Design fundamentals

The following are the relevant calculations which underly screw design and safe operation.

For detailed information on ball screw design, please refer to DIN 69051.

«Suitability test» rotational speed characteristics

When selecting a ball screw it is important to first ensure that the correct nut design for the ball return system required to support the maximum rotational speed demanded by the application is used (independet of the screw length).

The maximum rotational speed is based on the system's rotational speed characteristics and the outside screw diameter:

$$n_{max} = \frac{\text{rotational speed characteristic}}{d_1} \; [min^{-1}]$$

n_{max} = maximum rotational speed [min^{-1}]

Rotational speed characteristics [–] for

- single-thread ball return: 60 000 (Carry «…I» types)
- tube type ball return: 80 000 (Carry «…R» types)
- end cap ball return: 80 000 (Carry Speed-line «…E» types)

d_1 = outside screw diameter [mm]

Calculations at dynamic load:

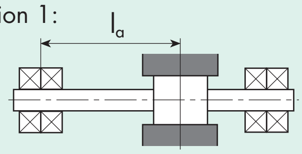
Critical rotational speed n_{per}

Permissible rotational speeds must differ substantially from the screw's own frequency.

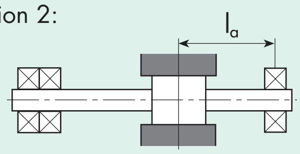
$$n_{per} = K_D \cdot 10^6 \cdot \frac{d_2}{l_a^2} \cdot S_n \; [min^{-1}]$$

n_{per} = permissible rotational speed [min^{-1}]
K_D = characteristic constant as a function of bearing configuration [–]
→ see below
d_2 = core diameter [mm]
l_a = bearing distances [mm]
→ see below
(always include maximum allowable l_a in calculation)
S_n = safety factor
usually S_n = 0.5…0.8 [–]

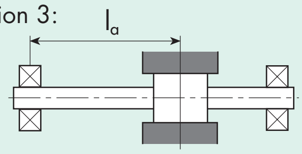
Configuration 1:
fixed – fixed
K_D = 276



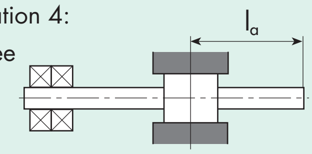
Configuration 2:
fixed – simple
K_D = 190



Configuration 3:
simple – simple
K_D = 122



Configuration 4:
fixed – free
K_D = 43



Nominal service life L_{10} or L_h

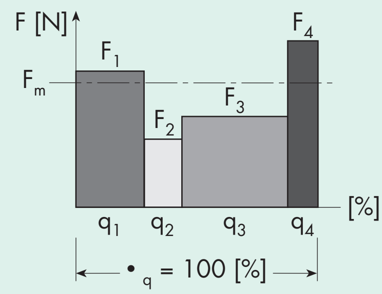
$$L_{10} = \left(\frac{C_{dyn}}{F_m}\right)^3 \cdot 10^6 \; [R]$$

$$L_h = \frac{L_{10}}{n_m \cdot 60} \; [h]$$

L_{10} = service life in revolutions [R]
L_h = service life in hours [h]
C_{dyn} = dynamic load [N]
F_m = average axial load [N]
$F_{1…n}$ = load per cycle unit [N]
n_m = average rotational speed [min^{-1}]
$n_{1…n}$ = rotational speed per cycle unit [min^{-1}]
$q_{1…n}$ = cycles [%]
100 = $\sum_q$ (sum of cycles $q_{1…n}$) [%]

Average axial load F_m
at constant rotational speed n_{const} and dynamic load C_{dyn}

$$F_m = \sqrt[3]{F_1^3 \frac{q_1}{100} + F_2^3 \frac{q_2}{100} + F_3^3 \frac{q_3}{100} + \ldots} \; [N]$$



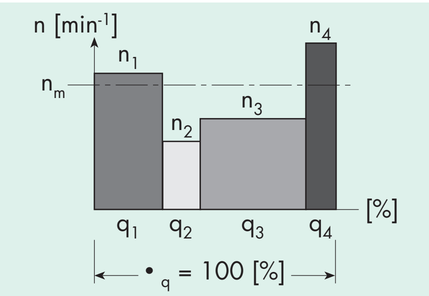
$$\rightarrow L_{10} = \left(\frac{C_{dyn}}{F_m}\right)^3 \cdot 10^6 \; [R]$$

$$\rightarrow L_h = \frac{L_{10}}{n_{const} \cdot 60} \; [h]$$

Calculations at dynamic load (continuation)**:**

Average rotational speed n_m
at constant load F_{const}
and variable rotational speeds $n_{1...n}$

$$n_m = n_1 \frac{q_1}{100} + n_2 \frac{q_2}{100} + n_3 \frac{q_3}{100} + \ldots \ [min^{-1}]$$



$$\rightarrow L_{10} = \left(\frac{C_{dyn}}{F_{const}}\right)^3 \cdot 10^6 \ [R]$$

$$\rightarrow L_h = \frac{L_{10}}{n_m \cdot 60} \ [h]$$

Average axial load F_m
at constant rotational speeds n_{const}
and dynamic load C_{dyn}

$$F_m = \sqrt[3]{F_1^3 \frac{q_1}{100} + F_2^3 \frac{q_2}{100} + F_3^3 \frac{q_3}{100} + \ldots} \ [N]$$

$$n_m = n_1 \frac{q_1}{100} + n_2 \frac{q_2}{100} + n_3 \frac{q_3}{100} + \ldots \ [min^{-1}]$$

$$\rightarrow L_{10} = \left(\frac{C_{dyn}}{F_m}\right)^3 \cdot 10^6 \ [R]$$

$$\rightarrow L_h = \frac{L_{10}}{n_m \cdot 60} \ [h]$$

Efficiency η (theoretical)
Depends upon the type of power transmission.

Case 1: torque → linear movement

$$\eta \approx \frac{\tan \alpha}{\tan (\alpha + \rho)} \ [–]$$

Case 2: axial force → torque

$$\eta' \approx \frac{\tan (\alpha - \rho)}{\tan \alpha} \ [–]$$

whereby

$$\tan \alpha \approx \frac{p}{d_0 \cdot \pi} \ [–]$$

η = efficiency [%]
η' = corrected efficiency [%]
p = pitch [mm]
d_0 = nominal screw diameter [mm]
ρ = angle of friction [°]
→ ρ = 0.30...0.60°

Efficiency η_p (practical)
The efficiency η for Carry ball screws is better than 0.9.

Driving torque M
Depends upon the type of power transmission.

Case 1: torque → linear movement

$$M_a = \frac{F_a \cdot p}{2000 \cdot \pi \cdot \eta} \ [Nm]$$

Case 2: axial force → torque

$$M_e = \frac{F_a \cdot p \cdot \eta'}{2000 \cdot \pi} \ [Nm]$$

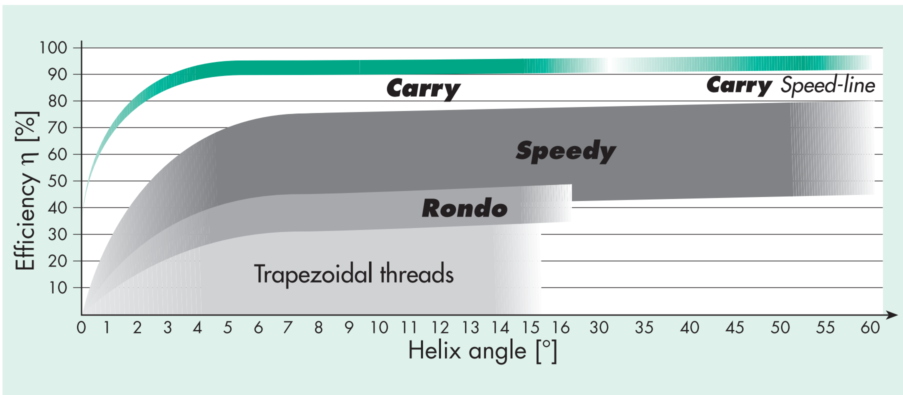
M_a = input torque [Nm], case 1
M_e = output torque [Nm], case 2
F_a = axial force [N]

Input performance P

$$P = \frac{M_a \cdot n}{9550} \ [kW]$$

P = input performance [kW]
n = rotational speed [min⁻¹]

A safety margin of 20 % is recommended when selecting drives.



Calculations at static load:

Permissible maximum load $F_{per.}$

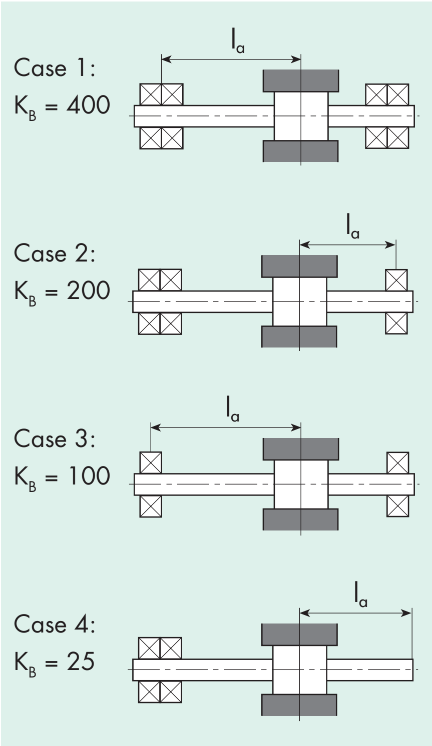
$$F_{per.} = \frac{C_{stat}}{f_s} \text{ [N]}$$

C_{stat} = static load [N]
f_s = operating coefficient
→ normal operation: 1…**2** [–]
→ shock load: 2…3 [–]

Permissible buckling force F_B

$$F_B = \frac{K_B}{S_B} \cdot \frac{d_2^{\,4}}{l_F^{\,2}} \cdot 10^3 \text{ [N]}$$

K_B = characteristic constant of load (depends on design) [–]
→ see below
d_2 = nominal screw diameter [mm]
l_F = force-transferring length [mm]
S_B = buckling safety factor
→ gen. S_B = 2…4 [–]



The following are the relevant calculations which underly high-helix screw design and safe operation.

Calculations at dynamic load:

Critical rotational speed n_{per}

Permissible rotational speeds must differ substantially from the screw's own frequency.

$$n_{per} = K_D \cdot 10^6 \cdot \frac{d_2}{l_a^2} \cdot S_n \ [min^{-1}]$$

n_{per} = permissible rotational speed [min^{-1}]
K_D = characteristic constant as a function of bearing configuration
→ see below
d_2 = core diameter [mm]
l_a = bearing distances [mm]
→ see opposite
(always include maximum allowable l_a in calculation)
S_n = safety factor
usually S_n = 0.5…0.8 [–]

Configuration 1:
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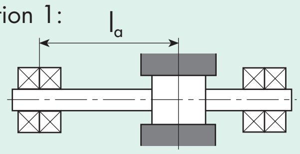
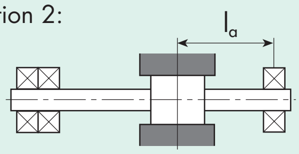
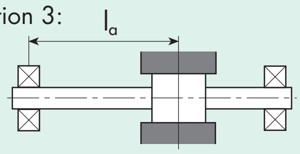
K_D = 276



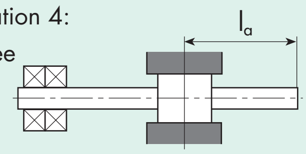
Configuration 2:
fixed –
simple
K_D = 190



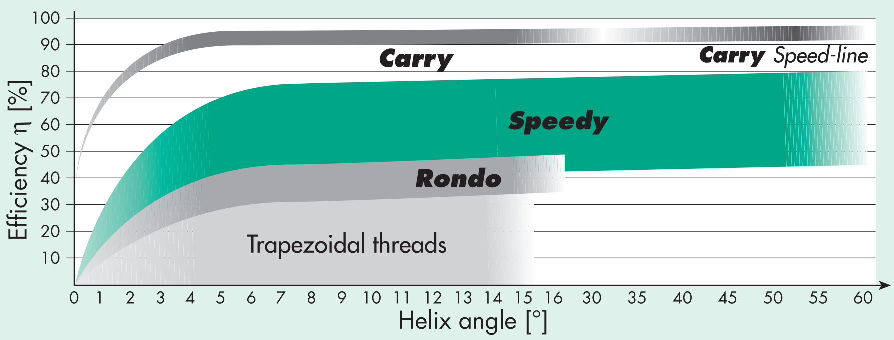
Configuration 3:
simple –
simple
K_D = 122



Configuration 4:
fixed – free
K_D = 43



Efficiency η_p (practical)
The efficiency η depends on the helix angle and reaches values from ~0.5 to 0.75.



Basic calculations

Driving torque M

Depends upon the type of power transmission.

Case 1: torque → linear movement

$$M_a = \frac{F_a \cdot p}{2000 \cdot \pi \cdot \eta} \text{ [Nm]}$$

Case 2: axial force → torque

$$M_e = \frac{F_a \cdot p \cdot \eta'}{2000 \cdot \pi} \text{ [Nm]}$$

M_a = input torque [Nm]
M_e = output torque [Nm]
F_a = axial force [N]
η = efficiency [%]
η' = corrected efficiency [%]
p = pitch [mm]

Input performance P

$$P = \frac{M_a \cdot n}{9550} \text{ [kW]}$$

P = input performance [kW]
n = rotational speed [min^{-1}]

A safety margin of 20 % is recommended when selecting drives.

Maximum authorized load depending on speed

$$F_{per.} = C_0 \cdot f_L \text{ [N]}$$

C_0 = static load rate [N]
f_L = load factor [–] for POM-C nuts

circumferential speed v_C [m/min]	load factor f_L [–]
5	0.95
10	0.75
20	0.45
30	0.37
40	0.12
50	0.08

Example

Parameters:
Speedy 10/50 with non-preloaded POM-C nut, d_0 = 10 mm, p = 50 mm and C_0 = 1250 N;
required moving speed v_S = 200 mm/sec.

We need to find: $F_{per.}$

We calculate n [min^{-1}],

$$n = \frac{v_S \text{ [mm/sec]} \cdot 60}{p \text{ [mm]}}$$

$$= \frac{200 \cdot 60}{50} = 240 \text{ min}^{-1}$$

circumferential speed v_C [m/min]

$$v_C = \frac{d_0 \text{ [mm]} \cdot \pi \cdot n \text{ [min}^{-1}\text{]}}{1000}$$

$$= \frac{10 \cdot \pi \cdot 240}{1000} = 7.53 \text{ m/min}$$

and find load factor f_L in above table:

f_L at v_C of 7.53 m/min ≈ 0.85 [–]

It follows:

$F_{per.} = C_0 \cdot f_L = 1250 \cdot 0.85 = 1062.5$ N

In other words, the maximum load for a Speedy 10/50 at v_S = 200 mm/sec. (→ n = 240 min^{-1}) is 1060 N.