

# **Servomech**®

*new ideas in linear motion*



## ***Ball screws and nuts***

CATALOGUE ED. 01/10

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## 1. DESIGN

### 1.1 Introduction

A ball screw is a mechanical system capable of converting rotary motion to linear motion or vice versa. An example of such a system, shown in Figure 1, is composed of a ball screw threaded shaft, a ball nut body, balls, ball recirculation elements and wipers (when present).

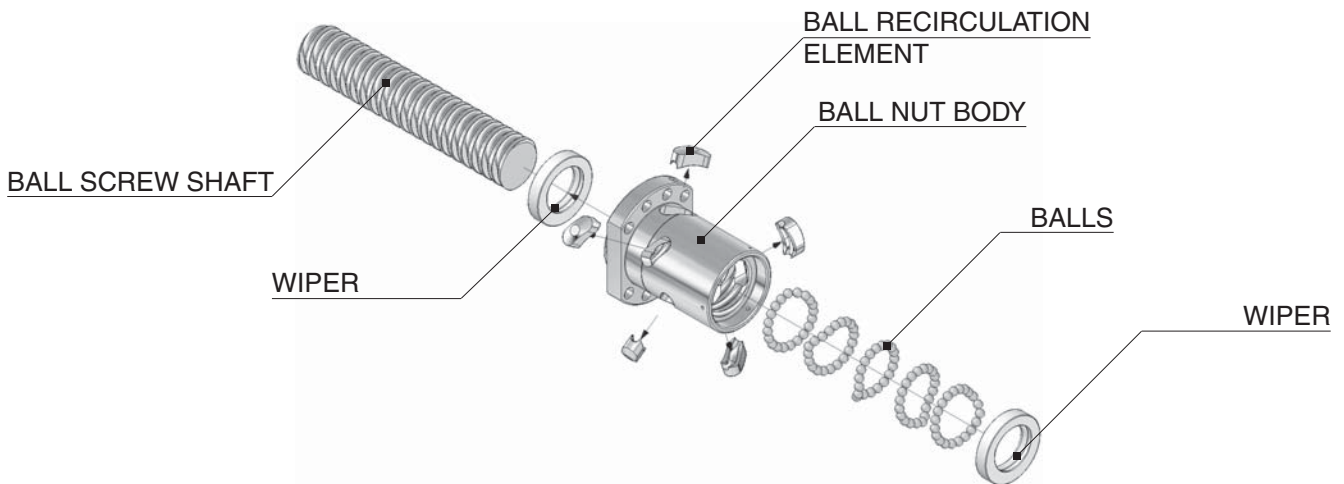


Fig. 1 - Ball screw assembly

One of the main ball screw features is a high efficiency obtained by rolling of balls between the screw shaft and the nut body. Where ball contacts shaft and nut body, the rolling friction occurs. This feature is one of main advantages compared to alternative solutions like an acme screws, where the screw thread surface slides directly on the nut thread surface, so the sliding friction occurs in the contact zone.

Ball screws can be classified as follows (in accordance with Standards ISO 3408 and DIN 69051):

- positioning ball screws,
- transport ball screws.

The difference between the two typologies is related to application requirements, where an accuracy and a position repeatability are the most important.

The positioning ball screw is used where high stiffness, high positioning accuracy and high repeatability is required. Ball screws with preloaded nut are mainly used in these applications.

The transport ball screw is used for moving a load where stiffness, accuracy and/or repeatability is not required.

The above mentioned standards ISO 3408 and DIN 6905 also define all ball screw constructive parameters.

### 1.2 Ball track profile

There are two thread track profiles:

- profile with a round groove - two flanks of the groove make a part of the same arch (both flanks centres coincide with the ball centre),
- profile with a gothic (ogival) groove - two flanks of the groove are two arches, their centres are moved respect to the ball centre, in order to obtain a required contact angle.

Generally, a small variation of thread profile geometry strongly influences performances of the ball screw system.

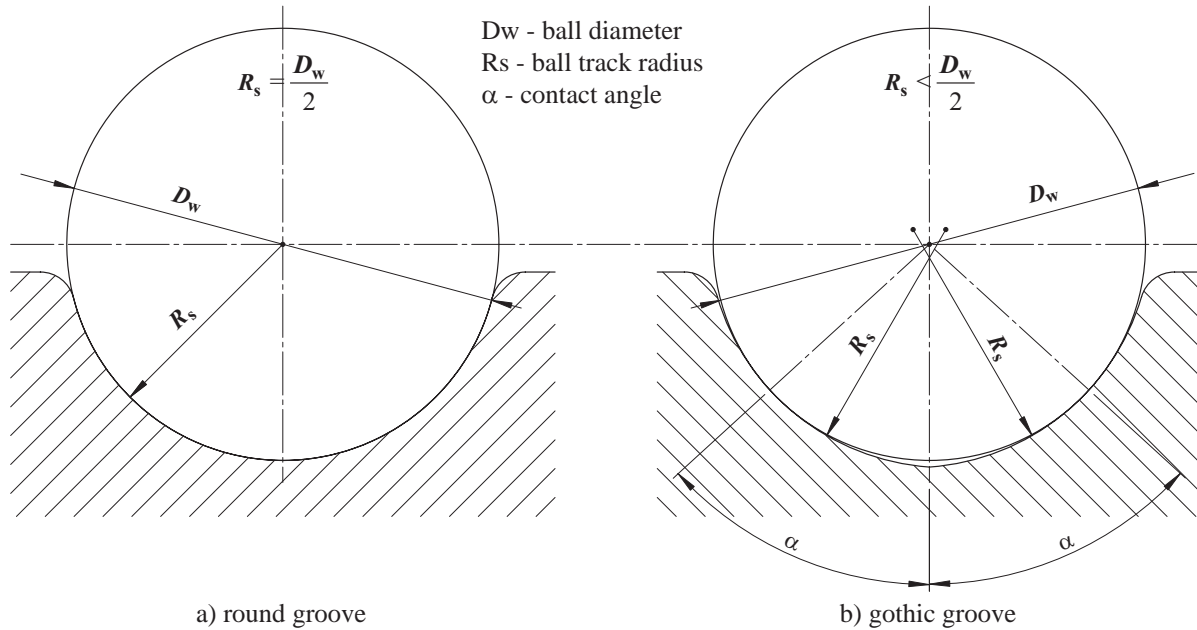


Fig. 2 - Ball track profile

The **round profile** (see Figure 2.a): the zone of contact between the ball and the ball track, where significant effects of sliding are present, is quite wide; consequences of sliding are a high wear of bodies in contact, a relatively high power losses (heating), a relatively low efficiency and life. The round profile is used in applications with a high load and a very low linear speed.

The **gothic profile** (see Figure 2.b): the zone of contact between the ball and the ball track, where significant effects of sliding are present, is very reduced; it helps lower wear of bodies in contact, lower power losses (heating), smaller thermal deformation (and, as consequence, higher accuracy and better repeatability of the system during positioning), higher efficiency and longer life.

SERVOMECH designs and manufactures ball screws, whose thread has got the gothic profile.

### 1.3 Ball recirculation

The continuous ball recirculation inside the thread is achieved by ball recirculating elements (called also liners or deflectors) fitted into the ball nut body.

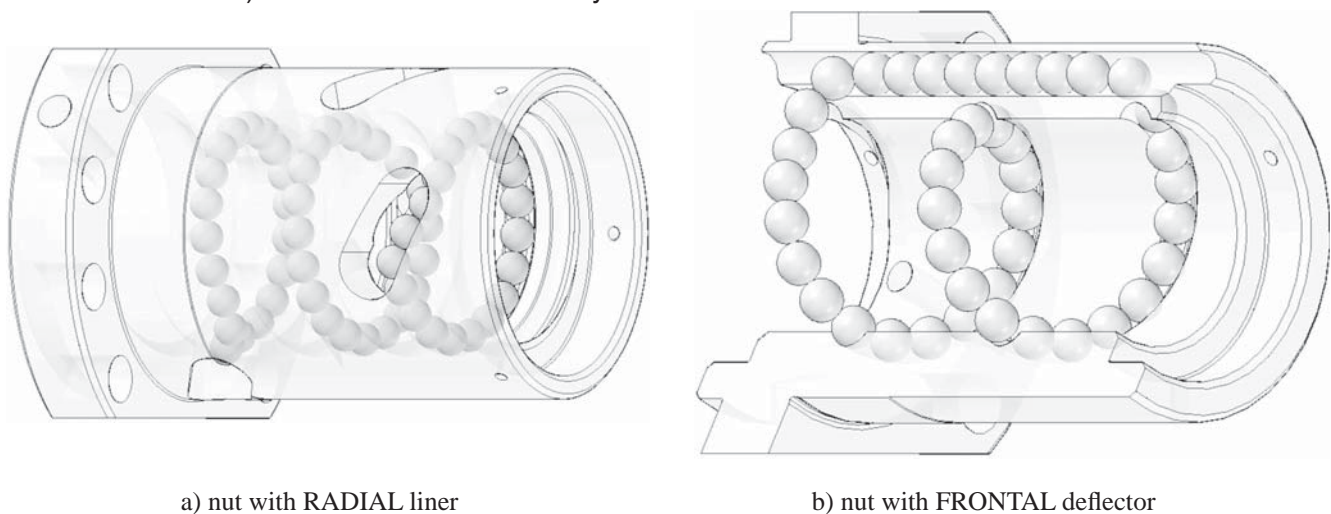


Fig. 3 - Ball recirculation

A solution used to obtain the ball recirculation affects:

- max. rotating speed of ball screw threaded shaft or ball nut,
- the system load capacity,
- ball nut axial stiffness.

A solution with the RADIAL liner (see Figure 3.a) is usually used in ball screws with lead not greater than 20 mm; the liner is fit in the groove present in the ball nut body and restricts a ball trajectory at one single revolution around the ball screw shaft.

A solution with the FRONTAL deflector (see Figure 3.b) is usually used in ball screws with lead greater than 10 mm and in all multiple-start ball screws. The recirculation of balls is obtained by deflectors fitted on ends of the ball nut body and joined with an axial hole (passing through the body); the deflectors deviate ball trajectory from the ball track to the axial hole or vice versa.

SERVOMECH designs and manufactures ball screws realizing various ball recirculating solutions, suitable for specific, concrete application and its conditions and requirements.

### 1.4 Backlash or preload

Depending on type of the ball nut used (preloaded or not preloaded ball nut), applying an axial, centric load to the ball screw shaft or nut, two effects may occur:

- backlash,
- elastic deformation.

The backlash is an axial displacement which nut or threaded shaft makes, without any relative rotation between them; it can be caused by inversion of nut or threaded shaft motion direction or by inversion of applied load direction.

The elastic deformation coincides to axial deflection of parts in contact, under the action of an unidirectional axial force applied.

In case of the ball screw with backlash, applying an axial force, both effects occur. The diagram in Figure 4.a shows the total axial displacement  $\Delta l$  of ball screw assembly related to the applied force  $F$ . In the left part of the diagram, where  $F = 0$ , the displacement  $S_a$  indicates the ball screw backlash, while in the right part of the diagram, where  $F > 0$ , the displacement  $\Delta l_{Fax}$  indicates the elastic deformation corresponding to force  $F_{ax}$ .

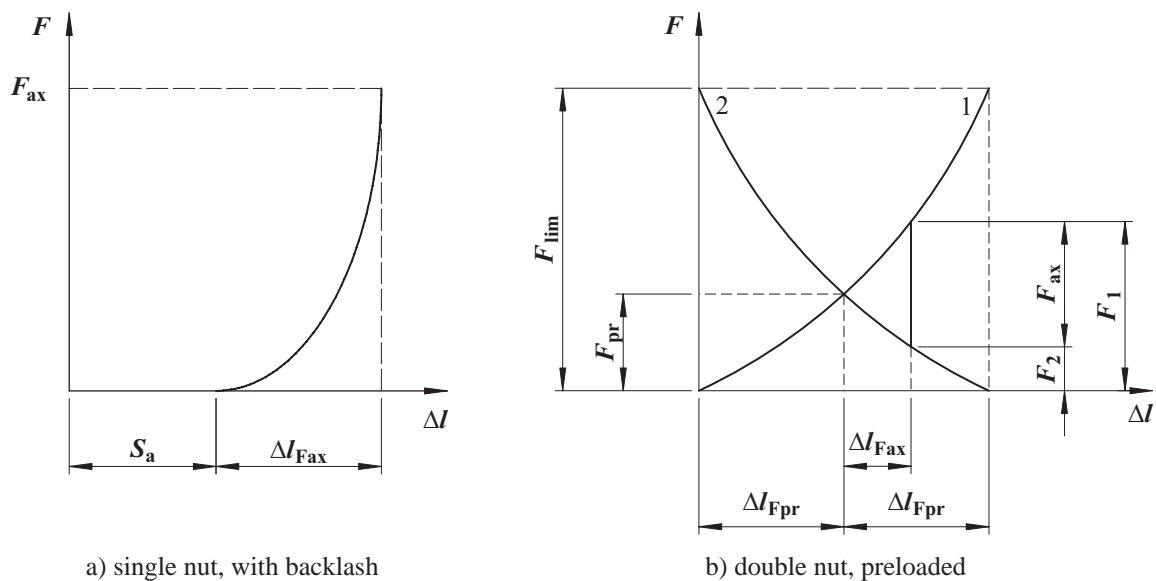


Fig. 4 - Ball screw assembly: axial load - axial displacement diagram

The preload is an axial force, generated different ways inside the ball screw assembly. Its purpose is to eliminate backlash and increase assembly stiffness.

This force must be determined accurately, in order to avoid ball screw life reduction (when the preload is too high) or, otherwise, positioning errors caused by backlash generated under working load (when the preload is too small). The preload depends on the applied axial load:



$$F_{pr} = \frac{F_{ax\ max}}{2.83}$$

where:

- $F_{pr}$  - preload force  
 $F_{ax\ max}$  - max. working load

The diagram on Figure 4.b shows the total ball track elastic deformation along the axis related to the level of the load applied on a preloaded ball screw.

The two curves (1 and 2) represent two semi-nuts of the same assembly. The intersection point represents the preload force  $F_{pr}$  operating on both semi-nuts without external axial load.

After the external load  $F_{ax}$  is applied, the force acting to the semi-nut 1 changes from  $F_{pr}$  to  $F_1$ , while the force acting to semi-nut 2 changes from  $F_{pr}$  to  $F_2$ , in order to keep a ratio  $F_1 = F_2 + F_{ax}$ .

The force  $F_{lim}$  represents the max. load, which will not cause detachment of balls and ball tracks. This force is  $2.83 \times F_{pr}$ .

Generally, for preloaded nuts, SERVOMECH recommends a preload force of  $F_{pr} = 0.08 \times C_a$ ; in a particular condition, this value can be reduced or increased, but in any case the max. value must not exceed  $F_{pr} = 0.12 \times C_a$ .

There are three methods of preloading:

- preload with four contact points (see Figure 5),
- compression preload (see Figure 6),
- traction preload (see Figure 7).

The first preloading method is valid for single nuts only and it is suitable for applications with low linear speed.

This method requires use of balls with effective diameter greater than nominal; this way, there will be four contact points for each ball, two between ball and nut and two between ball and threaded shaft, and the backlash will be eliminated (see Figure 5).

This solution doesn't allow an optimal rolling of balls because sliding between surfaces in contact may occur.

In this case, the preload force must not exceed  $F_{pr} = 0.04 \times C_a$ , in order to prevent ball screw overheating and consequently life reduction.

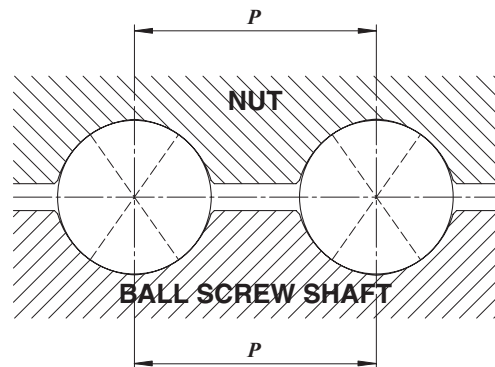


Fig. 5 - Preload with 4 contact points

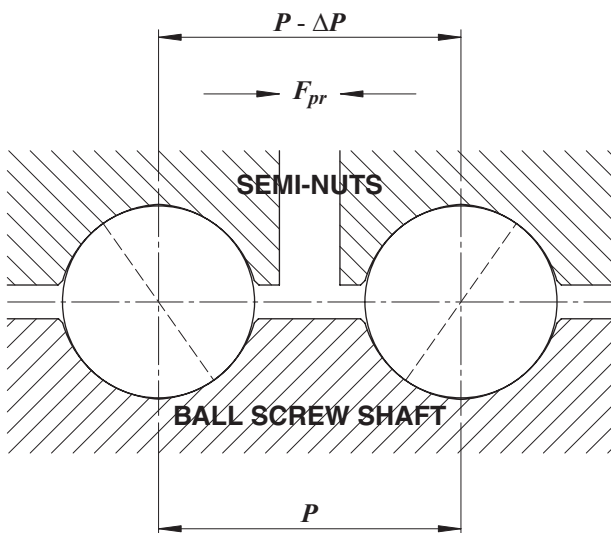


Fig. 6 - Compression preload

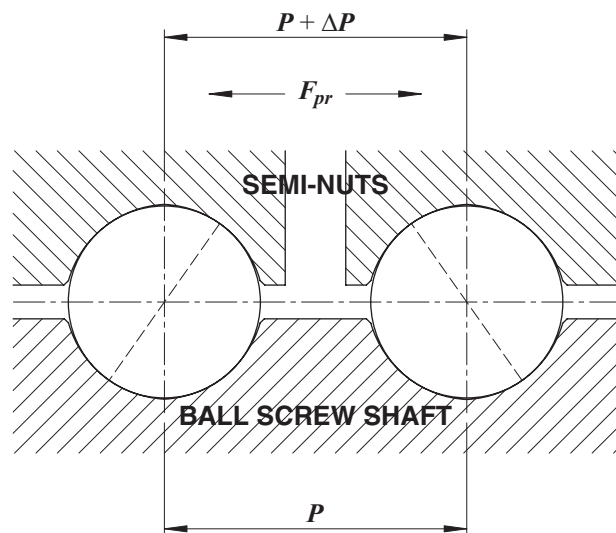


Fig. 7 - Traction preload

The second and the third preloading method can be applied either to ball screws with single nut or to ball screws with double nut, giving them an optimal efficiency. There are only two contact points for each ball, one between ball and nut and one between ball and threaded shaft, so a sliding between surfaces in contact may not occur.

In case of ball screws with single nut, the preload force is obtained by thread lead variation (called *shift*) during the finishing of the internal thread.

In case of ball screws with double preloaded nut - compression preload, the contact points have "X-configuration". The preload force tries to make two semi-nuts less distant and compresses the part of threaded shaft between them. This solution is applied to cylindrical preloaded nuts, fit into a housing and preloaded by means of locknut or cover and bolts.

In case of ball screws with double preloaded nut - traction preload, the contact points have "O-configuration". The preload force tries to make two semi-nuts more distant and pulls the part of threaded shaft between them. This solution is applied to preloaded nuts with flange or cylindrical, where the preloading force is obtained by interposing a spacer ring with calibrated thickness between the two semi-nuts.

The actual preload force depends on the distance  $\Delta P$ , so this distance must be determined very carefully, in order to avoid overloading or overheating of ball screw, with consequent reduction of its performances and life.

### 1.5 Accuracy grade

The accuracy grade is the quality level reached during the manufacturing of ball screws, which identifies relevant geometrical and dimensional parameters and defines specific tolerances.

SERVOMECH applies ISO 3408 and DIN 69051 regulations as reference standards for own production. Depending on the ball screw application typology, different accuracy grades are recommended:

Application typology	Accuracy grade recommended
positioning	1, 3, 5
transport	1, 3, 5, 7, 10

SERVOMECH manufactures ball screws in accordance with accuracy grade 3, 5, 7, 10.

### 1.6 Thread lead accuracy

Main parameters which contribute to determine the thread lead accuracy are:

- $l$  - threaded length of the shaft
- $l_u$  - threaded length of the shaft, subjected to the specified accuracy
- $l_e$  - threaded length of the shaft, not subjected to the specified accuracy  
(SERVOMECH considers its length equal to the thread nominal diameter)
- $l_0$  - nominal threaded length of the shaft
- $l_s$  - specified threaded length of the shaft
- $e_p$  - tolerance of the mean error, referred to  $l_u$  thread portion
- $e_{0a}$  - actual mean travel deviation, referred to nominal thread portion  $l_0$
- $C$  - linear compensation, valid for  $l_0$  thread portion (possibly required by customer)
- $e_{sa}$  - actual mean travel deviation, referred to specified thread portion  $l_s$
- $v_{up}$  - permissible travel variation, referred to  $l_u$  thread portion
- $v_{300p}$  - permissible travel variation, referred to 300 mm long thread portion
- $v_{300a}$  - actual travel variation, measured over 300 mm long thread portion
- $v_{2\pi p}$  - permissible travel variation, referred to a thread portion equivalent to  $2\pi$  radian (1 shaft revolution)
- $v_{2\pi a}$  - actual travel variation, measured over a thread portion equivalent to  $2\pi$  radian (1 shaft revolution)

Diagrams Figure 8.a and 8.b illustrate the graphical method of travel deviation evaluation.

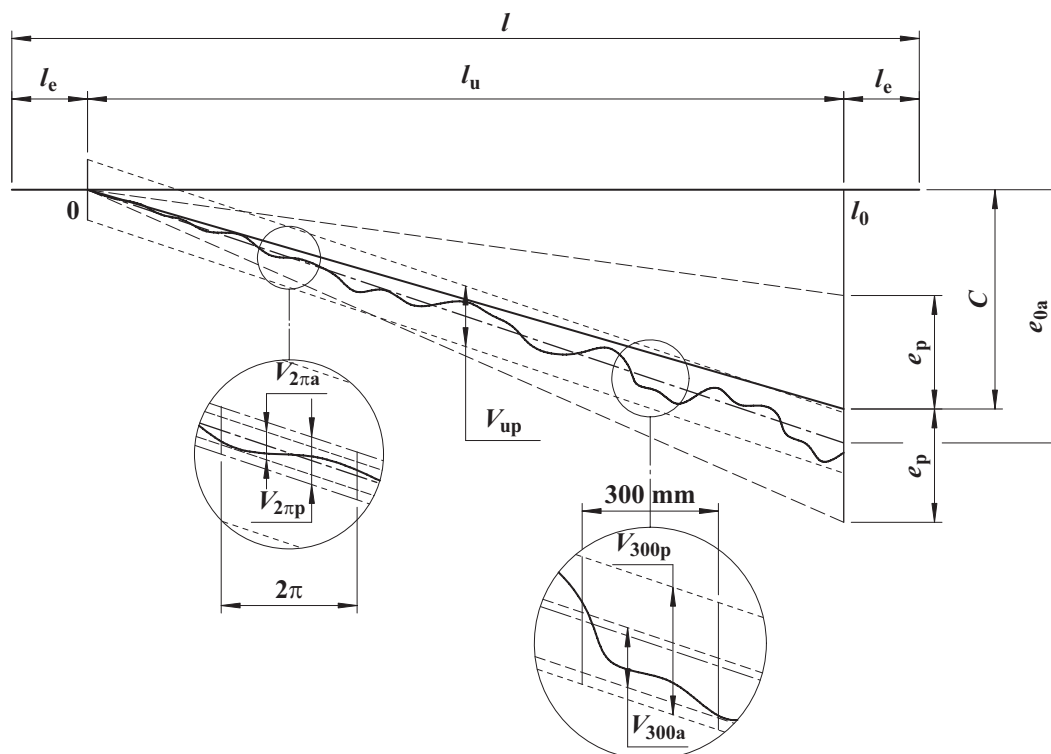


Fig. 8.a - Travel deviation in relation to nominal travel

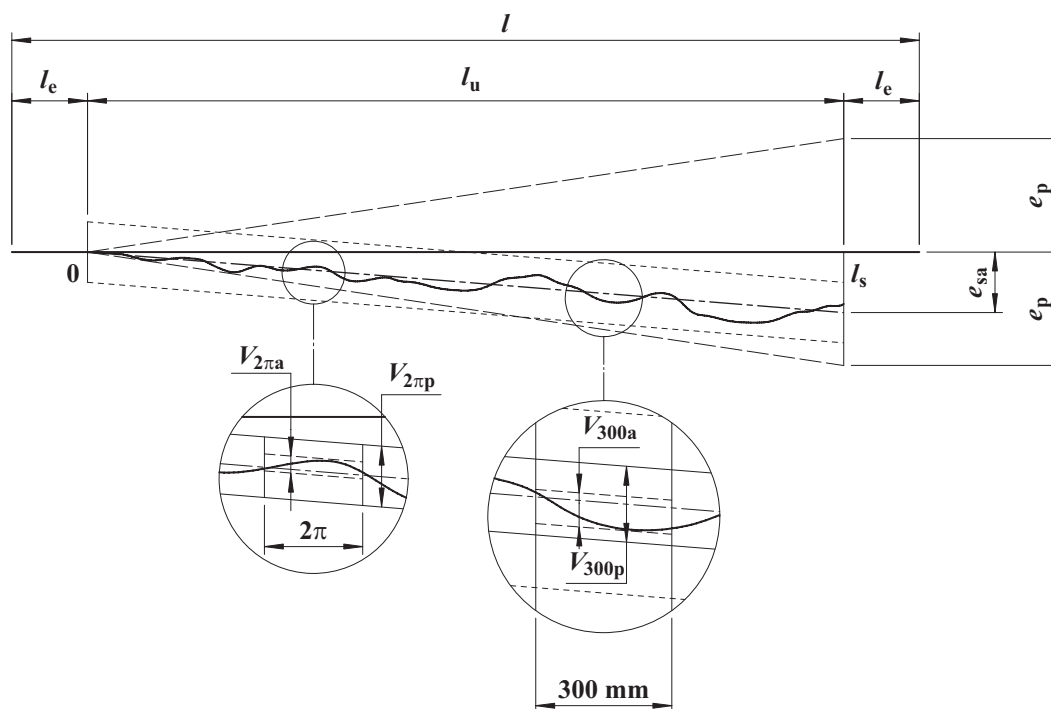


Fig. 8.b - Travel deviation in relation to specified travel



Positioning ball screws

Thread length $l_u$ [mm]		Travel variation allowed referred to $l_u$ $v_{up}$ [ $\mu$ m]				
		STANDARD TOLERANCE GRADE				
>	$\leq$	1	3	5	7	10
0	315	6	12	23	–	–
315	400	6	12	25	–	–
400	500	7	13	26	–	–
500	630	7	14	29	–	–
630	800	8	16	31	–	–
800	1 000	9	17	34	–	–
1 000	1 250	10	19	39	–	–
1 250	1 600	11	22	44	–	–
1 600	2 000	13	25	51	–	–
2 000	2 500	15	29	59	–	–
2 500	3 150	17	34	69	–	–
3 150	4 000	21	41	82	–	–
4 000	5 000	–	49	99	–	–
5 000	6 300	–	–	119	–	–

Positioning ball screws

Thread length $l_u$ [mm]		Tolerance of mean error, referred to $l_u$ $e_p$ [ $\mu$ m]				
		STANDARD TOLERANCE GRADE				
>	$\leq$	1	3	5	7	10
0	315	6	12	23	52	210
315	400	7	13	25	57	230
400	500	8	15	27	63	250
500	630	9	16	32	70	280
630	800	10	18	36	80	320
800	1 000	11	21	40	90	360
1 000	1 250	13	24	47	105	420
1 250	1 600	15	29	55	125	500
1 600	2 000	18	35	65	150	600
2 000	2 500	22	41	78	175	700
2 500	3 150	26	50	96	210	860
3 150	4 000	32*	62*	115*	260*	1 050*
4 000	5 000	39*	76*	140*	320*	1 300*
5 000	6 300	48*	92*	170*	390*	1 550*

\* - values calculated by linear extrapolation

Positioning and transport  
ball screws

$v_{300p}$ [ $\mu$ m]				
STD. TOLERANCE GRADE				
1	3	5	7	10
6	12	23	52	210

Positioning ball screws

$v_{2\pi p}$ [ $\mu$ m]				
STD. TOLERANCE GRADE				
1	3	5	7	10
4	6	8	–	–

Transport ball screws

$e_p$ [ $\mu$ m]				
STD. TOLERANCE GRADE				
1	3	5	7	10
$e_p = 2 \times \frac{l_u}{300} \times V_{300f}$				

## 2. SELECTION OF BALL SCREWS

### 2.1 Introduction

Elements that affect the functioning of the ball screw, as well as guidelines for proper sizing of ball screws, are indicated below.

In order to allow correct sizing of ball screws, the following points must be known:

- required life,
- detailed working cycle (all load levels, relative speed and working period of time),
- mounting conditions,
- environmental conditions,
- lubrication conditions.

During work, the load applied to the ball screw must be coaxial with a screw itself. It is essential for proper functioning of the ball screw and achievement of the required life. Any load not coaxial to the screw, caused by misalignment and/or other reasons, significantly reduces its life. In addition, it must be supported by guides or external support systems.

**Step 1:** determinate the **thread helix lead ( $P_h$ )**

$$P_h = \frac{v}{n}$$

where:

- $P_h$  [mm] - thread helix lead
- $v$  [mm/min] - linear speed (of threaded shaft or nut)
- $n$  [rpm] - rotating speed (of nut or threaded shaft)

As a first approximation, the rotating speed can be considered equal to the max. working speed.

**Step 2:** determinate the **equivalent axial dynamic load ( $F_m$ )**

The **equivalent axial dynamic load** is defined as that hypothetical axial load, constant in magnitude and direction, acting axially and centrically on a ball screw, which, if applied, would have the same influence on ball screw life as the actual loads to which the ball screw is subjected. It must be determined by dividing the working cycle at separate, distinct subcycles, each of them identified with the proper load level, rotation speed and time period.

The equivalent axial dynamic should be determined considering actual working conditions:

- **case 1:** linearly variable axial load at constant speed

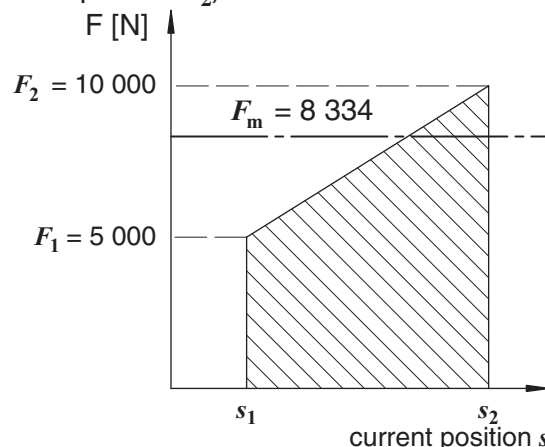
$$F_m = \frac{F_1 + 2 \times F_2}{3}$$

where:

- $F_1$  - axial load at the beginning of displacement (current position  $s_1$ )
- $F_2$  - axial load at the end of displacement (current position  $s_2$ )

example:

$i$	$F_i$ [N]	$F_m$ [N]
1	5 000	8 334
2	10 000	



- **case 2:** variable axial load at constant speed

$$F_m = \sqrt[3]{F_1^3 \times \frac{t_1}{t_1 + t_2 + t_3 + \dots} + F_2^3 \times \frac{t_2}{t_1 + t_2 + t_3 + \dots} + F_3^3 \times \frac{t_3}{t_1 + t_2 + t_3 + \dots} + \dots}$$

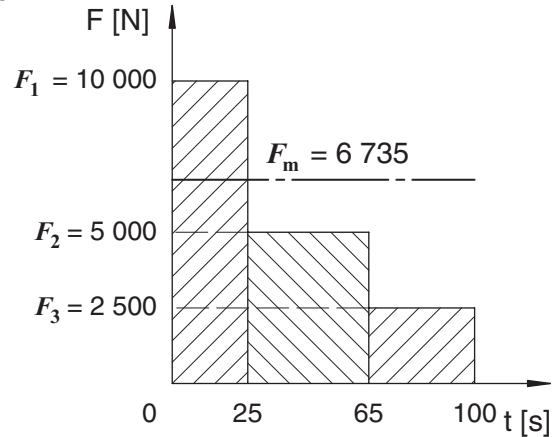
where:

$F_1, F_2, F_3, \dots$  - level of axial load, relative to a single subcycle

$t_1, t_2, t_3, \dots$  - time period of a single subcycle

example:

$i$	$t_i$ [s]	$F_i$ [N]	$F_m$ [N]
1	25	10 000	6 735
2	40	5 000	
3	35	2 500	



- **case 3:** variable axial load at variable speed

$$F_m = \sqrt[3]{F_1^3 \times \frac{n_1}{n_m} \times \frac{t_1}{t_1 + t_2 + t_3 + \dots} + F_2^3 \times \frac{n_2}{n_m} \times \frac{t_2}{t_1 + t_2 + t_3 + \dots} + F_3^3 \times \frac{n_3}{n_m} \times \frac{t_3}{t_1 + t_2 + t_3 + \dots} + \dots}$$

where:

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

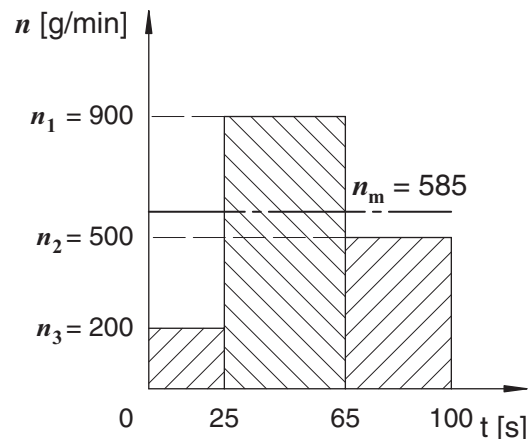
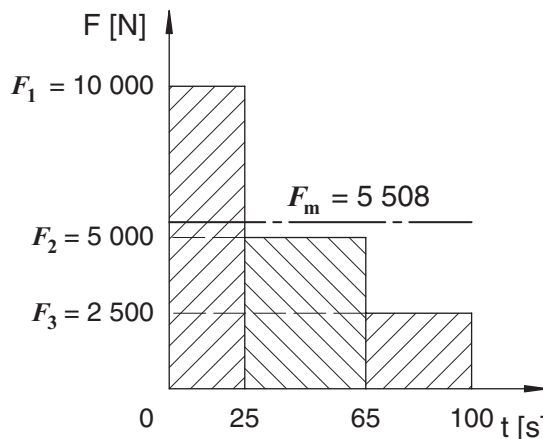
$F_1, F_2, F_3, \dots$  - level of axial load, relative to a single subcycle

$n_1, n_2, n_3, \dots$  - rotation speed, relative to a single subcycle

$t_1, t_2, t_3, \dots$  - time period of a single subcycle

example:

$i$	$t_i$ [s]	$n_i$ [rpm]	$F_i$ [N]	$n_m$ [rpm]	$F_m$ [N]
1	25	200	10 000	585	5 508
2	40	900	5 000		
3	35	500	2 500		



In case of the preloaded nut, the equivalent axial dynamic load must be calculated taking into account the preload force too, adding the preload force value to the level of the axial load of each single subcycle.

## 2.2 Basic rating life

The basic rating life is expressed as a number of revolutions of the threaded shaft with respect to the nut or vice versa, before fatigue effects and consequences occur on the threaded shaft and/or nut and/or rolling bodies.

The following formula is used for calculation of the **basic rating life** ( $L_{10}$ ):

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

where:

- $L_{10}$  [rev] - basic rating life of the ball screw
- $C_a$  [N] - basic dynamic axial load rating
- $F_m$  [N] - equivalent axial dynamic load
- $f_{sh}$  - shock factor
- $f_{sh} = 1$ : load without shocks
- $1 < f_{sh} \leq 1.3$ : load with light shocks
- $1.3 < f_{sh} \leq 1.8$ : load with medium shocks
- $1.8 < f_{sh} \leq 3$ : load with strong shocks

The above formula provides the 90 % ball screw reliability, i. e. the number of the threaded shaft or ball nut revolutions reached by 90% of ball screws, apparently identical, subject to the same loading conditions, the same motion and environmental conditions.

The basic rating life expressed in hours ( $L_{10h}$ ) is calculated as follows:

$$L_{10h} = \frac{L_{10}}{60 \times n_m}$$

In case it is necessary to determinate the ball screw life with greater reliability (**modified ball screw rating life**,  $L_{10m}$ ), the correction factor  $f_a$  must be applied:

$$L_{10m} = L_{10} \times f_a$$

Reliability [%]	90	95	96	97	98	99
Factor $f_a$	1	0.62	0.53	0.44	0.33	0.21

## 2.3 Basic dynamic axial load rating ( $C_a$ )

The **basic dynamic axial load rating** ( $C_a$ ) is a constant, centric axial load that a ball screw can theoretically endure for a basic rating life of 1 000 000 revolutions.

The dynamic load  $C_a$  helps to calculate the ball screw life. This parameter depends on the ball track profiles geometry, materials used as well as the hardness of components which constitute the ball screw itself, is supplied by the ball screw manufacturer.

In accordance with standards ISO 3408 and DIN 69051, the following formulae is used to calculate a corrective factor for the basic dynamic axial load rating  $C_a$  which takes into account a hardness of ball tracks  $f_{h0}$ :

$$f_{h0} = \left( \frac{\text{actual hardness HV } 10}{654 \text{ HV } 10} \right)^3 \leq 1$$

while the standard tolerance grade is taken into account with  $f_{ac}$  factor:

Standard tolerance grade	1, 3 or 5	7	10
Factor $f_{ac}$	1	0.9	0.7

## 2.4 Basic static axial load rating ( $C_{0a}$ )

The **basic static axial load rating** ( $C_{0a}$ ) is a static load, applied axially and centrically to the ball screw, which generates a total permanent deformation of ball and ball track, at the most heavily point of contact between them, of  $0.0001 \times D_w$  ( $D_w$  - ball diameter).

In applications where the load is applied to a stationary ball screw or a ball screw rotating at very low speed (less than 10 rpm) for short duration, the basic static axial load rating ( $C_{0a}$ ) should be greater than the load applied, in order to guaranty a proper working and integrity of the system. In case it is lower, the ball screw could suffer irreversible damage that would make it unusable.

A reasonable safety factor, related to the application, should be considered when comparing the basic static axial load rating ( $C_{0a}$ ) and the applied static load.

## 2.5 Max. allowed rotating speed ( $n_{max}$ )

Following factors restrict a ball screw rotating speed:

- A) external factors (threaded shaft length, shaft end supports),
- B) internal factors (ball material, geometry and material of recirculation elements).

A) Rotating speed of the threaded shaft must not reach critical level, in order to ensure a proper working of a ball screw system and to prevent imbalances which could damage the ball screw.

The critical rotating speed depends on the threaded shaft diameter, type of shaft end support (Figure 8) and the ball shaft length ( $L$ , Figure 8).

The following formulae is used to calculate the max. allowed rotating speed ( $n_{max}$ ). It restricts the rotating speed to 80 % of the critical value and it is valid for threaded shafts without an axial through hole:

$$n_{max} = 2.17 \times 10^8 \times K_1 \times \frac{d_3}{L^2}$$

where:

$n_{max}$	[rpm]	- max. allowed rotating speed
$d_3$	[mm]	- threaded shaft root diameter
$L$	[mm]	- length of threaded shaft between supports
$K_1$		- factor of shaft end support type:
		cantilever beam                      scheme 1 $K_1 = 0.144$
		simple supported beam            scheme 2 $K_1 = 0.447$
		supported cantilever beam       scheme 3 $K_1 = 0.694$
		constrained beam                   scheme 4 $K_1 = 1$

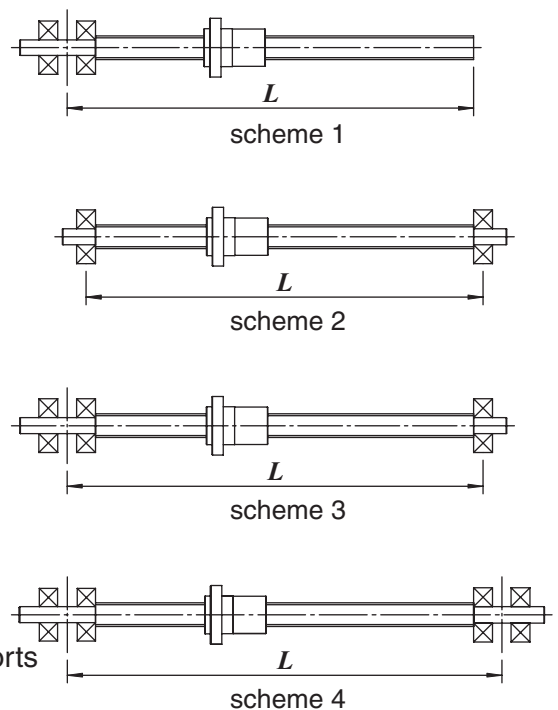


Fig. 8 - Threaded shaft supports scheme

B) The rotating speed can be determined as  $d_0 \times n$  ( $d_0$  [mm] - nominal thread diameter,  $n$  [rpm] - rotating speed). SERVOMECH consider following values:

- ( $d_0 \times n_{max}$ ) = 90 000, for rolled threads and recirculation elements in plastic material,
- ( $d_0 \times n_{max}$ ) = 120 000, for whirled threads and recirculation elements in steel.

**The max. allowed rotating speed of the ball screw system is the lower speed calculated using A and B criteria.**



## 2.6 Max. allowed buckling load ( $F_{c \max}$ )

When a push load is applied on the threaded shaft - nut system, a buckling load check must be done. The applied push load must not exceed a value calculated with the following formulae (EULER's formula), valid for threaded shafts without an axial through hole only:

$$F_{c \max} = K_2 \times \frac{\pi^2 \times E \times I}{L^2 \times s} = K_2 \times \frac{\pi^3 \times E \times d_3^4}{L^2 \times s \times 64}$$

where:

$F_{c \max}$ [N]	- max. allowed buckling load
$K_2$	- factor of shaft end support type (ref. Figure 8):
	cantilever beam                      scheme 1 $K_2 = 0.25$
	simple supported beam            scheme 2 $K_2 = 1$
	supported cantilever beam       scheme 3 $K_2 = 2$
	constrained beam                 scheme 4 $K_2 = 4$
$E$ [N/mm <sup>2</sup> ]	- modulus of elasticity
$I$ [mm <sup>4</sup> ]	- area moment of inertia
$d_3$ [mm]	- threaded shaft root diameter
$L$ [mm]	- length of threaded shaft between supports (rif. Figure 8)
$s$	- safety factor against buckling (the value used by SERVOMECH: $s = 3$ )

## 2.7 Efficiency ( $\eta$ )

As explained before, one of main ball screw features is a high efficiency, obtained thanks to balls interposed between the threaded shaft and nut, which roll during operation of the system and consequent presence of rolling friction in contact points, i.e. absence of sliding friction.

The graph shown in Figure 9 illustrates variation of efficiency of a threaded shaft - nut system related to the thread lead angle, for two significant examples: ball screw with gothic profile (curves *a* - direct efficiency, and *b* - backdriving efficiency) and acme screw (curve *c*).

In case of ball screw, the efficiency is (95 ... 96) %, while in case of acme screw, where the sliding friction is present during operation, the efficiency is (40 ... 42) %.

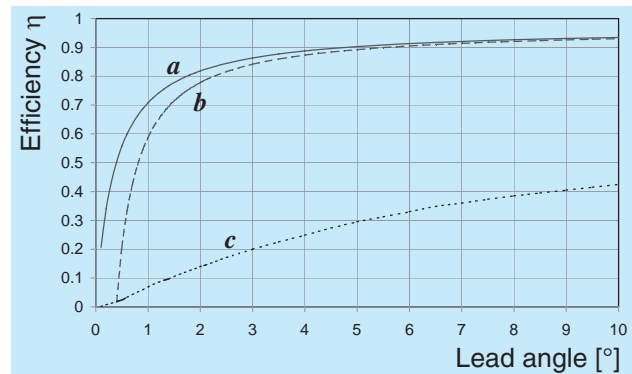


Fig. 9 - Ball screw efficiency

The theoretical **direct efficiency** ( $\eta$ ) of the system where the rotary motion of the screw is transformed into a translatory motion of the nut can be calculated as follows:

$$\eta = \frac{\operatorname{tg} \varphi}{\operatorname{tg}(\varphi + \gamma)}$$

where:

- $\varphi$  [°] - lead angle of the thread
- $\gamma$  [°] - friction angle ( $0.20 \leq \gamma \leq 0.55$ )

The theoretical **backdriving efficiency** ( $\eta'$ ) of the system where the translatory motion of the nut is transformed into a rotary motion of the screw can be calculated as follows:

$$\eta' = \frac{\operatorname{tg}(\varphi - \gamma)}{\operatorname{tg} \varphi}$$

It is important to keep in mind that the efficiency may vary depending on working conditions (load, speed, type of lubricant used).

## 2.8 Torque and power

The Figure 10 illustrates an example of a ball screw system, driven by an electric motor, through a mechanical transmission. Following formulas help to calculate parameters necessary to determine the motor size properly.

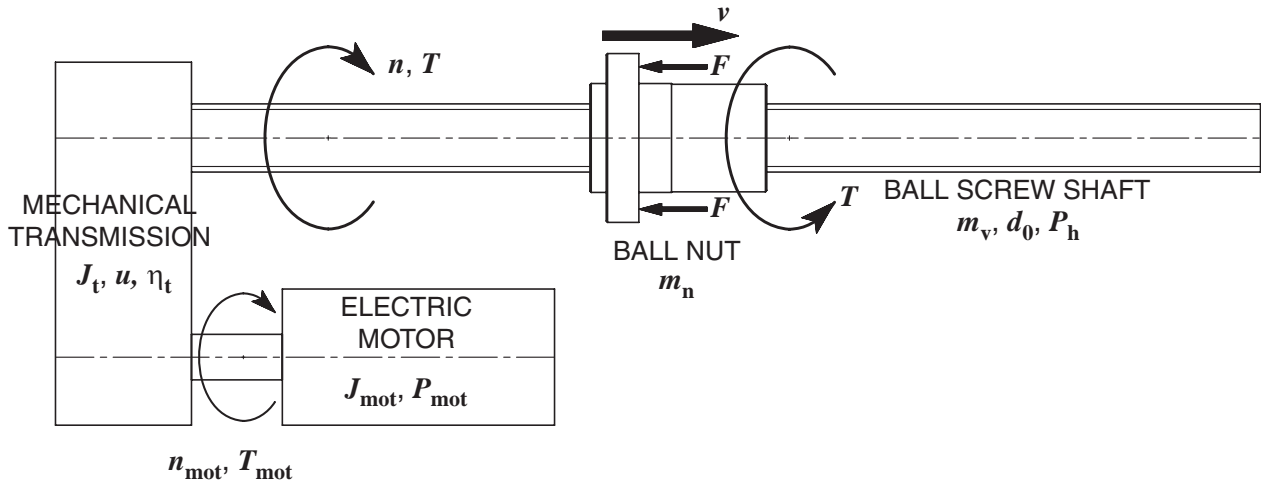


Fig. 10 - Schema vite a sfere azionato da motore elettrico tramite trasmissione meccanica

The torque ( $T$ ) to be applied on the threaded shaft in order to make translate the load ( $F$ ) applied on the nut, when the load and speed are constant:

$$T = \frac{F \times P_h}{2 \times \pi} \times \frac{1}{\eta}$$

where:

- $T$  [Nm] - torque to be applied on the threaded shaft
- $F$  [kN] - load applied on the nut
- $P_h$  [mm] - thread helix lead
- $\eta$  - direct efficiency of ball screw

In case of preloaded ball screw system, the torque ( $T_{pr}$ ) to be applied on the threaded shaft in order to overcome the preload force ( $F_{pr}$ ):

$$T_{pr} = \frac{F_{pr} \times P_h}{2 \times \pi} \times \left( \frac{1}{\eta} - \eta' \right)$$

where:

- $T_{pr}$  [Nm] - torque to be applied on the threaded shaft
- $F_{pr}$  [kN] - preload force
- $P_h$  [mm] - thread helix lead
- $\eta'$  - direct efficiency of ball screw

The braking torque ( $T_{fr}$ ) to be applied on the threaded shaft in order to prevent the axial motion of the nut subjected to load ( $F$ ):

$$T_{fr} = \frac{F \times P_h}{2 \times \pi} \times \eta'$$

where:

- $T_{fr}$  [Nm] - braking torque to be applied on the threaded shaft
- $F$  [kN] - load applied on the nut
- $P_h$  [mm] - thread helix lead
- $\eta'$  - backdriving efficiency of ball screw

The torque ( $T_{mot}^{nom}$ ) required on motor shaft, at constant motor shaft speed:

$$T_{mot}^{nom} = \frac{T + T_{pr} + T_{att}}{u \cdot \eta_t}$$

where:

- $T_{mot}^{nom}$  [Nm] - torque required on motor shaft
- $T$  [Nm] - torque to be applied on threaded shaft in order to make load  $F$
- $T_{pr}$  [Nm] - torque to be applied on threaded shaft in order to overcome the preload force  $F_{pr}$
- $T_{att}$  [Nm] - friction torque, due to guides and/or bearings
- $u$  - ratio of mechanical transmission between threaded shaft axis and motor axis
- $\eta_t$  - efficiency of mechanical transmission

The torque required on motor shaft to overcome inertia of rotating parts during acceleration ( $T_{mot}^{acc}$ ):

$$T_{mot}^{acc} = J \times \alpha \times 10^{-6}$$

where:

- $T_{mot}^{acc}$  [Nm] - torque required on motor shaft
- $J$  [kg·mm<sup>2</sup>] - moment of inertia of parts in motion, referred to motor axis

$$J = J_{mot} + J_t + m_v \times \left(\frac{d_0}{2}\right)^2 \times \frac{1}{u^2} + m_s \times \left(\frac{P_h}{2 \times \pi}\right)^2 \times \frac{1}{u^2}$$

where:

- $J_{mot}$  [kg·mm<sup>2</sup>] - moment of inertia of motor
- $J_t$  [kg·mm<sup>2</sup>] - moment of inertia of mechanical transmission, referred to motor axis
- $m_v$  [kg] - mass of ball screw shaft
- $d_0$  [mm] - nominal diameter of ball screw
- $m_s$  [kg] - mass of structure (including ball nut) along the ball screw axis
- $P_h$  [mm] - thread helix lead
- $u$  - ratio of mechanical transmission between ball screw axis and motor axis
- $\alpha$  [1/s<sup>2</sup>] - angular acceleration

$$\alpha = \frac{\pi \times (n_{mot2} - n_{mot1})}{30 \times \Delta t}$$

where:

- $n_{mot2}$  [g/min] - rotating speed at the end of acceleration
- $n_{mot1}$  [g/min] - rotating speed at the beginning of acceleration
- $\Delta t$  [s] - acceleration time

The torque required on motor shaft in transitional period ( $T_{mot}^{tot}$ ), during acceleration:

$$T_{mot}^{tot} = T_{mot}^{nom} + T_{mot}^{acc}$$

The power required on motor shaft ( $P_{mot}$ ), at constant rotating speed:

$$P_{mot} = \frac{T_{mot}^{nom} \times n_{mot}}{9549}$$

## 2.9 Axial stiffness

In applications where a high positioning accuracy is required, a ball screw axial stiffness becomes very important. The axial stiffness is defined as an axial load applied on the ball screw which generates a unit strain along the applied load direction.

In case of ball screws, the axial stiffness can be calculated as follows:

$$\frac{1}{R_{tot}} = \frac{1}{R_v} + \frac{1}{R_n} + \frac{1}{R_s}$$

where:

$R_{tot}$  [N/μm] - total axial stiffness of ball screw

$R_v$  [N/μm] - stiffness of threaded shaft, calculated as follows:

$$R_v = K_{Rv} \times \frac{d_3^2}{L}$$

where:

$K_{Rv}$  - factor of shaft end support type (ref. to Figure 8):

cantilever beam:  $K_{Rv} = 163.5$

constrained beam:  $K_{Rv} = 654$

$d_3$  [mm] - ball screw shaft root diameter

$L$  [mm] - length of threaded shaft between supports

$R_n$  [N/μm] - axial stiffness of ball nut

$R_s$  [N/μm] - axial stiffness supports (see Figure 4)

The value of the total axial stiffness is always lower than value of any single component stiffness.

The ball nut stiffness ( $R_n$ ) includes a stiffness of the ball nut body (depending on its shape and dimensions) and a stiffness of the zone of contact between track and balls.

In case an exact value of the ball nut stiffness is required, you are kindly asked to contact SERVOMECH Engineering Dpt.

## 2.10 Lubrication

The ball screw must be lubricated in a correct way with a right lubricant, in order to guarantee its proper work.

A choice of the lubricant must be done taking into account following points:

- temperature of parts during the work,
- rotating speed of ball screw shaft / ball nut,
- ball screw nominal diameter,
- load magnitude,
- protection level of ball screw system.

An **oil lubrication** is recommended in applications where the threaded shaft or the ball nut rotate at high speed and, at the same time, where heat to be removed (generated during the work of ball screw or by external sources) is quite high.

Modern oil lubrication systems work based on an air - oil lubricating method, where a very small amount of oil is discharged intermittently into a pipe carrying a constant flow of compressed air. Advantages: ball screw temperature remains stable during the work, the lubricant is uniformly distributed and a very small quantity of it is lost in the environment. In these systems, the heat is removed by an air flow.

The Figure 11 shows a diagram which helps to determine a right lubricant viscosity referred to actual working conditions.

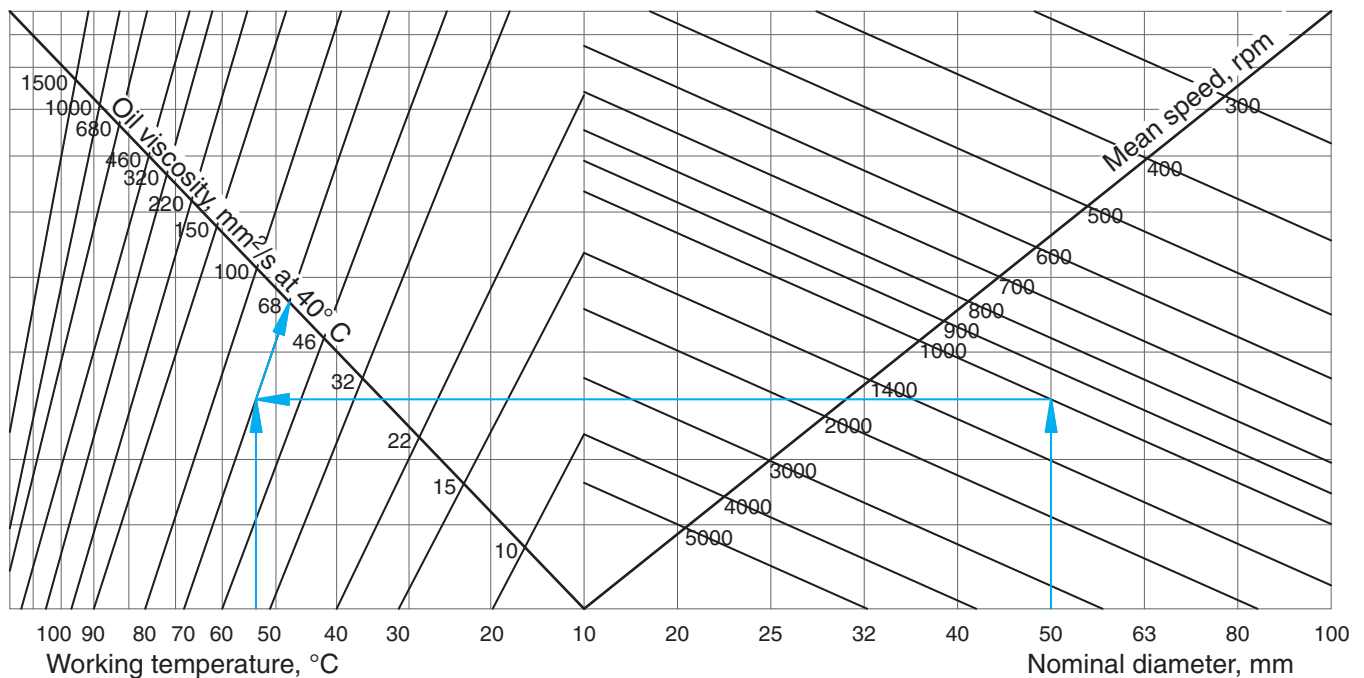


Fig. 11 - Required oil viscosity

**EXAMPLE:**

ball screw nominal diameter: 50 mm  
rotating speed: 1 000 rpm  
working temperature: 55° C

viscosity of oil to be used: 68 mm<sup>2</sup>/s  
(ISO VG 68)

Thanks to their performances, new generation grease allow constant operating temperature of 70° C at high load and at high speed. For this reason, the **grease lubrication** is becoming a commonly used solution. Furthermore, in some cases, a long grease life allows to remove the centralized lubrication system, with consequent cost reduction.

For a proper work and correct maintenance of ball screw, the grease quantity in the system must not exceed 2/3 of free volume between the threaded shaft and ball nut. This way, overheating and a reduction in effective life will be avoided.

SERVOMECH Engineering Dpt. can help you to determine a right grease quality.



### 3. SERVOMECH PRODUCTION RANGE

#### 3.1 Production capability

SERVOMECH manufactures ball screws using last generation technologies, starting from 3D modeling software, through in-process measuring instruments, up to CNC tool machines, kept continuously updated. This way, a high quality of products and services offered by SERVOMECH to its customers is guaranteed.

The production range of SERVOMECH ball screws is quite large, thanks to numerous combinations diameter - lead and the production process.

Threaded shafts for ball screws can be made using two different process:

- rolling,
- whirling.

The following tables indicate limits of SERVOMECH production range:

- nominal diameter  $d_0$  [mm], in accordance with standards ISO 3408 and DIN 69051:

14   16   20   25   32   40   50   63   80   100

- lead  $P_h$  [mm], in accordance with standards ISO 3408 and DIN 69051:

4   5   6   8   10   12   15   16   20   25   32   40

- max. thread length, which can be obtained by the rolling process:

Nominal diameter, $d_0$ [mm]		Max. thread length [mm]
>	≤	
12	20	2 700
20	63	5 700

- max. thread length, which can be obtained by the whirling process:

Nominal diameter, $d_0$ [mm]		Max. thread length [mm]
>	≤	
12	16	1 500
16	25	2 700
25	100	5 400

#### 3.2 Materials

Threaded shafts for SERVOMECH ball screws are manufactured in alloy steel 42 CrMo 4 or 50 CrMo 4 (UNI EN 10083-1), quenched. The hardness of ball tracks is within the range (58 ... 61) HRc.

Ball nuts are manufactured in alloy steel 18 NiCrMo 5 (UNI EN 10084). During the manufacturing process, ball nuts are case-hardened, in order to ensure a hardness of surfaces in contact within the range (58 ... 61) HRc.

Ball recirculating elements can be manufactured in different materials:

- plastic material, for applications with the parameter  $(d_0 \times n) \leq 90\,000$ ,
- alloy steel, case-hardened, with a surface hardness (50 ... 55) HRc, for applications with the parameter  $(d_0 \times n) > 90\,000$ .

#### 3.3 Geometry inspection

In order to ensure an optimal ball screw work and to guarantee the achievement of performances and life time required, it is necessary to carry out a series of geometry tests, in accordance to reference standards.

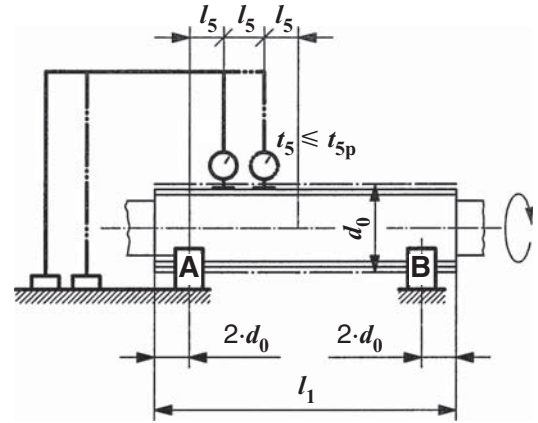
Some tests, defined by standards ISO 3408 and DIN69051, are described below, indicating instructions, necessary tools, equipment list and reference values.

**- test E5:**

Measurement of radial run-out  $t_5$  of ball screw threaded shaft outer diameter for ascertaining straightness related to V-blocks (identified as A and B), per length  $l_5$ .

Test:

- place the threaded shaft in two identical V-blocks, positioned in points A and B;
- set the dial gauge with measuring shoe perpendicular to the cylindrical surface, at the distance  $l_5$  from V-block;
- rotate the threaded shaft slowly while recording changes in the measurements;
- repeat the same procedure at measuring intervals  $l_5$ .



NOMINAL DIA. $d_0$ [mm]		$l_5$ [mm]	$t_{5p}$ [μm], per $l_5$				
>	≤		STD. TOLERANCE GRADE				
			1	3	5	7	10
6	12	80	20	25	32	40	80
12	25	160	20	25	32	40	80
25	50	315	20	25	32	40	80
50	100	630	20	25	32	40	80
100	200	1 250	20	25	32	40	80

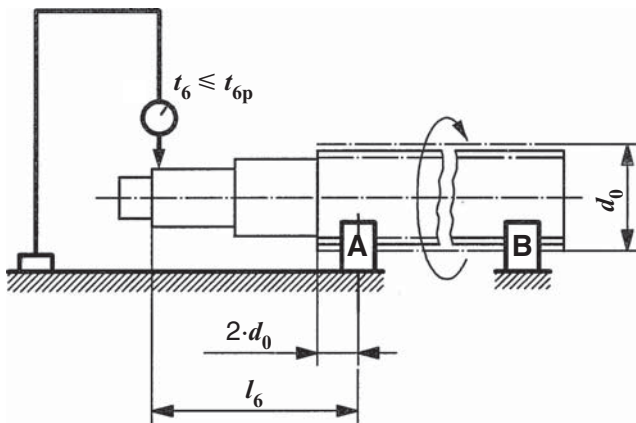
$l_1 / d_0$		$t_{5max p}$ [μm], per $l_1 > 4 \times l_5$				
>	≤	STD. TOLERANCE GRADE				
		1	3	5	7	10
–	40	40	50	64	80	160
40	60	60	75	96	120	240
60	80	100	125	160	200	400
80	100	160	200	256	320	640

**- test E6:**

Measurement of radial run-out  $t_6$  of bearing seat related to V-blocks (identified as A and B), per length  $l_6$ .

Test:

- place the threaded shaft in two identical V-blocks, positioned in points A and B;
- place the dial gauge perpendicular to the cylindrical surface of the bearing seat, at the distance  $l_6$  from V-block;
- rotate the threaded shaft slowly and record the dial gauge readings.



NOMINAL DIA. $d_0$ [mm]		$l$ [mm]	$t_{6p}$ [μm], per $l$				
>	≤		STD. TOLERANCE GRADE				
			1	3	5	7	10
6	20	80	10	12	20	40	63
20	50	125	12	16	25	50	80
50	125	200	16	20	32	63	100
125	200	315	–	25	40	80	125

for  $l_6 \leq l$ : apply above table values

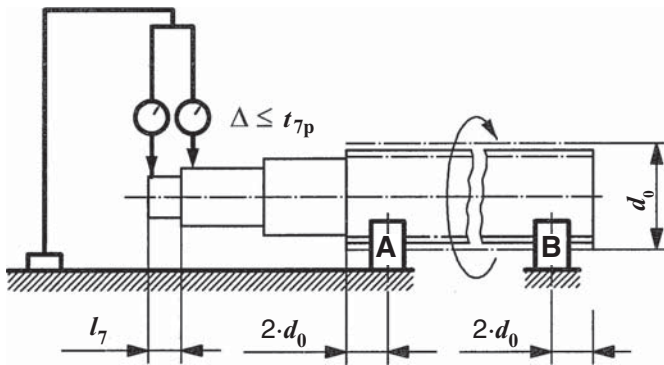
for  $l_6 > l$ : increase above table values by  $t_{6p} \times \frac{l_6}{l}$

**- test E7:**

Measurement of radial run-out  $t_7$  of journal diameter related to the bearing seat by determining the difference  $\Delta$ .

Test:

- place the threaded shaft in two identical V-blocks, positioned in points A and B;
- place the dial gauge perpendicular to the cylindrical surface, at the distance  $l_7$  from V-block;
- rotate the threaded shaft slowly and record the dial gauge readings.



NOMINAL DIA. $d_0$ [mm]		$l$ [mm]	$t_{7p}$ [μm], per $l_7$				
>	≤		STD. TOLERANCE GRADE				
6	20	80	5	6	8	12	16
20	50	125	6	8	10	16	20
50	125	200	8	10	12	20	25
125	200	315	–	12	16	25	32

for  $l_7 \leq l$ : apply above table values

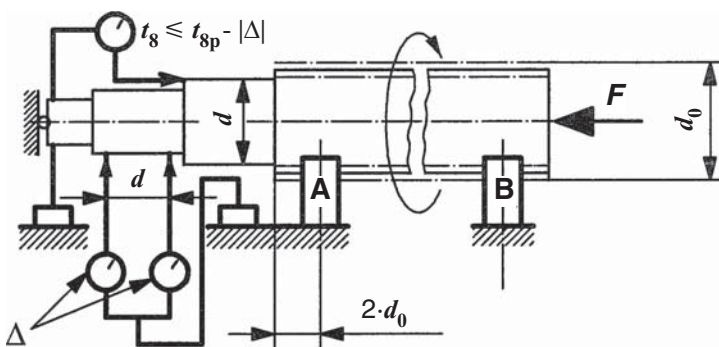
for  $l_7 > l$ : increase above table values by  $t_{7p} \times \frac{l_7}{l}$

- test E8:

Measurement of axial run-out  $t_8$  of bearing seat faces related to V-blocks (identified as A and B).

Test:

- place the threaded shaft in two identical V-blocks, positioned in points A and B;
- secure the ball screw shaft in the axial direction against movement (e.g. by placing a ball between the centres of the ball screw shaft and the mounting surface);
- place the dial gauge perpendicular to the end face of the journal and to the cylindrical surface of the corresponding diameter;
- rotate the screw shaft one revolution and record the dial gauge reading.



NOMINAL DIA. $d_0$ [mm]		$t_{8p}$ [μm]				
>	≤	STD. TOLERANCE GRADE				
6	63	3	4	5	6	10
63	125	4	5	6	8	12
125	200	–	6	8	10	16

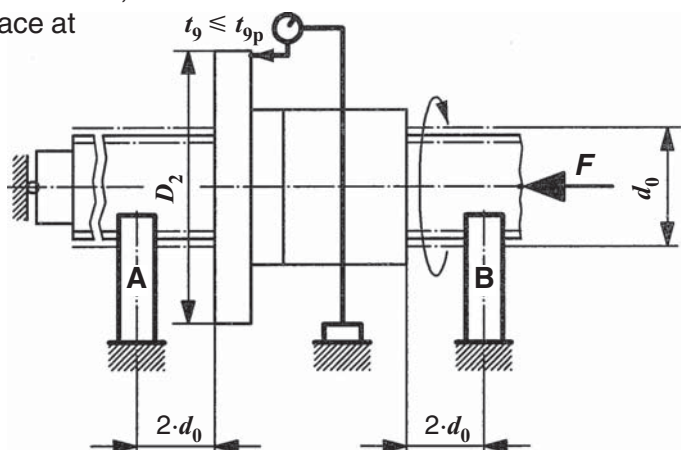
- test E9:

For preloaded ball nuts only - measurement of axial run-out  $t_9$  of ball nut location face related to V-blocks (identified as A and B).

Test:

- place the ball screw in two identical V-blocks, positioned in points A and B;
- secure the ball screw shaft in the axial direction against movement (e.g. by placing a ball between the centres of the ball screw shaft and the mounting surface);
- secure the ball nut against rotation on the ball screw shaft;
- place the dial gauge perpendicular to the flange face at the outer rim of the inspection diameter;
- rotate the ball screw shaft and record the dial gauge readings.

FLANGE DIA. $D_2$ [mm]		$t_{9p}$ [μm]				
>	≤	STD. TOLERANCE GRADE				
16	32	10	12	16	20	–
32	63	12	16	20	25	–
63	125	16	20	25	32	–
125	250	20	25	32	40	–
250	500	–	32	40	50	–

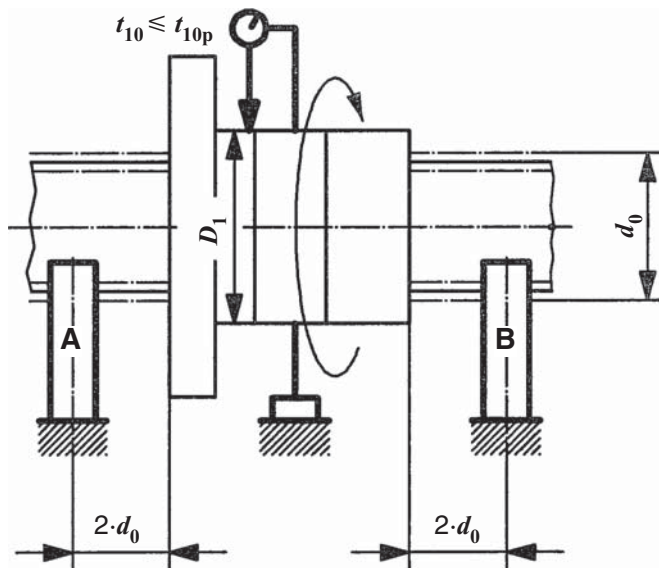


**- test E10:**

For preloaded and rotating ball nuts only - measurement of radial run-out  $t_{10}$ , of ball nut location diameter, related to V-blocks (identified as A and B).

Test:

- place the ball screw in two identical V-blocks, positioned in points A and B;
- secure the ball screw shaft in the axial direction against movement (e.g. by placing a ball between the centres of the ball screw shaft and the mounting surface);
- secure the ball nut against rotation on the ball screw shaft;
- place the dial gauge perpendicular to the cylindrical surface of ball nut location diameter;
- rotate the ball nut body slowly and record the dial gauge readings.



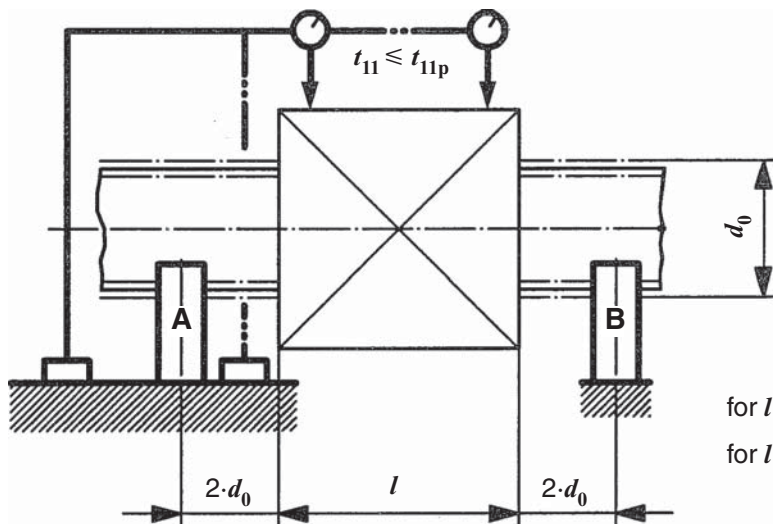
BALL NUT DIA. $D_1$ [mm]		$t_{10p}$ [ $\mu$ m]				
		STD. TOLERANCE GRADE				
>	$\leq$	1	3	5	7	10
16	32	10	12	16	20	—
32	63	12	16	20	25	—
63	125	16	20	25	32	—
125	250	20	25	32	40	—
250	500	—	32	40	50	—

**- test E11:**

For preloaded ball nuts only - measurement of deviation of parallelism  $t_{11}$  of rectangular ball nut (understood in a broad sense; this may also be a cylindrical ball nut mounted in an external casing, which has plans parallel to the nut axis) related to V-blocks (identified as A and B).

Test:

- place the ball screw in two identical V-blocks, positioned in points A and B;
- place the dial gauge perpendicular to the inspection surface;
- probe along the specified inspection length and record the dial gauge readings.



$t_{11p}$ [ $\mu$ m]				
STD. TOLERANCE GRADE				
1	3	5	7	10
16	20	25	32	—

for  $l \leq 100$ : apply above table values

for  $l > 100$ : increase above table values by  $t_{11p} \times \frac{l}{100}$

### 3.4 Mounting suggestions

To ensure a smooth operation of ball screws and compliance with performance requirements, it is necessary to very carefully align the ball screw to external guides and supports, including the error of parallelism between them shall not exceed 0.02 mm.

### 3.5 Working temperature

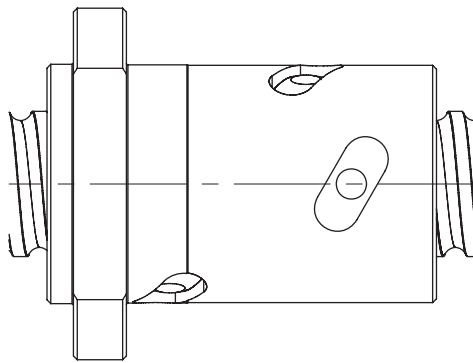
Ball screws manufactured by SERVOMECH can operate within a temperature range  $(-20 \dots +110)^{\circ} \text{C}$ .

Using special materials, it is possible to achieve temperature up to  $500^{\circ} \text{C}$ .

In case the application requires ball screws able to work at higher temperature then the above mentioned, you are kindly asked to contact SERVOMECH Engineering Dpt.

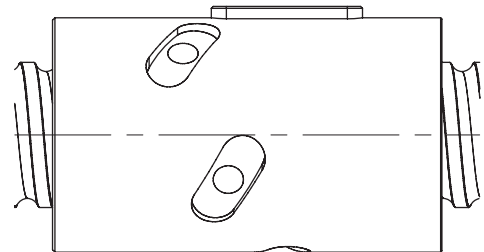
### 3.6 Ball nut types

Ball nut types, designed and manufactured by SERVOMECH, are shown below.



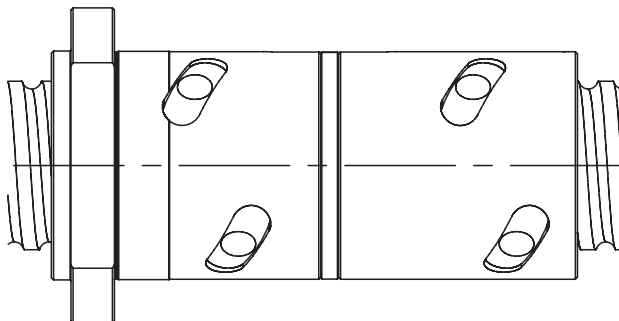
**SFN** single nut with flange,  
with or without backlash

**SFNP** single nut with flange,  
with preload obtained by *shift*

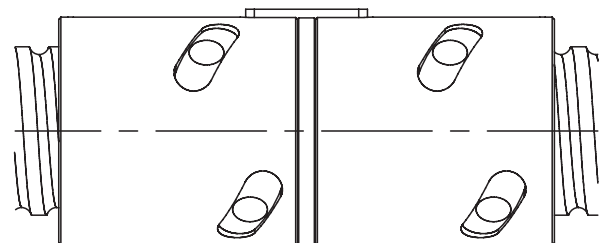


**SCN** single cylindrical nut,  
with or without backlash

**SCNP** single cylindrical nut,  
with preload obtained by *shift*



**DFN** double nut with flange



**DCN** double cylindrical nut



### 3.7 QUESTIONARY

page 1 of 2

#### CUSTOMER

Company:	Address:
Reference person:	Position of reference person:
Phone:	Fax:
E-mail:	Date:

#### APPLICATION DATA

Type of application	<input type="checkbox"/> transport	<input type="checkbox"/> positioning
DYNAMIC axial load	Rotating speed	Time
Max. N	rpm	%
Nominal N	rpm	%
Min. N	rpm	%
		100 % TOTAL
STATIC axial load: N		
RADIAL load: N		
Direction of load:	<input type="checkbox"/> unidirectional	<input type="checkbox"/> bidirectional
Vibrations / shocks:	<input type="checkbox"/> light	<input type="checkbox"/> medium <input type="checkbox"/> strong
Rotating part:	<input type="checkbox"/> threaded shaft	<input type="checkbox"/> nut
Mounting position:	<input type="checkbox"/> horizontal	<input type="checkbox"/> vertical <input type="checkbox"/> at angle of ____ °
Further details:	<input type="checkbox"/> axial tensioning	<input type="checkbox"/> balanced axis
End supports:	<input type="checkbox"/> constrained beam	<input type="checkbox"/> supported cantilever beam
	<input type="checkbox"/> simple supported beam	<input type="checkbox"/> cantilever beam
Working temperature:	°C	
Lubrication:	<input type="checkbox"/> oil lubricated	<input type="checkbox"/> grease lubricated Lubricant
Environment:	(type)	
	<input type="checkbox"/> clean	<input type="checkbox"/> normal <input type="checkbox"/> contaminated

specific protection required:

specific coating required:

specific material required:

Life REQUIRED	hours of work, with above described working cycle
	revolutions
	m of travel, subjected to above indicated load

### 3.7 QUESTIONARY

page 2 of 2

#### TECHNICAL FEATURES

	Nominal diameter	mm	Lead	mm	
	Thread helix		<input type="checkbox"/> right-hand	<input type="checkbox"/> left-hand	
SHAFT	Length		thread:	mm	
			total:	mm	
	Accuracy grade		<input type="checkbox"/> in accordance to ISO 3408	(specify)	
			<input type="checkbox"/> other	(specify)	
BALL NUT	Form		<input type="checkbox"/> with flange	<input type="checkbox"/> cylindric	
	Type		<input type="checkbox"/> single	<input type="checkbox"/> with backlash	
				<input type="checkbox"/> preloaded ( <i>shift</i> )	
				Preload force	N
			Preload torque	Nm	
		<input type="checkbox"/> double	Preload force	N	
			Preload torque	Nm	
	Design		<input type="checkbox"/> in accordance to DIN 69051		
			<input type="checkbox"/> personalized		
			<input type="checkbox"/> SERVOMECH		
		Dynamic load, $C_a$ :	N	Static load, $C_{0a}$ :	N
		Axial stiffness	N/ $\mu$ m		

#### ENQUIRY

Purpose	<input type="checkbox"/> new project	<input type="checkbox"/> existing application	<input type="checkbox"/> spare part
Prototype required	<input type="checkbox"/> no	<input type="checkbox"/> yes	(q.ty: )
Serial production	per month:		
	per year:		
	batches of		
Delivery required	weeks		

#### TECHNICAL DOCUMENTS SUPPLIED WITH ENQUIRY

Technical drawings	<input type="checkbox"/> (indicate number: )
Assembly drawings	<input type="checkbox"/> (indicate number: )
Application documents	<input type="checkbox"/>

### 3.8 Designation

	BS	-	40	-	10	-	1R	-	800	-	1 100	-	G5	-	C	-	2EM	-	SFN	-	D	-	5
NOMINAL DIAMETER																							
NOMINAL LEAD																							
Nr of THREAD STARTS and THREAD HELIX HANDING																							
R - right-hand L - left-hand																							
THREAD LENGTH																							
TOTAL SHAFT LENGTH																							
STANDARD TOLERANCE GRADE																							
THREAD MACHINING																							
R - rolling C - whirling																							
SHAFT ENDS MACHINING																							
1EM - only one shaft end machined 2EM - both shaft ends machined ENM - shaft ends not machined																							
BALL NUT TYPE																							
SFN - single nut with flange SFNP - single preloaded nut with flange DFN - double nut with flange SCN - single cylindrical nut SCNP - single cylindrical preloaded nut DCN - double cylindrical nut																							
BALL NUT DESIGN																							
C - nut at customer's drawing D - nut in accordance with DIN 69051 S - nut at SERVOMECH's drawing																							
Nr of LOADED TURNS																							



**SERVOMECH S.p.A.**

Via M. Calari, 1 - 40011 Anzola dell'Emilia (Bologna) - ITALY

Tel. +39.051.6501711 - Fax +39.051.734574 - [info@servomech.com](mailto:info@servomech.com)

[www.servomech.com](http://www.servomech.com)

