Ball Screw Technical Description

Feature of Ball Screws

●High mechanical efficiency
KSS Ball Screws are fitted with steel Balls, providing rolling contact between the Nut and Screw Shaft, allowing for mechanical efficiency of about 90% and reducing the required Torque to less than one-third that of conventional Lead Screws. The design of the KSS Ball Screws also allows linear motion to be converted into rotary motion easily (Fig. A-81).

●Axial play
With conventional Triangular and Trapezoidal Screw threads, reducing the Axial play increases the rotational Torque due to the sliding friction.
KSS Ball Screws, on the other hand, are very easily rotated, even with no Axial play. The use of Double Nuts also provides increased Rigidity.

●High precision
KSS Ball Screws are machined, assembled, and inspected using the technology of ultra-precision Lead Screw and Screw Gauge machining, under the temperature controlled room. High precision and accurate positioning ensure high reliability in use.

●Long service life
The Ball Screw movement results in virtually no wear, as the rolling-contact design, combined with the use of carefully selected heat-treated materials, results in an extremely low friction. This is the reason that high precision can be kept over long period.

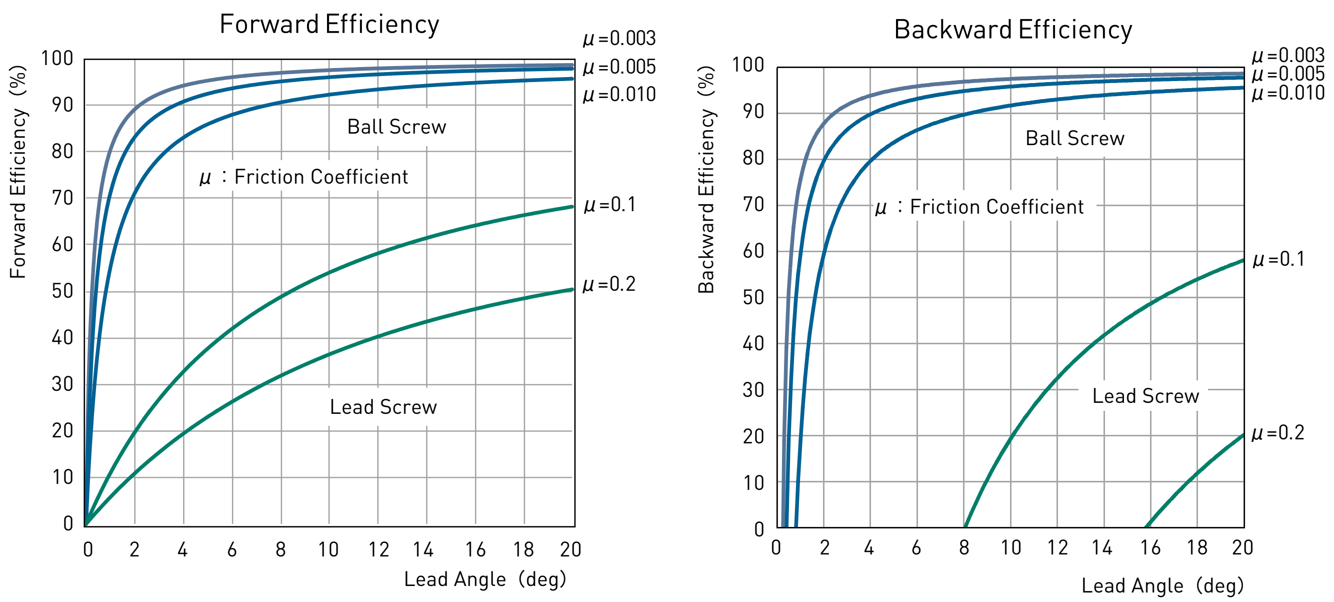
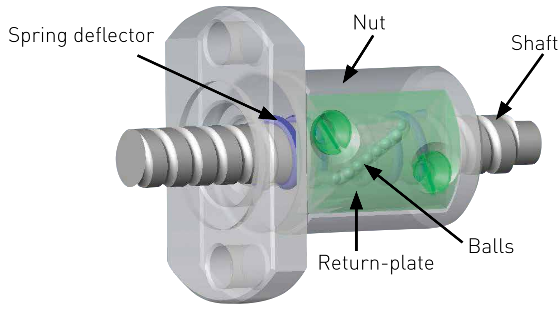


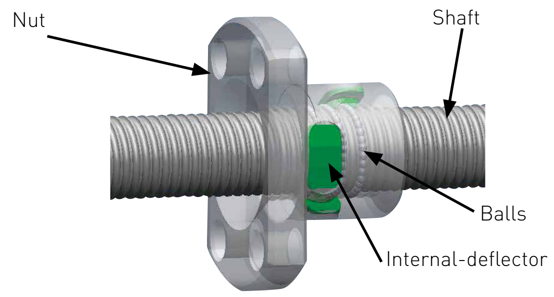
Fig. A-81 : Mechanical Efficiency

Construction of Ball Screws

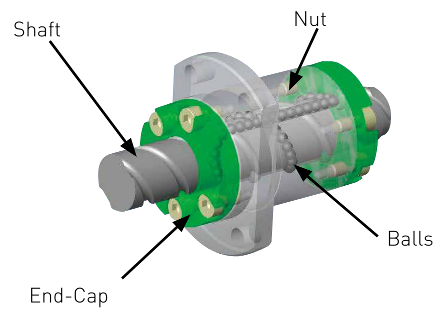
●Return-plate system
The Return-plate system uses coil-type deflectors incorporated inside the Nut to pick up the steel Balls and circulate them via the Return-plate channel. This system has the advantage of allowing the use of a Nut that is smaller in diameter than those employed in Return-tube systems. In addition, the upward-angle installation of the Return-plate ensures even smoother rotation.



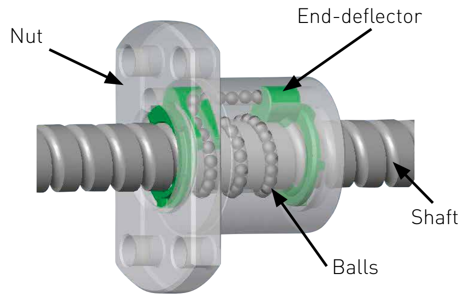
●Internal-deflector system
The Internal-deflector system employs a lightweight Miniature Ball Screw, which enables the Nut diameter and length to be reduced to the smallest possible size. The Balls bear the load while rolling along the screw groove between the Shaft and the Nut. The Balls are continuously circulated, transferred to the adjacent groove in the screw via the Internal-deflector channel and then back to the loaded groove area.



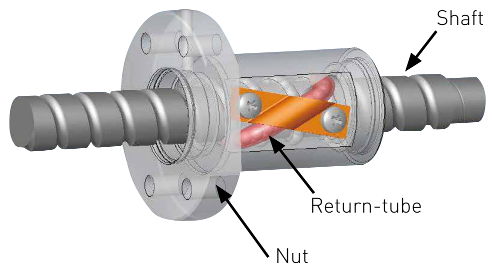
●End-cap system
The End-cap system is a recirculating system in which the Balls advance by rolling through the screw groove between the Nut and the Screw Shaft. The Balls are then returned via the holes in the Nut and the channels in the recirculating sections of the End-caps on either end of the Nut.



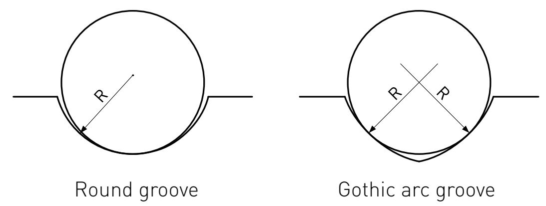
●End-deflector system
The Balls are circulated from End-deflector incorporated inside the Nut or outside the Nut through the hole in the Nut and the channels in the recirculating sections. Ball Nut diameter can be smaller than Return-plate system. This is suitable for the middle lead Ball Screws.



●Return-tube system
In the Return-tube system, Balls rolling between the Nut and the Shaft are picked up from the screw groove by the end of the Return-tube built into the Nut. Then, they flow back through the Return-tube to the screw groove.



●Thread Groove profile
Ball screws may have either a circular arc profile, formed of a single arc, or a gothic arc profile, formed from two arcs.
KSS Ball Screws feature a gothic arc profile.



The range of manufacturing for Ball Screws

The range of manufacturing for KSS Ball Screws is from φ1.8 to φ16mm as Shaft nominal diameter. Maximum limit of overall lengths are shown below. Maximum limit of overall lengths will vary depending on the Shaft end configuration, materials and KSS series. Please inquire KSS for details.

●Maximum limit of overall lengths for Precision Ball Screws

Unit:mm

Shaft nominal diameter \ Accuracy grade	C0	C1	C3	C5
4	90	120	160	170
6	140	180	240	250
8	200	250	330	350
10	260	320	420	450
12	320	390	510	550
14	380	460	600	660
16	450	540	700	770

Note 1)If required length exceeds the number in table above, please ask KSS representative.

●Maximum limit of overall lengths for Rolled Ball Screws(Ct7 & Ct10)

Unit :mm

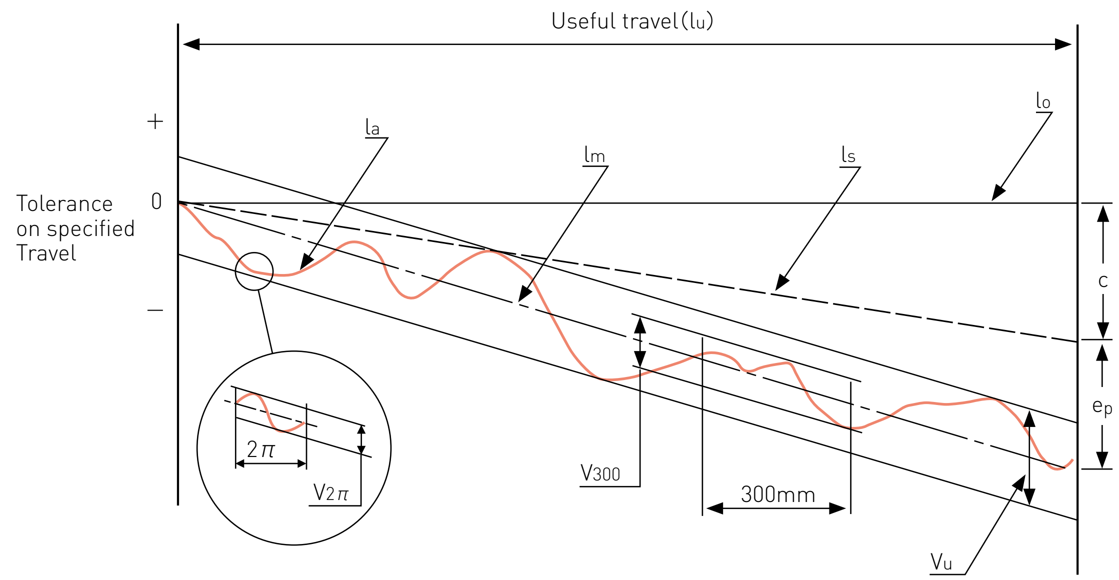
Shaft nominal diameter	Maximum length
4	240
5	300
6	350
8	450
10	650
12	700
13	700
14	700
15	1000

Note 1)If required length exceeds the number in table above, please ask KSS representative.
Note 2)Maximum limit of overall length for Rolled Ball Screws includes 25mm of incomplete thread area at both end.

Lead accuracy of Ball Screws

Ball Screw lead accuracy conforming to JIS B 1192-3 is specified by the tolerance on specified travel over the Nut effective travel amount, or Screw Shaft useful travel, travel variation and travel variation within arbitrary 300mm, and 1 revolution(2π rad) over the Screw Shaft useful travel.
Tolerance of each accuracy grades are shown in the Table A-83, 84, 85.

Fig. A-82：Travel deviation diagram



Nominal travel(l_0) : Travel in axial direction when rotated arbitrary number of revolution according to the Nominal lead

Specified Lead(Phs) : Lead given some amount of correction to the Nominal lead in order to compensate the deformation generated due to the temperature rise or the load.

Travel compensation(c) : Difference between the Specified travel and the Nominal travel within the valid travel.

Specified travel(ls) : Travel in axial direction when rotated arbitrary number of revolution according to the Specified lead.

Actual travel(la) : Actual travel of Ball Nut in axial direction in respect to an arbitrary angle of rotation of Ball Screw Shaft.

Actual mean travel(lm) : Straight line which represents the tendency of Actual travel. It is obtained by the least square method or a simple and appropriate approximation method from the curve indicating the Valid travel of Ball Nut.

Tolerance on specified travel(e_p) : Difference between the Actual mean travel and the Specified travel corresponding to the Valid travel of Ball Nut or the Useful travel of Ball Screw Shaft.

Travel variation(V_u) : Maximum width of the Actual travel curve between the two straight lines put in parallel to the Actual mean travel line, that corresponding to Valid travel of Ball Nut or Useful travel of Ball Screw Shaft.

Travel variation(V_{300}) : Maximum width of the Actual travel curve between the two straight lines put in parallel to the Actual mean travel line, that corresponding to arbitrary 300mm taken within Useful travel of Ball Screw Shaft.

Travel variation($V_{2\pi}$) : Maximum width of the Actual travel curve between the two straight lines put in parallel to the Actual mean travel line, that corresponding to arbitrary one revolution (2π rad) within Useful travel of Ball Screw Shaft.

Table A-83 : Tolerance on specified travel ($\pm e_p$) and permissible travel variation(Vu) of precision Ball Screws (for positioning : C series)

Unit : μm

Accuracy Grade			C0		C1		C3		C5	
Useful travel (mm)	Over	Up to	$\pm e_p$	Vu	$\pm e_p$	Vu	$\pm e_p$	Vu	$\pm e_p$	Vu
	—	100	3	3	3.5	5	8	8	18	18
	100	200	3.5	3	4.5	5	10	8	20	18
	200	315	4	3.5	6	5	12	8	23	18
	315	400	5	3.5	7	5	13	10	25	20
	400	500	6	4	8	5	15	10	27	20
	500	630	6	4	9	6	16	12	30	23
	630	800	7	5	10	7	18	13	35	25
	800	1000	8	6	11	8	21	15	40	27

Table A-84 : Permissible travel variation V_{300}, $V_{2\pi}$ (for positioning : C series)

Unit : μm

Accuracy grade	C0		C1		C3		C5	
Item	V_{300}	$V_{2\pi}$	V_{300}	$V_{2\pi}$	V_{300}	$V_{2\pi}$	V_{300}	$V_{2\pi}$
Permissible value	3.5	3	5	4	8	6	18	8

Table A-85 : Permissible travel variation V_{300} for Ct series(7,10 grade)

Unit : μm

Accuracy grade	Ct7	Ct10
V_{300}	52	210

Tolerance on specified travel(e_p) for Ct series is calculated as follows.

$$e_p = \pm \frac{lu}{300} \times V_{300}$$

Useful travel(mm)

Japan Industrial Standard of Ball Screw(JIS B1192)was revised in 1997, 2013 and 2018 in order to correspond to ISO.
Regarding accuracy grade, C series(current JIS C0, 1, 3, 5) and Cp, Ct series (standard corresponding to ISO) are established. KSS conforms to JIS B 1192-3(2018) and adopts C series for 0,1,3,5 grade, Cp, Ct series for 7,10 grade.

Ball Screw Run-out and location tolerances

Japan Industrial Standard of Ball Screw(JIS B1192)was revised in 1997, 2013 and 2018 in order to correspond to ISO. Regarding accuracy grade, C series(current JIS C0, 1, 3, 5) and Cp, Ct series (standard corresponding to ISO) are established. There are some differences between C series and Cp, Ct series in notation and tolerances for accuracy of Ball Screw mounting section. KSS uses notation in Fig. A-86 below and standard tolerance value, which conforms to C series standard, and KSS refers to Cp, Ct series standard in case of 7 and 10 grade. Moreover, in the revision of 2018, the notation of perpendicularity changed to "run-out of the mounting surface or end face", and geometric tolerance symbols changed from ⊥ to ↗.

Fig. A-86 : Description of Run-out and location tolerances for Ball Screws

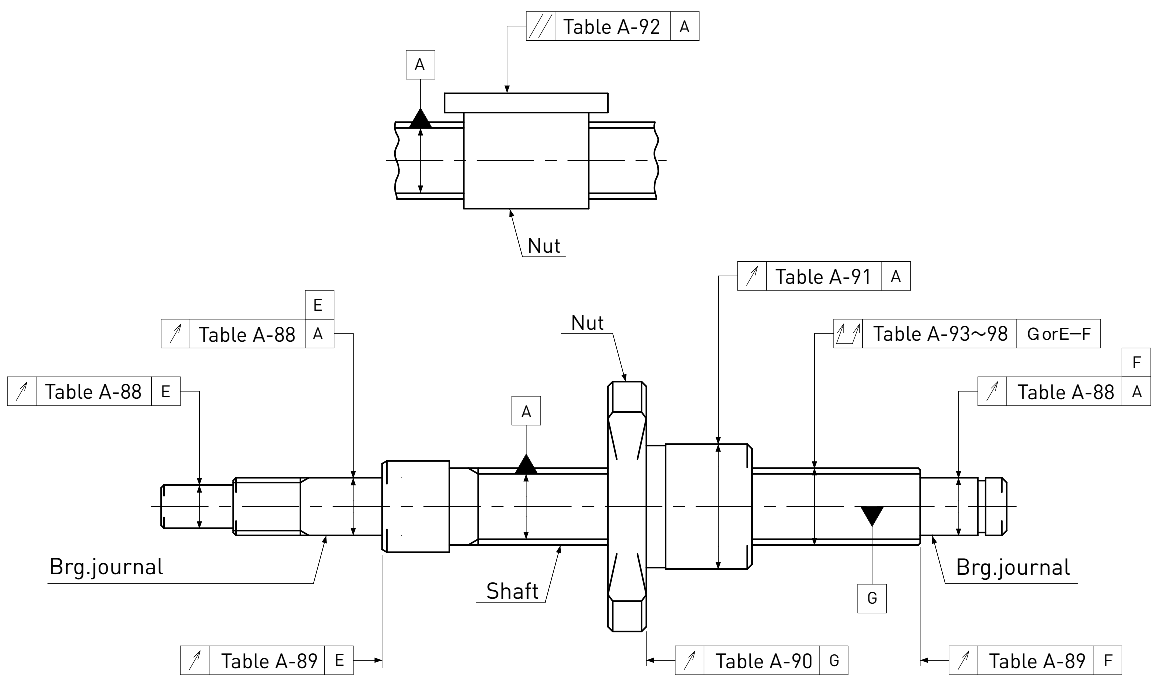


Table A-88：Radial Run-out of Bearing seat related to the centerline of screw groove and Radial Run-out of journal diameter related to the Bearing seat

Unit：μm

Shaft nominal diameter (mm)		Permissible deviation of Radial Run-out					
Over	Up to	C0	C1	C3	C5	C7	C10
—	8	3	5	8	10	14	40
8	12	4	5	8	11	14	40
12	20	4	6	9	12	14	40

This measurement item is affected by Total Run-out of the Screw Shaft, and so it must be corrected as follows. Find the corrected value from the Total Run-out tolerances given in Tables A-93~98 on page A809 ~A811 using the ratio of the total Shaft length to the distance between the supporting point and the measuring point(L_1,L_2) (see Fig. A-87), and add the values obtained to the tolerance given in Table A-88.

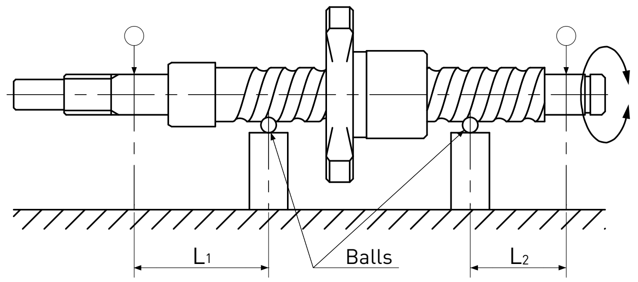


Fig. A-87：Compensation of Radial Run-out

$$\text{Compensation Value of Run-out} = \frac{\text{Tolerance of total Run-out (Table A-93~98)}}{\text{Total shaft length}} \times (L_1 \text{ or } L_2)$$

L_1 ,L_2：Distance btw supporting pt & measuring pt(mm)

Table A-89：Axial Run-out (Perpendicularity) of Shaft(Bearing) face related to the centerline of the Bearing seat

Unit：μm

Shaft nominal diameter (mm)		Permissible deviations of Axial Run-out (Perpendicularity)					
Over	Up to	C0	C1	C3	C5	C7	C10
—	8	2	3	4	5	7	10
8	12	2	3	4	5	7	10
12	20	2	3	4	5	7	10

Table A-90：Axial Run-out (Perpendicularity) of Ball Nut location face related to the centerline of Screw Shaft

Unit：μm

Nut outside diameter (mm)		Permissible deviations of Axial Run-out (Perpendicularity)					
Over	Up to	C0	C1	C3	C5	C7	C10
—	20	5	6	8	10	14	20
20	32	5	6	8	10	14	20
32	50	6	7	8	11	18	30

Table A-91：Radial Run-out of Ball Nut location diameter related to the centerline of Screw Shaft

Unit：μm

Nut outside diameter (mm)		Permissible deviations of Radial Run-out					
Over	Up to	C0	C1	C3	C5	C7	C10
—	20	5	6	9	12	20	40
20	32	6	7	10	12	20	40
32	50	7	8	12	15	30	60

Table A-92：Parallelism of rectangular Ball Nut related to the centerline of Screw Shaft

Unit：μm

Mounting length (mm)		Permissible deviations of Parallelism					
Over	Up to	C0	C1	C3	C5	C7	C10
—	50	5	6	8	10	17	30
50	100	7	8	10	13	17	30

Table A-93：Total Run-out in radial direction of Screw Shaft related to the centerline of Screw Shaft(C0)

Unit：mm

Shaft total length		Shaft nominal diameter		
	Over	—	8	12
	Up to	8	12	20
Over	Up to	Permissible deviations of total Run-out in radial direction		
—	125	0.015	0.015	0.015
125	200	0.025	0.020	0.020
200	315	0.035	0.025	0.020
315	400	—	0.035	0.025
400	500	—	0.045	0.035
500	630	—	0.050	0.040
630	800	—	—	0.050
800	1000	—	—	0.065

Table A-94：Total Run-out in radial direction of Screw Shaft related to the centerline of Screw Shaft(C1)

Unit：mm

Shaft total length		Shaft nominal diameter		
	Over	—	8	12
	Up to	8	12	20
Over	Up to	Permissible deviations of total Run-out in radial direction		
—	125	0.020	0.020	0.015
125	200	0.030	0.025	0.020
200	315	0.040	0.030	0.025
315	400	0.045	0.040	0.030
400	500	—	0.050	0.040
500	630	—	0.060	0.045
630	800	—	—	0.060
800	1000	—	—	0.075

Table A-95：Total Run-out in radial direction of Screw Shaft related to the centerline of Screw Shaft(C3)

Unit：mm

Shaft total length		Shaft nominal diameter		
	Over	—	8	12
	Up to	8	12	20
Over	Up to	Permissible deviations of total Run-out in radial direction		
—	125	0.025	0.025	0.020
125	200	0.035	0.035	0.025
200	315	0.050	0.040	0.030
315	400	0.060	0.050	0.040
400	500	—	0.065	0.050
500	630	—	0.070	0.055
630	800	—	—	0.070
800	1000	—	—	0.095

Table A-96：Total Run-out in radial direction of Screw Shaft related to the centerline of Screw Shaft(C5)

Unit：mm

Shaft total length		Shaft nominal diameter		
	Over	—	8	12
	Up to	8	12	20
Over	Up to	Permissible deviations of total Run-out in radial direction		
—	125	0.035	0.035	0.035
125	200	0.050	0.040	0.040
200	315	0.065	0.055	0.045
315	400	0.075	0.065	0.055
400	500	—	0.080	0.060
500	630	—	0.090	0.075
630	800	—	—	0.090
800	1000	—	—	0.120

Table A-97：Total Run-out in radial direction of Screw Shaft related to the centerline of Screw Shaft(C7)

Unit：mm

Shaft total length		Shaft nominal diameter		
	Over	—	8	12
	Up to	8	12	20
Over	Up to	Permissible deviations of total Run-out in radial direction		
—	125	0.060	0.055	0.055
125	200	0.075	0.065	0.060
200	315	0.100	0.080	0.070
315	400	—	0.100	0.080
400	500	—	0.120	0.095
500	630	—	0.150	0.110
630	800	—	—	0.140
800	1000	—	—	0.170

Table A-98：Total Run-out in radial direction of Screw Shaft related to the centerline of Screw Shaft(C10)

Unit：mm

Shaft total length		Shaft nominal diameter		
	Over	—	8	12
	Up to	8	12	20
Over	Up to	Permissible deviations of total Run-out in radial direction		
—	125	0.100	0.095	0.090
125	200	0.140	0.120	0.110
200	315	0.210	0.160	0.130
315	400	—	0.210	0.160
400	500	—	0.270	0.200
500	630	—	0.350	0.250
630	800	—	0.460	0.320
800	1000	—	—	0.420

Note)In case of Ct7, Ct10 grade, KSS may use the standard of Total Run-out based on slenderness ratio, which conforms to JIS B1192-2013.

Slenderness ratio		Total Run-out	
Over	Up to	Ct7	Ct10
—	40	0.080	0.160
40	60	0.120	0.240
60	80	0.200	0.400
80	100	0.320	0.640

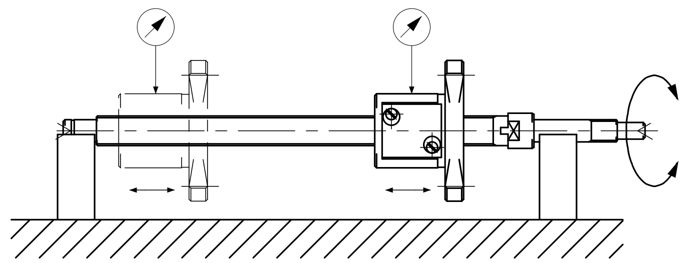
Slenderness ratio= lu/do

lu: Useful travel(mm)
do: Nominal diameter of Ball Screw(mm)

Measuring method of Ball Screw Run-out and location tolerances

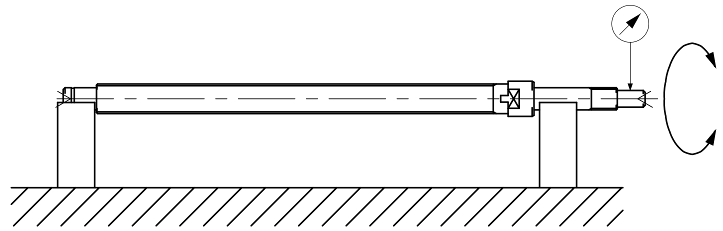
●Radial Run-out of Bearing seat related to the centerline of screw groove (Table A-88)

Place the Ball Screw in identical V-blocks at both Bearing seat. Place the dial gauge perpendicular to the Nut cylindrical surface. Rotate Screw Shaft slowly and record the dial gauge readings. Measurement should be done at near both ends of threaded part. Some cases, this measurement will be done by both centerhole support, and directly measured on Bearing seat.



●Radial Run-out of journal diameter related to the Bearing seat(Table A-88)

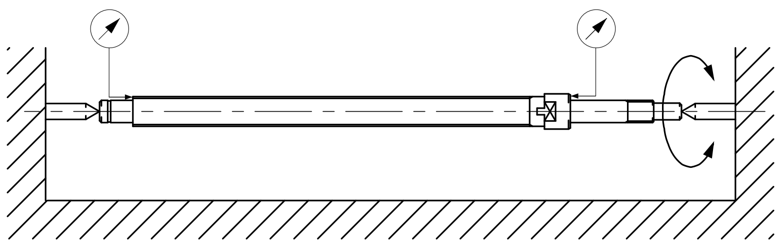
Place the Ball Screw in identical V-blocks at both Bearing seats. Place the dial gauge perpendicular to the journal cylindrical surface. Rotate the Screw Shaft slowly and record the dial gauge readings.



●Axial Run-out (Perpendicularity) of shaft(Bearing) face related to the centerline of the Bearing seat(Table A-89)

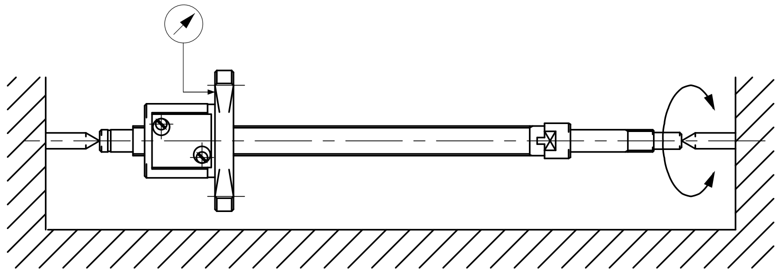
Support a Screw Shaft at both centers. Place the dial gauge perpendicular to the end face of the journal. Rotate the Screw Shaft slowly and record the dial gauge readings.

**This method is equivalent to the one, which is supported at both Bearing seats, because Bearing seats are ground related to both centers.



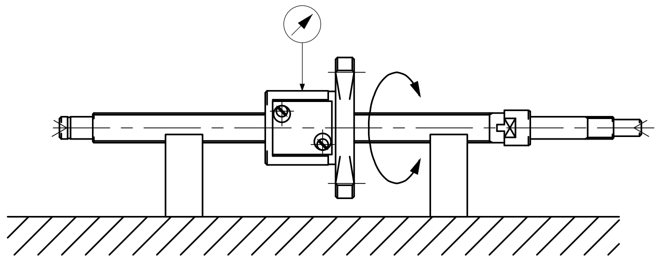
●Axial Run-out (Perpendicularity) of Ball Nut location face related to the centerline of Screw Shaft (Table A-90)

Support the Ball Screw at both centers. Place the dial gauge perpendicular to the flange face. Rotate the Screw Shaft with Ball Nut slowly and record the dial gauge readings. Secure the Ball Nut against rotation on the Screw Shaft.



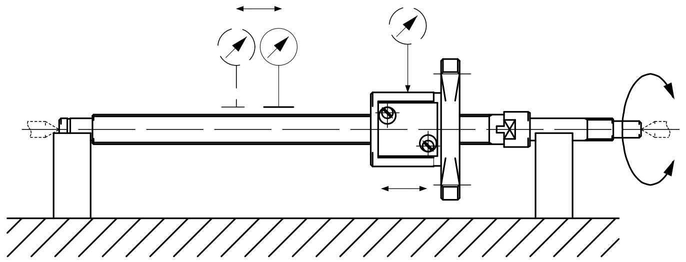
●Radial Run-out of Ball Nut location diameter related to the centerline of Screw Shaft (Table A-91)

Place the Ball Screw on V-blocks at adjacent sides of the Ball Nut. Place the dial gauge perpendicular to the cylindrical surface of Ball Nut. Secure the Screw Shaft against rotation of Ball Nut. Rotate Ball Nut slowly and record the dial gauge readings.



●Total Run-out in radial direction of Screw Shaft related to the centerline of Screw Shaft (Table A-93～98)

Place the Ball Screw in identical V-blocks at both Bearing seats, or support the Ball Screw at both centers. Place the dial gauge with measuring shoe at the several points over the full thread length.Rotate the Screw Shaft slowly and record the dial gauge readings. Maximum value of measurement should be the Total Run-out.



Material and Heat treatment, Surface hardness

Standard material of KSS Ball Screws, Heat treatment and Surface hardness are shown in table A-99, 100. However, they vary depending on series or model number. Please refer to KSS drawings.

Table A-99：Material, Heat treatment & Surface hardness for regular items

	Material	Heat treatment	Surface hardness
Screw Shaft	SCM415 (JIS G 4105)	Carburizing and quenching	HRC 58-62
	S55C (JIS G 4051)	Induction hardening	HRC min.58
Nut	SCM415 (JIS G 4105)	Carburizing and quenching	HRC 58-62

Note 1)Hardness on table shows surface hardness of thread part.
Note 2)S55C is applicable for Precision Rolled Ball Screws.

Table A-100：Material, Heat treatment & Surface hardness for stainless steel items

	Material	Heat treatment	Surface hardness
Screw Shaft	SUS440C (JIS G 4303)	Quenching and tempering	HRC min.55
Nut	SUS440C (JIS G 4303)	Quenching and tempering	HRC min.55

Note)Hardness on table shows surface hardness of thread part.

Permissible Axial load

It is recommended that Ball Screw Shafts be used almost exclusively under tension load conditions. However, in some applications, compression loads may exist, and under such conditions it must be checked that Shaft buckling will not occur.
Also, when the mounting span distance is short, there is a restriction on the permissible tension or compression load and the Basic Static Load Rating Coa unrelated to mounting.
Buckling load, permissible tension and permissible compression load can be calculated below.

●Permissible compression load calculation for buckling

$$P = \alpha \times \frac{n\pi^2 E \cdot I}{L^2} \quad \text{N} \quad \text{Formula for Oiler}$$

α : Safety Factor 0.5
E : Young's modulus 2.08 $\times 10^5$ N/mm^2(MPa)
I : Screw Shaft minimum moment of inertia of area

$$I = \frac{\pi}{64} d^4 \quad \text{mm}^4$$

d : Screw Shaft Root diameter mm
L : Mounting span distance mm
n : Factor for Ball Screw mounting method

Supported−Supported	n = 1
Fixed−Supported	n = 2
Fixed−Fixed	n = 4
Fixed−Free	n = 1/4

●Permissible tension, compression load calculation for Screw Shaft yield stress

$$P = \sigma \times A \quad \text{N}$$

σ : Permissible stress 98N/mm^2 (MPa)
A : Screw Shaft minimum section area

$$A = \frac{\pi}{4} d^2 \quad \text{mm}^2$$

d : Screw Shaft Root diameter mm

Permissible speed

For Screw Shaft rotation, the mounting method determines the established rotation limits. When this value is approached, resonance phenomenon will occur, and operation becomes impossible. There is also rotation limit which causes damages to recirculating parts. This limit is unrelated to mounting methods.

●Permissible speed calculation for critical speed

$$N = \beta \times \frac{60 \cdot \lambda^2}{2\pi} \times \sqrt{\frac{E \cdot I \cdot g}{\gamma \cdot A \cdot L^4}} \quad \text{min}^{-1}$$

β : Safety Factor 0.8
E : Young's modulus 2.08 $\times 10^5$ N/mm^2 (MPa)
I : Screw Shaft minimum moment of inertia of area

$$I = \frac{\pi}{64} d^4 \quad \text{mm}^4$$

d : Screw Shaft Root diameter mm
g : Gravity acceleration 9.8$\times 10^3$ mm/sec^2
γ : Material specific gravity 7,850kg/m^3 (7.7 $\times 10^{-5}$ N/mm^3)
L : Mounting span distance mm
A : Screw Shaft minimum section area

$$A = \frac{\pi}{4} d^2 \quad \text{mm}^2$$

λ :Factor for Ball Screw mounting method

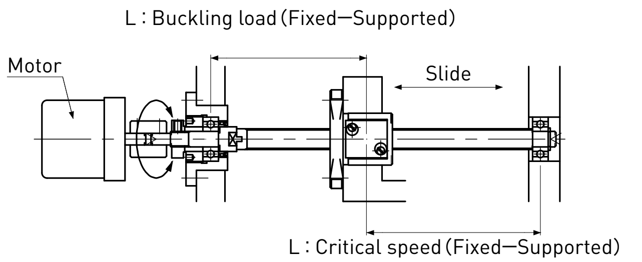
Supported−Supported	$\lambda = \pi$
Fixed−Supported	$\lambda = 3.927$
Fixed−Fixed	$\lambda = 4.730$
Fixed−Free	$\lambda = 1.875$

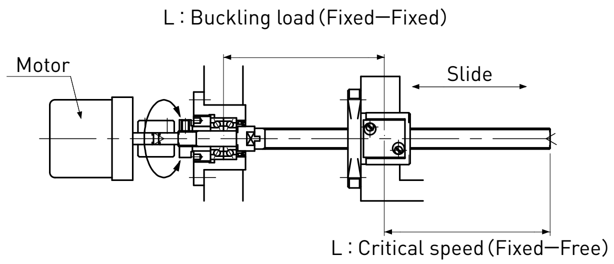
●Rotational speed limit for damage on recirculating parts

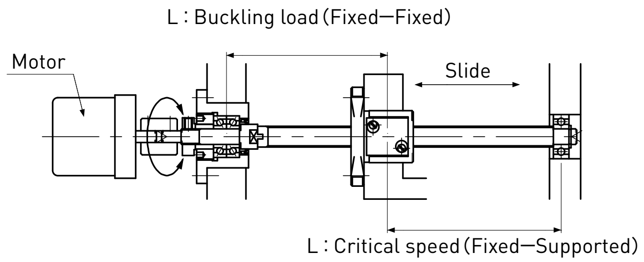
Generally, regarding critical speed for damage on recirculating parts, limitation is established by dn value, which is multiplied Shaft nominal diameter of revolution, but dn value cannot be applied to Miniature Ball Screws. For KSS Ball Screws, please consider rotational speed limit by damage on recirculating parts as 3,500 to 4,000min^{-1}. This value varies depending on operating conditions and environment. Please inquire KSS for details.
Moreover, possibilities of breakage of recirculating parts will be increased when using in high acceleration / deceleration.
Estimate criterion of the breakage in the recirculating section is depending on the internal specification of the Ball Screw, please ask KSS for more detail.

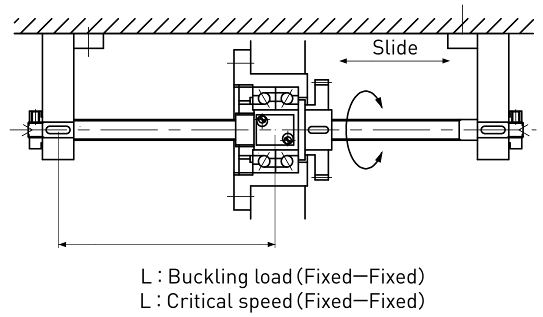
Ball Screw mounting methods

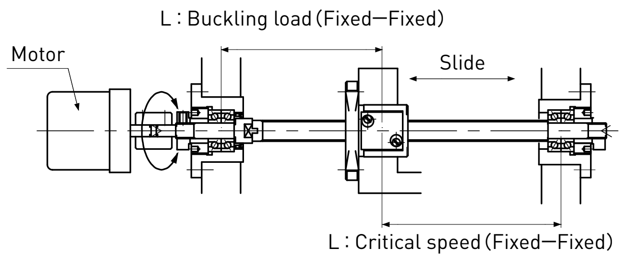
Typical Ball Screw's mounting methods are shown in Fig. A-101. Mounting configuration affects permissible Axial load in relation to buckling, as well as permissible speed in relation to critical speed. Please refer to below when studying strength and speed.

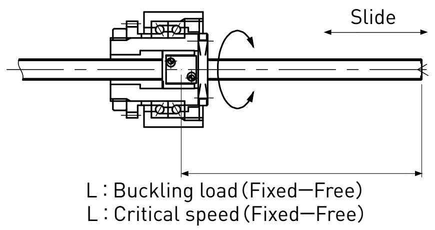












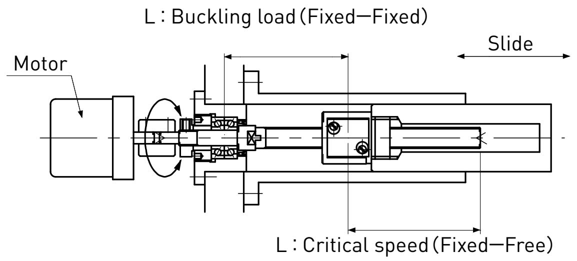


Fig. A-101：Ball Screw mounting methods

Axial play and Preload

For standard Single Nut Ball Screws under normal conditions, a slight Axial play exists between the Screw Shaft and Nut. Consequently, when Axial loads act on Single Nut Ball Screws, total amount of Axial play and Elastic displacement due to Axial load becomes backlash. In order to prevent this backlash in Ball Screws, the Axial play can be reduced to a negative value. That is what we call "Preload", which is the method of causing Elastic deformation to the Balls between the Screw Shaft and Nut in advance.

●Axial play

Symbol and permissible value for Axial play are shown in Table A-102.
Combination of accuracy grade and symbol are shown in Table A-103.

Table A-102：Symbol and permissible value for Axial play

Unit：mm

Symbol	0	02	05	20	50
Axial play	0 (Preloading)	0.002 max.	0.005 max.	0.02 max.	0.05 max.

Table A-103：Combination of accuracy grade and Axial play

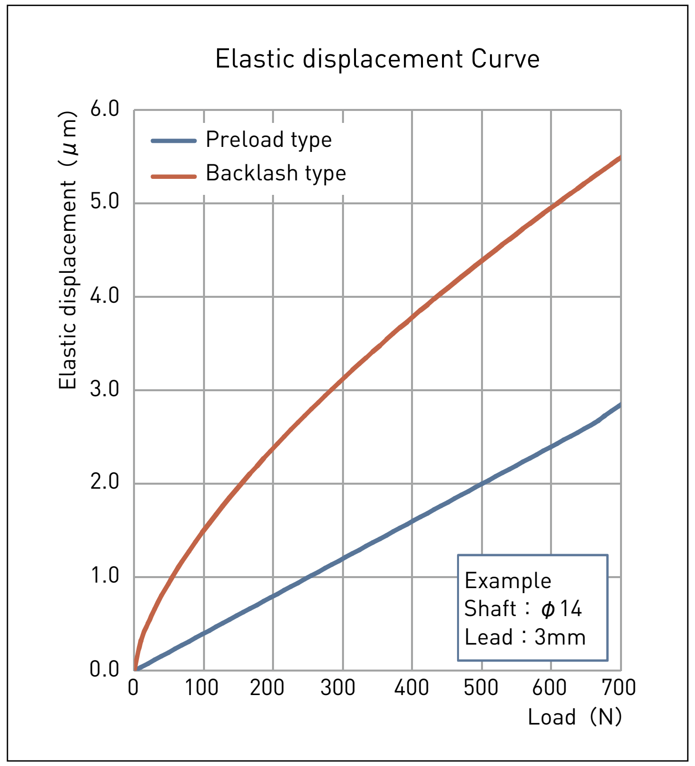
Symbol / Accuracy grade	0	02	05	20	50
C0	C0-0	−	−	−	−
C1	C1-0	C1-02	−	−	−
C3	C3-0	C3-02	C3-05	C3-20	C3-50
C5	−	−	C5-05	C5-20	C5-50
C7	−	−	−	C7-20	C7-50
C10	−	−	−	C10-20	C10-50

Note) When combinations other than the above are requested, please inquire KSS.

●Preload effect

Preload is not only used for removing Axial play, it also has the effect of reducing the amount of Axial displacement due to Axial load, and improving the Rigidity in Ball Screws. Fig. A-104 shows the difference of the amount of Elastic displacement (theoretical value) regarding Ball Screw with Axial play and Ball Screw with Preload under the Axial load.

Fig. A-104：Elastic displacement curve comparison between Backlash type and Preload type



●Proper amount of Preload

Although the amount of Preload should be determined by the required Rigidity and the permissible amount of backlash, when setting Preload, there are some concerning issues as follows.

1)Increased Dynamic Drag Torque
2)Heat generation, lowering of positioning accuracy due to the temperature rise.
3)Shortened life

Therefore, it is advisable to establish the amount of Preload at the lowest possible limits.

●Preload methods

Generally, a method of Double Nut Preload by inserting a spacer between two Nuts is adopted. KSS Ball Screw adopts「Oversized Ball Preload」by inserting Balls slightly bigger than space between Screw Shaft and Nut. As a result, it can eliminate Axial play even with a Single Nut and it is possible to maintain compact. Moreover, operating performance will never be deteriorated by using spacer Balls (Balls with slightly smaller diameter than those of the oversize Balls) alternatively with oversize Balls.

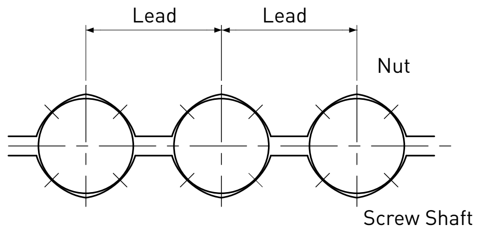


Fig. A-105：Preload by oversized Balls

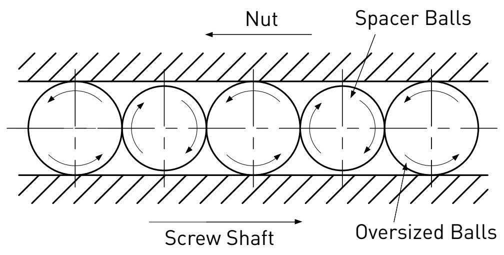
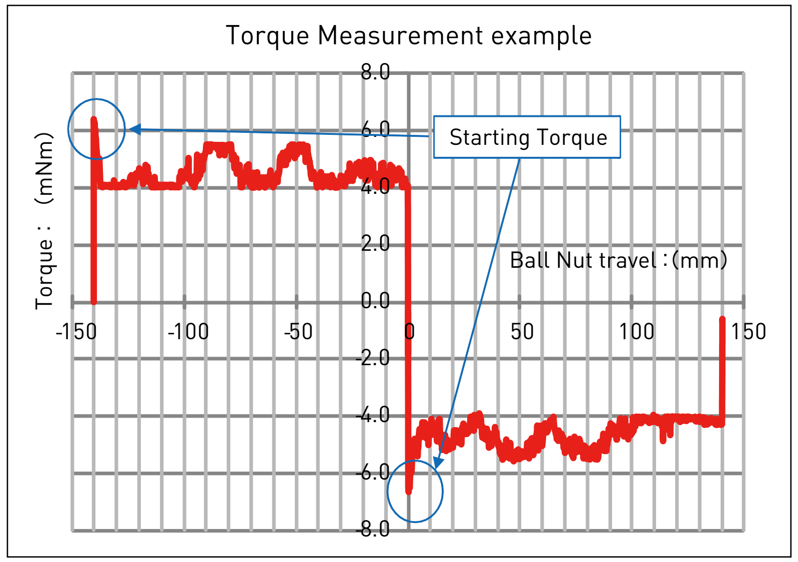


Fig. A-106：Spacer Balls

●Preload control

It is difficult to control Preload amount by measuring. Therefore, Preload of Ball Screw is controlled by measuring Preload Dynamic Drag Torque, which is converted from Preload amount. Amount of Preload Dynamic Drag Torque is decided with customers by specification drawing. Preload Dynamic Drag Torque is measured under specific condition to verify the amount of Axial play is 0. Dynamic Drag Torque installed actual machine will vary depending on lubricating condition, load condition and so on. Starting torque (Torque for starting Ball Screw) is slightly bigger than Dynamic Drag Torque.



*Torque wave in this diagram is exaggerated for explanation.

Fig. A-107：Dynamic Drag Torpue measurement

Rigidity in Linear Motion system

In precision machinery, to improve positioning accuracy of the drive screws or to increase Rigidity for load, the Rigidity of the entire Linear Motion system must be examined. Rigidity of entire Linear Motion system is as follows.

$$\frac{1}{K} = \frac{1}{K_1} + \frac{1}{K_2} + \frac{1}{K_3} + \frac{1}{K_4} \quad \mu m/N$$

K	: Total Rigidity of Linear Motion system	N/μm
K_1	: Screw Shaft Rigidity	N/μm
K_2	: Nut Rigidity	N/μm
K_3	: Support Bearing Rigidity	N/μm
K_4	: Nut, Bearing fitting part Rigidity	N/μm

●Total Rigidity of Linear Motion system K

$$K = \frac{Fa}{\delta} \quad N/\mu m$$

Fa	: Axial load applied to Linear Motion system	N
δ	: Elastic displacement of Linear Motion system	μm

●Screw Shaft Rigidity K_1

(1) In case of general mounting (Fixed-Free in axial direction) (Fig. A-108)

$$K_1 = \frac{A \cdot E}{\ell} \times 10^{-3} \quad N/\mu m$$

(2) In case of Fixed-Fixed mounting in axial direction (Fig. A-109)

$$K_1 = \frac{A \cdot E \cdot L}{\ell\,(L-\ell)} \times 10^{-3} \quad N/\mu m$$

The max. axial displacement occurs when $\ell = L/2$. The formula is as follows.

$$K_1 = \frac{4 \cdot A \cdot E}{L} \times 10^{-3} \quad N/\mu m$$

A : Screw Shaft minimum section area

$$A = \frac{\pi}{4} d^2 \quad mm^2$$

d	: Screw Shaft Root diameter	mm
E	: Young's modulus	2.08×10^5 N/mm² (MPa)
ℓ	: Axial distance between fixed point & Nut center	mm
L	: Mounting span distance	mm

Accordingly, the amount of Screw Shaft Elastic displacement δ due to Axial load Fa is as follows.

$$\delta = \frac{Fa}{K_1} \quad \mu m$$

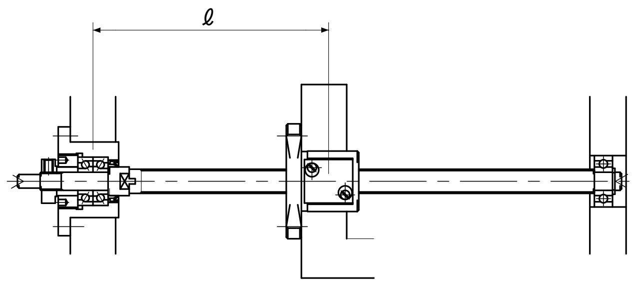


Fig. A-108 : Fixed-Free in axial direction

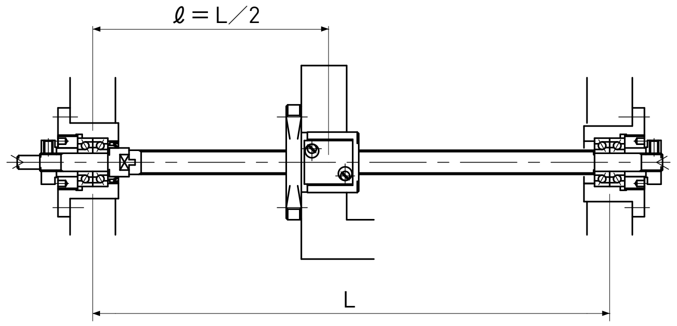


Fig. A-109 : Fixed-Fixed in axial direction

●Nut Rigidity K_2

Calculation formula of static Rigidity is defined by JIS B1192-4 established in 2018. KSS will use the formula which is defined by JIS to identify the static Rigidity.

(1) Rigidity of Single Nut with backlash

Theoretical static Rigidity(K_2) of the Single Nut with backlash is calculated by the formula as follows.

$$K_2 = f_{ar} \times (3/2) \times Fa/\delta \quad (N/\mu m)$$

K_2	: Theoretical Nut Rigidity	$N/\mu m$
Fa	: Axial Load	N
δ	: Amount of Elastic displacement at Axial Load Fa	μm
f_{ar}	: Correction factor = 0.67	

$$\delta = k \times Fa^{2/3} \quad (\mu m)$$

$$k = \frac{C}{Z^{2/3} \times Dw^{1/3} \times (\sin\alpha \times \cos\beta)^{5/3}}$$

k	: Rigidity characterization factor	
Z	: Quantity of loaded Ball)	個(qty.)
Dw	: Diameter of Ball	mm
α	: Contact angle to the thread groove	度(deg.)
β	: Lead angle	度(deg.)
C	: Coefficient depending on the material, shape and dimension	0.52～0.58

The theoretical static Rigidity K_2 of the Nut under an Axial load equivalent to 30% of the Basic Dynamic Load Rating Ca is described in dimension table. For Axial loads which are not 30% of the Basic Dynamic Load Rating Ca, it can be easily calculated by following formula.

$$K'_2 = K_2 \times \left(\frac{Fa}{0.3Ca}\right)^{1/3} \quad N/\mu m$$

K_2	: Nut Rigidity in dimension table	$N/\mu m$
Fa	: Axial load	N
Ca	: Basic Dynamic Load Rating	N

(2) Rigidity of preloaded Ball Nut

Theoretical static Rigidity(K_2) of the preloaded single Ball Nut will become a fixed value if axial load (Fa) is less than $2\sqrt{2}$ times of the preload amount(F_{pr}) regardless of the value of the axial load(Fa), and this will be calculated as follows.

$$K_2 = 2^{3/2} \times \frac{1}{k} \times F_{Pr}^{1/3} \quad N/\mu m$$

k	: Rigidity Characterization factor See formula stated above	
F_{pr}	: Preload amount	N

In case of Preload type Ball Screws, Rigidity varies depending on the dispersion of Preload Dynamic Drag Torque. Therefore, please inquire KSS for details.
If the axial load(Fa) will be more than $2\sqrt{2}$ times of the preload amount(F_{pr}), the calculation formula will be the same as the formula for single Nut Theoretical static Rigidity.

The theoretical static Rigidity K_2 under a Preload equivalent to 5% of the Basic Dynamic Load Rating Ca is described in dimension table. For Preload amounts other than the above, it can be easily calculated by following formula.

$$K'_2 = K_2 \times \left(\frac{F_{pr}}{0.05Ca}\right)^{1/3} \quad N/\mu m$$

K_2	: Nut Rigidity in dimension table	$N/\mu m$
F_{pr}	: Preload amount	N
Ca	: Basic Dynamic Load Rating	N

●Support Bearing Rigidity K_3

Support Bearing Rigidity varies depending on the type of Bearing and amount of Preload. Please inquire Bearing manufacturers.

●Nut, Bearing fitting part Rigidity K_4

Rigidity of Nut mounting part and Bearing mounting part vary depending on machine structure and design. KSS cannot mention the details but a design of high Rigidity must be considered.

●Screw Shaft torsion Rigidity

For positioning error due to torsion, this error is a relatively small compared to axial displacement. However, if investigation is required, the following formula may be used for calculation.

$$\theta = \frac{32TL}{\pi Gd^4} \times \frac{180}{\pi} \times 10 \quad deg$$

θ	: Torsion angle due to torsion moment	deg
T	: Torsion moment	N·cm
L	: Distance between Nut & Shaft end support	mm
G	: Modulus of Rigidity	8.3×10^4 N/mm² (MPa)
d	: Screw Shaft Root diameter	mm

Amount of axial displacement δa due to torsion angle is as follows.

$$\delta a = \ell \times \frac{\theta}{360} \times 10^3 \quad \mu m$$

ℓ : Lead mm

Basic Load Rating and Basic Rating Life

●Basic Dynamic Load Rating Ca and Basic Rating Life

The Basic Rating Life of Ball Screws means the total number of revolutions which 90% of the Ball Screws can endure. Failure is indicated by flaking caused by rolling fatigue on the surface of grooves or Balls. These figures are valid when a group of the same type Ball Screws are operated individually under the same conditions. The Basic Dynamic Load Rating Ca is the Axial load for which the Basic Rating Life is 1,000,000 revolutions. These values are listed under Ca in the dimension tables. Ball Screw's Basic Rating Life L_{10} can be estimated using Basic Dynamic Load Rating Ca in the following basic formula.

$$L_{10} = \left(\frac{Ca}{f \cdot Fa}\right)^3 \times 10^6 \quad \text{rev.}$$

Also, in place of the total number of revolutions, the Basic Rating Life can be expressed in hours: L_{10h} or traveled distance: L_{10d}, and these can be calculated through the following formulas.

$$L_{10h} = \left(\frac{1}{60 \cdot N}\right) \times L_{10} \quad \text{hours}$$

$$L_{10d} = \left(\frac{\ell}{10^6}\right) \times L_{10} \quad \text{km}$$

Ca	: Basic Dynamic Load Rating	N
Fa	: Axial load	N
N	: Revolution	min^{-1}
ℓ	: Lead	mm
f	: Load factor	
	f=1.0～1.2 for almost no vibration, no impact load	
	f=1.2～1.5 for slight vibration, impact load	
	f=1.5～3.0 for severe vibration, impact load	

Generally, Axial load on the most machine is not constant and it can be divided into several operating pattern. In this case, Basic Rating Life can be calculated to figure up equivalent Axial load Fam, equivalent Revolution N_m in the following formula.

Axial load N	Revolution min^{-1}	Frequency of use %
Fa_1	N_1	t_1
Fa_2	N_2	t_2
Fa_3	N_3	t_3

$$Fam = \left(\frac{Fa_1^3 \cdot N_1 \cdot t_1 + Fa_2^3 \cdot N_2 \cdot t_2 + Fa_3^3 \cdot N_3 \cdot t_3}{N_1 \cdot t_1 + N_2 \cdot t_2 + N_3 \cdot t_3}\right)^{1/3} \quad \text{N}$$

$$N_m = \frac{N_1 \cdot t_1 + N_2 \cdot t_2 + N_3 \cdot t_3}{t_1 + t_2 + t_3} \quad min^{-1}$$

Also, for Axial loads which vary linearly, the average Axial load Fam can be calculated approximately using the following formula.

$$Fam = \frac{Fa\,min + 2 \cdot Fa\,max}{3} \quad \text{N}$$

Fa min: Minimum Axial load N
Fa max: Maximum Axial load N

Note) As the Basic Rating Life varies due to lubricating conditions, and contaminations, Moment load or Radial load, etc., this should be considered a rough estimate only.

Load direction and Preload will be taken into consideration when calculate the Basic Rating Life by JIS B1192-5, which was established in 2018. Therefore, KSS uses a calculation formula of Basic Rating Life for Miniature Ball Screws that is conformed to JIS B1192-5.

●Life calculation considered the Load direction

Contact point of the Steel Balls changes based on Load direction (see Fig. A-110), therefore it is considered the lifetime when flaking occurred at any contact points, with calculating the Rating Life at each contact point of the Steel Balls.
The calculating formula is as follows.

$$L'_{10} = \left(L_{10(A)}^{-10/9} + L_{10(B)}^{-10/9}\right)^{-9/10} \quad \text{rev.}$$

L'_{10} : Merged Basic Rating Life of contact point A and B
$L_{10(A)}$: Basic Rating Life on contact point A
$L_{10(B)}$: Basic Rating Life on contact point B

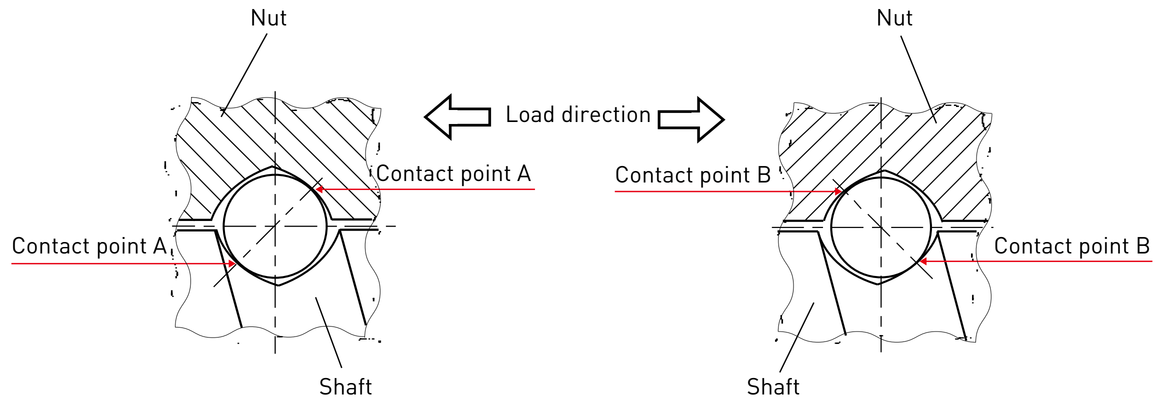


Fig. A-110 : Ball contact condition by load direction

●Life calculation considered the Preload

Preloaded Ball Screw is filled with oversized Balls, therefore each Steel Ball is contacted at four (4) points between Screw Shaft and Ball Nut. It is considered the lifetime when flaking occurred at any contact points, with calculating the Rating Life at each contact point.

The contact point of the Steel Balls is described in Fig. A-111, when Preload is effective by oversized Balls. The amount of Elastic displacement is described schematically by oval (contact ellipse).
Both contact point A and B are evenly contacted under no load from outside.

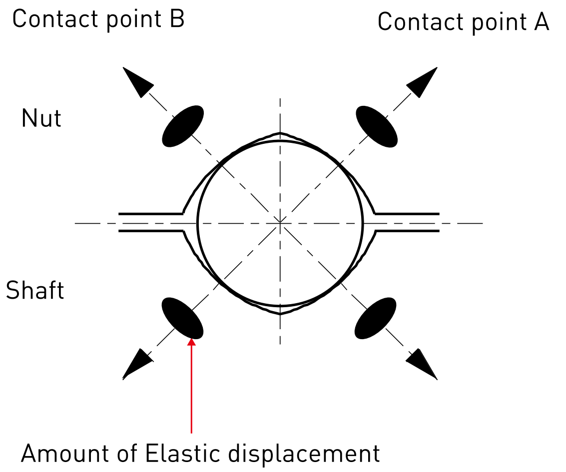


Fig. A-111：Ball Contact condition under Preload

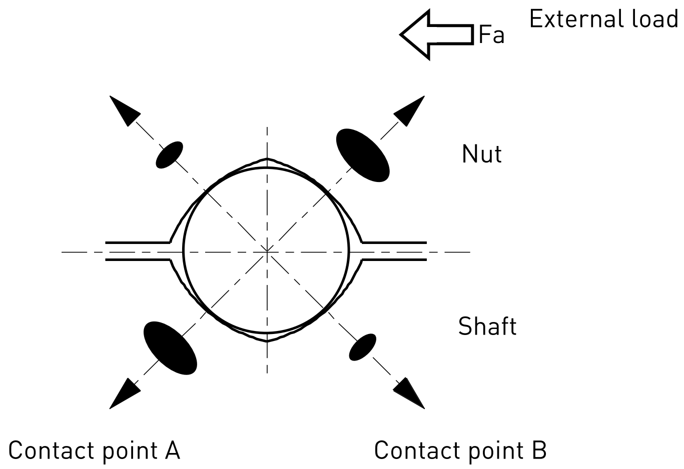


Fig. A-112：Ball contact condition under preload & external load

When external load (Fa) is applied, Elastic displacement increases at contact point A, and decreases at contact point B(see Fig. A-112).
In this case, the load at contact point A and B can be calculated as below based on the Hertz theory of Elastic displacement.
By substituted each values into the formula of Basic Rating Life, Rating Life of each contact point can be calculated.

In case of $Fa \leqq 2\sqrt{2}\,F_{pr}$

$$Fa_{(A)} = F_{pr} \times \left(1 + \frac{Fa}{2^{3/2} \times F_{pr}}\right)^{3/2} \qquad Fa_{(B)} = Fa_{(A)} - Fa$$

Fa	: Amount of external load	N
$Fa_{(A)}$	: Axial load applying on contact point A	N
$Fa_{(B)}$	: Axial load applying on contact point B	N
F_{pr}	: Preload	N

In case of $Fa > 2\sqrt{2}\,F_{pr}$

$$Fa_{(A)} = Fa \qquad Fa_{(B)} = 0$$

Note) Load direction of A and B is opposite.

Using the value calculated by the above formula, calculate the Rating Life at each contact point A and B ($L_{10(A)}$, $L_{10(B)}$), then merge both value to calculate the merged Basic Rating Life.

$$L_{10(A)} = \left(\frac{Ca}{f \cdot Fa_{(A)}}\right)^3 \times 10^6 \quad \text{rev.}$$

$$L_{10(B)} = \left(\frac{Ca}{f \cdot Fa_{(B)}}\right)^3 \times 10^6 \quad \text{rev.}$$

$$L'_{10} = \left(L_{10(A)}^{\,-10/9} + L_{10(B)}^{\,-10/9}\right)^{-9/10} \quad \text{rev.}$$

Note) As a rough estimation of Basic Rating Life,
we consider the Axis load as external load added by preload amount F_{pr} for some cases.

●Basic Static Load Rating Coa

The Basic Static Load Rating Coa is the Axial Static load at which the amount of permanent deformation (Ball + Raceway) occurring at the maximum stress contact point between the Ball and Raceway surfaces is 1/10,000 times the Ball diameter. These values are listed under Coa in the dimension tables. The Basic Static Load Rating Coa values apply to investigation of stationary state or extremely low Revolution load conditions (less than 10 min^{-1}). However, in most cases the amount of permanent deformation causes absolutely no problems under the general conditions. The maximum permissible load Fa max for the screw groove can be found by using the following formula.

$$Fa\ max = \frac{Coa}{fs} \quad N$$

fs：Static safety factor
fs=1～2 for normal operation
fs=2～3 for vibration, impact

●Hardness coefficient

For Surface hardness of less than HRC58(654 Hv10), the Basic Dynamic Load Rating Ca and the Basic Static Load Rating Coa must be adjusted. Adjustment is made by the following formula.

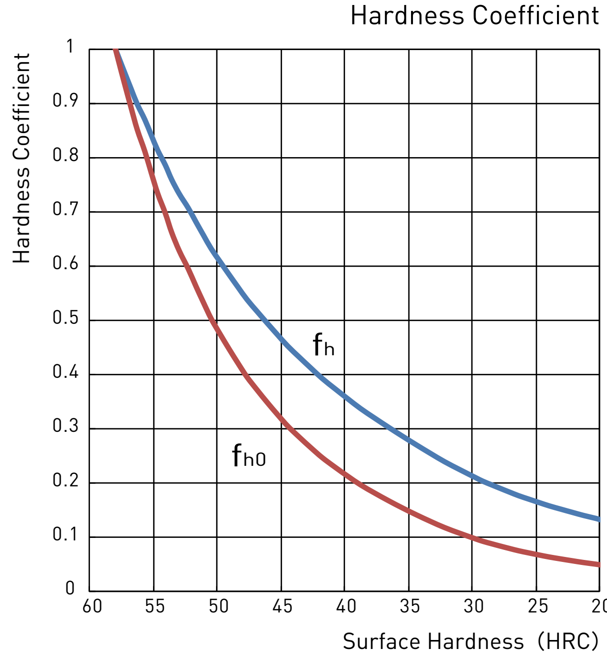
$Ca' = f_h \cdot Ca$ (N)
$Coa' = f_{h0} \cdot Coa$ (N)

$$f_h = \left(\frac{H_a}{654}\right)^2 \leqq 1$$

$$f_{h0} = \left(\frac{H_a}{654}\right)^3 \leqq 1$$

f_h, f_{h0}：Hardness coefficient
(See formula above and graph right)

H_a：Vickers hardness Hv10



Driving Torque

Driving Torque in Linear Motion System T is expressed according to the following formula.

$$T = T_1+T_2+T_3+T_4 \quad N\cdot m$$

T_1 : Acceleration Torque N·m
T_2 : Load Torque N·m
T_3 : Preload Dynamic Drag Torque N·m
T_4 : Additional Torque N·m

When Motor selection, Driving Torque in Linear Motion System is needed.
$T_1 \sim T_3$ can be calculated by the following formula

●Acceleration Torque T_1

$$T_1=\alpha \cdot I \quad N\cdot m$$

$$\alpha = \frac{2\pi N}{60\cdot t} \quad rad/sec^2$$

$$I = Iw\cdot A^2 + Is\cdot A^2 + I_A\cdot A^2 + I_B \quad kg\cdot m^2$$

$$Iw = m_W \times \left(\frac{\ell}{2\pi}\right)^2 \quad kg\cdot m^2$$

$$Is = m_S \times \left(\frac{d^2}{8}\right) \quad kg\cdot m^2$$

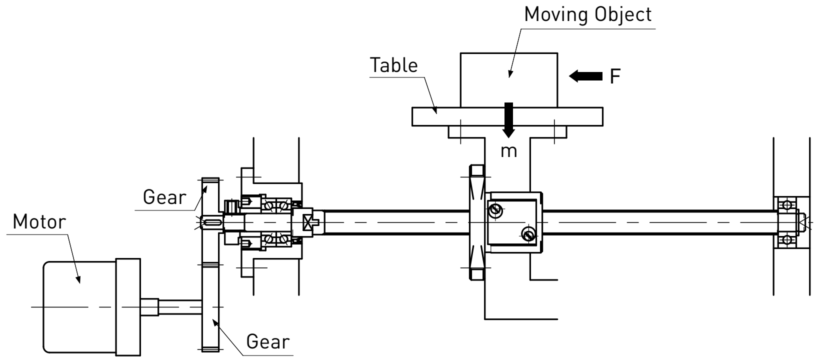
$$m_S = \pi \left(\frac{d}{2}\right)^2 \times L \times \gamma \quad kg$$

α	: Angular acceleration	rad/sec^2
I	: Inertia moment	kg·m^2
Iw	: Inertia moment of moving object by Motor axial conversion	kg·m^2
Is	: Inertia moment of Screw Shaft	kg·m^2
I_A	: Inertia moment of gears on screw side	kg·m^2
I_B	: Inertia moment of gears on motor side	kg·m^2
m_W	: Mass of moving object	kg
m_S	: Mass of Screw Shaft	kg
ℓ	: Lead	m
d	: Screw Shaft diameter	m
L	: Ball Screw length	m
γ	: Specific gravity	7,850 kg/m^3
A	: Reduction ratio	
N	: Motor speed	min^{-1}
t	: Acceleration time	sec

●Load Torque T_2

$$T_2 = \frac{P\cdot \ell \cdot A}{2\pi\eta} \times 10^{-3} = \frac{(F+\mu mg)}{2\pi\eta}\cdot \ell \cdot A \times 10^{-3} \quad N\cdot m$$

P	: Axial load	N
F	: Load	N
m	: Mass of moving object	kg
g	: Gravity accelerarations=9.8×10^3 mm/sec^2	
ℓ	: Lead	mm
μ	: Sliding surface friction coefficient	
η	: Efficiency=0.9	
A	: Reduction ratio	



●Preload Dynamic Drag Torque T_3

$$T_3 = 0.05 \times (\tan\beta)^{-0.5} \times \frac{F_{pr}\cdot \ell}{2\pi} \times 10^{-3} \quad N\cdot m$$

β	: Lead angle	deg
F_{pr}	: Preload	N
ℓ	: Lead	mm

●Additional Torque T_4

Described as Torque which occurs in addition to those listed above. For example, support Bearing friction Torque, oil seal resistance Torque, etc.

Rust prevention and Lubrication

●Rust prevention

KSS Ball Screws are applied anti-rust oil when shipping in case of no specific instruction. This oil should be removed before use. Wash Ball Screws with cleaned Kerosine and apply lubricant(Grease or Oil) on Ball Screws. As customer's request, specified Grease or Oil can be applied, but it should be noted that they are not suitable for long term storage purpose and rust might occur.

Note)Anti-rust oil is focused on anti-rust performance and it does not have lubricating function. Therefore, when using Ball Screws with anti-rust oil coating, the problems such as shortened Life, increase of Torque and abnormal heat generation occurs.

●Lubrication

In Ball Screw use, lubricant should be required. If lubricant is not applied with, the problem such as increase of Torque and shortened Life occurs. Applying lubricant can minimize temperature increases, decline of mechanical efficiency due to friction, and deterioration of accuracy caused by wear.
Ball Screw lubrication is divided into Greasing and Oiling. A regular lithium-soap-based Grease and ISO VG32-68 Oil (turbine Oil #1 to #3) are recommended. It is highly important to choose lubricant depending on customer's usage. Especially in case of Miniature Ball Screws, malfunction such as increase of Torque are caused by the stir resistance. KSS original Greases which maintains Ball Screw's smooth movement and have high lubricating performance are prepared. MSG No.1 is appropriate for high smooth requirement and high positioning usage(consistency 1). MSG No.2 is suitable for high speed and general usage(consistency 2).
Please refer to page B101「Original Grease for Miniature Ball Screws」.

Recommended lubricants for normal operating conditions

Lubricant	Type	Product name
Grease	Lithium-based Grease	KSS original Grease MSG No.2
Lubricating Oil	Sliding surface Oil or turbine Oil	Super Multi 68

●Inspection and replenishment

Grease inspection should be performed once every two to three months, and Oil inspection should be performed approximately weekly. Check the Oil or Grease amount and contamination at each inspection and replenish if needed.
When re-greasing, the old or discolored one should be wiped off as much as you can.

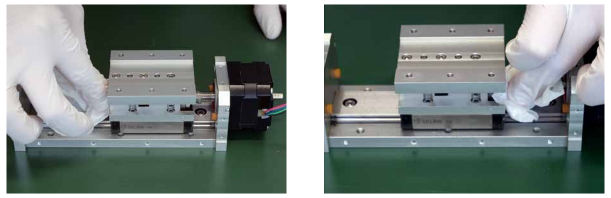
Inspection and replenishment Interval of lubricant

Lubrication	Inspection frequency	Inspection Items	Replenishment and replacement frequency
Automatic intermittent lubrication	Weekly	Oil level, contamination	Replenish at each inspection, depending on tank capacity
Grease	Every 2 to 3 months initially	Contamination, swarf contamination	Replenish annually or as necessary, depending on Inspection results The old or discolored grease should be wiped off before re-greasing.
Oil bath	Daily before operation	Oil suface check	Set a rule for replenishment as necessary, depending on amount of wear.

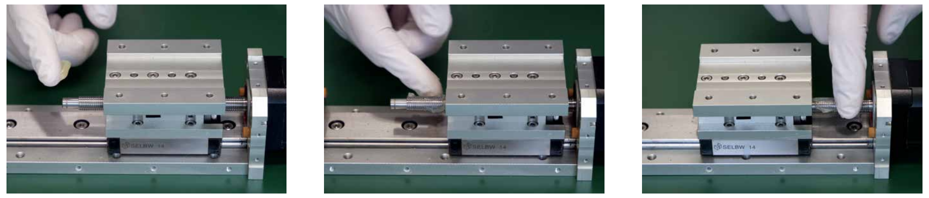
●Grease-up Procedure(Example)

1)It is desirable to wear rubber gloves, not to handle Ball Screw by bear hand.

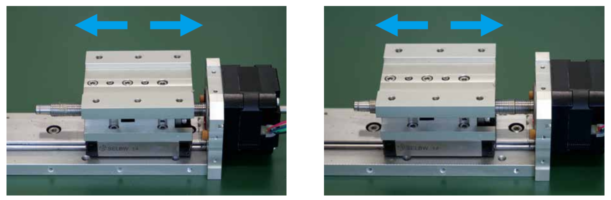
2)Wipe off discolored Grease on the Screw Shaft by using cloth or paper exclusive for wiping Grease or oil (e.g.: Kim Wipes by Kimberly-Clark Corp.).
Move the Ball Nut to wipe off remaining Grease inside the Ball Nut as much as possible.



3)There is no oil hole on the flange for KSS Ball Screws as standard design, apply Grease entirely throughout the Screw Shaft.
Please use the brush exclusive for applying Grease, or apply directly to the Screw Shaft by hand with wearing rubber gloves.If the Ball Nut has an oil hole, utilize it to fill in the new Grease.



4)In order to apply Grease entirely on the Screw Shaft, move the Ball Nut over full travel manually, or install in the device and do running-in.
Remove any remaining Grease on either end of the Screw Shaft.



Please consult KSS for details.

Dust prevention

In Ball Screws, if dust or other contaminations intrude into the Ball Nut, wear is accelerated, the screw groove will be damaged, circulation will be obstructed due to Ball fracture, damage of recirculation parts and so on. Eventually, the Ball Screws will cease to function. Where the possibility of dust or other contaminant exists, the screw thread section cannot be left exposed, and dust prevention measure such as a bellows or Telescopic pipe must be taken.

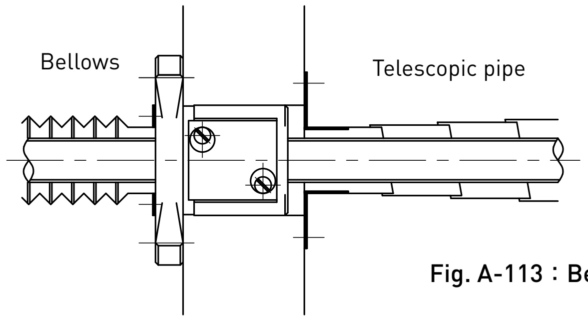


Fig. A-113：Bellows & Telescopic pipe

KSS Ball Screws are concentrated on compact design for a feature of Miniature Ball Screw. Therefore, all models in the catalogue are the dimension without seals. Please inquire KSS if seals are required. Please note that Nut dimension may change due to seal installation. Some models cannot install the seals.

Surface treatment

Surface treatment can be possible for the purpose of rust prevention. Very Low temp. Black Chrome treatment (BCr) is KSS standard surface treatment for the purpose of rust prevention. Please inquire KSS if other surface treatments are needed.

●Feature of KSS Ball Screws with Very Low temp. Black Chrome(BCr) coating

- Due to thin film thickness, mating part can be applicable with BCr.
- Due to strict production management, film thickness can be treated equally and smoothness is kept.
- High anti-rust ability is possible.
- To improve sliding characteristics, BCr＋fluorine resin coating is also available.

Photo A-114：Very Low temp. Black Chrome coating

●Examination data of anti-rust ability

Based on the salt spray corrosion test(JIS Z2371), anti-rust ability has been evaluated, as follows.

- Standard test piece：70mm×150mm×1mm(material＝SPCC)
- Data：Evaluated by appearance and rating number method
  after 24 hours of salt spray corrosion test.(The less number, the more corrosion)

	Rating number(Average)
Sample A(BCr coating)	9.3
Sample B(R coating)	9~8
Sample C(M coating)	3~4

Sample A　Sample B　Sample C

●About RoHS compliance

The amount of hexavalent Chromium in KSS Very Low temp. Black Chrome(BCr) coating is less value than the based on RoHS regulation.

Ball Scresw
Technical description

Traceability

KSS Ball Screws are manufactured from rigidly selected materials in our temperature controlled factory. They are manufactured using the latest production equipment, with consistent quality control supervision ranging from the production process to inspection and shipping.
Certificate of inspection, Photo A-115, or Inspection report, Photo A-116 can be provided as your request.
The Ball Screws produced by KSS have a serial number which is marked on the Nut (refer to the Photo A-117).
Record of inspection and production trail which is in correspondence to a production number, are stored in KSS and inspection data can be retrieved by inquiry of a serial number.

However,some products may not be applicable for serial number, please ask KSS for more detail.

CERTIFICATE OF INSPECTION

This is to Certify that this product has passed the inspection of KSS STANDARD.

KSS CO., LTD.

Head Office: 1-22-14 Yaguchi, Otaku, Tokyo 146-0093
TEL 03(3756)3921
Ojiya Plant: TEL 0258(82)5577

Photo A-115：Certificate of Inspection

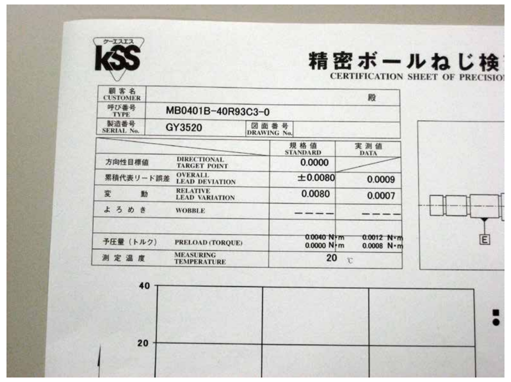


Photo A-116：Inspection report

Photo A-117：Serial Number

Calculation example of characteristic for Ball Screws.

Load direction and Preload will be taking into consideration when calculate the Basic Rating Life by JIS B1192-5, which was established in 2018.
Therefore, KSS uses a calculation formula of Basic Rating Life for Miniature Ball Screws that is conformed to JIS B 1192-5.

Example 1：Vertical Pick&Place

Ball Screw model and operating condition

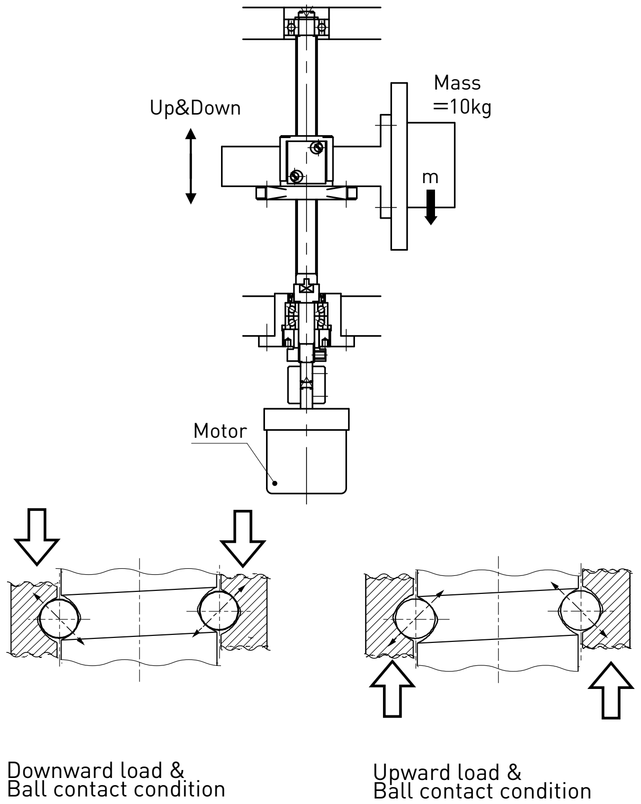
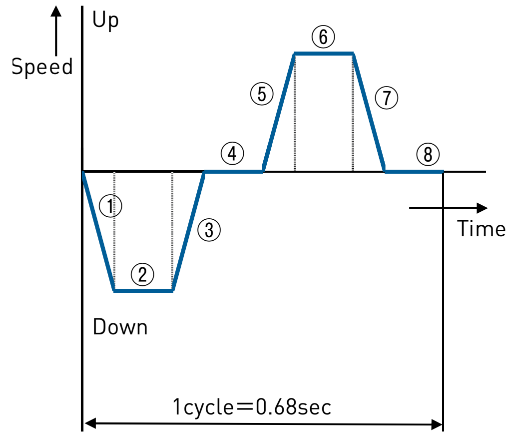


Fig. A-118：Load direction and Ball Contact condition

Operating pattern



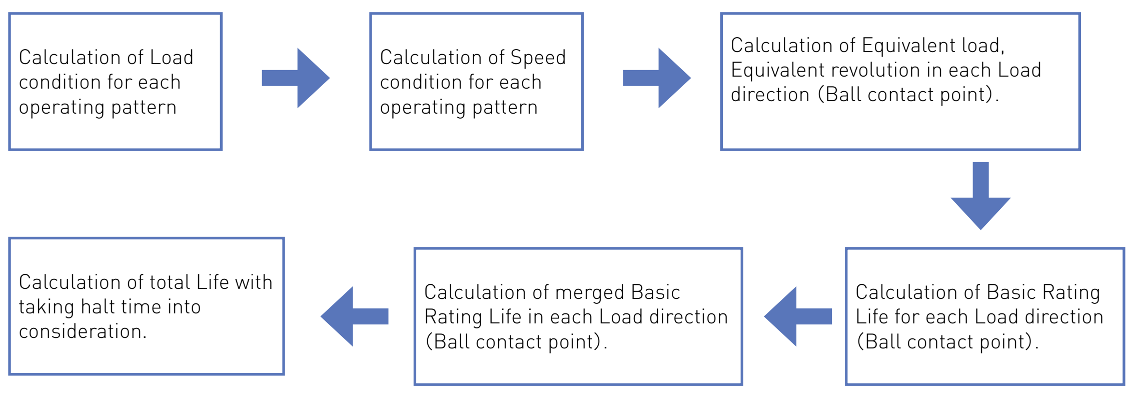
Load direction (Ball contact point) should be considered in calculation of lifetime for Vertical axis application. Load direction is defined as plus for downward, and as minus for upward. The status of Ball contact point is indicated in Fig. A-118.

Ball Screw spec.
- Shaft dia. = ϕ10mm
- Lead = 10mm
- Dynamic Capacity Ca = 3,300N
- Total length = 180mm
- Axial play = 20μm or less

Operating Pattern
- Max Speed = 0.4m/sec
  - ** 2,400 min^{-1} because of Lead 10mm
- Acceleration & Deceleration time = 0.02 sec
  - **①③⑤⑦ in diagram above
- Constant speed time = 0.2 sec
  - **②⑥ in diagram above
- Halt time = 0.1 sec
  - **④⑧ in diagram above
- Cycle time = 0.68sec

Calculation of Basic Rating Life

Basic Rating Life is calculated in the following procedure.



1) Calculation of Load condition from Operating pattern

Load condition of each operating pattern which is numbered is as follows.

①Down & Acceleration
$Fa_1 = mg - m\alpha = 10 \times 9.807 - 10 \times 20 = -101.9(N)$
②Down & Constant speed area
$Fa_2 = mg = 10 \times 9.807 = 98.1(N)$
③Down & Deceleration
$Fa_3 = mg + m\alpha = 10 \times 9.807 + 10 \times 20 = 298.1(N)$
④Halt
$Fa_4 = 0$
⑤Up & Acceleration
$Fa_5 = mg + m\alpha = 10 \times 9.807 + 10 \times 20 = 298.1(N)$
⑥Up & Constant speed area
$Fa_6 = mg = 10 \times 9.807 = 98.1(N)$
⑦Up & Deceleration
$Fa_7 = mg - m\alpha = 10 \times 9.807 - 10 \times 20 = -101.9(N)$
⑧Halt
$Fa_8 = 0$

Here,
m : Mass = 10 kg
g : Gravity Acceleration = 9.807 m/sec²
α : Acceleration
Acceleration up to 0.4m/sec
$\alpha = 0.4/0.02 = 20$ m/sec²

2) Calculation of Speed condition from Operating pattern

Speed condition (Revolution condition) of each operating pattern which is numbered is as follows.

Constant speed area (②、⑥) ;
0.4m/sec = 400 × 60 mm/min = 24,000mm/min = 2,400 min^{-1} (Lead 10mm)
Acceleration and deceleration area (①、③、⑤、⑦) ;
as an average revolution above, 2,400/2 = 1,200 min^{-1}

Calculation result of the load condition and speed condition (revolution) for each operating patterns are as below.

Condition	Axial load Fai(N)	Revolution Ni(min^{-1})	Frequency of use ti(sec)
①Down & Acceleration	−101.9	1,200	0.02
②Down & Constant speed	98.1	2,400	0.2
③Down & Deceleration	298.1	1,200	0.02
④Halt	0	0	0.1
⑤Up & Acceleration	298.1	1,200	0.02
⑥Up & Constant speed	98.1	2,400	0.2
⑦Up & Deceleration	−101.9	1,200	0.02
⑧Halt	0	0	0.1

plus(+) indicates downward load and minus(−) indicates upward load.

3) Calculation of Equivalent load, Equivalent revolution for in each Load direction (Ball contact point)

As we could calculate the applying load and direction in each operating pattern, now we calculate the Equivalent load and Equivalent revolution for each Load direction.
Calculation formula shown in page A825 will be used for calculating Equivalent load and Equivalent revolution.

$$Fam = \left(\frac{Fa_1^3 \cdot N_1 \cdot t_1 + Fa_2^3 \cdot N_2 \cdot t_2 + Fa_3^3 \cdot N_3 \cdot t_3 + ... + Fa_i^3 \cdot N_i \cdot t_i}{N_1 \cdot t_1 + N_2 \cdot t_2 + N_3 \cdot t_3 + ... + N_i \cdot t_i} \right)^{1/3} \ N$$

$$N_m = \frac{N_1 \cdot t_1 + N_2 \cdot t_2 + N_3 \cdot t_3 + ... + N_i \cdot t_i}{t_1 + t_2 + t_3 + ... + t_i} \ min^{-1}$$

Now calculation table should be re-arranged as below by load direction, and Equivalent load and Equivalent revolution in each load direction are as follows.

Condition	Downward load		Upward load		Frequency of use ti(sec)
	Axial load Fai(N)	Revolution Ni(min^{-1})	Axial load Fai(N)	Revolution Ni(min^{-1})	
①Down & Acceleration	−	−	101.9	1,200	0.02
②Down & Constant speed	98.1	2,400	−	−	0.2
③Down & Deceleration	298.1	1,200	−	−	0.02
④Halt	−	−	−	−	0.1
⑤Up & Acceleration	298.1	1,200	−	−	0.02
⑥Up & Constant speed	98.1	2,400	−	−	0.2
⑦Up & Deceleration	−	−	101.9	1,200	0.02
⑧Halt	−	−	−	−	0.1
Equivalence	Fam(d) =129.3	Nm(d) =2,290.9	Fam(u) =101.9	Nm(u) =1,200	Working duration : 0.48 sec Halt time : 0.2 sec 1 cycle : 0.68 sec

4) Calculation of Basic Rating Life for each Load direction (Ball contact point)

Then calculate the Basic Rating Life for downward load, upward load by using the value of Equivalent load, Equivalent revolution in each load direction (Ball contact point).

【Downward load】

Substitute the Equivalment Load Fam(d) and Revolution Nm(d) in the following formula in page A825.

$$L_{10h(d)} = \left(\frac{Ca}{f \cdot Fam(d)}\right)^3 \times \left(\frac{10^6}{60 \cdot Nm(d)}\right) = 69{,}991 \text{ hours}$$

Here, Basic Dynamic Load Rating Ca = 3,300N, Load factor f = 1.2.

【Upward load】

Calculate the upward load as same method as above.

$$L_{10h(u)} = \left(\frac{Ca}{f \cdot Fam(u)}\right)^3 \times \left(\frac{10^6}{60 \cdot Nm(u)}\right) = 272{,}988 \text{ hours}$$

5) Calculation of merged Basic Rating Life in each Load direction (Ball contact point)

Calculate the merged Basic Rating Life by combining the Basic Rating Life of each Load direction ($L_{10h(d)}$, $L_{10h(u)}$), with the calculation formula of page A826.

$$L'_{10h} = \left(L_{10h(d)}{}^{-10/9} + L_{10h(u)}{}^{-10/9}\right)^{-9/10} = 58{,}504 \text{ hours}$$

5) Calculation of total Life with taking halt time into consideration

Above calculation is only for the working duration, therefore calculate the total Life with taking halt time in each cycle into consideration.

$$L''_{10h} = L'_{10h} \times (\text{cycle time}) / (\text{working duration}) = 58{,}504 \times (0.68 / 0.48)$$

$$= 82{,}881 \text{ hours}$$

Calculation of Driving Torque for Linear Motion system

Calculate Driving Torque for Linear Motion system according to page A829. It is important for motor selection. In the above case, due to backlash type Ball Screw, Preload Dynamic Drag Torque does not occur. Therefore, calculate acceleration Torque T_1 and Load Torque T_2.

$$T = T_1 + T_2 + T_3 + T_4 \quad N\cdot m$$

- T_1 : Acceleration Torque N·m
- T_2 : Load Torque N·m
- T_3 : Preload Dynamic Drag Torque N·m
- T_4 : Additional Torque N·m

1) Calculation of acceleration Torque T_1

$$T_1 = \alpha \cdot I = \alpha\,(Iw + Is) \quad N\cdot m$$

- α : Angular acceleration rad/sec²
- I : Inertia moment kg·m²
- Iw : Inertia moment of moving object by motor axis conversion kg·m²
- Is : Inertia moment of Screw Shaft kg·m²

$$Iw = m_W \times (\ell/2\pi)^2 = 2.53 \times 10^{-5} \quad kg\cdot m^2$$

- m_W : Mass of moving object = 10 kg
- ℓ : Ball Screw Lead = 0.01 m

$$Is = m_S \times (d^2/8) = (d/2)^2 \pi \gamma \times L \times (d^2/8) = 0.139 \times 10^{-5} \quad kg\cdot m^2$$

- m_S : Mass of Screw Shaft kg
- γ : Specific gravity of Screw Shaft = 7,850 kg/m³
- d : Shaft dia. = 0.01 m
- L : Shaft length = 0.18 m

$$\alpha = (2\pi N)/60t = 12{,}566.4 \quad rad/sec^2$$

- N : Max speed = 2,400 min⁻¹
- t : Acceleration time = 0.02 sec

$$T_1 = 12{,}566.4 \times (2.53 + 0.139) \times 10^{-5} = 0.335 \; N\cdot m$$

2) Calculation of Load Torque T_2

$$T_2 = mg\ell/(2\pi\eta) = 0.173 \; N\cdot m$$

- m : Mass of moving object = 10 kg
- g : Gravity acceleration = 9.807 m/sec²
- ℓ : Ball Screw Lead = 0.01 m
- η : Ball Screw efficiency = 0.9

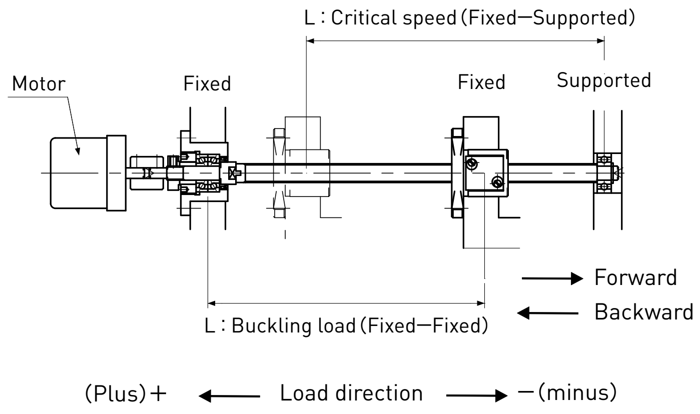
3) Calculation of Driving Torque T for Linear Motion system

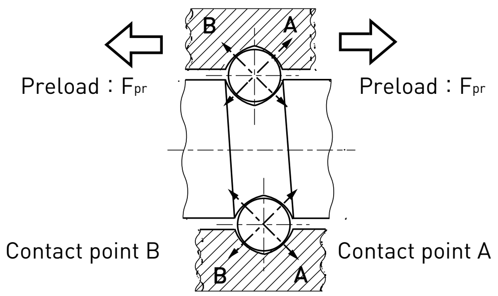
In case without consideration of Torque by support Bearings, Driving Torque of Ball Screw is as follows.

$$T = T_1 + T_2 = 0.335\; N\cdot m + 0.173\; N\cdot m = 0.508\; N\cdot m$$

Example 2 : Horizontal desk top small lathe

Ball Screw model and operating condition





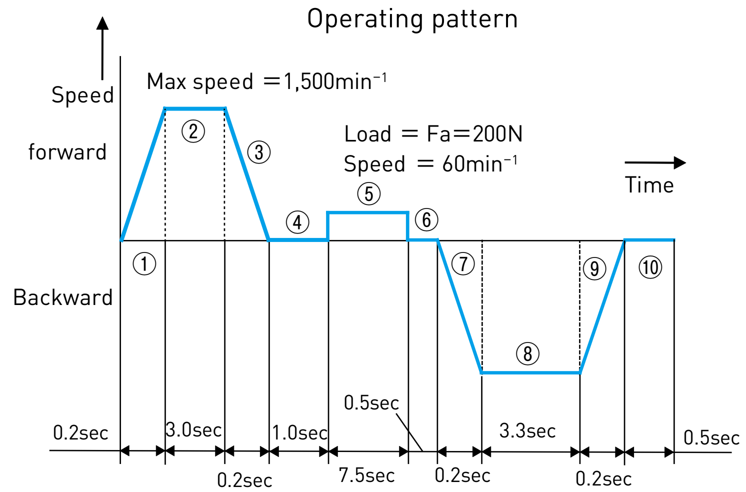


Fig A-119 : Operating condition and Ball Contact point

Ball Screw spec.
- Shaft dia. = ϕ12mm
- Lead = 2mm
- Shaft Root dia. d = ϕ10.6mm
- Dynamic Capacity Ca =1,900N
- Mounting span L = 400mm
- Axial play = 0μm or less
- Mass of moving object m = 10kg
- Sliding surface friction coefficient μ = 0.05
- Preload F_{pr} = 95N (Ca×5%)

Operating Pattern
- Max Speed = 50mm/sec
  - ** 1,500 min^{-1} because of Lead 2mm
- Operating pattern : see diagram above
  - ①⑦Acceleration = 0.2sec
  - ②Constant speed (forward) = 3.0sec
  - ③⑨Deceleration = 0.2sec
  - ④⑥⑩halt = 2.0sec (total)
  - ⑤Turning time = 7.5sec
  - ⑧Constant speed (backward) = 3.3sec
- Load Fa = 200N
- Cutting speed = 2mm/sec
- **60min^{-1} due to 2 mm lead

Calculation of permissible Axial load

1) Study of Buckling load

Calculate Buckling load according to the following formula in page A815.

$$P = \alpha \times \frac{n\pi^2 E \cdot I}{L^2} \quad N \qquad\qquad I = \frac{\pi}{64} d^4 \quad mm^4$$

Substitute safety factor α = 0.5, Young's modulus E = 2.08 × 10^5 N/mm^2 (MPa), Root diameter d = 10.6mm, Fixed — Fixed mounting factor n = 4, mounting span L = 400mm in formula above.

$$P = 15{,}900N$$

It is more than maximum Load so that there is no problem.

2) Study of permissible Load for yield stress

Calculate permissible Load for yield stress based on the formula in page A815.

$$P = \sigma \times A \quad N \qquad\qquad A = \frac{\pi}{4} d^2 \quad mm^2$$

Substitute permissible stress σ=98N/mm^2 (MPa), Root diameter d=10.6mm in the formula above.

$$P = 8{,}650 \quad N$$

It is more than maximum Load and there is no problem.

Calculation of permissible Revolution

Calculate permissible Revolution based on the formula in page A816

$$N = \beta \times \frac{60 \cdot \lambda^2}{2\pi} \times \sqrt{\frac{E \cdot I \cdot g}{\gamma \cdot A \cdot L^4}} \quad min^{-1}$$

$$I = \frac{\pi}{64} d^4 \quad mm^4 \qquad\qquad A = \frac{\pi}{4} d^2 \quad mm^2$$

Substitute safety factor β=0.8, Young's modulus E = 2.08×10^5 N/mm^2 (MPa), gravity acceleration g = 9.8×10^3 mm/sec^2, material specific gravity γ = 7.7×10^{-5} N/mm^3, Root diameter d = 10.6mm, Fixed — Support mounting factor λ = 3.927, mounting span L = 400mm in formula above.

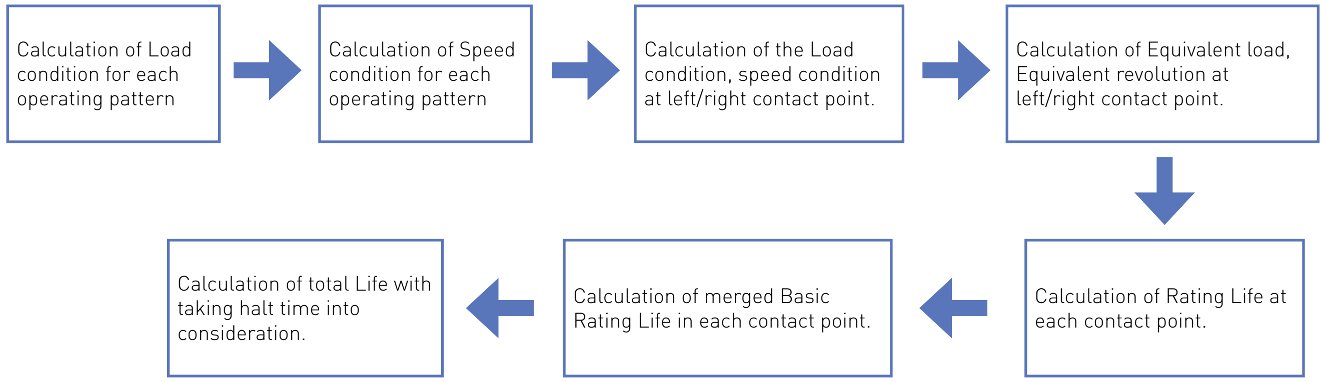
$$N = 10{,}000 \quad min^{-1}$$

Therefore, it is more than maximum Revolution and there is no problem.

Calculation of Basic Rating Life

Load direction and Preload will be taken into consideration when calculate the Basic Rating Life by JIS B1192-5, which was established in 2018. Therefore, KSS uses a calculation formula of Basic Rating Life for Miniature Ball Screws that is conformed to JIS B 1192-5.

In case when preload is effective by oversized Ball, the contact condition of the Ball is 4 points as per Fig. A-111. As explained in page A827, total Life can be calculated after calculation of Rating Life at contact point A and B due to the change of initial contact condition under the preload casused by external load.



1) Calculation of Load condition from Operating pattern

Load condition of each operating pattern which is numbered is as follows.

①Forward Acceleration
$Fa_1 = \mu mg + m\alpha = 0.05 \times 10 \times 9.807 + 10 \times 0.25 = 7.4(N)$
②Forward at constant speed area
$Fa_2 = \mu mg = 0.05 \times 10 \times 9.807 = 4.9(N)$
③Forward Deceleration
$Fa_3 = \mu mg - m\alpha = 0.05 \times 10 \times 9.807 - 10 \times 0.25 = 2.4(N)$
④Halt
$Fa_4 = 0$
⑤at Turning
$Fa_5 = \mu mg + Fa = 0.05 \times 10 \times 9.807 + 200 = 204.9(N)$
⑥Halt
$Fa_6 = 0$
⑦Backward Acceleration
$Fa_7 = -(\mu mg + m\alpha) = -(0.05 \times 10 \times 9.807 + 10 \times 0.25) = -7.4(N)$
⑧Backward at constant speed area
$Fa_8 = -\mu mg = -0.05 \times 10 \times 9.807 = -4.9(N)$
⑨Backward Deceleration
$Fa_9 = -\mu mg + m\alpha = -0.05 \times 10 \times 9.807 + 10 \times 0.25 = -2.4(N)$
⑩Halt
$Fa_{10} = 0$

Here,
m : Mass = 10 kg
g : Gravity Acceleration = 9.807 m/sec²
α : Acceleration
Acceleration which reaches up to 50mm/sec
$\alpha = 0.05/0.2 = 0.25$ m/sec²

2) Calculation of Speed condition from Operating pattern

Speed condition (Revolution condition) of each operating pattern which is numbered as follows.

Constant speed area(②、⑧) ;
50mm/sec=50×60 mm/min = 3,000mm/min = 1,500 min^{-1}(Lead 2mm)

Acceleration and deceleration area(①、③、⑦、⑨) ;
As above average revolution, 1,500/2= 750 min^{-1}

Calculation result of the load condition and speed condition (revolution) for each operating patterns are as below.

Condition	Axial load Fai(N)	Revolution Ni(min^{-1})	Frequency of use ti(sec)
①Forward Acceleration	7.4	750	0.2
②Forward at Constant speed	4.9	1,500	3.0
③Forward Deceleration	2.4	750	0.2
④Halt	0	0	1.0
⑤Turning	204.9	60	7.5
⑥Halt	0	0	0.5
⑦Backward Acceleration	−7.4	750	0.2
⑧Backward at constant speed	−4.9	1,500	3.3
⑨Backward Deceleration	−2.4	750	0.2
⑩Halt	0	0	0.5

3) Calculation of the Load condition at left/right contact point

Ball contact condition in 4 point between Balls and thread grooves by preload may changes by external load as shown in page 827(Fig. A-112). Based on the changed Elastic displacement, load applying on the contact point A and B will be calculated by formula below.

【If the direction of the external load is plus(+)】

$$Fai_{(A)} = F_{pr} \times \left(1 + \frac{Fai}{2^{3/2} \times F_{pr}}\right)^{3/2} \qquad Fai_{(B)} = Fai_{(A)} - Fai$$

【If the direction of the external load is minus(−)】

$$Fai_{(B)} = F_{pr} \times \left(1 + \frac{|Fai|}{2^{3/2} \times F_{pr}}\right)^{3/2} \qquad Fai_{(A)} = Fai_{(B)} - |Fai|$$

Here,
F_{pr} : Preloaded load = 95 N
Fai : Axial load in each condition(N)
(A),(B) : This means contact point

The calculation result of each load condition and revolution condition as per contact point A and B is shown in table A-120.

4) Calculation of Equivalent load, Equivalent revolution at left and right contact point

Load applying on contact point A and B is calculated under each operating condition, then Equivalent load and Equivalent revolution at each contact point will be calculated. However, the speed and frequency of use stay the same, only the load condition will be different.

Calculation formula shown in page A825 will be used for calculating Equivalent load and Equivalent revolution.

$$Fam = \left(\frac{Fa_1^3 \cdot N_1 \cdot t_1 + Fa_2^3 \cdot N_2 \cdot t_2 + Fa_3^3 \cdot N_3 \cdot t_3 + \ldots\ldots + Fa_i^3 \cdot N_i \cdot t_i}{N_1 \cdot t_1 + N_2 \cdot t_2 + N_3 \cdot t_3 + \ldots\ldots + N_i \cdot t_i} \right)^{1/3} \ \mathrm{N}$$

$$N_m = \frac{N_1 \cdot t_1 + N_2 \cdot t_2 + N_3 \cdot t_3 + \ldots\ldots + N_i \cdot t_i}{t_1 + t_2 + t_3 + \ldots\ldots + t_i} \quad \mathrm{min}^{-1}$$

The axial load applying on contact point A and B for each condition, Equivalent load and Equivalent revolution are as follows.

Table A-120 : Load & Revolution condition at each contact point

Condition	Axial load Fai(N)	Axial load at contact pt. A $Fai_{(A)}$(N)	Axial load at contact pt. B $Fai_{(B)}$(N)	Revolution Ni(min^{-1})	Frequency of use ti(sec)
① Forward Acceleration	7.4	99.0	91.6	750	0.2
② Forward at Constant speed	4.9	97.6	92.7	1,500	3.0
③ Forward Deceleration	2.4	96.3	93.9	750	0.2
④ Halt	0	—	—	0	1.0
⑤ Turning	204.9	222.3	17.4	60	7.5
⑥ Halt	0	—	—	0	0.5
⑦ Backward Acceleration	−7.4	91.6	99.0	750	0.2
⑧ Backward at constant speed	−4.9	92.7	97.6	1,500	3.3
⑨ Backward Deceleration	−2.4	93.9	96.3	750	0.2
⑩ Halt	0	—	—	0	0.5
Equivalence		$Fam_{(A)}$=109.0	$Fam_{(B)}$=94.0	Nm=719.2	Working duration : 14.6 sec Halt time : 2.0 sec 1 cycle : 16.6 sec

Note) Results of applying load at contact point A and B are all absolute number.

5) Calculation of Rating Life at each contact point

Calculate the Basic Rating Life at contact point A and B by using the value of Equivalent load, Equivalent revolution in each contact point A, B.

【Contact point A】

Substitute the Equivalent load $Fam_{(A)}$ and Equivalent revolution N_m in the following formula as shown in page A825.

$$L_{10h(A)} = \left(\frac{Ca}{f \cdot Fam_{(A)}} \right)^3 \times \left(\frac{10^6}{60 \cdot N_m} \right) = 71{,}029 \text{ hours}$$

【Contact point B】

Substitute the Equivalent load $Fam_{(B)}$ and Equivalent revolution N_m in the following formula as shown in page A825.

$$L_{10h(B)} = \left(\frac{Ca}{f \cdot Fam_{(B)}} \right)^3 \times \left(\frac{10^6}{60 \cdot N_m} \right) = 110{,}747 \text{ hours}$$

Here, Basic Dynamic Load Rating Ca = 1,900N, Load factor f = 1.2.

6) Calculation of merged Basic Rating Life in each contact point

Calculate merged Basic Rating Life of contact point A, B ($L_{10h(A)}$, $L_{10h(B)}$) by using formula in page A 826.

$$L'_{10h} = \left(L_{10h(A)}^{-10/9} + L_{10h(B)}^{-10/9} \right)^{-9/10} = 46{,}257 \text{ hours}$$

7) Calculation of total Life with taking halt time into consideration

Above calculation is only for the working duration, therefore calculate the total Life with taking halt time into consideration.

$$L''_{10h} = L'_{10h} \times (\text{cycle time}) / (\text{working duration}) = 46{,}257 \times (16.6 / 14.6)$$
$$= 52{,}594 \text{ hours}$$