1. Product overview

Rolled ballscrews

Page 34

- Flange nuts and cylindrical nuts
- Reduced axial play available
- Nominal diameter 8 – 63 mm
- Standardised end machining

Peeled ballscrews

Page 39

- Flange nuts and cylindrical nuts
- Single and double nuts
- Nominal diameter 16 – 80 mm
- Standardised end machining
- Safety nuts

Ground ballscrews

Page 48

- Flange nuts and cylindrical nuts
- Single and double nuts
- Nominal diameter 6 – 100 mm
- Preloaded or minimum axial play

Ballscrews for special requirements

Page 60

- Driven nut unit
- Ballscrews for heavy-duty operation

Shaft ends and accessories

Page 62

- Standard end machining
- Standard spindle bearings
- Nut housing

Ball Spline

Page 88

- Flange nuts with integrated bearing
- Combination of ball spline nut and ballscrew nut
- Ball spline as full or hollow shaft
- Nominal diameter 16-32 mm

2. General information

2.1 Properties
There are many benefits associated with HIWIN ballscrews including high efficiency, freedom from axial play, high rigidity and high lead accuracy. The characteristic properties and benefits of HIWIN ballscrews are described in detail below.

2.1.1 High efficiency in both directions
Thanks to the rolling contact between the shaft and nut, ballscrews can achieve an efficiency of up to 90 %.
The special surface treatment used on the ball tracks in HIWIN ballscrews reduces the frictional resistance between the ball and track.
The rolling motion of the balls only requires a low drive torque thanks to the high level of efficiency. Operating costs are therefore cut since less drive output is needed.

1	Linear to rotary motion
2	Ballscrews Rotary to linear motion
3	Ballscrews
μ	Efficiency [%]
φ	Lead angle [°]
φ_N	Lead angle for common transmission [°]
φ_U	Lead angle for reverse transmission [°]

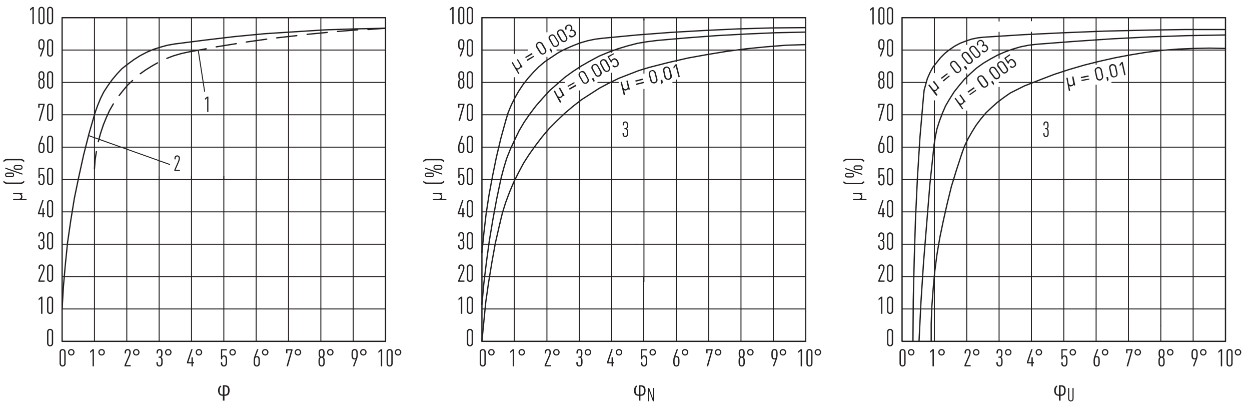
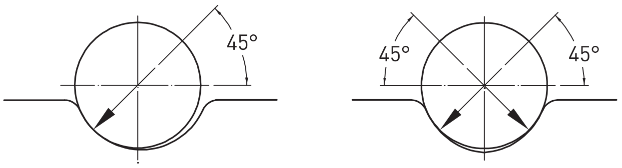


Fig. 2.1 **Mechanical efficiency of threaded shafts**

2.1.2 Zero play and high rigidity
The pointed profile HIWIN uses for ballscrew shafts and nuts allows the ballscrew nuts to be assembled without any axial play. A preload is usually used to achieve the good overall rigidity and repeatability.



Semi circular profile   Pointed profile (Gothic type)

Fig. 2.2 **Typical types of contact in ballscrews (semi circular type, Gothic type)**

2.1.3 High lead accuracy
For applications requiring very high levels of accuracy, our production meets the requirements of ISO and JIS standards; but we manufacture to customer specifications too.
Accuracy is guaranteed by testing with our laser measurement systems and documented for the customer.

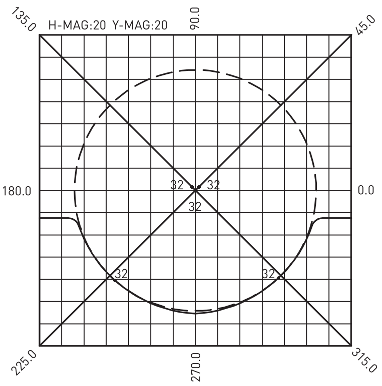
2.1.4 Reliable service life

Whereas the life of standard screw drives is determined by wear on the contact surfaces, HIWIN ballscrews can be used virtually up until the end of the metal's fatigue life. Great care is exercised in development, choice of material, heat treatment and manufacturing, as is demonstrated by the reliability and resilience of HIWIN ballscrews over their nominal service life. With every kind of ballscrew, the service life depends on several influencing factors including design aspects, material quality, maintenance and most importantly the dynamic load rating (C_{dyn}).
Profile accuracy, material properties and surface hardness are the fundamental factors affecting the dynamic load rating.

2.1.5 Low starting torque with smooth operation

The rolling friction of the balls in ballscrews only requires a very low starting torque. To achieve precise ball tracks, HIWIN uses a special design (adaptation factor) and special production procedures. This guarantees that the motor's drive torque remains in the range required.
In one particular step of manufacturing, HIWIN can check the profile of every single ball track. A sample report of this test is shown in Fig. 2.3.

Using computer-based measuring systems, the friction torque of every ballscrew is recorded and documented with great accuracy at HIWIN. Fig. 2.4 shows typical torque progress over travel.

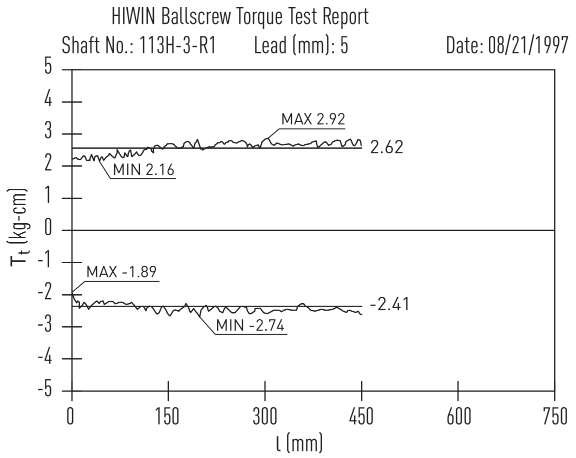


Work name:	SH	Measuring method:	X lead
		Pick up radius:	0.0256 mm
Model No.:	001H-2-3	Horizontal mag:	20.0000
Lot No.:	201536	Vertical mag:	20.0000
Operator:	L. J. F.	Measuring path:	7.0000 mm
Comment:		A:	0.0030 mm

No. of current code symbol

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	RC: −42.5259 mm
32	292 X:	2.1799 mm	Z: −2.3084 mm	RC: 43.3615 mm
32	292 X:	−0.0000 mm	Z: −0.0000 mm	RC: 3.1750 mm

Fig. 2.3 **Ball arch profile testing at HIWIN**



L Distance [mm]
T_C Torque [kg-cm]

Fig. 2.4 **Preload testing at HIWIN**

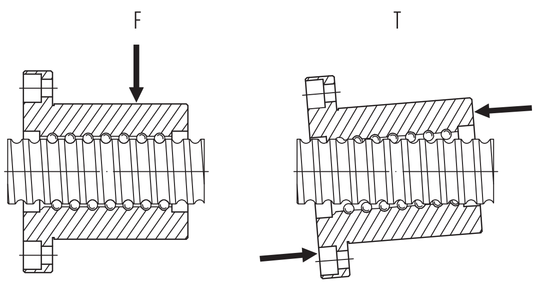
2.1.6 Special solutions

HIWIN manufactures ballscrews in line with customer drawings or with HIWIN standard end machining. For the ballscrew definition the requirements on the project planning sheet must be documented and checked. This ensures that the ballscrew is ideally adapted to the requirements in place.

3. Structural properties and selection of HIWIN ballscrews

3.1 Design information

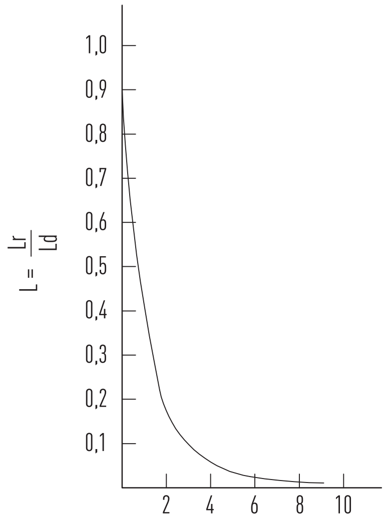
a) Select a suitable ballscrew for your application (see Table 3.6). The relevant requirements must be noted for installation. For precision-ground ballscrews with CNC machines, this means careful alignment and the corresponding type of installation; for applications requiring less precision, we recommend rolled ballscrews, which require less work when designing the type of installation and bearings.



F Radial load
T Moment load

Fig. 3.1 **Uneven load distribution,** caused by insufficient alignment of support bearing and ballscrew nut, incorrect configuration of mounting surface, incorrect angle or error in aligning the nut flange

b) It is particularly important that the bearing housing and ballscrew nut are assembled axially parallel; otherwise uneven load distribution would result. Radial and torque loads are also among the factors which result in uneven load distribution (see Fig. 3.1). This can cause functional limitations and shorten the service life (see Fig. 3.2).



Ball nut – FSWXB2

Specifications:
Shaft diameter: 40 mm
Lead: 10 mm
Ball diameter: 6.35 mm
Radiales play: 0.05 mm

Conditions:
Axial force Fa: 3000 N
Radial displacement: 0 mm

L Service life ratio
Lr realservice life
Ld desired service life
$\delta\alpha$ Assembly inclination

Fig. 3.2 **Impacts on life expectancy** of radial load caused by insufficient alignment

c) Select the right type of bearing for the ballscrew shaft. When used in CNC machines, we recommend angular ball bearings (angle = 60°) because of their higher axial load capacity and the fact that they permit zero-backlash or pre-loaded installation.
A selection of possible end machining processes and suitable floating and fixed bearings are listed in Chapter 8 onwards.

d) Precautionary measures must be taken to stop the ballscrew nut once the useful path has been exceeded (see Fig. 3.3). Travel against an axial fixed stop results in damage.

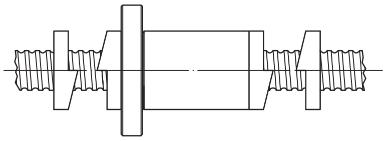


Fig. 3.3 **Mechanical stop which prevents the travel distance from being exceeded**

e) In environments with high levels of dust or metal debris, ballscrews should be provided with a telescopic or bellows shaft protection (see Fig. 3.4).

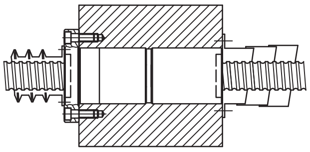


Fig. 3.4 **Telescopic or bellows shaft protection**

f) When using an internal or end cap ball recirculation system, the ball thread must be cut to the end of the shaft. The diameter of the adjacent bearing journal must be around 0.5 –1.0 mm less than the core diameter of the ball tracks d_r (see Fig. 3.5).

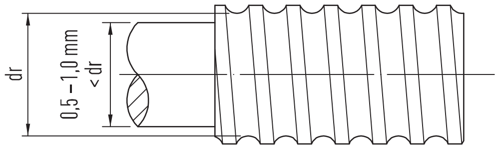


Fig. 3.5 **Special requirement of bearing journal with internal recirculation system**

g) While surface-hardening the shafts, 2 to 3 thread turns are left unhardened on the two ends adjacent to the bearings so that connection modifications are possible. These areas are marked with the symbol in HIWIN drawings (see Fig. 3.6). Please contact HIWIN if you have special requirements for these areas.

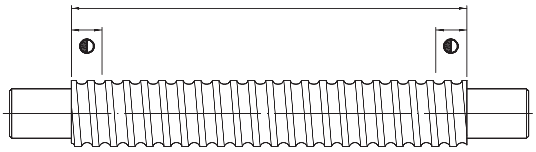


Fig. 3.6 **Area of surface hardening on a ballscrew shaft**

h) Excess preload results in increased friction torque which in turn causes heating and therefore a reduced service life. On the other hand, insufficient preload reduces rigidity and increases the risk of backlash. For details, see Sections 3.7 and 3.8.7.

3.2 Procedure for selecting a ballscrew

Table 3.1 shows the procedure for selecting a ballscrew. The usage requirements (A) can be used to determine the necessary ballscrew parameters (B). The ballscrew suited to the application can therefore be determined one step at a time following the information provided (C).

Table 3.1 **Procedure for selecting a ballscrew**

Step	Usage requirement (A)	Ballscrew parameter (B)	Reference (C)
1	Positioning accuracy	Lead accuracy	Table 4.1, Table 5.1, Table 6.1
2	Speed	Lead of screw drive	$p = \frac{v_{max}}{n_{max}}$
3	Total length of travel distance	Total length of thread	Total length = thread length + length of end machining Thread length = travel distance + length of nut + distance which cannot be used due to connection design (e.g. nut housing, bearing housing etc.)
4	1 Load conditions [%] 2 Speed conditions [%] (≤ 1/5 C recommended)	Average axial load Average speed	Formulas F 3.4 – F 3.9
5	Average axial force	Preload	Formula F 3.5
6	1 Nominal service life 2 Average axial load 3 Average speed	Dynamic load rating	Section 3.8.2, "Service life"
7	1 Dynamic load rating 2 Lead of ballscrew 3 Critical speed 4 Speed limitation by D_N value	Shaft diameter and nut type	Section 3.8.2, "Service life"
8	1 Diameter of ballscrew 2 Nut type 3 Preload 4 Dynamic load rating	Rigidity	Section 3.8.7, "Rigidity"
9	1 Ambient temperature 2 Length of ballscrew	Thermal deformation and final value of cumulative lead (T)	Section 3.8.8, "Thermal expansion"
10	1 Shaft rigidity 2 Thermal deformation	Preload	Section 3.8.8, "Thermal expansion"
11	1 Max. table speed 2 Max. start-up time 3 Configuration of ballscrew	Motor drive torque and configuration of motor	Section 3.8.3, "Drive torque and drive output of motor"

3.3 Ballscrew shafts

HIWIN offers rolled, peeled and ground ballscrews – depending on the application requirements. For the selection of the appropriate shaft the individual characteristics are listed in Table 3.2.

Table 3.2 **Procedure for the selection of a ballscrew**

	Rolled	Peeled	Ground
Profile			
Manufacturing process	Forming process	Cutting process	Grinding process
Typical applications	Transportation	Transportation and positioning	Positioning
Tolerance classes	T5 – T10	T5 + T7	T0 – T5
Nominal diameter [mm]	8 – 63	16 – 80	6 – 100
Max. shaft length [1] [mm]	500 – 5,600	3,300 – 6,500	110 – 10,000
Nut shapes	Flange nut Cylindrical nut	Flange nut Cylindrical nut Double nut	Flange nut Cylindrical nut Double nut
Availability	From stock	From stock	Upon request

[1] Depends on the diameter and the tolerance class

3.4 Ball recirculation systems

HIWIN ballscrews are available with three different recirculation systems.

The external recirculation system consists of the return tubes and the clamping plate. The balls are placed in the ball track between the ballscrew shaft and nut. At the end of the nut, they are guided out of the ball track and back to the start via a return tube; ball circulation is therefore a closed circuit (see Fig. 3.7).

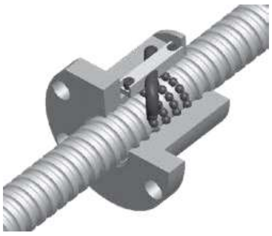


Fig. 3.7 **External recirculation type nut**

In the case of the internal single recirculation, the balls are each fed back to the beginning of a thread turn with the help of the deflecting parts. The balls undertake just one circuit around the shaft. The circuit is closed by a deflecting part in the ballscrew nut and allows the balls to return to the start via the rear of the thread. The position of the ball deflection in the nut gives the internal single recirculation system its name (see Fig. 3.8).

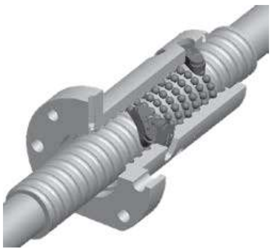


Fig. 3.8 **Internal single recirculation type nut**

The third type of return is the cassette recirculation system shown in Fig. 3.9. It has the same basic principle as the external return, however, the balls are returned via a channel in the ballscrew nut. The balls perform one complete cycle in the ballscrew nut. The cassette return is also called "internal total recirculation".

Fig. 3.9 **Cassette recirculation type nut**

3.5 Wiper variants

NBR wiper (N): the allrounder
The nitrile rubber wiper offers excellent sealing and wiping properties for most environmental conditions and is therefore used in almost all applications.

NBR-finger wiper (K): the one for the rough stuff
Wherever stubborn dirt prevails, it really cleans up. The finger wiper with its hard plastic fingers should not be missing in environments with coarse dirt particles.

Felt wiper (F): the most absorbent among the wipers
Felt has the property of absorbing liquids, storing them and releasing them again. This gives the felt wiper an ideal wiping effect and provides additional lubrication.

Felt-finger wiper (V): the duo
With our duo, consisting of a felt wiper and a finger wiper, dirt – whether coarse or fine – has no chance.

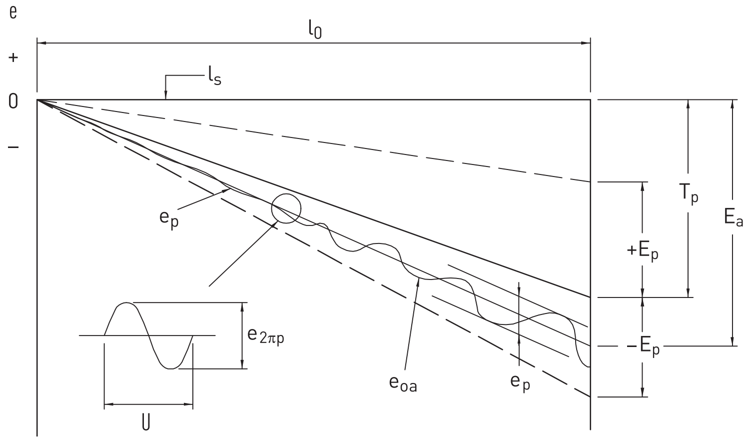
Table 3.3 **Wiper properties**

	NBR (N)	Felt (F)	NBR-finger (K)	Felt-finger (V)
Temperature resistance		+ +		+
Soiling	+		+ +	+
Friction reduction	+ +		+	
Tightness	+ +		+ +	
Emergency running		+ +		+ +
Chemical resistance	+ +	+	+	+

3.6 Accuracy of the HIWIN ballscrews

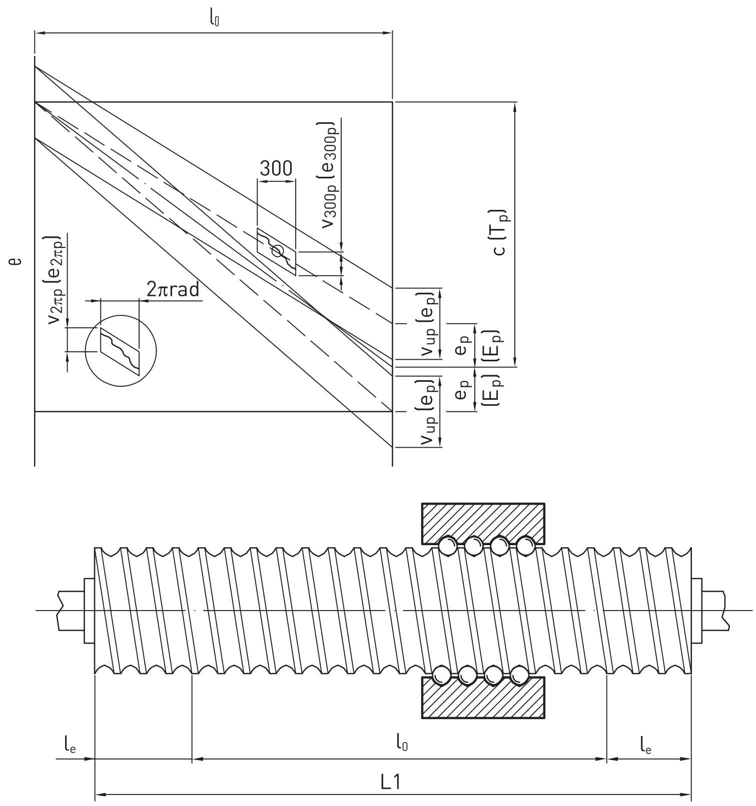
3.6.1 Tolerance class

HIWIN ballscrews are produced in various tolerance classes depending on the application's accuracy requirements.



T_p	Difference between nominal and actual path. This value is determined by the various requirements of the customer's application.
e	Path deviation
E_p	Maximum actual path deviation from nominal path over complete distance.
$e_{2\pi p}$	Path deviation within one revolution
E_a	Actual path, determined using laser measurement
e_p	Actual path deviation. Maximum deviation of total actual path from actual total nominal path in the corresponding area
e_{oa}	Average actual path deviation
l_0	Useful path
l_s	Nominal path

Fig. 3.10 **HIWIN measurement curve of lead of precision ballscrews**



l_0	Useful path
e	Path deviation
e_{300p}	Actual path deviation at 300 mm. Actual path deviation over 300 mm at any thread position
e_{oa} (E_a)	Average actual path deviation over useful path l_0
C (T_p)	Path compensation over useful path l_0
e_p (E_p)	Limit deviation of nominal path
v_{up} (e_p)	Permissible path deviation over useful path l_0
v_{300p} (e_{300p})	Permissible path deviation over 300 mm
$v_{2\pi p}$ ($e_{2\pi p}$)	Permissible path deviation over one revolution

l_0	Useful path
l_e	Path outside of the nominal path
L1	Total thread length

Fig. 3.11 **DIN ISO measurement curve of lead of ballscrews**

3.6.2 Travel fluctuation over 300 mm travel distance

As an international company, HIWIN produces ballscrews on the basis of DIN ISO 3408 in tolerance classes 0, 1, 3, 5, 7 and 10 and in accordance with the Japanese standard JIS in classes 0, 2 and 4. The tolerance classes as well as the permissible travel fluctuation v_{300p} over 300 mm path are listed in Table 3.4.

Table 3.4 **International standards for tolerance classes of ballscrews**

HIWIN tolerance class		T0	T1	T2	T3	T4	T5	T7	T10
v_{300p}	DIN ISO	3.5	6	–	12	–	23	52	210
	JIS	3.5	–	8	–	18	–	–	–

Unit: [µm]

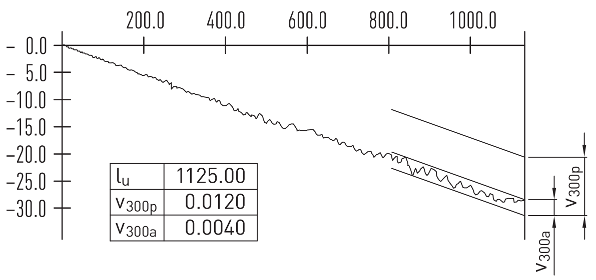


Fig. 3.12 **Travel fluctuation over 300 mm useful path**

v_{300a} Travel fluctuation over 300 mm at any position (measurement in accordance with DIN standard 69051-3-3)

3.6.3 Path deviation and travel fluctuation over useful path

Positioning ballscrews

For positioning ballscrews (peeled and ground) the permissible path deviations over the useful path l_u are listed in Table 3.5.

Table 3.5 **Tolerance classes of peeled and ground ballscrews**

HIWIN tolerance class		T0		T1		T2		T3		T4		T5	
Useful path l_u		e_p	v_{up}	e_p	v_{up}	e_p	v_{up}	e_p	v_{up}	e_p	v_{up}	e_p	v_{up}
above	below												
–	315	4	3.5	6	6	12	8	12	12	23	18	23	23
315	400	5	3.5	7	6	13	10	13	12	25	20	25	25
400	500	6	4.0	8	7	15	10	15	13	27	20	27	26
500	630	6	4.0	9	7	16	12	16	14	30	23	32	29
630	800	7	5.0	10	8	18	13	18	16	35	25	36	31
800	1,000	8	6.0	11	9	21	15	21	17	40	27	40	34
1,000	1,250	9	6.0	13	10	24	16	24	19	46	30	47	39
1,250	1,600	11	7.0	15	11	29	18	29	22	54	35	55	44
1,600	2,000	13		18	13	35	21	35	25	65	40	65	51
2,000	2,500	15		22	15	41	24	41	29	77	46	78	59
2,500	3,150	18		26	17	50	29	50	34	93	54	96	69
3,150	4,000			32	21	60	35	62	41	115	65	115	82
4,000	5,000			39		72	41	76	49	140	77	140	99
5,000	6,300			48		90	50	92		170	93	170	119
6,300	8,000					110	60					210	130
8,000	10,000											260	145
10,000	12,000											320	180

e_p [µm] Path deviation: Limit deviation of nominal path

v_{up} [µm] Travel fluctuation over useful path

Transportation ballscrews

For transportation ballscrews (rolled) the permissible path deviation over the useful path (tolerance for desired path) can be calculated with Formula F 3.1.

F 3.1

$$e_p = \pm \frac{l_u}{300} \times v_{300p}$$

e_p Path deviation: Limit deviation of nominal path
l_u Useful path
v_{300p} Permissible travel fluctuation over 300 mm path

Curves of lead accuracy when measuring on a laser measuring device according to DIN ISO 3408

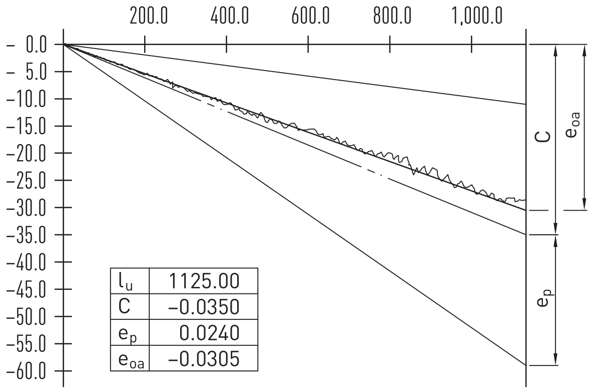


Fig. 3.13 **Average path deviation over useful path l_u**

l_u Useful path
C Travel compensation
e_p Path deviation: Limit deviation of nominal path
e_{oa} Average deviation of actual path

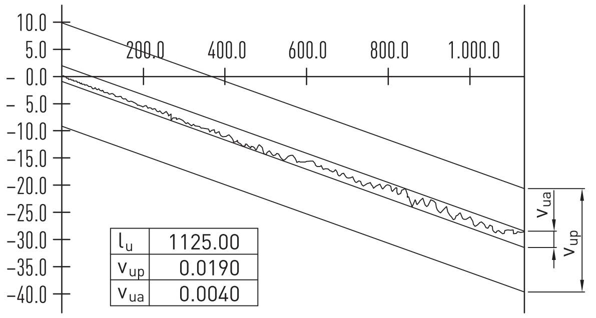


Fig. 3.14 **Travel fluctuation over useful path l_u**

l_u Useful path
v_{up} Permissible travel fluctuation over useful path
v_{ua} Actual travel fluctuation over useful path

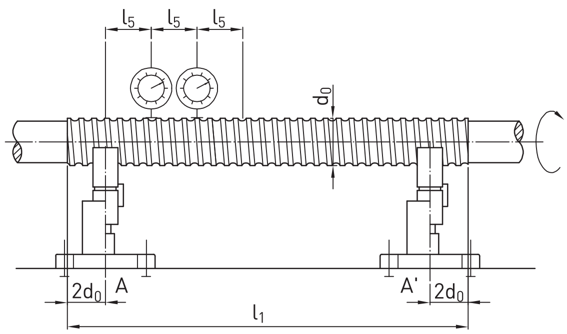
Table 3.6 **Recommended tolerance classes for various applications**

	Application	Axis	Tolerance class						
			T0	**T1**	**T2**	**T3**	**T4**	**T5**	**T7**
CNC machine tools	Turning	X	○	○	○	○			
		Z				○	○	○	
	Milling Bore milling	X		○	○	○	○	○	
		Y		○	○	○	○	○	
		Z			○	○	○	○	
	Machining centres	X		○	○	○	○		
		Y		○	○	○	○		
		Z			○	○	○		
	Coordinate drilling	X	○	○					
		Y	○	○					
		Z	○	○					
	Drilling	X				○	○	○	
		Y				○	○	○	
		Z					○	○	○
	Grinding	X	○	○	○				
		Y		○	○	○			
	Die sinking	X		○	○	○			
		Y		○	○	○			
		Z			○	○	○	○	
	Wire eroding	X		○	○	○			
		Y		○	○	○			
		U		○	○	○	○		
		V		○	○	○	○		
	Laser cutting	X			○	○	○		
		Y			○	○	○		
		Z			○	○	○		
Other machines	Punching machine	X				○	○	○	
		Y				○	○	○	
	Wood processing machines								○
	Precision industrial robots			○	○	○	○		
	Industrial robots							○	○
	Coordinate measuring device		○	○	○				
	Non-CNC machines					○	○	○	
	Transport units						○	○	○
	X-Y tables			○	○	○	○	○	
	Linear electric lifting cylinders							○	○
	Aircraft landing gear							○	○
	Wing control							○	○
	Gate valves								○
	Power-assisted steering systems								○
	Glass grinders				○	○	○	○	○
	Surface grinders						○	○	
	Induction hardening machine								○
	Electric machines			○	○	○	○	○	○

3.6.4 Tolerance details and measuring methods for HIWIN ballscrews

Table 3.7 **Radial runout t5 of ballscrew shaft outer diameter related to AA' per length l5 (measurement in accordance with DIN ISO 3408)**

Nominal Ø d_0 [mm]		Reference length [mm]	Tolerance class l_{5p} [µm] for l_5							
above	up to	l_5	T0	T1	T2	T3	T4	T5	T7	T10
6	12	80	16	20	23	25	25	32	40	80
12	25	160	16	20	23	25	25	32	40	80
25	50	315	16	20	23	25	25	32	40	80
50	100	630	16	20	23	25	25	32	40	80
100	200	1,250	16	20	23	25	25	32	40	80
l_1 / d_0			Tolerance class l_{5maxp} [µm] for $l_1 > 4l_5$							
above	up to		T0	T1	T2	T3	T4	T5	T7	T10
–	40		32	40	45	50	50	64	80	160
40	60		48	60	70	75	75	96	120	240
60	80		80	100	115	125	125	160	200	400
80	100		128	160	180	200	200	256	320	640



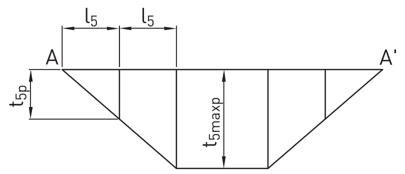
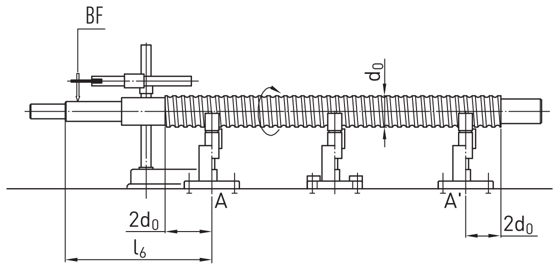


Table 3.8 **Radial runout $t_{6.1}$ of bearing seat related to AA' per unit length l (measurement in accordance with DIN ISO 3408)**

Nominal Ø d_0 [mm]		Reference length [mm]	Tolerance class $t_{6.1p}$ [µm] for l							
above	up to	l	T0	T1	T2	T3	T4	T5	T7	T10
6	20	80	6	10	11	12	12	20	40	63
20	50	125	8	12	14	16	16	25	50	80
50	125	200	10	16	18	20	20	32	63	100
125	200	315	–	–	20	25	25	40	80	125



BF Bearing seat

Table 3.9 **Radial runout $t_{6.2}$ of bearing seat related to the centre line of the screw part (measurement in accordance with DIN ISO 3408)**

Nominal Ø d_0 [mm]		Tolerance class $t_{6.2p}$ [µm]			
above	up to	T0	T1	T3	T5
–	8	3	5	8	10
8	12	4	5	8	11
12	20	4	6	9	12
20	32	5	7	10	13
32	50	6	8	12	15
50	80	7	9	13	17
80	125	–	10	15	20

Table 3.10 **Radial runout $t_{7.1}$ of journal diameter related to the bearing seat (measurement in accordance with DIN ISO 3408)**

Nominal Ø d_0 [mm]		Reference length [mm]	Tolerance class $t_{7.1p}$ [µm] for l							
above	**up to**	**l**	**T0**	**T1**	**T2**	**T3**	**T4**	**T5**	**T7**	**T10**
6	**20**	80	4	5	6	6	6	8	12	16
20	**50**	125	5	6	7	8	8	10	16	20
50	**125**	200	6	8	8	10	10	12	20	25
125	**200**	315	–	–	10	12	12	16	25	32

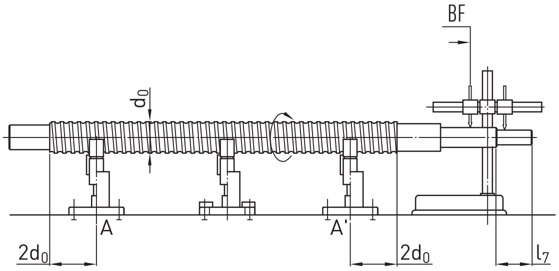


Table 3.11 **Radial runout $t_{7.2}$ of the journal diameter related to the centre line of the bearing seat (measurement in accordance with DIN ISO 3408)**

Nominal Ø d_0 [mm]		Tolerance class $t_{7.2p}$ [µm]			
above	**up to**	**T0**	**T1**	**T3**	**T5**
–	**8**	3	5	8	10
8	**12**	4	5	8	11
12	**20**	4	6	9	12
20	**32**	5	7	10	13
32	**50**	6	8	12	15
50	**80**	7	9	13	17
80	**125**	–	10	15	20

Table 3.12 **Axial runout $t_{8.1}$ of shaft (bearing) faces related to AA' (measurement in accordance with DIN ISO 3408)**

Nominal Ø d_0 [mm]		Tolerance class $t_{8.1p}$ [µm]							
above	**up to**	**T0**	**T1**	**T2**	**T3**	**T4**	**T5**	**T7**	**T10**
6	**63**	3	3	3	4	4	5	6	10
63	**125**	3	4	4	5	5	6	8	12
125	**200**	–	–	6	6	6	8	10	16

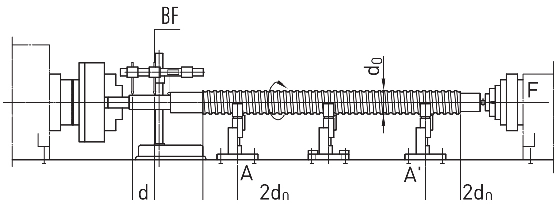


Table 3.13 **Axial runout $t_{8.2}$ of the shaft faces related to the centre line of the screw shaft (measurement in accordance with DIN ISO 3408)**

Nominal Ø d_0 [mm]		Tolerance class $t_{8.2p}$ [µm]			
above	**up to**	**T0**	**T1**	**T3**	**T5**
–	**8**	2	3	4	5
8	**12**	2	3	4	5
12	**20**	2	3	4	5
20	**32**	2	3	4	5
32	**50**	2	3	4	5
50	**80**	3	4	5	7
80	**125**	–	4	6	8

Table 3.14 **Axial runout t_9 of ballscrew nut location face related to AA' (for preloaded ballscrew nuts only) (measurement in accordance with DIN ISO 3408)**

Flange diameter D_2 [mm]		Tolerance class t_{9p} [µm]							
above	**up to**	**T0**	**T1**	**T2**	**T3**	**T4**	**T5**	**T7**	**T10**
16	**32**	8	10	10	12	12	16	20	–
32	**63**	10	12	12	16	16	20	25	–
63	**125**	12	16	16	20	20	25	32	–
125	**250**	16	20	20	25	25	32	40	–
250	**500**	–	–	15	32	32	40	50	–

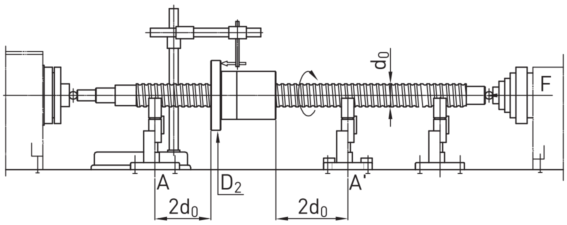


Table 3.15 **Radial runout t_{10} of ballscrew nut location diameter related to AA' (for preloaded and rotating ballscrew nuts only) (measurement in accordance with DIN ISO 3408)**

Outer diameter D_1 of ballscrew nut [mm]		Tolerance class t_{10p} [µm]							
above	**up to**	**T0**	**T1**	**T2**	**T3**	**T4**	**T5**	**T7**	**T10**
16	**32**	8	10	10	12	12	16	20	–
32	**63**	10	12	12	16	16	20	25	–
63	**125**	12	16	16	20	20	25	32	–
125	**250**	16	20	20	25	25	32	40	–
250	**500**	–	–	–	32	32	40	50	–

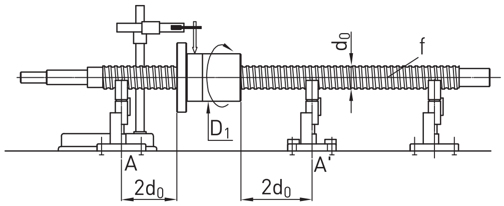
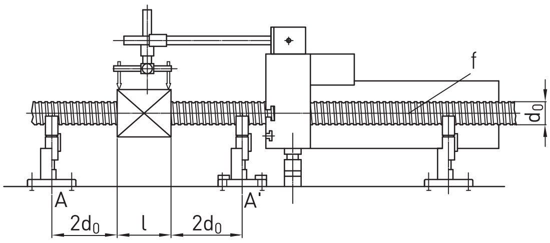


Table 3.16 **Parallelism deviation t_{11} of rectangular ballscrew nut related to AA' (for preloaded ballscrew nuts only) (measurement in accordance with ISO 3408)**

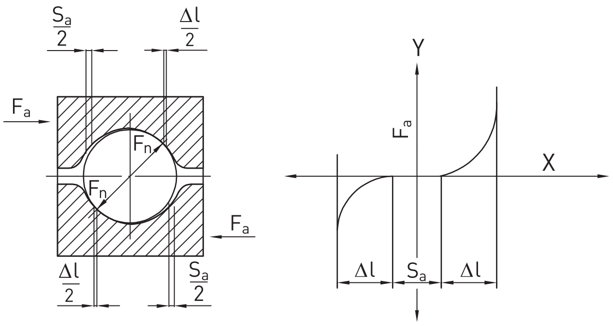
Tolerance class t_{11p} [µm] / 100 mm, cumulative							
T0	**T1**	**T2**	**T3**	**T4**	**T5**	**T7**	**T10**
14	16	16	20	20	25	32	–



f fixed

3.7 Preload and play

The axial force F_a, caused by outer drive forces or inner preload forces, produces two kinds of axial play. Firstly, axial play S_a, that originates from the air between the ball and ball track. Secondly, the spring compression play $\Delta\ell$, caused by the force F_n, which acts vertically on the point of contact.
By default, rolled and peeled ballscrews are delivered with slight play. This is sufficient for most applications, and has the advantage that the ballscrews run smoothly and a low starting torque is required.
If increased demands are placed on the positioning accuracy and rigidity, the ballscrew should be used with no axial play or preloaded. For preloading, different methods are available, which are explained below.



F_a Axial load
X Deviation

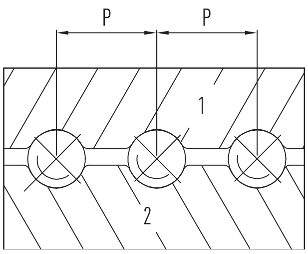
Fig. 3.15 **Gothic arch profile and preload**

3.7.1 HIWIN types of preload

Preload can be generated either with double nuts, or single nuts with lead offset or in the case of preloaded single nuts by adjusting the ball size.

Preloaded single nuts

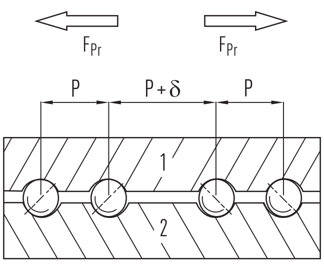
There are two kinds of preload for the single nuts. One of these is the "preload method with oversized balls". This involves balls which are slightly larger than the space in the ball tracks between spindle and nut; the ball therefore makes contact at four points (see Fig. 3.16).



P Lead
1 Nut
2 Screw shaft

Fig. 3.16 **Preload from ball size**

The other method is known as "preload from lead offset" (see Fig. 3.17). The nut is ground such that it is offset from the central lead. This type of preload takes the place of the classic double nut preload and offers the benefit that a compact single nut with good rigidity can be used with low preload forces. This method is not, however, suited to use with high preloads and high leads. The recommend preload force is less than 5 % of the dynamic load rating (C).

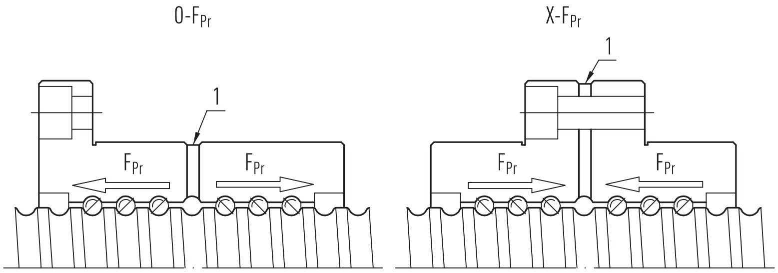


F_{Pr} Preload force
P Lead
1 Nut
2 Shaft

Fig. 3.17 **Preload from lead offset**

Preloaded double nuts

The preload is generated by inserting a spacer between the nuts (see Fig. 3.18). The O preload results from fitting an oversized spacer which pushes the halves of the nut apart. The X preload is generated with an undersized spacer which pulls the nuts together.



F_{Pr} Preload force
1 Spacer

Fig. 3.18 **Preload from spacer**

3.7.2 Effects of preload

Preload increases the thread's friction torque and therefore causes increases in temperature during operation. To ensure a long service life and low increase in temperature, the maximum preload should not exceed 5 % of the dynamic load rating for single nuts and 10 % for double nuts.
Furthermore preload has an effect on the running characteristics. Besides an increase in idle torque it leads to fluctuations in idle torque, especially with ballscrews with high tolerance classes. (see Section 3.7.3).

Basically, ballscrews should only be preloaded when it is absolutely necessary to avoid axial play.

3.7.3 Idle torque fluctuation

(1) Measuring method

Preload produces a friction torque between nut and threaded shaft. This is measured by moving the threaded shaft at constant speed while holding the nut with a special locking device (see Fig. 3.19).
The force F_{Pr} measured by the force sensor is used to calculate the idle torque of the threaded shaft.

F 3.2

$$T_d = \frac{K_p \times F_{pr} \times P}{2000 \times \pi}$$

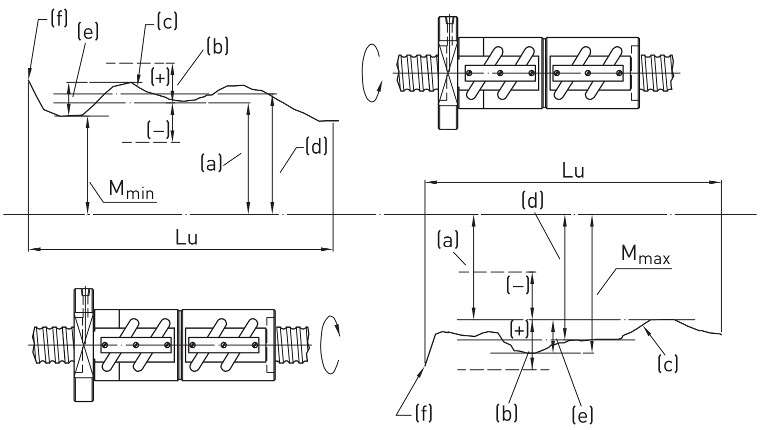
T_d Idle torque of preloaded nut
F_{Pr} Preload force
P Lead
K_P Preload friction coefficient
K_P $= \frac{1}{\eta_1} - \eta_2$ (between 0.1 and 0.3)
h_1, η_2 are the mechanical efficiencies of the ballscrew

(2) Measurement conditions

1. Without wiper
2. Speed: 100 rpm
3. Dynamic viscosity of lubricant 61.2 – 74.8 cSt [mm/s] at 40 °C, complying with ISO VG 68 or JIS K2001

(3) The result of the measurement is displayed using standard depiction of idle torque; the nomenclature is shown in Fig. 3.19.

(4) Fluctuations in idle torque (incorporated in the tolerance class definition) are listed in Table 3.17.



(a) Idle torque
(b) Fluctuations in idle torque
(c) Friction torque currently measured
(d) Average measured friction torque
(e) Measured starting torque
Lu Useful path of nut
M_{min} Minimum torque
M_{max} Maximum torque

Fig. 3.19 **Nomenclature for measuring idle torques**

Table 3.17 **Fluctuation range of idle torque with preload in % (in accordance with DIN ISO 3408)**

Basic friction torque T_{p0} [Nm]		Length of useful path of thread [mm]																				
		4,000 mm maximum														over 4,000 mm						
		Slenderness ratio ≤ 40 Tolerance class							40 < Slenderness ratio < 60 Tolerance class							Tolerance class						
Above	**Up to**	**T0**	**T1**	**T2**	**T3**	**T4**	**T5**	**T7**	**T0**	**T1**	**T2**	**T3**	**T4**	**T5**	**T7**	**T0**	**T1**	**T2**	**T3**	**T4**	**T5**	**T7**
0.2	**0.4**	30	35	40	40	45	50	–	40	40	50	50	60	60	–	–	–	–	–	–	–	–
0.4	**0.6**	25	30	35	35	40	40	–	35	35	40	40	45	45	–	–	–	–	–	–	–	–
0.6	**1.0**	20	25	30	30	35	35	40	30	30	35	35	40	40	45	–	–	–	40	43	45	50
1.0	**2.5**	15	20	25	25	30	30	35	25	25	30	30	35	35	40	–	–	–	35	38	40	45
2.5	**6.3**	10	15	20	20	25	25	30	20	20	25	25	30	30	35	–	–	–	30	33	35	40
6.3	**10.0**	–	–	15	15	20	20	30	–	–	20	20	25	25	35	–	–	–	25	23	30	35

Note:
1. Slenderness ratio = thread length of shaft/nominal diameter of shaft [mm]
2. To calculate the idle torque, see Formula F 3.2
3. For more information, please contact HIWIN

3.8 Calculations

Bases of calculations in accordance with DIN ISO 3408.

3.8.1 Load ratings

Dynamic load rating C_{dyn} (theoretical)

The dynamic load rating describes the load at which 90 % of all ballscrews reach a life expectancy of 1×10^6 revolutions (C). The reliability factor can be taken into account in accordance with Table 3.18. The dynamic load rating is listed in the dimensions tables for the nuts.

Static load rating C_0

The static load rating describes the load which causes permanent deformation of the ball track of more than 0.0001 of the ball diameter. In order to calculate the maximum static load rating, the static structural safety S_0 of the application conditions must be taken into account.

F 3.3

$$S_0 \times F_{amax} < C_0$$

S_0 Static structural safety
C_0 Static load rating (dimensions table for nut)
F_{amax} Max. static axial load

3.8.2 Service life

a) Average speed n_m

F 3.4

$$n_m = n_1 \times \frac{t_1}{100} + n_2 \times \frac{t_2}{100} + n_3 \times \frac{t_3}{100} + \ldots$$

n_m Average speed, total [rpm]
n_n Average speed in phase n [rpm]
t_n Amount of time in phase n [%]

b) Preload

F 3.5

$$F_{pr} = \frac{f_{pr}}{100\,\%} \times C_{dyn}$$

F 3.6

$$F_{lim} = 2^{3/2} \times F_{pr}$$

F_{pr} Preload force
C_{dyn} Dynamic load rating
f_{pr} Preload factor in %
Single nut $f_{pr} \leq 5\,\%$
Double nut $f_{pr} \leq 10\,\%$
F_{lim} Disengagement force

Distinction of cases:

$F_n > F_{lim}$ No influence from preload: $F_{bn} = F_n$
$F_n < F_{lim}$ Influence from preload: Formula F 3.7

F 3.7

$$F_{bn} = \left(1 + \frac{F_n}{2^{3/2} \times F_{pr}}\right)^{3/2} \times F_{pr}$$

F_n Axial loading in phase n
F_{bn} Operating axial loading in phase n

F_n must be calculated for all phases and used in Formula F 3.7.

c) Average operating load F_{bm}

- With alternating load and constant speed

F 3.8

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

F_{bm} Average operating load [N]
F_{bn} Operating axial loading in phase n
f_p Operating condition factor
f_p 1.1 – 1.2 operation without impact
1.3 – 1.8 operation under normal conditions
2.0 – 3.0 operation with high impact and with vibrations
3.0 – 5.0 short-stroke applications < 3 × nut length

- With alternating load and alternating speed:

F 3.9

$$F_{bm} = \sqrt[3]{F_{b1}^3 \times \frac{n_1}{n_m} \times \frac{t_1}{100} \times f_{p1}^3 + F_{b2}^3 \times \frac{n_2}{n_m} \times \frac{t_2}{100} \times f_{p2}^3 + F_{b3}^3 \times \frac{n_3}{n_m} \times \frac{t_3}{100} \times f_{p3}^3 \ldots}$$

d) Axial loading on both sides:

- Service life in revolutions

F 3.10

$$L_1 = \left(\frac{C_{dyn}}{F_{bm1}}\right)^3 \times 10^6 \qquad L_2 = \left(\frac{C_{dyn}}{F_{bm2}}\right)^3 \times 10^6$$

F 3.11

$$L = \left(L_1^{-10/9} + L_2^{-10/9}\right)^{-9/10}$$

L_1 Service life in revolutions, forward motion
L_2 Service life in revolutions, backward motion
C_{dyn} Dynamic load rating [N]
F_{bm1} Average operating load, forward motion
F_{bm2} Average operating load, backward motion
L Service life in revolutions

- Conversion of service life into operating hours

F 3.12

$$L_h = \frac{L}{n_m \times 60}$$

L_h Service life in operating hours
n_m Average speed [rpm], see Formula F 3.4

- Conversion of distance travelled [km] into operating hours:

F 3.13

$$L_h = \left(\frac{L_{km} \times 10^6}{P}\right) \times \frac{1}{n_m \times 60}$$

L_h Service life in operating hours
L_{km} Service life in distance travelled [km]
P Lead [mm]
n_m Average speed [rpm]

- The modified service life with different reliability factors is calculated using

F 3.14

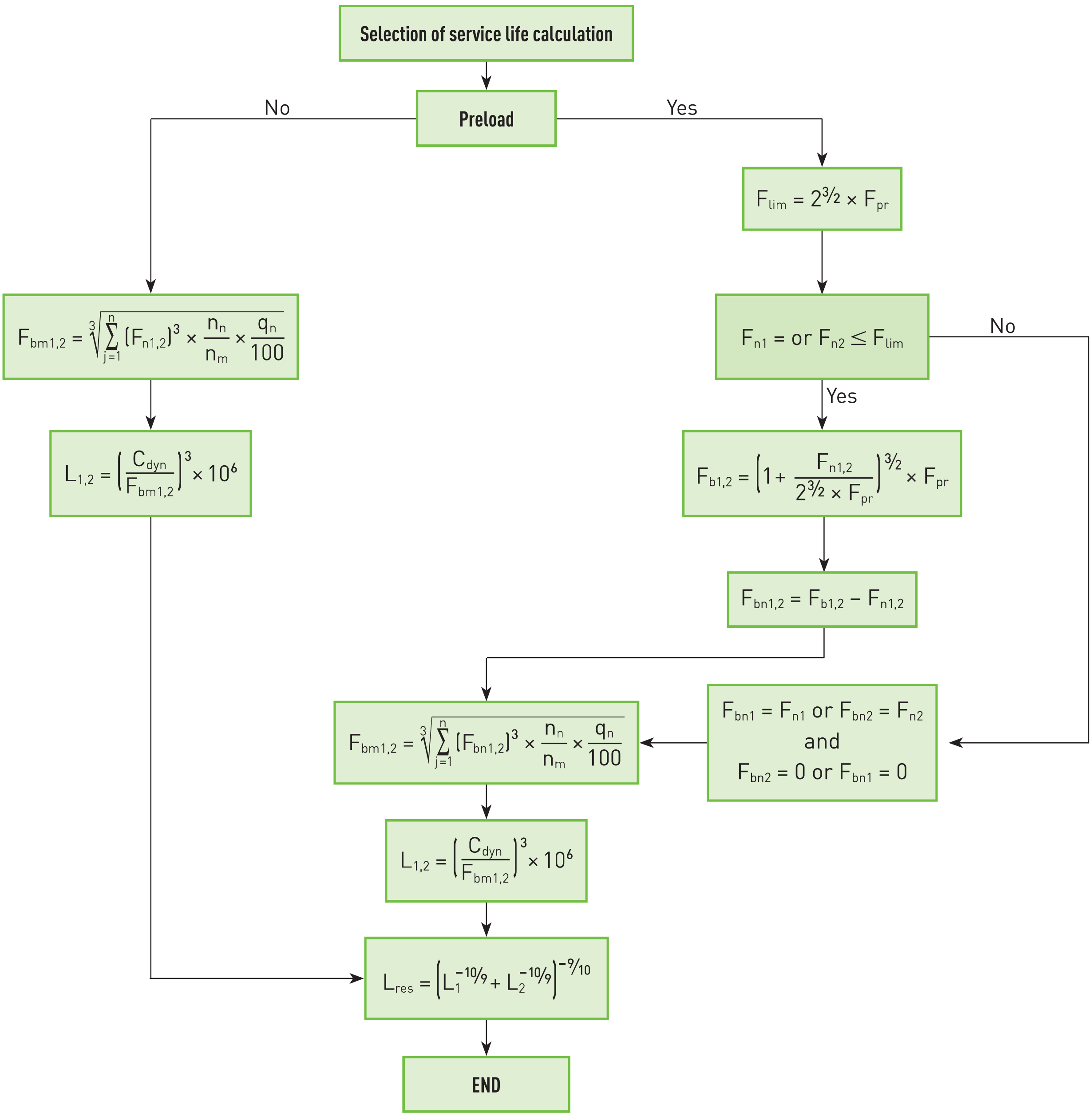
$$L_m = L \times f_r \qquad L_{hm} = L_h \times f_r$$

f_r Reliability factor (see Table 3.18)

Table 3.18 **Reliability factor for calculating service life**

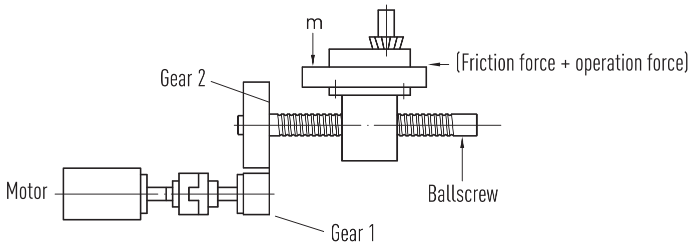
Resilience %	Reliability factor f_r
90	1.00
95	0.63
96	0.53
97	0.44
98	0.33
99	0.21

Flow chart for calculating service life



3.8.3 Drive torque and drive output of motor

Fig. 3.20 shows the influencing parameters of a feed system with ballscrew.
Below you will find the formula for calculating the drive torque required of the motor:



(1) Motor, (2) Gear 1, (3) Gear 2, (4) (Friction force + operation force),
(5) Ballscrew

Fig. 3.20 **Load trend of a system with ballscrew**

- Normal operation (conversion of rotary motion into linear motion)

F 3.15
$$T_a = \frac{F_w \times P}{2{,}000 \times \pi \times \eta_1}$$

T_a Drive torque for normal operation [Nm]
T_c Drive torque for reverse operation [Nm]
F_w Effective axial load [N], friction force + operating force
P Lead [mm]
η_1 Mechanical efficiency (0.85 – 0.95), normal operation
η_2 Mechanical efficiency (0.75 – 0.85), reverse operation

- Reverse operation (conversion of linear motion into rotary motion)

F 3.16
$$T_c = \frac{F_w \times P \times \eta_2}{2{,}000 \times \pi}$$

- Drive torque of motor

For normal operation:

F 3.17
$$T_M = (T_a + T_b + T_d)\,\frac{N_1}{N_2}$$

T_M Motor drive torque [Nm]
T_b Friction torque of support bearing [Nm]
T_d Idle torque [Nm]
N_1 Number of teeth on driving gear wheel
N_2 Number of teeth on driven gear wheel

For acceleration:

F 3.18
$$T'_a = J \times \alpha$$

T'_a Motor drive torque during acceleration [Nm]
J Inertia torque of system [Nm²]
α Angular acceleration [rad/s²]
t_a Acceleration start-up time [s]
n_1 Initial speed [rpm]
n_2 Final speed [rpm]

F 3.19
$$\alpha = \frac{2\pi \times \Delta n}{60 \times t_a}$$

F 3.20
$$\Delta n = n_2 - n_1$$

F 3.21
$$J = J_M + J_{G1} + J_{G2} \times \left(\frac{N_1}{N_2}\right)^2 + \frac{1}{2} m_r \times \left(\frac{d_n}{2000}\right)^2 \times \left(\frac{N_1}{N_2}\right)^2 + m_l \times \left(\frac{P}{2000\pi}\right)^2 \times \left(\frac{N_1}{N_2}\right)^2$$

= motor inertia + equivalent gear inertia + inertia of ballscrew (see Fig. 3.20)

m_r Mass of rotating parts [kg]
m_l Mass of components moved in linear fashion [kg]
d_n Nominal diameter of ballscrew [mm]
J_M Motor inertia [kgm²]
J_{G1} Inertia of drive gear [kgm²]
J_{G2} Inertia of driven gear[kgm²]

Total drive torque:

F 3.22
$$T_{Ma} = T_M + T'_a$$

T_{Ma} Total drive torque [Nm]

- Drive output

F 3.23

$$P_A = \frac{T_{pmax} \times n_{max}}{9,550}$$

- Acceleration time check

F 3.24

$$t_a = \frac{J}{T_{M1} - T_L} \times \frac{2\pi \times n_{max}}{60} \times f$$

P_A Maximum reliable drive output [kW]
T_{pmax} Maximum drive torque (safety factor × T_{max}) [Nm]
n_{max} Maximum speed [rpm]
t_a Acceleration start-up time [s]
J Total inertia torque [kgm²]
T_{M1} Nominal torque of motor [Nm]
T_L Drive torque at nominal speed [Nm]
f Safety factor = 1.5

3.8.4 Buckling load

F 3.25

$$F_k = 4.072 \times 10^5 \left(\frac{f_k \times d_k^4}{l_s^2} \right)$$

F 3.26

$$F_{kmax} = 0.5 \times F_k$$

F_k Permissible load [N]
F_{kmax} Max. permissible load [N]
d_k Core diameter of threaded shaft [mm]
l_s Unsupported shaft length [mm] (see Fig. 3.21)
f_k Factor for different types of assembly (buckling load)

Fixed bearing – fixed bearing	$f_k = 1.0$
Fixed bearing – supported bearing	$f_k = 0.5$
Supported bearing – supported bearing	$f_k = 0.25$
Fixed bearing – no bearing	$f_k = 0.0625$

3.8.5 Critical speed

F 3.27

$$n_k = 2.71 \times 10^8 \left(\frac{f_n \times d_k}{l_s^2} \right)$$

F 3.28

$$n_{kmax} = 0.8 \times n_k$$

n_k Critical speed [rpm]
n_{kmax} Max. permissible speed [rpm]
d_k Core diameter of threaded shaft [mm]
l_s Unsupported shaft length [mm] (see Fig. 3.21)
f_n Factor for different types of assembly (critical speed)

Fixed bearing – fixed bearing	$f_n = 1.0$
Fixed bearing – supported bearing	$f_n = 0.692$
Supported bearing – supported bearing	$f_n = 0.446$
Fixed bearing – no bearing	$f_n = 0.147$

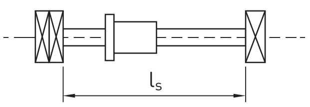


Fig. 3.21 **Definition of „Unsupported shaft length“**

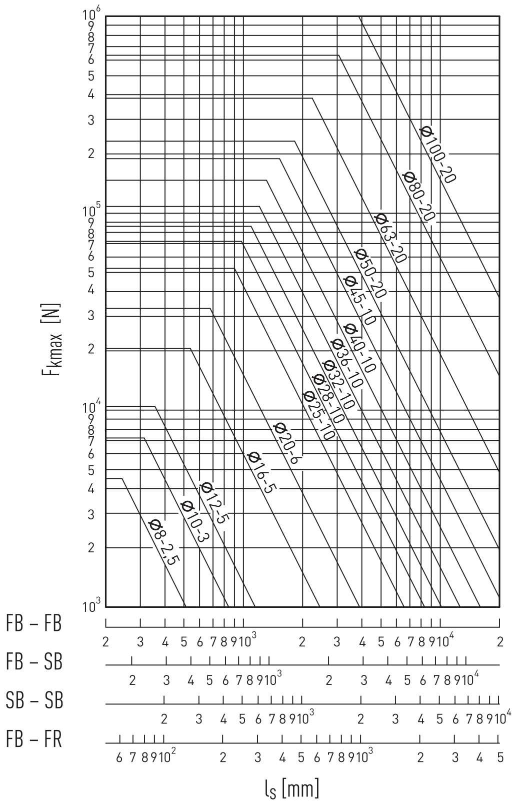


Fig. 3.22 **Buckling load for different diameters and lengths of threaded shafts**

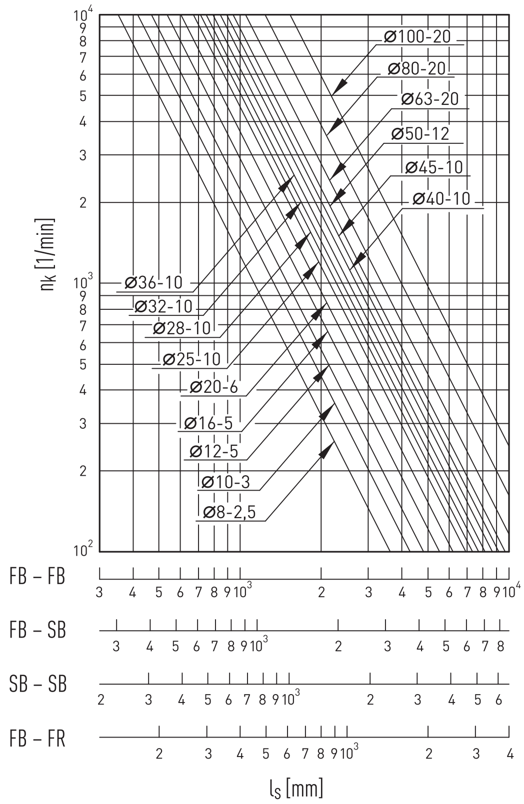


Fig. 3.23 **Critical speed for different diameters and lengths of threaded shafts**

l_s Unsupported shaft length [mm]
F_{kmax} Critical axial load [N]
n_k Critical speed [rpm]
FB Fixed bearing
FR Free
SB Support bearing

3.8.6 D_N value for working speed of a ballscrew

The specific speed value D_N has a huge influence on the behaviour of the ballscrew in terms of noise and heat development and service life of the recirculation system.

For HIWIN ballscrews

F 3.29

$$D_N = d_s \times n_{max}$$

d_s Shaft diameter [mm]
n_{max} Max. speed [rpm]

$D_N \leq 90{,}000$ for ballscrews with cassette recirculation
$D_N \leq 120{,}000$ for ballscrews with cassette recirculation and no axial play
$D_N \leq 150{,}000$ for high-speed ballscrews (upon request)

3.8.7 Rigidity

Rigidity describes the flexibility of a machine element. The overall rigidity of a ballscrew is determined by the axial rigidity of the nut/shaft system, the contact rigidity of the ball track and the rigidity of the threaded shaft. The following factors should also be taken into account when fitting the ballscrew in a machine: rigidity of support bearings, assembly conditions of nuts with table etc.
The rigidity of the nut/shaft unit and the ball and ball track can be combined to produce the rigidity of the nut Rn, which is listed in the dimensions tables for the different types of nuts.

- Rigidity of a ballscrew

F 3.30

$$\frac{1}{R_{bs}} = \frac{1}{R_s} + \frac{1}{R_n}$$

R_{bs} Overall rigidity of a ballscrew [N/μm]
R_s Rigidity of threaded shaft [N/μm]
R_n Rigidity of nut [N/μm]

Rigidity of threaded shaft

F 3.31

$$R_{s1} = \frac{\pi \times d_c^2 \times E}{4 \times l_1 \times 10^3}$$

fixed – floating/unsupported

F 3.32

$$R_{s2} = \frac{\pi \times d_c^2 \times E}{4 \times l_1 \times 10^3} \times \frac{l_2}{l_2 - l_1}$$

fixed – fixed

F 3.33

$$d_c = PCD - D_k \times \cos\alpha$$

R_s Rigidity of threaded shaft [N/µm]
d_c Diameter on which the force acts on the ballscrew shaft
E Elasticity module [N/mm²]
α Contact angle between ball and track [°]
PCD Ball centre diameter of circle [mm]
D_k Nominal diameter of ball [mm]
l_1 Distance between bearing and nut [mm]
l_2 Distance between bearing and bearing [mm]

Rigidity of nut

The nut rigidity can be checked using an axial force corresponding to the maximum possible preload of 10 % of the dynamic load rating (C_{dyn}) (this is listed in the dimensions tables for the nuts). With a lower preload, the nut rigidity can be determined by extrapolation:

F 3.34

$$R_n = 0.8 \times R \times \left(\frac{F_{pr}}{0.1 \times C_{dyn}}\right)^{1/3}$$

R_n Rigidity of nut [N/µm]
R Rigidity in accordance with dimensions table [N/µm]
F_{pr} Preload [N]
C_{dyn} Dynamic load rating from dimensions table [N]

The rigidity of a single nut with play can be calculated as follows with an external axial load of 0.28 C_{dyn}:

F 3.35

$$R_n = 0.8 \times R \times \left(\frac{F_{bm}}{0.28 \times C_{dyn}}\right)^{1/3}$$

R_n Rigidity of nut [N/µm]
R Rigidity in accordance with dimensions table [N/µm]
F_{bm} Average operating load [N]
C_{dyn} Dynamic load rating from dimensions table [N]

The axial rigidity of a feed system includes that of the support bearing and assembly table. The total rigidity should be noted with care when configuring the system.

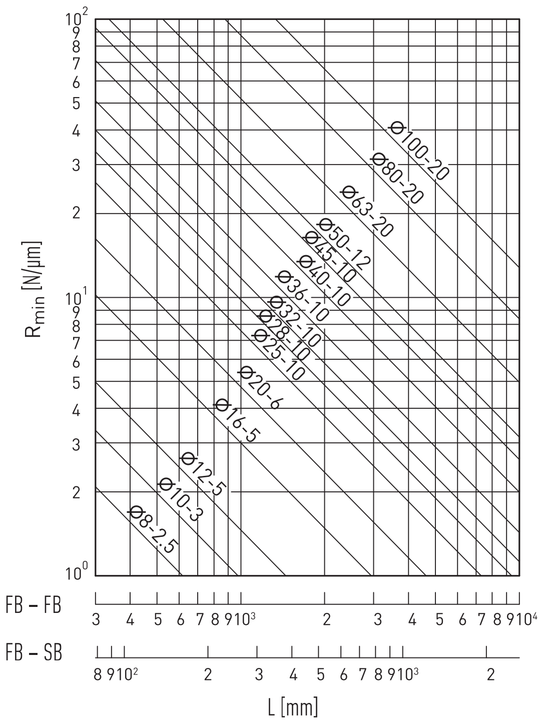
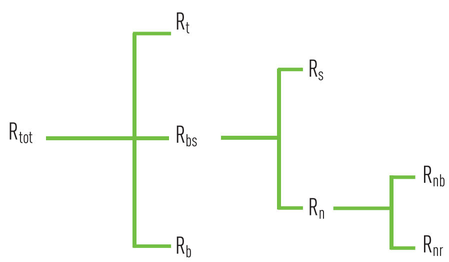


Fig. 3.24 **Rigidity diagram for ballscrews**



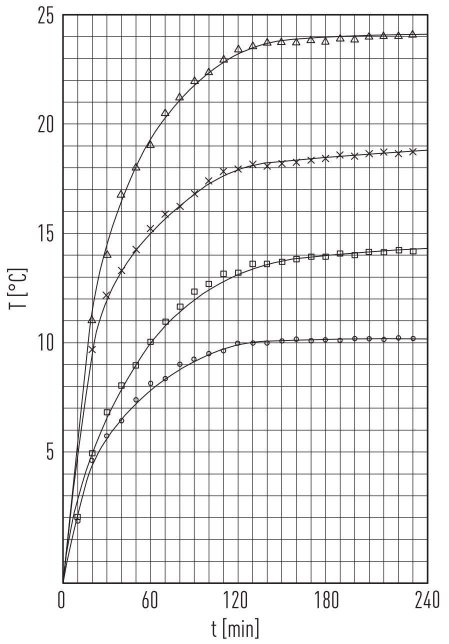
L Length of spindle [mm]
R_{min} Minimum rigidity of spindle [N/µm]
R_{tot} Total rigidity of feed system
R_t Rigidity of assembly table
R_b Rigidity of support bearing
R_{bs} Rigidity of ballscrew
R_s Rigidity of threaded shaft
R_n Rigidity of ballscrew nut
R_{nb} Rigidity of balls and ball track
R_{nr} Rigidity of nut/shaft system with radial load

Fig. 3.25 **Rigidity factors for feed systems with ballscrews**

3.8.8 Thermal expansion

An increase in temperature in ballscrew shafts during operation impacts on the accuracy of a machine's feed system, since the threaded shaft extends through the thermal stress.
The following factors affect the temperature increase in ballscrews:
1) Preload
2) Lubrication
3) Stretching of the shaft

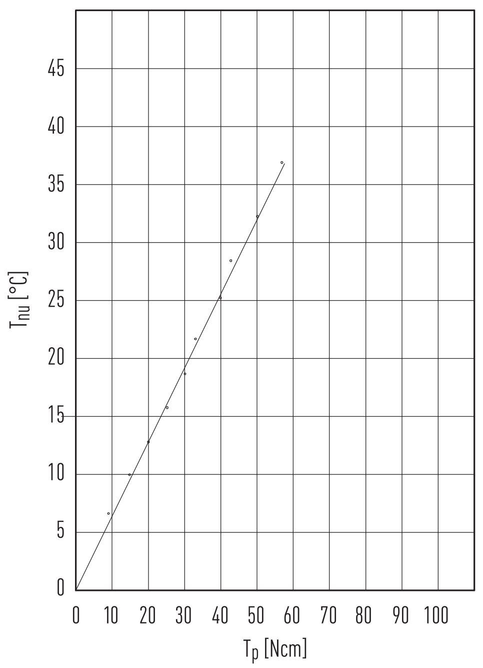
Fig. 3.26 shows the relationship between operating speed, preload and temperature increase. Fig. 3.27 shows the temperature increase in the nut depending on idle torque.



Ballscrew data R40-10-B2-FDW

Δ = 1,500 rpm with 2,000 N preload
× = 1,500 rpm with 1,000 N preload
□ = 500 rpm with 2,000 N preload
○ = 500 rpm with 1,000 N preload

Fig. 3.26 **Relationship between operating speed , preload and temperature increase**



Spindle diameter	R40
Lead	10
Ball diameter	6,35
Circuits	2,5 × 2
Speed	2.000 U/min
Running time	1,5 sec
Stop time	1 sec
T_{nu}	Temperature in nut [°C]
T_p	Idle torque [Ncm]

Fig. 3.27 **Relationship between temperature increase in the ballscrew and idle torque**

The thermal expansion of the threaded shaft can be determined using formula F 3.36.
The expansion can be compensated by stretching of the shaft. For further information please consult HIWIN.

F 3.36

$$\Delta L = 11.6 \times 10^{-6} \times \Delta T \times l_{s;total}$$

ΔL Thermal expansion of threaded shaft [mm]
ΔT Temperature increase in threaded shaft [°C]
$L_{s;total}$ Shaft length + shaft end (left/right) [mm]

3.9 Material and heat treatment

3.9.1 Materials of the components

Table 3.19 **Material overview**

	Material numbers according to DIN EN 10027		
Components	**Rolled ballscrews**	**Peeled ballscrews**	**Ground ballscrews**
Shaft	1.1213	1.1213 1.7225	1.7228
Nut [1)]	1.6523 [1)]		
Ball	1.3505		

[1)] Special nuts 16MnCr5B

3.9.2 Heat treatment

Table 3.20 shows the hardness of the main components used in HIWIN ballscrews. The surface hardness of the ballscrew affects both the dynamic and the static load rating. The dynamic and static load ratings listed in the dimensions tables are based on a surface hardness equivalent to HRC 60. For surface hardnesses of less than this, the load ratings can be determined using the following calculation.

F 3.37

$$C_0' = C_0 \times f_{H0} \qquad f_{H0} = \left(\frac{H_R}{60}\right)^3 \leq 1$$

F 3.38

$$C' = C_{dyn} \times f_H \qquad f_H = \left(\frac{H_R}{60}\right)^2 \leq 1$$

With hardness levels f_H and f_{H0}

C_0' Corrected static load rating
C_0 Static load rating at 60 HRC
H_r Real hardness (HRC)

C' Corrected dynamic load rating
C_{dyn} Dynamic load rating at 60 HRC
H_r Real hardness (HRC)

Table 3.20 **Hardness levels of components used for HIWIN ballscrews**

Components	Hardening method	Hardness (HRC)
Shaft	Carburizing	58 – 62
Nut	Carburizing or induction hardening	58 – 62
Ball		62 – 66

3.10 Lubrication

HIWIN ballscrews can be lubricated with grease, semi-fluid grease or oil depending on the application. They are supplied preserved as standard and must never be taken into service without initial lubrication. For information about the initial greasing, amounts of lubricant and lubrication intervals, please consult the assembly instructions "Ballscrews".

Table 3.21 **Information about checking and topping up lubricant**

Lubrication method	Information about checking
Oil	Check oil level once a week and check oil for contamination If contaminated, we recommend changing the oil
Grease	Check grease for contamination every two to three months If contaminated, replace old grease with new grease Always replace grease on an annual basis