Technology

TECHNICAL INFORMATION

The following chapters only show a small portion of the wide field of applications for ball screws. In order to meet all the technical and commercial demands for such a wide variety of different tasks, a deep understanding of the technology of ball screws is absolutely necessary. We have collected extensive information about ball screws in this section of this catalogue and hope that you will find it useful.

Please be aware that although we edited this information as carefully as possible, we cannot be held responsible for missing or incorrect information.

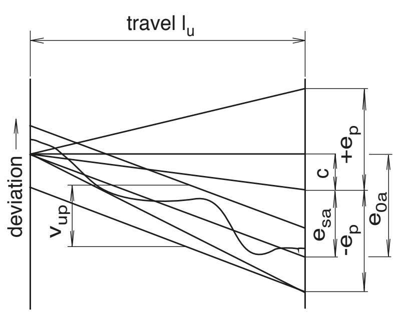
PRECISION (LEAD ERROR, FRICTION TORQUE, CRITICAL TOLERANCES)

Under the headline „precision" DIN / ISO standards are explained as they apply to ball screws, how accuracies are defined, and the acceptance or specification criteria derived from these standards.

- Lead accuracy
- Friction torque
- Roundness, concentricity and squareness of relevant surfaces

In general, all acceptance criteria should be reviewed and agreed upon between Steinmeyer and our customer. This is especially relevant for those applications where special demands are required, for example a lead accuracy of class 5 but friction torque variation consistent with accuracy class 1. In this example, Steinmeyer produces a ball screw with economical class 5 lead accuracy but with much lower friction torque variation.

LEAD ACCURACY PER DIN 69051 / ISO 3408



Both DIN and ISO standards use the following terms and definitions to describe lead accuracy. The corresponding JIS designations are given in parentheses:

- Lead compensation c is used to compensate lead errors resulting from thermal growth or pre-tensioning of the ball screw shaft (JIS: T).
- The permissible lead deviation e_p is an averaged lead error over the entire useful travel (JIS: E)
- The permissible lead fluctuation v_{up} over the entire useful travel is defined as the vertical distance of two straight lines parallel to the line representing e_p which enclose the entire lead error graph (JIS: e).
- The lead fluctuation v_{300p} represents the same for any 300 mm interval (JIS: e_{300}).
- And finally the lead wobble $v_{2\pi p}$ is the lead error within one revolution (JIS: $e_{2\pi}$).

Ball screws are normally categorized in accuracy classes, which not only define lead accuracy, but a number of different quality criteria like squareness and concentricity of mating surfaces, shaft straightness, friction torque fluctuation etc. Although this seems to be a trouble-free approach, ball screw users often choose to specify these other criteria by defining them in the source control drawing, while using ISO or DIN standard accuracy classes to only describe lead accuracy.
Steinmeyer uses accuracy classes 1, 3, 5, 7 and 10 per ISO / DIN standard. Accuracy classes 0, 2 and 4 are not contained in these standards, but have been added to match the JIS 1902 standard.

Positioning ball screws P
Transport ball screws T

The DIN standard differentiates between positioning ball screws and transport ball screws.

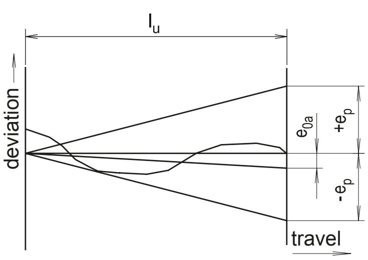
Positioning ball screws are normally used in high-precision applications (like machine tool) and are usually equipped with a ground ball thread.

Transport ball screws are predominantly used for travelling and moving applications. Typical applications are axes for handling systems. Ball thread of such screws is usually rolled or whirled.
Per DIN standard the tolerance classes for positioning ball screws are described as "P" while the transport ball screws classes are described as "T".

Steinmeyer uses this designation as per DIN:

P0 – P5 for positioning ball screws
T5 – T10 for transport ball screws

Lead error over the entire useful travel



l_u [mm]		Limit e_p for the average lead error e_{0a} [µm]							
		Tolerance class							
from	to	P0	P1	P2	P3	P4	P5	T7	T10
-	200	3	5	7	10	15	20		
200	315	4	6	8	12	18	23		
315	400	5	7	9	13	19	25		
400	500	6	8	10	15	21	27		
500	630	6	9	11	16	23	30		
630	800	7	10	13	18	27	35		
800	1000	8	11	15	21	31	40	$e_p = 2 \cdot \frac{l_u}{300} \cdot V_{300p}$	
1000	1250	9	13	18	24	35	46		
1250	1600	11	15	21	29	42	54		
1600	2000	-	18	25	35	50	65		
2000	2500	-	22	30	41	59	78		
2500	3150	-	26	36	50	72	96		
3150	4000	-	32	44	62	88	115		
4000	5000	-	-	-	76	108	140		
5000	6300	-	-	-	92	131	170		

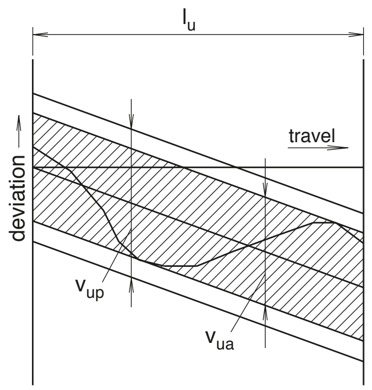
TECHNICAL TIP

The lead error e_p is defined via a straight line, which represents the optimum approximation of the actual lead error graph. This line will then be shifted parallel so it crosses the origin of the axis. This means that there can be a deviation at the beginning of the travel. Also, the lead error at the end of travel can actually be greater than the maximum permissible error as long as the average lies within the tolerance limits!

Nevertheless, a definition of the lead accuracy via e_p is a stronger criteria than the commonly used deviation per 300 mm, be-cause the error may not be accumulated. For example, the maximum permissible lead error e_p of a screw 900 mm long is less than three times the permissible error in 300 mm.

PRECISION

Lead error fluctuation over entire travel



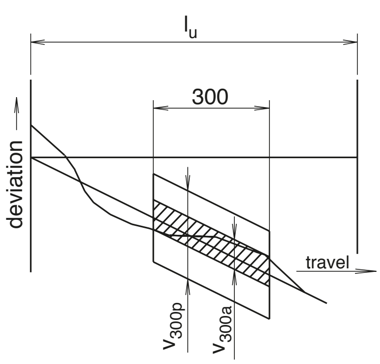
l_u [mm]		Limits v_{up} for the variation v_{ua} [µm] Accuracy class [1 n/a for rolled and whirled ball screws]							
from	to	P0	P1	P2	P3	P4	P5	T7	T10
-	200	3	5	7	10	15	20[1]	-	-
200	315	4	6	8	12	18	23[1]	-	-
315	400	4	6	8	12	19	25[1]	-	-
400	500	4	7	8	13	20	26[1]	-	-
500	630	4	7	8	14	22	29[1]	-	-
630	800	5	8	9	16	24	31[1]	-	-
800	1000	6	9	10	17	26	35[1]	-	-
1000	1250	6	10	11	19	29	39[1]	-	-
1250	1600	7	11	13	22	33	44[1]	-	-
1600	2000	-	13	15	25	38	51[1]	-	-
2000	2500	-	15	18	29	44	59[1]	-	-
2500	3150	-	17	21	34	52	69[1]	-	-
3150	4000	-	21	25	41	62	82[1]	-	-
4000	5000	-	-	-	49	74	99[1]	-	-
5000	6300	-	-	-	58	88	119[1]	-	-

TECHNICAL TIP

The fluctuation of the lead error is defined by two lines parallel to the line representing the average lead error, which include the entire lead error graph.

The lead error fluctuation is applicable to precision ground positioning screws in accuracy classes 0 - 5 only. It does not apply to general purpose screws which are typically either rolled or whirled in classes 5 - 10.

Lead error fluctuation per 300 mm



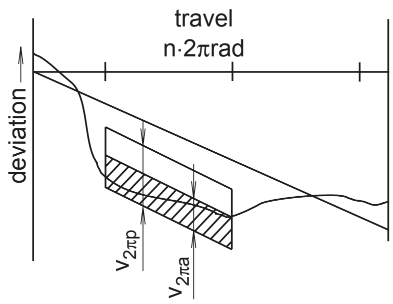
The variation over 300 mm is the most common definition. However it is also important to know whether the allowed variation is cumulative, or not. If the allowable variation is specified as "cumulative" then the permitted error over the full travel of a screw can result in a total error which is equivalent to the next lower accurancy class.

Example:
Stroke 900 mm, accuracy class P3, 3 x 300 mm would result in a total deviation of 3 x 12 µm = 36 µm. This result is equivalent to the total error of accuracy class P5 (35 µm)! However, the admissible deviation e_p for such a screw in accuracy class P3 is 21 µm.

Limits v_{300p} for the fluctuation v_{300a} [µm] Accuracy class							
P0	P1	P2	P3	P4	P5	T7	T10
4	6	8	12	18	23	52	210

Lead error fluctuation over one revolution (lead wobble)

The variation of the lead within one revolution, the so-called wobble-error usually is sinusoidal. Here the DIN standard allows relatively coarse tolerances. If needed, this tolerance can be restricted.



Limit $v_{2\pi p}$ for the lead error $v_{2\pi a}$ [µm] [1 n/a to rolled or whirled ball screws]							
Accuracy class							
P0	P1	P2	P3	P4	P5	T7	T10
3	4	5	6	7	8[1]	-	-

Ball screws are globally defined by the ISO standard 3408, which is compatible to DIN 69051 to a large extent. In Japan the JIS 1902 standard is valid while in the USA the ANSI standard B92.1 is still occasionally used.

Concerning nut dimensions, the ISO standard has achieved acceptance, at least where nuts are equipped with the space-saving internal deflection system.

Quite often old data or specifications from other standards are used to define a ball screw. Therefore, please be cautious when trying to directly compare parameters such as load capacity or stiffness.

While JIS and ISO standard are quite similar when describing accuracies or load capacities, the ANSI standard is quite different, especially regarding load capacity.

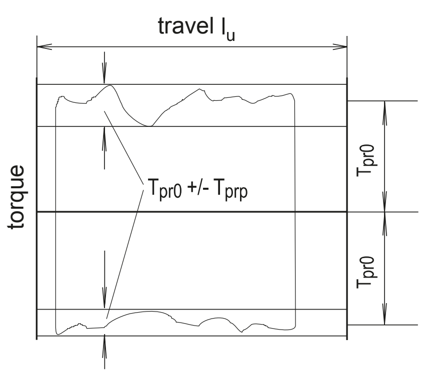
Steinmeyer exclusively uses the ISO definitions.

PRECISION

TOLERANCE OF TORQUE

The friction torque tolerance is solely defined for preloaded ball screws with double nuts and is mainly affected by the accuracy class, preload value and diameter/length ratio of a ball screw. The permissible torque fluctuation is given in % based on the nominal idling torque of a preloaded ball screw.

Variation of friction torque



Friction torque (nominal) T_{pr0} [Ncm] from	to	Friction torque variation in [%] For short screws (L ≤ 40 · d_N) For longer screws see table below! Accuracy class P0	P1	P3	P5	T7
5	10	40	45	50	60	-
10	20	35	40	45	50	-
20	40	30	35	40	50	-
40	60	25	30	35	40	-
60	100	20	25	30	35	40
100	250	15	20	25	30	35
250	630	10	15	20	25	30
630	1000	-	-	15	20	30

Friction torque (nominal) T_{pr0} [Ncm] from	to	Friction torque variation in [%] For long screws (L ≥ 40 · d_N) Accuracy class P0	P1	P3	P5	T7
5	10	-	-	-	-	-
10	20	50	50	60	60	-
20	40	40	40	50	60	-
40	60	35	35	40	45	-
60	100	30	30	35	40	45
100	250	25	25	30	35	40
250	630	20	20	25	30	35
630	1000	-	-	20	25	35

The values of the interclasses can be determined by interpolating.

TECHNICAL TIP

Steinmeyer can provide a friction torque chart for preloaded ball screws upon request (for an extra charge!). The test method described in the ISO / DIN standard is to run the ball screw, with a thin film of oil, at 100 rpm and without wipers. Other test methods can be agreed upon. The values shown here apply to the test method per ISO / DIN standard.
Tolerances for the friction torque of preloaded single nuts are specified individually.

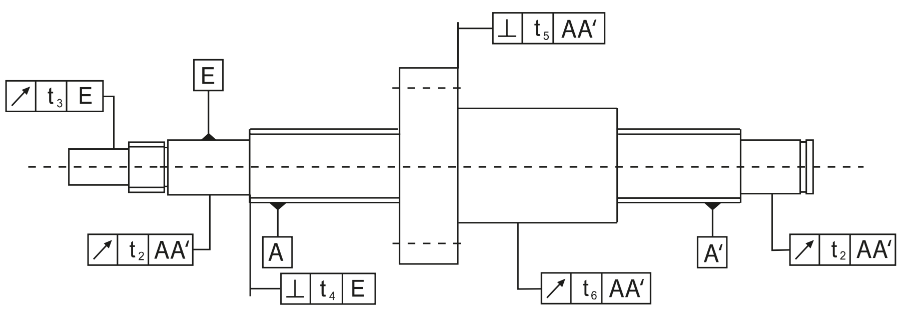
Aerospace ball screws are often tested for idling torque, although they are normally not preloaded. This test serves to determine proper operation of wipers and seals. This test can be part of the ATP; it is normally done with the ball nut greased and ready for shipment.

GENERAL GEOMETRIC TOLERANCES (ACCEPTANCE CRITERIA)

The values listed below represent general rules for run-out tolerances of ball screws. For specific applications tolerances may vary.

Tolerances of run-out and perpendicularity

Steinmeyer recommends supporting the screw using the outside diameter for all inspection of geometric tolerances. This will ensure optimum repeatability of the measurement. In some cases the center holes are used as reference.



Run-out t_2

Nominal-∅ d_N [mm]	Run-out t_2 [µm] Accuracy class P0	P1	P2	P3	P4	P5	T7	T10
3 - 6	5	7	7	8	10	-	-	-
8 - 10	5	7	7	9	10	10	20	-
12	5	7	8	9	10	10	20	-
16 - 20	5	7	9	10	12	13	20	-
25 - 32	6	8	10	11	12	14	25	-
36 - 50	7	9	12	13	15	16	25	-
60 - 125	8	10	13	14	16	18	25	-

Run-out t_3

Nominal-∅ d_N [mm]	Run-out t_3 [µm] Accuracy class P0	P1	P2	P3	P4	P5	T7	T10
3 - 6	3	4	6	7	8	10	-	-
8 - 10	4	5	7	8	9	11	12	15
12	4	5	7	8	9	11	13	17
16 - 20	4	6	8	9	10	12	15	18
25 - 32	5	7	9	10	12	13	16	19
36 - 50	6	8	11	12	14	15	18	21
60 - 125	7	9	12	13	15	17	20	23

PRECISION

Perpendicularity t_4

Nominal-∅	Perpendicularity t_4 [µm]							
	Accuracy class							
d_N [mm]	P0	P1	P2	P3	P4	P5	T7	T10
3 - 6	2	2	2	3	3	3	-	-
8 - 10	2	2	2	3	3	4	5	7
12	2	2	2	3	3	4	6	8
16 - 20	2	3	3	4	4	5	7	9
25 - 32	2	3	4	4	4	5	7	9
36 - 50	2	3	4	4	4	5	7	9
60 - 125	3	4	5	5	6	7	10	13

Perpendicularity t_5

Nominal-∅	Perpendicularity t_5 [µm]							
	Accuracy class							
d_N [mm]	P0	P1	P2	P3	P4	P5	T7	T10
3 - 6	6	7	8	9	9	10	-	-
8 - 10	6	7	8	9	9	10	15	-
12	6	7	8	9	9	10	20	-
16 - 20	7	8	9	10	10	12	25	-
25 - 32	7	8	9	10	10	12	32	-
36 - 50	8	9	10	10	12	13	32	-
60 - 125	9	10	11	12	13	15	40	-

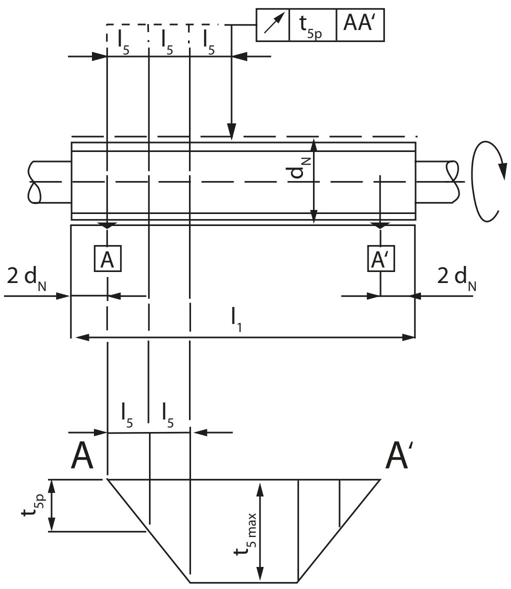
Run-out t_6

Nominal-∅	Run-out of pilot diameter t_6 [µm]							
	Accuracy class							
d_N [mm]	P0	P1	P2	P3	P4	P5	T7	T10
3 - 6	5	6	7	8	9	10	-	-
8 - 10	5	6	7	8	9	10	20	-
12	5	6	7	8	9	10	20	-
16 - 20	5	6	7	8	9	10	20	-
25 - 32	5	6	7	8	9	10	20	-
36 - 50	6	7	8	8	10	11	25	-
60 - 125	7	8	9	10	12	13	32	-

Steinmeyer's specified parameters for concentricity and perpendicularity are considerably lower compared to the DIN values.

Run-out tolerances

Run-out Tolerance t_{5p} of the shaft outside diameter for the length l_5 (the shaft straightness with reference to AA') - according to DIN 69051 part 3 or ISO 3408-3.



d_N from	d_N to	l_5	Run-out tolerance t_{5p} for the length l_5 [µm] – Accuracy class P1	P3	P5	T7	T10
3	12	80	20	25	32	40	80
12	25	160					
25	50	315					
50	100	630					
100	200	1250					

l_1 / d_N from	l_1 / d_N to	Run-out tolerance $t_{5\,max}$ for the length $l_1 \geq 4l_5$ [µm] – Accuracy class P1	P3	P5	T7	T10
	40	40	50	64	80	80
40	60	60	75	96	120	240
60	80	100	125	160	200	400
80	100	160	200	256	320	640

TECHNICAL TIP

Steinmeyer recommends supporting the screw by using V-blocks for all inspection of geometric tolerances. This will ensure optimum repeatability of the measurement. If necessary, dual gages can be used to measure the concentricity of two surfaces with respect to one another, e.g. the concentricity of drive journal and bearing journal. This measuring method includes the perpendicularity and run-out tolerance of the nut already.

optiSLITE

Optimized ball screws for best running features.

Microscopic irregularities on the surface of the spindle thread can cause vibration and uneven running. Through the use of innovative production technologies, the smoothness of the thread surface of miniature ball screws can be significantly improved, whereby the operating characteristics of the optiSLITE ball screws are remarkably improved.

The improved running characteristics are achieved by increased material contact. The material contact area Rmr (c) indicates the percentage of material filled path lengths depending on the depth of cut "c". The plateau-like surface of the optiSLITE technology provides clean, smooth running, while offering improved lubricating properties.

More information about optiSLITE-Technology on page 55.

PRECISION

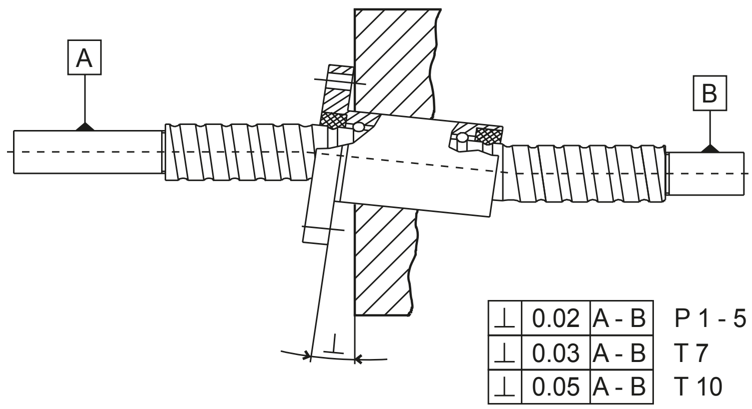
MOUNTING TOLERANCES

TECHNICAL TIP

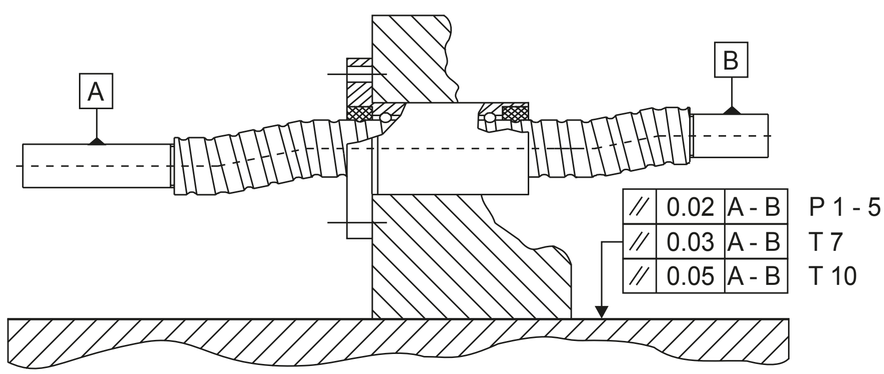
Steinmeyer recommends maintaining the mounting tolerances shown on this page. Optimum alignment of the screw with the guideways and square and concentric mounting of the nut will ensure proper operation of the drive system and long life of the ball screw.

After installation, check that the screw spins freely and without excessive friction over its entire travel. If there is any binding or considerable increase in effort necessary to turn the screw, especially near the support bearings, this indicates the alignment of the screw and the guideways should be improved. Binding indicates excessive side loads which can and will shorten the service life of the ball screw unless corrected.

Perpendicularity



Parallelism



PRELOAD AND RIGIDITY

NUT DESIGNS

This section deals with the various nut types and their preload. We also explain the different rigidity (stiffness) values.

TECHNICAL TIP

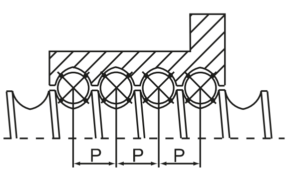
Preload primarily serves to eliminate play. But at the same time preload increases rigidity, which means the displacement of the nut under load is reduced.

Another reason why preload may be necessary is to prevent balls from skidding during high acceleration, or to ensure better load distribution if side loads on the ball nut cannot be avoided.

There are a number of ways to preload a ball nut. For a discussion of preload and its effects we have to first distinguish between preload with two contact points per ball (one in the nut, one in the shaft), and preload with 4-point contact (two contact points in the nut and two in the shaft), which requires the use of a „gothic arch" profile.

Steinmeyer always uses a gothic arch profile, so both kinds of preload are possible.

Single nut with 4-point-contact



Single nuts, as defined by Steinmeyer, are one-piece nuts without any shift or offset in their I.D. ball thread. They can only be preloaded by ball oversize and will always have 4-point contact. This differentiates them from all other preloaded nuts and their 4 point contact has an impact on technical data and some calculations.

4-point contact influences the kinematics of the balls with the following results:

- Efficiency of a single nut with preload is always somewhat lower than that of a double nut
- Single nuts with preload show a more pronounced reaction to manufacturing tolerances, which is why they are normally not suitable for very long screws

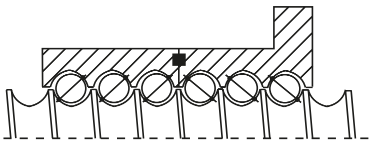
Caution: these statements are only true for single nuts with preload. Without preload, single nuts have 2-point contact with the same ball kinematics as double nuts!

Advantages of the single ball nut are:

- Cost effectiveness
- Compact envelope
- No unloading of balls when subjected to peak loads (see preload chart)

PRELOAD AND RIGIDITY

Double nut (UNILOCK)



Double nuts from Steinmeyer do not depend on the usual spacer ring or other hardware to separate the two nuts. Our patented UNILOCK coupling ensures a robust connection of the two nut halves. Thus Steinmeyer's double nut is almost as compact and stiff as a single-piece design. Moreover, it prevents radial slippage of the two halves, so the UNILOCK double nut cannot be misaligned by rough handling. The coupling is rugged and absolutely tight to prevent loss of lubricant.

Advantages of a double nut are:

- 2-point contact for higher rigidity and efficiency
- Simplified production compared to lead-offset nuts when the nut is long and helix angles are large
- Easier factory setting of preload without changing of balls, thus economical

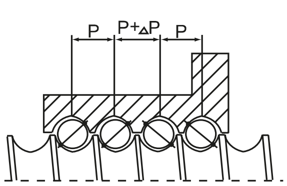
Disadvantages are:

- Longer nut body compared to single nuts, and somewhat higher cost
- As in all nuts with 2-point contact, the maximum thrust should not exceed 2.8 times the preload

TECHNICAL TIP

Steinmeyer labels all ball nuts that have 2-point ball contact a „double nut", regardless whether the nut body is made of two pieces or one. This definition makes sense because both of these nuts have very similar properties, and because all life calculations for these nuts are done in the same way. 4-point contact ball nuts on the other hand require different calculations, specifically how preload is handled in the life calculations.
Steinmeyer's patented UNILOCK coupling of two-piece double nuts results in a connection that is nearly as stiff and compact as a single-piece nut. This is why we do not distinguish between single-piece double nut design (the lead-offset nut) and a two-piece design (UNILOCK double nut) in this catalog. Steinmeyer will select whatever design is most suitable for the particular size of ball nut.All nuts with 2-point contact can be recognized by the „5" in the 2nd number of their series designation (for example 1516, 3526 etc.).

Lead-offset nut



When preloading a UNILOCK double nut, the two halves are rotated relative to each other until the balls are slightly compressed. Thus the threads of the second nut are no longer directly „in phase" with the threads of the first nut. There is a slight offset between the two nut halves.

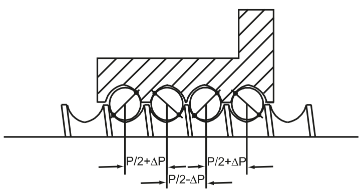
The lead offset nut uses the same principle but the offset is created during machining of the threads in a nut made from a single piece of steel. Fine tuning of the preload is then accomplished by selecting „oversized" balls like in a single nut. Due to the offset, balls have 2-point contact.

Advantage: ■ Slightly more compact than a double nut (shorter length)

Disadvantage: ■ More threads have to be machined from one end of the nut. Especially in nuts with many turns and large helix angle, grinding I.D. threads can be difficult or even impossible

NB: Steinmeyer often uses hard turning for such long nuts, eliminating the problems associated with long and relatively weak grinding arbors.

Pitch-offset nut



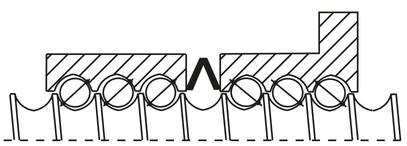
Like the lead-offset nut, the pitch-offset nut uses a shift machined in the I.D. threads to enable 2-point contact of balls. The only difference is that the pitch-offset nut uses a two-start thread and the offset is between the two starts.

Each thread start has its own set of balls and ball returns.

Advantage: ■ Pitch-offset nuts are extremely compact

Disadvantage: ■ Pitch-offset preload can only be used in nuts with two (or more) starts

Spring preloaded double nut



Almost exclusively used with miniature ball screws, the spring preload maintains perfectly constant preload regardless of wear and manufacturing tolerances. Two nuts are installed in a housing. One is fixed, while the second one can slide longitudinally (a pin keeps it from rotating). A spring located in between the two nuts keeps the preload constant.

The advantages are obvious:

- Manufacturing tolerances will have no impact on preload and friction
- Preload can be very light and the ball nut will never have play regardless of wear etc.

There are also some disadvantages:

- The double nut with its housing is bulkier and more expensive
- The slip-fit of the second nut makes alignment more challenging
- The maximum thrust is limited by the spring - when exceeded, the spring will collapse causing play in the nut

TECHNICAL TIP

The spring preloaded nut excels when it is not possible to keep manufacturing tolerances low enough to ensure very low and constant friction. This is the case when near zero friction is necessary while at the same time absolutely no play is acceptable, or if the screw is extremely long.

Please be aware that the spring preloaded nut has a maximum operating force equal to about 2/3 of the preload. We can incre-ase preload to some degree upon request, but if peak loads exceeding the preload force cannot be avoided, then a single nut with rigid preload is preferred (miniature screw series 1112, 1214, 1412).

PRELOAD AND RIGIDITY

STIFFNESS

TECHNICAL TIP

Higher preload results in increased rigidity. But rigidity increases only with the cubic root of preload. To compare stiffnesses based on different preload settings, multiply rigidity values by the cubic root of the ratio of the preloads. For example:

$$R_2 = R_1 \cdot \sqrt[3]{\frac{F_{pr2}}{F_{pr1}}}$$

where F_{pr1} and F_{pr2} are the preload settings to be compared, and R_1 and R_2 are the corresponding rigidities.

The rigidity of a ball screw not only has a strong impact on positioning accuracy, but also plays an important role concerning the dynamic behavior of a linear drive. The importance to the latter is normally underestimated.

- Rigidity according to ISO 3408 or DIN 69051 is a value labeled $R_{b/t}$. This value is obtained from theoretical elastic deformations obtained from the theory of Hertzian pressure with the variables of track conformity, contact angle and the number of load carrying balls. $R_{b/t}$ is a relatively high number.
- When the deformation of the nut body (diameter widening, longitudinal expansion) is included, the value is labeled R_{nu}, and is already significantly lower than $R_{b/t}$.
- To obtain a real value, the theoretical vaule of R_{nu} has to be corrected further, since not all balls carry the same load. Depending on the accuracy grade of the ball screw, the correction factor varies. The reduced value, which is closer to reality and measurable, is designated $R_{nu,ar}$.

Steinmeyer publishes only $R_{nu,ar}$ values in our catalog and on our website. Please be sure to compare these values only to similarly defined rigidities from other manufacturers, and also ensure that the comparison is based on the same preload value you want to use (see technical tip regarding the impact of preload on rigidity).

Rigidity of the nut and preload

Besides number and size of loaded balls, track conformity and contact angle, the main driver for the rigidity of the nut is preload. However, the possibility to significantly increase rigidity by raising preload is very limited because rigidity increases only as the cubic root of preload. But the increase in heating is directly proportional plus the reduction of service life is substantial since preload increase is raised to power three in the life equation.

This is why preload should not be set too high: For a nut with 2-point contact, 10% of the dynamic load capacity is a reasonable number. Nuts with 4-point contact should not be preloaded to more than approximately 8% of their dynamic capacity.

Steinmeyer publishes rigidity values for these preload settings:

- 10% C_a for nuts with 2-point contact (series x5xx)
- 8% C_a for nuts with 4-point contact (series x4xx)

Make sure when comparing with other manufacturer‘s rigidity data to check not only the definition of terms but also that the comparison is done using the same preload.

TECHNICAL TIP

According to the ISO / DIN standard, the nut theoretical stiffness R_{nu} is converted into the actual nut stiffness $R_{nu,ar}$ by applying a correction factor which depends on the accuracy class of the ball screw. For example a ball screw with accuracy class 1 would be more rigid than a screw of otherwise same design with an accuracy class 5. The reason for this is that better geometric tolerances of a more accurate screw, e.g. the lead wobble, result in more even ball loading.

Upon request (and extra cost), Steinmeyer will provide rigidity test protocols with every screw.

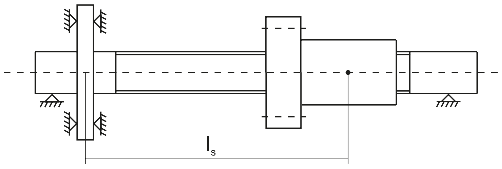
ETA+ for better rigidity - static and dynamic

ETA+ ball screws are not only significantly more rigid in a classic sense, which only considers the elastic deformation of a stationary screw under varying axial loads. They also have a much better „dynamic stiffness", which means that the screw has a „linear behavior" and delivers an axial movement perfectly proportional to its rotation, regardless of thrust, speed and direction. Even reversing will not cause deviations from this.

This linearity is especially significant in closed-loop controls, allowing considerably increased dynamics with substantially reduced following error. Unfortunately, this is not yet reflected in current ISO or DIN standards.

The screw stiffness depends on the elasticity modulus of the screw material, cross sectional area of the screw and the unsupported screw length.

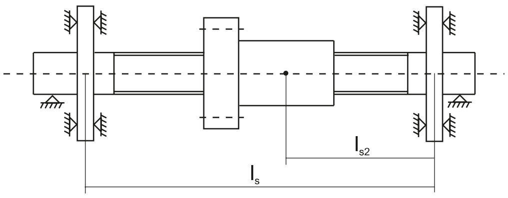
Fixed-free mounting



On fixed-free bearing configuration the screw stiffness is calculated as follows:

$$R_{s1} = A \cdot \frac{E}{l_s} \cdot 10^{-3}$$

Fixed-fixed mounting



This is the calculation of the rigidity for fixed-fixed mounting method:

$$R_{s2} = 2 \cdot A \cdot \frac{E}{l_{s2}} \cdot 10^{-3}$$

A: Screw cross section [mm²]

E: Young´s modulus [N/mm²]

(for DIN 1.1213 = 210,000 N/mm²)

PRELOAD AND RIGIDITY

Total rigidity of a linear drive

Elasticity of a linear drive includes elastic deformations of thrust bearings, ball screw shaft and ball nut. This total, based on the „inverse" equation below, is what matters to the performance of the drive. But only the nut stiffness is normally given in the technical data of a ball screw.

$$R_t = \frac{1}{\left(\frac{1}{R_{nu,ar}} + \frac{1}{R_s} + \frac{1}{R_b}\right)}$$

R_t: Total rigidity [N/µm]
$R_{nu,ar}$: Actual nut rigidity [N/µm]
R_s: Stiffness of the screw shaft [N/µm]
R_b: Axial stiffness of the thrust bearing [N/µm]

The rigidity of the thrust bearing can be obtained from the literature of the manufacturer. Rigidity of the screw shaft can be calculated from the elasticity modulus of steel (210,000 N/mm^2), the shaft's cross section and the length of the loaded portion of the shaft. For screws with large lead / diameter ratio, torsion also plays an important role and must be considered as well.

If you want to do the calculation of axial and torsional stiffness yourself, just use the nominal diameter of the ball screw minus the ball diameter to calculate the cross section and moment of interia. Or, contact us and we will do this for you.

TECHNICAL TIP

Installing thrust bearings at both ends of the screw yields four times the axial shaft stiffness compared to a single thrust bearing at one end and no support on the opposite end. A factor of 2 results because forces are transmitted through the shaft on both sides of the nut. A second factor of 2 also applies because the weakest point is now in the middle of the shaft rather than at the extreme end.Thrust bearings at both ends normally require pre-tensioning the screw, against the support bearings, in order to avoid compressive loads on the shaft from thermal expansion (which may cause buckling). Make sure to check the impact this additional force has on the bearing life.

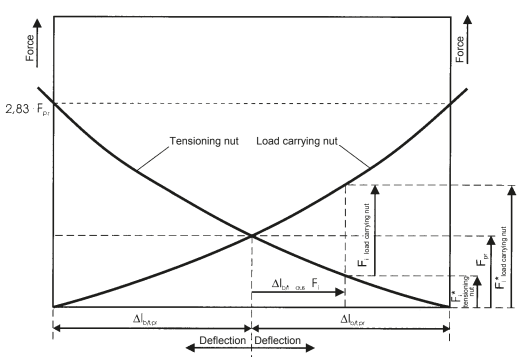
Linear drives with rotating nut and stationary screw allow a simple way of increasing torsional stiffness of the shaft by transmitting moments into the surrounding structure at both ends of the shaft. Then the same effect as described above applies to the torsional stiffness: A factor of 2 for twice the cross section, and again a factor of 2 for half the distance to the weakest point.

Ball screws are normally loaded with axial forces. Service life is determined by material fatigue in most cases. However, in some applications, abrasive or adhesive wear may cause a loss of preload and thus be considered a failure of the ball screw.
In the ISO 3408 or DIN 69051 standards, the math used to estimate ball screw life is based on material fatigue only. This is generally correct, but if the loading is very low and mean speeds are very high, this method may return an erroneous life estimation which cannot be reached in reality. If your application falls within this category, please consult our engineers.

ACCOUNTING FOR PRELOAD IN THE LIFE CALCULATION

In this step, we modify axial forces to reflect the impact of preload. The preload graph shows how internal forces in the ball nut are affected by external forces and preload.

Preload graph



The preload graph includes two lines which represent the forces and resulting deformation for both nut halves in a double (or pitch shift) nut. The center of the chart, where the two lines cross, shows the situation of the nut with no external forces. Both nut halves carry the same load - which is the preload F_{pr}.

If an external force is added, then the load in one nut increases, while it decreases in the other. The resulting difference of the two forces is equal to the external load.

The load in each nut can be read from the two lines and this load is to be used for life calculations. As a simple approximation, the nut with the higher load has to carry the preload plus approximately half of the external load. Using this approximation for F_{i^*} is accurate enough.

Calculating modified loads for nuts with 2-point-contact

$$F_i^* = F_{pr} + \frac{F_i}{2}$$

F_i^*: Modified external load [N]
F_{pr}: Preload [N]
F_i: External load [N]

As a quick and simple approximation for the internal load in nuts with 2-point contact, just add or subtract half of the external thrust F_i to or from the preload F_{pr}. Whether you call a certain thrust direction positive or negative does not matter. But you have to keep this direction orientation the same for the whole duty cycle. The nut with the highest load then determines service life.

SERVICE LIFE CALCULATIONS

Calculating modified loads for nuts with 4-point contact

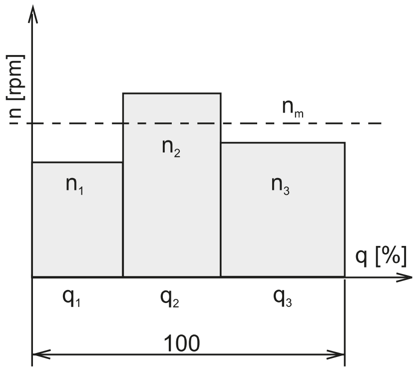
$$F_i^* = \frac{5}{4} F_{pr} + \left| \frac{F_i}{2} \right|$$

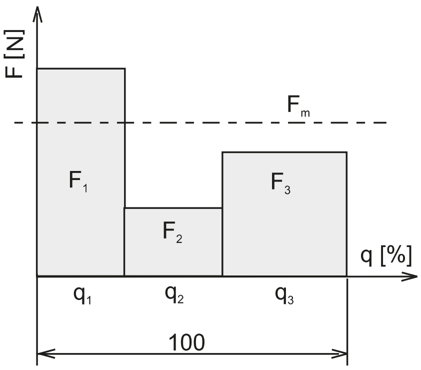
When calculating modified loads for nuts with 4-point contact, all external thrust has to be added regardless of load direction. The factor of 1.25 (5/4) is necessary to account for the fact that the balls have 4 contact points, causing fatigue much sooner than in a nut with 2 contact points. This approximation is accurate enough in cases with normal loads and duty cycle.

THE DUTY CYCLE AND ITS EQUIVALENT LOAD

To calculate the impact of actual duty on the fatigue life of the ball screw, it is necessary to convert the varying loads F_i into a mean load F_m which will have the same effect on life as the actual duty cycle. This mean load Fm is therefore called „equivalent" load.

To simplify things, the actual duty is normally divided into steps. Then the maximum thrust (or, if the ball screw is preloaded, its modified load), speed and duration are used to calculate the equivalent load. Instead of using time and speed $n_i \cdot q_i$, the absolute number of revolutions can be used in this calculation.





TECHNICAL TIP

The duty cycle of machine tools is normally provided as:
x_1% Rapid with thrust y_1
x_2% Roughing with thrust y_2
x_3% Finishing with thrust y_3
x_4% Dwell

etc. Such values can be modified to reflect preload and then directly entered into the equation for F_m.
If the duty cycle provided includes a detailed description of moves, the respective revolutions can be entered instead of $q_i \cdot n_i$

EQUIVALENT LOAD

$$F_m = \sqrt[3]{\frac{q_1 \cdot n_1 \cdot F_1^3 + q_2 \cdot n_2 \cdot F_2^3 + \ldots + q_n \cdot n_n \cdot F_n^3}{q_1 \cdot n_1 + q_2 \cdot n_2 + \ldots + q_n \cdot n_n}}$$

$$n_m = \frac{q_1 \cdot n_1 + q_2 \cdot n_2 + \ldots + q_n \cdot n_n}{q_1 + q_2 + \ldots + q_n}$$

The mean or equivalent load is calculated using the above equation.

F_m: Dynamic equivalent load [N]
n_m: Equivalent speed [rpm]
q_i : Time percentage [%]
n_i : Speed [rpm]
F_i : Thrust (which may need to be modified - due to preload - first) [N]

The service life estimate obtained from ISO / DIN calculations represents a dependable method to predict useful life of a ball screw under the conditions and for the duty cycles found in most machines. Other necessary conditions include proper lubrication, suitable protection from contamination, and operation of the ball screw at temperatures not exceeding 70° C.

Should the calculated life fall outside the range mentioned on the following page, or if there are special requirements or unusual conditions, please contact our application engineers.

TECHNICAL TIP

By definition, the dynamic load capacity is the load at which the ball screw will reach a useful life of 1 million revolutions.
More accurately, this means 90% of a sufficiently large number of identical ball screws will reach this useful life. This life is designated as L_{10} and is usually the basis for ball screw selection in normal machine design. For higher reliabilities than 90%, an additional correction factor must be applied. In regular machine design, such higher reliabilities are normally not needed because there are much greater uncertainties, for example in the estimate of thrust.

FATIGUE LIFE

$$F_m = \frac{C_a}{\sqrt[3]{\frac{L_{10}}{10^6}}}$$

F_m: Permissible mean load for a given dynamic load capacity and life [N]

$$C_a = F_m \cdot \sqrt[3]{\frac{L_{10}}{10^6}}$$

C_a: Dynamic load capacity required for a given mean load and life [N]

$$L_{10} = \left(\frac{C_a}{F_m}\right)^3 \cdot 10^6$$

L_{10}: Fatigue life for a given dynamic load capacity and mean load [revolutions]

SERVICE LIFE CALCULATIONS

The useful life L_{10} can be expected to be reached by 90% of a sufficiently large number of identical ball screws having a load capacity C_a, when subjected to the mean load F_m.

As an example, the life L_5, which is 62% of L_{10}, will be reached by 95% of the ball screws. Aerospace applications require a more detailed reliability analysis, for which the FMEA (Failure Mode and Effects Analysis) and fault trees are used to determine MTBF (Mean Time Between Failure) or MTBUR (Mean Time Between Unscheduled Removal).

Reliability (probability of survival)	90 %	95 %	96 %	97 %	98 %	99 %
Correction factor	1	0.62	0.53	0.44	0.33	0.21

TECHNICAL TIP

Calculation of the life expectancy is based, according to ISO / DIN standards, on Hertzian pressure. Theoretical load capacities are then modified with factors derived from experience. These corrected values allow very dependable predictions of ball screw life, under normal conditions, if the calculated life expectancy falls within this range:

$$3 \cdot 10^7 \leq L_{10} \leq 10^9 \quad \text{[revolutions]}$$

ANSI load capacity: According to ANSI ball screw standards, the dynamic load capacity is the load whereby the ball screw reaches a life of 1 million inches of travel (ISO/DIN: 1 million revolutions). For a direct comparison with ISO/DIN numbers, the ANSI load capacity must be converted as follows

LOAD CAPACITY ACCORDING TO ANSI STANDARD

According to the ANSI standard, dynamic load capacity of a ball screw is the load under which it will reach a life of 1 million inches of travel. If the lead is smaller than 1 inch, then the load capacity per ANSI definition is smaller than the load capacity of the same ball screw expressed according to ISO / DIN standard. This is because the ANSI load capacity defines a load for which the resulting life in this case (lead less than 1 inch) is greater than 1 million revolutions.

The opposite holds true for a ball screw with lead larger than 1 inch. For such a case, the ANSI load capacity will be higher than the ISO load capacity, although the ball screw itself is identical!

To compare load capacity of a ball screw per ANSI standard to one per ISO / DIN standard, a conversion must be made. This equation will also convert lbf into Newtons:

$$C_a = P_i \cdot 4.45 \cdot \sqrt[3]{\frac{25.4}{P}}$$

C_a: DIN / ISO-load capacity [N]

P_i: Load capacity per ANSI [lbf]

P: Lead [mm]

TECHNICAL TIP

Preload is usually defined as a percentage of dynamic load capacity. Nuts with 4-point contact normally have a preload of 5 - 8% of dynamic load capacity, while for nuts with 2-point contact it is 8 - 10%. Keep in mind this refers to ISO/ DIN load capacity only. If the ANSI standard is used, then the percentages may change!

MAXIMUM LOAD (LIMIT LOAD)

There are five ways a ball screw may fail due to overload:

- Excessive dynamic loading, which means the screw makes too many revolutions under a certain load resulting in material fatigue. This can be avoided by selecting a ball screw with sufficient dynamic load capacity (or by reducing the number of revolutions and/or reducing the load). This is the subject of the load capacity discussion.
- Exceeding the static load capacity, which causes instant and permanent damage to the ball screw due to brinelling of balls and races, and prevents any further normal operation of the ball screw. Static load capacities are listed as technical data.
- Buckling of the shaft under compressive load. Buckling load value depends on bearing method and free length of the loaded ball screw shaft.
- Failure of the nut body or of the bolts that connect it to the slide. This may happen even before the static capacity is reached. Safe loads are discussed on the following pages.
- Radial loads. It means the load capacities given in this catalogue apply only to pure axial loading. As there are always tolerances in the alignment of bearings and linear guideways, there may be a small amount of radial force, which should be minimized. Under normal conditions, a radial load less than 5% of the minimum axial load will not cause any problems. When considering a ball screw for use under radial load, please consult Steinmeyer engineers.

TECHNICAL TIP

A reasonable load for a ballscrew, which may be sustained for significant travel, is about 10% of its dynamic capacity.
A mean load of 10% of its dynamic capacity results in a theoretical life of 1 billion revolutions, which is the upper limit of the range where the life equation is valid. Mean loads of a reasonably sized ball screw will therefore be somewhat higher than this, but normally not exceed 20% of its dynamic capacity.

For short peak loads, the loading may be higher, but normally the loading of a ball nut with 2-point contact should not exceed 2.8 times the preload. And preload is around 5% - 10% of dynamic capacity.

As a rule of thumb, this all means the load range for a ball screw is really about 10% to 30% of its dynamic capacity.

BUCKLING

There are several analytical ways to demonstrate safety from buckling. In machine design, the most frequently used is a simple calculation using formulas based on Euler equations.

Other, more accurate methods include non-linear FEM analysis and more involved mathematics. These methods are normally used in aerospace applications, where excess safety margins are not possible due to weight limitations. Please contact us if you require such an analysis.

On the following page we describe a simple form of buckling analysis.

MAXIMUM LOAD (LIMIT LOAD)

TECHNICAL TIP

Because of the logarithmic scale on the chart below, data for long screws or screws with large diameter may be difficult to read. You may prefer to use this equation instead:

$$P_B = \frac{m \cdot d_N^4}{l_s^2} \cdot 10^4$$

P_B: Buckling [N]

d_N: Nominal-Ø [mm]

l_s: unsupported screw length [mm]

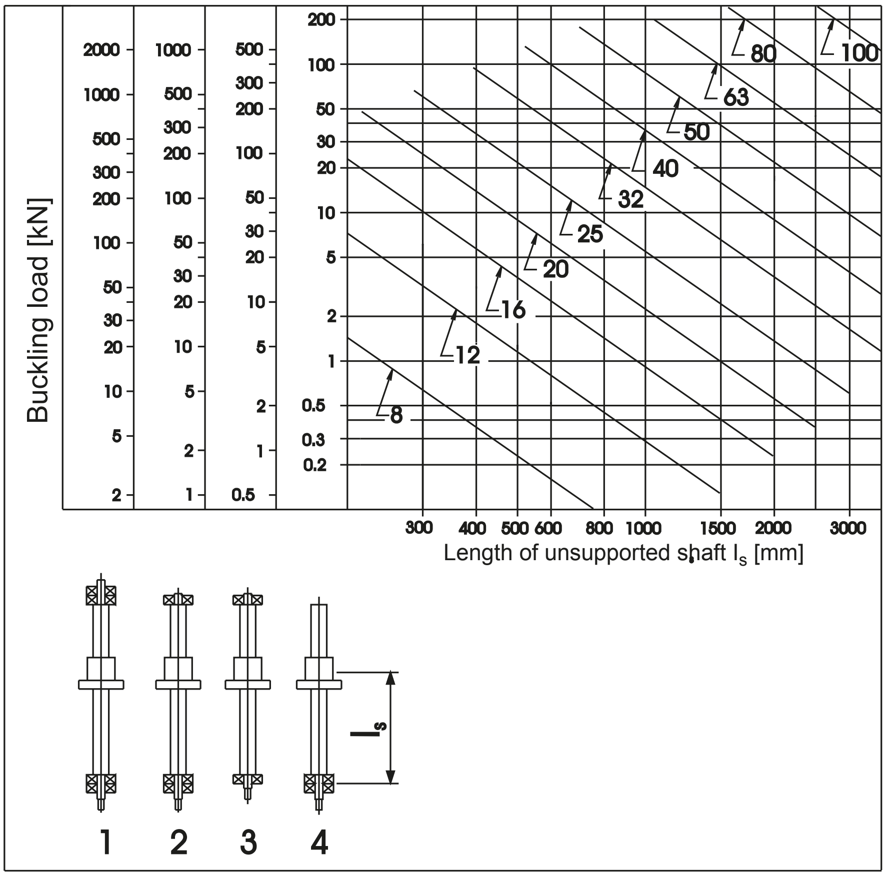
m: bearing coefficient

The following factors refer to the bearing methods shown in the chart. Choose the appropriate one and use it as the variable m in the equation to the left:

Fixed - fixed (1):	22.4
Fixed - supported (2):	11.2
Supported - supported (3):	5.6
Fixed - free (4):	1.4

For safety reasons, a factor of 0.5 should be applied to the buckling load obtained.

$$F_{max} = 0.5 \cdot P_B$$



FRACTURE LOAD

Some ball screws cannot be loaded all the way to their static capacity. Screws with high dynamic load capacity (which might be selected to obtain a long enough service life at a much lighter load) will necessarily have a high static capacity. But the term „static capacity" is misleading, since the ball screw may actually fail due to fracture of the nut flange, nut body, or connecting bolts before reaching this load!

Additional limitations (e.g. buckling) may apply

Maximum loads for flange nut according to DIN 69051					Standard nut	UltraThrust nut
DIN 69051						
Nominal diameter [mm]	DIN 69051 bolt pattern	Dynamic bolt tension Fb* [kN]	Static bolt tension Fb* [kN]	Bolt torque [Nm]	Maximum permissible axial load [kN]	Maximum permissible axial load [kN]
5/6	4xM3	5	20	1.5	C_{0a}	
8	4xM3	5	20	1.5	C_{0a}	
10	4xM4	7	28	3	C_{0a}	
12	4xM4	7	28	3	7	
16	6xM5	12	40	6	12	
20	6xM6	16	63	10	16	
25	6xM6	16	63	10	16	
32	6xM8	32	100	25	32	
40	8xM8	40	150	25	40	
50	8xM10	80	225	49	80	120
60/63	8xM10	80	225	49	80	180
80	8xM12	125	320	86	125	200
100	8xM12	125	320	86	125	200-250
100	8xM16	250	630	210	250	250
125	8xM16	250	630	210	150[1]	300
125	8xM16	250	630	210	250[2]	350

[1] Flange 25 mm thick
[2] Flange 30 mm thick
* Bolts DIN ISO 4762, Strength 8.8 (90% load, safety factor 0.8, μ = 0.14)

Please note: The highest permissible load is the minimum of static capacity C_{0a} (to prevent brinelling) and fracture load (to prevent failure of ball nut or bolts). Necessary condition for both is proper alignment and squareness of the mounting surfaces and also concentric applied load.

TECHNICAL TIP

The structural strength of ball screws for aerospace applications is first predicted using analytical methods including FEM analysis. In some cases, tests are performed in the development phase to demonstrate safety and avoid costly and time consuming redesigns once the qualification phase of the project has started.
For the final qualification, tests under exactly the same conditions as in the aircraft must be passed. These tests often require test rigs specific to the aircraft program, called „iron bird". The documentation of these tests becomes part of the aircraft qualification.
In addition to static tests (limit load, ultimate load), fatigue tests are conducted. In these fatigue tests, an alternating or pulsating load is applied to a ball screw that is not operating. This is not to be confused with endurance tests, which aim at fatigue also (to confirm dynamic load capacity), but with the pulsating load generated by balls running repetitively over the same spot of the ball track.

ROTATIONAL SPEED

TECHNICAL TIP

The maximum nut speed should not be exceeded under any circumstances. There are, however, special executions available that are suitable for higher speeds, so please inquire. Critical speed may be exceeded in certain cases - contact our application engineers for further advice. And critical speed is not a significant limitation in applications with rotating nuts. When operating at prolonged high speeds, heating of the ball screw may become the limiting factor. Hollow screws for internal cooling are available, but this requires an expensive additional system with its often troublesome rotating unions. Nuts with cooling jacket, which prevents heat migration into the slide, have been tested but are not practical. Another possibile solution to running at high speeds for prolonged time is Steinmeyer's HIGH ETA+ PERFORMANCE technology. This advanced ball screw design produces less than half the heat compared to a regular ball screw of same size. So steady-state temperatures remain much lower. This may eliminate the need to use a forced cooling system for the ball screw shaft. Pre-tensioning amounts to compensate for thermal expansion are lower too, significantly reducing the burden on the support bearings from tensioning forces.

Xi-Plus

Optimized ball screws for best running features.

Microscopic irregularities on the surface of the spindle thread can cause noise and vibration as well as uneven running characteristics.

Through the use of innovative production technologies, the smoothness of the thread surface on the ball nut of precision ball screws can be significantly improved. Xi-Plus ball screws are characterized by markedly improved running characteristics, smooth running and low noise.

Comparative measurements on ball screws show that frequencies which generate audible noise can be completely eliminated. Moreover a reduction up to 40% in the variation of friction torque can be achieved, resulting in a significant improvement in the running smoothness.

More information about
Xi-Plus-Technology on page 89.

CRITICAL SPEED

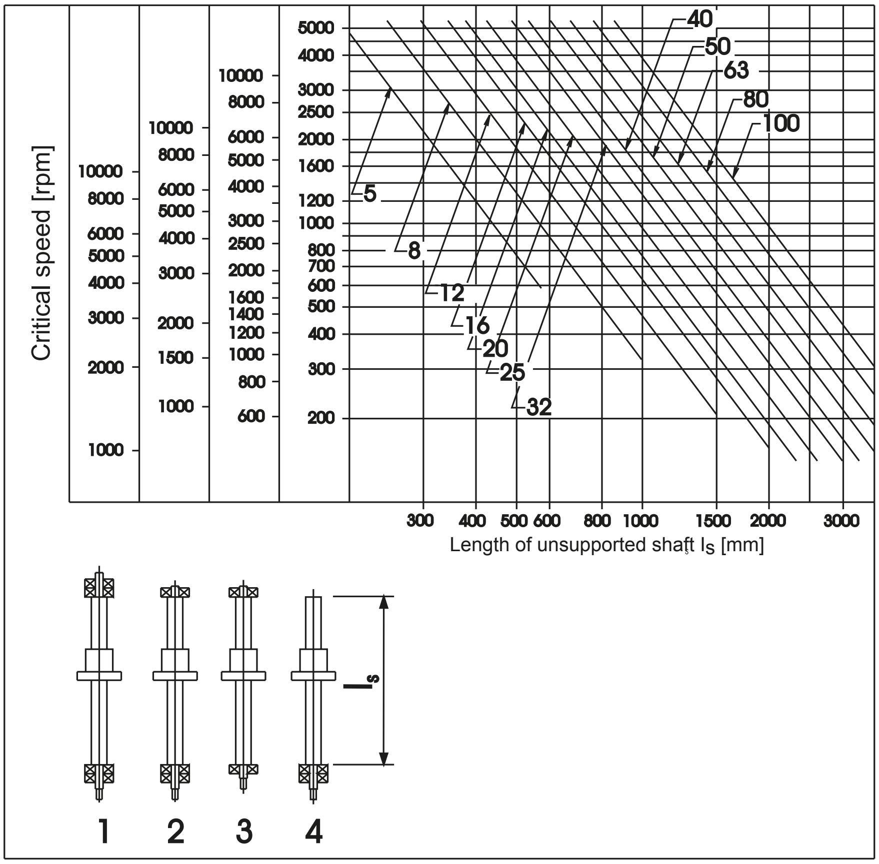
Critical speed is the first (lowest) speed at which the ball screw shaft is in resonance. In applications with rotating shafts it limits the rpm of the screw. Variables that influence it are shaft diameter, unsupported length and support bearing configuration.

Similar to buckling, critical speed depends on the support bearing configuration. Fixed support bearings are assumed to resist angular deflection of the shaft, while simple support bearings do not. A bearing assembly consisting of two simple bearings with a spacer would however qualify as "fixed" bearing for the purpose here.

For long screws, we recommend using the following equation. Make sure to select the proper factor for the bearing configuration used:

$$n_k = k \cdot d_N \cdot \frac{1}{l_s^2} \cdot 10^7$$

n_k:	Critical Speed [rpm]	Fixed - fixed (1):	25.5
d_N:	Nominal diameter [mm]	Fixed - supported (2):	17.7
l_s:	Unsupported length [mm]	Supported - supported (3):	11.5
k:	Support bearing factor	Fixed - free (4):	3.9



TECHNICAL TIP

Critical speed is the first resonant frequency (speed) of the rotating shaft. Resonance in a rotating shaft can be catastrophic and even break the shaft. However, not all ball screws will necessarily exhibit such behavior since the nut represents another support of the shaft, permanently changing the unsupported length of the shaft and its resonant frequency. Nevertheless Steinmeyer recommends operating a ball screw only up to a maximum speed not to exceed approximately 80% of the critical speed, or to discuss the possibility of higher speed with our engineers. For ball screws with rotating nuts, critical speed may be exceeded if the run-out of the rotating nut is kept within tight tolerances. A perfectly concentric nut will not „pump" critical amounts of energy into the shaft even if it is at or near its critical speed.

Pre-tensioning and critical speed: Contrary to general belief, pre-tensioning the ball screw shaft does not change the critical frequency of the shaft. It is the stiffness against bending that resists centrifugal forces in a rotating shaft, not tension.

ROTATIONAL SPEED

MAXIMUM SPEED

A second limitation is imposed by the mass forces upon balls. It depends on internal construction of the ball nut and in particular the ball return, and ball diameter (or mass).

The DN value does not take lead and ball size into consideration. In general, ball screws with very small balls have somewhat lower speed limits than screws with larger balls. We strongly recommend observing the speeds in the two tables below instead of DN values.

Maximum speeds [rpm] nominal diameter 3 - 125 mm (rolled ball screws 30% lower)

Lead P	3	5	6	8	10	12	16	20	25	28	32	36	40	50	60	63	80	100	125
0,5	4500	2900	2900	1800															
1	4500	4500	4500	3000	2000	2000													
1,5		4500	4500	3500	2500	2500													
2		4500	4500	4500	3700	3700	2800	2200	1800										
2,5		4500	4500	4500	4000	4000	3500	4100	2500										
3		4500	4500	4500	4500	4500	4000	4100	3000										
4			4500	4500	4500	4500	4300	4100	3600		2800								
5			4500	4500	4500	4500	4300	4100	3800	3800	3300	3300	2600	2100		1700	1200	1000	800
6			4500	4500	4500	4500	4300	4100	3800	3800	3400	3400	3000	2200		1700	1200	1000	800
8				4500	4500	4500	4300	4100	3800	3800	3400	3400	3000	2400		1800	1200	1000	800
10					4500	4500	4300	4100	3800	3800	3400	3400	3000	2500		2000	1500	1200	1000
12						4500	4300	4100	3800	3800	3500	3500	3000	2500		2000	1500	1200	1000
15						4500	4300	4100	3800	3800	3600	3600	3000	2500		2000	1500	1200	1000
20						4500	4300	4100	4000	4000	4000	4000	3000/ 4000	2500/ 3000		2000/ 2500	1500	1600	1000
25							4300	4100	4000	4000	4000	4000	4000	3000	2500	2000	2000	1600	1200
30							4300	4100	4000	4000	4000	4000	4000	3000	2500	2000	2000	1600	1200
35								4100	4000				4000	3000	2500	2000	2000	1600	1200
40								4100	4000				4000	3000	2500	2000	2000	1600	1200
50									4000					3000	2500		2000	1600	

Maximum speeds [rpm] for UltraThrust ball screws diameter 32 - 125 mm

Lead P	32	36	40	50	63	80	100	125
10	3000	3000	3000	2500				
12	3000		3000	2500				
16				2500	2000			
20					2000	1500	1200	1000
25					2000	1500	1200	1000
30			3500					
40				2800	2200	1800	1400	1100
50						1800	1400	1100

First number: Maximum speed with internal ball return (1XXX)
Second number: Maximum speed with external ball return (3XXX)

TECHNICAL TIP

DN values only yield a preliminary idea of the maximum speed, because there are more factors than just the nominal diameter and the type of ball return that affect speed rating. Specifically the mass of balls plays an important role, which is not reflected in DN ratings. Also, the DN method to determine maximum speeds is not applicable for very small screws.

DN VALUE

TECHNICAL TIP

When using only the DN method to determine maximum rpm for a 3 mm ball screw, the result would be 120.000 / 3 = 40,000 rpm, which is certainly not realistic. The DN method returns unreasonably high numbers when used for very small ball screws! Always check the maximum speed tables.

The concept of DN is a simplified way of determining the maximum rotational speed of a ball screw. DN is simply the multiplication of nominal diameter of the ball screw in mm times the maximum allowable speed in rpm. Keep in mind that for very small and very large screws this will not return valid numbers.

DN values allow easy comparison between different ball screw designs. More sophisticated ball return systems result in higher DN values and, conversely, lower DN values are associated with less sophisticated ball return methods. DN values provide direct correlation to ball velocity.

$$DN = n_{max} \cdot d_N$$

n_{max}: Maximum speed [rpm]
d_N: Nominal diameter [mm]
DN: Driving speed value [mm/min]

Most ball screws available today have maximum DN values between 60,000 and 120,000, and in some cases even higher. However, Steinmeyer recommends observing the maximum speeds published here for each size. Use the following values for orientation purposes and of course for Steinmeyer ball screws only:

- Internal return (Series 1xxx): DN $\leq$ 120,000
- External return (UltraSpeed and end-cap return) (Series 2xxx and 3xxx): DN $\leq$ 160,000
- High-load ball screws (UltraThrust) (Series 9xxx with ball diameter 15 mm and 19 mm): DN $\leq$ 120,000

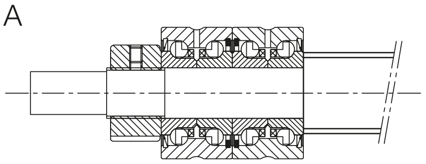
Always check the maximum speed tables on page 36.

BEARING JOURNAL DESIGN AND PRE-TENSIONING BALL SCREWS

Journal design

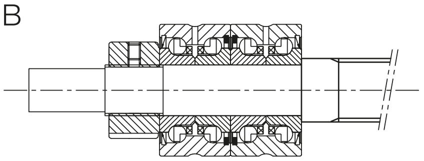
Support bearings facilitate rotation of the shaft while transmitting axial loads from the ball screw into the surrounding structure. They have to do that with minimum friction and the smallest possible deformation under changing loads. Modern ball screws have high load capacity and rigidity, which places high demands on bearings as well. Only high-end bearings which have been designed specifically for use with ball screws can match the load capacity and rigidity numbers. Using inferior bearings yields unsatisfactory results. It is also important that the interface between ball screw and support bearings is designed properly.

A



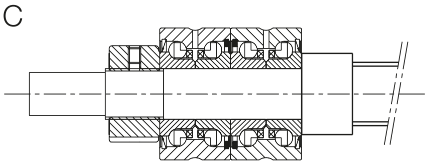
A: The simplest journal design is a bearing seat that is small compared to the root diameter of the ball thread. If the shoulder surface between the bearing journal diameter and the root diameter (= nominal diameter - ball diameter) of the ball thread is sufficient to support the bearing preload plus the maximum thrust with reasonable surface pressure, then this cost-effective solution is recommended.

B



B: Should the shoulder surface be too small, then the shaft could actually bend when the locknut is tightened to preload the bearings. In this case, a full shoulder is needed, that is the ball thread has to be incomplete instead of cutting through the shoulder - sometimes referred to as a „dead start thread". This is possible for ground and whirled screws, but impossible for rolled screws. Make sure that the other end of the ball thread is complete to allow mounting of the nut. Two incomplete ends of the ball thread means costly complications of nut mounting!

C



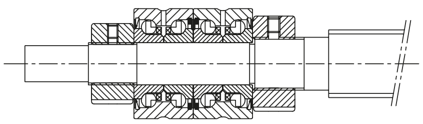
C: If the shoulder surface is still insufficient, then a collar becomes necessary. Collars can be heat shrunk onto the ball screw if there is still some difference between bearing bore diameter and screw O.D. Shrunk collars are also possible for rolled screws. If there is not enough shoulder for the shrunk collar to prevent axial slippage, then the collar has to be solid, which means considerable machining time. This is the most expensive solution.

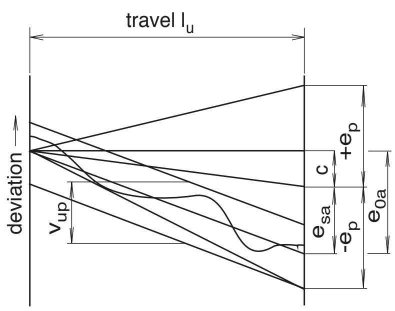
TECHNICAL TIP

If a long screw is to be pre-tensioned to compensate for thermal expansion, thrust bearings are required at both ends. One of them needs to be adjustable to set exact tension of the shaft. A way to do that is by using a second locknut thread. Other possibilities include shims to set pre-tension. Always make sure to check elongation of the screw with a dial gage while tensioning it.

Sufficient shoulder area is important, since creeping of the metal and loss of bearing preload results from excessive surface loading. If the shoulder surface is not symmetric (for example when the thread cuts through it and the shoulder surface below the root is insufficient to support the bearing), then cocking of the bearing will cause bending of the shaft with unsatisfactory runout of the ball screw.

PRE-TENSIONING A BALL SCREW





The purpose to pre-tension a ball screw is to compensate for thermal expansion during operation and thus avoid compressive forces as the shaft expands between the two fixed bearings. The goal is to stretch the screw sufficiently to maintain at least some tension at the highest anticipated temperature.

TECHNICAL TIP

There is only a minimal change in the resonant frequency (critical speed) of the shaft when tensioned. But tensioning is necessary when the screw is supported by fixed bearings at both ends. The fixed bearings do of course result in raising critical speed.

Ball screws may be ordered with a negative lead compensation c, so that after tensioning the lead deviation is near zero.

Anticipating the temperature

$$\Delta l_T = l_s \cdot \Delta t \cdot \alpha$$

Δt: Temperature rise [°C]

α: Coeff. of thermal expansion [1/°C]
(for DIN 1.1213 = 11.5 x 10^{-6}/°C)

l_s: Shaft length [mm]

Steinmeyer is capable of determining the steady-state temperature for your duty cycle. Please contact us for details.

Calculating the elongation

The elongation of the shaft should at least be equal to the thermal expansion for the anticipated steady-state temperature, or slightly larger.

$$\Delta l_p \geq \Delta l_T$$

Δl_p: Amount of elongation [mm]

Δl_T: Thermal expansion [mm]

Calculating the tensioning force

$$F_T = E \cdot A \cdot \varepsilon$$

$$\varepsilon = \frac{\Delta l_p}{l_s}$$

F_T: Tensioning Force [N]

E: Elasticity modulus [N/mm²]

A: Shaft cross section [mm²]

ε: Elongation relative to length

Pre-tensioning requires considerable forces. In applications with rotating shaft, the support bearings have to withstand these forces, which becomes a main consideration for the selection of the bearings.

To calculate the tensioning force, assume that the root diameter of the screw is approximately the nominal diameter minus the ball diameter.

BEARING JOURNAL DESIGN

SELECTING SUPPORT BEARINGS

TECHNICAL TIP

Miniature ball screws are normally supported using either a pair of angular contact bearings on one end, or by a single-row bearing at both ends of the screw, preloaded against one another. It is better to use bearings with increased play here, since this will establish a more desirable contact angle when the bearings are preloaded. Bearings with minimal or no play may generate excessive forces on their balls when preloaded this way, causing potential premature bearing failure and rough motion.

Support bearings must be able to carry not only the thrust produced by the ball screw, but also any additional forces from pre-tensioning the ball screw shaft, plus any side forces generated by belt drives. Ball nuts with many ball circles and/or large balls and shafts with considerable pre-tension may make it difficult to find bearings with sufficient load capacity, especially when the bore (ID) of the bearing has to be no larger than the shaft's root diameter, and the journal shoulder diameter no larger than the ball screw's nominal diameter.

This discussion is only meant to highlight areas of concern. We cannot give detailed recommendations on which bearing to select. Criteria for the selection include:

- Axial dynamic load capacity of the bearing should be approximately equal to the dynamic load capacity of the ball screw, or higher if the screw is pre-tensioned.
- Minimum shoulder diameter for the bearing's inner ring should be no greater than the root diameter of the ball screw (journal shape A), or no greater than the screw's nominal diameter (journal shape B).
- The bearing should be suitable for the same lubrication method (grease/oil) and equipped with the proper seals for that lubricant. Speed ratings must be sufficient with the lubricant selected.

Steinmeyer recommends using INA support bearings. The following table gives examples of typical ballscrew / bearing assemblies. However, it is not possible to cover all combinations in this catalog. Please refer to our engineering service for further information.

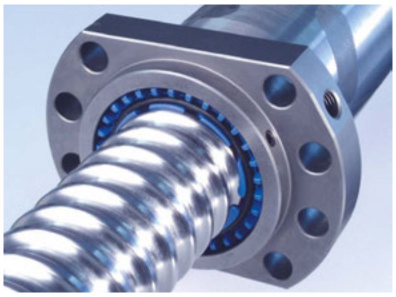
Ballscrew nominal diameter d_N [mm]	INA-bearing with journal configuration acc. to fig.		
	A	B	C
16	ZKLN1034	-	ZKLN1242
20	ZKLN1242	-	ZKLN1545
25	ZKLN1747	-	ZKLN2052
32 (P≤5)	ZKLN2557	-	-
32 (P>5)	ZKLN2052	ZKLN2557	-
40 (P≤5)	ZKLN3062	-	-
40 (P>5)	-	ZARN3062LTN	-
50 (P≤5)	ZKLN4075	-	-
50 (P>5)	-	ZARN4075LTN	-
63 (P≤5)	ZKLN5090	-	-
63 (P>5)	ZARN4090LTN	ZARN45105LTN	-
80	-	ZARN50110LTN	-
100	-	ZARN60120LTN	-

This brief overview cannot give a final selection aid to determinate an optimum bearing solution. Radial loads due to drive belt tension or increased axial loads due to pre-tensioning a ballscrew need to be considered, too.

LUBRICATION AND WIPERS / SEALS

Supply of fresh lubricant and wipers must be considered together. The selection not only depends on environmental conditions, but also on loads and speeds. In this section we discuss whether grease or oil is the right lubricant, and which grease or oil should be used.

Plastic Wiper

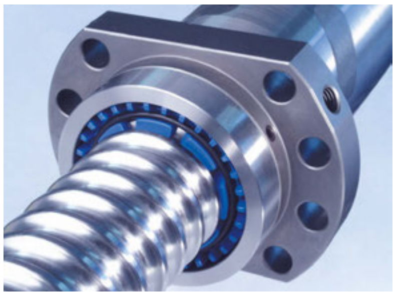


Plastic wipers, sometimes called „labyrinth seals", are widely used in machine applications. They prevent contamination of the nut from chips and other larger particles, while at the same time letting some oil exit the nut. Combined with automatic grease or oil lubrication, they help to flush the nut. The result is higher reliability of the ball screw.

Felt Wiper

Felt wipers are an excellent solution for ball screws operated in an environment with abrasive or otherwise troublesome dirt as in grinding or woodworking machines (for example oil absorbing particles like wood chips). The felt wiper seals the nut effectively, even from fine particles, and also stores some lubricant. Felt wipers are contact wipers and add some friction.

Combination Wiper



Combination wipers are plastic wipers with an additional felt wiper (inside the ball nut). They are used where felt wipers are desirable, but need to be protected from water or water based fluids. Felt tends to absorb water, so it should not be used without the additional plastic wiper in such cases.

Combination wipers add to the nut length - please consult Steinmeyer for details.

TECHNICAL TIP

If the duty cycle is such that an EHD lubrication film can build over a significant part of the motion, then oil lubrication with a properly selected oil grade and viscosity will always outperform grease in terms of wear. On the other hand, grease has an edge at slow speeds because it offers better wear protection under mixed friction or boundary friction conditions. Grease can also be used for long-term or for-life lubrication. Lubricant loss with grease is lower than with oil.

A general discussion for oil vs. grease lubrication and the results of a scientific test of several commonly used lubricants by the tribology lab of CSEM at Neuchâtel/Switzerland (www.csem.ch) is shown on the next pages.

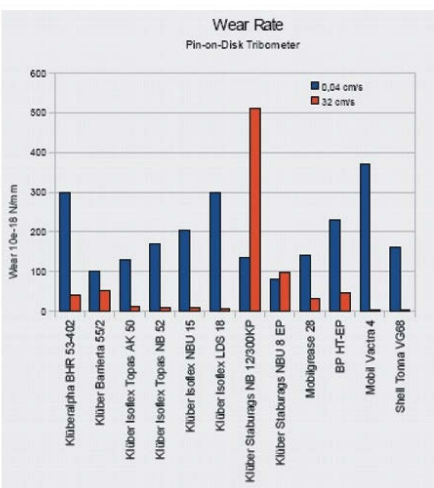
LUBRICATION AND WIPERS / SEALS

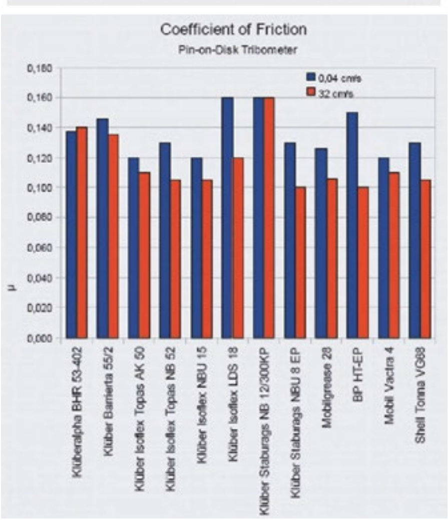
TRIBILOGY OIL / GREASE

On behalf of Steinmeyer and under the scope of a EUREKA project sponsored by the KTI (Commission for Technological Interchange) in Switzerland, the tribology lab of CSEM (Centre Suisse d'Electronique et Microtechnique) designed and conducted a test for commonly used lubricants under the specific tribological conditions found in ball screws. A modified pin-on-disc tribometer was used, with the pin replaced by a regular bearing ball made from 52100 steel, which rode on a rotating disc made from the same material, heat treated and ground similar to the raceway of a ball screw.

The disc and the ball closely resembled the friction partners in a ball nut in terms of material and surface finish.

Surface loads and relative speeds in two test series were selected to simulate a slow moving ball screw and a fast moving one. Since the pin-on-disc tribometer causes pure sliding instead of the sliding/rolling typical for a ball screw, the speeds were adjusted to the sliding portion of typical mid-size ball screw kinematics, running at about 50 rpm (0,04 cm/s in the tribometer) and at approximately 1000 rpm (32 cm/s in the tribometer). This simulated as closely as possible the real tribological conditions in a ball screw, both in terms of lubrication film build-up, and wear.





Results:

First of all, the coefficient of friction does not seem to be correlated to the wear rate. Some lubricants yielded low friction, but higher wear at the same time!

- The wear rates of some greases show that these lubricants can really be seen as „universal", because they offer moderate wear protection at both slow and fast speeds.
- At the higher speed, both oils built a hydrodynamic lube film, which can be concluded from the extremely low wear rates.
- A plain mineral oil without wear-inhibiting additive (Vactra4) yielded excessive wear rate at slow speed, despite its high viscosity. It was obviously unable to protect the friction partners once speed fell below the threshold to build a hydrodynamic film. It was squeezed out of the contact patch and did not sufficiently reduce wear in boundary conditions.
- Oil with high-pressure, wear inhibiting additives (EP oil), which is able to bond its hydrocarbon molecules to metal surfaces through chemical or physical reaction, performs much better during conditions of boundary friction.

With the exception of a high-pressure grease, which caused the lube film to collapse at high speeds, all greases performed well and yielded acceptable wear rates throughout the test scenario. At the same time it was proven that greases are unable to build a perfect fluid film like oils of proper viscosity, so the wear rate with grease lubrication is higher than the wear rate of oil when an EHD film is present.

Theory of elasto-hydrodynamic (EHD) lubrication

Life calculations for ball screws assume sufficient lubrication, which means that there is an adequate lubrication film. In case of oil lubrication this means that whenever the speed is sufficient, a fluid film develops which separates the contact partners as much as possible. This requires that

- sufficient oil is available at all times
- contamination is minimal
- lubricant is in adequate condition
- viscosity is selected so that over most of the duty cycle a hydrodynamic lube film is maintained

The necessary condition to form such a fluid film is described by the theory of elasto-hydrodynamic lubrication. Whether a fluid film, able to withstand the pressure in the friction contact zone, will build depends on the actual viscosity of the lubricant and the speed and, to a lesser extent, the pressure. However, a certain minimum load is required (for example by preloading the ball screw) to cause a consistent rolling motion of the balls.

Whether an EHD film will build can be determined from the viscosity ratio $k = u / u_1$. The operational viscosity u is the viscosity the lubricant exhibits under the conditions in the contact patch in terms of speed, temperature and pressure. The viscosity needed to build a sufficient EHD film is u_1.

The viscosity ratio can be classified in 3 parts.

- $k \geq 4$ full EHD lubrication - contact partners are mostly separated
- $0.4 \leq k < 4$ mixed friction - lubricant with wear inhibiting additives is necessary, since EHD film is only partially able to separate contact partners (EP - grease, CLP - oil)
- $k < 0.4$ no separation - accelerated wear through micro welding will occur

Since a high viscosity ratio is desirable, oil with high viscosity seems to be the solution. But high oil viscosity also causes high temperatures, which in turn could lower the actual viscosity again. High viscosity oil is also difficult to deliver to all lubrication points and will not aid in cooling the ball nut.

We recommend using oil viscosities as close as possible to the ones per the following table.

LUBRICATION AND WIPERS / SEALS

OIL LUBRICATION

Oil lubrication requires an oil port in the nut and wipers. Steinmeyer ships all ball screws lubricated ready for use. If the screw is to be used with oil lubrication, please notify us when ordering - we will then pack the nut with a special grease, which requires no cleaning prior to use. The grease will be washed away with time and be dissolved in the oil. The grease fill protects the screw during shipping and storage, and keeps it lubricated until the oil supply is operational.

Oil should be injected approximately four times per hour. Recirculating oil systems should include a filter with a 10 micron mesh and a cooler to keep the oil temperature below 50° C as it enters the ball nut. Oil mist is only recommended when there are no wipers and contamination is very low. Oil bath lubrication can only be used for very low speeds. Oil drip and loss lubrication is possible, but oil quantities must be observed.

Recommended oil quantities

Nominal diameter [mm]	Oil quantity in cm^3/hr for oil circulation		Oil quantity in mm^3/min for minimum quantity lubrication (MQL) or oil mist lubrication
	without extra cooling	with extra cooling	
16	0.12	0.3	1
20	0.15	0.4	1.5
25	0.2	0.5	2
32	0.25	0.7	2.5
40	0.3	0.9	3
50	0.4	1.5	4
63	0.5	2	5.0
80	0.6	3.0	6.0
100	0.8	4.0	8.0
125	1.0	5.0	10.0

Suitable oil grades

Only oils with wear inhibiting additives should be used. These have the ability to lubricate in conditions of boundary friction, when speeds for EHD-lubrication are insufficient. We strongly recommend CLP grade gear oil per DIN 51517-3 or equivalent. Do not use way oils or hydraulic oils, even if they are labeled „high pressure"!

Recommended oil viscosity for EHD lubrication

Nominal diameter [mm]	Mean speed [rpm]	Viscosity u1 [mm²/s]	Lubricant temperature [°C]	Viscosity grade ISO VG	Actual viscosity u [mm²/s]
up to 16	10	ca. 3000	30	680	ca. 3000
	50	500	35	320	500
	200	180	40	220	220
	500	70	45	100	80
	1000	40	50	68	42
	2000	28	55	68	35
20 to 32	10	ca. 1200	30	460	ca. 1200
	50	350	35	320	500
	200	120	40	150	150
	500	50	45	68	50
	1000	30	50	46	30
over 40	10	ca. 900	30	320	ca. 900
	50	250	35	220	300
	200	80	40	100	100
	500	35	45	46	35

TECHNICAL TIP

Felt or combination wipers seal the nut. It is therefore important, when using such wipers, that the recommended oil flow rates are not exceeded. The nut may become overfilled causing overheating at higher speeds.

Service life of oil lubricated ball screws exceeds the life that can be expected with grease lubrication in many cases, but only if EHD lubrication can be reached over most of the duty cycle.

LUBRICATION AND WIPERS / SEALS

GREASE LUBRICATION

Re-lubrication of ball screws should be done with grease having the same, or compatible, thickener as the grease used for factory lubrication. The base oil viscosity should also be approximately the same. If not specified otherwise, Steinmeyer uses Kluber Staburags NBU 8 EP for factory grease fill of the nut. Please note that the factory grease fill is only enough for initial operation, until the first required re-lubrication.

Manual re-lubrication

Ball screws with regular plastic wipers should be re-greased every 500 hours of continuous operation, or four times a year. Ball screws with felt wipers have a recommended re-greasing interval of 1000 hours of operation, or twice a year.

If the ball screw is fitted with felt or combination wipers and is adequately protected from dirt and liquids (e.g. water, coolant), then the re-lubrication interval may be extended to 2000 hours or once a year. In such cases, Steinmeyer will pack the ball nut with long-term grease Kluber Isoflex NBU 15, and impregnate the felt wiper with an oil compatible with this grease. Please consult with our application engineers if you want your ball screw prepared for long-term lubrication.

Automatic grease lubrication

There are two possible ways to automatically re-lubricate ball screws. Either the nut is connected to an automatic lubrication pump with multiple ports, or a lubrication cartridge is used. Both require a grease which is not too viscous and can be reliably pumped through piping to its destination. Grease cartridges powered by chemical gas generators keep the grease pressurized all the time, which can cause the base oil to separate from the thickener resulting in clogged lines and lubrication breakdown. We recommend electromechanical cartridges, which use a gear motor and a lead screw to pump the lubricant. These do not keep the grease under continuous pressure, thus preventing such problems.

Normally, liquid greases NLGI class 0 or 00 are used for automatic grease lubrication, because they can be pumped through long lines more easily. With large enough pipe diameters, grease class 1 or even 2 can be used as well - but should first be tested.

The intervals and quantities are shown on page 47.

TECHNICAL TIP

Mineral oil based lubricants degrade with time in two ways:

- They are attacked by aggressive chemicals (for example oxygen) from their environment, which causes a polymerization of the hydrocarbon or oxidation. The lubricant turns into a solid with time, but the process is slow in a normal environment, so re-lubrication in normal intervals replaces the lubricant before it becomes too degraded. However, this chemical aging limits the storage period of pre-greased components to 2 - 5 years. Specifics can be obtained from grease manufacturers.
- During use, the hydrocarbon molecules are subjected to mechanical stresses when they are passed through the contact patches. This mechanical stress causes the molecule chains to break up over time, reducing the viscosity. The maximum usage time of the lubricant can be calculated - please contact us.

Manual re-lubrication

Nomina diameter [mm]	Grease quantity on the shaft [g]	Grease quantity of the nut [g]	
		Single nut	Double nut
3	0.15	-	-
5	0.3	-	-
8	0.7	-	-
12	1	-	-
16	4	0.2	0.3
20	7	0.5	0.8
25	10	0.7	1
32	17	1.1	1.7
40	25	1.8	2.7
50	35	2.4	3.6
63	50	3	4.5
80	70	4	6

Automatic re-lubrication

Nominal diameter [mm]	Grease quantity per shot [cm³]		Lube interval [h] *
	Single nut	Double nut	interval
16	0.01	0.03	4 - 8
20	0.03	0.06	4 - 8
25	0.03	0.06	4 - 8
32	0.06	0.1	4 - 8
40	0.06	0.1	4 - 8
50	0.1	0.16	4 - 8
63	0.16	0.2	4 - 8
80	0.2	0.3	4 - 8

* The automatic re-lubrication should use grease class NLGI 00 or NLGI 000 and grease quantity as specified per shot. A re-lubrication interval should be between 4 and 8 operating hours or after a distance of 500 m. In addition, every time the machine axis is initially turned on a lubrication impulse should also occur.

TECHNICAL TIP

For non-aerospace applications, when re-lubricating manually using a grease gun, make sure not to pump more grease than indicated in the table. Exceeding the recommended quantities will not only overfill the nut, but can actually pressurize it and push the wipers out of their locations!

Ball screws for aerospace use are normally greased to a point where fresh grease appears either at both ends of the nut underneath the wipers or ice scrapers, or at a relief valve. Filling the nut completely is desirable in aerospace applications because the grease keeps water from entering the nut. Due to the low average speeds of most aerospace actuator screws there is no risk of overheating from excessive amounts of grease inside the nut, as would be the case in machine tools with their rapid motion.

LUBRICATION AND WIPERS / SEALS

Recommended Grease

Application	Kluber	Low temp. limit [C°]	Upper temp. limit [C°]	Base oil viscosity [mm²/s] at 40°C	Lubcon	Low temp. limit [°C]	Upper temp. limit [°C]	Base oil viscosity [mm²/s] at 40°C
General purpose, long-term grease for high surface pressure	Staburags NBU 8 EP	-20	140	100	Turmogrease PHS 1002	-40	160	105
Long-term grease with felt wipers	Isoflex NBU 15	-30	130	23	Turmogrease Highspeed L252	-50	120	25
Long-term grease without felt wipers, low speeds only	Stabu-rags NBU 12/300KP	-20	140	220	Turmogrease CAK 4002	-20	150	400
Low friction grease	Isoflex LDS 18 Spezial A	-50	120	15	Thermoplex 2 TML Spezial	-70	130	20
High temperature grease	Klüberalpha BHR 53-402	-40	260	400	Turmotemp Super 2 EP	-30	280	500
Low temperature grease	Isoflex PDL 300 A	-70	110	9	Thermoplex TTF 122	-70	150	12
Vacuum grease, clean room	Barrierta L55/2	-40	260	400	Turmotemp II/400 KL	-30	260	400
					Turmotemp II/400 RS2	-30	260	550
Food grade grease	Klübersynth UH1 14-151	-45	120	150	Turmosynth-grease ALN 2501	-40	160	250

Hybrid ball screws

Hybrid design of ball screw means screw and nut are made from steel and balls are made from ceramic.
Most suitable ceramic is Silicon Nitride in HIP-quality (not isostatic pressed).

Hybrid ball screws are designed to operate with very minimal lubrication. When adequate lubrication is available there is no difference compared to conventional ball screws equipped with steel balls.

The high hardness and Young´s modulus of Silicon Nitride (Si_3N_4) results in a higher applied load to the steel-ball race. Hybrid ball screws however must be assembled using smaller steel spacer balls which means that only 50% of the balls are carrying load. Therefore, the application of hybrid ball screws must be carefully reviewed.

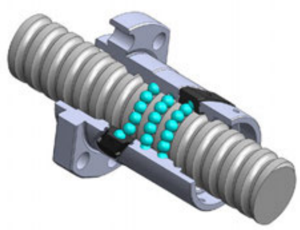
BALL RETURN

TECHNICAL TIP

Ball nuts require a means to recirculate balls. Without it, the ball path would not be closed and balls would fall out at the rear end of the nut. Design of the ball return is the determining factor for the maximum speed at which the ball nut can safely operate. This is normally expressed by the D_N-value. The better the ball return system deals with mass forces of the balls, the higher the D_N value. Manufacturers typically quote D_N values from 60.000 for basic tube returns to 160.000 and higher, e.g. the UltraSpeed return from Steinmeyer.

Steinmeyer uses all commonly known designs for ball returns. However, the multi-liner and the tube return are only used for aerospace „build-to-print" applications, since staying with a previously qualified design simplifies the qualification procedure.

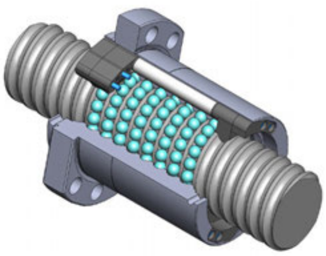
Track-to-track (internal return)



The track-to-track return uses ball deflectors to lift the balls across the O.D. of the shaft and guide them directly into the next (or previous) track. Internal ball return is very compact and yields the smallest nut diameters among all ball return systems. It is also the ball return of choice for very small ball sizes and small leads.

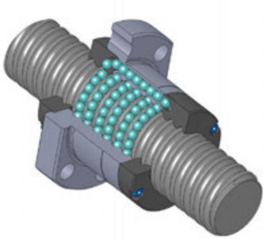
Each deflector serves one turn, which is one circuit (or ball circle).

Through-the-nut return (external return)



Steinmeyer's „UltraSpeed" return is normally used for lead/diameter ratios greater than 0.5. It is normally used with dual start threads. Balls are lifted off the shaft using a deflector at one end of the nut and then guided through a bore (internal to the nut body) to the other end of the nut, where a similar piece guides the balls back onto the thread. One pair of deflectors serves one circuit (i.e. one of the threads) which includes several turns.

End cap return



End cap return works very much like the previously described through-the-nut return, with the exception that the ball deflector function is executed using a (plastic) cap at both ends of the ball nut. Each cap serves as the ball return and also includes wipers. End cap return is normallly used for very large lead/diameter ratios. This style ball nut is only available as a flange nut.

MATERIALS AND PROCESSES

Materials					
Application	Material No.	AMS designation	DIN / ISO designation	ANSI designa-tion	Aerospace grade
Shaft	1.1213		Cf53	1050	
	1.3505		100Cr6	52100	
	1.4021	AMS 5351	X20Cr13		
	1.4112	AMS 7445D	X90CrMoV18	440 B	
	1.4108	AMS 5898	X30CrMoN15-1		•
	1.4123	AMS 5925	XD15W		•
	1.4125	AMS 5618	X105CrMo17	440 C	•
		AMS 5844		MP35N	•
General purpose	1.4545	AMS 5659		15-5 PH	•
	1.4548	AMS 5643	X5CrNiCuNb17-4-4	17-4 PH	•
Nuts	1.3505		100Cr6		
	1.4108	AMS 5898	X30CrMoN15-1		•
Balls	1.3505		100Cr6		
	1.4108	AMS 5898	X30CrMoN15-1		•
	1.4125	AMS 5618	X105CrMo17	440 C	•

Processes				
SAE designation		AS 7003 / NADCAP cer-tified	Certified through 3rd party/customer audit	External process
AS 7102	Heat treatment	•	•	•
AS 7114	Induction hardening	•		•
AS 7108	Chemical processes	•		•
AS 1701	NDT	•		•
AS 7117	Surface treatment	•		•
AS 7115	Elastomer Seals/ Gaskets	•	•	•
AS 7200	Sealants	•	•	•
AS 7101	Non-standard treatment	•	•	•

DEFINITIONS

A: Shaft cross section [mm^2]
α: Coefficient of thermal expansion [1/°C]
c: Lead compensation
C_{0a}: Static axial load capacity [N]
C_a: Dynamic axial load capacity [N]
$C_{aerf.}$: Required dynamic axial load capacity [N]
$\Delta l_{b/t}$: Axial elastic deformation due to external load F_i [µm]
$\Delta l_{b/t,pr}$: Axial elastic deformation due to preload F_{pr} [µm]
Δl_p: Elongation of the shaft [mm]
Δl_T: Thermal expansion [mm]
d_N: Nominal diameter [mm]
DN: Driving speed value [mm/min]
Δt: Temperature increase [°C]
„E": Tolerance of the cumulative lead error (JIS)
E: Elasticity modulus [N/mm2]
ε: Elongation relative to length
e_{0a}: Mean lead deviation over entire travel l_u
$e_{2\pi}$: Lead error in one revolution (JIS)
e_{300}: Lead error in 300 mm (JIS)
e_p: Tolerance for the average lead deviation over entire travel l_u
e_{sa}: Actual lead deviation over entire travel l_u
F_i: External load [N]
F_i^*: Modified external load [N]
F_m: Dynamic equivalent load [N]
F_{m*}: Modified dynamic equivalent load [N]
F_{pr}: Preload [N]

F_T: Tensioning force [N]
i: Number of ball circles
JIS: Japanese Industrial Standard
k: Support coefficient (critical speed)
L_{10}: Nominal life 90% reliability [revolutions]
l_s: Unsupported shaft length [mm]
l_u: Travel [mm]
m: Support coefficient (buckling)
n_i: Speed [rpm]
n_k: Critical speed [rpm]
n_m: Equivalent Speed [rpm]
n_{max}: Maximum Speed [rpm]
P: Lead [mm]
P_B: Buckling load [N]
P_i: Load capacity per ANSI [lbf]
q_i: Time percentage [%]
R_b: Axial bearing rigidity [N/µm]
$R_{nu,ar}$: Actual nut rigidity [N/µm]
R_s: Shaft rigidity [N/µm]
R_t: Rigidity of ball screw [N/µm]
T: Travel compensation (JIS)
T_{pr}: Preload torque
$v_{2\pi a}$: Lead fluctuation in one revolution
v_{300a}: Lead fluctuation in 300 mm
v_{300p}: Tolerance of the lead fluctuation in 300 mm
v_{ua}: Lead deviation over entire travel lu
v_{up}: Tolerance of the lead deviation over entire travel l_u

NUMBERING SYSTEM LARGE BALL SCREWS

BALL SCREWS	1 4 1 6 / 5 . 40 . 800 . 900 L P 3 P
1 nut with internal return 2 nut with end cap return 3 nut with external return 9 nut with z-deflectors (Heavy Duty)	1
1 single nut with connection thread 2 cylindrical single nut 3 cylindrical double nut 4 flange single nut 5 flange double nut	4
1 single start thread, ground 2 dual start thread, ground 3 single start thread, rolled 4 dual start thread, rolled 5 single start thread, whirled	1
0 spring preloaded double nut 2 standard nut (Ø 3-16 mm) 4 customized nut 6 nut according to DIN 69051	6
lead (mm)	5
nominal diameter (mm)	40
thread length (mm)	800
overall length (mm)	900
E ETA+ (High Performance) K mounted with ceramic balls* L lefthand thread* R stainless steel* S pitch-offset-nut* X Xi-Plus *	L
P positioning ball screw T transport ball screw	P
tolerance class	3
P preloaded nut	P

Note:

This item number describes a ball screw with flange single nut acc. to DIN with internal return. Single start thread, ground. Lead 5 mm, nominal diameter 40 mm, thread length = 800 mm, overall length = 900 mm, left hand thread, tolerance class P3, preloaded nut.

You can get further information about our Xi-Plus -ball screws on page 34 and 89.

Please follow the advices on page 12 to 20.

*upon request

NUMBERING SYSTEM MINIATURE BALL SCREWS

BALL SCREWS	xxxx / 2.16.189.271 L P 3 P
1412 | 1432	Flange nut
2412 | 2422 | 2432 | 2442	Flange nut with end cap return
1214 | 1234	Cylindrical nut
1112 | 1132	Nut with connecting thread
1510 | 1530	Nut with connecting thread (spring preloaded)

- lead (mm)
- nominal diameter (mm)
- thread length (mm)
- overall length (mm)
- K mounted with ceramic balls*
L lefthand thread*
O optiSLITE*
R stainless steel*
- P positioning ball screw
T transport ball screw
- accuracy class
- P preloaded nut

1. Selection of Screw
Choose screw type A, B or customized

2. Selection of Nut
According to the table above

Ordering Example ball screw:
1412/2.16.189.271 P3P
Ball screw with flange single nut, series 1412, screw type B, backlash-free, stroke 100 mm, accuracy class 3

Ordering Example ball nut:
1412/2.16.1,5.3
Flange single nut, series 1412, mounted on customized screw shaft, ball-Ø 1,5 mm, 3 circuits. Please specify the accuracy class.

You can get further information about our optiSLITE-ball screws on page 19 and 55.

Please follow the advices on pages 12 to 20.

*upon request