5 Ball screw drives

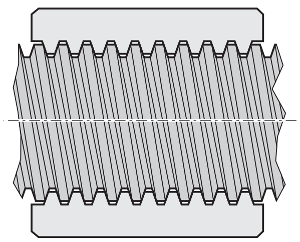
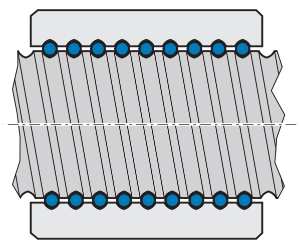
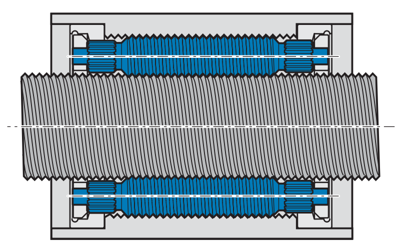
5.1 Principles

5.1.1 System technology

Screw drive overview

In linear motion technology, the generation of "push-pull" or drive motion is just as important as precise guidance of the machine parts. Alongside rack and pinion drives and linear motors, screw drives (screw-and-nut systems) play an important role as feed mechanisms.

These units convert rotary motion into linear motion. The most important representatives in this group of systems are acme screw drives, ball screw drives and planetary roller screw drives.

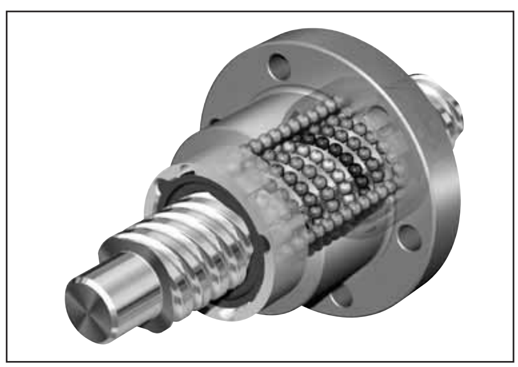
Screw drive type	Description
Acme screw drive 	▮ Screw drive with sliding contact between the screw and the nut
Ball screw drive 	▮ Screw drive with rolling contact between the screw, rolling elements and nut ▮ Rolling elements: balls
Planetary roller screw drive 	▮ Screw drive with integral planetary gear ▮ Screw drive with rolling contact between the screw and the rolling elements and between the rolling elements and the nut ▮ Rolling elements: planetary rollers

In linear motion technology, ball screw drives are the most commonly used option. In the following sections, balls screw drives are dealt with in more detail.

DIN standard

The most important technical specifications and definitions for ball screw drives are stipulated in DIN 69051, Parts 1 to 6. This family of standards covers ball screw drives for use in machine tools, but also applies to other industry sectors.

DIN 69051 Part 1 defines ball screw drives as follows:
An assembly comprising a ball screw shaft and a ball nut and which is capable of converting rotary motion into linear motion and vice versa. The rolling elements of the assembly are balls.



Screw and nut of a ball screw assembly

5.1 Principles

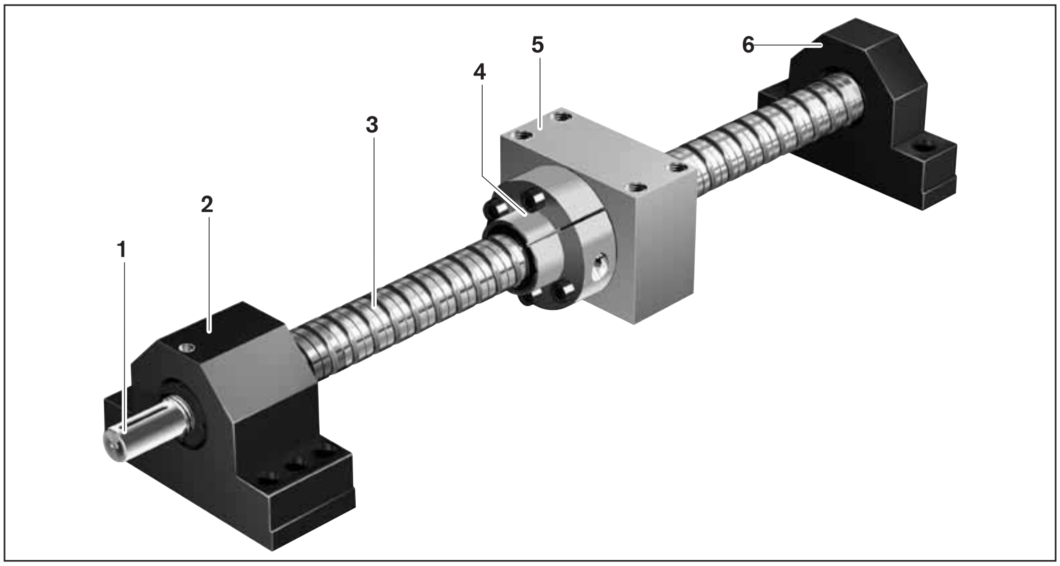
5.1.1 System technology

5.1.1.1 Structural design of a ball screw assembly

Ball screw assemblies generally consist of the following components:

- Ball nut with continuously recirculating rolling elements
- Nut housing (optional)
- Ball screw
- End bearings

The ball nut is installed in the component to be moved (table/carriage) either directly or using a nut housing.



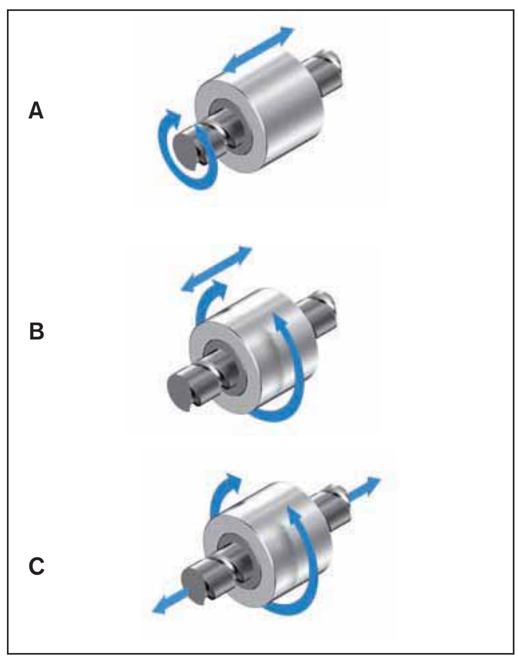
Structural design of a ball screw assembly

1 Screw journal
2 End bearing (here: fixed bearing)
3 Ball screw
4 Ball nut
5 Nut housing or customer's carriage element
6 End bearing (here: floating bearing)

Operating principle

Most ball screw assemblies are driven by a motor attached to the screw journal (1). The nut, or nut and carriage assembly, is positioned by means of the screw's rotation (A). There are also ball screw drives which operate according to a different principle, that of the driven nut (see section 5.3.2). In this case the nut is driven directly and the screw does not turn. Depending on the application, either the nut (B) or the screw (C) will be fixed in position.

The individual elements of ball screw drives are covered in detail in the following sections.



Operating principles of ball screw assemblies

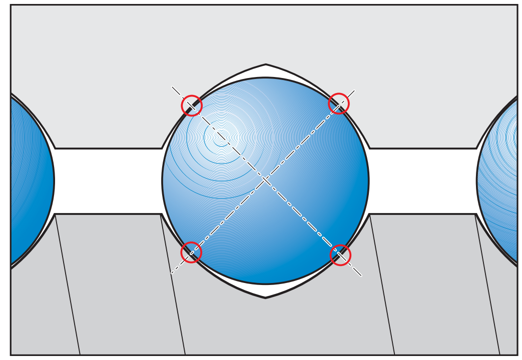
5 Ball screw drives

5.1 Principles

5.1.1 System technology

Screw

The balls run along a helical ball track (the thread, generally with a gothic profile) formed in a shaft. The ball nuts, too, are threaded, and it is the interaction of the ball movement along the screw ball track and along the ball nut raceways which converts rotary motion into linear motion.

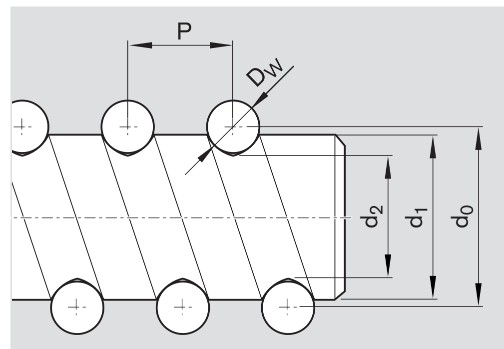


Gothic profile of the ball tracks in the screw and nut and contact points on the rolling elements

Screw dimensions

Screws are specified by means of defined geometric parameters. These parameters are also generally used to specify the complete ball screw assembly.

P	=	lead (linear travel/revolution)	(mm)
d_0	=	nominal screw diameter (ball center-to-center diameter)	(mm)
d_1	=	screw outside diameter	(mm)
d_2	=	screw core diameter	(mm)
D_W	=	ball diameter	(mm)



Screw geometry

Screw sizes

Screw sizes are specified according to the nominal screw diameter d_0, the lead P and the ball diameter D_W: d_0 x P x D_W
The specification for the lead P also includes the direction of rotation of the screw thread (R for right-hand or L for left-hand).

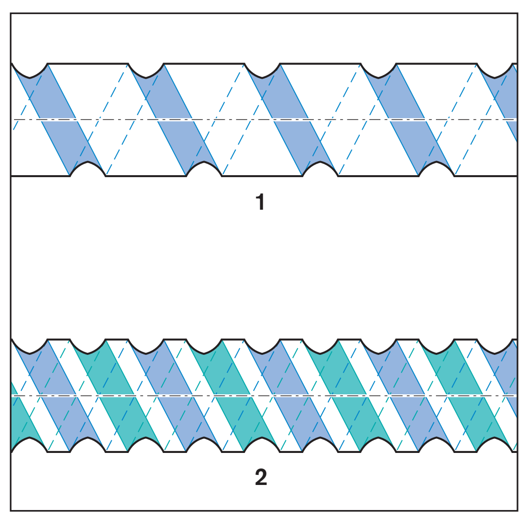
Example: 32 x 5R x 3.5 for a screw with a nominal diameter of 32 mm, lead of 5 mm, right-hand thread, and a ball diameter of 3.5 mm.

Multi-start screws

Depending on the screw diameter, lead and ball diameter, screws can also be produced with more than one ball track. These screws are commonly called multi-start screws.

Screws with up to four starts are technically feasible today and have also been produced where appropriate. When used in combination with multi-start nuts, the resulting assemblies can achieve higher load ratings and therefore also have a longer life expectancy.

In general, ball screws are produced with a right-hand thread. For special applications (e.g. closing or clamping movements) screws with a left-hand thread or with right and left-hand thread can be used.



Single-start (1) and two-start (2) screw

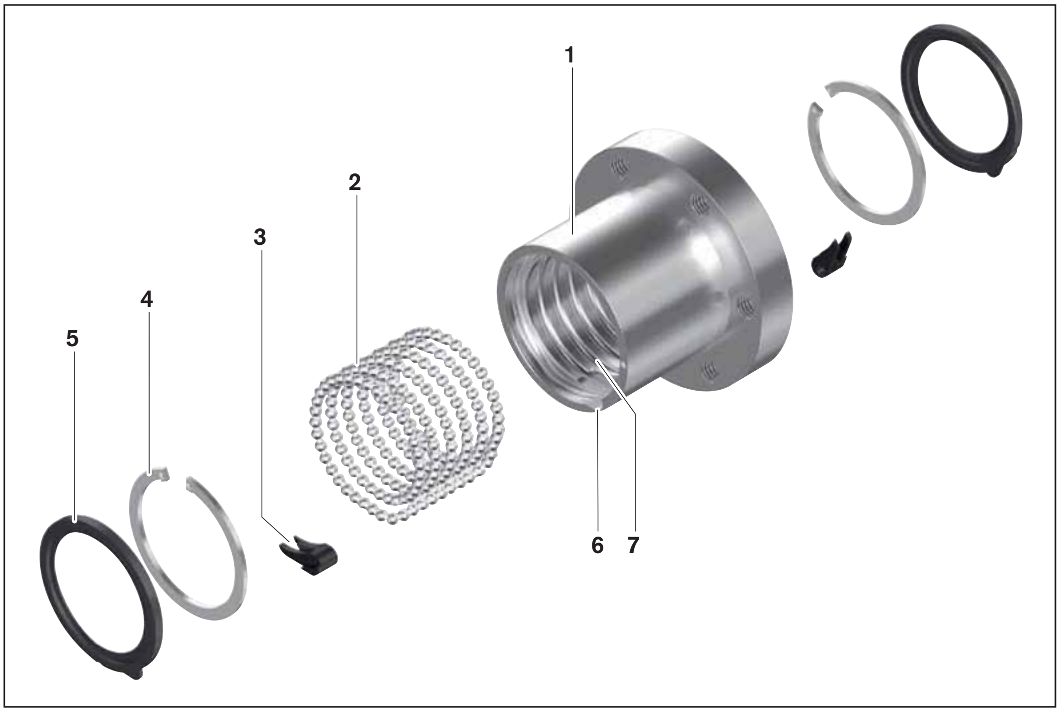
5.1 Principles

5.1.1 System technology

Ball nut

The ball nut is fastened to the moving machine part in the adjacent structure and converts the screw's rotary motion into linear motion via the recirculating balls.

The main elements of a ball nut are illustrated in the following example.

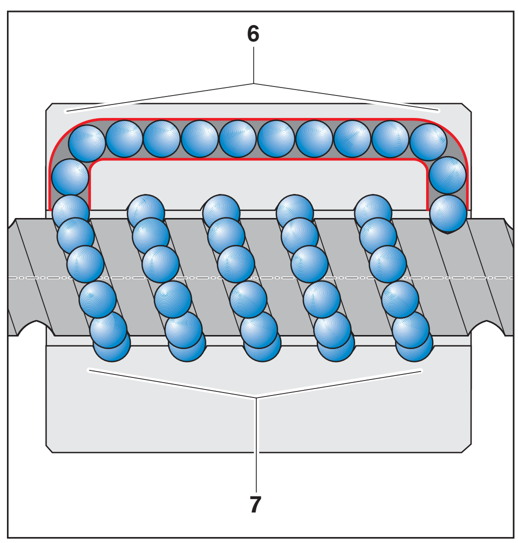


Structural design of a ball nut

1 Nut body
2 Rolling elements (balls)
3 Recirculation piece (ball pick-up)
4 Ball recirculation retaining ring
5 Sealing system
6 Ball return zone
7 Load-carrying zone

Rolling element circuit

The rolling element circuit in a ball nut consists of a load-carrying zone (7) and a return zone (6). In the load-carrying zone, the rolling elements transfer the arising axial forces from the screw to the nut and vice versa. The balls execute several turns around the screw while they are in the load-carrying zone, according to the number of ball track turns in the nut. In the example shown, there are 5 ball track turns. In the return zone, the balls are not loaded and are simply guided back to the load-carrying zone. The recirculation piece (3) picks up the balls at the end of the load-carrying zone and guides them into the return zone and from the return zone back into the load-carrying zone. Various ball recirculation systems have evolved over the course of technical development. The most important of these are described in the following paragraphs.



Principle of rolling element recirculation

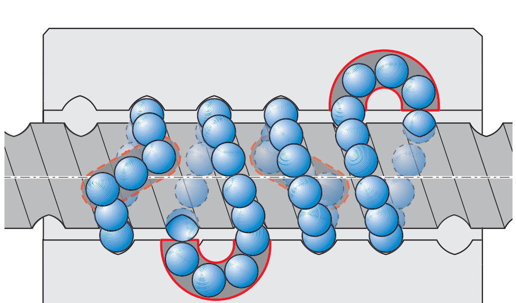
5 Ball screw drives

5.1 Principles

5.1.1 System technology

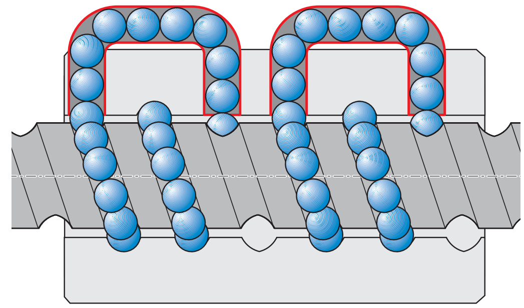
Recirculation systems

Single-turn recirculation



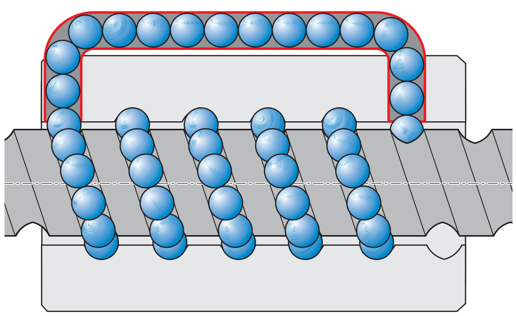
The single-turn recirculation system recirculates the balls from just one ball track turn. A recirculating piece inserted into the nut body guides the balls over the screw shoulder and into the neighboring thread turn.

Multiple-tube recirculation



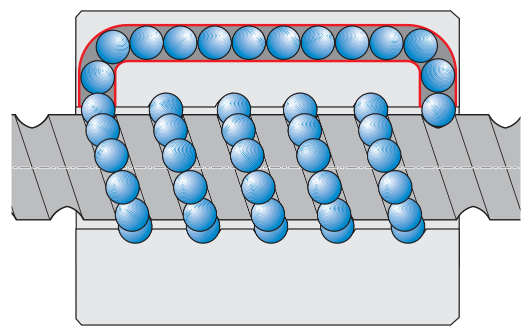
In this particular recirculation system, the balls are brought back to the threading by tubes spanning two or three ball track turns. A nut with several ball circuits will have several recirculation tubes.

Single-tube full recirculation



In this tube-type recirculation system, the balls in a circuit are returned via a single tube that spans the entire nut length and all the ball track turns.

Integrated single-bore full recirculation



In this recirculation system, there is again only one recirculation duct spanning all the ball track turns. The balls are returned through a recirculation bore inside the nut. A nut designed for use with multi-start screws can comprise several ball circuits and will therefore have several recirculation bores.

All the above recirculation systems can be found in ball screw assemblies currently offered on the market. The best technical solution is the integrated single-bore full recirculation system with tangential ball pick-up.

The advantages of this system are:

- High load ratings by using the whole length of the nut
- Quieter running through integration of the recirculation tube into the body of the nut, resulting in a highly rigid return channel with no noise-increasing effect (does not act as a resonance body)
- Uniform torque profile due to tangential ball pick-up

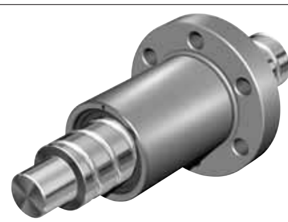
5.1 Principles

5.1.1 System technology

Nut body

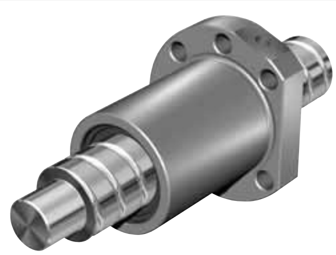
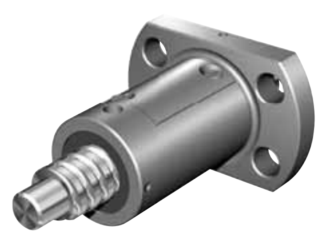
There are various nut forms and flange designs available for incorporating ball nuts into the surrounding structure.

Nut forms

Nut form	Fastening to the adjacent structure and further system characteristics
Screw-in nut 	▮ Screw-in nuts are inserted directly into a mating thread in the adjacent structure.
Cylindrical nut 	▮ Cylindrical nuts are used for compact constructions. ▮ The torque is transferred by a key in the nut body. ▮ The cylindrical nut is axially secured by a ring nut.
Flanged nut 	▮ The flange is used to bolt the nut to the adjacent structure. ▮ The position of the flange (at the end or in the center) depends on the nut series.

Flanged nuts

The different flange shapes on the flanged nuts enable fixing even in restricted spaces. The number of flange holes is adapted to the load rating and/or the potential load on the respective ball nut.

Full flange	Single-flat flange	Double-flat flange
		
“Speed” series nut with full circular flange	“Standard” series nut with a flat on one side of the flange	“Miniature” series nut with flange flattened on both sides

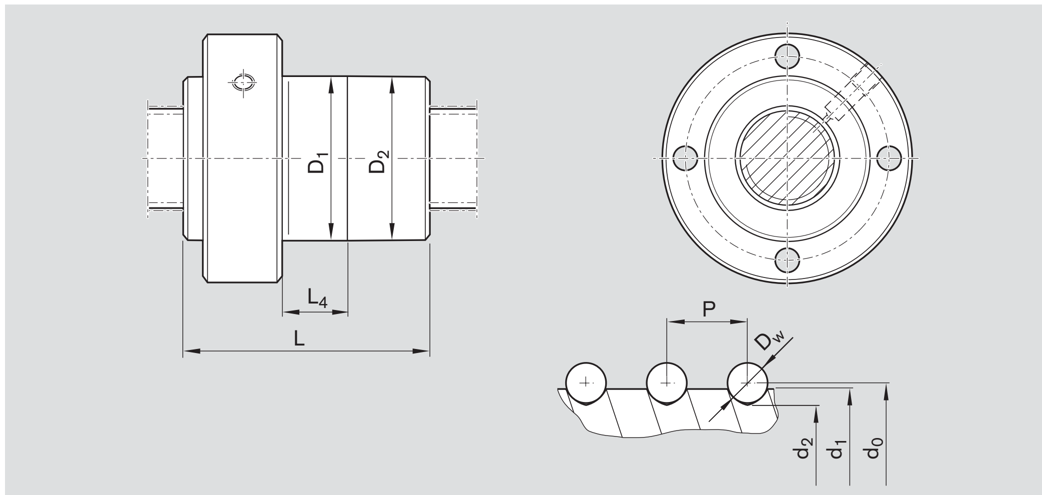
5 Ball screw drives

5.1 Principles

5.1.1 System technology

Nut dimensions

The dimensions of the nuts are specified using the same geometric parameters as for the screw. The number of ball track turns around the screw is also given.



Ball nut geometry

P	=	lead (R = right-hand, L = left-hand)	(mm)
d_0	=	nominal screw diameter	(mm)
d_1	=	screw outside diameter	(mm)
d_2	=	screw core diameter	(mm)
D_W	=	ball diameter	(mm)
a	=	number of load-carrying turns per thread	(–)
b	=	number of load-carrying threads on the screw	(–)
i	=	number of ball track turns for single-start screws (i = a x b for multi-start screws)	(–)
D_1	=	centering diameter of the ball nut body	(mm)
D_2	=	outside diameter of the ball nut body ($D_2 < D_1$)	(mm)
L	=	length of the ball nut	(mm)
L_4	=	length of centering zone	(mm)

Nut sizes

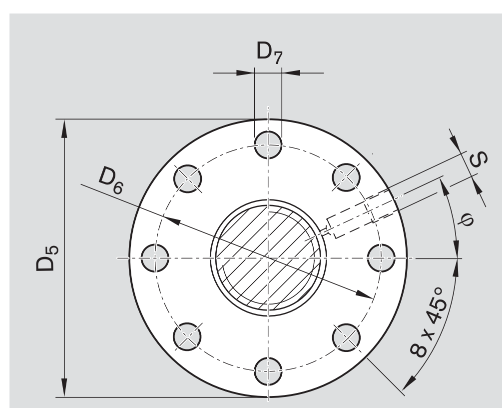
The nominal screw diameter d_0, lead P, thread direction, ball diameter D_W and the number of ball track turns form the nut's size designation: d_0 x P x D_W – i

Example: 32 x 5R x 3.5 – 4 for a nut with a nominal diameter of 32 mm, lead of 5 mm, right-hand thread, ball diameter 3.5 mm, 4 ball track turns for a single thread.

Flanged nut connection dimensions

Rexroth offers its flanged nuts with flange connection dimensions per DIN 69051 Part 5 or according to Rexroth specifications for fastening the nuts to the adjacent structure.

D_5	=	flange diameter	(mm)
D_6	=	pitch circle diameter of the through-holes for bolts	(mm)
D_7	=	diameter of the through-holes for bolts	(mm)
S	=	lube hole thread	(–)
φ	=	lube hole angle	(°)



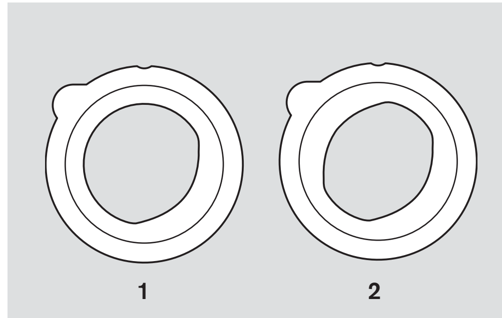
Example of the mounting hole pattern for a flanged ball nut

5.1 Principles

5.1.1 System technology

Sealing system

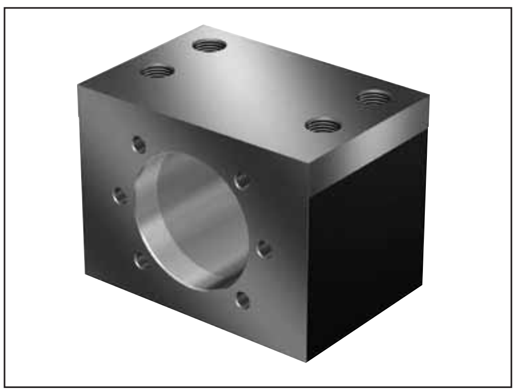
To cater for different applications, various types of seals are available for ball screw drives. Low-friction seals are used for handling applications, applications with clean or covered axes, or for applications requiring very low torque. Standard seals are suitable for use in normal plant and machinery environments. Reinforced seals are used in very dirty environments, e.g. in the woodworking industry.



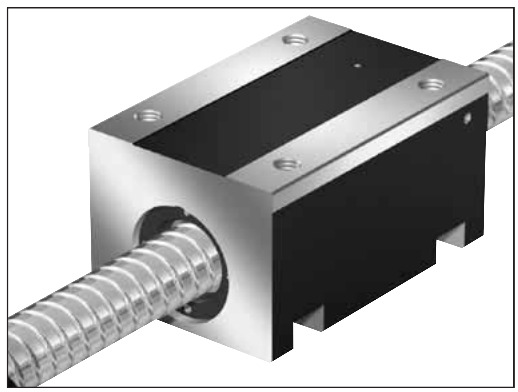
Seals for single-start (1) and two-start (2) screws

Nut housings

Rexroth provides nut housings for easy and low-cost fastening of the nut to the adjacent structure. Nut housings are precision components that can be installed with a minimum of effort. They eliminate the need for customer-built mounting brackets or expensive processing of cast iron parts.



Nut housing for flanged nut



Nut housing for cylindrical nut

End bearings

Normally, a fixed-floating bearing combination is selected for the screw's end fixity. Generally the fixed bearing is on the drive side. Rexroth offers matched bearing-pillow block units that eliminate the need for costly customer-built designs and the search for suitable bearings (see also section 5.1.4.2). Today, preloaded angular-contact thrust ball bearings are generally used for fixed bearings. For the floating bearings, deep groove ball bearings will suffice in most cases.

End fixity			
Bearings		Pillow block units	
Fixed bearing	Floating bearing	Fixed bearing unit	Floating bearing unit
			

5 Ball screw drives

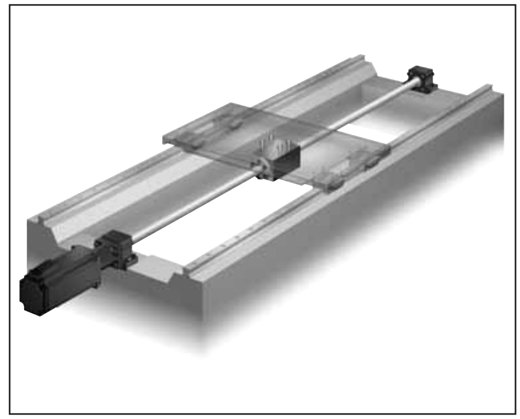
5.1 Principles

5.1.1 System technology

5.1.1.2 Load ratings

Load-bearing capability

Unlike linear motion guides, ball screw drives can only absorb axial forces. They may not be subjected to radial forces or torque loads. These loads must be taken up by the system's linear motion guides.

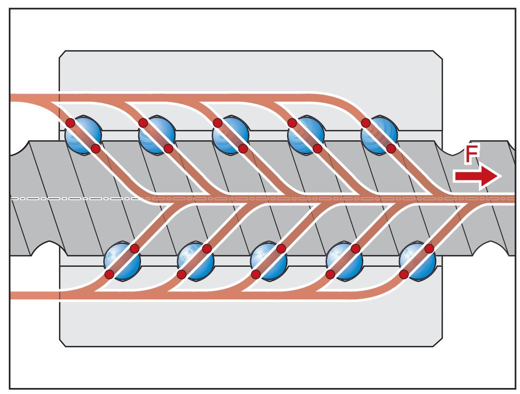


Example of a system with a ball screw drive and two rail guides

Load ratings

The load-carrying capacity of a ball screw assembly in the axial direction is described by the ball nut's static load rating C_0 and dynamic load rating C (for precise definitions, see below). The load ratings are the most important parameters describing the system's performance capability. Details of the load ratings C and C_0 can be found in the product catalogs. The dynamic load ratings have been confirmed by endurance tests at Rexroth. The methods for calculating the load ratings are defined in the standard DIN 69051 Part 4.

Depending on the conditions of use, the screw (buckling risk) and the end bearings can limit the permissible loads or affect the choice of product. Explanations regarding buckling and the end bearings can be found in section 5.1.3.



Flow of forces in the ball screw assembly

Definition of the dynamic load rating C

The axial force of constant magnitude and direction under which a ball screw can theoretically achieve a nominal service life of one million revolutions.

Definition of the static load rating C_0

The static load in the direction of loading which results in a permanent overall deformation of approximately 0.0001 times the ball diameter at the center of the most heavily loaded ball/raceway contact.

5.1 Principles

5.1.1 System technology

5.1.1.3 Preload

Zero backlash
Rigidity

The advantages of ball screw drives as opposed to acme screw drives include the fact that, in addition to the significantly lower friction, the nut can also be preloaded. This makes it possible to achieve zero backlash, which increases the nut's rigidity. Rexroth offers nuts with backlash or with preload, depending on the type of nut. The preload can be adjusted to suit the particular application.

Depending on the method used to achieve preloading, the system will have either two-point or four-point contact. The different ways of adjusting the preload are explained below:

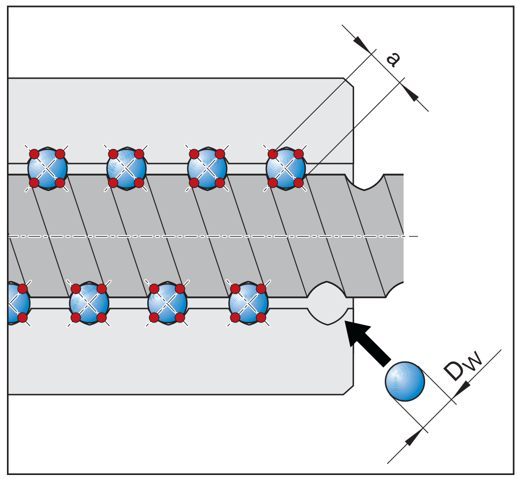
- Ball size selection
- Adjustable-preload single nut
- Double nut
- Shifted single nut

Preloading by ball size selection

Just as with profiled rail systems, this type of ball nut can be preloaded by inserting balls with a specific oversize d_{OS}. When the nut is mounted on the screw, the balls are already slightly elastically deformed without any external load having been applied. This results in four-point contact between the balls and the raceways. The advantages of this type of preload generation are the low costs and the short overall length of the nut.

$$D_W = a + d_{OS}$$

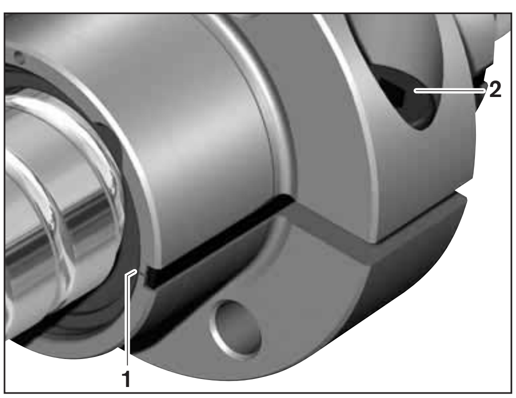
D_W	=	ball diameter	(mm)
a	=	distance between the contact points in the screw and the nut	(mm)
d_{OS}	=	oversize	(mm)



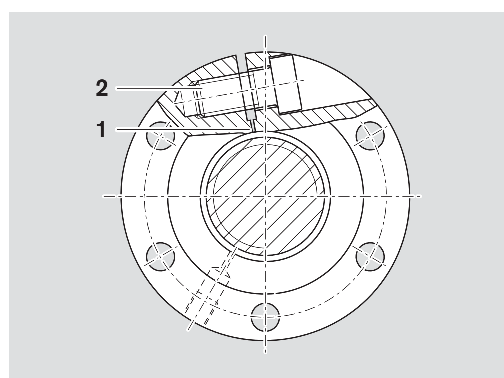
Preloading by inserting oversized balls

Adjustable-preload single nut

With this type of nut, the preload is adjusted via the effective diameter of the nut raceway. The nut body is provided with a narrow slot, whose width (and therefore the preload) is adjusted with an adjusting screw. The balls run under a slight external preload, which induces four-point contact. The advantage of this system is the infinitely variable adjustability of the preload.



Slot and adjusting screw on an adjustable-preload single nut



Preload adjustable via a tangential adjusting screw

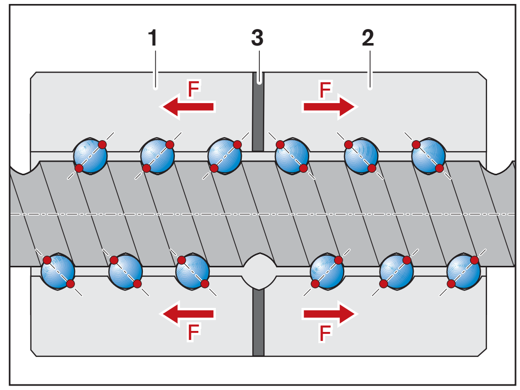
1 Slot
2 Adjusting screw

5.1 Principles

5.1.1 System technology

Double nut

In this type of preload generation, two single nuts are tensioned against each other to a defined level and then secured. This produces a two-point contact combined with an O-arrangement. The operating force is transferred by either of the two nut halves, depending on the direction of the applied load. In the standard series, the level of preload is adjusted via the thickness of the spacer ring (3). Both of the single nuts in the double nut unit will normally have backlash. The advantages of this system are the very good frictional torque characteristics. However, it takes a great deal of effort to match the nuts up with the spacer ring and secure the assembly, and this operation is therefore cost-intensive. The very long overall length compared to single nuts is also a disadvantage.

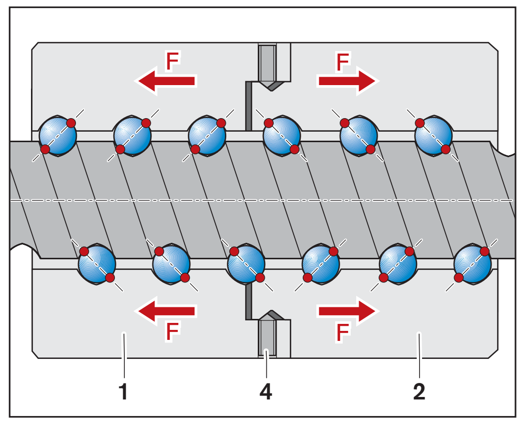


Double nut with spacer ring

Double nut for machine tools

In addition to the spacer ring versions, Rexroth also offers a series with increased load ratings for machine tools and heavy-duty applications. In double nuts, the preload is created by rotating the nut halves relative to each other. The nut halves are then fixed with clamping screws.

The single nuts in the machine tool series are preloaded via ball selection or by shifting (see below).



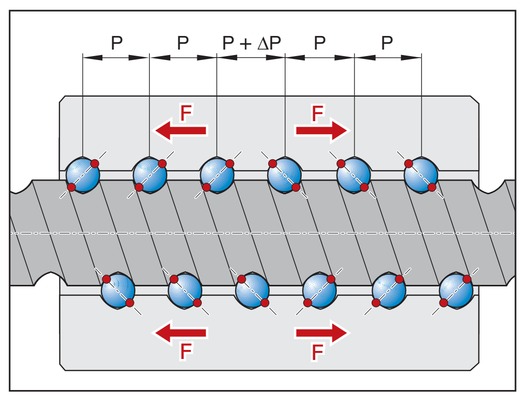
Double nut from the machine tool series

Shifted single nut

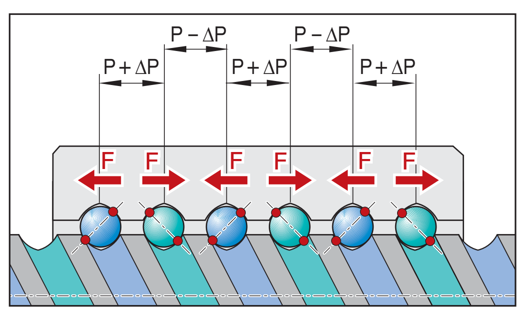
With shifting, the operating principle of the double nut is applied to a single nut. A specific lead offset ΔP is produced in the thread at a point half-way along the nut. The ball track turns before and after the lead offset are thereby tensioned against each other. Just as with double nuts, a two-point contact is produced combined with an O-arrangement. The offset (dimension ΔP) is chosen according to the level of desired preload. This is termed shifting within a ball track turn. Nuts of this type are cheaper to produce than double nuts. The long nut length remains a disadvantage, however. Systems with multi-start screws allow shifting between threads. In principle, this is the effect that would be achieved by integrating the two halves of a double nut into a single nut body.

Shifting within a ball track turn

Shifting between threads



Shifting within a ball track



Shifting between threads

1 Nut A	**4** Clamping screw
2 Nut B	**F** Preload force
3 Spacer ring	**P** Lead

5.1 Principles

5.1.1 System technology

5.1.1.4 Rigidity

Definition of rigidity

Rigidity is understood as being the resistance to elastic deformation. The rigidity R denotes the force required to produce a certain deformation in the direction of loading.

$$R = \frac{\Delta F}{\Delta l} \quad (5\text{-}1)$$

R	=	rigidity	(N/µm)
ΔF	=	change in force	(N)
Δl	=	elastic deformation	(µm)

Rigidity of a ball screw assembly

The rigidity of a ball screw assembly is influenced not only by the ball nut's rigidity but also by all the adjoining parts such as bearings, housing bores, nut housings, etc.

Definition of the overall axial rigidity R_{tot}

The overall axial rigidity R_{tot} is comprised of the component rigidity of the bearing R_{aL}, the screw R_S and the nut R_{nu}.

$$\frac{1}{R_{tot}} = \frac{1}{R_{aL}} + \frac{1}{R_S} + \frac{1}{R_{nu}} \quad (5\text{-}2)$$

R_{tot}	=	overall axial rigidity	(N/µm)
R_{aL}	=	rigidity of the bearing	(N/µm)
R_S	=	rigidity of the screw	(N/µm)
R_{nu}	=	rigidity of the nut unit	(N/µm)

The component with the lowest rigidity is therefore the determining factor for the ball screw assembly's overall axial rigidity R_{tot}. In many cases, the rigidity R_S of the screw will be significantly lower than the rigidity R_{nu} of the nut unit.

In an assembly of size 40 x 10 ($d_0 \cdot P$), for example, the rigidity R_{nu} of the nut unit will be two or three times higher than the rigidity R_S of a screw with a length of 500 mm.

Rigidity of the bearing R_{aL}

The rigidity of the bearing corresponds to the value provided by the bearing manufacturer. Details of the rigidities of Rexroth bearings are shown in the ball screw product catalogs.

Rigidity of the nut unit R_{nu}

The rigidity in the area of the preloaded nut unit is calculated according to DIN 69051 Part 5. Details of the rigidities of the nut units are also given in the product catalogs.

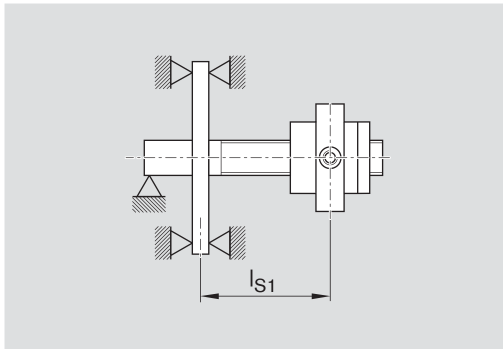
Rigidity of the screw R_S

The rigidity of the screw depends on the screw's cross-sectional area, the screw length, the position of the nut unit on the screw and the type of bearing used. The rigidity of the screw is calculated according to DIN 69051 Part 6. The following two installation examples illustrate the method for calculating the screw rigidity. The product catalogs contain corresponding tables which can be used to compare the screw and nut rigidities at the pre-selection stage. The screw rigidity details relate to a screw length of 1 meter.

5.1 Principles

5.1.1 System technology

Example 1: Ball screw shaft fixed at one end



The complete formula for calculating the screw rigidity R_{S1} is:

(5-3)

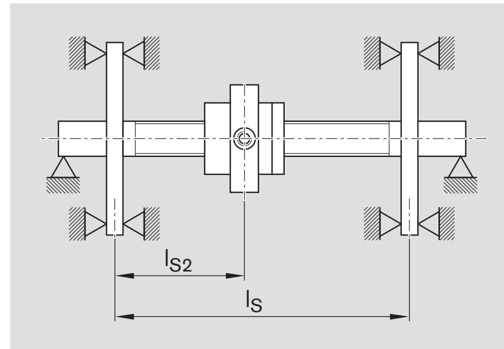
$$R_{S1} = \frac{\pi \cdot (d_0 - D_W \cdot \cos\alpha)^2 \cdot E}{4 \cdot l_{S1} \cdot 10^3} \quad \left(\frac{N}{\mu m}\right)$$

By inserting the values for the material ($E = 210{,}000$ N/mm²) and the ball track geometry ($\alpha = 45°$) and combining the dimensionless values we obtain the following simplified formula:

(5-4)

$$R_{S1} = 165 \cdot \frac{(d_0 - 0.71 \cdot D_W)^2}{l_{S1}} \quad \left(\frac{N}{\mu m}\right)$$

Example 2: Ball screw shaft fixed at both ends



The complete formula for calculating the screw rigidity R_{S2} is:

(5-5)

$$R_{S2} = \frac{\pi \cdot (d_0 - D_W \cdot \cos\alpha)^2 \cdot E}{4 \cdot l_{S2} \cdot 10^3} \cdot \frac{l_S}{l_S - l_{S2}} \quad \left(\frac{N}{\mu m}\right)$$

The simplified formula for calculating the screw rigidity R_{S2} is:

(5-6)

$$R_{S2} = 165 \cdot \frac{(d_0 - 0.71 \cdot D_W)^2}{l_{S2}} \cdot \frac{l_S}{l_S - l_{S2}} \quad \left(\frac{N}{\mu m}\right)$$

The lowest screw rigidity R_{S2min} occurs at the centre of the screw ($l_{S2} = l_S/2$) and thus equals:

(5-7)

$$R_{S2min} = 660 \cdot \frac{(d_0 - 0.71 \cdot D_W)^2}{l_S} \quad \left(\frac{N}{\mu m}\right)$$

R_{S1}	=	rigidity of screw with shaft fixed at one end	(N/µm)
R_{S2}	=	rigidity of screw with shaft fixed at both ends	(N/µm)
E	=	elasticity modulus	(N/mm²)
d_0	=	nominal diameter	(mm)
D_W	=	ball diameter	(mm)
l_{S1}	=	distance between bearing and nut	(mm)
l_{S2}	=	distance between bearing and nut	(mm)
l_S	=	distance between bearing and bearing	(mm)
α	=	contact angle between the ball and the raceway	(°)

5.1 Principles

5.1.1 System technology

5.1.1.5 Accuracy

Standard

DIN 69051 Part 3 (ISO 3408-3) defines the acceptance conditions and the acceptance tests for ball screw assemblies.
Different tolerance grades are specified, with a distinction being made between positioning drives and transport drives.

The most important terms, tolerances and tests are explained in the following paragraphs:
- Travel deviations and variations
- Run-outs and location deviations
- Drag torque variations

Travel deviations and variations

Even with the most advanced production techniques, it is impossible to produce a ball screw with no deviations. The amount of travel deviation is evaluated according to a series of tolerance grades. The evaluation is performed in three steps that are explained below, with reference to the chart on the opposite page.

Travel compensation

If required, a target value for the travel deviation is determined before beginning the actual evaluation process. This target travel deviation, known as the travel compensation c, is the desired deviation from the nominal lead within the useful travel. It is determined by the user and depends on the conditions of use and the specific application. The standard value for travel compensation is zero.

Evaluation over the entire useful length

The travel deviation is evaluated first of all over the entire useful length.
The actual travel deviation is recorded over the useful travel l_u. Since the actual travel deviation is difficult to evaluate, the mean actual travel deviation is determined as the geometric mean of the measurements recorded over the useful travel.
The difference between the travel compensation value and the mean actual travel deviation at the end of the useful travel constitutes the tolerance for mean actual travel deviation e_p, which gives an indication of the screw's average precision over the useful travel l_u. However, a screw might exhibit widely varying accuracy errors that virtually canceled each other out over the useful travel.
The tolerance for mean actual travel deviation e_p would then indicate a high level of precision, even though the screw displayed significant errors.
For this reason, the bandwidth of travel variation around the mean actual travel deviation must also be analyzed. To do this, two lines are drawn parallel to the line for mean actual travel deviation to form an "envelope" enclosing the actual travel deviation curve. The distance between these parallel lines is called the permissible travel variation within the useful travel and is denoted by the symbol ν_{up}. This is the bandwidth for travel variations.

The e_p value is verified for both positioning ball screws and transport ball screws. For positioning ball screws, the ν_{up} valve is verified in addition. The figures for these values can be found in the product catalogs.

l_u		e_p (µm) tolerance grade					ν_{up} (µm) tolerance grade		
>	≤	1	3	5	7	9	1	3	5
400	**500**	8	15	27	63	200	7	13	26
500	**630**	9	16	30	70	220	7	14	29

Extract from the ball screws catalog: values for e_p and ν_{up} according to the useful travel and the tolerance grade

Evaluation over a reference length

A second evaluation step is performed for a reference length of 300 mm. In this case, the value ν_{300p} for the specified tolerance grade may not be exceeded at any point on the screw within the 300mm length.
The tolerance for travel variations within 300 mm of travel is verified for positioning ball screws and for transport ball screws.

	Tolerance grade				
	1	3	5	7	9
ν_{300p} (µm)	6	12	23	52	130

Extract from the ball screws catalog: values for ν_{300p} according to the tolerance grade

5 Ball screw drives

5.1 Principles

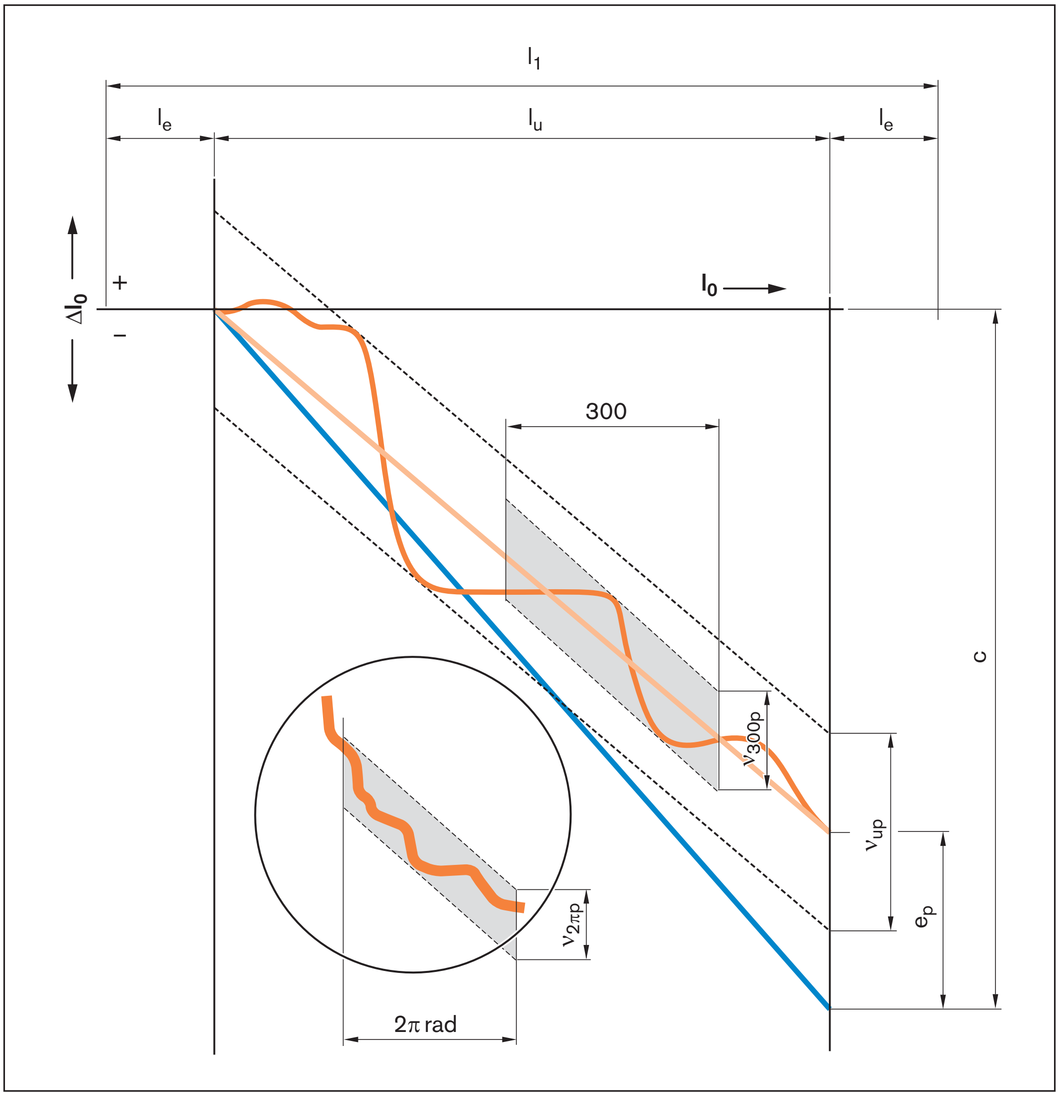
5.1.1 System technology

Evaluation per revolution

The third step is to evaluate the travel deviation per revolution. This value is called the permissible travel deviation within one revolution (2π rad) and is denoted by the symbol $\nu_{2\pi p}$.
This check is only performed for positioning ball screws (precision ball screws).

	Tolerance grade				
	1	3	5	7	9
$\nu_{2\pi p}$ (µm)	4	6	8	10	10

Extract from the ball screws catalog: values for $\nu_{2\pi p}$ according to the tolerance grade



Sample chart for evaluating travel deviations and variations in ball screw assemblies

- Travel compensation (target travel deviation)
- Actual travel deviation
- Mean actual travel deviation
- -- Permissible travel variation (tolerance "envelope")

l_0 = nominal travel
l_1 = thread length
Δl_0 = travel deviation
l_e = excess travel (non-usable length)
l_u = useful travel

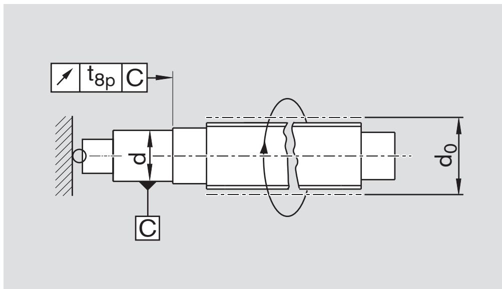
c = travel compensation for useful travel (standard: c = 0)
e_p = tolerance for mean actual travel deviation
ν_{up} = permissible travel variation within useful travel l_u
ν_{300p} = permissible travel deviation within 300 mm travel
$\nu_{2\pi p}$ = permissible travel deviation within one revolution (2π rad)

5.1 Principles

5.1.1 System technology

Run-outs and location deviations (geometric accuracy)

For screws, screw ends and complete ball screw assemblies, DIN 69051 Part 3 (ISO 3408-3) specifies various permissible run-outs (radial and axial) and location deviations. As with travel deviations, various tolerance grades are defined for run-outs and location deviations.



Example: Axial run-out t_{8p} of the shaft (bearing) face of the ball screw shaft in relation to the bearing diameter

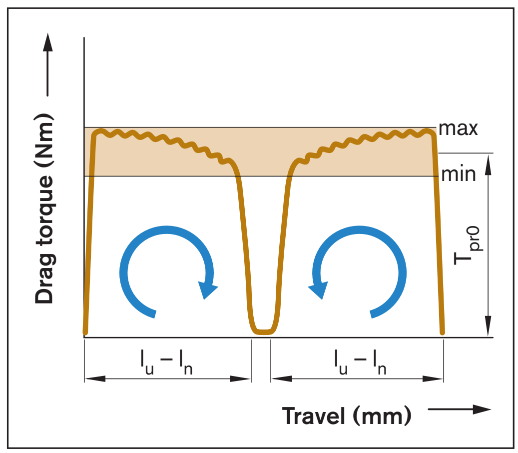
Nominal diameter d_0 (mm)		Axial run-out t_{8p} (µm) for tolerance grade		
>	≤	1	3	5, 7, 9
6	63	3	4	5
63	125	4	5	6
125	200	–	6	8

Extract from the ball screws catalog: values for t_{8p} according to the tolerance grade

Drag torque variations

As an additional criterion for evaluating a ball screw drive, DIN 69051 Part 3 specifies the dynamic drag torque. This should ideally remain constant over the entire travel and in both directions of movement.

T_{pr0} = dynamic drag torque without seals (Nm)
$l_u - l_n$ = useful travel minus length of the ball nut (mm)



Qualitative representation of the dynamic drag torque

If required, measurement reports for travel deviations, run-outs and location deviations, and drag torque variations can be requested from Rexroth.

5.1.1.6 Dynamic drag torque

The overall dynamic drag torque T_0 is the sum of the nut unit's dynamic drag torque without seals T_{pr0} and the dynamic drag torque of the two seals T_{RD}. These dynamic drag torque values are given in tables in the product catalogs.

(5-8) $T_0 \quad T_{pr0} + T_{RD}$

T_0 = overall dynamic drag torque (Nm)
T_{pr0} = dynamic drag torque without seals (Nm)
T_{RD} = dynamic drag torque of the two seals (Nm)

5.1 Principles

5.1.1 System technology

5.1.1.7 Characteristic speed and maximum linear speed

Rexroth ball screws can be operated at very high speeds due to their internal ball recirculation system. Characteristic speeds of up to 150,000 mm/min are possible, depending on the nut type. The theoretically possible maximum linear speed v_{max} can be calculated from the characteristic speed and the screw lead P. The values for v_{max} can be found in the product catalogs.

In practice, the actually attainable linear speeds will depend heavily on factors such as the preload and the duty cycle. They are generally restricted by the critical screw speed, see section 5.1.3.3.

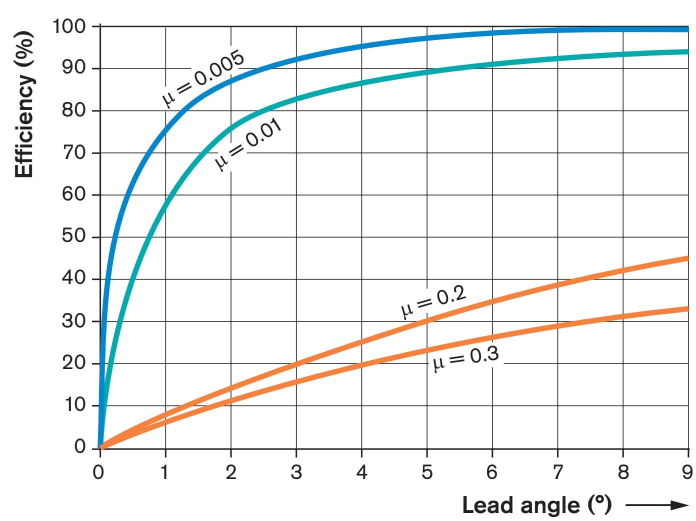
$_{0} \cdot n \leq 150000$ mm/min

$$\text{(5-9)} \qquad v_{max} = \frac{(d_0 \cdot n) \cdot P}{d_0} = \frac{150000\,\frac{\text{mm}}{\text{min}} \cdot P}{d_0}$$

$d_0 \cdot n$	=	characteristic speed	(mm/min)
d_0	=	nominal screw diameter	(mm)
n	=	rotary speed	(min^{-1})
v_{max}	=	theoretical maximum linear speed	(mm/min)
P	=	lead	(mm)

5.1.1.8 Mechanical efficiency

Due to the use of rolling elements, sliding friction is largely avoided (with the exception of contact seals, for example). Ball screw assemblies therefore have a very high degree of mechanical efficiency. As a result, they can be put to very effective use as powerful machine components for a wide variety of applications.



Comparison: Efficiency of ball screw assemblies versus that of acme screws

- Ball screw assembly with two-point contact
- Ball screw assembly with four-point contact
- Acme screw

μ = friction coefficient

5.1 Principles

5.1.1 System technology

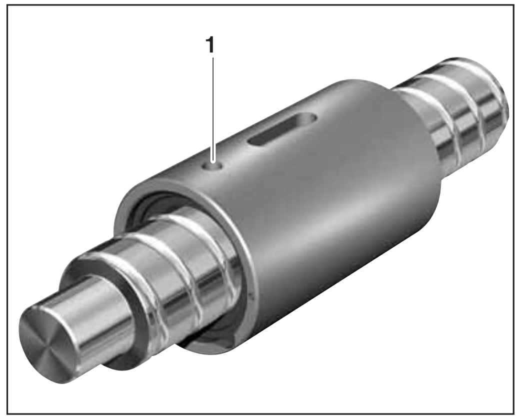
5.1.1.9 Lubrication

Lube port

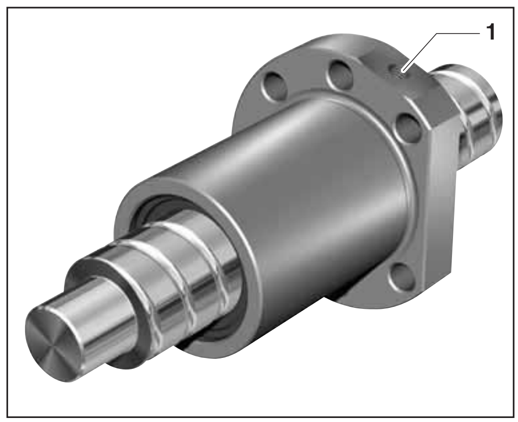
Just like every other type of rolling bearing, ball screws must be adequately lubricated. They can be lubricated with oil or grease. Normally, the ball nut is lubricated via a lube port (1). In the case of flanged nuts, the lube port is situated on the flange. The lubricant quantities depend on the size of the ball nut. The in-service lubrication intervals depend on the lead and the loads applied. All the relevant lubrication details are provided in the Rexroth product catalogs.

Short stroke

During a very short stroke, the balls do not make complete turns and the lubricant is not distributed optimally in the nut. This can result in premature wear. To avoid this, occasional longer strokes should be performed, which can also be designed as lubricating strokes for simultaneous in-service lubrication.



Cylindrical ball nut



Flanged ball nut

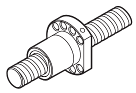
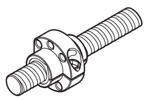
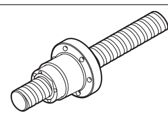
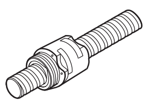
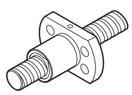
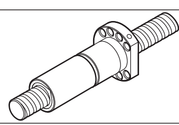
5 Ball screw drives

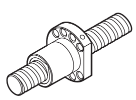
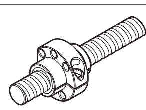
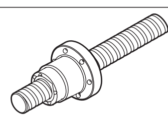
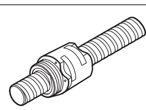
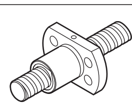
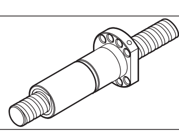
5.1 Principles

5.1.2 Product selection

5.1.2.1 Guide to choosing the right product

System characteristics

Ball nut type		Load-carrying capability	Rigidity	Accuracy	Friction
Single nut, Standard series		+++	+++	+++	++
Adjustable nut, Standard series		+++	+++	+++	++
Single nut, Speed series		++	+	++	++
Single nut, eLINE series		+	+	+	++
Single nut, Miniature series		+	+	++	++
Double nut		+++	+++	+++	+++

Ball nut type		Speed	Noise characteristics[1)]	Lubrication requirement	Costs
Single nut, Standard series		++	++	++	++
Adjustable nut, Standard series		++	++	++	++
Single nut, Speed series		+++	+++	++	++
Single nut, eLINE series		+	+	++	+++
Single nut, Miniature series		+	++	+++	++
Double nut		++	++	+	+

+++ Very good ++ Good + Satisfactory o Adequate

1) at the same linear speed

5.1 Principles

5.1.2 Product selection

5.1.2.2 Product selection procedure

When choosing a ball screw assembly, it is not sufficient simply to calculate the life expectancy. There is always a risk that the screw might buckle under excessive axial loading. The permissible axial screw load must therefore also be checked. In systems with driven screws, the critical speed must be taken into account when determining the maximum linear speed. To ensure that the overall system will operate reliably and safely, the end bearings and the drive unit must also be checked by performing the necessary calculations.

The following procedure is recommended for selection and dimensioning of a ball screw drive.

Procedure		Section
Step 1	Define the requirements	5.1.3.1
Step 2	Select the appropriate ball screw assembly	5.1.2.1 5.1.2.3
Step 3	Calculate the life expectancy	5.1.3.2
Step 4	Calculate the critical speed	5.1.3.3
Step 5	Calculate the permissible axial screw load (buckling)	5.1.3.4
Step 6	Calculate the end bearings	5.1.3.5
Step 7	Calculate the drive torque and the drive power	5.1.3.6
Result	Ordering details with part numbers	(Product catalog)

Rexroth provides a special design calculation service for selecting the appropriate ball screw drives.

5.1.2.3 Pre-selection

For pre-selection, the desired service life and an initial estimation of the average load can be used as a basis for calculating the required load rating of the ball screw.

$$C = F_m \cdot \sqrt[3]{\frac{L}{10^6}} \tag{5-10}$$

C = dynamic load rating (N)
F_m = equivalent dynamic axial load (N)
L = nominal life in revolutions (–)

Once the load rating has been determined, a suitable ball nut with the next highest load rating can be selected in order to perform the actual design calculations.

5.1 Principles

5.1.3 Calculations

5.1.3.1 Defining the requirements

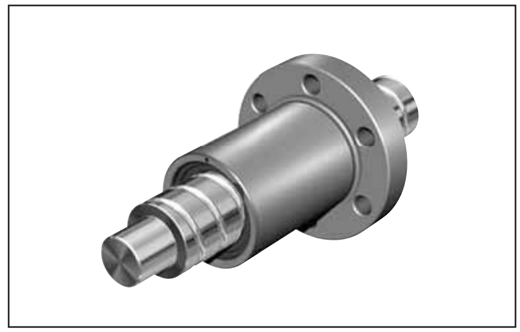
A number of different geometric and operating parameters have to be defined before a ball screw can be dimensioned. All further design calculations are then based on these values. The required parameters relate to:

- the ball screw drive
- the application layout
- the dynamic cycle
- the load scenario

Ball screw drive

Specific details of the pre-selected ball screw assembly are required in order to perform the design calculations. These are the ball nut type, the size, the nut dimensions and its specific characteristics. All of these values can be found in the Rexroth product catalogs.

Ball screw assembly details		
Parameter		**Unit**
Nominal diameter	d_0	mm
Lead	P	mm
Ball diameter	D_W	mm
Number of ball track turns	i	–
Dynamic load rating	C	N
Static load rating	C_0	N
Preload factor	X_{pr}	–
Maximum linear speed	v_{max}	m/min

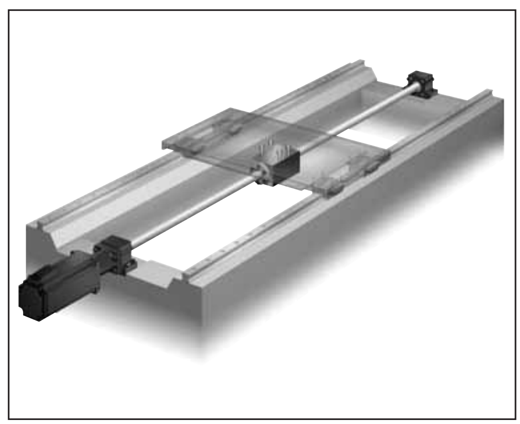


Example: Ball screw assembly with flanged nut from the Standard series, as determined at the pre-selection stage

Application layout

Application layout is a collective term that covers all the relevant geometric parameters. As a rule, this is determined from a drawing of the machine or installation showing all the design dimensions. Details of the masses moved and of the type of bearings used for the end fixity of the unit (e.g. fixed-floating) are also required. The required service life of the installation will depend on the specific application. Motors, gear units and transmission ratios are also considered when determining the application layout.

Application layout details		
Parameter		**Unit**
Mass of the table	m	kg
Required service life in revolutions	L_{req}	–
Required service life in hours	$L_{h\,req}$	h
Screw length	l_1	mm
Maximum stroke length	l_n, l_K	mm
Bearing coefficients	f_{nk}, f_{Fk}	–



Example: Application layout of a ball screw assembly with fixed-floating end bearings and motor, combined with 2 guide rails and 4 runner blocks

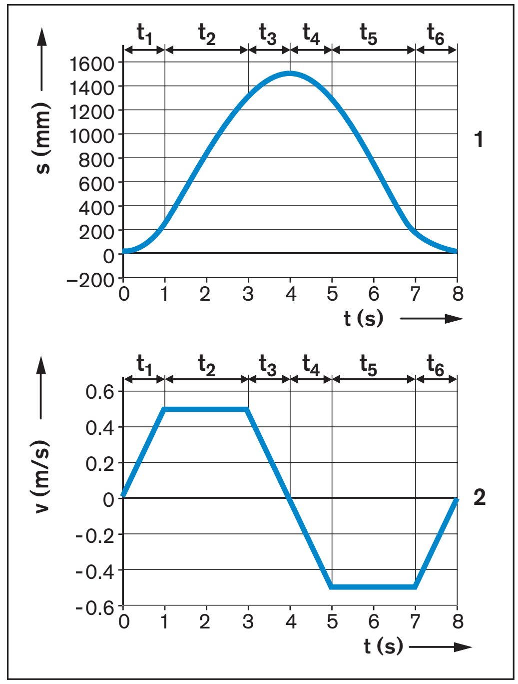
5.1 Principles

5.1.3 Calculations

Dynamic cycle

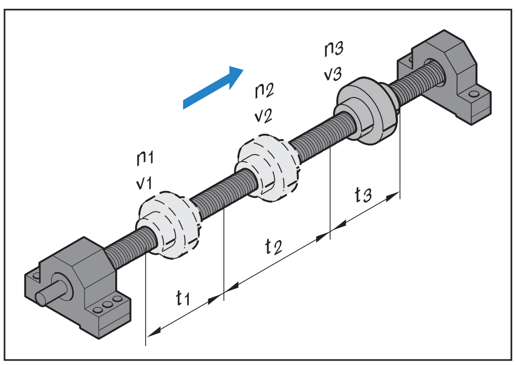
The next step is to determine a reference cycle for the application. This cycle represents the expected dynamic motion sequences and forms the basis for calculating the nominal life. Cycles consist of several phases representing the individual operating steps that the ball screw drive is to perform (e.g. acceleration, braking, processing/machining, etc.). The time, travel, linear speed, acceleration and rotary speed must be determined for each phase. The ball screw drive's duty cycle is required for calculating the life expectancy of the ball screw drive in the specific machine or installation.

Dynamic cycle details		
Parameter		**Unit**
Phase	n	–
Time for phase 1 … n	$t_1 \dots t_n$	s
Travel in phase 1 … n	$s_1 \dots s_n$	mm
Linear speed in phase 1 … n	$v_1 \dots v_n$	m/s
Acceleration in phase 1 … n	$a_1 \dots a_n$	m/s^2
Rotary speed in phase 1 … n	$n_1 \dots n_n$	min^{-1}
Duty cycle of the machine	$DC_{machine}$	%
Duty cycle of the ball screw drive	DC_{BS}	%

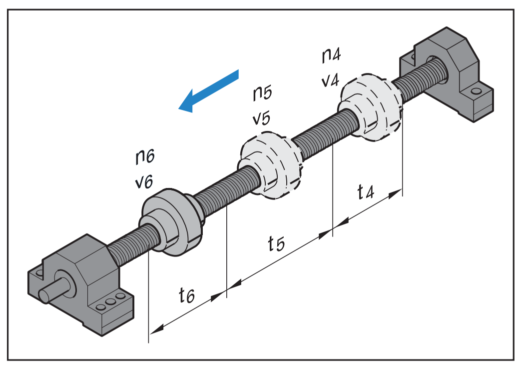


Example of a simple dynamic cycle

1 Travel-time curve
2 Speed-time curve



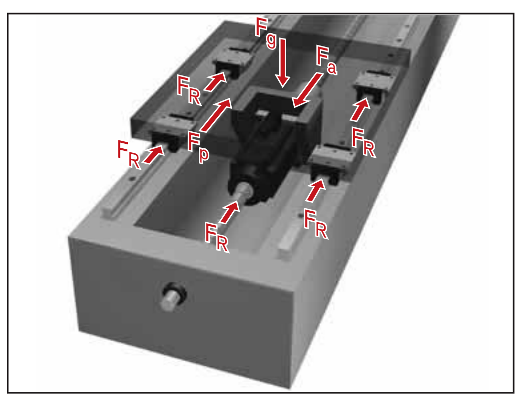
Example of a simple dynamic cycle: forward stroke



Example of a simple dynamic cycle: return stroke

Load scenario

A ball screw drive can only take up forces acting in the axial direction. All other loads must be carried by the guide units. Depending on the application, the axial forces may include weight forces F_g, acceleration forces F_a, process forces F_p, and friction forces F_R.



Example showing an axially effective process force F_p

5.1 Principles

5.1.3 Calculations

The table below provides a summary of the forces that may arise in a system with a ball screw drive.

Force	Formula	Description
Weight force	$F_g = m \cdot g$	The effective weight force F_g is calculated from the mass m and the acceleration due to gravity $g = 9.81\ m/s^2$.
Acceleration force	$F_a = m \cdot a$	The effective acceleration force is the force that must be applied to accelerate a mass.
Friction force	$F_R = \mu \cdot F_N$	The effective friction force is opposed to the direction of movement. Its magnitude is determined, among other factors, by the ball screw assembly's preload, load, sealing and lubrication as well as by the end bearings and the guides.
Process force	F_p	The effective process forces will depend on the specific processing operation. These may be, for instance, forces arising during molding/extrusion, forming, machining, etc.

When performing calculations, particular attention must be paid to the direction in which the individual forces act.

5.1.3.2 Life expectancy

The nominal life calculation for ball screw assemblies is the same as that used for rotary rolling bearings and is similar to the nominal life calculation for linear motion guides. It is usually performed using the number of revolutions and the rotary speed. For precise calculation of the nominal life the load and dynamic data used must be as accurate as possible.

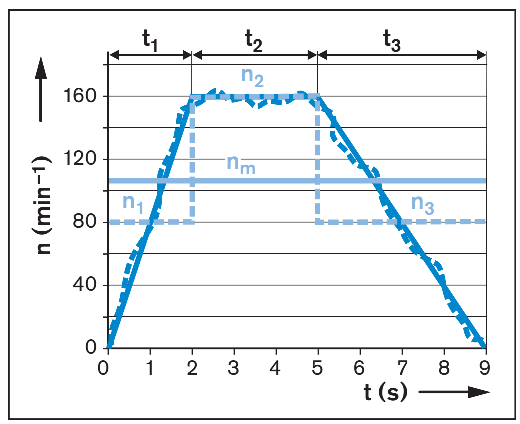
For applications with varying speed and load parameters, the average rotary speed n_m and the equivalent (average) dynamic load F_m must be calculated first. The nominal life calculation is then performed using these average values.

Average rotary speed

If the rotary speed varies in steps over n phases of the speed cycle, the average rotary speed is calculated from the discrete speed values for the individual phases. For definitions of the terms “cycle” and “discrete time steps,” see Chapter 2.

$$(5\text{-}11)\quad n_m = \frac{|n_1| \cdot q_{t1} + |n_2| \cdot q_{t2} + \ldots + |n_n| \cdot q_{tn}}{100\%}$$

n_m	=	average rotary speed	(min^{-1})
$n_1 \ldots n_n$	=	rotary speed in phases 1 … n	(min^{-1})
$q_{t1} \ldots q_{tn}$	=	discrete time steps in phases 1… n	(%)



Example of a simple rotary speed cycle

- Actual speed profile
- Approximated speed profile
- Average speed in phase n
- Average speed over the entire speed cycle

5.1 Principles

5.1.3 Calculations

Taking preload into account

In the case of preloaded ball nut systems, the preload must be taken into account when calculating the nominal life. To determine whether the preload will have an effect on the service life, the internal preload force of the ball nut must first be calculated.

$$F_{pr} = X_{pr} \cdot C \tag{5-12}$$

F_{pr}	=	internal axial load on the ball nut due to the preload	(N)
X_{pr}	=	preload factor	(–)
C	=	dynamic load rating	(N)

Preload	Preload factor X_{pr}
2% of C	0.02
3% of C	0.03
5% of C	0.05
7% of C	0.07
10% of C	0.10

The preload force can be used to calculate the load point at which the raceways inside the ball nut are relieved due to the force exerted by an external load, i.e. the preload becomes zero.

$$F_{lim} = 2.8 \cdot F_{pr} \tag{5-13}$$

This effect is known as “lift-off” and represents the limit for the external load. The symbol for the lift-off force is F_{lim}.

F_{lim}	=	lift-off force	(N)
F_{pr}	=	preload force	(N)

A distinction therefore has to be made between two cases:

Case 1: $F > F_{lim}$
If the external axial force acting on the ball screw assembly in phase n is greater than the lift-off force, then the preload need not be considered when calculating the nominal life.

Preload may be disregarded:
$F_n > 2.8 \cdot F_{pr}$

$$F_{eff\,n} = |F_n| \tag{5-14}$$

Case 2: $F \le F_{lim}$
If the external axial force acting on the ball screw assembly in phase n is less than or equal to the lift-off force, then the preload will have an effect on the nominal life. The effective load $F_{eff\,n}$ must be calculated.

Preload must be considered:
$F_n \le 2.8 \cdot F_{pr}$

$$F_{eff\,n} = \left(\frac{|F_n|}{2.8 \cdot F_{pr}} + 1\right)^{\frac{3}{2}} \cdot F_{pr} \tag{5-15}$$

F_n	=	load on ball screw assembly during phase n	(N)
F_{pr}	=	preload force	(N)
$F_{eff\,n}$	=	effective axial load during phase n	(N)

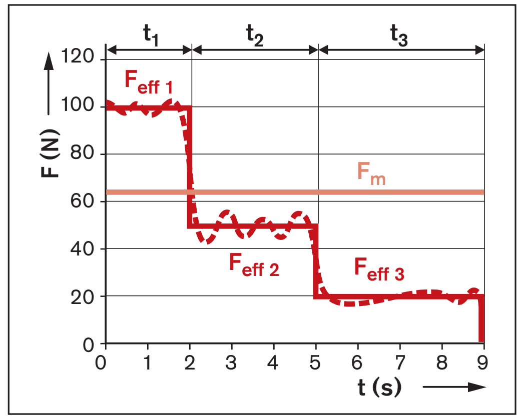
5 Ball screw drives

5.1 Principles

5.1.3 Calculations

Equivalent dynamic axial load

If the load on the ball screw assembly varies in steps, the average axial load must be determined before performing the nominal life calculation. The equivalent dynamic axial load F_m is obtained from the individual loads F_n during the phases n.



Example of a simple load cycle at constant rotary speed

- Actual force profile
- Approximated force profile
- Equivalent dynamic load

Equivalent dynamic axial load at constant speed:

$$F_m = \sqrt[3]{\left(F_{eff\ 1}\right)^3 \cdot \frac{q_{t1}}{100\%} + \left(F_{eff\ 2}\right)^3 \cdot \frac{q_{t2}}{100\%} + \ldots + \left(F_{eff\ n}\right)^3 \cdot \frac{q_{tn}}{100\%}} \qquad (5\text{-}16)$$

Equivalent dynamic axial load at varying speed:

$$F_m = \sqrt[3]{\left(F_{eff\ 1}\right)^3 \cdot \frac{|n_1|}{n_m} \cdot \frac{q_{t1}}{100\%} + \left(F_{eff\ 2}\right)^3 \cdot \frac{|n_2|}{n_m} \cdot \frac{q_{t2}}{100\%} + \ldots + \left(F_{eff\ n}\right)^3 \cdot \frac{|n_n|}{n_m} \cdot \frac{q_{tn}}{100\%}} \qquad (5\text{-}17)$$

F_m	= equivalent dynamic axial load	(N)
$F_{eff\,1} \ldots F_{eff\,n}$	= effective load during phases 1 … n	(N)
n_m	= average speed	(min^{-1})
$n_1 \ldots n_n$	= speed during phases 1 … n	(min^{-1})
$q_{t1} \ldots q_{tn}$	= discrete time steps for phases 1 … n	(%)

5.1 Principles

5.1.3 Calculations

Nominal life

The nominal life is expressed by the number of revolutions or number of operating hours at constant speed that will be attained or exceeded by 90% of a representative sample of identical ball screws before the first signs of material fatigue become evident.

Nominal life in revolutions

The nominal life in revolutions is designated as L and is calculated using the following formula:

$$L = \left(\frac{C}{F_m}\right)^3 \cdot 10^6 \qquad (5\text{-}18)$$

L	=	nominal life in revolutions	(–)
C	=	dynamic load rating	(N)
F_m	=	equivalent dynamic axial load on the ball screw	(N)

Nominal life in hours

The nominal life in hours L_h is calculated from the average rotary speed:

$$L_h = \frac{L}{n_m \cdot 60} \qquad (5\text{-}19)$$

L_h	=	nominal life in hours	(h)
L	=	nominal life in revolutions	(–)
n_m	=	average speed	(min^{-1})

Machine operating hours

Since the required service life of the machine is generally also specified, the life of the ball screw must be recalculated in terms of the duty cycle.

$$L_{h\ machine} = L_h \cdot \frac{DC_{machine}}{DC_{BS}} \qquad (5\text{-}20)$$

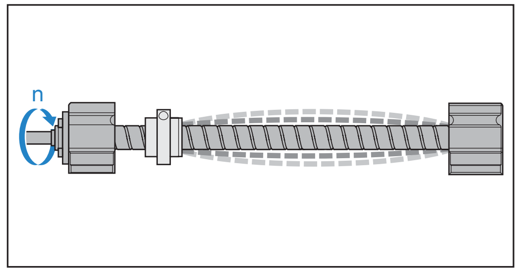
$L_{h\ machine}$	=	nominal machine service life in hours	(h)
L_h	=	nominal ball screw service life in hours	(h)
$DC_{machine}$	=	machine duty cycle	(%)
DC_{BS}	=	ball screw duty cycle	(%)

5.1 Principles

5.1.3 Calculations

5.1.3.3 Critical speed

The rotation of the screw causes bending vibrations (also known as screw whip). The frequency of these vibrations is the screw's rotation frequency. The "critical speed" is the rotary speed that is equivalent to the first order frequency of the screw. If the ball screw assembly is operated at the critical speed, resonance occurs, which can lead to destruction of the system. To avoid this, the critical speed must be determined when performing the design calculations for the ball screw.



Bending vibrations (screw whip)

Critical speed n_k

The critical speed n_k depends on:

- the type of end bearings, coefficient f_{nk}
- the screw's core diameter d_2
- the critical screw length l_n, i.e. the maximum unsupported screw length.
  In the case of ball nuts with backlash, the critical screw length is the same as the bearing-to-bearing length l_1. In preloaded systems, the position of the ball nut is taken into account.

The product catalog contains charts for quickly checking the calculation results. When dimensioning and selecting ball screw drives, the operating speed should never be more than 80% of the critical speed. The characteristic speed and the maximum permissible linear speed must not be exceeded.

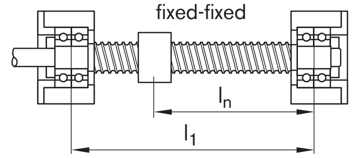
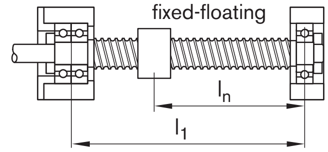
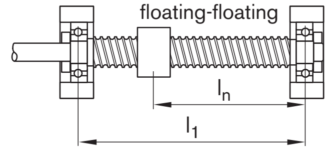
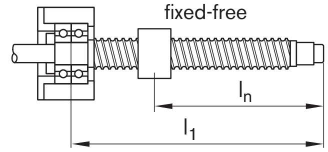
The following measures can be taken to ensure that the screw speed remains outside the critical speed range:

- Increase the screw diameter.
- Choose appropriate end bearings.
- Use preloaded ball nuts instead of nuts with backlash.
- Use screw supports (see section 5.3.1.1).

(5-21) $$n_k = f_{nk} \cdot \frac{d_2}{l_n^2} \cdot 10^7 \quad (\text{min}^{-1})$$

(5-22) $$n_{k\,perm} = n_k \cdot 0.8$$

n_k	=	critical speed	(min^{-1})
$n_{k\,perm}$	=	permissible operating speed	(min^{-1})
f_{nk}	=	coefficient as a function of the end bearings	(–)
d_2	=	screw core diameter (see product catalog)	(mm)
l_1	=	bearing-to-bearing distance	(mm)
l_n	=	critical screw length for preloaded nut systems (For nuts with backlash: $l_n = l_1$)	(mm)

End fixity	Coefficient f_{nk}
fixed-fixed 	27.4
fixed-floating 	18.9
floating-floating 	12.1
fixed-free 	4.3

Driven nuts

Rexroth also offers drive units with driven nuts. When using driven nuts, less energy is introduced into the vibratory system because eccentricities within the rotating system are avoided and good axial and radial run-out is maintained. If resonance should occur, the lower energy input ensures that the consequences will be far less destructive for an optimized system with driven nut than they would be for a system with a rotating screw. For drive units with driven nut, see section 5.3.2.

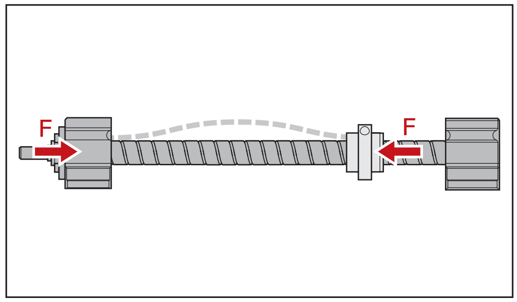
5.1 Principles

5.1.3 Calculations

5.1.3.4 Permissible axial load on screw (buckling load)

Buckling

Buckling stress is a special instance of compressive stress. If a rod (in this case, the screw shaft) is subjected to a compressive force in the axial direction, it will begin to bend in the shape of a bow. The change in shape increases rapidly with increasing load.



Buckling caused by axial loading

Axial loads occur in ball screw drives as a result of acceleration, friction and weight and process forces. The resultant stress depends on:

- the end bearings, coefficient f_{Fk}
- the screw's core diameter d_2
- the effective buckling length l_k of the screw, i.e. the maximum unsupported screw length in the direction of the force's flow between the ball nut and the end bearing.

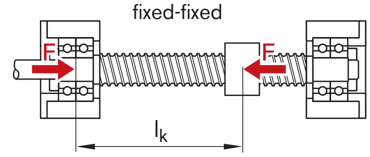
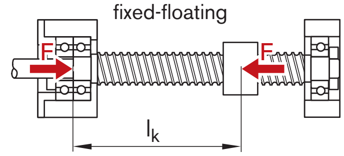
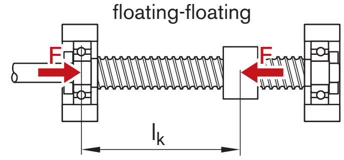
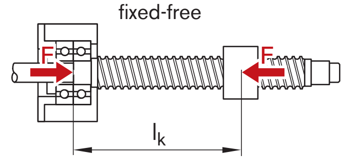
The permissible axial screw load F_k can be calculated from these variables. Just as for the critical speed, the product catalog contains charts allowing a quick cross-check on the buckling load. When dimensioning and selecting ball screw drives, a buckling safety factor of at least 2 should be used when calculating the permissible axial load.

Permissible axial screw load F_k

(5-23) $$F_k = f_{Fk} \cdot \frac{d_2^4}{l_k^2} \cdot 10^4 \quad (N)$$

(5-24) $$F_{k\,perm} = \frac{F_k}{2}$$

F_k	=	theoretical buckling load of the screw	(N)
$F_{k\,perm}$	=	permissible axial load on the screw in service	(N)
f_{Fk}	=	coefficient as a function of the end bearings	(–)
d_2	=	screw core diameter (see product catalog)	(mm)
l_k	=	effective buckling length of the screw	(mm)

End fixity	Coefficient f_{Fk}
fixed-fixed 	40.6
fixed-floating 	20.4
floating-floating 	10.2
fixed-free 	2.6

The following measures can be taken to avoid buckling:

- Increase the screw diameter.
- Choose appropriate end bearings.

5.1 Principles

5.1.3 Calculations

5.1.3.5 End bearings

The end bearings are calculated using the values determined for the rotary speed and the loads on the ball screw assembly. The nominal life must be calculated in accordance with the manufacturer's specifications for the particular type of bearing used.

The calculation method and the corresponding formulas for calculating the bearings used by Rexroth are given in the product catalog.

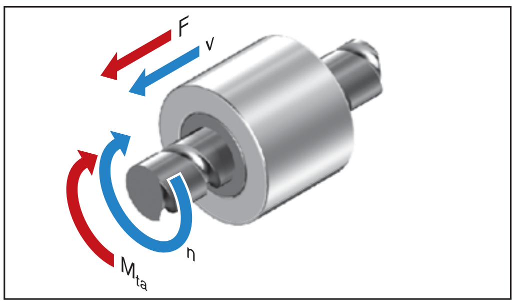
5.1.3.6 Drive torque and drive power

The following formulas can be used for an initial estimation of the required drive torque and power.

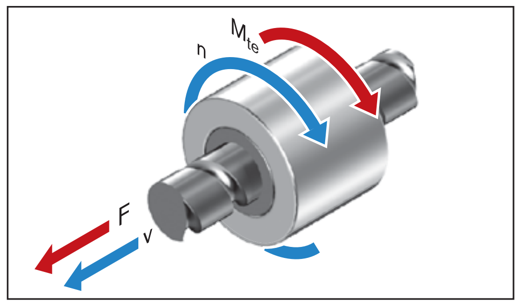
Definitions of drive torque and transmitted torque

Drive torque M_{ta}:
An applied drive torque M_{ta} causes the screw to rotate. As a reaction to the screw's rotation, a linear force F is generated in the ball nut, which causes linear motion of the nut.

Transmitted torque M_{te}:
The screw moves under the action of a thrust force F in the axial direction. As a reaction, a transmitted torque M_{te} is generated, causing the nut to rotate, provided there is no self-locking effect due to the lead angle.



Conversion of rotary to linear motion



Conversion of linear to rotary motion

$$M_{ta} = \frac{F \cdot P}{2000 \cdot \pi \cdot \eta} \quad (\mathrm{Nm}) \tag{5-25}$$

$$M_{te} = \frac{F \cdot P \cdot \eta'}{2000 \cdot \pi} \quad (\mathrm{Nm}) \tag{5-26}$$

M_{ta}	=	drive torque	(Nm)
M_{te}	=	transmitted torque	(Nm)
F	=	operating load	(N)
P	=	lead	(mm)
η	=	mechanical efficiency η ≈ 0.9 for drive torque η′ ≈ 0.8 for transmitted torque	(–)

Drive power

$$P_a = \frac{M_{ta} \cdot n}{9550} \quad (\mathrm{kW}) \tag{5-27}$$

P_a	=	drive power	(kW)
M_{ta}	=	drive torque	(Nm)
n	=	rotary speed	(min^{-1})

This calculation serves merely to arrive at an initial estimation, since it takes only the ball screw into account. For precise calculation, additional influencing factors such as the guide units, the end bearings, mass moments of inertia and dynamic drag torques must also be taken into account. The overall dynamic drag torque T_0 is the sum of the dynamic drag torque of the preloaded nut T_{pr0} and the dynamic drag torque of the two seals T_{RD} (see section 5.1.1.6).

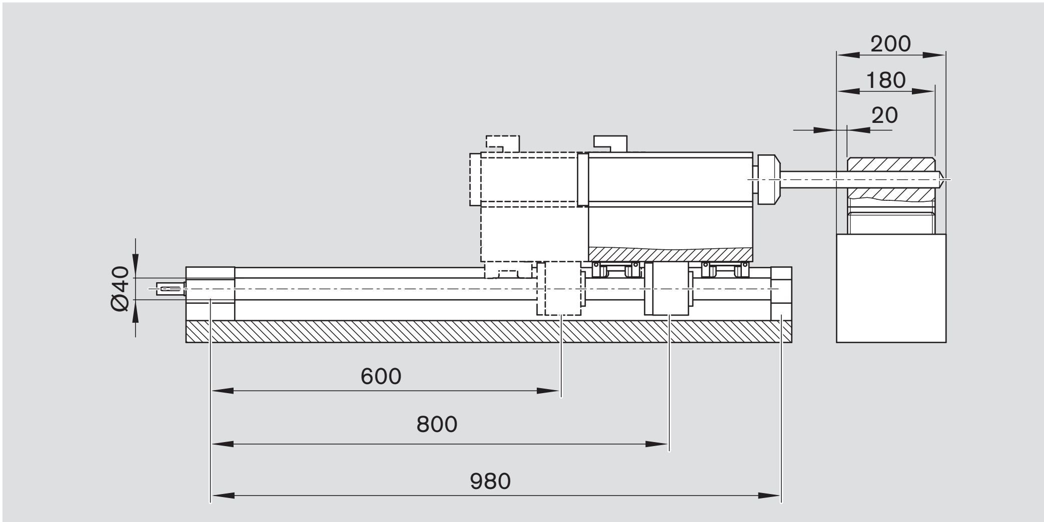
Details of T_{pr0} and T_{RD} are given in the product catalogs. Precise configuration and calculation of the drive unit should ideally be done in collaboration with a motor and controller manufacturer. A check must also be made to determine whether the screw ends are strong enough to transfer the drive torque. Details of the screw ends can also be found in the product catalogs.

5.1 Principles

5.1.3 Calculations

5.1.3.7 Calculation example

Drilling station



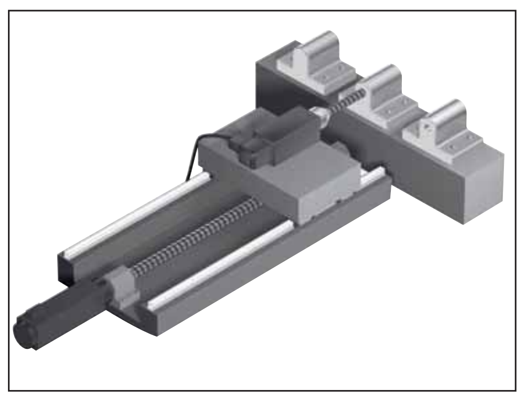
Dimensions for calculating the ball screw drive

A ball screw drive is to be used for the feed motion in the drilling station of a transfer line. Positioning is controlled using a glass measuring scale.

The following example illustrates the complete procedure for calculating the ball screw drive.

Technical data of the application

- Weight of the carriage including attachments $m_1 = 400$ kg
- The carriage is guided by 4 roller runner blocks on 2 guide rails
- Total force required to move the carriage
- $F_R = 150$ N
- The maximum axial load is the process force during drilling $F_p = 4500$ N
- End fixity of the screw: fixed-floating
- Unsupported screw length: 800 mm
- Maximum linear speed $v_{max} = 0.5$ m/s at $n_{max} = 3000$ min^{-1}
- Drive using a servo-motor via a timing belt with a transmission ratio of $i = 0.5$
- The required service life for the complete drilling station is at least 6 years operating 360 days a year in 3 shifts.
  $L_{h\,machine} = 6\,a \cdot 360\,d/a \cdot 24\,h/d$
  $L_{h\,machine} = 51840\,h$



Calculation example for a drilling station

Ball screw assembly

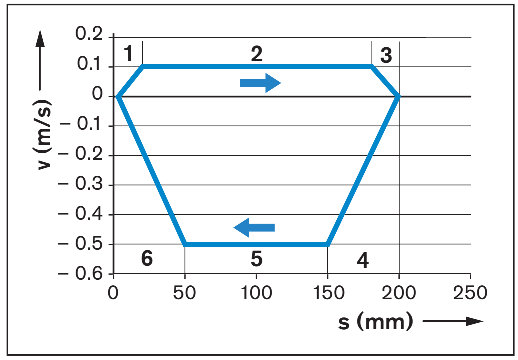
During pre-selection, a ball screw assembly size 40 x 20 (d_0 x P) with a preload of 5% of C was chosen.

5.1 Principles

5.1.3 Calculations

Dynamic cycle

A reference cycle is defined for calculating the ball screw assembly. In this reference cycle, a bore is drilled during phase 2 of the forward stroke. This is followed by a fast return stroke (phases 4 to 6). The complete cycle is executed in 3 seconds. After the reference cycle the workpiece is changed, which also takes 3 seconds. This results in a duty cycle for the ball screw drive of 50% of the machine duty cycle, which must be taken into account in the machine's service life.

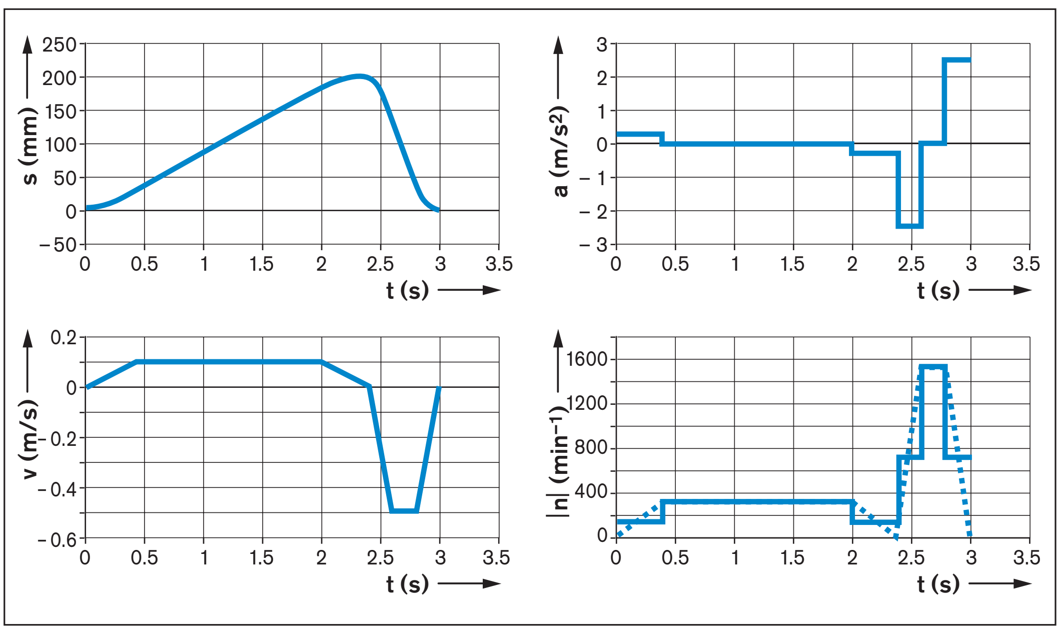


Reference cycle

Phase n	Travel coordinates s_x	Travel s_n	Linear speed v_n	Time t_n	Acceleration a_n	Rotary speed values |n|	Average rotary speed $\|n_n\|$	Description
1	0 mm	20 mm	0 m/s	0.4 s	0.25 m/s²	0 min⁻¹	150 min⁻¹	Acceleration
	20 mm		0.1 m/s			300 min⁻¹		
2	20 mm	160 mm	0.1 m/s	1.6 s	0 m/s²	300 min⁻¹	300 min⁻¹	Constant motion
	180 mm		0.1 m/s			300 min⁻¹		Drilling
3	180 mm	20 mm	0.1 m/s	0.4 s	–0.25 m/s²	300 min⁻¹	150 min⁻¹	Deceleration
	200 mm		0 m/s			0 min⁻¹		
4	200 mm	–50 mm	0 m/s	0.2 s	–2.5 m/s²	0 min⁻¹	750 min⁻¹	Acceleration
	150 mm		–0.5 m/s			1500 min⁻¹		Return stroke
5	150 mm	–100 mm	–0.5 m/s	0.2 s	0 m/s²	1500 min⁻¹	1500 min⁻¹	Constant motion
	50 mm		–0.5 m/s			1500 min⁻¹		Return stroke
6	50 mm	–50 mm	–0.5 m/s	0.2 s	2.5 m/s²	1500 min⁻¹	750 min⁻¹	Deceleration
	0 mm		0 m/s			0 min⁻¹		Return stroke

Only constant operating parameters for each phase are used in the nominal life calculation. Therefore, the average rotary speed n_m must be determined for the phases with acceleration or deceleration.

The curves below show the profiles for all the relevant kinematic parameters (travel s_n, linear speed v_n, acceleration a_n and rotary speed $|n_n|$) over the reference cycle.



Kinematic data for the reference cycle

5.1 Principles

5.1.3 Calculations

Discrete time steps

The discrete time steps are required for determining the average rotary speeds and the loads.

$$t = \sum t_n = t_1 + t_2 + t_3 + t_4 + t_5 + t_6 = 0.4\ s + 1.6\ s + 0.4\ s + 0.2\ s + 0.2\ s + 0.2\ s = 3\ s$$

$$q_{tn} = \frac{t_n}{t} \cdot 100\%$$

$$q_{t1} = \frac{t_1}{t} \cdot 100\% = \frac{0.4\ s}{3\ s} \cdot 100\% = 13.3\%$$

$$q_{t2} = \frac{t_2}{t} \cdot 100\% = \frac{1.6\ s}{3\ s} \cdot 100\% = 53.3\%$$

. . .

Loads

The dynamic data can now be used in the following step to calculate the loads F_n occurring during the individual phases. This is done by adding the individual forces.

- The friction force F_R acts against the direction of travel throughout the entire cycle.
- The acceleration force F_a acts during acceleration and deceleration in phases 1, 3, 4 and 6.
- The process force F_p acts only in phase 2.
- The carriage's weight force F_g has no component acting in the axial direction of ball screw because of the horizontal layout. F_g is taken up completely by the guide units and has no effect on the load on the ball screw drive.

$$F = F_a + F_R + F$$

$$F_1 = F_{a1} + F_{R1} + F_{p1} = 100\ N + 150\ N + 0\ N = 250\ N$$

$$F_2 = F_{a2} + F_{R2} + F_{p2} = 0\ N + 150\ N + 4500\ N = 4650\ N$$

. . .

The intermediate results for the loads F_n and the corresponding discrete time steps q_{tn} are shown in the table below:

Phase n	Travel s_n	Time t_n	Discrete time step q_{tn}	Acceleration a_n	Acceleration force F_a	Friction force F_R	Process force F_p	Load F_n
1	20 mm	0.4 s	13.3%	0.25 m/s²	100 N	150 N	0 N	250 N
2	160 mm	1.6 s	53.3%	0 m/s²	0 N	150 N	4500 N	4650 N
3	20 mm	0.4 s	13.3%	–0.25 m/s²	–100 N	150 N	0 N	50 N
4	–50 mm	0.2 s	6.7%	–2.5 m/s²	–1000 N	–150 N	0 N	–1150 N
5	–100 mm	0.2 s	6.7%	0 m/s²	0 N	–150 N	0 N	–150 N
6	–50 mm	0.2 s	6.7%	2.5 m/s²	1000 N	–150 N	0 N	850 N

5 Ball screw drives

5.1 Principles

5.1.3 Calculations

Average rotary speed

Because the speed and load varies in this application, in the next step the average rotary speed n_m over the entire cycle is calculated according to the formula (5-11).

$$n_m = \frac{|n_1| \cdot q_{t1} + |n_2| \cdot q_{t2} + ... + |n_n| \cdot q_{tn}}{100\%}$$

$$n_m = \frac{150\,\text{min}^{-1} \cdot 13.3\% + 300\,\text{min}^{-1} \cdot 53.3\% + 150\,\text{min}^{-1} \cdot 13.3\% + 750\,\text{min}^{-1} \cdot 6.7\% + 1500\,\text{min}^{-1} \cdot 6.7\% + 750\,\text{min}^{-1} \cdot 6.7\%}{100\%}$$

$$n_m = 400.80\ \text{min}^{-1}$$

Taking preload into account

For the load values F_n acting on the ball screw in the phases n, it must be established whether the system's preload will have an effect on the life expectancy. To do this, the preload force must first be determined.

The following ball screw has been selected:

- Size 40 x 20
- Dynamic load rating C = 37,900 N
- Preload 5% of C ($X_{pr} = 0.05$)

Preload force F_{pr} according to formula (5-12): $F_{pr} = X_{pr} \cdot C = 0.05 \cdot 37900\ \text{N} = 1895\ \text{N}$

To calculate the nominal life as accurately as possible, it must now be ascertained whether this preload force must be taken into account in the calculation.

Lift-off force F_{lim} according to formula (5-13): $F_{lim} = 2.8 \cdot F_{pr} = 2.8 \cdot 1895\ \text{N} = 5306\ \text{N}$

In all phases F_n is $< F_{lim}$.
The preload must therefore be taken into account in the nominal life calculation.

Effective axial load on the ball screw assembly according to formula (5-15):

$$F_{eff\ n} = \left(\frac{|F_n|}{2.8 \cdot F_{pr}} + 1\right)^{\frac{3}{2}} \cdot F_{pr}$$

$$F_{eff1} = \left(\frac{|F_1|}{2.8 \cdot F_{pr}} + 1\right)^{\frac{3}{2}} \cdot F_{pr} = \left(\frac{250\ \text{N}}{2.8 \cdot 1895\ \text{N}} + 1\right)^{\frac{3}{2}} \cdot 1895\ \text{N} = 2030\ \text{N}$$

$$F_{eff\ 2} = \left(\frac{|F_2|}{2.8 \cdot F_{pr}} + 1\right)^{\frac{3}{2}} \cdot F_{pr} = \left(\frac{4650\ \text{N}}{2.8 \cdot 1895\ \text{N}} + 1\right)^{\frac{3}{2}} \cdot 1895\ \text{N} = 4871\ \text{N}$$

. . .

Intermediate results: effective axial load

Phase n	Load value $\|F_n\|$	Effective load $F_{eff\ n}$
1	250 N	2030 N
2	4650 N	4871 N
3	50 N	1922 N
4	1150 N	2543 N
5	150 N	1976 N
6	850 N	2368 N

5.1 Principles

5.1.3 Calculations

Equivalent dynamic axial load

After calculating the loads in the individual phases, all the required data are now available for determining the equivalent dynamic axial load F_m.

F_m according to formula (5-17):

$$F_m = \sqrt[3]{\left(F_{eff\,1}\right)^3 \cdot \frac{|n_1|}{n_m} \cdot \frac{q_{t1}}{100\%} + \left(F_{eff\,2}\right)^3 \cdot \frac{|n_2|}{n_m} \cdot \frac{q_{t2}}{100\%} + \ldots + \left(F_{eff\,n}\right)^3 \cdot \frac{|n_n|}{n_m} \cdot \frac{q_{tn}}{100\%}}$$

$$F_m = \sqrt[3]{(2030\ \text{N})^3 \cdot \frac{150\ \text{min}^{-1}}{400.80\ \text{min}^{-1}} \cdot \frac{13.3\%}{100\%} + (4871\ \text{N})^3 \cdot \frac{300\ \text{min}^{-1}}{400.80\ \text{min}^{-1}} \cdot \frac{53.3\%}{100\%} + \ldots + (2368\ \text{N})^3 \cdot \frac{750\ \text{min}^{-1}}{400.80\ \text{min}^{-1}} \cdot \frac{6.7\%}{100\%}}$$

$$F_m = 3745\ \text{N}$$

Nominal life in revolutions

The equivalent dynamic load F_m can be used to calculate the nominal life in revolutions L according to formula (5-18):

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

$$L = \left(\frac{37900\ \text{N}}{3745\ \text{N}}\right)^3 \cdot 10^6 = 1036.366 \cdot 10^6$$

Nominal life in hours

The nominal life in hours is then calculated using formula (5-19):

$$L_h = \frac{L}{n_m \cdot 60\ \frac{\text{min}}{\text{h}}}$$

$$L_h = \frac{1036.366 \cdot 10^6}{400.80\ \frac{1}{\text{min}} \cdot 60\ \frac{\text{min}}{\text{h}}} = 43096\ \text{h}$$

Machine operating hours

The ball screw drive's duty cycle is 50% of the machine duty cycle. According to formula (5-20), the maximum service life of the machine will therefore be:

$$L_{h\ machine} = L_h \cdot \frac{DC_{machine}}{DC_{BS}}$$

$$L_{h\ machine} = 43069\ \text{h} \cdot \frac{100\%}{50\%} = 86191\ \text{h}$$

This value is far above the required 51,840 h. However, since the calculations were performed for a reference cycle only and the operating parameters may change over the years, the selected ball screw size and version are retained.

5 Ball screw drives

5.1 Principles

5.1.3 Calculations

Critical speed

The nominal life calculation is now followed by the necessary additional calculations and checks. The first step is to check the critical speed n_k. In this application, the screw is operated with a fixed-floating bearing configuration.

Parameter		Value
Fixed-floating bearing coefficient	f_{nk}	18.9
Core diameter of screw	d_2	33.8 mm
Critical screw length	l_n	800 mm
Maximum operating speed of screw	n_{max}	1500 min^{-1}

Critical speed n_k according to formula (5-21):

$$n_k = f_{nk} \cdot \frac{d_2}{l_n^2} \cdot 10^7 \qquad (min^{-1})$$

$$n_k = 18.9 \cdot \frac{33.8}{800^2} \cdot 10^7 \qquad (min^{-1})$$

$$n_k = 9982\ min^{-1}$$

According to formula (5-22), the permissible maximum operating speed is:

$$n_{k\,perm} = n_k \cdot 0.8 = 9982\ min^{-1} \cdot 0.8 = 7986\ min^{-1}$$

Therefore, the critical speed will not be a restricting factor in this specific application.

Permissible axial screw load

The permissible axial screw load is calculated to check the screw's buckling safety factor. This calculation is based on the maximum load on the ball screw, taking the preload into account as well. The maximum load occurs in phase 2. For the screw's effective buckling length, the unsupported screw length of 800 mm is used as an approximation. The excess travel of 20 mm, during which there is no axial load, is therefore disregarded.

Parameter		Value
Maximum effective load	$F_{eff\,2}$	4871 N
Fixed-floating bearing coefficient	f_{Fk}	20.4
Effective buckling length of screw	l_k	800 mm

Theoretical buckling load of the screw shaft F_k according to formula (5-23):

$$F_k = f_{Fk} \cdot \frac{d_2^4}{l_k^2} \cdot 10^4 \qquad (N)$$

$$F_k = 20.4 \cdot \frac{33.8^4}{800^2} \cdot 10^4 \qquad (N)$$

$$F_k = 416\,023\ N$$

The permissible axial load should be calculated with a safety factor of at least 2. Since this calculation example concerns a reference cycle that takes no disruptions into account (e.g. tool wear, tool breakage or collision in fast approach mode), a safety factor of 6 is chosen here.

$$F_{k\,perm} = \frac{416023\ N}{6} = 69\,337\ N$$

The two checks performed therefore show that neither the critical speed nor buckling represent a problem for the chosen ball screw assembly. The ball screw assembly can therefore be safely used in this application.

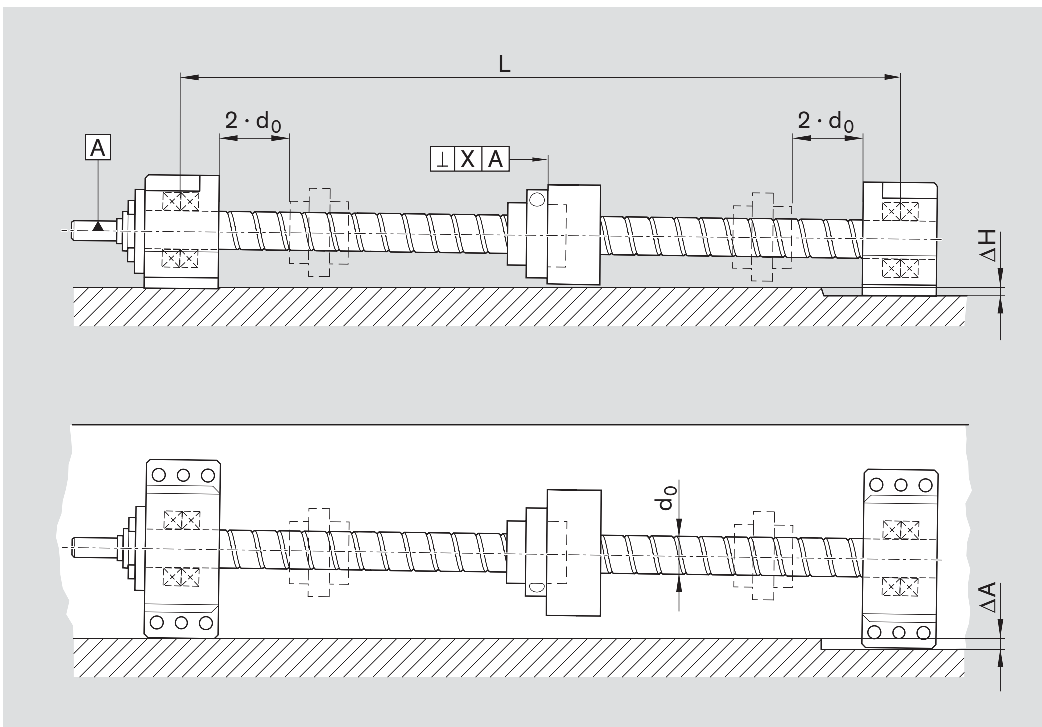
5.1 Principles

5.1.4 Design notes

To ensure that a ball screw drive can actually achieve the calculated service life and performance, its system-related requirements and limitations must be taken into account at the design stage. Screw drives are not suitable for transferring radial forces and torques that may be caused by misalignments during installation. The following sections illustrate the most important principles for achieving designs that will be compatible with the ball screw system and its requirements.

5.1.4.1 Adjoining structures and installation tolerances

When using ball screw drives, the specified installation tolerances must be observed when designing and building the adjoining structures. The first basic principle is: The higher the ball screw drive's precision and preload, the more accurate the adjoining structures must be. This applies in particular to applications in which the nut travels right up to the end bearings since, in this area, the risk of distortive stresses and therefore of additional loads is very high.



Height offset, lateral offset and details of the perpendicularity between the screw shaft axis and the location face of the nut housing

L	=	bearing-to-bearing distance	(mm)
d_0	=	nominal diameter of screw	(mm)
X	=	permissible deviation from perpendicularity: The tolerance applies to a surface that must lie between two planes spaced at a distance X from each other, which are perpendicular to the reference axis A.	(mm)
ΔH	=	permissible height offset	(mm)
ΔA	=	permissible lateral offset	(mm)

5.1 Principles

5.1.4 Design notes

Installation tolerances

The tables at right show the most important recommended installation tolerances for ball screw assemblies according to the respective preload. These tolerances include the perpendicularity of the nut housing (or adjoining structure) relative to the screw axis. The given tolerances for the height offset ΔH and lateral offset ΔA of the end bearings must also be observed.

Through appropriate design measures and mounting procedures, it is possible to avoid the need to fabricate highly accurate and therefore cost-intensive adjoining structures. Design engineers should always check whether reference edges, locating pin holes and the centering diameter on the nut are really required to assure the functionality of the system in service. Appropriate procedures are described in section 5.1.5 "Mounting instructions."

Installation tolerances for $L < 1000$ mm, minimum distance between the nut and the end bearings $< 2 \cdot d_0$:

Preload	X mm	ΔH mm	ΔA mm
Backlash	0.05	0.05	0.05
2% of C	0.04	0.04	0.04
5% of C	0.03	0.03	0.03
7% of C	0.01	0.01	0.01
10% of C	0.01	0.01	0.01

Installation tolerances for $L > 1000$ mm, minimum distance between the nut and the end bearings $> 2 \cdot d_0$:

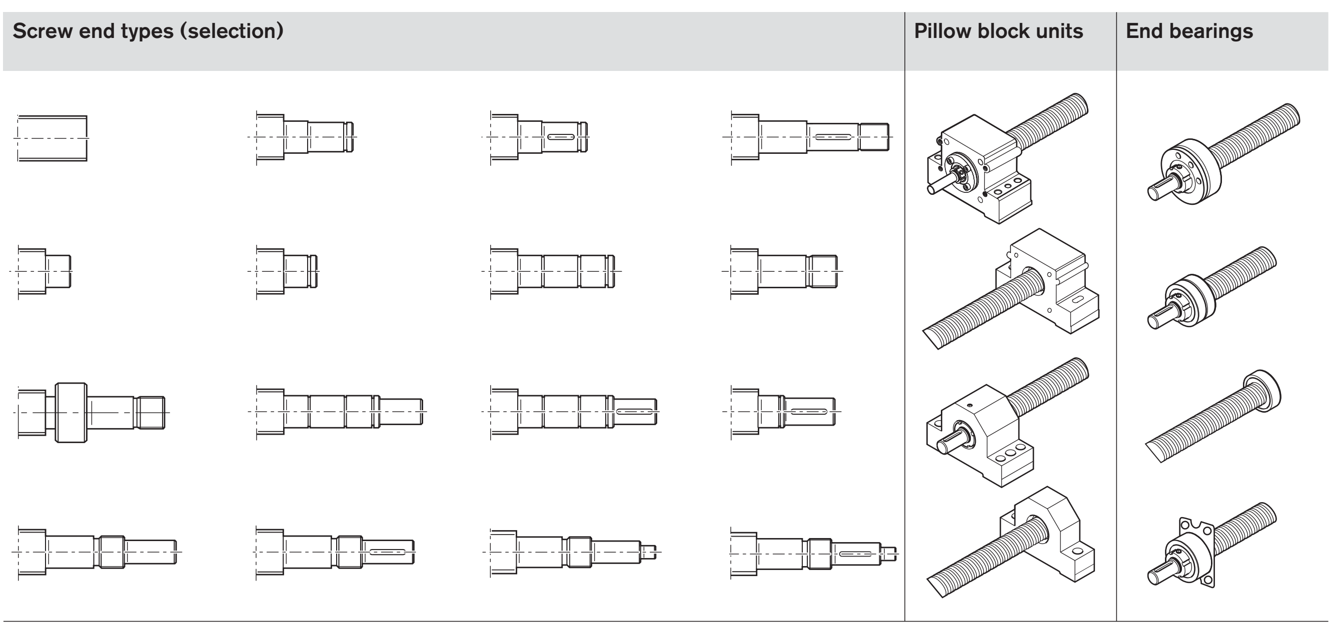
Preload	X mm	ΔH mm	ΔA mm
Backlash	0.10	0.10	0.10
2% of C	0.08	0.08	0.08
5% of C	0.05	0.05	0.05
7% of C	0.02	0.02	0.02
10% of C	0.02	0.02	0.02

5.1.4.2 Guidelines for economical constructions

Use of standard elements

Dimensioning the screw ends and selecting suitable bearings is time-consuming and costly. To simplify the design process for customers and reduce costs, Rexroth offers standardized solutions for end bearings and screw end machining as appropriate for the individual sizes and application areas.

The following illustration shows a selection of the available screw end types and bearing options. Further advantages of using well-proven standard elements are fast delivery and simpler logistics. Customers can order perfectly matched components from a single source.



5.1 Principles

5.1.4 Design notes

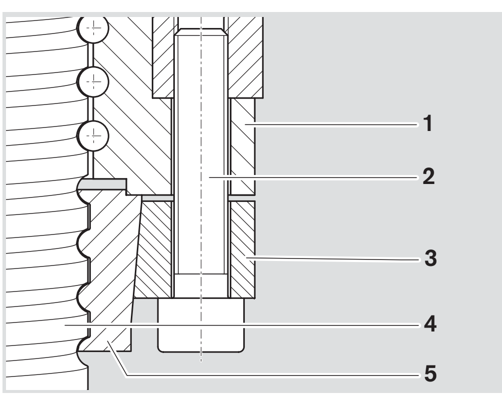
As a rule, the more precise a ball screw assembly is, the more expensive it will be. Cost-conscious design engineers will therefore only use as precise a ball screw drive as is necessary.

The following advice is provided to assist engineers during the design and selection process:

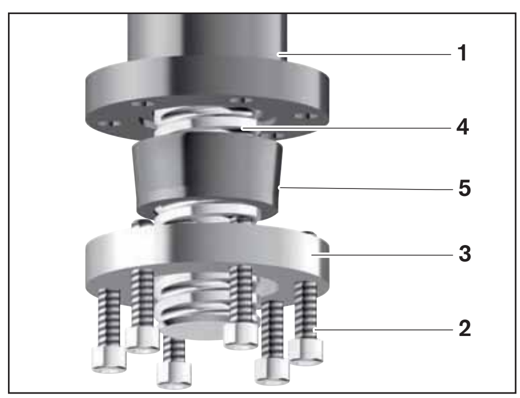
- It is not necessary to use a precision screw if a direct position measuring system is available on the travel axis. The required positioning accuracy can be achieved using the position measuring system and the electronic controls.
- High accuracy can also be achieved by one-time tuning even without a direct measurement system, if the lead deviations are entered in the control system (see section 5.1.1.5).
- Only use a preloaded system when necessary. The higher the preload, the higher the requirements will be for the accuracy of the adjoining structures.
- Check whether cylindrical nuts can be used. Cylindrical nuts make it possible to build more compact structures. No fixing holes have to be drilled for this type of nut.
- Use screw-in nuts whenever possible.
- Only center the nut in the housing if necessary. Centering increases the processing costs and makes higher demands on manufacturing tolerances.
- Whenever possible, use standardized nuts. Common ball nut sizes and types, as specified in the DIN 69051 standard, are usually more affordable. Rexroth identifies the nut types belonging to this category in its product catalogs. Using such nuts also offers the advantage of ready availability if a replacement is required.
- Dimension the ball screw drive correctly. The more well-founded the design data is, the more accurate the calculations will be, which helps to avoid expensive oversizing of the assembly.

5.1.4.3 Safety nuts for vertical applications

In vertical applications, it should be remembered that a failure of the ball nut could result in uncontrolled dropping of the entire carriage. A safety nut can be used to protect against such crashes. Safety nuts are profiled with a negative profile to that of the screw. In normal service, there is a clearance between the two profiles. In the event of failure of the ball nut, the profiles on the safety nut and the screw come into contact, acting as a jam to prevent the ball nut and the carriage assembly from falling.
The safety nut is fastened to the ball nut by means of a clamping ring. For vertical applications it must always be fitted below the ball nut, as this is the only position that can ensure the safety function.



Structural design of a safety nut



Mounting configuration with safety nut

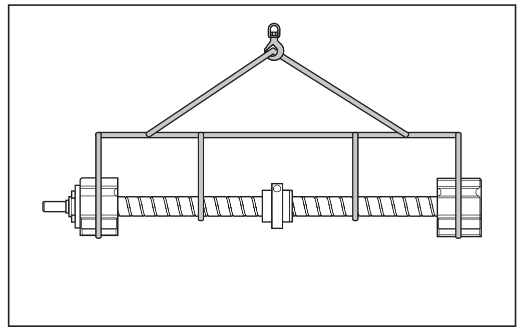
1 Ball nut
2 Socket head cap screws
3 Clamping ring
4 Screw
5 Safety nut

5.1 Principles

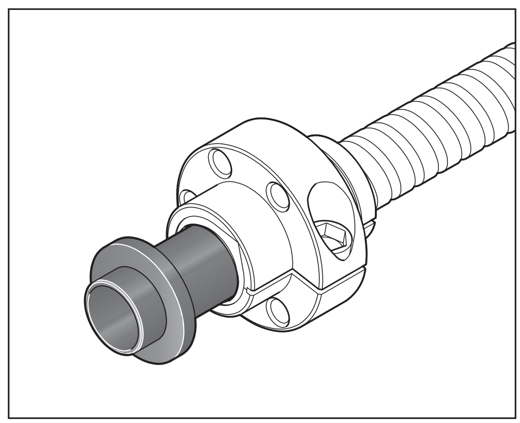
5.1.5 Mounting instructions

Ball screw drives are high-value machine parts that have a significant influence on the precision and service life of the entire machine. For this reason, the following ground rules must be followed during installation.

- Appropriate lifting equipment must be used for long screws. The screw should be supported at several points along its length to avoid excessive deflection (risk of permanent deformation).
- All mounting surfaces on the adjoining structure must be clean and burr-free.
- For all screw connections, the permissible tightening torque must not be exceeded.
- The ball screw drive must be aligned parallel to the installed guide units.
- After installation, check whether the torque is constant over the entire stroke. If it is, the screw has been properly aligned. If the torque varies, the system can be optimized by re-aligning the end bearings.
- If it is necessary to remove the ball nut from the screw, a mounting arbor must be used to avoid losing balls. Disassembly should only be carried out by properly qualified and authorized personnel.
- Double nuts are systems that have been precisely matched to the screw and should not be removed from it, as the set preload would then be lost.
- Ball nuts that have been installed without initial lubrication must be thoroughly lubricated before the unit is started up for the first time.
- If any work such as drilling, welding or painting has to be performed near the ball screw drive after installation, the screw drive must be protected with a cover before starting the work.
- The relevant mounting instructions contain additional important advice and recommendations.



When lifting, support the assembly at a sufficient number of points.



Mounting arbor

5.2 Ball nuts

5.2.1 Single nuts

5.2.1.1 System characteristics

Most common ball nuts

High performance
Compact design

Single nuts are the most commonly used ball nuts. They are available in all the usual sizes, preloads and degrees of accuracy. They provide high performance in a compact unit. The systems with two (and sometimes even four) ball tracks achieve very high load ratings, thus offering long service life for customer applications.

Ball nut series

The different series of single nuts cover a wide variety of applications:

- Standard series
- Miniature series
- eLINE series
- ECOplus series with recirculation caps
- Speed series with recirculation caps
- Machine Tool series

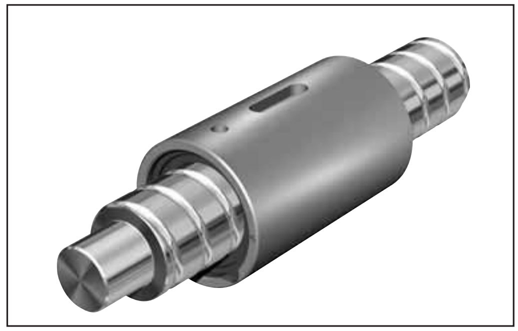
The single nuts in the Standard series essentially have the structural design described in section 5.1.1.1.

Unlike the Standard nuts, ECOplus ball nuts have no recirculation pieces but instead full plastic recirculation caps. This configuration makes them very economical.

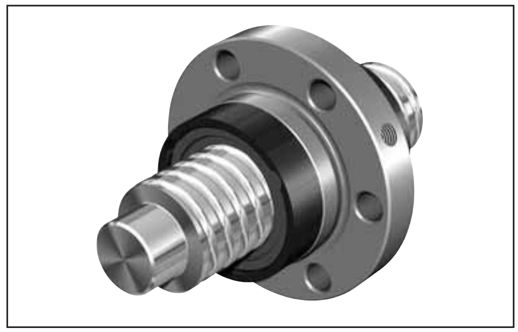
Single nuts in the Speed series are characterized by their high maximum permissible linear speeds. This is due to the fact that their leads are equal to or greater than the nominal diameter.

Single nuts in the Machine Tool series are preloaded. The preload is achieved by shifting.

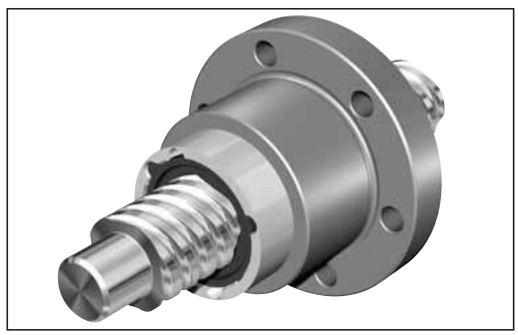
Examples of single nuts from the Rexroth range are shown in the illustrations at right.



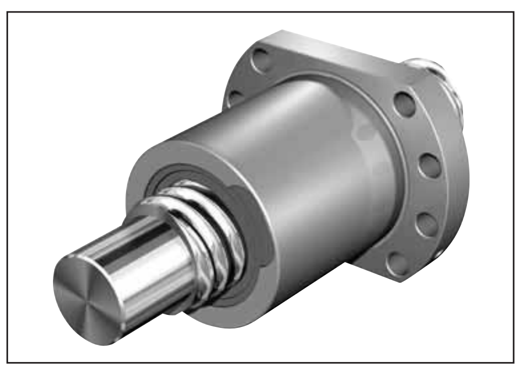
Cylindrical single nut from the Standard series



Single nut from the ECOplus series with plastic recirculation caps



Single nut from the Speed series with plastic recirculation caps



Single nut from the Machine Tool series

5.2.1.2 Application areas

The range of applications for single nuts is as broadly diversified as the range of versions available. Single nuts are used in practically every machinery construction sector.

5.2 Ball nuts

5.2.2 Standard series single nuts

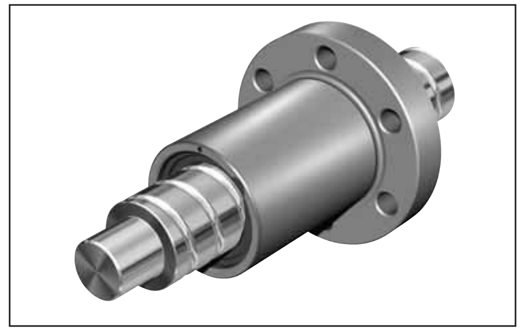
5.2.2.1 System characteristics

Most common series

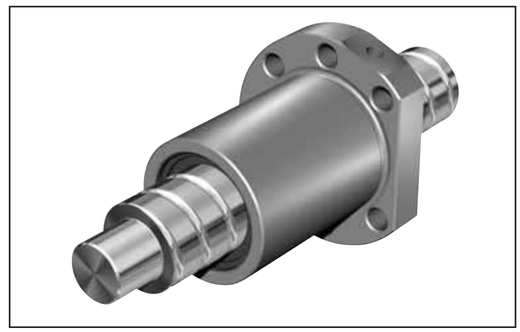
Standard single nuts are the most common series with the greatest variety of ball nut types. The Standard series comprises flanged nuts and cylindrical nuts. Two of the flanged nuts are also available in adjustable preload versions. Adjustable preload nuts have a narrow, adjustable slot running the length of the nut body. This design allows both zero backlash and adjustment of the preload to user requirements.

Standardized design

Single nuts in the Standard series are available with mounting dimensions per DIN 69051 part 5 or with Rexroth mounting dimensions. Matching nut housings and end bearings are also available in several versions for all Standard single nuts.



Standard series flanged nut

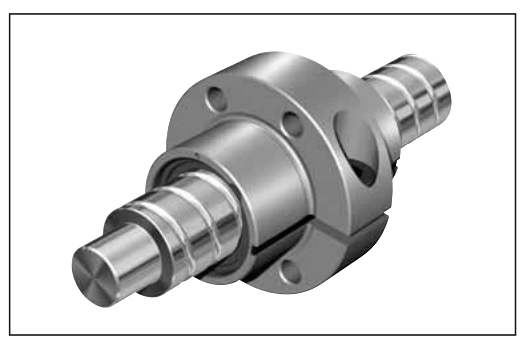


Standard series flanged nut with single flat

5.2.2.2 Application areas

Because of the wide variety of nut types, Standard series single nuts have a very broad range of applications. They can be used in virtually every area of the machinery construction sector.

Standard series cylindrical nut



Standard series adjustable preload nut

5.2 Ball nuts

5.2.3 Miniature series single nuts

5.2.3.1 System characteristics

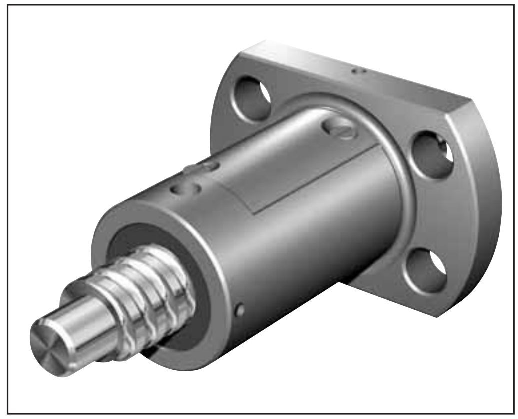
Nominal diameter less than 12 mm

Miniature ball screw assemblies are conventionally understood to be systems with a nominal diameter of less than 12 mm. Miniaturized nut geometries are achieved through the use of optimized recirculation systems and very small balls.

Low preload

These ball screws are usually not preloaded or only very slightly preloaded to ensure the smoothest possible travel.

The illustration at right shows a typical nut from the miniature series.



Miniature series flanged nut

5.2.3.2 Application areas

Because of their very compact design, miniature ball screw drives are used in all kinds of technical applications where the available space is limited.

Typical application areas are:

- Semi-conductor production and processing
- Medical technology (diagnosis, dispensing and rehabilitation)
- Automation (jigs and fixtures, handling systems, grippers and robots)
- Electrical engineering (switches)
- Process technology (valve and flap actuation)
- Manufacturing technology (small machines for miniature metal-cutting processes)

5.2 Ball nuts

5.2.4 eLINE series single nuts

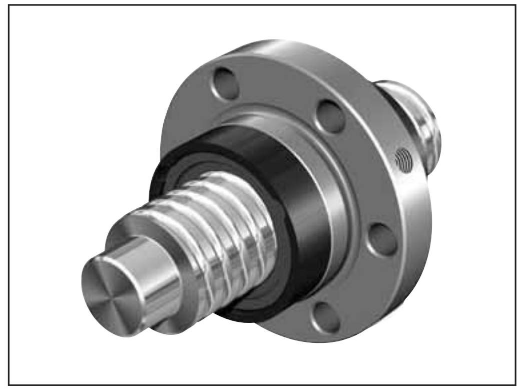
5.2.4.1 System characteristics

Economical

eLINE ball screw drives are economical assemblies for applications that do not make very high demands on accuracy, speed and rigidity. The use of alternative, rationalized manufacturing processes and a small range of standardized components makes it possible to produce very affordable drive solutions. eLINE ball nuts are supplied without preload on rolled screws with a lower level of accuracy.

Without preload

The illustrations at right show two typical nuts from the eLINE series.



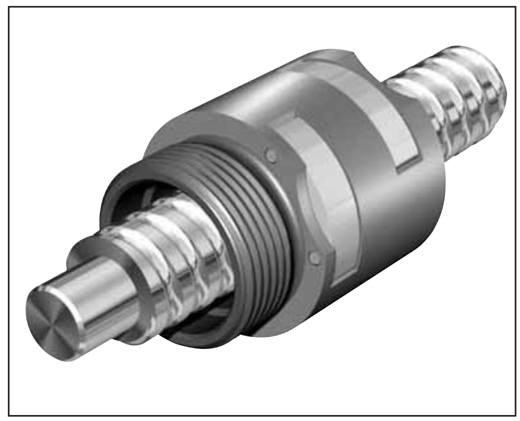
eLINE series flanged nut with recirculation caps

5.2.4.2 Application areas

These low-cost ball screw drives can be utilized in every technical field of application. Despite certain limitations in terms of performance (e.g. no preloading possible), they are vastly superior to acme screws.

Typical application areas are:

- Factory automation (workpiece transport and jigs and fixtures)
- Adjustable axes (woodworking and bending presses)
- Linear actuators
- Ergonomics (table height adjustment)
- Transportation technology (door operation and lifting equipment)



eLINE series screw-in nut

5.2 Ball nuts

5.2.5 Double nuts

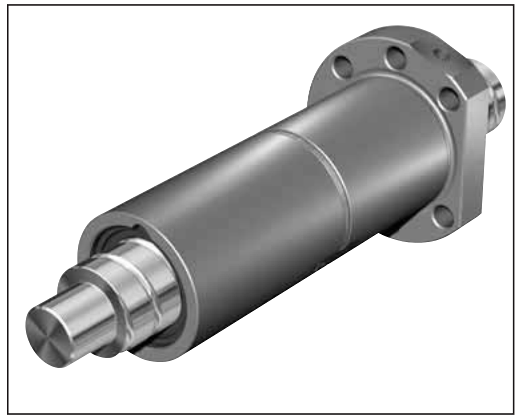
5.2.5.1 System characteristics

Preload
Two-point contact

Double nuts are classic examples of preloaded ball nuts with two-point contact. The preload is generated by tensioning the two halves of the double nut against each other on the screw (see section 5.1.1.3). Two-point contact ensures very smooth operation even with very high preloads.

Modular construction

Double nuts are relatively expensive because of their two-piece design (two complete nuts). The modular construction also results in very long component lengths, which must be taken into account when designing the adjoining structure.



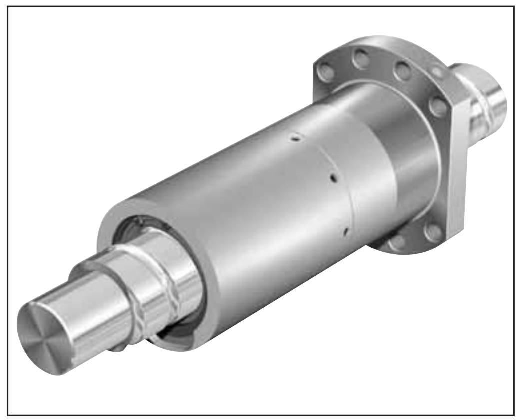
Standard series double nut

Double nut series

Rexroth offers double nuts in two series:
- Standard series
- Machine Tool series

The double nuts are preloaded and run on special screws. The Machine Tool series has specifically optimized load ratings and travel speeds.

The illustrations at right show two typical double nuts.



Machine Tool series double nut

5.2.5.2 Application areas

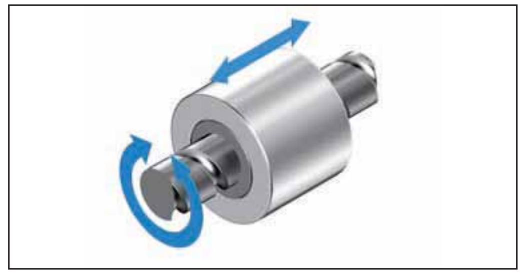
Double nuts are mainly used in machine tools. The nuts are generally installed in axes with high accuracy and rigidity requirements. These may be, for instance, the main axes in grinding machines and machining centers.

5.3 Drive units

5.3.1 Drive units with driven screw

5.3.1.1 System characteristics

These readily available drive units offer users the opportunity to rapidly integrate economical drive solutions with minimal design and manufacturing effort. They combine all the performance characteristics of a classic ball screw drive in one unit. When paired with Rexroth linear guides, they offer machine designers full design freedom for every application.

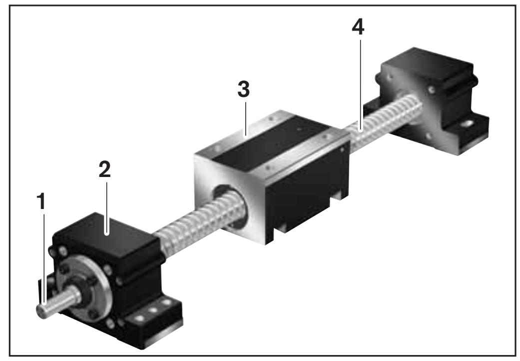


Operating principle of the driven screw

Rexroth offers drive units with driven screw in two forms:
- Open drive unit AOK
- Drive unit AGK with enclosure and sealing strip

Open drive units (AOK)

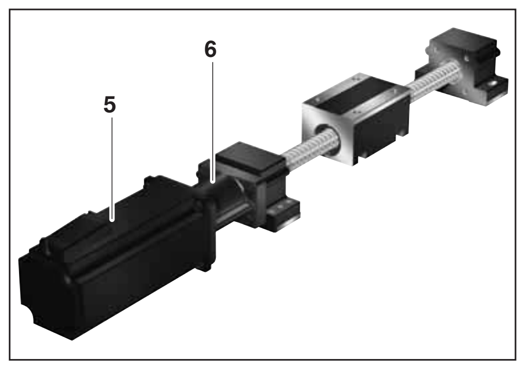
The AOK drive unit is the classic ball screw assembly with pillow blocks and pre-assembled ball nut enclosure. A motor and gear unit can be supplied along with the unit, if so requested by the customer.



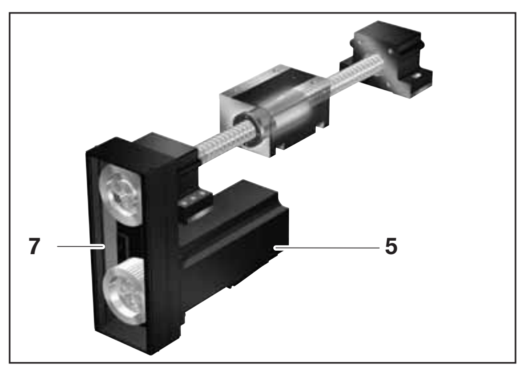
AOK drive unit

The drive units comprise a precision screw and a cylindrical single nut (with zero backlash or preloaded). The aluminum ball nut enclosure is finished on all sides and has reference edges on both sides. The pillow block units are made of robust extruded aluminum profile with reference edges on both sides and mounting holes as well as a locating feature for motor mounting.

The illustrations at right show AOK drive units with various motor attachment options.



AOK drive unit with motor mount, coupling and motor



AOK drive unit with side drive timing belt and motor

1 Screw journal
2 Pillow block unit
3 Nut enclosure with ball nut
4 Screw
5 Motor
6 Motor mount, coupling
7 Side drive timing belt

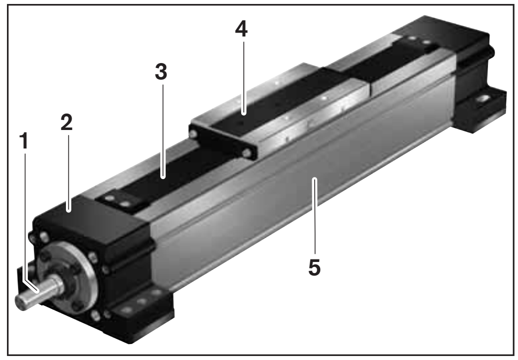
5.3 Drive units

5.3.1 Drive units with driven screw

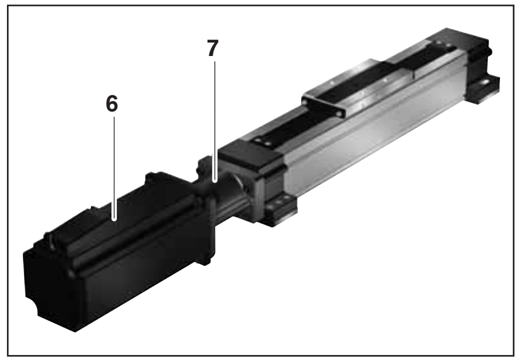
Closed drive unit (AGK)

The closed drive unit with ball screw assembly has the same basic structure as the AOK drive unit, but is additionally provided with an enclosure and sealing strip. This eliminates the need to design and install protective structures. The unit comes complete with aluminum extrusion profile encapsulation and a steel or polyurethane sealing strip.

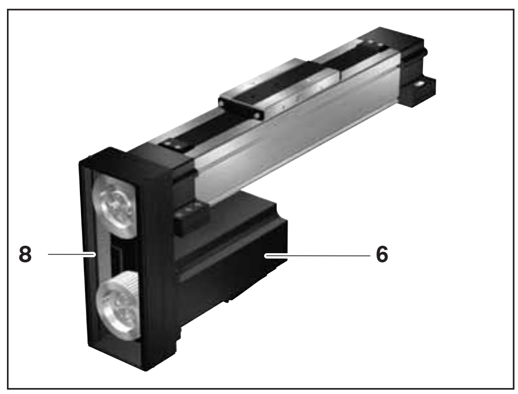
The illustrations at right show AGK drive units with various motor attachment options.



AGK drive unit



AGK drive unit with motor mount, coupling and motor

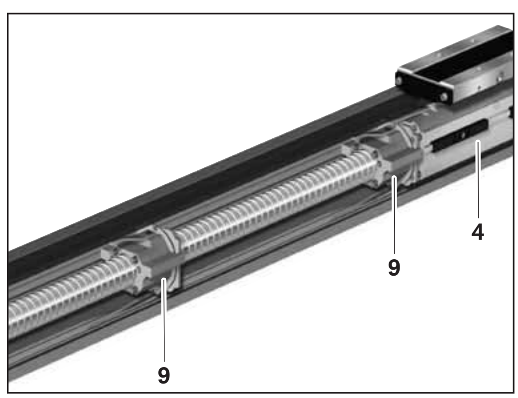


AGK drive unit with side drive timing belt and motor

1 Screw journal
2 Drive side pillow block
3 Sealing strip
4 Carriage with ball nut
5 Enclosure
6 Motor
7 Motor mount and coupling
8 Side drive timing belt
9 Screw support

Screw support (SS)

The AGK drive unit is available with optional screw supports (SS). These traveling screw supports are located on either side of the ball nut and support the screw radially against the enclosure. This allows the screw to rotate at high speed even in applications with long strokes. The number of screw supports is freely selectable and depends on the maximum permissible linear speed and the critical screw speed for the specific application.



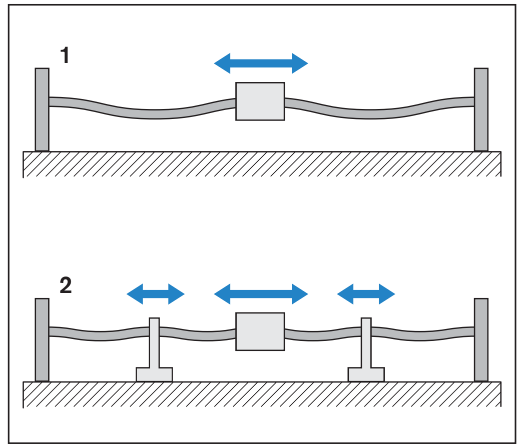
AGK drive unit with screw supports

5.3 Drive units

5.3.1 Drive units with driven screw

Operating principle of the screw support

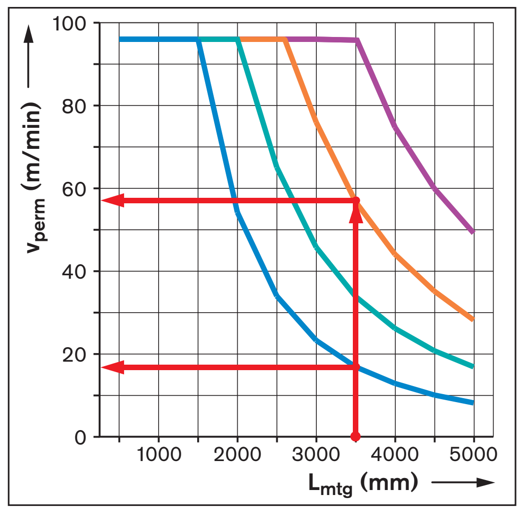
The traveling screw supports act as floating bearings and reduce the free screw length between the nut and the end bearings. This increases the critical screw speed while reducing both screw deflection and the resonant energy introduced into the system due to screw whip.



1 Ball screw assembly without screw supports
2 Ball screw assembly with one traveling screw support on each side of the nut

Operating principle of the screw support

The effect of the screw supports on the maximum permissible linear speed is clear from the following example (see chart at right):
A drive unit with a size 32 ball screw assembly with a lead of 32 mm and a screw length of 3500 mm can operate at a maximum linear speed of 17 m/min without screw supports. With 2 screw supports on each side of the ball nut, a maximum linear speed of 57 m/min is possible without reaching the critical screw speed range.



v_{perm}	=	maximum permissible linear speed	(m/min)
L_{mtg}	=	mounting length (screw length)	(mm)

Comparison of the maximum permissible linear speeds as a function of the number of screw supports, taking an AGK 32 drive unit with a 32x32 ball screw as an example

- Without screw support
- With 1 screw support (on each side)
- With 2 screw supports (on each side)
- With 3 screw supports (on each side)

5.3.1.2 Application areas

Drive units with driven screws and screw supports are suitable for applications with very long strokes. The encapsulated design makes this solution ideal for woodworking environments and for use in water jet cutting applications.

5.3 Drive units

5.3.2 Drive units with driven nut

5.3.2.1. System characteristics

The requirement for higher dynamics as well as competition from linear motors (see Chapter 6, section 6.8.2.2) have led to the development of systems where the nut is driven instead of the screw.


Operating principle of a driven nut

Advantages

This concept provides marked advantages over the driven screw design:

- Since the screw does not rotate, the attainable rotary speed is no longer limited by the critical screw speed. However, even with stationary screws, the natural frequency must still be taken into account. On the other hand, resonance (that is, when the rotational frequency of the nut is the same as the first order frequency) is far less critical than in systems with driven screws because of the significantly lower energy introduced by the driven nut. This is due to the optimized radial and axial run-out of Rexroth's driven nuts. The systems can theoretically be operated up to the maximum rotary speed as determined by the characteristic speed (see section 5.1.1.7). Nevertheless, the resonance range must be traversed as rapidly as possible to avoid unnecessary stresses on the system.
- The screw does not have to be set into rotation. This reduces the system's overall mass moment of inertia.
- The end fixity for the non-rotating screw can be of a simpler and therefore more economical design.
- Since the screw is stationary, it can be stretched (tensioned) with relatively little effort. This makes it possible to compensate for length variations due to temperature fluctuations.
- Thermal influences can also be compensated for by using a hollow screw with a cooling system.

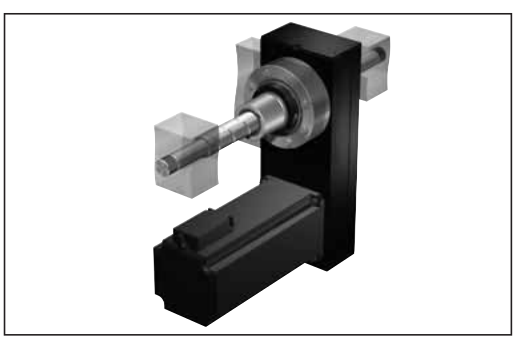
The disadvantage of such a system is that the motor is moved along with the carriage and therefore provision must be made for the corresponding amount of space and for cable management.

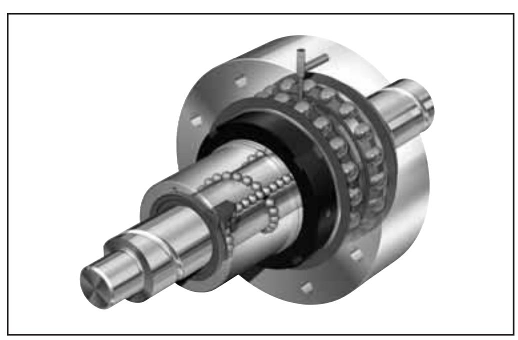
Rexroth offers drive units with driven nut in two forms:

- Drive unit with FAR driven nut with side drive timing belt and motor
- MHS drive unit with directly driven nut and hollow shaft motor

FAR drive unit with belt-driven nut

In drive units with FAR belt-driven nuts, the ball nut is driven by the motor via a toothed belt. FAR drive units are available as complete functional units consisting of the ball nut assembly, a side drive timing belt and an AC servo motor.


FAR drive unit with side drive timing belt


Ball nut of a FAR drive unit

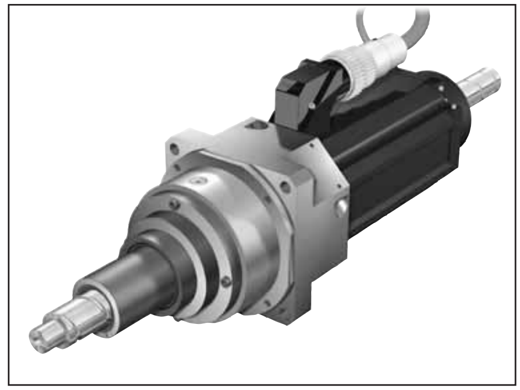
5.3 Drive units

5.3.2 Drive units with driven nut

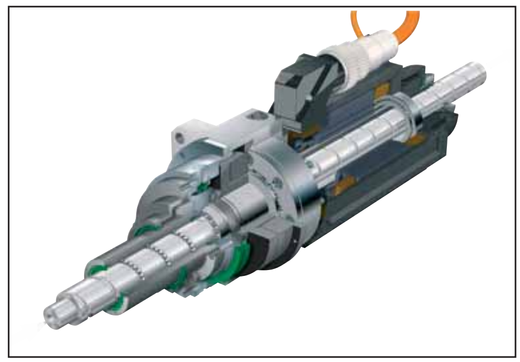
MHS drive unit with hollow shaft motor

In MHS drive units, the nut is driven directly by a hollow shaft motor. The screw of the ball screw assembly is passed through the hollow rotor shaft of the servo motor.

Arranging the servo motor and the ball screw on one axis helps to save space. The nut is connected to the hollow shaft motor without any additional transmission elements. Transmission elements such as drive belts or couplings are eliminated together with their normally negative effects on precision.



Functional unit consisting of a ball nut and hollow shaft motor mounted on the screw

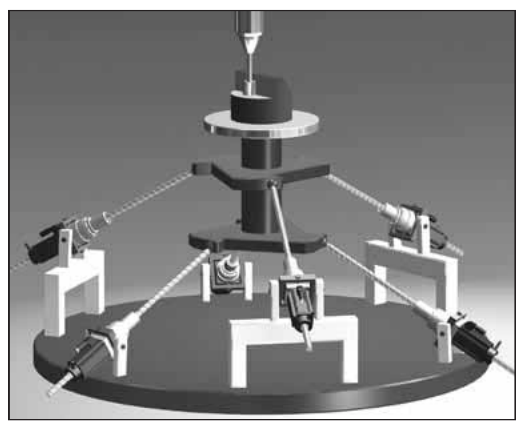


Structural design of the MHS drive unit

5.3.2.2 Application areas

The advantages of driven nuts can be seen most clearly in applications with long strokes, e.g. grinding machine tables.

Drive units with directly driven nut and MHS hollow shaft motor are the ideal solution for highly dynamic applications. The dynamic potential of the ball screw drive can be utilized to the fullest extent. Systems with driven nuts have been successfully incorporated in hexapods, for example. In these systems, the directly driven nuts are axially fixed and the screws perform the linear movements.



Model of a hexapod with six MHS drive units