{d} \quad (each)$$

Q：Load of one steel ball (kgf)

n：Number of steel ball

k：Constant determined based on material, shape, dimensions $k \fallingdotseq 5.7\times10^{-4}$

β：Angle of contact（45°）

P：Axial load (kgf)

d：Steel ball diameter (mm)

ξ：Accuracy, internal structure coefficient

m：Effective number of balls

D_0：Steel ball center diameter (mm)

$$D_0 = \frac{\ell}{\tan\alpha \cdot \pi}$$

ℓ：Lead (mm)

α：Lead angle

(b) In case of double nuts

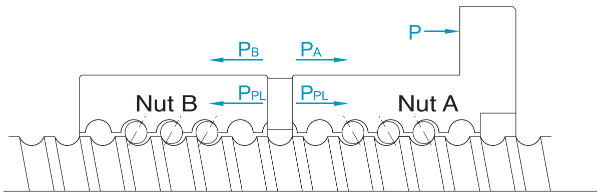


Fig. 4.6 Preloaded for the double nuts

When an axial load P of approximately 3 times of the preload load P_{PL} is exerted, for the purpose of eliminating the preload P_{PL} on nut B, please set the preload load P_{PL} at no more than 1/3 of the maximal axial load (0.25Ca should be taken as the standard maximal preload load). With respect to the displacement value, it should be of 1/2 of the single nut displacement when axial load is 3 times of the preload.

$$K_N = \frac{P}{\delta_{NW}} = \frac{3P_{PL}}{\delta_{NS}/2} = \frac{6P_{PL}}{\delta_{NS}} \text{ (kgf / mm)}$$

δ_{NS} : Displacement of single nut (mm)

δ_{NW} : Displacement of double nuts (mm)

(Explanation of the rigidity of double nuts)

As shown in Diagram 5.1 and 5.2 , when a preload P_{PL} is applied on the 2 nuts A,B, both nuts A & B would produce flexural deformations that will reach point X. If an external force P is exerted from here, nut A would move from point X to point X1, while nut B would move from X to X2.

Then, based on the computing formula for displacement δ_{NS} of the single nut, we can obtain:

$\delta_o = aP_{PL}^{2/3}$

while displacements of nuts A & B are

$\delta_A = aP_{PL}^{2/3}$

since displacements of nuts A & B generated due to exertion of external force P are equal, therefore

$\delta_A - \delta_0 = \delta_0 - \delta_B$

or if P is the only external force P that exerts on nuts A,B, if P_A increases

$P_A - P_B = P$

$\delta_B = 0$

for preventing the external force applied on nut B being absorbed by nut A thus decreasing, so

when $\delta_B = 0$

$aP_A^{2/3} - aP_{PL}^{2/3} = aP_{PL}^{2/3}$

$P_A^{2/3} = 2P_{PL}^{2/3}$

$P_A = \sqrt{8}\, P_{PL} \fallingdotseq 3P_{PL}$

or based on $\delta_A - \delta_0 = \delta_0$

$\delta_0 = \frac{\delta_A}{2}$

thus it can also be judged from Fig. 5.3 that, when axial load is 3 times of preload load , for a single nut with 1/2 displacement, the rigidity is 2 times as high.

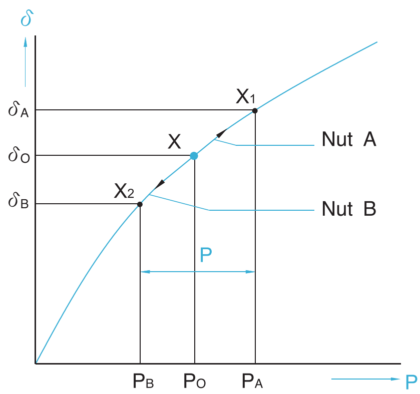


Fig. 4.7

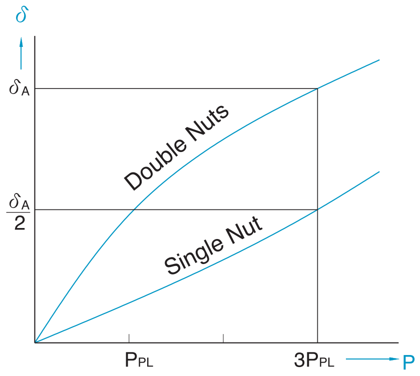


Fig. 4.8

(3) Axial rigidity K_B and displacement δ_B of bracing shaft

$$K_B = \frac{P}{\delta_B} \text{ (kgf / mm)}$$

The rigidity of the assembled diagonal thrust ball bearing that is used as the bracing bearing for the ball screw and is widely utilized in the field of precision machines can be found from the following formula.

$$\delta_B = \frac{2}{\sin\beta}\left(\frac{Q^2}{d}\right)^{1/3}$$

$$Q = \frac{P}{n \cdot \sin\beta} \text{ (kgf)}$$

Q : Load of one steel ball (kgf)

β : Angle of contact (45°)

d : Steel ball diameter (mm)

P : Axial load (kgf)

n : Number of steel balls

(4) Axial rigidity K_H and displacement δ_H of installation portions of nuts and bearings.

In early stage of machine development, special attentions should be paid to the requirement of high rigidity for the installation portion.

$$K_H = \frac{P}{\delta_H} \text{ (kgf / mm)}$$

4.2.1 Horizontal reciprocating moving mechanism

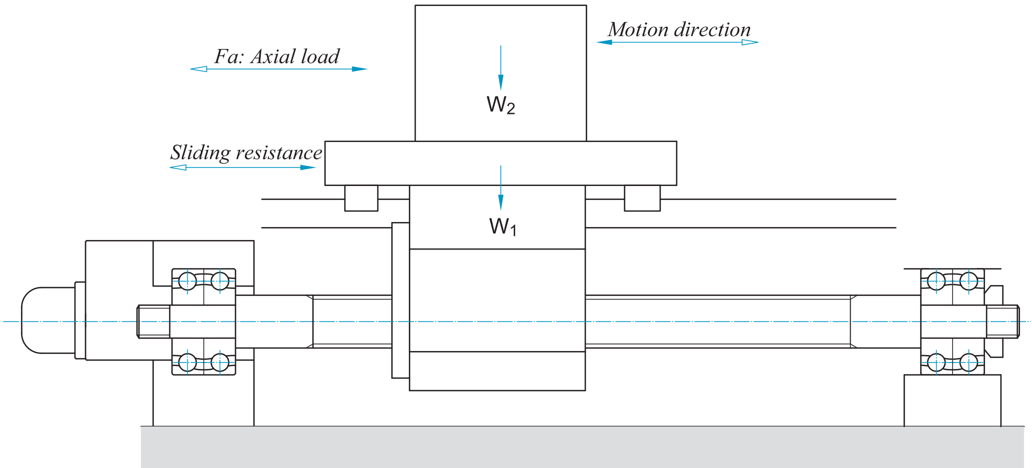


Fig.4.9 Horizontal reciprocating moving mechanism

For reciprocal operation to move work horizontally (back and forth) in an conveyance system, the axial load (Fa) can be gotten using the following equations:

Acceleration (leftward)	$Fa_1 = \mu \times mg + f + ma$
Constant speed (leftward)	$Fa_2 = \mu \times mg + f$
Deceleration (leftward)	$Fa_3 = \mu \times mg + f - ma$
Acceleration (rightward)	$Fa_4 = -\mu \times mg - f - ma$
Constant speed (rightward)	$Fa_5 = -\mu \times mg - f$
Deceleration (rightward)	$Fa_6 = -\mu \times mg - f + ma$

Here:

a : Acceleration

$a = \dfrac{V_{max}}{t}$ V_{max} : Rapid feed speed

t : time

m : Total weight(table weight + work piece weight)

μ : Sliding surface friction coefficient

f : Non-load resistance

4.2.2 Vertical reciprocating moving mechanism

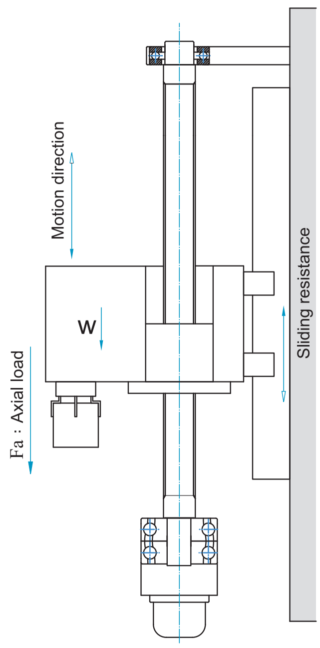


Fig.4.10 Vertical reciprocating moving mechanism

5.1 Preload and Effect

5.1.1 Ball Screw’s Preload and Effect

In order to get high positioning accuracy, there are two ways to reach it. One is commonly known as to clear axial play to zero. The other one is to increase Ballscrew rigidity to reduce elastic deformation while taking axial load. Both two ways are done by preloading.

(1) Methods of preloading

a. Double-nut method:

A spacer inserted between two nuts exerts a preload. There are two ways for it.

One is illustrated in Fig.5.1 That is to use a spacer with thickness complies with required magnitude of preload. The spacer makes the gap between Nut A and B to be bigger, hence to produce a tension force on Nut A and B. It is

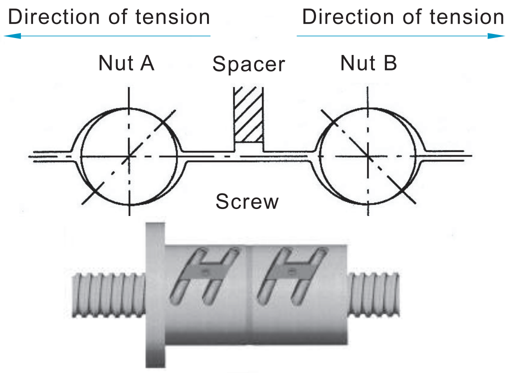


Fig.5.1 Extensive preload

b. Single-nut method:

As that illustrated on Fig. 5.2 using oversize balls onto the space between Ballnut and screw to get required preload. The balls shall make four-point contact with grooves of Ballnut and screw.

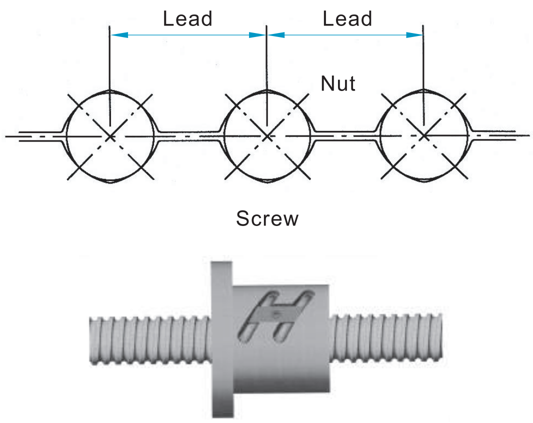


Fig.5.2 Four-point contact preload

(2) Relation between preload force and elastic deformation

Fig 5.3 Nuts A and B are assembled with preloading spacer. The preload forces on Nut A and B are F_{ao}, but with reversed direction. The elastic in fig.5.4 deformation on both Nuts are δ_{a0}

$$\delta_A = \delta_{a0} + \delta_{a1}$$

$$\delta_B = \delta_{a0} - \delta_{a1}$$

The load in nut A and nut B are:

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

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

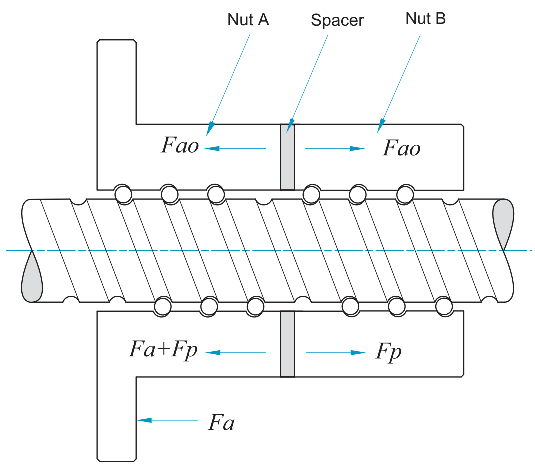


Fig.5.3 Double-nut positioning preload

It means Fa is offset with an amount Fa' because of the deformation of Nut B decreases. As a result, the elastic deformation of Nut A is reduced. This effect shall be continued until the deformation of Nut B becomes zero, that is, until the elastic deformation δ_{al} caused by the external axial force equals δ_{a0}, and the preload force applied to Nut B is completely released. The formula related the external axial force and elastic deformation is shown as below:

$$\delta_{a0} = K \times Fao^{2/3} \text{ and } 2\delta_{a0} = K \times F_l^{2/3}$$

$$(F_l / Fao)^{2/3} = (2\delta_{a0} / \delta_{a0}) = 2$$

$$F_l = 2.8Fao \fallingdotseq 3Fao$$

Therefore, the preload amount of a ballscrew is recommended to set as 1/3 of its axial load. Too much preload for a Ballscrew shall cause temperature raise and badly affect its life. However, taking the life and efficiency into consideration, the maximum preload amount of a Ballscrew is commonly set to be 10% of its rated basic dynamic load.

Shown on Fig 5. 5 with the axial load to be three times as the preload, the elastic displacement for the non-preloaded ball Nut is two times as that of the preloaded Nut.

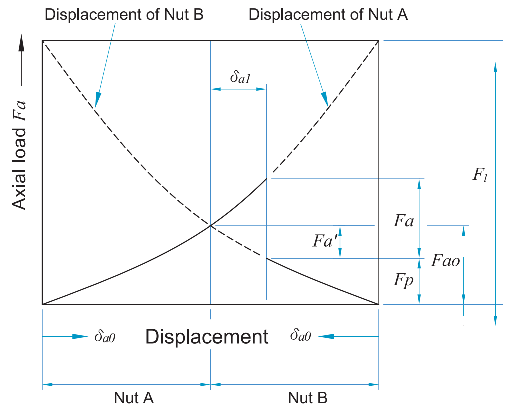


Fig.5.4 Positioning preload diagram

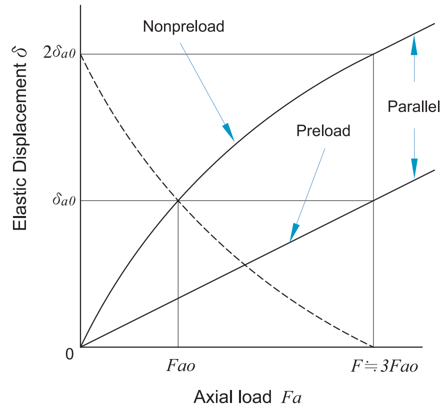


Fig.5.5 Elastic Displacement of the Ball Screw

5.2 Positioning Accuracy

5.2.1 Causes of error in positioning accuracy

Lead error and rigidity of feed system are common causes of feed accuracy error. Other causes like thermal deformation and feed system assembly are also playing important roles in feed accuracy.

5.2.2 Considering thermal displacement

If the screw-shaft temperature increases during operation, the heat elongates the screw shaft, thereby reducing the positioning accuracy. Expansion and shrinkage of a screw shaft due to heat can be calculated using equation (5.6).

$$\Delta L_\theta = \rho \cdot \theta \cdot L \quad \text{............(5.6)}$$

Here

ΔL_θ : Thermal displacement (μm)

ρ : Thermal-expansion coefficient ($12\ \mu m/m^\circ C$)

θ : Screw-shaft temperature change ($^\circ C$)

L : Ballscrew length (mm)

That is to say, an increase in the screw shaft temperature of 1 expands the shaft by 12 μm per meter. The higher the Ballscrew speed, the greater the heat generation. Thus, temperature increases reduce positioning accuracy. Where high accuracy is required, anti-temperature-elevation measures must be provided as follows:

(1) To control temperature:
- Selecting appropriate preload.
- Selecting correct and appropriate lubricant.
- Selecting larger lead for the Ballscrew and decrease the rotation speed.

(2) Compulsory cooling:
- Ballscrew with hollow cooling.
- Lubrication liquid or cooling air can be used to cool down external surface of Ballscrew.

(3) To keep off effect upon temperature raise:
- Set a negative cumulative lead target value for the Ballscrew.
- Warm up the machine to stable machine's operating temperature.
- Pretension by using on Ballscrew while installing onto the machine.

6. Life

6.1 Life of the Ballscrew

Even though the Ballscrew has been used with correct manner, it shall naturally be worn out and can no longer be used for a specified period. Its life is defined by the period from starting use to ending use caused by nature fail.

a. Fatigue life - Time period for surface flaking off happened either on balls or on thread grooves.
b. Accuracy life - Time period for serious loosing of accuracy caused by wearing happened on thread groove surface, hence to make Ballscrew can no longer be used.

6.2 Fatigue Life

The basic dynamic rate load (Ca) of the Ballscrew is used to calculate its fatigue life

6.2.1 Basic dynamic rate load Ca

The basic dynamic rate load (Ca) is the revolution of 10^6 that 90% of identical Ballscrew units in a group, when operated independently of one another under the same conditions, can achieve without developing flaking.

6.2.2 Fatigue life

(1) Calculating life:

There are three ways to show fatigue life:
a. Total number of revolutions.
b. Total operating time.
c. Total travel.

$$L = \left(\frac{Ca}{Fa \times f_W} \right)^3 \times 10^6$$

$$L_t = \frac{L}{60 \times n}$$

$$L_s = \frac{L \times l}{10^6}$$

Here
L : Fatigue life (total number of revolutions)
L_t : Fatigue life (total operating time)
L_s : Fatigue life (total travel)
Ca : Basic dynamic rate load
Fa : Axial load
n : Rotation speed
l : Lead
f_W : Load factor (refer to Table 6.1)

Table6.1 Load factor f_W

Vibration and impact	Velocity(V)	f_W
Light	V<15 (m/min)	1.0~1.2
Medium	15<V<60 (m/min)	1.2~1.5
Heavy	V>60 (m/min)	1.5~3.0

Too long or too short fatigue life are not suitable for Ballscrew selection. Using longer life make the Ballscrew's dimensions too large. It's an uneconomical result. Following table is a reference of the Ballscrew's fatigue life.

Machine center....................................20,000 hrs
Production machine..............................10,000 hrs
Automatic controller..............................15,000 hrs
Surveying instruments...........................15,000 hrs

(2) Mean load:

When axial load changed constantly. It is required to calculate the mean axial load (Fm) and the mean rotational speed (Nm) for fatigue life. Setting axial load (Fa) as Y-axis; rotational number ($n.t$) as X-axis. Getting three kind curves or lines:

a. Gradational variation curve (Fig.6.1)

Mean load can be calculated by using equation):

$$F_m = \left(\frac{F_1^3 \cdot n_1 \cdot t_1 + F_2^3 \cdot n_2 \cdot t_2 + \cdots + F_n^3 \cdot n_n \cdot t_n}{n_1 \cdot t_1 + n_2 \cdot t_2 + \cdots + n_n \cdot t_n} \right)^{\frac{1}{3}}$$

Mean rotational speed can be calculated by using equation :

$$N_m = \frac{n_1 \cdot t_1 + n_2 \cdot t_2 + \cdots + n_n \cdot t_n}{t_1 + t_2 + \cdots + t_n}$$

Axial load (kgf)	Rotation speed (rpm)	Time Ratio (Sec)
F_1	n_1	t_1
F_2	n_2	t_2
⋮	⋮	⋮
F_n	n_n	t_n

b. Similar straight line (Fig.6.2)

When mean load variation curve like similar straight line.

Mean rotational speed can be calculated using equation (6.6)

$F_m = 1/3(F_{min} + F_{max})$ .... (6.6)

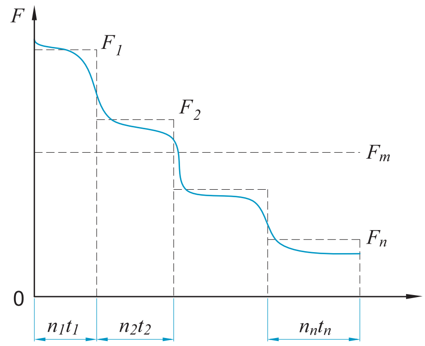


Fig. 6.1 Gradational variation curve's load

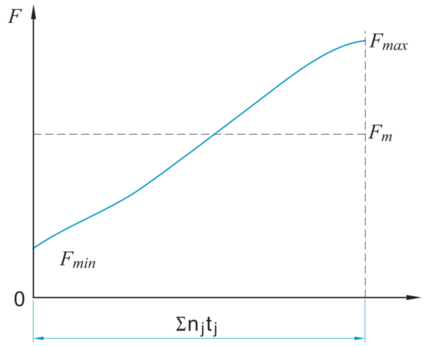


Fig. 6.2 Similar straight line's load

c. Sine curve there are two cases (Fig.6.3)

1. When mean load variation curve shown as the diagram below.

   Mean rotational speed can be calculated by using equation (6.3.1):

   $F_m = 0.65F_{max}$

2. When mean load variation curve shown as the diagram below.

   Mean rotational speed can be calculated by using equation (6.3.2):

   $F_m = 0.75F_{max}$

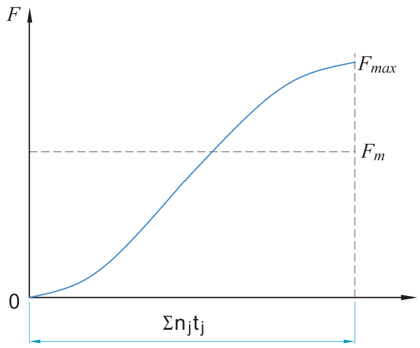


Fig. 6.3.1 Variation like Sine curve's load (1)

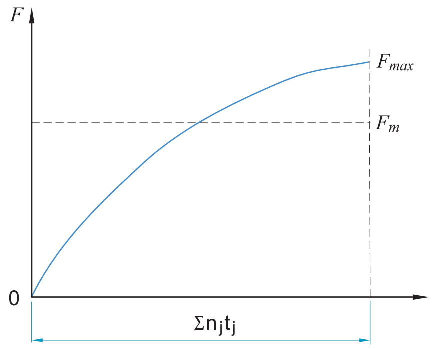


Fig. 6.3.2 Variation like Sine curve's load (2)

6.3 Material and Hardness

Material and Hardness of GTEN Ballscrews refer to Table 6.2

Table6.2 Material and hardness of GTEN Ballscrews

Denomination	Material	Heat treating	Hardness (RHC)
Precision ground	50CrMo4 QT	Induction hardening	58~62
Rolled	S55C	Induction hardening	58~62
Nut	SCM415H	Carburized hardening	58~62

6.4 Lubrication

Lithium base lubricants are used for Ballscrew lubrication.
Their viscosity are 30~40 cst (40℃) and ISO grades of 32~100.
Selecting:

1. Low temperature application: Using the lower viscosity lubricant.
2. High temperature, high load and low speed application: Using the higher viscosity lubricant.

Table6.3 Checking and supply interval of lubricant

Manner	Checking interval	Checking item	Supply or replacing interval
Automatic interval oil supply	every week	Oil volume and purity	To supply on each check, its volume depends on oil tank capacity.
Lubricating grease	Within 2-3 months after starting operation of machine	Foreign matter	Normally supply once a year as per the result of check
Oil bath	everyday before operation of machine	Oil surface	To supply as per wasting condition

6.5 Dustproof

Same as the rolling bearings, if there is the particles such as chips or water get into the ballscrew, the wearing problem shall be deteriorated. In some serious cases, ballscrew shall then be damaged. In order to prevent these problems from happening, there are wipers assembled at both ends of ball nut to scrape chips and dust. There is also the "O-Ring" at the wipers to seal the lubrication oil from leaking from ball nut.

6.6 Heat Treating Inspection Certificate

GTEN BALL SCREW TECHNOLOGY CO., LTD.

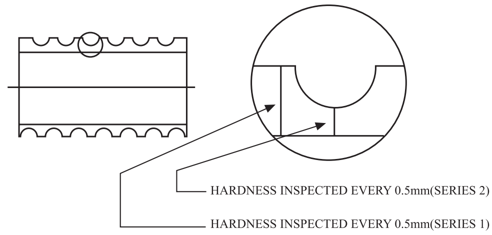
REPORT FOR HEAT TREATING INSPECTION

AMTRI VERITAS — NATIONAL ACCREDITATION OF CERTIFICATION BODIES — APPROVED TO BS EN ISO 9001 :1994 CERTIFICATE No.9AV1279

SPECIMEN#	8040		
CUSTOMER		P.O.NUMBER	SPECIFICATION
PRODUCT	BALL SCREW	980405-1	R25-5T4-FSI-300-395-C3
MATERIAL	50CrMo4 QT	980405-2	R25-5T4-FSI-500-600-C3
HEAT TREAT	INDUCTION SURFACE HARDENING		

ITEM	INSPECTION DATA
HARDNESS	58-62 HRC AT SURFACE
CASE DEPTH	2.0mm BELOW THREAD ROOT
MICRO-	Martensite IN SURFACE AREA
STRUCTURE	Sorbite IN CORE AREA
TEMPERING	AT 160 DEGREES CELCIUS

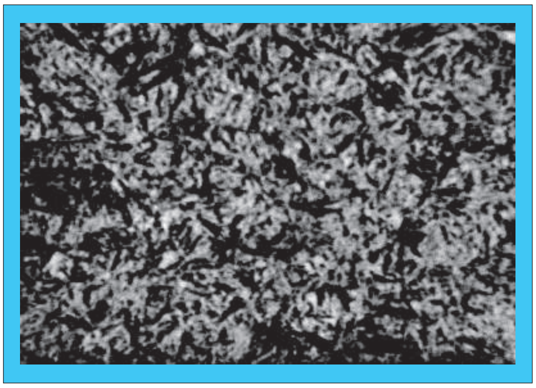
HEAT TREATED ARE
(SEE SKETCH)

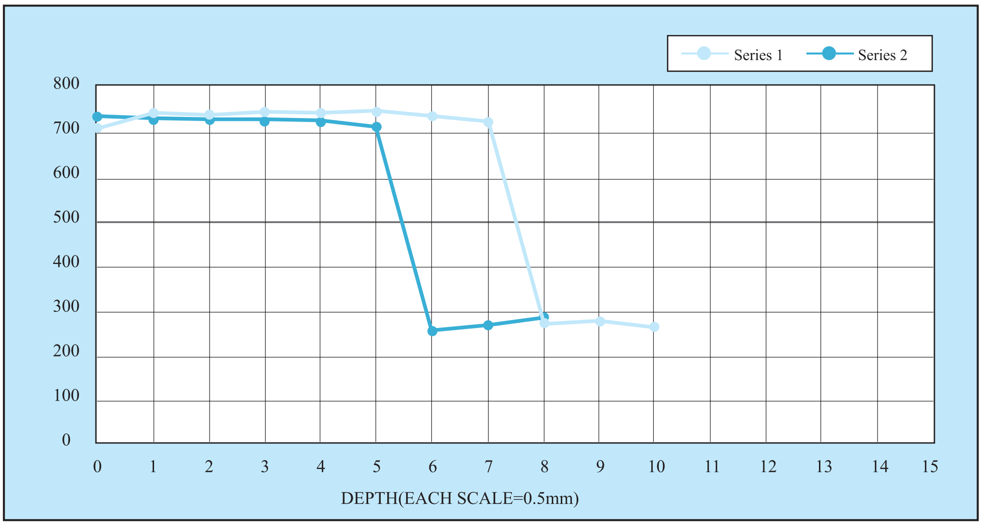


DEPTH	Series 1	Series 2
0	717	733
1	738	730
2	735	728
3	744	728
4	741	725
5	746	712
6	733	255
7	725	267
8	276	283
9	276	
10	262	
11		
12		
13		
14		
15		

MICROSTRUCTURE

X500





HV VS. HRC	
HV	HRC
800	64.0
780	63.3
760	62.5
740	61.8
720	61.0
700	60.1
690	59.7
680	59.2
670	58.8
660	58.3
650	57.8
640	57.3
630	56.8
620	56.3
610	55.7
600	55.2
590	54.7
580	54.1
570	53.6
560	53.0
540	51.7
520	50.5
500	49.1
480	47.7
460	46.1
440	44.5
420	42.7
400	40.8
380	38.8
360	36.6
340	34.4
320	32.2
300	29.8
280	27.1
260	24.0
240	20.3

REMARKS		PASS OR NOT		Q.C.CHIEF		INSPECTOR	

6.7 Key Points for Ball Screws Selection and Calculation

Key points for ball screws selection

When ball screws are subjected to selection, it is a most fundamental rule that you must first clearly find out what the operation conditions are before going ahead with the final design. Moreover, the elements of your selection include load weight, stroke, torque, position determination accuracy, tracking motion, hardness, lead stroke, nut inside diameter, etc., all elements are mutually related, any change to one of the elements will lead to the changes of other elements, special attention should always be paid to the balance among the elements.

1.Setting of operation conditions

(a) Machine service life time reckonig of H (hr)

H = ☐ × ☐ × ☐ × ☐

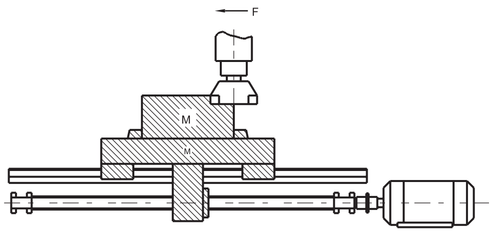
hours/day   days/ year   life years   Running

(b) Mechanical conditions

Calculation Date / Difference Operations	Speed/rotations	Cutting resistance	Sliding resistance	Time used
Fast feed	m / min / min^{-1}	kgf	kgf	%
Light cutting	/			
Medium cutting	/			
Heavy cutting	/			

(c) Position determination accuracy
Feed accuracy error factor includes load accuracy and system rigidity.Thermal displacement due to heat generation and positional error of the guide system is also important factors.

Calculation for ball screws selection



Design conditions

1. Working table weight 300 Kg
2. Working object weight 400 Kg
3. Maxima 700 mm
4. Fast feed speed 10 m/min
5. Minimal disassembly ability 10 μm/stroke
6. Driving motor DC motor (MAX 1000 min^{-1})
7. Guiding surface friction coefficient (μ= 0.05～0.1)
8. Running rate 60 %
9. Accuracy review items
10. Inertia generated during acceleration/deceleration can be neglected because the time periods involved are comparatively small.

1.Setting of operation conditions

(a) Machine service life time reckonig of H (hr)

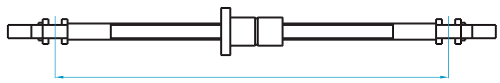
H = 12 hrs × 250 days × 10 years × 0.6 Running
=18000hr

(b) Mechanical conditions

Calculation Date / Difference Operations	Speed/rotations	Cutting resistance	Sliding resistance	Time used
Fast feed	10m/min/1000min^{-1}	0 kgf	70 kgf	10 %
Light cutting	6 / 600	100	70	50
Medium cutting	2 / 200	200	70	30
Heavy cutting	1 / 100	300	70	10

Sliding resistance = (300+400) × 0.1=70 kgf

Key points for ball screws selection	Calculation for ball screws selection
2. Ball screws lead stroke ℓ (mm) $\ell = \dfrac{\text{Fast feed stroke (m/min)} \times 1000}{\text{Max. Rotating speed (min}^{-1}\text{) of motor}}$ (mm)	2. Ball screws lead stroke ℓ (mm) $\ell = \dfrac{10000}{1000} = 10$ (mm) Minimal disassembly $= \dfrac{10\text{mm}}{1000\text{ stroke}} = 0.01$ mm/stroke
3. Computation of average load Pe (kgf) $Pe = \left(\dfrac{P_1^3 n_1 t_1 + P_2^3 n_2 t_2 + \cdots + P_n^3 n_n t_n}{n_1 t_1 + n_2 t_2 + \cdots + n_n t_n} \right)^{1/3}$ $Pe = \dfrac{2Pmax + Pmin}{3}$ $pe \fallingdotseq 0.65 Pmax$ $pe \fallingdotseq 0.75 Pmin$	3. Computation of average load Pe (kgf) Pe = $\left(\dfrac{70^3 \times 1000 \times 10 + 170^3 \times 600 \times 50 + 270^3 \times 200 \times 30 + 370^3 \times 100 \times 10}{1000 \times 10 + 600 \times 50 + 200 \times 30 + 100 \times 10} \right)^{1/3}$ $= \left(\dfrac{31.7 \times 10^{13}}{4.7 \times 10^4} \right)^{1/3}$ $\fallingdotseq 189$ kgf
4. Average number of rotations n_m $n_m = \dfrac{n_1 t_1 + n_2 t_2 + \cdots + n_n t_n}{100}$	4. Average number of rotations n_m $n_m = \dfrac{1000 \times 10 + 600 \times 50 + 200 \times 30 + 100 \times 10}{100}$ $= \dfrac{4.7 \times 10^4}{100}$ $= 470 \text{ min}^{-1}$
5. Calculation of required dynamic rated load Ca Ca = Pe • fs	5. Calculation of required dynamic rated load Ca Ca = 189 × 5 = 945 (kgf)
6. Calculation of required static rated load Coa Coa = Pmax • fs	6. Calculation of required static rated load Coa Coa = 369 × 5 = 1845 (kgf)
7. Selection of nut type Ca > 945 Coa > 1845 Select the nut types with basic dynamic rated load and basic static rated load as specified above.	7. Selection of nut type Choose SF I 4010 on the catalogue Ca = 3178 kgf Coa = 9480 kgf

Key points for ball screws selection	Calculation for ball screws selection
8. Calculation of life confirmation Lt (h) $$L_t = \left(\frac{Ca}{Pe \cdot fw}\right)^3 \cdot \frac{1}{60 n_m} \cdot 10^6$$	8. Calculation of life confirmation Lt (h) $$L_t = \left(\frac{3178}{189 \cdot 2}\right)^3 \cdot \frac{1}{60 \cdot 470} \cdot 10^6$$ = 20479 (h)
9. Determination of screw length Screw length = Maximal stroke + Nut length + 2 × reserved length at shaft end	9. Determination of screw length Screw length = 700+93+2×81 = 874 mm
10. Mounting distance of screw length	10. Mounting distance of screw length(F-F support) 
11. Permissible axial load	11. Permissible axial load Omitted because of F-F support
12. Permissible revolution speed n and dm $$n = \alpha \times \frac{60\lambda^2}{2\pi L^2}\sqrt{\frac{EIg}{\gamma A}} = f\,\frac{dr}{L^2}\times 10^7 \text{ (rpm)}$$ dm=Shaft dia. × Maximal speed	12. Permissible revolution speed n and dm $$n = \frac{21.9\times 35.2\times 10^7}{1200^2}$$ $= 5353\text{min}^{-1} > n\text{max}$ dm = 40 × 1000 = 40000 < 50000
13. Countermeasure against thermal displacement and rigidity	13. Countermeasure against thermal displacement and rigidity (a) It is estimated there would be a temperature rise of 2~5℃ with the ball screws of the general machinery, take temperature rise of 2℃ to computer the extension of ball screw. $$\Delta \ell = \alpha \cdot t \cdot L$$ $=11.7\times10^{-6}\times 2\times 700$ mm ≒ 0.016 mm $$F_P = \frac{EA\Delta \ell}{L}$$ $$= \frac{2.06\times 10^4\times\frac{\pi\times 35.2^2}{4}\times 0.016}{700}$$ ≒ 458 kgf

Key points for ball screws selection	Calculation for ball screws selection
(Reference) Force exerted on ball screw when inertia is considered ◎ When used horizontally 1. During acceleration $P_{ACC} = Mg \times \mu + \dfrac{M \times V}{60 \times \Delta t}$ 2. During deceleration $P_{DEC} = Mg \times \mu - \dfrac{M \times V}{60 \times \Delta t}$ ◎ When used vertically 1. During acceleration while descending, during deceleration while ascending $P_U = Mg - \dfrac{M \times V}{60 \times \Delta t}$ 2. During acceleration while ascending, during deceleration while descending $P_D = Mg + \dfrac{M \times V}{60 \times \Delta t}$ M : Mass of moving object (kg) g : Acceleration of gravity ($9.8 m/s^2$) V : Velocity (m/min) Δt : Acceleration /deceleration time (s) μ : Friction coefficient	Deviation can be corrected by estimating the temperature rise per extension of 0.016mm, and taking into consideration of the pre-tension of 458 kgf . (b) Rigidity (1) Directional rigidity $\delta_{SF} = \dfrac{PL}{4AE} = \dfrac{27 \times 1200}{4 \times \dfrac{\pi \times 35.2^2}{4} \times 2.06 \times 10^4}$ $= 0.00036$ mm $K_S = \dfrac{370}{0.00036} = 10.3 \times 10^5$ kgf / mm (2) Rigidity of steel ball and nut groove $n = \dfrac{41.8 \times \pi \times 2.5}{6.35} = 52$ $Q = \dfrac{370}{52 \sin 45^\circ} = 10$ $\delta_{NS} = \dfrac{0.00057}{\sin 45^\circ} \left(\dfrac{10^2}{6.35} \right)^{1/3} \times \dfrac{1}{0.7}$ $= 2.9 \times 10^{-3}$mm $K_N = \dfrac{370}{2.9 \times 10^{-3}} = 1.28 \times 10^5$kgf/mm (3) Rigidity of bracing bearings Where, nut rigidity 50 kgf / mm $\delta_B = \dfrac{370}{50 \times 2} = 3.7\ \mu m$ $K_B = \dfrac{370}{0.0037} = 1 \times 10^5$kgf/mm ◎ $\delta_{TOTAL} = 0.36 + 2.9 + 3.7 = 6.96\ \mu m$
14. Confirmation of the ball screw life	14. Confirmation of the ball screw life L = 20479(h) > 18000 (h)

Driving Torque

Driving torque T_S of the transmission shaft

$T_S = T_P + T_D + T_F$ (in fixed speed)

$T_S = T_G + T_P + T_D + T_F$ (when accelerating)

T_G : Acceleration torque (1)

T_P : Load torque (2)

T_D : Preload torque (3)

T_F : Friction torque (4)

(1) Acceleration T_G

$T_G = J\alpha$ (kgf • cm)

$$\alpha = \frac{2\pi n}{60\Delta t} \quad (\text{rad/s}^2)$$

J : Moment of inertia (kgf • cm • s^2)

α : Angular acceleration (rad/s^2)

n : Revolutions (min^{-1})

Δt : Starting time (sec)

(2) Load torque T_P

$$T_P = \frac{P \bullet \ell}{2\pi\eta_1} \quad (\text{kgf} \cdot \text{cm})$$

$P = F + \mu M$

P : Axial load (kgf)

ℓ : Lead (cm)

η_1 : Positive efficiency

The efficiency when rotating motion is altered to linear motion

F : Cutting force (kgf)

μ : Friction coefficient

M : Mass of moving object (kg)

g : Acceleration of gravity (9.8 m/s^2)

$$T_P = \frac{P \bullet \ell \bullet \eta_2}{2\pi}$$

η_2 : Reverse efficiency

The efficiency when linear motion returns to rotating motion

(3) Preload torque T_D

$$T_D = \frac{K \bullet P_{PL} \bullet \ell}{\sqrt{\tan\alpha} \bullet 2\pi} \quad (\text{kgf} \cdot \text{cm})$$

K : Internal coefficient (0.05 is usually adopted)

P_{PL} : Preload kgf)

ℓ : Lead (cm)

α : Lead angle

(4) Friction torque T_F

$T_F = T_B + T_O + T_J$ (kgf • cm)

T_B : Friction torque of bracing shaft

T_O : Friction torque of free shaft

T_J : Friction torque motor shaft

The friction torque of the bracing shaft would be affected by the lubrication oil. Or special attention has to be paid to unexpected excessive friction torque which may be generated when oil seal is overly tight, or may result in temperature rise.

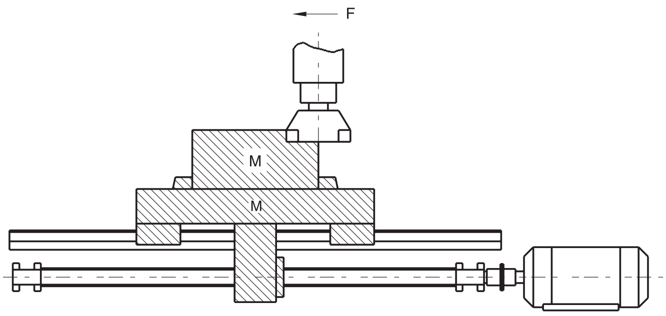


Fig. 7.1 Moment of inertia of load

【For reference】 Moment of inertia of load (Table 7.1)

$J = J_{BS} + J_{CU} + J_W + J_M$

J_{BS} : Moment of inertia Ball screws shaft

J_{CU} : Moment of inertia Coupler

J_W : Moment of inertia Linear motion part

J_M : Moment of inertia Roller shaft part of motor shaft

Table 7.1 Conversion formula for moment of inertia of load

Moment of inertia converted from motor shaft \ Formula	J
Cylinder load	$\frac{\pi\rho L D^4}{32}$
Linearly moving object	$\frac{M}{4}\left(\frac{V\ell}{\pi \bullet N_M}\right)^2 = \frac{M}{4}\left(\frac{P}{\pi}\right)^2$
Unit	kg • m^2
Moment of inertia during deceleration	$J_M = \left(\frac{J\ell}{N_M}\right)^2 \cdot J\ell$

ρ : Density (kg/m^3) $\rho = 7.8 \times 10^3$

L : Cylinder length (m)

D : Cylinder diameter (m)

M : Mass of the linear motion part (kg)

$V\ell$: Velocity of the linearly moving object (m/min)

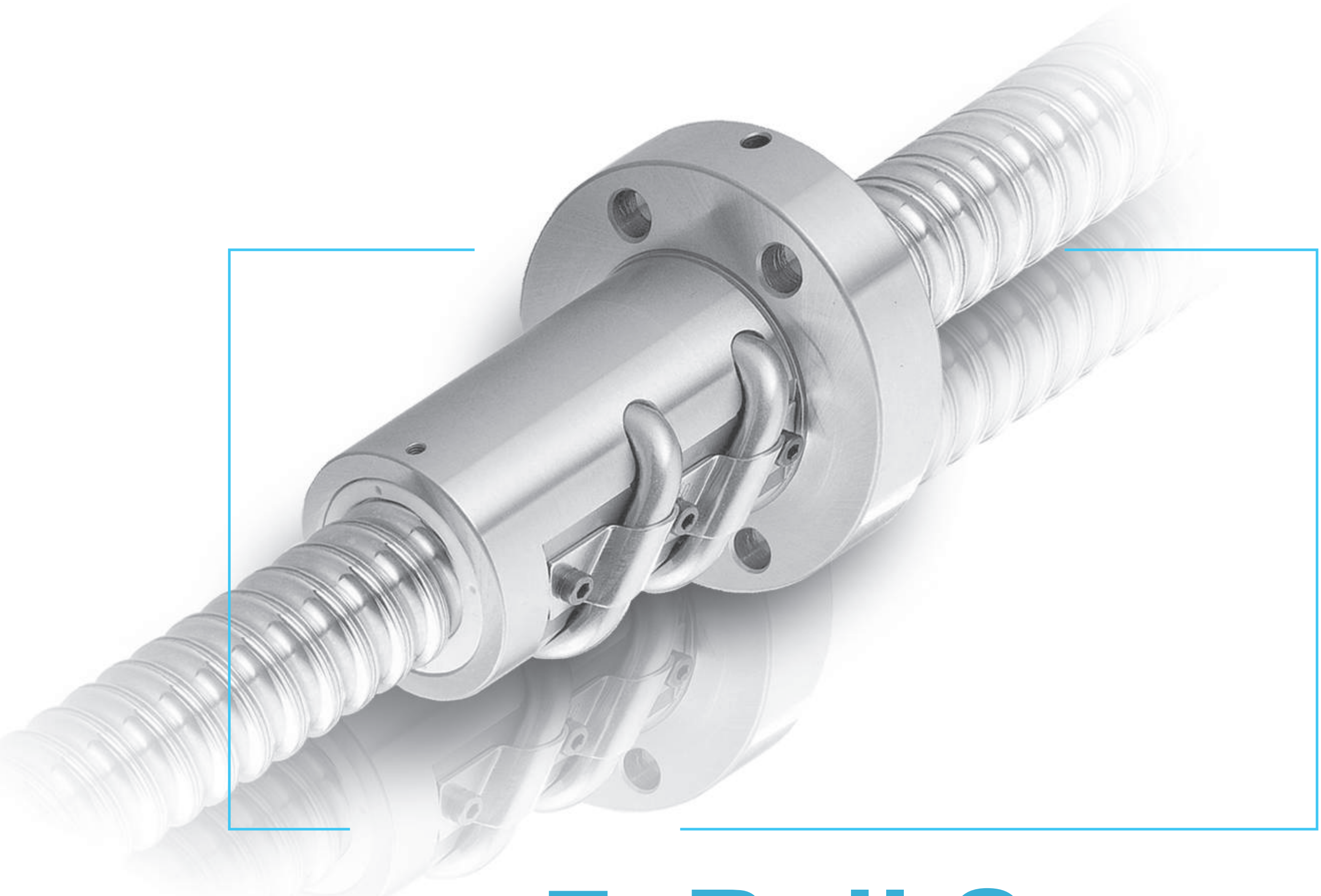
N_M : Motor shaft revolutions (min^{-1})

P : The moving magnitude of the linearly moving object per every rotation of the motor (m)

$N\ell$: Rotations in longitudinal moving direction (min^{-1})

$J\ell$: Rotations in longitudinal moving direction (min^{-1})

J_M : Moment of inertia in motor direction

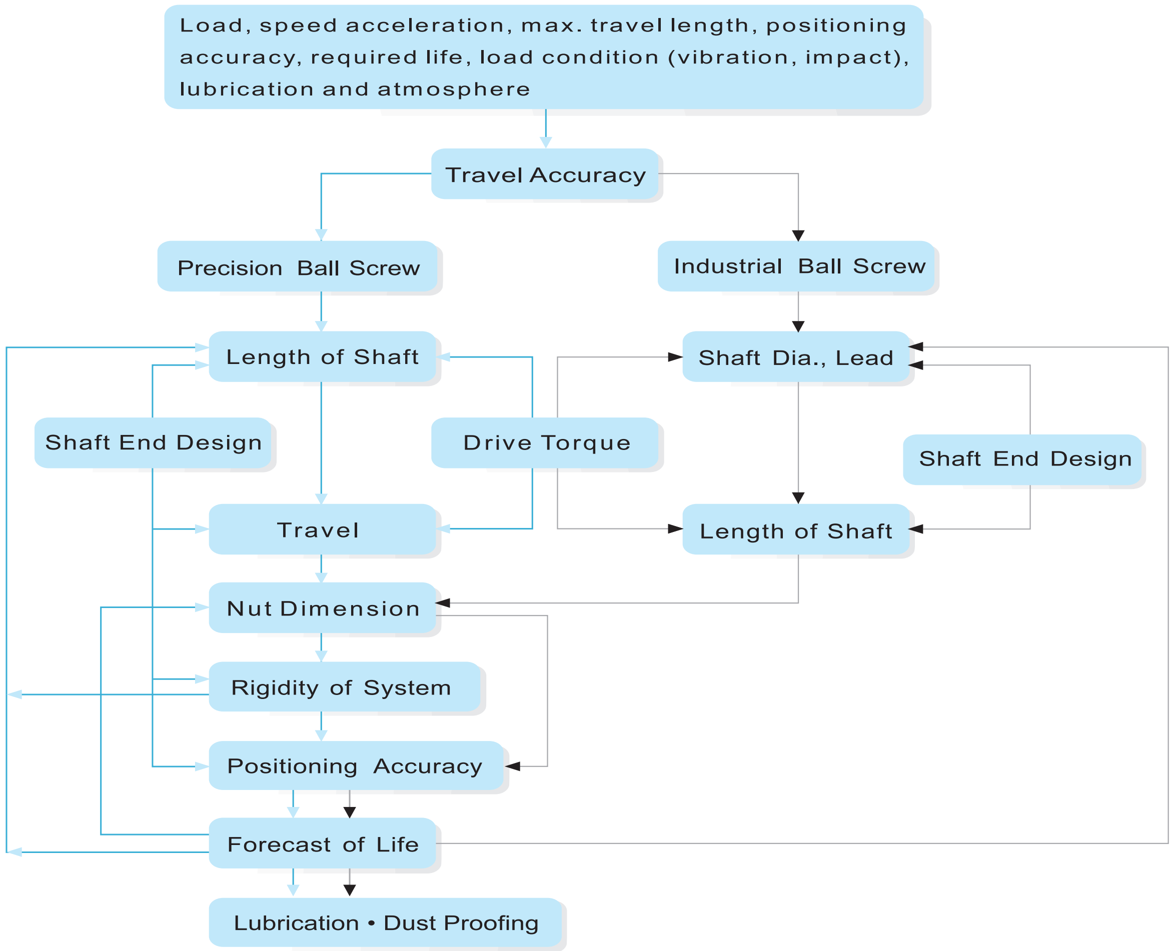


7. Ball Screw

Selecting Correct Type of Ballscrew

Condition

- Accuracy
- Screw Shaft Design
- Drive Torque
- Nut Design
- Rigidity
- Positioning Accuracy
- Life Design
- Lubrication and safety design



GTEN Ball Screw Size List

Dia. \ Lead	1	2	2.5	3	4	5	5.08	6	10	12.7	16	20	25	32	40	50
6	⊙															
8	⊙	⊙	⊙													
10		⊙		⊙	⊙											
12		⊙			⊙	⊙			⊙	⊙						
14		⊙			⊙	⊙										
15												⊙				
16		⊙			⊙	⊙	⊙		⊙		⊙					
20					○	⊙			⊙			⊙				
25					⊙	⊙			⊙			⊙	⊙			
32					○	⊙		⊙	⊙			⊙		⊙		
40						⊙		⊙	⊙			⊙			⊙	
50						○			⊙			⊙				⊙
63									⊙			⊙			⊙	
80									⊙			⊙				

• means rolled ball screw ○ means ground ball screw