# Recommendations for selection

Only basic selection parameters are included. To make the very best selection of a ball screw, the designer should specify such critical parameters as the load profile, the linear or rotational speed, the rates of acceleration and deceleration, the cycle rate, the environment, the required life, the lead accuracy, the stiffness, and any other special requirement. If in doubt, please consult an SKF ball screw specialist before placing an order.

## Basic dynamic load rating ($C_a$)

The dynamic rating is used to compute the fatigue life of ball screws. It is the axial load constant in magnitude and direction, and acting centrally under which the nominal life (as defined by ISO) reaches one million revolutions.

#### Nominal fatigue life $L_{10}$

The nominal life of a ball screw is the number of revolutions (or the number of operating hours at a given constant speed) which the ball screw is capable of enduring before the first sign of fatigue (flaking, spalling) occurs on one of the rolling surfaces.

It is however evident from both laboratory tests and practical experience that seemingly identical ball screws operating under identical conditions have different lives, hence the notion of **nominal life**. It is, in accordance with ISO definition, the life achieved or exceeded by 90% of a sufficiently large group of apparently identical ball screws, working in identical conditions (alignment, axial and centrally applied load, speed, acceleration, lubrication, temperature and cleanliness).

#### Service life

The actual life achieved by a specific ball screw before it fails is known as “service life“. Failure is generally by wear, not by fatigue (flaking or spalling) ; wear of the recirculation system, corrosion, contamination, and, more generally, by loss of the functional characteristics required by the application. Experience acquired with similar applications will help to select the proper screw to obtain the required service life. One must also take into account structural requirements such as the strength of screw ends and nut attachments, due to the loads applied on these elements in service.

#### Equivalent dynamic loads

The loads acting on the screw can be calculated according to the laws of mechanics if the external forces (e.g. power transmission, work, rotary and linear inertia forces) are known or can be calculated. It is necessary to calculate the equivalent dynamic load : this load is defined as that hypothetical load, constant in magnitude and direction, acting axially and centrally on the screw which, if applied, would have the same influence on the screw life as the actual loads to which the screw is subjected.
Radial and moment loads must be taken by linear bearing systems. It is extremely important to resolve these problems **at the earliest conceptual stage**. These forces are detrimental to the life and the expected performance of the screw.

***Fluctuating load***
When the load fluctuates during the working cycle, it is necessary to calculate the equivalent dynamic load : this load is defined as that hypothetical load, constant in magnitude and direction, acting axially and centrally on the screw which, if applied, would have the same influence on the screw life as the actual loads to which the screw is subjected.
Additional loads due, for example to misalignment, uneven loading, shocks, and so on, must be taken in account. Their influence on the nominal life of the screw is generally taken care of, consult SKF for advice.

## Static load carrying capacity ($C_{oa}$)

Ball screws should be selected on the basis of the basic static load rating $C_{oa}$ instead of on bearing life when they are submitted to continuous or intermittent shock loads, while stationary or rotating at very low speed for short duration. The permissible load is determined by the permanent deformation caused by the load acting at the contact points. It is defined by ISO standards as the purely axially and centrally applied static load which will create, by calculation, a total (rolling element + thread surface) permanent deformation equal to .0001 of the diameter of the rolling element.
A ball screw must be selected by its basic static load rating which must be, at least, equal to the product of the maximum axial static load applied and a safety factor “$s_o$“. The safety factor is selected in relation with past experience of similar applications and requirements of running smoothness and noise level (1).

(1) SKF can help you to define this value in relation with the actual conditions of service.

## Critical rotating speed for screw shafts

The shaft is equated to a cylinder, the diameter of which is the root diameter of the thread. The formulas use a parameter the value of which is dictated by the mounting of the screw shaft (whether it is simply supported or fixed). As a rule the nut is not considered as a support of the screw shaft. Because of the potential inaccuracies in the mounting of the screw assembly, a safety factor of .80 is applied to the calculated critical speeds.
Calculations which consider the nut as a support of the shaft, or reduce the safety factor, require practical tests and possibly an optimization of the design (1).

## Permissible speed limit

The permissible speed limit is that speed which a screw cannot reliably exceed at any time. It is generally the limiting speed of the recirculation system in the nut. It is expressed as the product of the rpm and the nominal diameter of the screw shaft (in mm). The speed limits quoted in this catalogue are **the maximum speeds that may be applied through very short periods** and in optimized running conditions of alignment, light external load and preload with monitored lubrication. Running a screw continuously at the permissible speed limit may lead to a reduction of the calculated life of the nut mechanism.

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High speed associated with high load requires a large input torque and yields a relatively short nominal life (1).
In the case of high acceleration and deceleration, it is recommended to either work under a nominal external load or to apply a light preload to the nut to avoid internal sliding during reversal. The value of preload of screws submitted to high velocity must be that preload which ensures that the rolling elements do not slide (1).
Too high a preload will create unacceptable increases of the internal temperature.

***The lubrication*** of screws rotating at high speed must be properly considered in quantity and quality. The volume, spread and frequency of the application of the lubricant (oil or grease) must be properly selected and monitored). At high speed the lubricant spread on the surface of the screw shaft may be thrown off by centrifugal forces. It is important to monitor this phenomenon during the first run at high speed and possibly adapt the frequency of re-lubrication or the flow of lubricant, or select a lubricant with a different viscosity. Monitoring the steady temperature reached by the nut permits the frequency of re-lubrication or the oil flow rate to be optimized.

## Efficiency and back-driving

The performance of a screw is mainly dependant on the geometry of the contact surfaces and their finish as well as the helix angle of the thread. It is, also, dependant on the working conditions of the screw (load, speed, lubrication, preload, alignment, etc…).
The ***«direct efficiency»*** is used to define the input torque required to transform the rotation of one member into the translation of the other. Conversely, the ***«indirect efficiency»*** is used to define the axial load required to transform the translation of one member into the rotation of the other one. It is used, also, to define the braking torque required to prevent that rotation.

It is safe to consider that these screws are reversible or back-driveable under almost all circumstances.
It is therefore necessary to design a brake mechanism if backdriving is to be avoided (gear reducers or brake).

***Preload torque :***
Internally preloaded screws exhibit a torque due to this preload. This persists even when they are not externally loaded. Preload torque is measured when assembly is lubricated with ISO grade 64 oil.

***Starting torque :***
This is defined as the torque needed to overcome the following to start rotation :

a) the total inertia of all moving parts accelerated by the energy source (including rotation and linear movement).

b) the internal friction of the screw/nut assembly, bearing and associated guiding devices.

In general, torque to overcome inertia (a) is greater than friction torque (b).
The coefficient of friction of the high efficiency screw when starting $\mu_s$ is estimated at up to double the dynamic coefficient $\mu$, under normal conditions of use.

## Axial play and preload

Preloaded nuts are subject to much less elastic deformation than non-preloaded nuts. Therefore they should be used whenever the accuracy of positioning under load is important.

Preload is that force applied to a set of two half nuts to either press them together or push them apart with the purpose of eliminating backlash or increasing the rigidity or stiffness of the assembly. The preload is defined by the value of the preload torque (see under that heading in the previous paragrah). The torque depends on the type of nut and on the mode of preload (elastic or rigid).

(1) SKF can help you to define this value in relation with the actual conditions of service.

## Static axial stiffness of a complete assembly

It is the ratio of the external axial load applied to the system and the axial displacement of the face of the nut in relation with the fixed (anchored) end of the screw shaft. The inverse of the ***rigidity of the total system*** is equal to the sum of all the inverses of the rigidity of each of the components (screw shaft, nut as mounted on the shaft, supporting bearing, supporting housings, etc…). Because of this, the rigidity of the total system is always less than the smallest individual rigidity.

***Nut rigidity***

When a preload is applied to a nut, firstly, the internal play is eliminated, then, the Hertzian elastic deformation increases as the preload is applied so that the overall rigidity increases. The theoretical deformation does not take into account machining inaccuracies, actual sharing of the load between the different contact surfaces, the elasticity of the nut and of the screw shaft. The practical stiffness values given in the catalogue are lower than the theoretical values for this reason. The rigidity values given in the SKF ball screw catalogue are individual practical values for the assembled nut. They are determined by SKF based on the value of the selected basic preload and an external load equal to twice this preload.

***Elastic deformation of screw shaft***

This deformation is proportional to its length and inversely proportional to the square of the root diameter. According to the relative importance of the screw deformation (see rigidity of the total system), too large an increase in the preload of the nut and supporting bearings yields a limited increase of rigidity and notably increases the preload torque and therefore the running temperature. Consequently, the preload stated in the catalogue for each dimension is optimum and should not be increased.

## Screw shaft buckling

The column loading of the screw shaft must be checked when it is submitted to compression loading (whether dynamically or statically). The maximum permissible compressive load is calculated using the Euler formulas. It is then multiplied by a safety factor of 3 to 5, depending on the application.

The type of end mounting of the shaft is critical to select the proper coefficients to be used in the Euler formulas.
When the screw shaft comprises a single diameter, the root diameter is used for the calculation. When the screw comprises different sections with various diameters, calculations becomes more complex (1).

## Manufacturing precision

Generally speaking, the precision indication given in the designation defines the lead precisions see page 9 - lead precision according to ISO - (ex. G5 - G7…).
Parameters other than lead precision correspond to our internal standards (generally based on ISO «class 7»).
If you require special tolerances (for example class 5) please specify when requesting a quotation or ordering.

## Materials and heat treatments

Standard screw shafts are machined from steel which is surface hardened by induction (42CrMo4-NF EN10083-1 for diameters > 20 mm and 2C45 for diameters < 20 mm).

Standard nuts are machined in steel which is through hardened (100 Cr6 - NFA 35.565 or equivalent for diameters > 20 mm and carbon steel for diameters < 20 mm).
Hardness of the contact surfaces is 56-60 HRc, depending on diameter, for standard screws.
Most assemblies made of stainless material have a surface hardness in the range 42 to 58 HRc, depending on the type. The load rating of the catalogue are given only for standard screws.

## Number of circuits of balls

A nut is defined by the number of ball turns which support the load. The number is changing, according to the product and the combination diameter/lead.
It is defined by the number of circuits and their type.

## Re-circulation inserts

The standard products have been fitted with composite ball re-circulation inserts.
System performance is improved because of the smoother ball re-circulation. This results from the improved precision of the moulded insert when compared to the former steel insert. If the product is used in severe applications, or the insert is used to prevent collapse, a steel version is available. In such cases, the specifier should consult SKF Linear Motion to obtain the optimum solution.

## Working environment

Our products have not been developed for use in an explosive atmosphere, consequently we cannot take any responsability for the use in this field.

NOTE : 42 CrMo, an AFNOR reference is similar to AISI 4140; 100Cr6 is similar to AISI 52100.

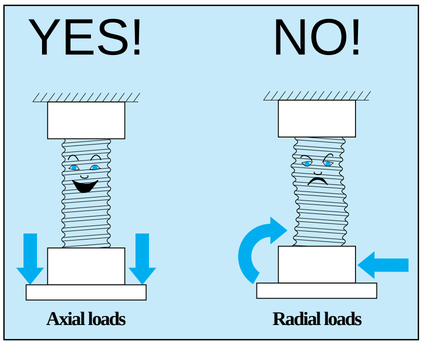
(1) SKF can help you to define this value in relation with the actual conditions of service.

# Recommended assembly procedure

Ball screws are precision components and should be handled with care to avoid shocks. When stored out of the shipping crate they must lie on wooden or plastic vee blocks and should not be allowed to sag.
Screw assemblies are shipped, wrapped in a heavy gauge plastic tube which protects them from foreign material and possible pollution. They should stay wrapped until they are used.

**Radial and moment loads**

Any radial or moment load on the nut will overload some of the contact surfaces, thus significantly reducing life.




**Alignment**

SKF linear guidance components should be used to ensure correct alignment and avoid non-axial loading. The parallelism of the screw shaft with the guiding devices must be checked. If external linear guidance proves impractical, we suggest mounting the nut on trunnions or gimbals and the screw shaft in self-aligning bearings.
Mounting the screw in tension helps align it properly and eliminates buckling.

**Lubrication**

Good lubrication is essential for the proper functioning of the screw and for its long term reliability (1).
Before shipping, the screw is coated with a protective fluid that dries to a film. ***This protective film is not a lubricant.***
Depending on the selected lubricant, it may be necessary to remove this film before applying the lubricant (there may be a risk of non-compatibility).
If this operation is performed in a potentially polluted atmosphere it is highly recommended to proceed with a thorough cleaning of the assembly.

**Designing the screw shaft ends**

Generally speaking, when the ends of the screw shaft are specified by the customer's engineering personnel, it is their responsability to check the strength of these ends. However, we offer in pages 15 and 17 of this catalogue, a choice of standard machined ends. As far as possible, we recommend their use.
Whatever your choice may be, please keep in mind that no dimension on the shaft ends can exceed $d_o$ (otherwise traces of the root of thread will appear or the shaft must be made by joining 2 pieces).
A minimum shoulder should be sufficient to maintain the internal bearing.

**Operating temperature**

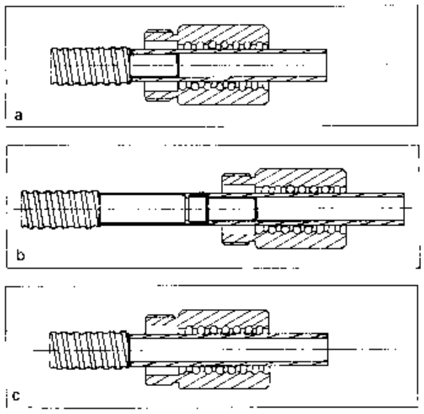
Screws made from standard steel (see page 7) and operating under normal loads can sustain temperatures in the range minus 20°C plus 110° Celsius.
Between 110° Celsius and 130° Celsius, SKF must be notified so that it adapts the annealing procedure and checks that the application can be successful with a hardness below the standard minimum value (see page 7).

Above 130° Celsius, steels adapted to the temperature of the application should be selected (100Cr6, special steel, etc…).
Consult SKF for advice.
Operating at high temperature will lower the hardness of the steel, alter the accuracy of the thread and may increase the oxidability of the materials or change lubricant properties.

**Separating the nut from the screw shaft**


Never screw the nut off the shaft without a mandrel to prevent the balls coming out.

1. Remove the retaining strap
2. Hold the sleeve against the ball track (a). If the sleeve does not go over the diameter next to the ball track, adhesive tape can be used (b) or the sleeve held against the unmachined end (c).




3. Without forcing, engage the nut in the screw thread.

**Starting-up the screw**

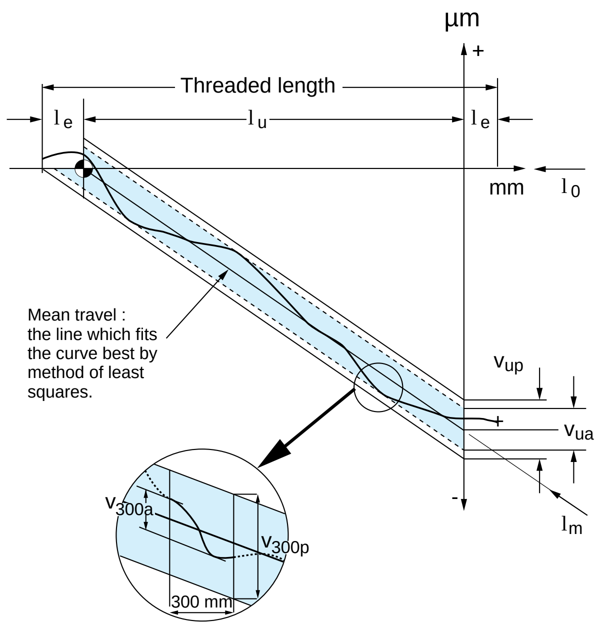
After the assembly has been cleaned, mounted and lubricated, it is recommended that the nut is allowed to make several full strokes at low speed ; to check the proper positioning of the limit switches or reversing mechanism before applying the full load and the full speed.

NOTE :
Intructions for most operations like mounting a nut on a screw shaft, a wiper on a nut, etc… are available in separate sheets delivered with the product : please refer to them.

# Lead precision according to ISO

Lead precision is measured at 20°C on the useful stroke $l_u$ , which is the threaded length decreased, at each end, by the length $l_e$ equal to the screw shaft diameter.

| | G5 | | G7 | | G9 | |
|---|---|---|---|---|---|---|
| $V_{300p}$, µm | 23 | | 35 | | 87 | |
| $l_u$ mm | $e_p$ µm | $V_{up}$ | $e_p$ | $V_{up}$ | $e_p$ | $V_{up}$ |
| 0 - 315 | 23 | 23 | 52 | 35 | 130 | 87 |
| (315) - 400 | 25 | 25 | 57 | 40 | 140 | 100 |
| (400) - 500 | 27 | 26 | 63 | 46 | 155 | 115 |
| (500) - 630 | 32 | 29 | 70 | 52 | 175 | 130 |
| (630) - 800 | 36 | 31 | 80 | 57 | 200 | 140 |
| (800) - 1000 | 40 | 34 | 90 | 63 | 230 | 155 |
| (1000) - 1250 | 47 | 39 | 105 | 70 | 260 | 175 |
| (1250) - 1600 | 55 | 44 | 125 | 80 | 310 | 200 |
| (1600) - 2000 | 65 | 51 | 150 | 90 | 370 | 230 |
| (2000) - 2500 | 78 | 59 | 175 | 105 | 440 | 260 |
| (2500) - 3150 | 96 | 69 | 210 | 125 | 530 | 310 |
| (3150) - 4000 | 115 | 82 | 260 | 150 | 640 | 370 |
| (4000) - 5000 | 140 | 99 | 320 | 175 | 790 | 440 |
| (5000) - 6000 | 170 | 119 | 390 | 210 | 960 | 530 |




$l_u$ = useful travel

$l_e$ = excess travel (no lead precision required)

$l_o$ = nominal travel

$l_s$ = specified travel

c = travel compensation (difference between $l_s$ and $l_o$ to be defined by the customer, for instance to compensate an expansion)

$e_p$ = tolerance over the specified travel

V = travel variation (or permissible band width)

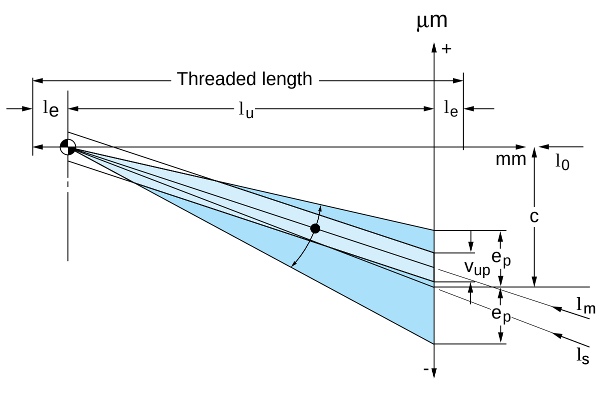
$V_{300p}$ = maximum permitted travel variation over 300 mm

$V_{up}$ = maximum permitted travel variation over the useful travel $l_u$

$V_{300a}$ = measured travel variation over 300 mm

$V_{ua}$ = measured travel variation over the useful travel

**Case with value of c specified by the customer.**




**Case with c = 0 = standard version in case of no value given by the customer.**