High-performance lubricant for Linear Motion Systems
(not released for the USA)

Product description Dynalub 510

Dynalub 510 is an NLGI grade 2 lithium-based high-performance grease specially developed for linear motion systems. It is notable for offering excellent water resistance and protection against corrosion, and is suited for use at temperatures of between –20 °C and 80 °C.

Applications
Under conventional environmental conditions this ground-fiber, homogeneous grease is ideally suitable for the lubrication of linear elements:
- for loads of up to 0.5 \( C_{\text{dyn}} \)
- also for short-stroke applications \( \geq 1 \) (mm)

Technical data

For further details, see
"Safety Data Sheet Dynalub 510"
R310EN 2052 (2004.04)

<table>
<thead>
<tr>
<th>Materialnummer</th>
<th>Packing unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>R3416 037 00</td>
<td>1 x 400 g</td>
</tr>
</tbody>
</table>

Chemical composition
Mineral oil, special lithium soap, agents

<table>
<thead>
<tr>
<th>Designation</th>
<th>KP2K-20</th>
</tr>
</thead>
<tbody>
<tr>
<td>Appearance</td>
<td>Light-brown/beige, ground-fiber</td>
</tr>
<tr>
<td>Service temperature range</td>
<td>–20 °C to +80 °C</td>
</tr>
</tbody>
</table>

NLGI grade
2

Worked penetration
265-295 1/10 mm

Water resistance
0-60, 1-90

Melting point in °C
> 165

Flash point in °C
> 200 – base oil

Basic oil viscosity
100 mm\(^2\)/s 40 °C

Density at +25°C approx. 0.92 g/cm\(^3\)

Flow pressure at –20°C < 1400 hPa

EMCOR test 0/0

Copper corrosion
2 (24 h/100 °C)

Four ball tester welding load > 2000 N

Four ball tester impression diameter 0.93 (400 N, 1 h)

Shelf life in original container 2 years

Materialnummer Packing unit
R3416 037 00 1 x 400 g

Product description Dynalub 520

Dynalub 520 is an NLGI grade 00 lithium-based high-performance grease specially developed for linear motion systems. It is notable for offering excellent water resistance and protection against corrosion, and is suited for use at temperatures of between –20 °C and 80 °C.

Applications
Under conventional environmental conditions this ground-fiber, homogeneous grease is ideally suited for the lubrication of miniature linear elements and for use in centralized lubrication systems.

Technical data

For further details, see
"Safety Data Sheet Dynalub 520"
R310EN 2053 (2004.04)

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<td>1 x 400 g</td>
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Chemical composition
Mineral oil, special lithium soap, agents

<table>
<thead>
<tr>
<th>Designation</th>
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<tbody>
<tr>
<td>Appearance</td>
<td>Light-brown/beige, ground-fiber</td>
</tr>
<tr>
<td>Service temperature range</td>
<td>–20 °C to +80 °C</td>
</tr>
</tbody>
</table>

NLGI grade
00

Worked penetration
400-430 1/10 mm

Water resistance
1-90

Melting point in °C
> 160

Flash point in °C
> 200 – base oil

Basic oil viscosity
100 mm\(^2\)/s 40 °C

Density at +25°C approx. 0.92 g/cm\(^3\)

Flow pressure at –20°C < 700 hPa

EMCOR test 0

Copper corrosion
0-1 (24 h/100 °C)

Four ball tester welding load 1800 N

Four ball tester impression diameter 0.80 (400 N, 1 h)

Shelf life in original container 2 years

Materialnummer Packing unit
R3416 043 00 1 x 400 g
### Design Calculations

#### Average speed and average load

- Where the speed fluctuates, the average speed $n_m$ is calculated as follows:

$$n_m = \frac{[n_1] \cdot q_{t1} + [n_2] \cdot q_{t2} + \ldots + [n_n] \cdot q_{tn}}{100\%}$$

where $n_1, n_2, \ldots, n_n$ are speeds in phases 1 ... n (min$^{-1}$), $n_m$ is average speed (min$^{-1}$), and $q_{t1}, q_{t2}, \ldots, q_{tn}$ are discrete time step in phases 1 ... n (%).

The following applies for the effective equivalent bearing load:

$$F > 2.8 \times X_{pr} \cdot C$$

$$F_{eff \_n} = \left| F_{n} \right|$$

$$F \leq 2.8 \times X_{pr} \cdot C$$

$$F_{eff \_n} = \left[ \frac{\left| F_{n} \right|}{2.8 \times X_{pr} \cdot C} + 1 \right]^{\frac{3}{2}} \times X_{pr} \cdot C$$

where $C$ is dynamic load rating (N), $F_{eff \_n}$ is effective equivalent axial load during phase n (N), $F_n$ is axial load during phase n (N), and $X_{pr}$ is preload factor (-).

- Where the load fluctuates and the speed is constant, the average load $F_m$ is calculated as follows:

$$F_m = \sqrt[3]{\frac{F_{\_eff \_1}^3 \cdot q_{t1}}{100\%} + \frac{F_{\_eff \_2}^3 \cdot q_{t2}}{100\%} + \ldots + \frac{F_{\_eff \_n}^3 \cdot q_{tn}}{100\%}}$$

where $F_{\_eff \_1}, F_{\_eff \_2}, \ldots F_{\_eff \_n}$ are effective equivalent axial load during phases 1 ... n (N), $F_m$ is equivalent dynamic axial load (N), and $q_{t1}, q_{t2}, \ldots, q_{tn}$ are discrete time step for $F_{\_eff \_1}, \ldots F_{\_eff \_n}$ (%).
Where both the load and the speed fluctuate, the average load $F_m$ is calculated as follows:

$$F_m = \sqrt[3]{\left(\frac{F_{\text{eff}1}}{n_{m1}}\cdot q_{t1}\right)^3 + \left(\frac{F_{\text{eff}2}}{n_{m2}}\cdot q_{t2}\right)^3 + \ldots + \left(\frac{F_{\text{eff}n}}{n_{mn}}\cdot q_{tn}\right)^3}$$

- $F_{\text{eff}1}, F_{\text{eff}2}, \ldots, F_{\text{eff}n}$ = effective equivalent axial load during phases 1 ... n (N)
- $F_m$ = equivalent dynamic axial load (N)
- $n_{m1}, n_{m2}, \ldots, n_{mn}$ = speeds during phases 1 ... n (min$^{-1}$)
- $q_{t1}, q_{t2}, \ldots, q_{tn}$ = discrete time step for $F_{\text{eff}1}, \ldots, F_{\text{eff}n}$ (%)

### Nominal life

**Service life in revolutions $L$**

$$L = \left(\frac{C}{F_m}\right)^3 \cdot 10^6 \quad \Rightarrow \quad C = F_m ^3 \left(\frac{L}{10^6}\right)^{\frac{1}{3}} \quad \Rightarrow \quad F_m = \frac{C}{L^{\frac{1}{3}}} \cdot 10^6$$

- $C$ = dynamic load rating (N)
- $F_m$ = equivalent dynamic axial load (N)
- $L$ = service life in revolutions

**Service life in hours $L_h$**

$$L_h = \frac{L}{n_{m} \cdot 60}$$

- $L_h$ = service life (h)
- $L$ = service life in revolutions
- $n_{m}$ = average speed (min$^{-1}$)

$$L_h_{\text{machine}} = L_h \cdot \frac{\text{DC}_{\text{machine}}}{\text{DC}_{\text{ball screw}}}$$

- $L_h_{\text{machine}}$ = nominal service life of the machine (h)
- $L_h$ = nominal service life of the ball screw drive (h)

### Drive torque and drive power

**Drive torque $M_{ta}$**

for conversion of rotary motion into linear motion:

$$M_{ta} = \frac{F_L \cdot P}{2000 \cdot \pi \cdot \eta} \quad F_L = \text{thrust force} \quad (N)$$

$$M_{ta} \leq M_p$$

- $F_L$ = thrust force (N)
- $M_p$ = maximum permissible drive torque (Nm)
- $M_{ta}$ = drive torque (Nm)
- $P$ = lead (mm)
- $\eta$ = mech. efficiency (approx. 0.9) (-)

**Transmitted torque $M_{te}$**

for conversion of linear motion into rotary motion:

$$M_{te} = \frac{F_L \cdot P \cdot \eta'}{2000 \cdot \pi} \quad F_L = \text{thrust force} \quad (N)$$

$$M_{te} \leq M_p$$

- $F_L$ = thrust force (N)
- $M_p$ = maximum permissible drive torque (Nm)
- $M_{te}$ = transmitted torque (Nm)
- $P$ = lead (mm)
- $\eta'$ = mech. efficiency ($\eta'$ approx. 0.8) (-)

The dynamic drag torque must be taken into account for preloaded nuts.

**Drive power $P_a$**

$$P_a = \frac{M_{ta} \cdot n}{9550} \quad M_{ta} = \text{drive torque} \quad (Nm)$$

$$n = \text{speed} \quad (\text{min}^{-1})$$

$$P_a = \text{drive power} \quad (kW)$$
Design Calculations

Calculation example
Service life

Operating conditions
The service life of the machine should be 40,000 operating hours with the ball screw operating 60% of the time.

Calculation procedure
Average speed \( n_m \)

\[
\begin{align*}
n_m &= \frac{6}{100} \cdot |10| + \frac{22}{100} \cdot |30| + \frac{47}{100} \cdot |100| + \frac{25}{100} \cdot |1000| \\
n_m &= 304 \text{ min}^{-1}
\end{align*}
\]

Average load \( F_m \) for variable load and variable speed

\[
F_m = \sqrt{\left( \frac{50000}{304} \right)^2 \left( \frac{6}{100} \right) + \left( \frac{25000}{304} \right)^2 \left( \frac{22}{100} \right) + \left( \frac{8000}{304} \right)^2 \left( \frac{47}{100} \right) + \left( \frac{2000}{304} \right)^2 \left( \frac{25}{100} \right)}
\]

\[
F_m = 8757 \text{ N}
\]

Required service life \( L \) (revolutions)
The service life \( L \) can be calculated by transposing the formulas 7 and 8:

\[
L = L_h \cdot n_m \cdot 60
\]

\[
L_h = L_{h, \text{machine}} \cdot \frac{DC_{\text{ball screw}}}{DC_{\text{machine}}}
\]

\[
L_h = 40000 \cdot \frac{60}{100} = 24000 \text{ h}
\]

\[
L = 24000 \cdot 304 \cdot 60
\]

\[
L = 437760000 \text{ revolutions}
\]

Basic dynamic load rating \( C \)

\[
C = 8757 \cdot \frac{3}{2} \sqrt{437760000 \cdot \frac{10^6}{10^6}}
\]

\[
C = 66492 \text{ N}
\]

Result and selection
The ball screw can now be selected from the Dimension Tables:

e.g. ball screw, size 63 x 10R x 6 - 6, with preloaded single nut with flange FEM-E-S, dynamic load rating \( C = 88800 \text{ N} \), part number R1512 640 13.

Note: Take into account the dynamic load rating of the screw bearing used!

Cross check
Service life of the selected ball screw in revolutions

\[
L = \left( \frac{88800}{8757} \right)^{10^6} \cdot 10^6
\]

\[
L = 1042 \cdot 10^6 \text{ revolutions}
\]

Service life in hours \( L_h \)

\[
L_h = \frac{1042 \cdot 10^6}{304 \cdot 60}
\]

\[
L_h = 57167 \text{ hours}
\]

The life of the selected ball screw assembly is thus greater than the required service life of 24,000 hours (including operating hours). A smaller ball screw could therefore be selected.
Design Calculations

Critical speed $n_{cr}$
The critical speed $n_{cr}$ depends on the diameter of the screw, the type of end fixity and the free length $l_{cr}$. No allowance must be made for guidance by a nut without preload. The operating speed should not reach more than 80% of the critical speed.

The characteristic speed and the max permissible linear speed must be taken into account, see "Technical Notes".

Example

<table>
<thead>
<tr>
<th>Screw diameter</th>
<th>Length $l_{cr}$</th>
<th>End fixity</th>
<th>$f_{ncr}$ value</th>
</tr>
</thead>
<tbody>
<tr>
<td>63 mm</td>
<td>2.4 m</td>
<td>II (fixed–supported)</td>
<td>27.4</td>
</tr>
</tbody>
</table>

According to the graph, the critical speed is $1850 \text{ min}^{-1}$.
The permissible operating speed is thus $1850 \text{ min}^{-1} \times 0.8 = 1480 \text{ min}^{-1}$.

The maximum operating speed in our calculation example of $n_4 = 1000 \text{ min}^{-1}$ is therefore below the permissible operating speed.

$$n_{cr} = f_{ncr} \cdot \frac{d_2}{l_{cr}^3} \cdot 10^7 \text{ (min}^{-1})$$

$$n_{crp} = 0.8 \cdot n_{cr} \text{ (min}^{-1})$$

For screw ends form 31 the end fixity can be assumed to be "fixed".

$n_{cr}$ = critical speed (min$^{-1}$)
$n_{crp}$ = permissible operating speed (min$^{-1}$)
$f_{ncr}$ = corrector value determined by bearing
$d_2$ = root diameter (see Dimension Tables) (mm)
$l_{cr}$ = critical length for preloaded nut systems (mm)
$l_s$ = distance between bearing and bearing (mm)
$l_{cr} = l_s$ for non-preloaded nut systems

Diagram:
- Critical speed
- Permissible operating speed
- Corrector value
- Length $l_{cr}$
- End fixity

$\text{Example}\n\begin{align*}
\text{Screw diameter} &= 63 \text{ mm} \\
\text{Length} \ l_{cr} &= 2.4 \text{ m} \\
\text{End fixity} \ II &= \text{fixed – supported} \\
\text{Example}\n\text{According to the graph, the critical speed is} \ 1850 \ \text{min}^{-1}. \\
\text{The permissible operating speed is thus} \ 1850 \ \text{min}^{-1} \times 0.8 = 1480 \ \text{min}^{-1}. \\
\text{The maximum operating speed in our calculation example of} \ n_4 = 1000 \ \text{min}^{-1} \text{ is therefore below} \ \text{the permissible operating speed.}
\end{align*}$
Permissible axial load on screw $F_c$ (buckling load)

The permissible axial load on the screw $F_c$ depends on the diameter of the screw, the type of end fixity and the effective free (unsupported) length $l_c$.

A safety factor of $s \geq 2$ should be taken into consideration when determining the permissible axial load.

**Example**

<table>
<thead>
<tr>
<th>Screw diameter</th>
<th>Length $l_c$</th>
<th>End fixity</th>
</tr>
</thead>
<tbody>
<tr>
<td>63 mm</td>
<td>2.4 m</td>
<td>II (fixed – supported)</td>
</tr>
</tbody>
</table>

According to the graph, the theoretically permissible axial load is 360 kN. A permissible axial load on the screw of 360 kN : 2 = 180 kN is achieved when applying the safety factor 2. This therefore lies above the maximum operating load of $F_1 = 50$ kN used in our calculation example.

$$F_c = F_{cp} \cdot \frac{d_2^4}{l_c^2} \cdot 10^4 (N)$$

$$F_{cp} = \frac{F_c}{2} (N)$$

$F_c$ = theoretically permissible axial load on screw

$F_{cp}$ = permissible axial load during operation

$f_{fc}$ = corrector value determined by bearing

$d_2$ = root diameter (mm), see Dimension Tables

$l_c$ = unsupported threaded length (mm)
Design Notes, Mounting Instructions

**Bearing design**
For customer-machined screw ends, please consider the design notes given for screw ends and housings.

For Rexroth screw end designs, see “End Machining Details.”

Rexroth delivers complete drive systems, including the end bearings. Calculations are performed with the formulas used in the antifriction bearing industry.

**Mounting**

**Angular-contact thrust ball bearings and deep-groove ball bearings**
When mounting the angular-contact thrust ball bearings LGF and LGN, ensure that the mounting forces are exerted only on the bearing rings. Never apply mounting forces via the anti-friction bearing elements or the seal rings! The two sections of the inner raceway may not be separated during assembly or disassembly for any reason! Tighten the mounting screws for screw-down or flange-mounted bearings in crosswise sequence. The mounting screws may be subjected only to tension amounting to a maximum of 70% of their yielding point. The screw-down (LGF) bearings have a groove on the cylindrical surface of the outer raceway for disassembly. The individual bearings of the bearing pair series LGF-C... and LGN-C... are marked on the cylindrical surfaces of the outer raceways (see Figure). The markings reveal the bearing sequence. The sealing rings should face outward after proper mounting.

**Slotted nut NMA, NMZ**
The bearings are preloaded by tightening the nuts.

In order to prevent settling phenomena, we recommend first tightening the slotted nut by twice the value of the tightening torque $M_a$ and then easing the load. Only then should the slotted nut be retightened to the specified tightening torque $M_a$. The two set screws are then alternately tightened using a hexagon socket wrench. The components are disassembled in the reverse order, i.e. the set screws are to be removed before the slotted nut. The slotted nuts can be used several times when properly assembled and disassembled by competent personnel. The inner raceways of the bearings are dimensioned in such a way as to achieve a defined bearing preload sufficient for most applications when the slotted nut is tightened ($M_a$ in accordance with Dimension Table).
Lubrication, Mounting the Housing

Mounting the housing SEB
Tighten the pillow block mounting screws in crosswise sequence. See table for max. tightening torque. The housing nut fixes the entire bearing unit in the housing. Use a threadlocking adhesive to secure the housing nut in place.

⚠️ Note:
Take care to align the screw and nut assembly, the bearings and the guideway precisely with one another. The Rexroth gauge is a useful aid here.

Mounting screw Locating pins

Tightening torques for fastening screws according to VDI 2230 for $\mu_g = \mu_h = 0.125$

Lubrication of the end bearings
Bearings for ball screw assemblies are lubricated with grease for a lifetime of reliable service. It should be noted, however, that grease lubrication does not facilitate the dissipation of heat in the bearings. The bearing temperature should therefore not exceed 50°C, particularly in machine tool applications. Angular-contact thrust ball bearings of the series LGF, LGN are lubricated for life with grease KEP-35 per DIN 51825. For regreasing, the quantities stated in the table below can be applied via the lube ports provided on the bearings.

The maximum interval can be assumed to be 350 million revolutions, in which case the larger of the two quantities should be used. As a rule, the initial grease quantity will therefore last for the entire service life of a ball screw assembly.

<table>
<thead>
<tr>
<th>Designation</th>
<th>Quantity (g)</th>
<th>Designation</th>
<th>Quantity (g)</th>
</tr>
</thead>
<tbody>
<tr>
<td>LGN-B-0624</td>
<td>0.3 / 0.2</td>
<td>LGN-B-1242</td>
<td>0.4 / 0.3</td>
</tr>
<tr>
<td>LGN-B-1034</td>
<td>0.3 / 0.2</td>
<td>LGF-B-1255</td>
<td>0.4 / 0.3</td>
</tr>
<tr>
<td>LGN-B-1747</td>
<td>0.5 / 0.4</td>
<td>LGF-B-1762</td>
<td>0.5 / 0.4</td>
</tr>
<tr>
<td>LGN-B-2052</td>
<td>0.8 / 0.5</td>
<td>LGF-B-2088</td>
<td>0.4 / 0.6</td>
</tr>
<tr>
<td>LGN-B-2557</td>
<td>1.0 / 0.6</td>
<td>LGF-B-2575</td>
<td>2.0 / 1.2</td>
</tr>
<tr>
<td>LGN-B-3062</td>
<td>1.0 / 0.6</td>
<td>LGF-B-3080</td>
<td>2.0 / 1.2</td>
</tr>
<tr>
<td>LGN-B-3572</td>
<td>1.6 / 0.9</td>
<td>LGF-B-3590</td>
<td>3.5 / 2.5</td>
</tr>
<tr>
<td>LGN-A-4075</td>
<td>2.0 / 1.2</td>
<td>LGN-A-4090</td>
<td>6.0 / 3.5</td>
</tr>
<tr>
<td>LGN-A-5090</td>
<td>2.5 / 1.5</td>
<td>LGN-A-50110</td>
<td>9.0 / 5.5</td>
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</table>

Steel/steel material pairing

<table>
<thead>
<tr>
<th>Strength class for $O_1$; $O_2$</th>
<th>M5</th>
<th>M6</th>
<th>M8</th>
<th>M10</th>
<th>M12</th>
<th>M14</th>
</tr>
</thead>
<tbody>
<tr>
<td>$8.8$</td>
<td>5.5</td>
<td>9.5</td>
<td>23</td>
<td>46</td>
<td>80</td>
<td>125</td>
</tr>
<tr>
<td>$12.9$</td>
<td>9.5</td>
<td>16.0</td>
<td>39</td>
<td>77</td>
<td>135</td>
<td>215</td>
</tr>
</tbody>
</table>

Steel/aluminum and aluminum/aluminum material pairings

<table>
<thead>
<tr>
<th>Strength class for $O_1$; $O_2$</th>
<th>M5</th>
<th>M6</th>
<th>M8</th>
<th>M10</th>
<th>M12</th>
<th>M14</th>
</tr>
</thead>
<tbody>
<tr>
<td>$8.8$</td>
<td>4.8</td>
<td>8.5</td>
<td>20</td>
<td>41</td>
<td>70</td>
<td>110</td>
</tr>
<tr>
<td>$12.9$</td>
<td>4.8</td>
<td>8.5</td>
<td>20</td>
<td>41</td>
<td>70</td>
<td>110</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Size</th>
<th>$d_0 \times P$</th>
<th>$O_1$</th>
<th>$O_2$</th>
<th>$O_3$ – Tapered pin (hardened)</th>
<th>$O_4$ – Straight pin (DIN 6325)</th>
</tr>
</thead>
<tbody>
<tr>
<td>8 x 2.5</td>
<td>8</td>
<td>M5 x 20</td>
<td>M6 x 16</td>
<td>4 x 20</td>
<td></td>
</tr>
<tr>
<td>12 x 5</td>
<td>8</td>
<td>M5 x 20</td>
<td>M6 x 16</td>
<td>4 x 20</td>
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<tr>
<td>16 x 5</td>
<td>11</td>
<td>M8 x 35</td>
<td>M10 x 25</td>
<td>8 x 40</td>
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<tr>
<td>16 x 10</td>
<td>11</td>
<td>M8 x 35</td>
<td>M10 x 25</td>
<td>8 x 40</td>
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</tr>
<tr>
<td>20 x 5</td>
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<td>M8 x 35</td>
<td>M10 x 25</td>
<td>8 x 40</td>
<td></td>
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<td>11</td>
<td>M8 x 35</td>
<td>M10 x 25</td>
<td>8 x 40</td>
<td></td>
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<tr>
<td>25 x 5</td>
<td>14</td>
<td>M10 x 40</td>
<td>M12 x 30</td>
<td>10 x 50</td>
<td></td>
</tr>
<tr>
<td>25 x 10</td>
<td>14</td>
<td>M10 x 40</td>
<td>M12 x 30</td>
<td>10 x 50</td>
<td></td>
</tr>
<tr>
<td>25 x 25</td>
<td>14</td>
<td>M10 x 40</td>
<td>M12 x 30</td>
<td>10 x 50</td>
<td></td>
</tr>
<tr>
<td>32 x 5</td>
<td>14</td>
<td>M10 x 40</td>
<td>M12 x 30</td>
<td>10 x 50</td>
<td></td>
</tr>
<tr>
<td>32 x 10</td>
<td>14</td>
<td>M10 x 40</td>
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<td>10 x 50</td>
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</tr>
<tr>
<td>32 x 32</td>
<td>14</td>
<td>M10 x 40</td>
<td>M12 x 30</td>
<td>10 x 50</td>
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<td>M12 x 50</td>
<td>M14 x 35</td>
<td>10 x 50</td>
<td></td>
</tr>
<tr>
<td>40 x 10</td>
<td>16</td>
<td>M12 x 50</td>
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<td>16</td>
<td>M12 x 50</td>
<td>M14 x 35</td>
<td>10 x 50</td>
<td></td>
</tr>
</tbody>
</table>
End Bearings

Design Calculations

Resulting and equivalent bearing loads

For angular-contact thrust ball bearings LGN and LFG
Angular-contact thrust ball bearings are preloaded. The chart shows the resulting axial bearing load $F_{ax}$ as a function of preload and axial operating load $F_{Lax}$. For a purely axial load $F_{comb} = F_{ax}$.

$$F_{comb} = X \cdot F_{rad} + Y \cdot F_{ax}$$

$$F_{ax} = \text{resulting axial bearing load (N)}$$
$$F_{comb} = \text{combined equivalent bearing load (N)}$$
$$F_{rad} = \text{radial bearing load (N)}$$

<table>
<thead>
<tr>
<th>$\alpha = 60^\circ$</th>
<th>$X$</th>
<th>$Y$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$F_{rad} \leq 2.17$</td>
<td>1.90</td>
<td>0.55</td>
</tr>
<tr>
<td>$F_{rad} &gt; 2.17$</td>
<td>0.92</td>
<td>1.00</td>
</tr>
</tbody>
</table>

$\alpha = \text{pressure angle}$
$F_{ax} = \text{resulting bearing load}$
$F_{Lax} = \text{operating load}$
$X, Y = \text{dimensionless factor}$

If the radial operating forces are not insignificant, the equivalent bearing loads are calculated according to formula 20.

Bearings for ball screw assemblies are also able to accommodate tilting moments. As a rule, the moments that usually occur due to the weight and drive motion of the screw do not need to be incorporated in the calculation of the equivalent bearing load.

Permissible static axial load for bearing series LGF
The permissible static axial load of LGF-series bearings in screw-down direction is:

$$F_{0ax p} \leq \frac{C_0}{2}$$

$$F_{0ax p} = \text{permissible static axial bearing load (N)}$$

The static axial load rating $C_0$ is stated in the Dimension Tables.