1 Introduction

Ballscrews, also called a ball bearing screws, recirculating ballscrews, etc., consist of a screw spindle and a nut integrated with balls and the balls' return mechanism, return tubes or return caps. Ballscrews are the most common type of screws used in industrial machinery and precision machines. The primary function of a ballscrew is to convert rotary motion to linear motion or torque to thrust, and vice versa, with the features of high accuracy, reversibility and efficiency. HIWIN provides a wide range of ballscrews to satisfy your special requirements.

The combination of state-of-the-art machining technology, manufacturing experiences, and engineering expertise makes HIWIN ballscrew users "High-Tech Winners". HIWIN uses precise procedures to create exact groove profiles, either by grinding or precision rolling. Accurate heat treatment is also used to ensure the hardness of our ballscrews. These result in maximum load capacity and service life.

HIWIN precision ballscrews provide the most smooth and accurate movement, together with low drive torque, high stiffness and quiet motion with predictable lengthened service life. HIWIN rolled ballscrews also provide smooth movement and long life for general applications with less precision in lower price. HIWIN has modern facilities, highly skilled engineers, quality manufacturing and assembly processes, and uses quality materials to meet your special requirements.

It is our pleasure to provide you with the technical information and selection procedure to choose the right ballscrews for your applications through this catalogue.

2 Technological Features of HIWIN Ballscrews

2.1 Characteristics of HIWIN Ballscrews

There are many benefits in using HIWIN ballscrews, such as high efficiency and reversibility, backlash elimination, high stiffness, high lead accuracy, and many other advantages. Compared with the contact thread lead screws as shown in (Fig. 2.1), a ballscrew add balls between the nut and spindle. The sliding friction of the conventional screws is thus replaced by the rolling motion of the balls. The basic characteristics and resultant benefits of HIWIN ballscrews are listed in more details as follows:

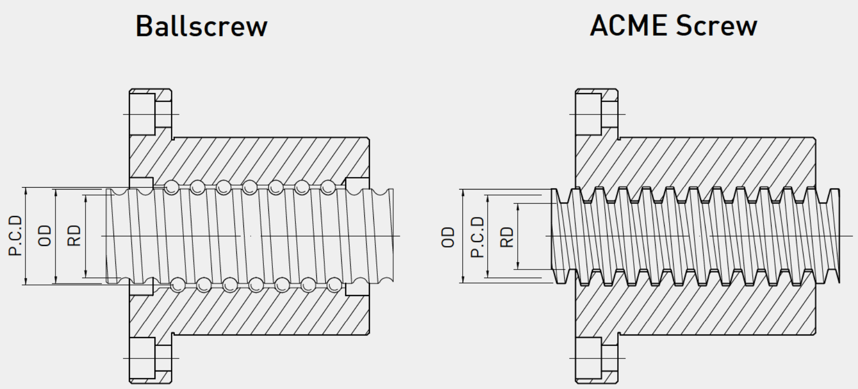


Fig 2.1 Basic configuration of ballscrews and contact thread lead screws

(1) High efficiency and reversibility

Ballscrews can reach an efficiency as high as 90% because of the rolling contact between the screw and the nut. Therefore, the torque requirement is approximately one third of that of conventional screws. It can be seen from Fig. 2.2 that the mechanical efficiency of ball screws are much higher than conventional lead screws.

HIWIN ballscrews have super surface finish in the ball tracks which reduce the contact friction between the balls and the ball tracks. Through even contact and the rolling motion of the balls in the ball tracks, a low friction force is achieved and the efficiency of the ballscrew is increased. High efficiency renders low drive torque during ballscrew motion. Hence, less drive motor power is needed in operation resulting in lower operation cost.

HIWIN uses a series of test equipment and testing procedures to guarantee the efficiency.

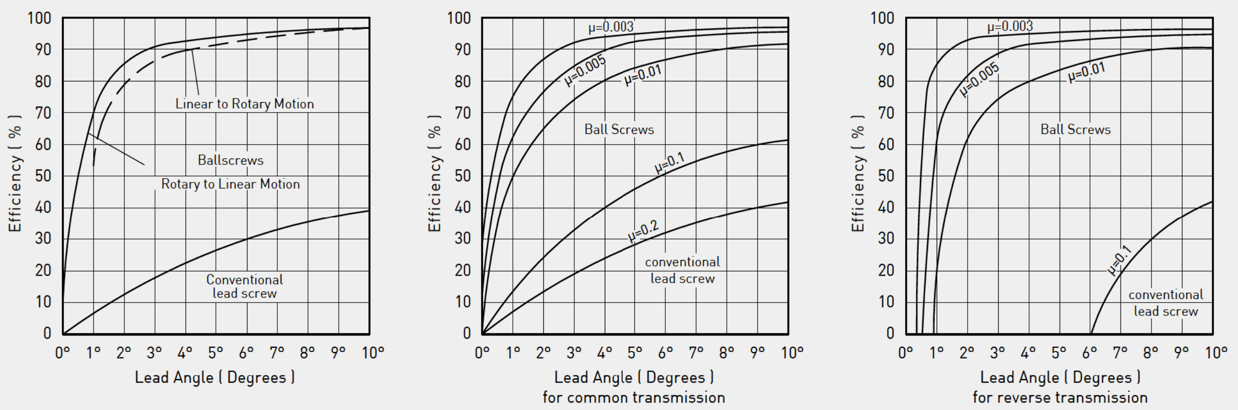


Fig 2.2 Mechanical efficiency of ballscrews

(2) Backlash elimination and high stiffness

Computer Numerically Controlled (CNC) machine tools require ballscrews with zero axial backlash and minimal elastic deformation (high stiffness). Backlash is eliminated by our special designed Gothic arch form balltrack (Fig. 2.3) and preload.

In order to achieve high overall stiffness and repeatable positioning in CNC machines, preloading of the ballscrews is commonly used. However, excessive preload increases friction torque in operation. This induced friction torque will generate heat and reduce the life expectancy. With our special design and fabrication process, we provide optimized ballscrews with no backlash and less heat losses for your application.

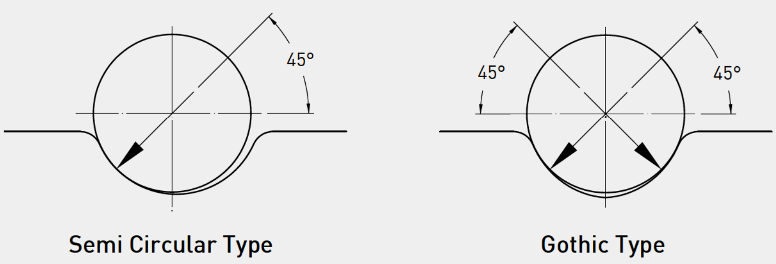


Fig2.3 Typical contact types for ballscrews

(3) High lead accuracy

For applications where high accuracy is required, HIWIN modern facilities permit the achievement of ISO, JIS and DIN standards or specific customer requirements.

This accuracy is guaranteed by our precise laser measurement equipment and reported to each customer.

(4) Predictable life expectancy

Unlike the useful life of conventional screws which is governed by the wear on the contact surfaces, HIWIN's ballscrews can usually be used till the metal fatigue. By careful attention to design, quality of materials, heat treatment and manufacture, HIWIN's ballscrews have proved to be reliable and trouble free during the period of expected service life. The life achieved by any ballscrew depends upon several factors including design, quality, maintenance, and the major factor, dynamic axial load (C).

Profile accuracy, material characteristics and the surface hardness are the basic factors which influence the dynamic axial load.

It is recommended that the life at average axial load should be a minimum of $1x10^6$ revs). High quality ballscrews are designed to conform with the B rating (i.e. 90% probability of achieving the design life). Fifty percent of the ballscrews can exceed 2 to 4 times of the design life.

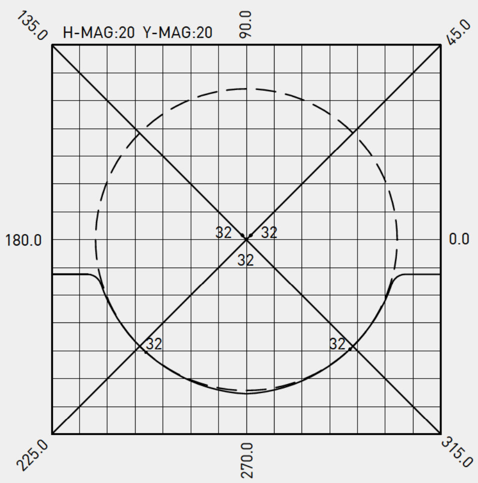
(5) Low starting torque and smooth running

Due to metal to metal contact, conventional contact thread lead screws require high starting force to overcome the starting friction. However, due to rolling ball contact, ballscrews need only a small starting force to overcome their starting friction.

HIWIN uses a special design factor in the balltrack (conformance factor) and manufacturing technique to achieve a true balltrack. This guarantees the required motor torque to stay in the specified torque range.

HIWIN has special balltrack profile tracing equipment to check each balltrack profile during the manufacturing process. A sample trace is shown in Fig. 2.4.

HIWIN also uses computer measurement equipment to accurately measure the friction torque of ballscrews. A typical distance-torque diagram is shown in Fig. 2.5.



Work name : S.H
Model No. : 001H-2-3
Lot No. : 201536
Operator : L.J.F.
Comment :

Measure node: X pitch
Pick up radius: 0.0256mm
Horizontal mag: 20.0000
Vertical mag: 20.0000
Measure length: 7.0000 mm
Measure pitch: 0.0030 mm

No. code symbol actual
32 292 X: 0.1816 mm Z: 0.1980 mm RC : 3.4438 mm
32 292 X: -0.1911 mm Z: 0.2022 mm RC : 3.4532 mm
32 292 X: -2.1464 mm Z: -2.3399 mm A : -42.5259 mm
32 292 X: 2.1799 mm Z: -2.3084 mm A : 43.3615 mm
32 292 X: -0.0000 mm Z: -0.0000 mm RC : 3.1750 mm
*Original point set

Fig 2.4 Balltrack checking by HIWIN profile tracer

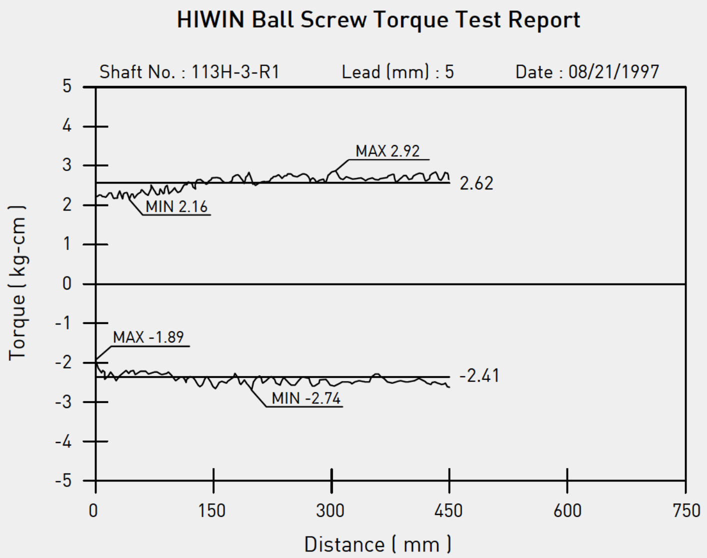


Fig 2.5 HIWIN preload checking diagram

(6) Quietness

High quality machine tools require low noise during fast feeding and heavy load conditions.

HIWIN achieves this by virtue of its return system, balltrack designs, assembly technique, and careful control of surface finish and dimensions.

(7) Short lead time

HIWIN has a fast production line and can stock ballscrews to meet short lead times.

(8) Advantages over hydraulic and pneumatic actuators

The ballscrew used in an actuator to replace the traditional hydraulic or pneumatic actuator has many advantages, i.e. fast response, no leakage, no filtering, energy savings and good repeatability.

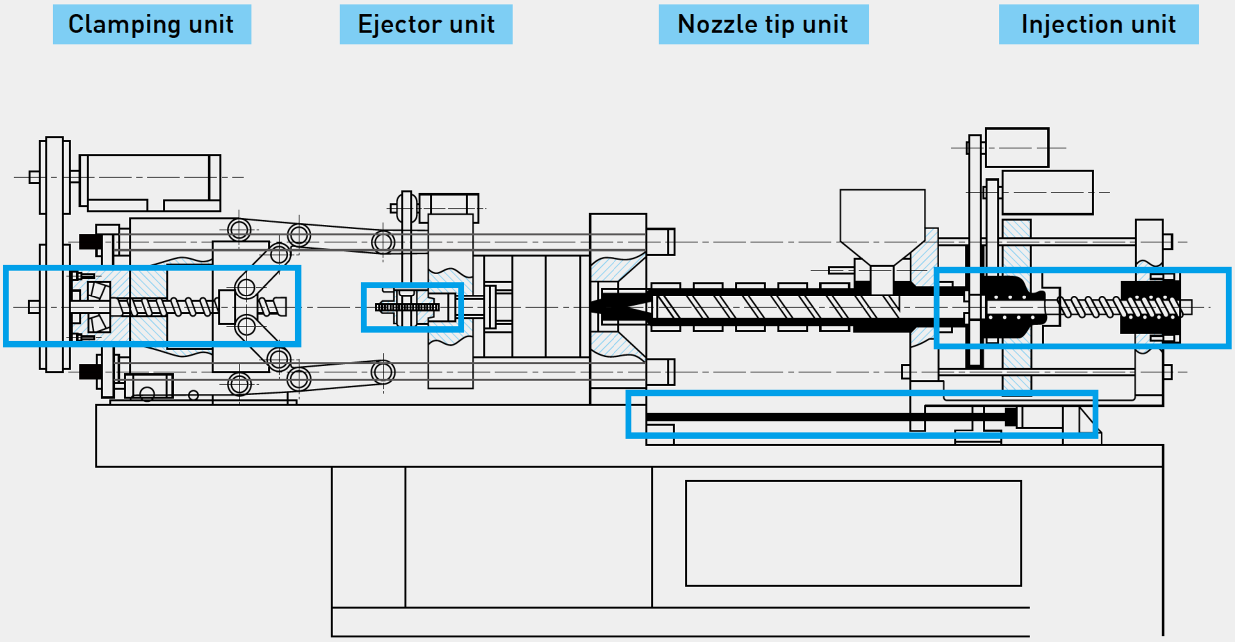


Fig 2.6 All-electric injection molding machine

2.2 Applications for Ballscrews

HIWIN ballscrews are used in the following fields and the recommended application grade can be found in Table 4.5.

1. **CNC machinery :** CNC machine center, CNC lathe, CNC milling machine, CNC EDM, CNC grinder, wire cutting machine, boring machine, etc.
2. **Precision machine tools :** Milling machine, grinder, EDM, tool grinder, gear manufacturing machine, drilling machine, planer, etc.
3. **Industrial machinery :** Printing machine, paper-processing machine, automatic machine, textile machine, drawing machine, special purpose machine,injection molding machine, etc.
4. **Electronic machinery :** Robot measuring instrument, X-Y table, medical equipment, surface mounting device, semi-conductor equipment, factory automation equipment, etc.
5. **Transport machinery :** Material handling equipment, elevated actuator, etc.
6. **Aerospace industry :** Aircraft flaps, thrust open-close reverser, airport loading equipment, fin actuator, etc.
7. **Miscellaneous :** Antenna leg actuator, valve operator, etc.

3 Classification of Standard Ballscrews

3.1 Standard Ballscrew Spindle

HIWIN recommends our standard regular ballscrews for your design. However, high lead, miniature or other special types of ballscrews, may also be available upon your request. Table 3.1 shows the standard ballscrew spindles which are available.

3.2 Nut Configuration

The circuiting systems of nut of HIWIN ball screw can be divided into: external circuit, internal circuit, end caps, and Super S. For each circuiting way the features are as follows: external recirculation type, internal recirculation type, endcap recirculation type,and Super S. The features of these types are specified below.

3.2.1 Type of return tube design

(1) External recirculation type

a. structure

The first, called the external recirculation type ballscrew, consists of the screw shaft,the ball nut, the steel balls, the return tubes and the fixing plate. The steel balls are introduced into the space between the screw shaft and the ball nut. The balls are diverted from the ball tracks and carried back by the ball guide return tube form a loop. Since the return tubes are located outside the nut body, this type is called the external recirculation type ball screw Fig. 3.1.

b. features

(a) Adapted to wide kinds of shaft diameters and leads of ball screw

(b) Complete specifications

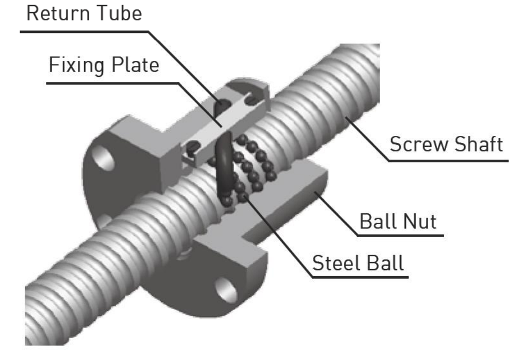


Fig. 3.1 External recirculation type nut with return tubes

(2) Internal recirculation type

a. structure

The second design, called the internal recirculation type ballscrew, consists of the screw spindle, the ball nut, the steel balls and the ball return caps. The steel balls make only one revolution around the screw spindle. The circuit is closed by a ball return cap in the nut allowing the balls to cross over adjacent ball tracks. Since the ball return caps are located inside the nut body, this is called the internal recirculation type ballscrew Fig. 3.2.

b. features

(a) Adapted to normal leads

(b) Outer diameter of nut is small

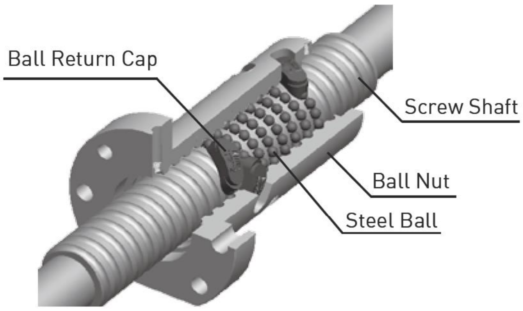


Fig. 3.2 Internal recirculation type nut with return caps

(3) Endcap recirculation type

a. structure

The third design is called endcap recirculation type ball screw Fig. 3.3.The basic design of this return system is the same as the external recirculation type nut Fig. 3.5 except that the return tube is made inside the nut body as a through hole. The balls in this design traverse the whole circuit of the ball tracks within the nut length. Therefore, a short nut with the same load capacity as the conventional design can be used.

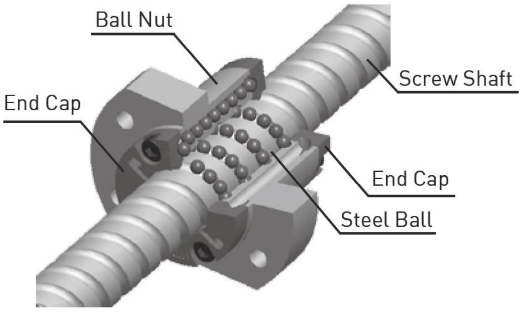


Fig. 3.3 Endcap recirculation type nut with return system

b. features

(a) Adapted to high lead

(b) Outer diameter of nut is middle

(c) Single nut only

(4) Super S

a. structure

The forth design is called Super S recirculation type ballscrew which consists of screw shaft, the ball nut, the steel balls and the cassette (Fig.3.4). The basic design of this return system is the same as the endcap recirculation type. Instead of using endcap, cassette is used in the recirculation. The balls in this design traverse the whole circuit of the ball tracks by passing through the cassette within the nut length.

b. features

(a) Quietness

(b) Compact and lightweight

(c) High acceleration and deceleration

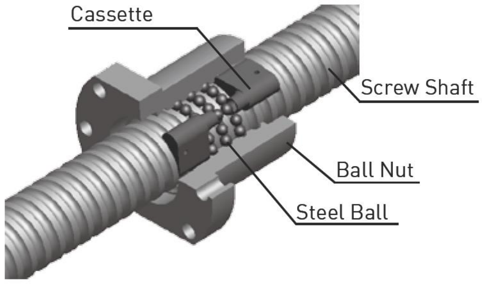


Fig. 3.4 Cassette recirculation type nut with return system

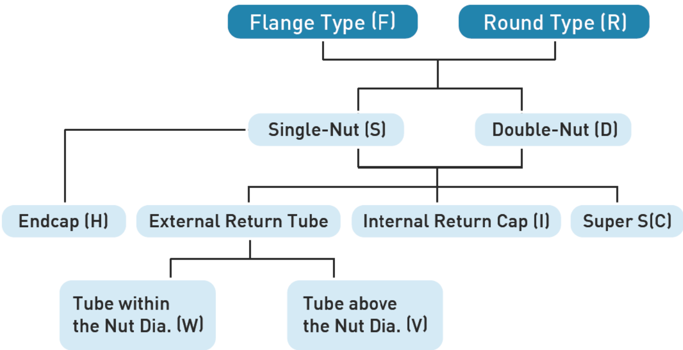
Table 3.1: The comparison chart of ball screw diameter vs lead and recirculation

dia. \ lead	1	1.5	2	2.5	3	3.175	3.5	4	4.23	5	5.08	6	6.35	8	10	12	12.7	15	16	20	24	25	25.4	28	30	32	35	36	40	50	60	64	80
6	I	I	I									H																					
8	I	I	I	I	I			I						H																			
10	I	I	I	T, I	I			T, I		T, I		T		T	T, H S																		
12	I	I	I	I	T, I			T, I		T, I S		T			T, S	T, H				T, H													
14	I		I	I	I			T, I		T, I	T, I	T		T	T, H S																		
15			I				I	I		T, I					T, S			T, H		T, H S					H								
16	I	I	I	T, I	T, I			T, I		T, I	T, I	T, I		T, I	T, H S		T		T, H S							H							
18					I			I		I																							
20			I	T, I	T, I			T, I		T, I S	T, I	T, I S	T, I	T, I S	T, I S	T	T	T	T	T, H S									H, S				
22										T	T			T	T																		
25			I	T, I				T, I		T, I S	T, I	T, I S	T, I	T, I S	T, I S	T, I S	T	I, S	T, S	T, H S		T, H S	T							H	H		H
28								T, I	T, I	T, I	T, I	T, I S	T	T, I S	T, I S	T, I S		T	T, S														
30							T			T, I				I	I				T	S													
32			I	T, I	I	T, I	I	T, I		T, I S	T, I S	T, I S	T, I	T, I S	T, I S	T, I S	T, I	T, I S	T, I S	T, I S, H		T, S	T			T, S H			S, H	H		H	
36								T		T, I		T, I S	T, I	T, I S	T, I S	T, I S	T		T, S	T, S H	T, S	T						T, S H					
38										I	T, I		T	S	I, S	T, S		S	S	S	T, I	T, S	T						S, H				
40			I	I				I		T, I S	T, I	T, I S	T, I	T, I S	T, I S	T, I S	T, I	T, I	T, I S	T, I S, H		T, I S			S	T			T, S H	T			S
45								I		T, I	T	T, I	I	T, I S	T, I S	T, I S	T		T, I S	T, I S		T, S							S				
50								T		T, I S	T, I	T, I	T, I	T, I S	T, I S	T, I S	I	I, S	T, I S	T, I S		T, S	S		T, I S	T	S		T, S H	T, S H			
55												T, I S	I	T, I	T, I	T, I S	T, I		T, S	T, S					T					H			
63										I	I	I	T, I	T, I	T, I S	T, I S	T, I	I	T, I S, H	T, I S	I	T, S			T, I S	T, S			T, S H	T, S			
70														T	T, I	T, I		T	T, I S	T, I S	T					T			T				
80										I		I		I	T, I S	T, I S			T, I S	T, I S	T	T, I S	I		T, I S				T, I S, H	T, S			
100															T, I	T, I	I		T, I	T, I S	T	T, I S				T			T		T		
120																			I	I, S		T, S		T									
125																				I, S						T							

Note: T : Return Tube I : Internal recirculation S : Super S H : End Cap

(2) Type of nuts

The type of nuts to select depends on the application requirements. HIWIN standard nuts are classified by three letters as follows (see also Chapter 5 for details):



* Other types of nut shape can also be made upon your design.

- The special high-lead double-start nut is classified by adding D in front of the above three letters.
- The compression preload nut is classified by adding P in front of the above three letters.
- The offset pitch preload single nut is classified by adding O in front of the above letters.

Examples :

RDI means round type, double nut with internal return caps.

FSW means flange type, single nut with external return tube within the nut diameter.

DFSV means two-start, flange, single nut with external return tube above the nut diameter.

(3) Number of circuits

The HIWIN nomenclature for the number of circuits in the ballnut is described as follows:

For the external type design:

A : 1.5 turns per circuit
B : 2.5 turns per circuit
C : 3.5 turns per circuit
D : 4.5 turns per circuit
E : 5.5 turns per circuit

For the internal type design:

T : 1.0 turn per circuit

For end cap type design:

U : 2.8 turns per circuit (high lead)
S : 1.8 turns per circuit (super high lead)
V : 0.8 turns per circuit (extra high lead)

For Super S Series:

K : 1 turn per circuit

Example :

B2 : designates 2 external return tube ball circuits. Each circuit has 2.5 turns.

T3 : designates 3 internal return ball circuits. Each circuit has a maximum of 1 turn.

S4 : designates 4 internal return ball circuits. Each circuit has 1.8 turns.

K5 : designates 5 internal return ball circuits. Each circuit has 1 turn.

HIWIN recommends that number of circuits for the external type design be 2 for 2.5 or 3.5 turns (that is, B2 or C2), and 3, 4 or 6 circuits for the internal type. Those shapes are shown in Fig. 3.5 and Fig. 3.6.

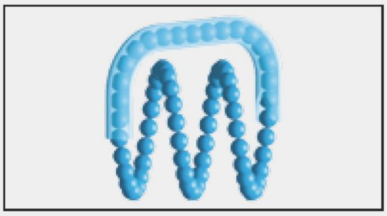


Fig 3.5 Circuit for external return tube

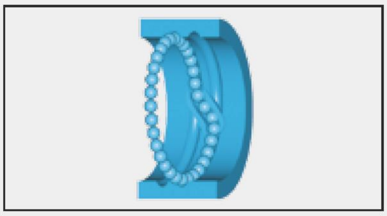


Fig 3.6 Circuit for internal return cap

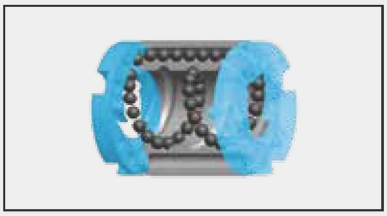


Fig 3.7 Circuit for Endcap

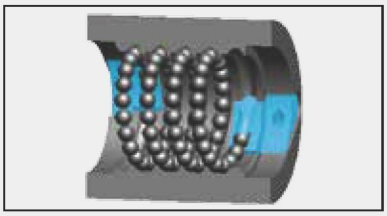


Fig 3.8 Circuit for Super S

3.3 Spindle End and Journal Configuration

Mounting methods

Bearing mounting methods on the end journals of ballscrews are crucial for stiffness, critical speed and column buckling load. Careful consideration is required when designing the mounting method. The basic mounting configuration are shown as follows Fig. 3.9.

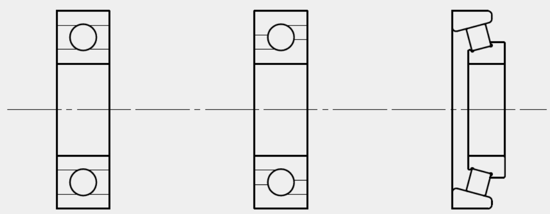
Spindle end journal configurations

The most popular journal configurations are shown in Fig. 3.10.
Table 3.2 lists the recommended dimensions and the bearings for the configurations of Fig. 3.10.

Table 3.2 Dimension for spindle ends

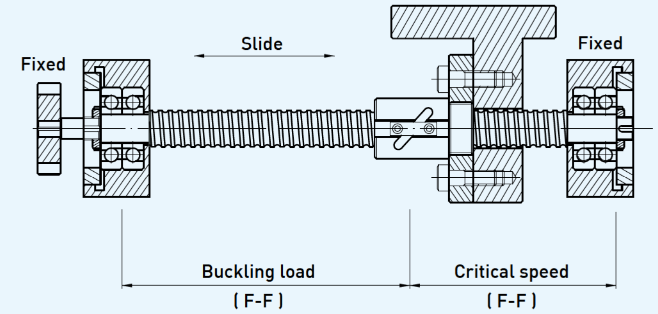
Model	d1	d5	d6	d7	d8	E	L3	L4	L5	L6	L7	L8	L9	L10	L11	L12	L13	bxt1	Recommended Bearing	
																			I.II.III DIN625	III.IV.V DIN625 628 720
10	10	8	7.6	M8x0.75	6	6	16	7	29	26	0.9	39	50	56	18	10	12	3.0x1.8	608	738B
12	12	8	7.6	M8x0.75	6	6	16	7	29	26	0.9	39	50	56	18	10	12	3.0x1.8	608	738B
14	14	10	9.6	M10x0.75	8	8	20	9	37	34	1.15	45	54	62	20	10	14	3.0x1.8	6200	7200BTVP
16	16	12	11.5	M12x1	10	8	21	10	41	38	1.15	46	56	66	20	10	14	4.0x2.5	6201	7301BTVP
20	20	15	14.3	M15x1	12	-	22	11	47	44	1.15	55	70	84	25	13	16	5.0x3.0	6202	7202BTVP
25	25	17	16.2	M17x1	15	-	23	12	49	46	1.15	56	72	86	25	13	16	5.0x3.0	6203	7203BTVP
28	28	20	19	M20x1	16	-	26	14	58	54	1.35	68	82	100	28	20	18	6.0x3.5	6204	7602020TVP
32	32	25	23.9	M25x1.5	20	-	27	15	64	60	1.35	79	94	116	36	22	26	7.0x4.0	6205	7602025TVP
36	36	25	23.9	M25x1.5	20	-	27	15	64	60	1.35	79	94	116	36	22	26	7.0x4.0	6205	7602025TVP
40	40	30	28.6	M30x1.5	25	-	28	16	68	64	1.65	86	102	126	42	22	32	8.0x4.0	6206	7602030TVP
45	45	35	33.3	M35x1.5	30	-	29	17	80	76	1.65	97	114	148	50	24	40	10.0x5.0	6207	7602035TVP
50	50	40	38	M40x1.5	35	-	36	23	93	88	1.95	113	126	160	60	24	45	12.0x5.0	6308	7602040TVP
55	55	45	42.5	M45x1.5	40	-	38	25	93	88	1.95	125	138	168	70	24	50	14.0x5.5	6309	7602045TVP
63	63	50	47	M50x1.5	45	-	33	27	102	97	2.2	140	153	188	80	27	60	14.0x5.5	6310	7602050TVP
70	70	55	52	M55x2.0	50	10	44	29	118	113	2.2	154	167	212	90	27	70	16.0x6.0	6311	7602055TVP
80	80	65	62	M65x2.0	60	10	49	33	132	126	2.7	171	184	234	100	30	80	18.0x7.0	6313	7602065TVP
100	100	75	72	M75x2.0	70	10	53	37	140	134	2.7	195	208	258	120	30	90	20.0x7.5	6315	7602075TVP

* We reserve the right to modify and improve data value without prior notice.
* Different diameters and leads are available upon request.

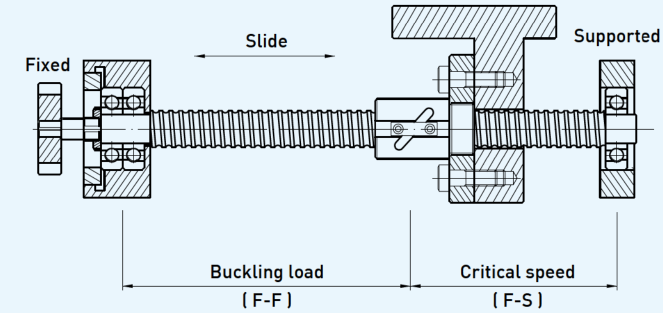


DIN 625   DIN 628   DIN 720

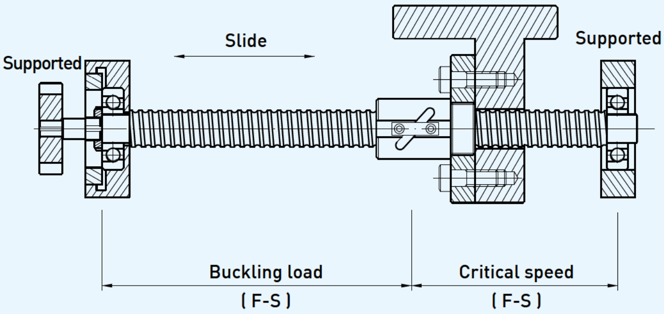
A. Both ends fixed.



B. One end fixed other end supported.



C. Both ends supported.



D. One end fixed other end free.

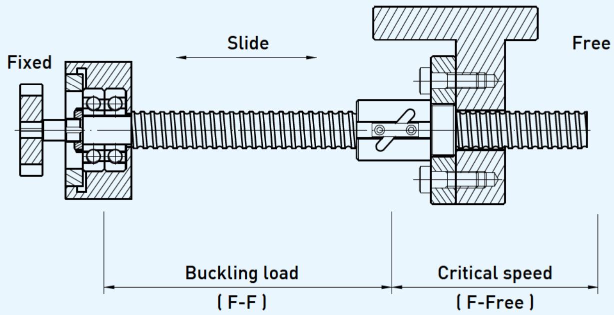
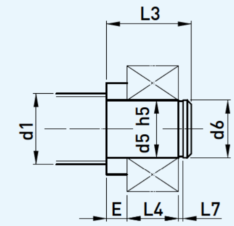
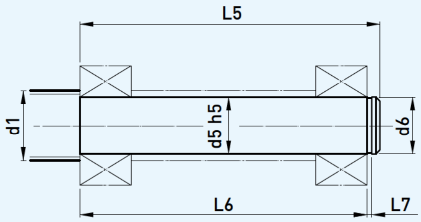


Fig 3.9 Recommended mounting methods for the ballscrew end journals

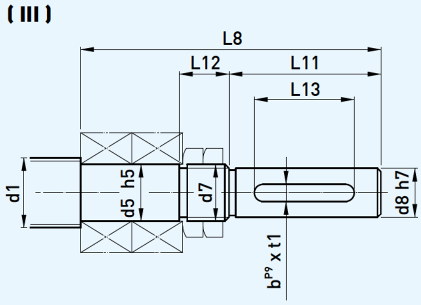
(I)



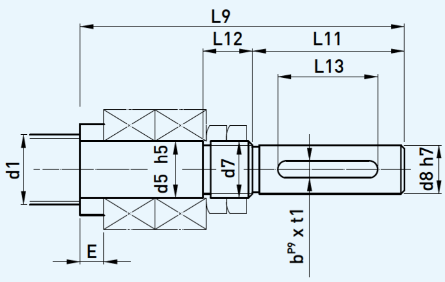
(II)



(III)



(IV)



(V)

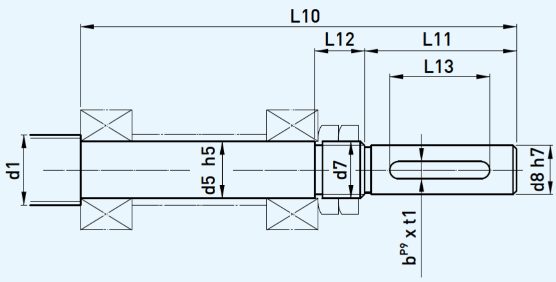


Fig 3.10 Configurations of spindle ends

4 Design and Selection of HIWIN Ballscrews

4.1 Fundamental Concepts for Selection & Installation

(1) A ballscrew must be thoroughly cleaned in white spirit and oil to protect against corrosion. Trichloroethylene is an acceptable degreasing agent, ensuring the ball track free from dirt and damage (paraffin is not satisfactory). Great care must be taken to ensure that the ball track is not struck by a sharp edged component or tool, and metallic debris does not enter the ball nut (Fig. 4.1).

(2) Select a suitable grade ballscrew for the application (ref. Table 4.5). Install with corresponding mounting disciplines. That is, precision ground ballscrews for CNC machine tools demand accurate alignment and precision bearing arrangement, where the rolled ballscrews for less precision applications, such as packaging machinery, require less precise support bearing arrangement.

It is especially important to eliminate misalignment between the bearing housing center and the ballnut center, which would result in unbalanced loads (Fig. 4.2). Unbalanced loads include radial loads and moment loads (Fig. 4.2a). These can cause malfunction and reduce service life (Fig.4.2b).

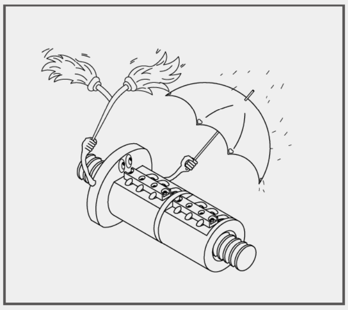


Fig 4.1 Carefully clean and protect

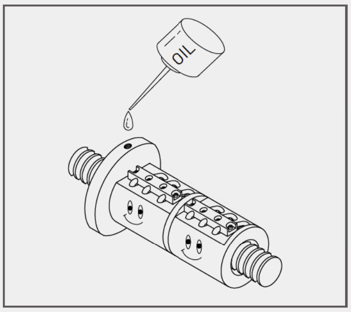


Fig 4.2 Oil lubrication method.

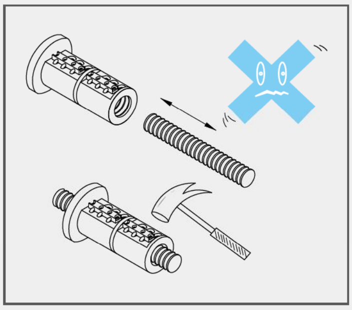


Fig 4.3 Carefully protect the nut

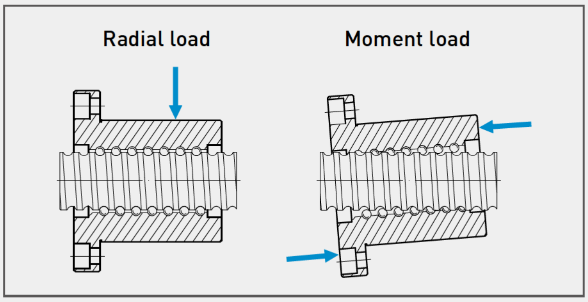


Fig 4.2(a) Unbalance load caused by misalignment of the support bearings and nut brackets, inaccurate alignment of the guide surface, inaccurate angle or alignment of the nut mounting surface

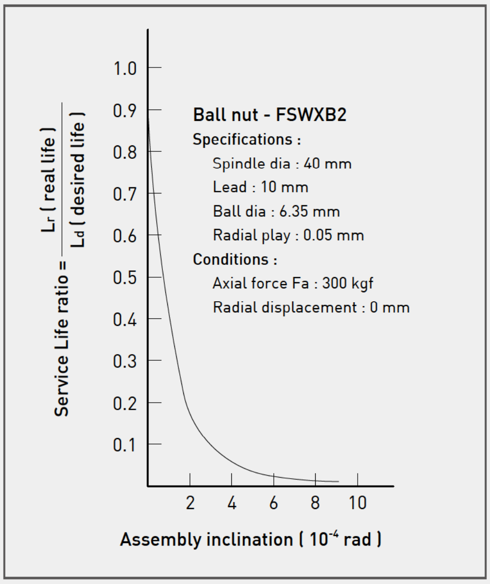
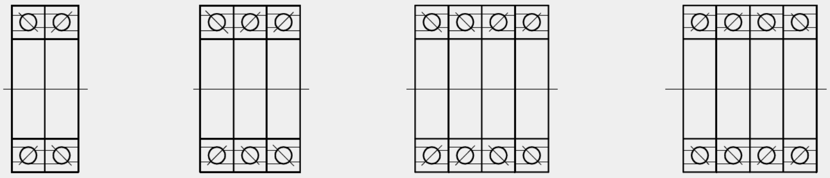


Fig 4.2(b) The effect on service life of a radial load caused by misalignment

(3) To achieve the ballscrews' maximum life, recommend the use of antifriction bearing oils.Oil with graphite and MoS_2 additives must not be used. The oil should be maintained over the balls and the balltracks.
(4) Oil mist bath or drip feeds are acceptable. However, direct application to the ball nut is recommended (Fig. 4.3).
(5) Select a suitable support bearing arrangement for the screw spindle. Angular contact ball bearings (angle=60°) are recommended for CNC machinery. Because of higher axial load capacity and ability to provide a clearance-free or preloaded assembly (Fig. 4.4).



Duplex DF | Triplex DTF | Quadruplex DTDF | Quadruplex DTDB

Fig 4.4 Different arrangement of ballscrew support bearings

(6) A dog stopper should be installed at the end to prevent the nut from over-travelling which results in damage to ballscrew assembly (Fig 4.5).
(7) In environments contaminated by dust or metallic debris, ballscrews should be protected using telescopic or bellow-type covers. The service life of a ballscrew will be reduced to about one-tenth normal condition if debris or chips enter the nut. The bellow type covers may need to have a threaded hole in the flange to fix the cover. Please contact engineers when special modifications are needed (Fig 4.6).

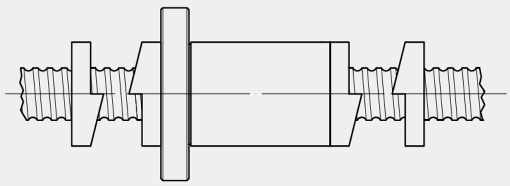


Fig 4.5 A dog stopper to prevent the nut from over travelling

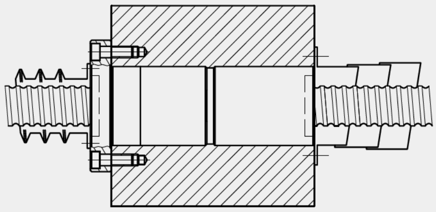


Fig 4.6 Ballscrew protection by telescopic or bellow type covers

(8) If you select an internal recirculation type or an endcap recirculation type ballscrew, one end of the ball thread must be cut through to the end surface. The adjacent diameter on the end journal must be 0.5 ~ 1.0 mm less than the root diameter of the balltracks (Fig 4.7).
(9) After heat treating the ballscrew spindle, both ends of the balltracks adjacent to the journal have about 2 to 3 leads left soft, for the purpose of machining. These regions are shown in (Fig. 4.8) with the mark " ◐ " on HIWIN drawings.
Please contact engineers if special requirements are needed in these regions.

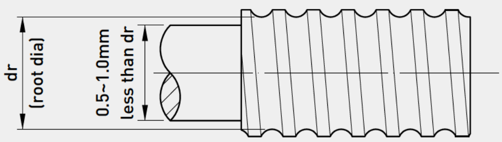


Fig 4.7 Special arrangement for the end journal of an internal recirculation screw

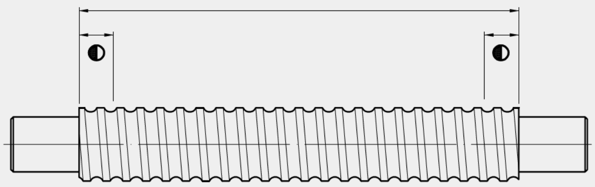


Fig 4.8 The heat treatment range of the ballscrew spindle

(10) Excessive preload increases the friction torque and generates heat which reduces the life expectancy. But insufficient preload reduces stiffness and increases the possibility of lost motion. Recommends that the maximum preload used for CNC machine tools should not exceed 8% of the basic dynamic load C.

(11) When the nut needs to be disassembled from/assembled to the screw spindle, a tube with an outer dia. 0.2 to 0.4 mm less than the root diameter (ref. M37) of the balltracks should be used to release/connect the nut to from/to the screw spindle via one end of the screw spindle shown in Fig. 4.9.

(12) As shown in Fig 4.10, the support bearing must have a chamfer to allow it to seat properly and maintain proper alignment. HIWIN suggests the DIN 509 chamfer as the standard construction for this design (Fig. 4.11).

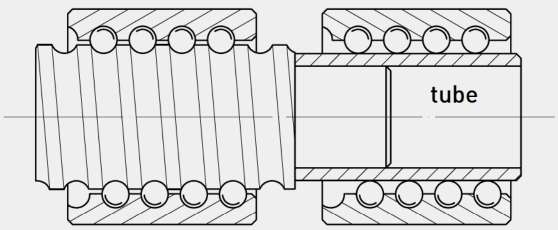


Fig 4.9 The method of separating the nut from the screw spindle

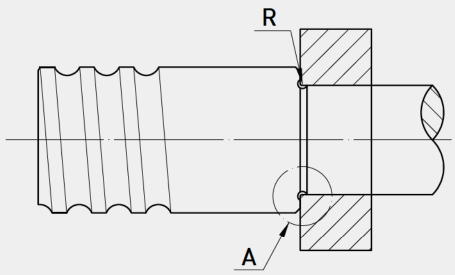
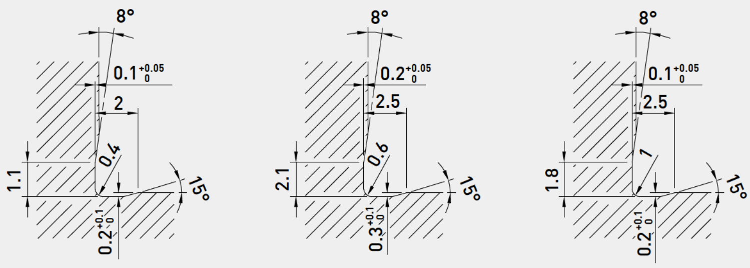


Fig 4.10 Chamfer for seating the face of bearing end



F0.4X0.2DIN509 F0.6X0.3DIN509 F1X0.2DIN509

Fig 4.11 Suggested chamfer dimension per DIN 509 for the "A" dimension in Fig 4.10

4.2 Ballscrews Selection Procedure

The selection procedure for ballscrews is shown in (Table 4.1) From the known design operation condition, (A) select the appropriate parameter of ballscrew, (B) follow the selection procedure step by step via the reference formula, and (C) find the best ballscrew parameters which can be met for the design requirements.

Table 4.1 Ballscrew selection procedure

Step	Design operation condition (A)	Ballscrew parameter (B)	Reference formula(C)
Step 1	Positioning accuracy	Lead accuracy	Table 4.2
Step 2	(1) Max. speed of DC motor (Nmax) (2) Rapid feed rate (Vmax)	Ballscrew lead	$\ell \geq \frac{V_{max}}{N_{max}}$
Step 3	Total travel distance	Total thread length	Total length = thread length+journal end length Thread length = stroke+nut length+100 mm (unused thread)
Step 4	(1) Load condition (%) (2) Speed condition (%)	Mean axial load Mean speed	M7~M10
Step 5	Mean axial force (≤1/5 C is the best)	Preload	M1
Step 6	(1) Service life expectancy (2) Mean axial load (3) Mean speed	Basic dynamic load	M13~M14
Step 7	(1) Basic dynamic load (2) Ballscrew lead (3) Critical speed (4) Speed limited by Dm-N value	Screw diameter and nut type (select some range)	M31~M33 and dimension table
Step 8	(1) Ballscrew diameter (2) Nut type (3) Preload (4) Dynamic load	Stiffness (check the best one via lost motion value)	M34~M40
Step 9	(1) Surrounding temperature (2) Ballscrew length	Thermal displacement and target value of cumulative lead (T)	M41 and 4.6 temperature rising effect
Step 10	(1) Stiffness of screw spindle (2) Thermal displacement	Pretension force	M45
Step 11	(1) Max. table speed (2) Max. rising time (3) Ballscrew specification	Motor drive torque and motor specification	M19~M28

4.3 Accuracy Grade of HIWIN Ballscrews

Precision ground ballscrews are used in applications requiring high positioning accuracy and repeatability, smooth movement and long service life. Ordinary rolled ballscrews are used for application grade less accurate but still requiring high efficiency and long service life. Precision grade rolled ballscrews have an accuracy between that of the ordinary grade rolled ballscrews and the higher grade precision ground ballscrews. They can be used to replace certain precision ground ballscrews with the same grade in many applications.

HIWIN makes precision grade rolled ballscrew up to C6 grade. Geometric tolerances are different from those of precision ground screws (See Chapter 6). Since the outside diameter of the screw spindle is not ground, the set-up procedure for assembling precision rolled ballscrews into the machine is different from that of ground ones. Chapter 7 contains the entire description of rolled ballscrews.

(1) Accuracy grade

There are numerous applications for ballscrews from high precision grade ballscrews, used in precision measurement and aerospace equipment, to transport grade ballscrews used in packaging equipment. The quality and accuracy classifications are described as follows: lead deviation, surface roughness, geometrical tolerance, backlash, drag torque variation, heat generation and noise level.

HIWIN precision ground ballscrews are classified to 7 classes. In general, HIWIN precision grade ballscrews are defined by the so called "υ_{300p}" value see Fig 4.12 and rolled grade ballscrews are defined differently as shown in Chapter 7.

Fig. 4.12 is the lead measuring chart according to the accuracy grade of the ballscrews. The same chart by the DIN system is illustrated in Fig. 4.13. From this diagram, the accuracy grade can be determined by selecting the suitable tolerance in Table 4.2. Fig. 4.14 shows HIWIN's measurement result according to the DIN standard. Table 4.2 shows the accuracy grade of precision grade ballscrews in HIWIN's specification.The relative international standard is shown in Table 4.3.

The positioning accuracy of machine tools is selected by e_p value with the υ_{300p} variation. The recommended accuracy grade for machine applications is shown in Table 4.5. This is the reference chart for selecting the suitable ballscrews in different application fields.

(2) Axial play (Backlash)

If zero axial play ballscrews (no backlash) are needed, preload should be added and the preload drag torque is specified for testing purpose. The standard axial play of HIWIN ballscrews is shown in Table 4.4.For CNC machine tools, lost motion can occur in zero-backlash ballscrews through incorrect stiffness. Please consult our engineers when determining stiffness and backlash requirements.

(3) Geometrical tolerance

It is crucial to select the ballscrew of the correct grade to meet machinery requirements. Table 4.6 and Fig 4.15 are helpful for you to determine the tolerance factors, which are based on certain required accuracy grades.

Table 4.2 HIWIN accuracy grade of precision ballscrew Unit: 0.001mm

Accuracy Grade		C0		C1		C2		C3		C4		C5		C6	
$\upsilon_{2\pi p}$		3		4		4		6		8		8		8	
υ_{300p}		3.5		5		6		8		12		18		23	
Thread length \ Item		e_p	υ_u	e_p	υ_u	e_p	υ_u	e_p	υ_u	e_p	υ_u	e_p	υ_u	e_p	υ_u
above	below														
-	315	4	3.5	6	5	6	6	12	8	12	12	23	18	23	23
315	400	5	3.5	7	5	7	6	13	10	13	12	25	20	25	25
400	500	6	4	8	5	8	7	15	10	15	13	27	20	27	26
500	630	6	4	9	6	9	7	16	12	16	14	30	23	30	29
630	800	7	5	10	7	10	8	18	13	18	16	35	25	35	31
800	1000	8	6	11	8	11	9	21	15	21	17	40	27	40	35
1000	1250	9	6	13	9	13	10	24	16	24	19	46	30	46	39
1250	1600	11	7	15	10	15	11	29	18	29	22	54	35	54	44
1600	2000			18	11	18	13	35	21	35	25	65	40	65	51
2000	2500			22	13	22	15	41	24	41	29	77	46	77	59
2500	3150			26	15	26	17	50	29	50	34	93	54	93	69
3150	4000			30	18	32	21	60	35	62	41	115	65	115	82
4000	5000							72	41	76	49	140	77	140	99
5000	6300							90	50	100	60	170	93	170	119
6300	8000							110	60	125	75	210	115	210	130
8000	10000											260	140	260	145
10000	12000											320	170	320	180

Table 4.3 International standard of accuracy grade for ballscrews Unit: 0.001mm

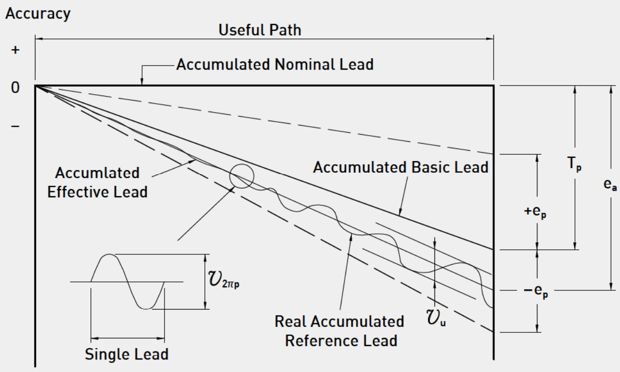
Grade		Ground									
								Rolled			
		C0	C1	C2	C3	C4	C5	C6	C7	C8	C10
υ_{300p}	ISO, DIN		6		12		23		52		210
	JIS	3.5	5		8		18		50		210
	HIWIN	3.5	5	6	8	12	18	23	50	100	210

Table 4.4 Standard combination of grade and axial play Unit: 0.001mm

Grade	C0	C1	C2	C3	C4	C5	C6
Axial Play	5	5	5	10	15	20	25

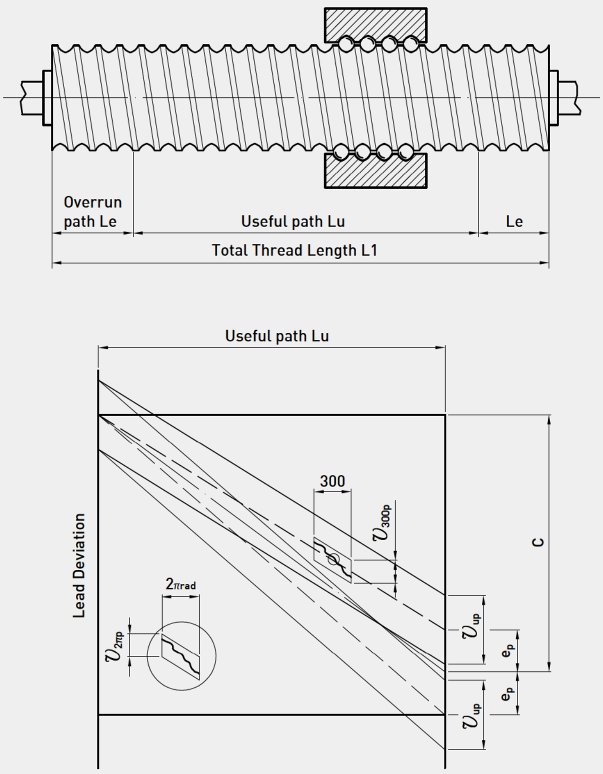
Table 4.5 Recommended accuracy grade for machine applications

Application grade		AXIS	Accuracy grade									
			0	1	2	3	4	5	6	7	8	10
CNC Machinery Tools	Lathes	X	•	•	•	•	•					
		Z				•	•	•				
	Milling machines Boring machines	X		•	•	•	•	•				
		Y		•	•	•	•	•				
		Z			•	•	•	•				
	Machine Center	X		•	•	•	•					
		Y		•	•	•	•					
		Z			•	•	•					
	Jig borers	X	•	•								
		Y	•	•								
		Z	•	•								
	Drilling machines	X				•	•	•				
		Y				•	•	•				
		Z					•	•	•			
	Grinders	X	•	•	•							
		Y		•	•	•						
	EDM	X		•	•	•						
		Y		•	•	•						
		Z			•	•	•	•				
	Wire cut EDM	X		•	•	•						
		Y		•	•	•						
		U		•	•	•	•					
		V		•	•	•	•					
	Laser Cutting Machine	X			•	•	•					
		Y			•	•	•					
		Z			•	•	•					
General Machinery	Punching Press	X				•	•	•				
		Y				•	•	•				
	Single Purpose Machines			•	•	•	•	•	•			
	Wood working Machines								•	•	•	•
	Industrial Robot (Precision)			•	•	•	•					
	Industrial Robot (General)							•	•	•	•	
	Coordinate Measuring Machine		•	•	•							
	Non-CNC Machine					•	•	•				
	Transport Equipment						•	•	•	•	•	•
	X-Y Table			•	•	•	•	•				
	Linear Actuator							•	•	•	•	
	Aircraft Landing Gear							•	•	•	•	
	Airfoil Control							•	•	•	•	
	Gate Valve								•	•	•	•
	Power steering								•	•	•	
	Glass Grinder				•	•	•	•	•			
	Surface Grinder						•	•				
	Induction Hardening Machine								•	•	•	•
	Electromachine			•	•	•	•	•	•			
	All-electric injection molding machine							•	•	•	•	•



- T_p : Target point of accumulated lead. This value is determined by customers' different application requirements.
- e_p : Total reference lead deviation. Maximum deviation for accumulated reference lead line over the full length.
- $\upsilon_{2\pi p}$: Single lead variation.
- e_a : Real accumulated reference lead measured by laser system.
- υ_u : Total relative lead deviation. Maximum deviation of the real accumulated lead from the real accumulated reference lead in the corresponding range.
- υ_{300p} : Lead deviation over path of 300mm. The above deviation in random 300 mm within thread length.

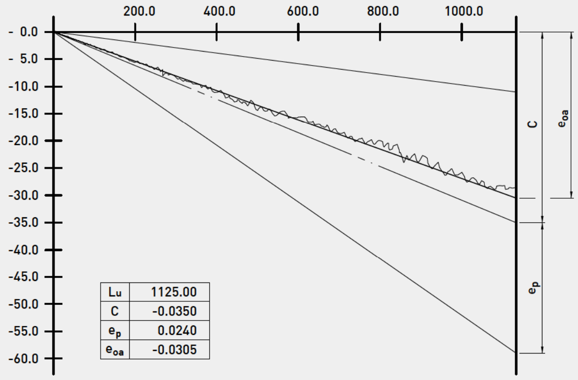
Fig 4.12 HIWIN lead measuring curve of precision ballscrew



- e_{oa} : Average lead deviation over useful path Lu. A straight line representing the tendency of the cumulative actual lead. This is obtained by the least square method and measured by the laser system. The value is added by path compensation over the useful path and the mean travel deviation.
- C : Path compensation over useful path Lu. Selection parameter:This value is determined by customer and maker as it depends on different application requirements.
- e_p : Mean travel deviation.
- υ_{up} : Lead variation over useful path Lu.
- υ_{300p} : Lead variation over path of 300 mm.
- $\upsilon_{2\pi p}$: Lead variation over 1 rotation.

Fig 4.13 DIN lead measuring curve of precision ballscrew

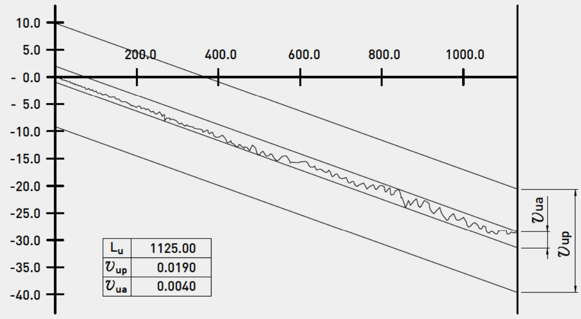
AVERAGE LEAD DEVIATION OVER USEFUL PATH LU



- $e_{oa}(E_a)$:

Lead deviation over useful thread length relative to the nominal deviation.
(This measurement is made according to DIN standard 69051-3-1).
$C(T) - e_p(Ep) \le e_{oa}(Ea) \le C(T) + e_p(E_p)$

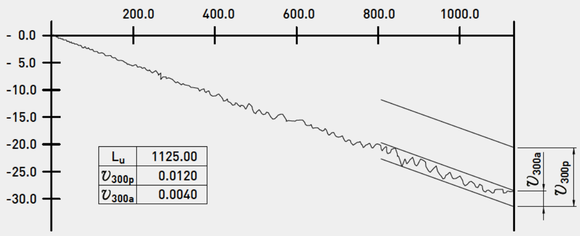
LEAD VARIATION OVER USEFUL PATH LU



- $\upsilon_{ua}(e_a)$:

Total relative lead variation over useful thread length.
(This measurement is made according to DIN standard 69051-3-2).
$\upsilon_{ua}(e_a) \le \upsilon_{up}(e_p)$

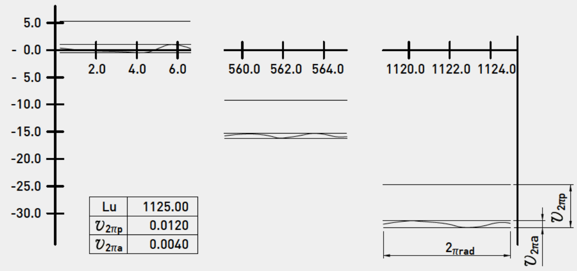
LEAD VARIATION OVER PATH OF 300MM



- $\upsilon_{300a}(e_{300a})$:

Relative lead variation in random 300mm length within thread length.
(This measurement is made according to DIN standard 69051-3-3).
$\upsilon_{300a}\ (e_{300a}) \le \upsilon_{300p}\ (e_{300p})$

LEAD VARIATION OVER 1 ROTATION



- $\upsilon_{2\pi a}(e_{2\pi a})$:

Single lead variation over 2p.
(This measurement is made according to DIN standard 69051-3-4).
$\upsilon_{2\pi a}(e_{2\pi a}) \le \upsilon_{2\pi p}(e_{2\pi p})$

Fig 4.14 Lead accuracy measuring chart from dynamic laser measurement equipment according to DIN 69051 standard

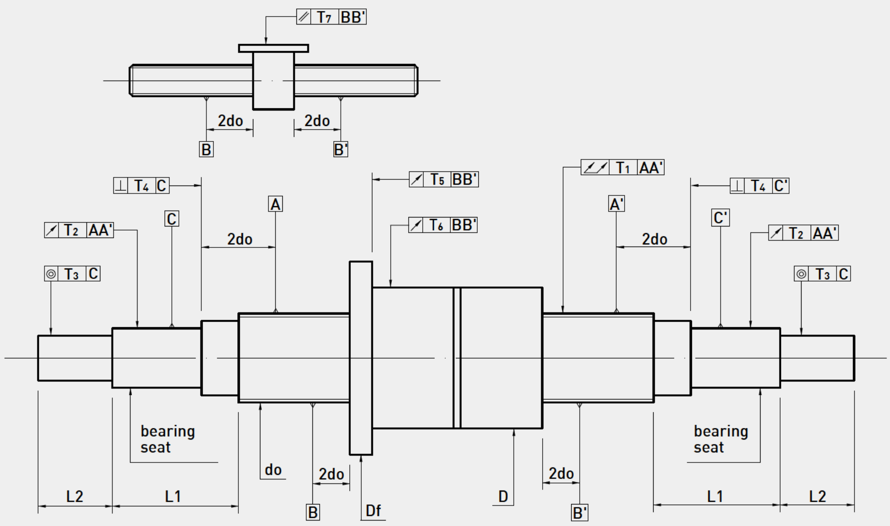
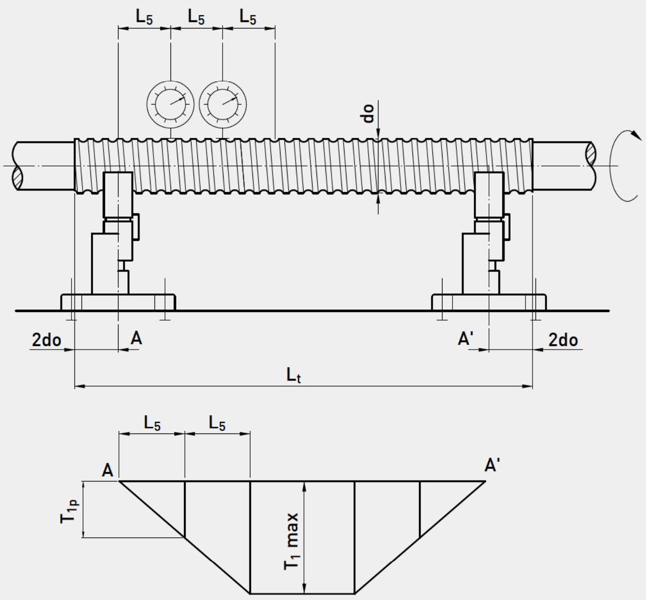


Fig 4.15 Geometrical tolerance of HIWIN precision ground ballscrew

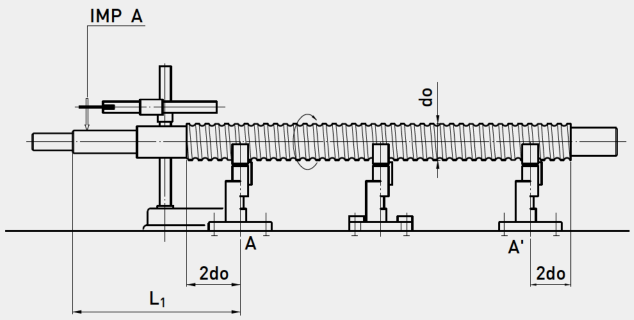
Table 4.6 Tolerance table and measurement method for HIWIN precision ballscrews



T1: True running deviation of external diameter relative to AA' (This measurement is made according to DIN 69051 and JIS B1192)

Nominal Diameter do (mm)		reference length	T_{1p} [μm] For HIWIN tolerance class							
above	up to	L5	0	1	2	3	4	5	6	7
6	12	80								
12	25	160								
25	50	315	20	20	20	23	25	28	32	40
50	100	630								
100	200	1250								

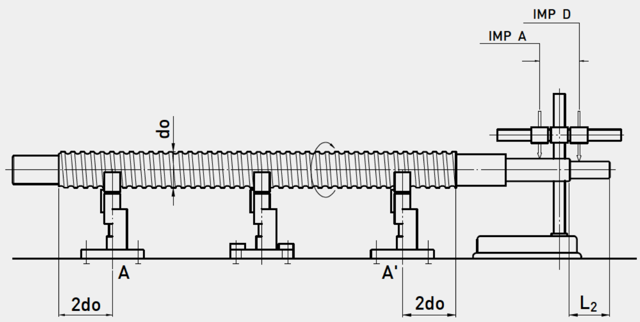
Lt/do		T_{1MAX} [μm] (for $L_t \ge 4L_5$) For HIWIN tolerance class							
above	up to	0	1	2	3	4	5	6	7
	40	40	40	40	45	50	60	64	80
40	60	60	60	60	70	75	85	96	120
60	80	100	100	100	115	125	140	160	200
80	100	160	160	160	180	200	220	256	320



T2: Run out deviation of bearing relative to AA' (This measurement is made according to DIN 69051 and JIS B1192)

Nominal Diameter do (mm)		reference length	T_{2P} [μm] (for $L_1 \le L_r$) For HIWIN tolerance class							
above	up to	Lr	0	1	2	3	4	5	6	7
6	20	80	6	8	10	11	12	16	20	40
20	50	125	8	10	12	14	16	20	25	50
50	125	200	10	12	16	18	20	26	32	63
125	200	315	-	-	-	20	25	32	40	80

if $L_1 > L_r$, then $t_{2a} \le T_{2p} \frac{L_1}{L_r}$

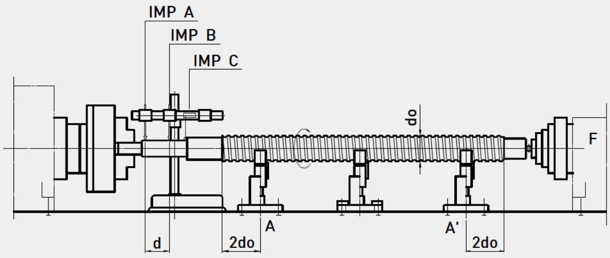


T3: Coaxial deviation relative to AA' (This measurement is made according to DIN 69051 and JIS B1192)

Nominal Diameter do (mm)		reference length	T_{3P} [μm] (for $L_2 \le L_r$) For HIWIN tolerance class							
above	up to	Lr	0	1	2	3	4	5	6	7
6	20	80	4	5	5	6	6	7	8	12
20	50	125	5	6	6	7	8	9	10	16
50	125	200	6	7	8	9	10	11	12	20
125	200	315	-	-	-	10	12	14	16	25

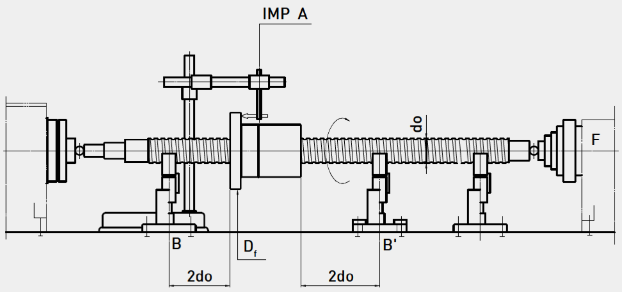
if $L_2 > L_r$, then $t_{3a} \le T_{3p} \frac{L_2}{L_r}$

Table 4.6 Tolerance table and measurement method for HIWIN precision ballscrews



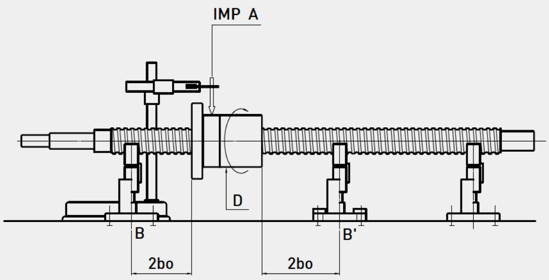
T4 : Run-out deviation of bearing end shoulder relative to AA' (This measurement is made according to DIN 69051 and JIS B1192)

Nominal Diameter do (mm)		T_{4P} [μm] For HIWIN tolerance class							
above	up to	0	1	2	3	4	5	6	7
6	63	3	3	3	4	4	5	5	6
63	125	3	4	4	5	5	6	6	8
125	200	-	-	-	6	6	8	8	10



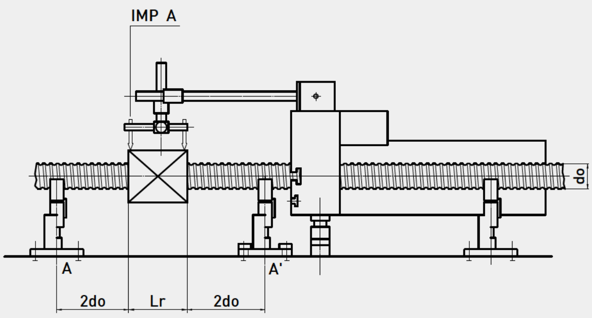
T5 : Face running deviation of locating face (only for nut) relative to BB' (This measurement is made according to DIN 69051 and JIS B1192)

Nut Flange Diameter D_f (mm)		T_{5P} [μm] For HIWIN tolerance class							
above	up to	0	1	2	3	4	5	6	7
-	20	5	6	7	8	9	10	12	14
20	32	5	6	7	8	9	10	12	14
32	50	6	7	8	8	10	11	15	18
50	80	7	8	9	10	12	13	16	18
80	125	7	9	10	12	14	15	18	20
125	160	8	10	11	13	15	17	19	20
160	200	-	11	12	14	16	18	22	25
200	250	-	12	14	15	18	20	25	30



T6 : Run-out deviation of external diameter (only for nut) relative to BB' (This measurement is made according to DIN 69051 and JIS B1192)

Nut Diameter Diameter D (mm)		T_{6P} [μm] For HIWIN tolerance class							
above	up to	0	1	2	3	4	5	6	7
-	20	5	6	7	9	10	12	16	20
20	32	6	7	8	10	11	12	16	20
32	50	7	8	10	12	14	15	20	25
50	80	8	10	12	15	17	19	25	30
80	125	9	12	16	20	24	22	25	40
125	160	10	13	17	22	25	28	32	40
160	200	-	16	20	22	25	28	32	40
200	250	-	17	20	22	25	28	32	40



T7 : Deviation of parallelism (only for nut) relative to BB' (This measurement is made according to DIN 69051 and JIS B1192)

Mounting basic length (mm) Lr		T_{7P} [μm] / 100mm For HIWIN tolerance class							
above	up to	0	1	2	3	4	5	6	7
-	50	5	6	7	8	9	10	14	17
50	100	7	8	9	10	12	13	15	17
100	200	-	10	11	13	15	17	24	30

4.4 Preload Methods

The specially designed Gothic ball track can make the ball contact angle around 45°. The axial force F_a which comes from an outside drive force or inside preload force, causes two kinds of backlash. One is the normal backlash, S_a caused by the manufacturing clearance between ball track and ball. The other is the deflection backlash, $\Delta\ell$ caused by the normal force F_n which is perpendicular to the contact point.

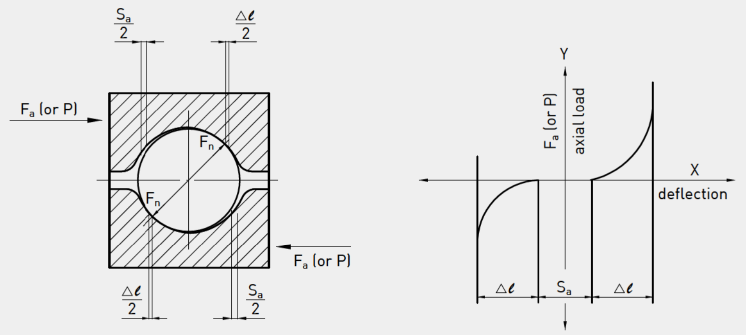


Fig 4.16 Gothic form profile and preloading relation

The clearance backlash can be eliminated by the use of an preload internal force P. This preload can be obtained via a double nut, an offset pitch single nut, or by adjusting the ball size for preloaded single nuts.

The deflection backlash is caused by the preload internal force and the external loading force and is related to that of the effect of lost motion.

(1) Double nut preloading

Preload is obtained by inserting a spacer between the 2 nuts (Fig. 4.17). "Tension preload" is created by inserting an oversize spacer and effectively pushing the nuts apart. "Compression pre-load" is created by inserting an undersize spacer and correspondingly pulling nuts together. Tension preload is primarily used for precision ballscrews. However, compression preload type ballscrews are also available upon your request. If pretension is necessary to increase stiffness, please contact us for the amount of pretension to be used in the ballscrew journal ends. (0.02mm to 0.03mm per meter is recommended, but the T value should be selected according to the compensation purpose).

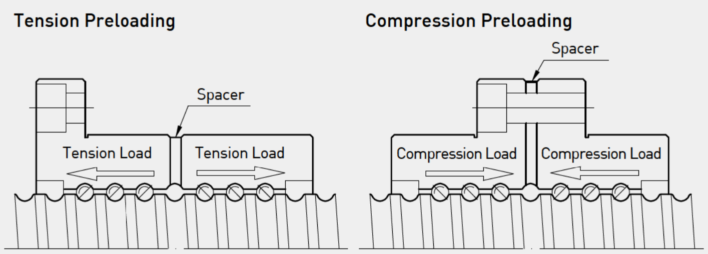


Fig 4.17 Preload by spacer

(2) Single nut preloading

There are two ways of preloading a single nut. One is called "the oversized-ball preloading method". The method is to insert balls slightly larger than the ball groove space (oversized balls) to allow balls to contact at four points (Fig. 4.18).

The other way is called "The offset pitch preloading method" as shown in Fig. 4.19. The nut is ground to have a δ value offset on the center pitch. This method is used to replace the traditional double nut preloading method and has the benefit of a compact single nut with high stiffness via small preload force. However, it should not be used in heavy duty preloading. The best preload force is below 5% of dynamic load (C).

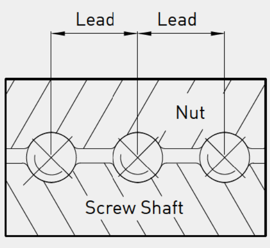


Fig 4.18 Preload by ball size

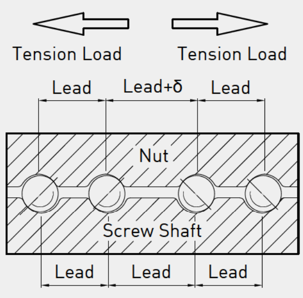


Fig 4.19 Offset type preloading

(3) Preload calculation

$$p = \frac{F_{bm}}{2.8} \quad \text{M1}$$

P : preload force (kgf)

F_{bm} : Mean operating load(kgf)

(Ref.M8~M10)

$$T_d = \frac{K_p \times P \times \ell}{2\pi} \quad \text{M2}$$

Preload drag torque (Fig. 4.20)

T_d : preload drag torque (kgf-mm)

P : preload (kgf)

ℓ : lead (mm)

K_p : preload torque coefficient **

K_p : $\frac{1}{\eta_1}$ - η_2 (is between 0.1 and 0.3)

η_1 , η_2 are the mechanical efficiencies of the ballscrew.

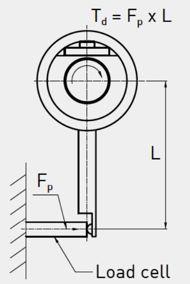


Fig 4.20 :
Preload drag torque measuring method(according to JIS B1192)

(1) For common transmission (to convert rotary motion to linear motion)

$$\eta_1 = \frac{\tan(\alpha)}{\tan(\alpha+\beta)} = \frac{1-\mu\tan\alpha}{1+\mu/\tan\alpha} \quad \text{M3}$$

(2) For reverse transmission (to convert linear rotary motion to rotary motion)

$$\eta_2 = \frac{\tan(\alpha-\beta)}{\tan(\alpha)} = \frac{1-\mu/\tan\alpha}{1+\mu\tan\alpha} \quad \text{M4}$$

$$\alpha = \tan^{-1}\frac{\ell}{\pi D_m} \quad \text{M5}$$

$$\beta = \tan^{-1}\mu \quad \text{M6}$$

α : lead angle (degrees)

D_m : pitch circle diameter of screw shaft (mm)

l : lead (mm)

β : friction angle (0.17º~0.57º)

μ : friction coefficient (0.003~0.01)

$$** \; K_p = \frac{0.05}{\sqrt{\tan\alpha}}$$

(4) Uniformity of preload drag torque

(1) Measuring method

Preload creates drag torque between the nut and screw. It is measured by rotating the screw spindle at constant speed while restraining the nut with a special fixture as shown in Fig. 4.20. The load cell reading force Fp is used to calculate the preload drag torque of the ballscrew.

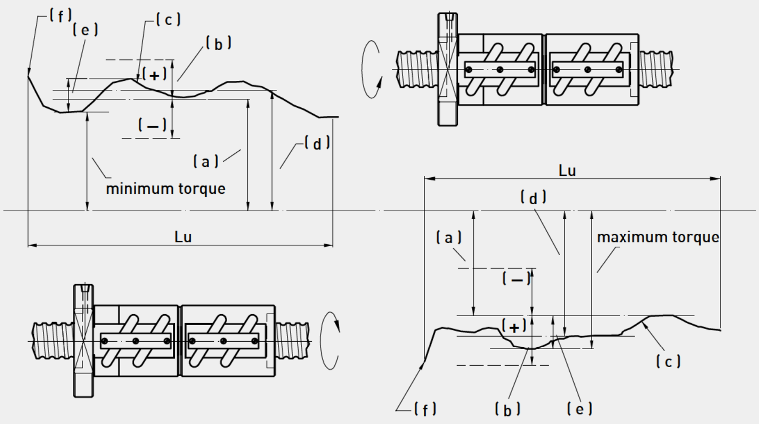
HIWIN has developed a computerized drag torque measuring machine which can accurately monitor the drag torque during screw rotation. Therefore, the drag torque can be adjusted to meet customer requirements (Fig. 2.5). The measurement standard for preload drag torque is shown in Fig. 4.21 and Table 4.7.

(2) Measuring conditions

1. Without wiper.
2. The rotating speed, 100 rpm.
3. The dynamic viscosity of lubricant, 61.2 ~74.8 cSt (mm/s) 40˚C, that is, ISO VG 68 or JIS K2001.
4. The return tube up.

(3) The measurement result is illustrated by the standard drag torque chart. Its nomenclature is shown in Fig. 4.21.

(4) The allowable preload drag torque variation as a function of accuracy grade is shown in Table 4.7.



(a) : basic drag torque.
(b) : Variation of basic preload drag torque.
(c) : Actual torque.
(d) : Mean actual preload drag torque.
(e) : Variation value of actual preload drag torque.
(f) : Starting actual torque.
Lu : Useful travelling distance of nut

Fig 4.21 Nomenclature of drag torque measurement

Table 4.7 : Variation range for preload drag torque (According to JIS B1192) Unit: ± %

(1) Basic Dragtorque (kgf - cm)		Useful stroke length of thread (mm)																							
		4000 mm maximum																over 4000 mm							
		Slender ratio ≤ 40								40 < Slender ratio < 60															
		Accuracy grade								Accuracy grade								Accuracy grade							
Above	Up To	0	1	2	3	4	5	6	7	0	1	2	3	4	5	6	7	0	1	2	3	4	5	6	7
2	4	30	35	40	40	45	50	60	-	40	40	50	50	60	60	70	-	-	-	-	-	-	-	-	-
4	6	25	30	35	35	40	40	50	-	35	35	40	40	45	45	60	-	-	-	-	-	-	-	-	-
6	10	20	25	30	30	35	35	40	40	30	30	35	35	40	40	45	45	-	-	-	40	43	45	50	50
10	25	15	20	25	25	30	30	35	35	25	25	30	30	35	35	40	40	-	-	-	35	38	40	45	45
25	63	10	15	20	20	25	25	30	30	20	20	25	25	30	30	35	35	-	-	-	30	33	35	40	40
63	100	-	15	15	15	20	20	25	30	-	-	20	20	25	25	30	35	-	-	-	25	23	30	35	35

Note :
1. Slender ratio=Thread length of spindle/ Nominal spindle O.D.(mm)
2. Refer to the designing section of the manual to determine the basic preload drag torque.
3. Table 4.9 shows the conversion table for Nm.
4. For more information, please contact our engineering department.

4.5 Calculation Formulas

Service life

• The average number of rpm, n_{av}

$$n_{av} = n_1 \times \frac{t_1}{100} + n_2 \times \frac{t_2}{100} + n_3 \times \frac{t_3}{100} + \ldots\ldots \quad \text{(M7)}$$

n_{av} : average speed (rpm)

n : speed (rpm)

$\frac{t_1}{100}$: % of time at speed n_1 etc.

- The average operating load F_{bm}

(1) With variable load and constant speed

$$F_{bm} = \sqrt[3]{F_{b1}^3 \times \frac{t_1}{100} \times f_{p1}^3 + F_{b2}^3 \times \frac{t_2}{100} \times f_{p2}^3 + F_{b3}^3 \times \frac{t_3}{100} \times f_{p3}^3 \ \cdots\cdots} \quad \text{M8}$$

F_{bm} : average operating load (kgf); F_b : working axial load

f_p : operation condition factor

f_p : 1.1 ~ 1.2 when running without impact

1.3 ~ 1.8 when running in the normal condition

2.0 ~ 3.0 when running with heavy impact and vibration

(2) With variable load and variable speed

$$F_{bm} = \sqrt[3]{F_{b1}^3 \times \frac{n_1}{n_{av}} \times \frac{t_1}{100} \times f_{p1}^3 + F_{b2}^3 \times \frac{n_2}{n_{av}} \times \frac{t_2}{100} \times f_{p2}^3 + F_{b3}^3 \times \frac{n_3}{n_{av}} \times \frac{t_3}{100} \times f_{p3}^3 \ \cdots\cdots} \quad \text{M9}$$

(3) With linear variable load and constant speed

$$F_{bm} \fallingdotseq \frac{F_{b\,min} \times f_{p1} + 2 \times F_{b\,max} \times f_{p2}}{3} \quad \text{M10}$$

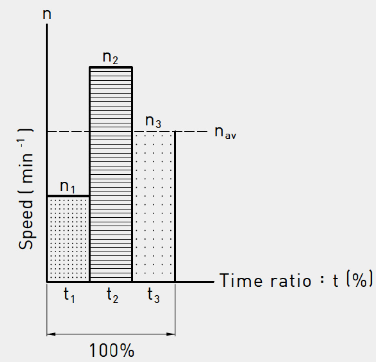


Fig 4.22 Equivalent speed

Example 4.5 - 1

A HIWIN ballscrew is subjected to the following operating conditions. Calculate the average running speed and operating load.

Operating Condition :

For smooth running without impact fp = 1.1

Condition	Axial load (kgf)	Revolution (rpm)	Loading time ratio (%)
	(Fb)	(n)	(t)
1	100	1000	45
2	400	50	35
3	800	100	20

Calculation

$$n_{av} = 1000 \times \frac{45}{100} + 50 \times \frac{35}{100} + 100 \times \frac{20}{100} = 487.5\text{rpm (ref.M7)}$$

$$F_{bm} = \sqrt[3]{100^3 \times \frac{1000}{487.5} \times \frac{45}{100} \times 1.1^3 + 400^3 \times \frac{50}{487.5} \times \frac{35}{100} \times 1.1^3 + 800^3 \times \frac{100}{487.5} \times \frac{20}{100} \times 1.1^3} = 318.5\ kgf$$

The resultant axial force, Fa

For a single nut without preload

$$F_a = F_{bm} \qquad \text{M11}$$

For a single nut with preload P

$$F_a \leq F_{bm} + P \qquad \text{M12}$$

Expected service life

For single nut

• Service life represented in revolutions :

$$L = \left(\frac{C}{F_a}\right)^3 \times 10^6 \qquad \text{M13}$$

L : Service life in running revolution (revolutions)

C : dynamic load rating (kgf) (10^6 rev)

For symmetrical preload double nut arrangement

(a) Service life represented in revolutions :

$$F_{bm}(1) = P\left(1 + \frac{F_{bm}}{3P}\right)^{3/2} \qquad L(1) = \left(\frac{C}{F_{bm}(1)}\right)^3 \times 10^6$$

$$F_{bm}(2) = F_{bm}(1) - F_{bm} \qquad L(2) = \left(\frac{C}{F_{bm}(2)}\right)^3 \times 10^6$$

$$L = [\,L(1)^{-10/9} + L(2)^{-10/9}]^{-9/10} \qquad \text{M14}$$

L = Service life in running revolution (revolutions)

C : Preload force (kgf)

(b) conversion from revolutions to hours :

$$L_h = \frac{L}{n_{av} \times 60} \qquad \text{M15}$$

L_h : Service life in hours (hours)

n_{av} : Average speed (rpm, Ref. M7)

(c) Conversion from travel distance to hours:

$$L_h = \left(\frac{L_d \times 10^6}{\ell}\right) \times \frac{1}{n_{av} \times 60} \qquad \text{M16}$$

L_h : Running life (in hours)

L_d : Running life (in distance, Km)

ℓ : Ballscrew lead (mm per rev)

n_{av} : Average running speed (rpm)

(d) the modified service life for different reliability factors is calculated by

$$L_m = L \times f_r \qquad \text{M17}$$

$$L_{hm} = L_h \times f_r \qquad \text{M18}$$

with the reliability factor fr (Table 4.8)

Table 4.8 Reliability factor for service life

Reliability %	f_r
90	1
95	0.63
96	0.53
97	0.44
98	0.33
99	0.21

Example 4.5 - 2

By the example 4.5-1, if the design service life of the ballscrew is 3500 hours, lead = 10mm, single nut with zero backlash, find the nominal diameter of the HIWIN ballscrew.

Calculation

$$P = \frac{F_{bm}}{2.8} = \frac{318.5}{2.8} = 114\ kgf \quad \text{(Assume zero backlash when } F_{bm} = 318.5 \text{ kgf)}$$

$$F_a = F_{bm} + p = 318.5 + 114 = 432.5\ kgf \quad \text{(Ref formula M1)}$$

$$L = L_h \times n_{av} \times 60 = 3500 \times 487.5 \times 60 = 1.02375 \times 108 \quad \text{(revolutions)}$$

$$C' = F_a \left(\frac{L}{10^6}\right)^{1/3} = 432.5 \times \left(\frac{1.02375 \times 10^8}{10^6}\right)^{1/3} = 2023\ kgf \quad C' \leq \text{rating}$$

So, from the dimensions table of HIWIN ballscrews, select FSV type nut with spindle nominal diameters equals 32mm and C1 circuits which can satisfy this application.

Example 4.5 - 3

If the ballscrew nominal diameter=50mm, lead=8mm, and service life L=7x10⁶ revolutions, find the permissible load on the screw spindle.

Calculation

From the dimensions table of HIWIN ballscrew, the FSV type ballscrew with nominal diameter=50 mm, lead=8 mm and B3 type return tube has the dynamic load rating C=5674.

$$Fa = C \div \left(\frac{L}{10^6}\right)^{1/3} = 5674 \div \left(\frac{7 \times 10^6}{10^6}\right)^{1/3} = 2966\ kgf$$

Drive torque and drive power for the motor

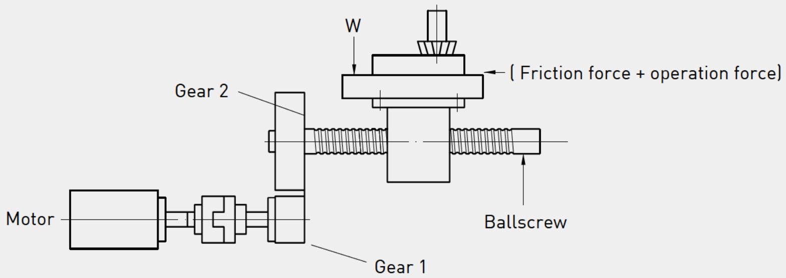


Fig 4.23 Load operation by ballscrew

Fig. 4.23 shows the terms for a feed system operated by ballscrew. The formula for motor drive torque is given below :

(a) Common transmission (to convert rotary motion to linear motion)

$$T_a = \frac{F_b \times \ell}{2\pi\eta_1} \quad \text{M19}$$

T_a = Drive torque for common transmission (kgf-mm)

F_b = Axial load (kgf)

$F_b = F_{bm} + \mu \times W$ (for horizontal motion)

ℓ = Lead (mm)

η_1 = Mechanical efficiency (0.9~0.95, Ref. M3)

W = Table wight + Work piece weight (kgf)

μ = Friction coefficient of table guide way

(b) Reverse transmission (to convert linear motion to rotary motion)

$$T_c = \frac{F_b \times \ell \times \eta_2}{2\pi} \quad \text{M20}$$

η_2 = Mechanical effciency (0.9~0.95, Ref. M4)

T_c = Torque for reverse transmission (kgf-mm)

(c) Motor drive torque

For normal operation :

$$T_M = (T_a + T_b + T_d) \times \frac{N_1}{N_2} \quad \text{M21}$$

T_M = Motor drive torque (kgf-mm)

T_b = Friction torque of supporting bearing (kgf-mm)

T_d = Preload drag torque (kgf-mm, Ref. M2)

N_1 = Number of teeth for driver gear

N_2 = Number of teeth for driven gear

For acceleration operation :

$$T'a = J\alpha \quad \text{M22}$$

$T'a$: Motor drive torque during acceleration (kgf)

J : System inertia (kgf-mm-sec^2)

α : Angular acceleration (rad/sec^2)

$$\alpha = \frac{2\pi N_{dif}}{60 t_a} \quad \text{M23}$$

$N_{dif} = rpm_{stage2} - rpm_{stage1}$

t_a = acceleration rising time (sec)

$$J = J_M + J_{G1} + J_{G2}\left(\frac{N_1}{N_2}\right)^2 + \frac{1}{2g} W_s \left(\frac{D_N}{2}\right)^2 \left(\frac{N_1}{N_2}\right)^2 + \frac{W}{g}\left(\frac{\ell}{2\pi}\right)^2 \left(\frac{N_1}{N_2}\right)^2 \quad \text{M24}$$

= Motor inertia + Equivalent gear inertia + Ballscrew inertia + Load inertia (Fig.4.23)

W_S : Ballscrew weight (kgf)

D_N : Ballscrew nominal diameter (mm)

g : Gravity coefficient (9800 mm/sec^2)

J_M : Inertia of motor (kgf-mm-sec^2)

J_{G1} : Inertia of driver gear (kgf-mm-sec^2)

J_{G2} : Inertia of driver gear (kgf-mm-sec^2)

Total operating torque :

$$T_{Ma} = T_M + T'_a \quad \text{......} \quad \text{M25}$$

T_{Ma} = Total operating torque (kgf)

The inertia of a disc is calculated as following :
For disc with concentric O.D.

$$J = \frac{1}{2g} \pi \rho_d R^4 L \quad \text{......} \quad \text{M26}$$

J : Disc inertia (kgf • mm • sec²)
ρ_d : Disc specific weight (7.8 × 10^{-6} kgf/mm³) for steel
R : Disc radius (mm)
L_ℓ : Disc lenght (mm)
g : Gravity coefficient (9800 mm/sec²)

(d) Drive power

$$P_d = \frac{T_{p\max} \times N_{\max}}{974} \quad \text{......} \quad \text{M27}$$

P_d : Maximum drive power (watt) safety
T_{pmax} : Maximum drive torque (safety factor × T_{ma}, kgf-mm)
T_{max} : Maximum rotation speed (rpm)

(e) Check the acceleration time

$$t_a = \frac{J}{T_{M1} - T_L} \times \frac{2\pi N_{\max}}{60} \bullet f \quad \text{......} \quad \text{M28}$$

t_a = Acceleration rising time
J = Total inertia moment
$T_{M1} = 2 \times T_{mr}$
T_{Mr} = Motor rated torque
T_L = Drive torque at rated feed
f = Safety factor = 1.5

Table 4.9 : Shows the conversion relationship of different measurement units for the motor torque or preload drag torque.

Table 4.9 Conversion table for motor torque

kgf - cm	kgf - mm	Nm	kpm (kgf - m)	OZ - in	ft - l bf
1	10	9.8×10^{-2}	10^{-2}	13.8874	7.23301×10^{-2}
0.1	1	9.8×10^{-3}	1.0×10^{-3}	1.38874	7.23301×10^{-3}
10.19716	1.019716×10^{2}	1	0.1019716	1.41612×10^{2}	0.737562
10^{2}	10^{3}	9.80665	1	1.38874×10^{3}	7.23301
7.20077×10^{-2}	0.720077	7.06155×10^{-3}	7.20077×10^{-4}	1	5.20833×10^{3}
13.82548	1.382548×10^{2}	1.35582	0.1382548	1.92×10^{2}	1

Example 4.5 - 4

Consider the machining process driven by the motor and ballscrew as Fig. 4.24.
Table weight W_1 = 200 kgf
Work weight W_2 = 100 kgf
Friction coefficient of slider μ = 0.02

Operating condition : Smooth running without impact

Axial feed force (kgf)	Revolution (rpm)	Loading time ratio (%)
100	500	20
300	100	50
500	50	30

Acceleration speed : 100 rad/sec^2
Motor Condition : Motor diameter : 50 mm, Motor length : 200 mm,
Gear condition : Driver gear diameter G1 : 80 mm, Thickness : 20 mm, Teeth : 30
Driven gear diameter G2 : 240 mm, Thickness : 20 mm, Teeth : 90
Ballscrew condition :
Nominal diameter : 50 mm, Pitch : 10 mm
Length : 1200 mm, Weight : 18 kgf
No backlash when axial feed force = 300 kgf
Bearing torque T_b = 10 kgf-mm
Mechanical efficiency η_1 = 0.80

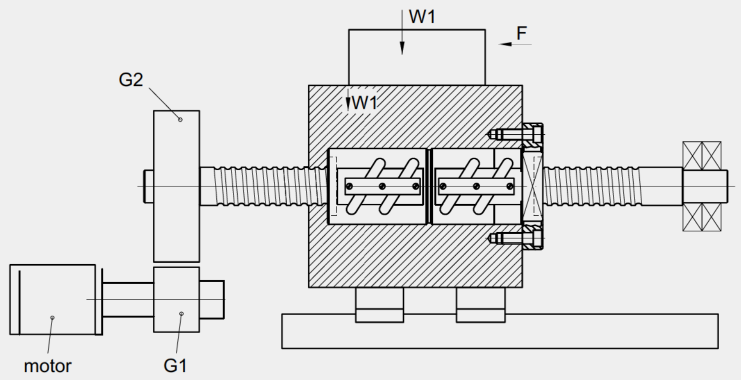


Fig 4.24 Milling process in the machine

Calculation
(1) Motor drive torque in normal rating condition :

$$n_{av} = 500 \times \frac{20}{100} + 100 \times \frac{20}{100} + 50 \times \frac{20}{100} = 165\ rpm \quad \text{(Ref. M7)}$$

$$F_1 = 100, F_2 = 300, F_3 = 500$$

$$F_{bm} = \sqrt[3]{100^3 \times 1 \times \frac{20}{100} \times \frac{500}{165} \times 300^3 \times 1 \times \frac{50}{100} \times \frac{100}{165} + 500^3 \times 1 \times \frac{30}{100} \times \frac{50}{165}} = 272\ kgf \quad \text{(Ref. M9)}$$

$$P = \frac{300}{2.8} \fallingdotseq 110\ \text{kgf (axial feed force} = 300\ kgf\text{, Ref. M1)}$$

$$F_b = F_{bm} + \mu W = 270 + (200 + 100) \times 0.02 = 278\ kgf$$

$$T_a = \frac{F_b \times \ell}{2\pi\eta_1} = \frac{278 \times 10}{2\pi \times 0.80} = 553\ kgf \bullet mm \quad \text{(Ref. M19)}$$

$$T_d = 0.2 \times \frac{P \times \ell}{2\pi} = \frac{0.2 \times 110 \times 10}{2\pi} = 35\ kgf \bullet mm \quad \text{(Ref. M2)}$$

$$T_M = (T_a + T_b + T_d) \times \frac{N_1}{N_2} = (535 + 10 + 35) \times \frac{30}{90} = 199\ kgf \bullet mm \quad \text{(Ref. M21)}$$

(2) Motor torque in acceleration operation :

(I) Inertia of motor

$$J_M = \frac{1}{2 \times 9800} \times \pi \times 7.8 \times 10^{-6} \times (25)^4 \times 200 = 0.1\ kgf \bullet mm \bullet \sec^2$$

(II) Inertia of gear

$$J_{Gear(eq)} = J_{G1} + J_{G2} \times \left(\frac{N_1}{N_2}\right)^2$$

$$J_{G1} = \frac{1}{2 \times 9800} \times \pi \times 7.8 \times 10^{-6} \times \left(\frac{80}{2}\right)^4 \times 20 = 0.064\ kgf \bullet mm \bullet \sec^2$$

$$J_{G2} = \frac{1}{2 \times 9800} \times \pi \times 7.8 \times 10^{-6} \times \left(\frac{240}{2}\right)^4 \times 20 = 5.18\ kgf \bullet mm \bullet \sec^2$$

$$J_{Gear(eq)} = 0.064 + 5.18 \times \left(\frac{30}{90}\right)^2 = 0.640\ kgf \bullet mm \bullet \sec^2$$

(III) Inertia of ballscrew

$$J_{ballscrew} = \frac{1}{2 \times 9800} \times 18 \times \left(\frac{50}{2}\right)^2 \left(\frac{30}{90}\right)^2 = 0.064\ kgf \bullet mm \bullet \sec^2$$

(IV) Inertia of load

$$J_{load} = \frac{300}{9800} \times \left(\frac{10}{2 \times \pi}\right)^2 \times \left(\frac{30}{90}\right)^2 = 0.009\ kgf \bullet mm \bullet \sec^2$$

(V) Total inertia

$$J = 0.1 + 0.64 + 0.064 + 0.009 = 0.813\ kgf \bullet mm \bullet \sec^2$$

(3) Total motor torque:

$$T'_a = J \bullet \alpha = 0.813 \times 100 = 81.3\ kgf \bullet mm$$

$$T_{Ma} = T_M + T'_a = 199 + 81.3 = 280\ kgf \bullet mm$$

(4) Drive power:

$$T_{p\max} = 2 \times 280 = 560\ kgf \bullet mm \quad \text{(safety factor = 2)}$$

$$P_d = \frac{560 \times 1500}{974} = 862\ \text{W} = 1.16\ Hp$$

(5) Selection motor:

Select the DC motor rated torque : $T_{Mr} > 1.5T_M$, and maximum motor torque : $T_{Max} > 1.5T_{pmax}$

Thus the DC servo motor with following specification can be chosen.

Rated output : 950 w

Rated torque : 30 kgf-cm (300 kgf • mm)

Rated rotational speed : 2000 rpm

Maximum torque : 65 kgf x cm (650 kgf • mm)

Moment of inertia of motor : 0.20 kgf • mm • sec^2

(6) Check the acceleration time:

$$T_L = \left(\frac{F_a \times \ell}{2\pi\eta_1} + T_b + T_d\right) \times \frac{N_1}{N_2} = \left(\frac{100 \times 10}{2\pi \times 0.8} + 10 + 35\right) \times \frac{30}{90} = 81.3\ kgf \bullet mm$$

$$t_a \geq \left(\frac{0.879}{300 \times 2 - 81.3}\right) \times \frac{2\pi \times 2000}{60} \times 1.5 = 0.53\ \sec$$

Buckling load

The ballscrew shaft when subjected to an axial compressive force may be undergo a visibly large deflection. The axial force is called the buckling load.

$$F_k = 40720\left(\frac{N_f d_r^4}{L_t^2}\right) \quad \text{M29}$$

$$F_p = 0.5\,F_k \quad \text{M30}$$

F_k = Permissible load (kgf)	fixed - fixed	N_f = 1.0
F_p : Maximum permissible speed (kgf)	fixed - supported	N_f = 0.5
d_r : Root diameter of screw shaft (mm)	supported - supported	N_f = 0.25
L_t : distance between support bearing (mm)	fixed - free	N_f = 0.0625
N_f : Factor for different mounting types	♦1kgf = 9.8N;1daN=10N	

The buckling load diagram for different spindle diameter and support method is shown in Fig 4.25.

Critical speed

The critical speed is said to exist when the rotational frequency of a shaft equals the first natural frequency of the shaft. This will cause the ball screw to bend under the stress of vibration coupled with the centrifugal forces due to the rotation and cause the shaft to vibrate violently. Therefore, the rotational speed of the ball screw should be set to below the value indicated by critical speed.

$$N_c = 2.71 \times 10^8 \times \frac{M_f d_r}{L_t^2} \quad \text{M31}$$

$$N_p = 0.8 N_c \quad \text{M32}$$

N_c = critical speed (rpm)	fixed - fixed	M_f = 1
N_p = Maximum permissible load (rpm)	fixed - supported	M_f = 0.689
d_r : Root diameter of screw shaft (mm)	supported - supported	M_f = 0.441
L_t : distance between support bearing (mm)	fixed - free	M_f = 0.157
M_f : Factor for different mounting types		

The critical speed for different spindle and support method is shown in (Fig 4.26).

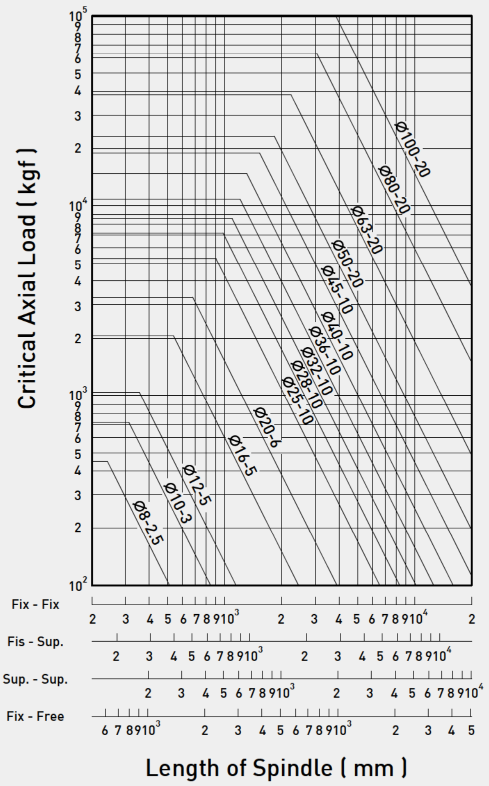


Fig 4.25 Shows the buckling load for different screw spindle diameter and length

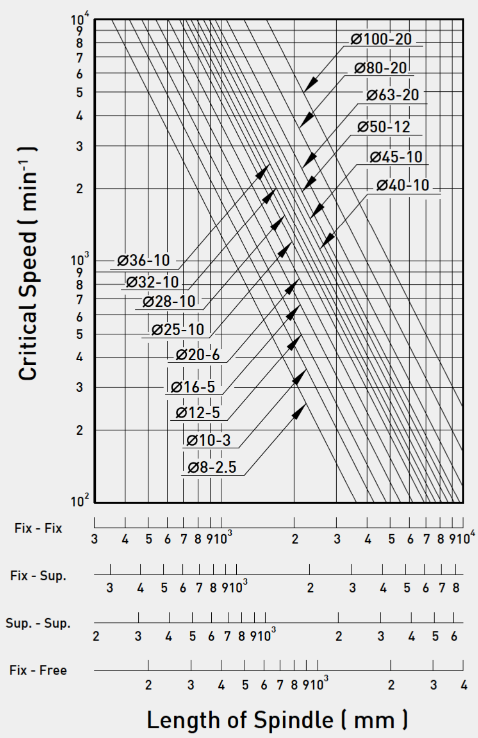
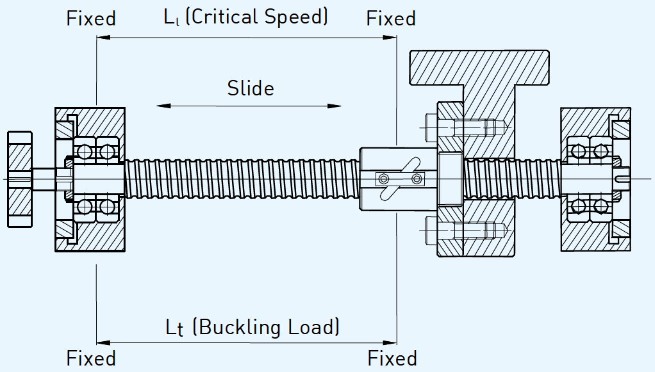


Fig 4.26 shows the critical speed for different screw spindle diameter and length

Supporting Conditions for Calculation of Buckling Load and Critical Speed

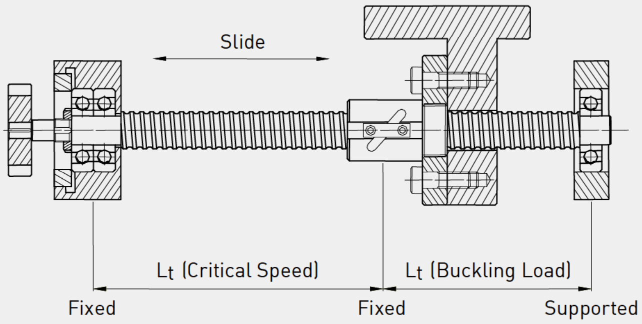
1

Critical Speed: fixed-fixed
Buckling Load: fixed-fixed



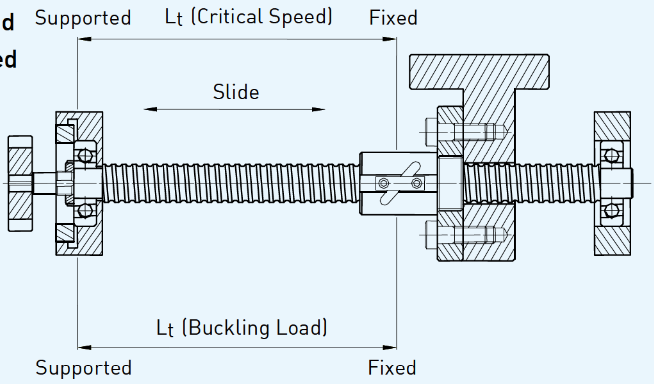
2

Critical Speed: fixed-supported
Buckling Load: fixed-fixed



3

Critical Speed: fixed-supported
Buckling Load: fixed-supported



4

Critical Speed: fixed-free
Buckling Load: fixed-fixed

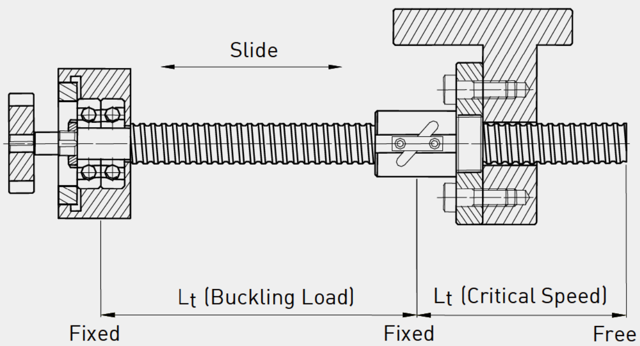


Fig 4.27 Supporting conditions for screw shaft ball nut

D_m-N value for ballscrew surface speed

D_m-N value has a strong influence over ballscrew noise, working temperature and service life of return system. For HIWIN ballscrew,

$D_m \times N \leq 70{,}000$ ······ M33

D_m : Pitch circle diameter (mm)

N : Maximum speed (rpm)

Ballscrew structure enhancement designed by HIWIN when Dm-N value ranges from 70,000 to 180,000 . If D_m-N value above 180,000 , please consult our company.

Stiffness

Stiffness is an indication of the rigidity of a machine. The stiffness of the ballscrew is determined by nut-spindle rigidity via axial load, balltrack contact rigidity and screw spindle rigidity. When assembling the ballscrew in the machine, the stiffness of support bearing, mounting condition of nut with machine table etc. also should be considered. Fig 4.28 shows the relation of total stiffness of the machine feed system.

From testing, the stiffness of nut-spindle relation and ball and balltrack relation can be combined into the stiffness of nut, Kn, and listed in dimension table of different nut type. The stiffness of the ballscrew is shown as :

$$\frac{1}{K_{bs}} = \frac{1}{K_s} + \frac{1}{K_n}$$ ······ M34

K_{bs} : Total stiffness of ballscrew (kgf/μm)

The stiffness of the screw spindle is shown as :

$$K_s = 67.4 \frac{d_r^2}{L_1} \quad \text{(Fixed-Fixed)}$$ ······ M35

$$K_s = 16.8 \frac{d_r^2}{L_1} \quad \text{(Fixed-Free)}$$ ······ M36

The stiffness chart is shown in Fig 4.29

d_r : Root diameter of screw spindle (mm) $\fallingdotseq D_m - D_b$ ······ M37

D_b : Diameter of ball (mm)

K_s : Screw spindle stiffness (kgf/μm)

K_n : Nut stiffness (kgf/μm)

The stiffness of the nut is tested using an axial force equal to the highest possible preload of 10% dynamic load (C) and is shown in the dimension table of each nut. When the preload is less than this value, the stiffness of the nut is calculated by extrapolation method as :

$$K_n = 0.8 \times K \left(\frac{P}{0.1C} \right)^{1/3}$$ ······ M38

K_n : Stiffness of nut

K : Stiffness in the dimension table

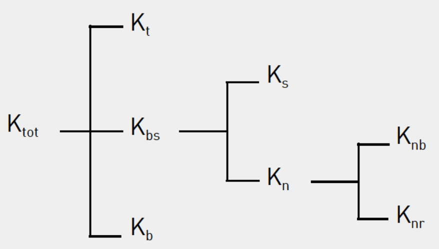
P : Preload

C : Dynamic load on dimension table

Single nut with backlash is calculated when the external axial force is equal to 0.28 C, thus :

$$K_n = 0.8 \times K \left(\frac{F_b}{2.8 \times 0.1C} \right)^{1/3}$$ ······ M39

The axial stiffness of the whole feed system includes the stiffness of support bearings and nut mounting table. The designer should consider the total stiffness carefully.



K_{tot} : Total stiffness of machine feed system
K_t : Table mounting stiffness
K_b : Support bearing stiffness
K_{bs} : Ballscrew stiffness
K_s : Ballscrew spindle stiffness
K_n : Ballscrew nut stiffness
K_{nb} : Ball and balltrack stiffness
K_{nr} : Nut-spindle stiffness by radial load

Fig 4.28 Stiffness distribution for ballscrew feed system

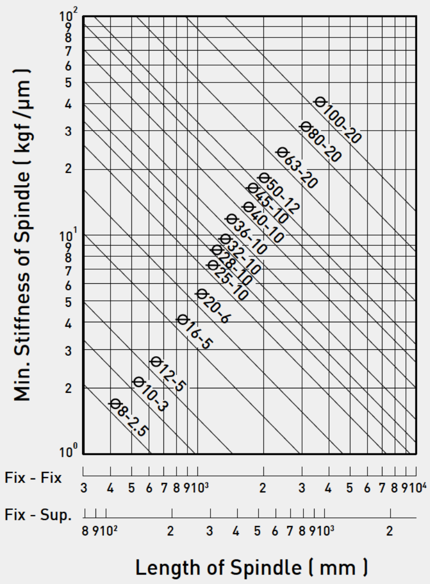


Fig 4.29 Stiffness chart for ballscrew spindle

Thermal expansion

$\Delta L = 11.6 \times 10^{-6} \times \Delta T \times L_s$ M40

ΔL : Thermal expansion of screw spindle (mm)

ΔL : (°C) Temperature rise at screw spindle

L_s : Total length of screw spindle (mm)

The T value should be chosen to compensate for the temperature rise of the ballscrew.
HIWIN recommends a T value of -0.02 ~ -0.03 per meter for CNC machine tools.

Basic dynamic axial load rating C (theoretical)

The dynamic load is the load at which 90% of the ballscrews will achieve the service life of 1 x 10^6 rev (C). The reliability factor can be adjusted by Table 4.8. The dynamic load is shown on the dimension table of each nut type.

Basic static axial load rating Co (theoretical)

The static load is the load which will cause the balltrack to have a plastic deformation exceeding 0.0001x ball diameter. To calculate the maximum static load of a ballscrew, the static safety factor S_f of the application condition should be considered.

$S_f \times F_a(\max) < Co$ M41

S_f : Static factor = 2.5 max

Co : Static load from the dimension table of the nut type

$F_a(\max)$: Maximum static axial load

Example 4.5 - 5

Ballscrew specification: 1R40-10B2-FSW-1000-1200-0.012
Pitch circle diameter D_m = 41.4 mm
Ball diameter : 6.35 mm
Root diameter d_r = 34.91 mm
Column load : fixed - supported
Critical speed : fixed - supported
Stiffness of bearing K_b = 105 kgf/μm

Lead ℓ =10 mm
Turns = 2.5x2
Lead angle α = 4.4°
Friction angle β = 0.286°
Preload P = 250 kgf
Mean axial force F_b = 700 kgf
N_f =0.5 ; L_t =1000 mm ; M_f = 0.692

Calculation

(1) Buckling load F_p

$$F_k = 40720 \times \frac{N_f d_r^4}{L_t^2} = 40720 \times \frac{0.5 \times 34.91^4}{1000^2} = 30240\ kgf \ \text{(Ref. M29)}$$

$$F_p = 0.5 \times \text{Fk} = 0.5 \times 30240 = 15120 \text{ kgf}$$

(2) Critical speed N_p

$$N_c = 2.71 \times 10^8 \times \frac{0.689 \times 34.90}{1000^2} = 6516\ rpm$$

$$N_p = 0.8 \times N_c = 0.8 \times 6516 = 5213\ rpm$$

(3) Mechanical efficiency η (theoretical)

(I) Common transmission

$$\eta_1 = \frac{\tan\alpha}{\tan(\alpha+\beta)} = \frac{\tan(4.396^\circ)}{\tan(4.396^\circ + 0.286^\circ)} = 0.938 \ \text{(Ref. M3)}$$

(II) Reverse transmission

$$\eta_2 = \frac{\tan(\alpha+\beta)}{\tan\alpha} = \frac{\tan(4.396^\circ + 0.286^\circ)}{\tan(4.396^\circ)} = 0.934 \ \text{(Ref. M4)}$$

(4) Stiffness K

$$K_s = 16.8 \frac{d_r^2}{L_1} = 16.8 \times \frac{34.91^2}{1000} = 20.5\ kgf/\mu m \qquad p = 250 < 0.1\text{C}(=537)$$

$$\therefore K_n = 0.8 \times \left(\frac{P}{0.1C}\right)^{1/3} = 0.8 \times 74 \times \left(\frac{250}{0.1 \times 5370}\right)^{1/3} = 46 \text{ kgf}/\mu m$$

$$\frac{1}{K} = \frac{1}{K_s} + \frac{1}{K_n} = \frac{1}{20.5} + \frac{1}{46} \qquad K = 14.18 \text{ kgf}/\mu m$$

(5) Lost motion during axial force F_b = 700kgf

$$\frac{1}{K_t} = \frac{1}{K} + \frac{1}{K_b} = \frac{1}{14} + \frac{1}{105} \qquad K_t = 12.35 \text{ kgf}/\mu m$$

$$\delta/2 = \frac{F}{K} = \frac{700}{12.4} = 56\ \mu m = 0.056\ mm$$ (each way) Total lost motion δ =2x0.056=0.112 mm

If the preload increases to 2x250=500 kgf then K_n=58 kgf/μm and K=15.1 kgf/μm. Total stiffness K_t=13.2 kgf/μm and total lost motion δ =0.106 mm. The difference is only 6 μm (5% change). comparing with 250 kgf, preloaded nut, but the temperature rise caused by 500 kgf preload is heavy. The spindle stiffness is sometimes more important than the nut stiffness. The best way to increase the stiffness of the system is not in the heavy preloading of the ballscrew nut. If the support method changes to fixed-fixed, then K_s=82 kgf/μm and K_t becomes 23 kgf/μm. The total lost motion d=0.061 mm. The difference is 51μm (45%).

Manufacturing range

The maximum length to which a ballscrew can be manufactured depends on spindle diameter and accuracy grade (Table 4.10). Since high accuracy ballscrews require a high degree of straightness to the screw spindle, the higher the slender ratio (length/diameter), the more difficult to manufacture and the less the spindle stiffness.

HIWIN recommends the maximum lengths shown in Table 4.10.

If a longer length is required, please contact with HIWIN engineer.

Table 4.10 General manufacturing range of HIWIN screw spindle vs. diameter and accuracy grade Unit : mm

Total length \ O.D. / Grade	6	8	10	12	16	20	25	28	32	36	40	45	50	55	63	70	80	100
C0	110	170	300	400	600	700	1000	1000	1200	1300	1500	1600	1800	2000	2000	2000	2000	2000
C1	110	170	400	500	720	950	1300	1500	1800	1800	2300	2500	3100	3500	4000	4000	4000	4000
C2	140	200	500	630	900	1300	1700	1800	2200	2200	2900	3200	4000	5000	5200	5500	6300	6300
C3	170	250	500	630	1000	1400	1800	2000	2500	3200	3500	4000	4500	5000	6000	7100	10000	10000
C4	170	250	500	630	1000	1400	1800	2000	2500	3200	3500	4000	4500	5000	6000	7100	10000	10000
C5	170	250	500	630	1410	1700	2400	2500	3000	3200	3800	4000	5000	5500	6900	7100	10000	10000
C6	400	800	1000	1200	1500	1800	2500	3000	3000	4000	4000	4000	5600	5600	6900	7100	10000	10000
C7	400	800	1000	1200	3000	3000	4000	4000	4500	4500	5600	5600	5600	5600	6900	7100	10000	10000

■ Please consult with HIWIN in this area

Heat treatment

HIWIN's homogenous heat treatment technique gives the ballscrew maximum life capability. Table 4.11 shows the hardness value of hardness in each component of HIWIN ballscrews. The surface hardness of the ballscrew affects both dynamic and static load value. The dynamic and static values shown in the dimension table are the values for a surface hardness equal to HRC 60. If the surface hardness is lower than this value, the following formula will give you the calibration result.

$$C'o = Co \times f_{HO} \qquad f_{HO} = \left(\frac{Real\ Hardness\ (HRC)}{60} \right)^3 \leq 1 \quad \text{..........} \quad \text{M42}$$

$$C' = C \times f_H \qquad f_H = \left(\frac{Real\ Hardness\ (HRC)}{60} \right)^2 \leq 1 \quad \text{..........} \quad \text{M43}$$

Where f_H and f_{HO} are the hardness factor.

$C'o$: Calibrated static load
Co : Static load
C' : Calibrated dynamic load
C : Dynamic load

Table 4.11 Hardness of each component of HIWIN ballscrew

Item	Treat Method	Hardness (HRC)
Spindle	Carburizing or Induction Hardening	58 - 62
Nut	Carburizing	58 - 62
Ball		62 - 66

4.6 Temperature Rise Effect on Ballscrews

The temperature rise of ballscrew during the working period will influence the accuracy of the machine feed system, especially in a machine designed for high speed and high accuracy.

The following factors have the effect of raising the temperature in a ballscrew.

(1) Preload (2) Lubrication (3) Pretension

Fig 4.30 shows the relation of working speed, preload and temperature rise. Fig 4.31 shows the relation of nut temperature rise to preload friction torque. From Fig 4.30, Fig 4.31 and example 4.5-5, doubling the preload of the nut will increase the temperature about 5 degrees, but the stiffness increase only by about 5% (few μm).

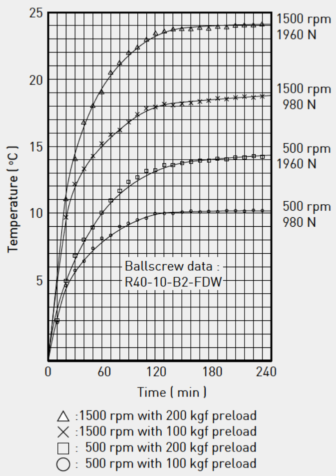


Fig 4.30 The relation of working speed, preload and temperature rise

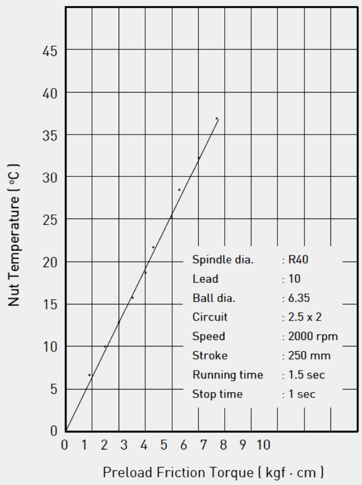


Fig 4.31 The relation of nut temperature rise to preload friction torque

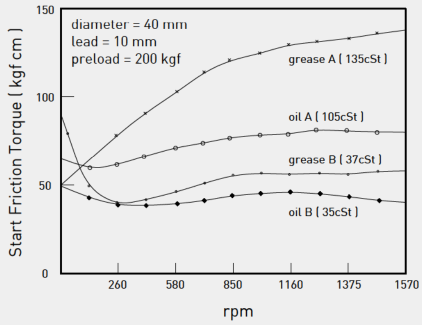


Fig 4.32 The influence of oil viscosity on the friction torque

(1) Preload effect

To avoid any lost motion in the machine feed system, increasing the rigidity of the ballscrew nut is important. However, to increase the rigidity of the ballscrew nut, it is necessary to preload the nut to a certain level.

Preloading the nut will increase the friction torque of the screw, making it more sensitive to an increase in temperature during working period.

HIWIN recommends using a preload of 8% of the dynamic load for medium and heavy preload, 6% ~ 8% for medium preload, 4% ~ 6% for light and medium and below 4% for light preload.

The heaviest preload should not exceed 10% of the dynamic load for best service life and a low temperature rise effect.

(2) Lubrication effect

The selection of lubricant will directly influence the temperature rise of the ballscrew.

HIWIN ballscrews require appropriate lubrication either by greasing or oiling. Antifriction bearing oil is recommended for ballscrew oil lubrication. Lithium soap based grease is recommended for ballscrew greasing. The basic oil viscosity requirement depends on the speed, working temperature and load condition of the application. (Fig 4.32) shows the relation of oil viscosity, working speed and rise in temperature.

When the working speed is higher and the working load is lower, a low viscosity oil is better. When the working speed is lower and the working load is heavy, a high viscosity oil is preferred.

Generally speaking, oil with a viscosity of 32 ~ 68 cSt at 40°C (ISO VG 32-68) is recommended for high speed lubrication (DIN 51519) and viscosity above 90 cSt at 40°C (ISO VG 90) is recommended for low speed lubrication.

In high speed and heavy load applications the use of a forced coolant is necessary to lessen the temperature. The forced lubrication of coolant can be done by a hollow ballscrew.

Fig 4.33 shows the comparison of a ballscrew applied with coolant and without coolant. Fig 4.34 shows a typical application for hollow ballscrew in machine tools. The inspection and replenishing of the ballscrew lubricant is listed in Table 4.12.

(3) Pretension effect

When the temperature rises in the ballscrew, the effect of thermal stress will elongate the screw spindle. It can make the spindle length unstable.Ø

The elongating relationship can be calculated according to M41. This elongation can be compensated via the pretension force. For the purpose of pretension, there is a negative T value indicated in the design drawing to compensate the pretension value.

Since a large pretension force will cause the burn down of the supporting bearing, HIWIN recommends using pretension when the temperature rise is below 5°C. Also, if the diameter of the screw spindle is greater than 50 mm, it is not suitable for pretension. A large spindle diameter requires a high pretension force, causing burn down of the supporting bearing.

HIWIN recommends a T compensation value of about 3°, (about -0.02~0.03 for each 1000 mm screw spindle).

Since different applications require different T values, please contact HIWIN engineer.

The pretension force is calculated as :

$P_f = K_s \times \Delta L$

K_s : Stiffness of screw spindle (kgf/μm)

P_f : Pretension force (kgf)

ΔL : Pretension valus (μm)

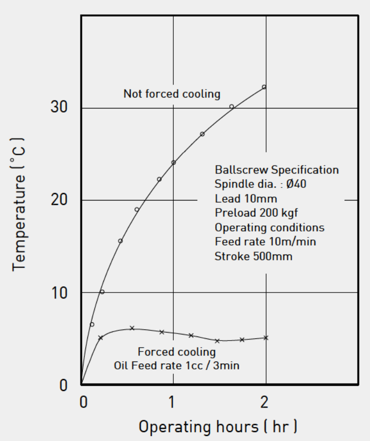


Fig 4.33 Ballscrew temperature rise with the coolant and without the coolant

Table 4.12 : Inspection and replenishment of Lubricant

Lubrication Method	Inspection & Replenishment Guide
Oil	• Check the oil level and clean the contamination once a week. • When contamination happens, replacing the oil is recommended. • Lubrication suggestion : Lubrication amount apply onto Ballscrew per 15 minute $\frac{\text{Ballscrew outer diameter(mm)}}{56\sim60}$ c.c.
Grease	• Inspect for contamination of chips every 2 or 3 months. • If contamination happens, remove old grease and replace with new grease. • Injection amount is about half of internal space within nut every 2 months or 100 km stroke.

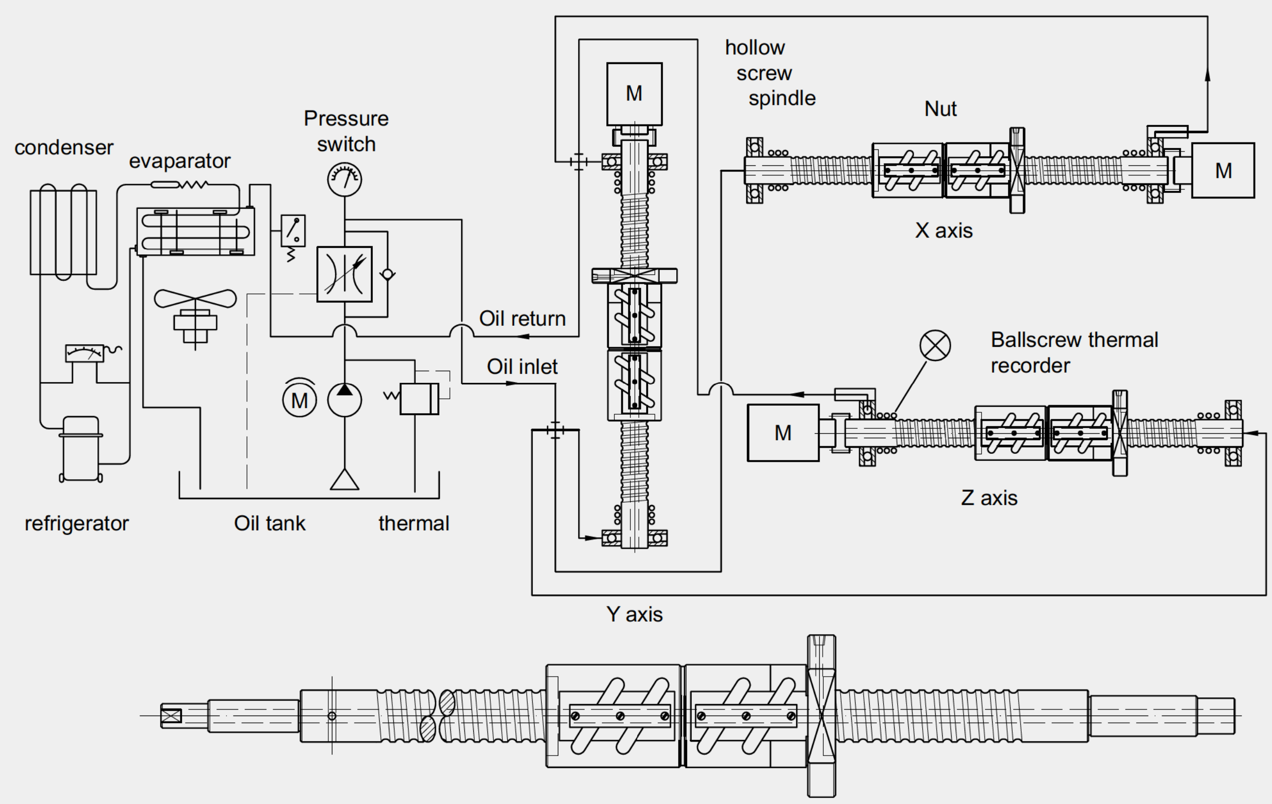


Fig 4.34 High accuracy machine tools with hollow ballscrew lubrication

5 Specification Illustration

HIWIN manufactures ballscrews according to customers' blueprints or specifications. Please read the following information for understanding out ballscrew designing.

1. Nominal diameter.
2. Thread lead.
3. Thread length, total length.
4. End journal configuration.
5. Nut configuration
6. Accuracy grade (lead deviation, geometrical tolerance).
7. Working speed.
8. Maximum static load, working load, preload drag torque.
9. Nut safety requirements.
10. Lubrication hole position.

HIWIN Ballscrew Nomenclature

HIWIN ballscrews can be specified as follows :

1R40 - 10B2 - PFDWE2 - 800 - 1000 - 0.0035 - M

Start type
1.Single start
2.Double start
3.Triple start
4.Four start
5:Five Start

Right hand screw

Nominal diameter

Lead

Number of turns

Preload type
P : Compression type
O : Offset type
D : High lead double start
T : High lead triple start
Q : High lead quatemary start
V : High lead five start

Nut shape
S : Square nut
R : Round
F : Nut with flange

Note :
M : Stainless
H : Hollow Shaft
L : heavy load

Lead deviation in random 300mm travel path thread length

Total length

Thread length

Optional Functions :
E2 : Self-lubrication
R1 : Rotating Nut
C1,C2 : Cool Type

Circulation type
W : Tubes within nut body
V : Tubes above nut body
B : Bonded tube
I : Internal cap
H : End cap
C : Super S series

Nut type
S : Single nut
D : Double nut

Number of turns

A : 1.5, B: 2.5, C: 3.5	T3 : 3	S1 : 1.8x1	U1 : 2.8x1	K2 : 2
A2 : 1.5x2	T4 : 4	S2 : 1.8x2	U2 : 2.8x2	K3 : 3
B2 : 2.5x2	T5 : 5	S4 : 1.8x4	V2 : 0.8x2	K4 : 4
C1 : 3.5x1	T6 : 6			

Note : 1. Different diameters and leads are available upon request.
2. Right hand thread is standard, left hand thread is available upon request.
3. Longer lengths are available upon request.
4. Stainless steel is available upon request, only if the ball size is less than 2.381 mm.
5. Complete questionnaire on page 173~174 and consult with HIWIN engineers.
6. If you need to order DIN 69051 type, please mark "DIN".
7. Number of turns = turns per circuit x number of circuits.
Please refer to page 6 for detailed illustration.