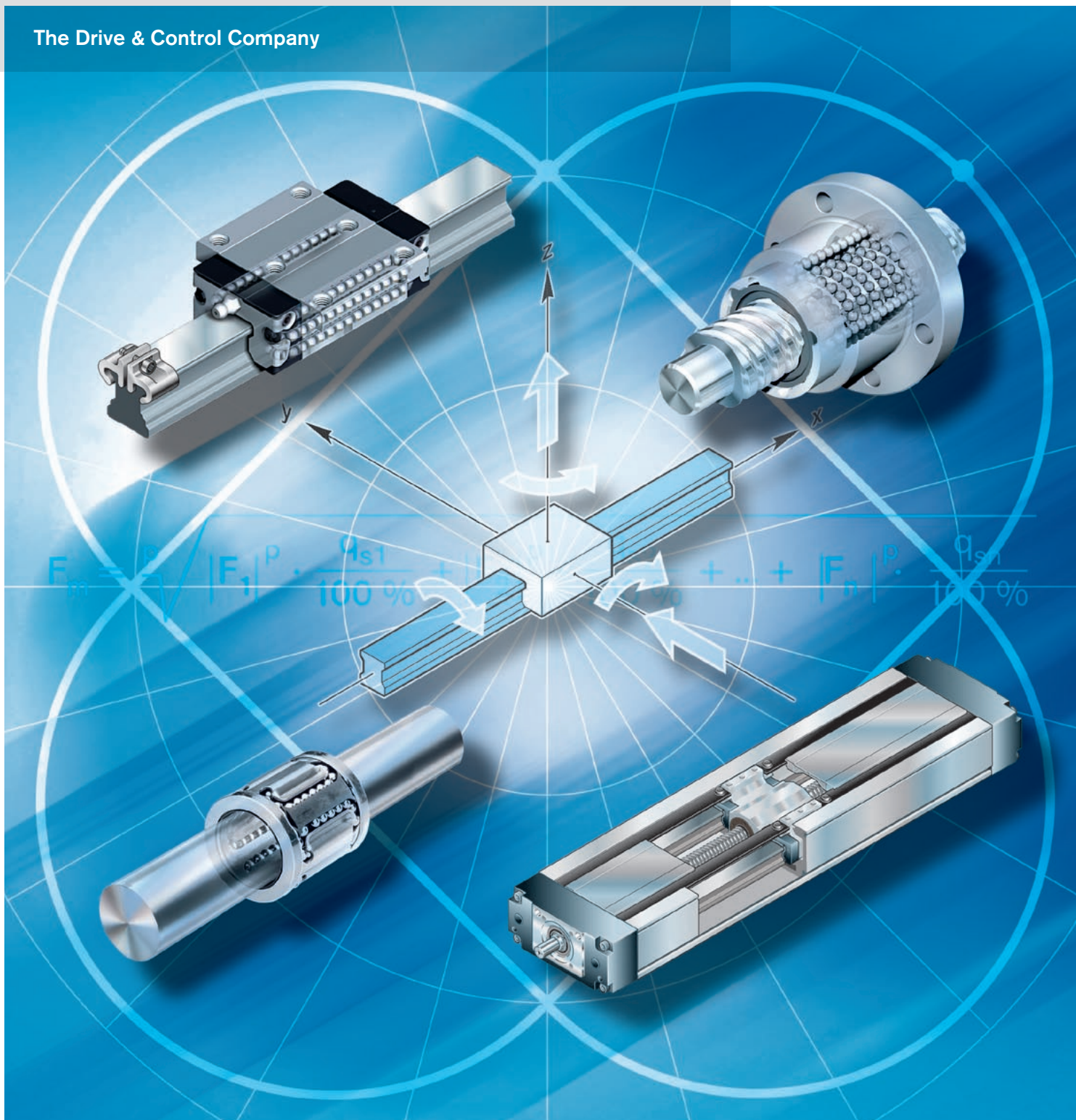




Linear Motion Technology Handbook

The Drive & Control Company



Linear Motion and Assembly Technologies



1 Introduction

1.1 Foreword

Reliable guidance and precise positioning – Rexroth provides a complete linear motion technology range for these tasks, from guides through to drive units. Linear motion technology components are the interface between static and moving machine elements. They significantly affect the machine characteristics. Linear motion technology comes into play whenever precision and high load-bearing capability are required, as is above all the case in machine construction and automation. Rexroth's guidance components are profiled rail systems and linear bushings and shafts. Ball screw assemblies are the drive components used for positioning. Both of these functions are combined in linear motion systems. However, Rexroth offers much more than just linear motion products. As a global equipment provider for machinery and plant construction, Rexroth provides all the relevant drive, control and motion technologies – from mechanics, hydraulics and pneumatics through to electronics.

This linear motion technology handbook provides specialized knowledge about Rexroth's linear motion technology products, giving users insights into the world of linear motion. The handbook is not designed to replace the Rexroth product catalogs but simply as a supplement to them. The dimensions, performance data and product versions, etc. must still be taken from the catalogs. The handbook, however, contains extensive advice on system characteristics, product selection, design and calculation. It is designed for all linear motion technology users.

The handbook is divided into a general Principles chapter, equally applicable to all Rexroth products, and into additional special chapters on the individual linear motion technology components.

The Principles chapter describes the physical background knowledge for linear motion technology. This includes rolling contact with all its usual practical manifestations, as well as generally accepted methods for calculating nominal life. Also described are system characteristics common to all products, such as preload, rigidity, accuracy and friction. The following chapters on Profiled Rail Systems, Linear Bushings and Shafts, Ball Screw Drives and Linear Motion Systems refer to the respective Rexroth products and their characteristics. These chapters cover additional basic knowledge, system properties, advice on product selection, and design hints for users of these products. A substantial part of the handbook covers how to calculate, dimension and configure the guidance and drive components. This includes detailed calculation of the components' life expectancy, calculation of the static load safety factors, determination of the critical screw speed, and drive dimensioning. The structural design and the functionalities of the individual types, versions and components are also described. The reader is therefore provided with an overview of each product's special characteristics.

We hope that you will enjoy reading and using this handbook.

Bosch Rexroth AG
The Drive & Control Company
Linear Motion and Assembly Technologies

1 Introduction

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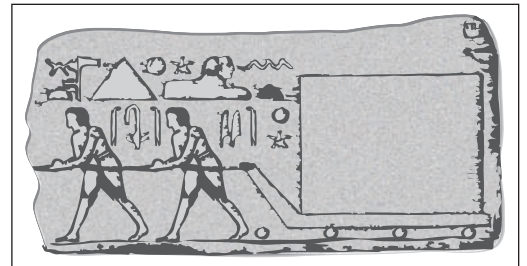
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2 Principles

2.1 Historical development

Linear motion

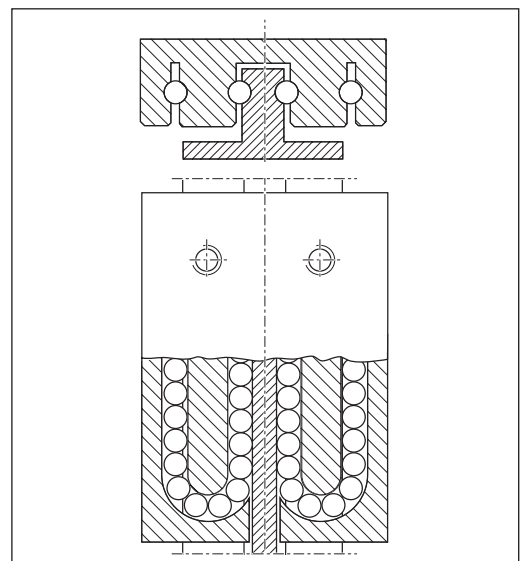
When building the pyramids, the Egyptians had already encountered the problem of how to move heavy loads. This was solved by using tree trunks laid under blocks of stone. Water was also applied as a lubricant to reduce friction.



Egyptian linear motion guide

Rolling contact profiled rail systems

This basic principle is still used today in modern linear motion guides. The rolling elements nowadays, however, no longer have to be carried to the desired position by hand but instead recirculate within the guide system itself. The requirements regarding rigidity, load-bearing capacity and resistance to movement have also changed. Applications today place the highest demands on precision and economy.



Historical illustration of a ball rail system

Round guides

In 1957, "Deutsche Star" signed a license contract to manufacture ball bushings in accordance with the patents held by the US-based Thomson company. "Deutsche Star" therefore became the leading manufacturer of linear bearings in Europe.



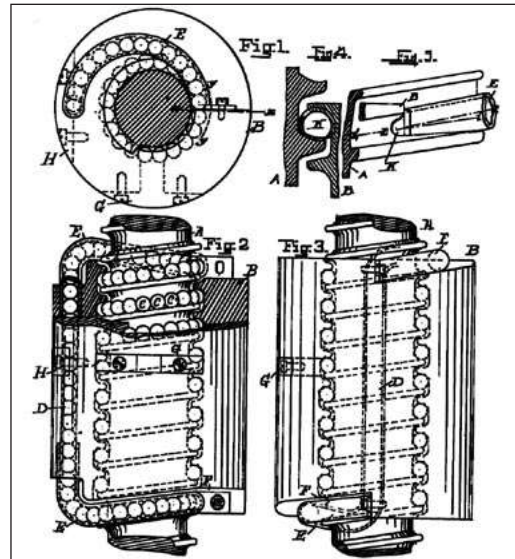
"Deutsche Star" catalog

2 Principles

2.1 Historical development

Ball screw drive

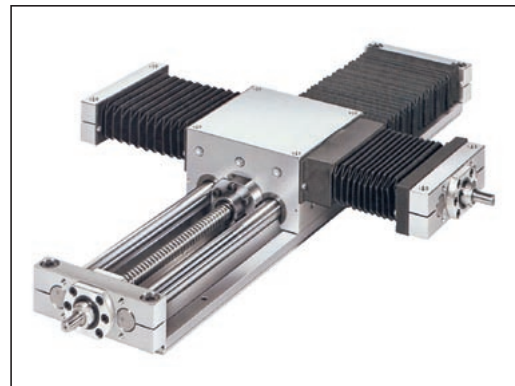
Sliding screw drives were already used in Antiquity to convert rotary motion into linear motion. The ball screw drive was first mentioned in literature in the 19th century. It replaced sliding friction with rolling friction. It was first used industrially in the 1940s, when General Motors built ball screw drives into vehicle steering systems. Further industrial applications soon followed. Since then, the design and manufacturing processes have made enormous progress. Today, ball screw drives are found in a broad range of industries.



Ball screw drive from a historical patent

Linear motion systems

Linear motion systems are ready-to-install drive and guidance units. This makes it easier for users to design and assemble their applications. It is not necessary to calculate and dimension the individual components, since the linear motion systems are installed as complete units. The first linear motion systems built by the former "Deutsche Star" consisted of linear bushings and shafts and a ball screw or pneumatic drive. These transfer tables were also offered as two-axis X-Y tables. Meanwhile, many different guide and drive unit variants have been incorporated into linear motion systems. Today, customers can select the optimal linear motion system from a broad range of Rexroth products.



X-Y table from the "Deutsche Star" product range

2 Principles

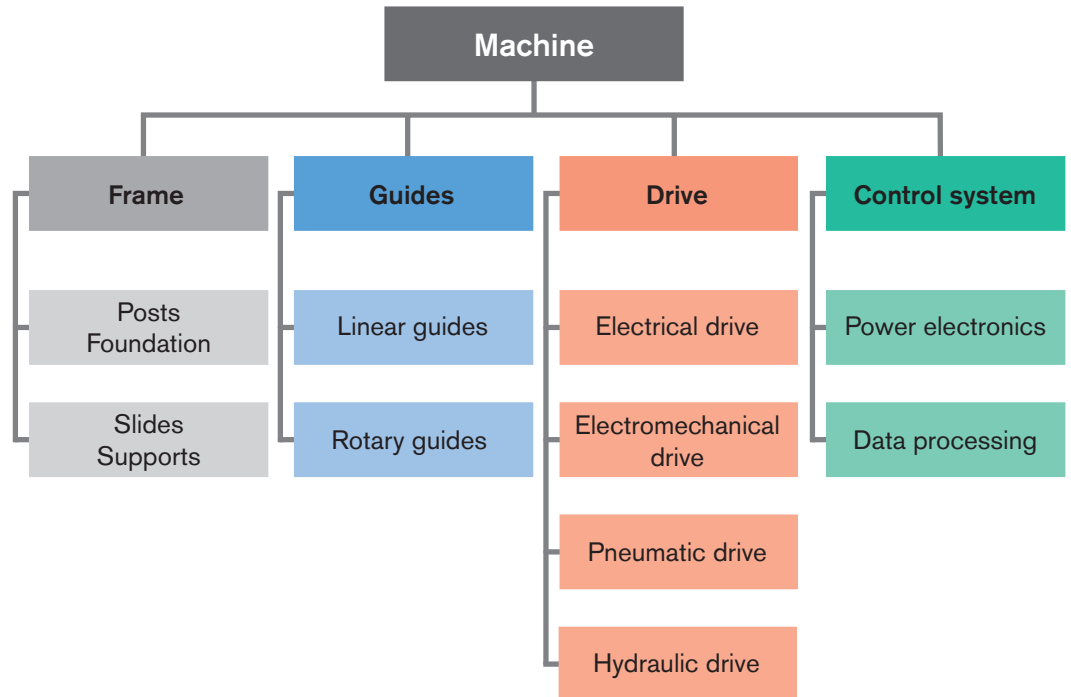
2.2 Technical principles

2.2.1 Elements of a machine

Basic structure of a machine

Rexroth's product range includes linear guides and drive units in a very wide variety of designs. For better understanding of these, it is useful first

of all to take a closer look at the basic structure of a machine and its most important components.



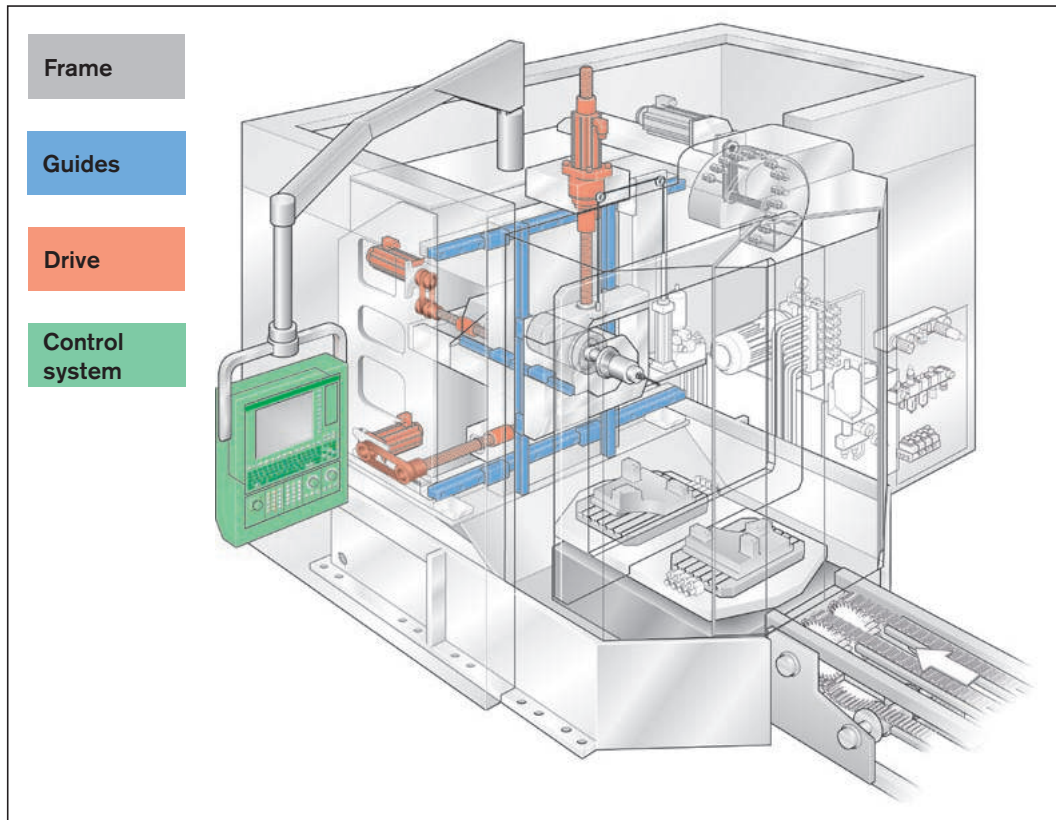
2 Principles

2.2 Technical principles

2.2.1 Elements of a machine

Frame	A machine's frame consists of stationary components (posts, foundation) and moving components (slides, supports). There are various designs to suit the corresponding application (standard machine base, gantry design, etc.)	The frame's purpose is to anchor the machine and to transmit forces.
Guides	These are responsible for the guidance and power transmission of the moving machine components. The machine's accuracy is due in no small	measure to the accuracy of the guidance system. Based on the movement, a distinction is made between linear guidance and rotary guidance.
Drives	Drives convert electrical, hydraulic or pneumatic energy into mechanical energy. Electromechanical drives are a special form of drive incorporating transmission elements (e.g. ball screw drives). A distinction is made between main drives, which	execute relative movements (e.g. between a tool and a workpiece), and auxiliary drives, which execute positioning movements (e.g. workpiece transport or tool changing).
Control system	The control system coordinates the requisite movements of the machine, i.e. the moving parts' speed and acceleration. The power electronics serves the motors and high-powered actuators,	whereas the data processing system covers the limit switches, measuring systems, field bus systems and the safety circuits.

Elements of a machine (example)



Machine with typical linear components shown in color

2 Principles

2.2 Technical principles

2.2.2 Guides

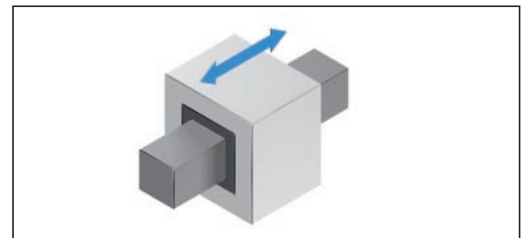
Guides are differentiated according to the type of motion, the type of contact points and the rolling element recirculation principle used.

2.2.2.1 Differentiation of guides according to the type of motion

Machines could not execute movements without guidance components. Depending on the guide's design, forces and moments can be transmitted in certain directions between moving and non-moving components. Guides can generally be differentiated according to their type of motion.

Linear guides

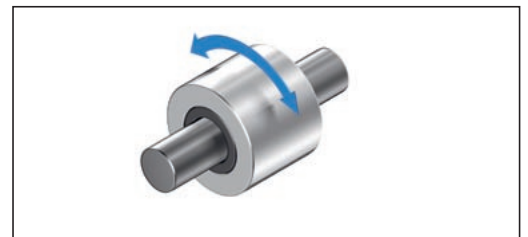
Linear motion takes place along an axis.
Examples: ball rail systems, dovetail sliders



Linear guide

Rotary guides

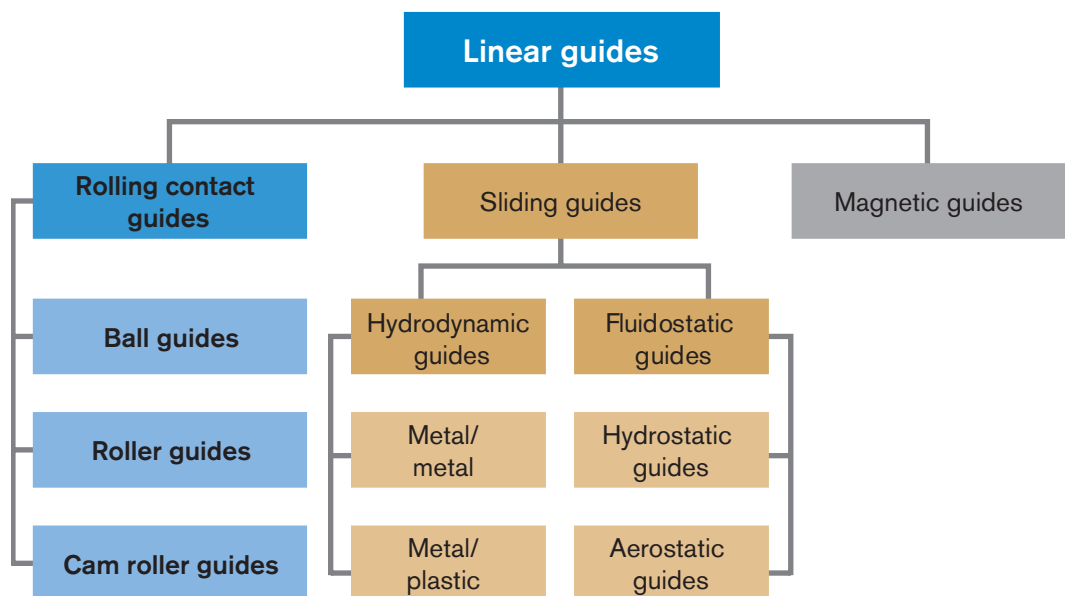
Rotary motion takes place about an axis.
Examples: deep groove ball bearings, radial sliding bearings



Rotary guide

2.2.2.2 Differentiation of linear guides according to the type of contact points

Linear guides can be differentiated according to the physical operating principle of the contact point, as is shown in the following diagram.








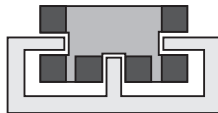


2 Principles

2.2 Technical principles

2.2.2 Guides

2.2.2.3 Operating principle of linear guides

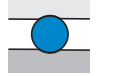

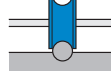
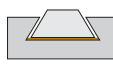
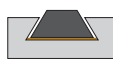
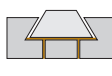
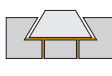
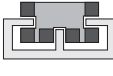
Linear guides		Operating principle	
Rolling contact guides	Ball guide		There are balls between the moving and fixed machine parts.
	Roller guide		There are rollers between the moving and fixed machine parts.
	Cam roller guide		There are cam rollers supported on ball bearings between the moving and fixed machine parts.
Hydrodynamic sliding guides	Metal/metal		Both machine parts are in contact during standstill. When movement starts, a lubricating film gradually forms between the moving and the fixed machine element. The lubricating film only separates the moving and the fixed element of the machine completely at higher sliding speeds.
	Metal/plastic		The operational principle is the same as for metal/metal. The metal/plastic material combination reduces friction when movement starts, until a complete lubricating film forms.
Fluidostatic sliding guides	Hydrostatic guide		A pump supplies liquid lubricant to the guide. The moving part rises. Between the moving and the fixed element there is a film of lubricant under pressure.
	Aerostatic guide		A compressor supplies compressed air to the guide. The moving and the fixed machine element are separated by the compressed air.
Magnetic guides			The moving and fixed machine elements are separated by magnetic force. The moving part "floats." The guide is therefore non-contacting.

2 Principles

2.2 Technical principles

2.2.2 Guides

2.2.2.4 Linear guide characteristics

Characteristics	Rolling contact guides			Hydrodynamic sliding guides		Fluidostatic sliding guides		Magnetic guide
	Ball guide	Roller guide	Cam roller guide	Metal/metal	Metal/plastic	Hydrostatic guide	Aerostatic guide	Magnetic suspension
								
Load-bearing capability	+++	+++	++	+++	+++	+++	o	+++
Rigidity	++	+++	+	+++	++	+++	o	+
Accuracy	++	++	++	+	+	++	++	+++
Friction characteristics	++	++	++	+	+	+++	+++	+++
Speed	+++	+++	+++	+	+	+++	+++	+++
Damping characteristics	+	+	+	+++	+++	+++	+++	+++
Operating safety	+++	+++	+++	+++	+++	+	+	+
Standardization	+++	+++	+++	+	+	o	o	o
Service life	++	++	++	++	++	+++	+++	+++
Costs	++	++	++	+++	+++	+	+	o

+++ Very good
++ Good

+ Satisfactory
o Adequate

The table shows that rolling contact guides score excellent ratings for the most commonly demanded characteristics. When the price-performance ratio is taken into account, it is no surprise that

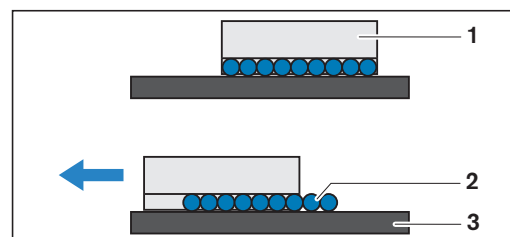
rolling contact guides have replaced conventional sliding guides more and more in recent years and now represent the standard in machine components.

2.2.2.5 Differentiation of rolling contact guides according to rolling element recirculation

Rolling element recirculation

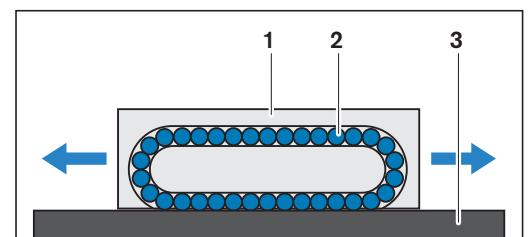
The type of contact point is not the only way to differentiate rolling contact guides. They also subdivide into guides with and guides without recirculation of the rolling elements.

In rolling contact guides without rolling element recirculation, the rolling elements (2) move at half the speed of the runner block (1) and therefore only cover half the distance. Rolling contact guides without rolling element recirculation therefore have only a limited stroke.



Rolling contact guide without rolling element recirculation

In rolling contact guides with rolling element recirculation, the rolling elements (2) recirculate within the runner block (1) and move together with the runner block in relation to the guide rail (3). The stroke is limited only by the rail length.



Rolling contact guide with rolling element recirculation

2 Principles

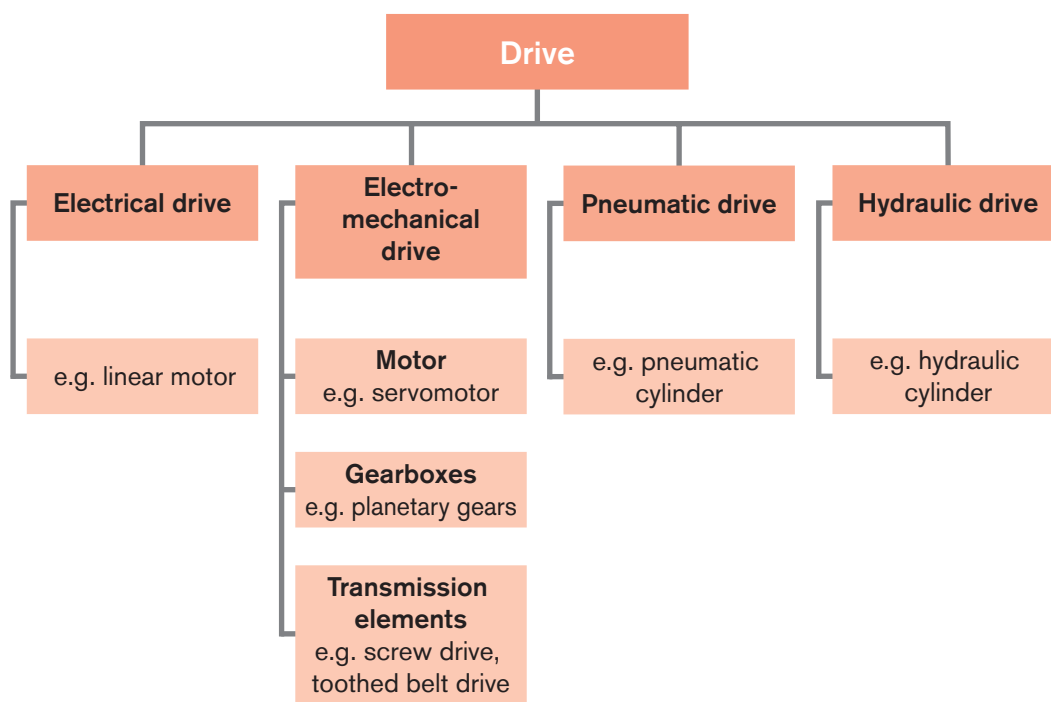
2.2 Technical principles

2.2.3 Drive

2.2.3.1 Drive types

Electrical, electromechanical, pneumatic or hydraulic drives can be used for main and auxiliary drives.

Among the electromechanical drives, the ball screw drive belongs to the sub-category of transmission elements. It is also frequently called a feed component.



2 Principles

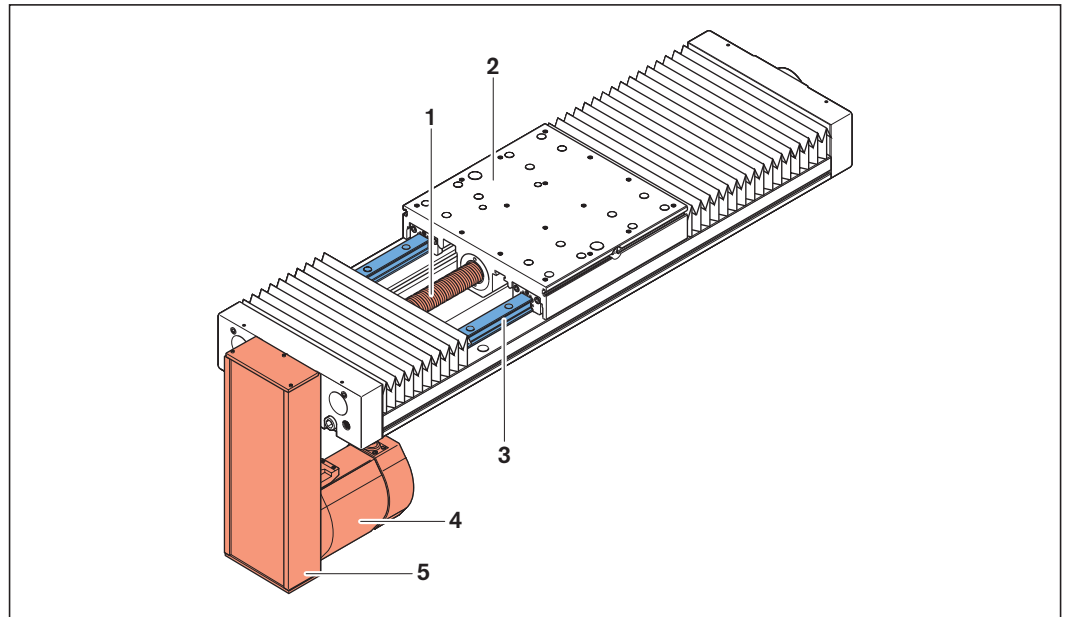
2.2 Technical principles

2.2.3 Drive

2.2.3.2 Screw drive

Structural design

The following illustration of a Ball Rail Table TKK shows the typical structural design of a drive unit with ball screw drive together with rail guides.



Ball Rail Table TKK with ball screw drive and ball rail system

- 1 Ball screw drive
- 2 Carriage
- 3 Guide rail
- 4 Motor
- 5 Gear unit (here: timing belt side drive)

Screw drive

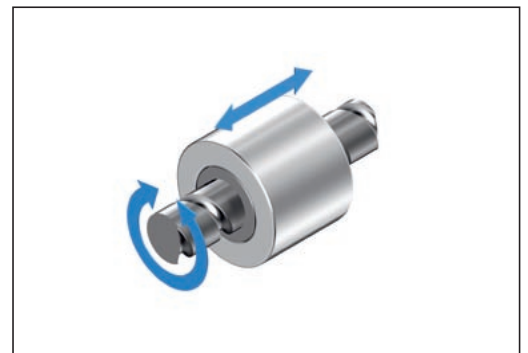
In a screw drive, a rotational movement takes place about an axis with a defined screw lead. Here rotary motion is converted into linear motion and vice versa.

In mechanical engineering, screw drives are classified as drive elements (transmission elements, feed elements).

Examples: ball screws (BS), acme screws

**DIN 69051 Part 1
Ball screw drive**

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



Screw drive

2 Principles

2.3 Rolling contact

2.3.1 Rolling contact of balls and rollers

In linear motion technology, balls or rollers are used as the rolling elements. Balls and rollers have different characteristics because of their different geometries.

2.3.1.1 Contact areas in balls and rollers

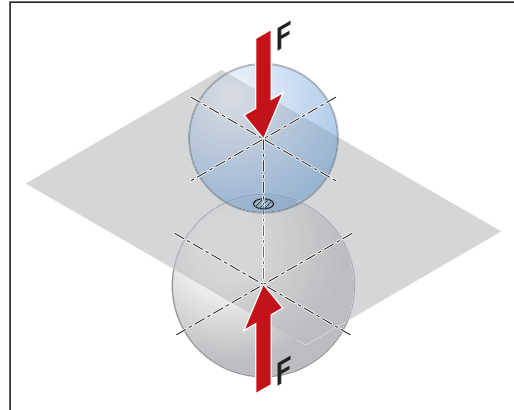
Point contact in the case of balls

The rolling contact considerations for balls are based on the Hertz theory. This deals with the behavior of two curved bodies when they are pressed against each other by an external force.

The Hertz theory

The elastic deformation, the dimensions of the compression areas, the maximum surface pressure and the sub-surface stresses occurring during the rolling contact of balls can be calculated using the Hertz theory.

The simplest case is the contact of a sphere with a plane (idealized point contact). In this case, there is a relatively small circular contact area, leading to a very high surface pressure.

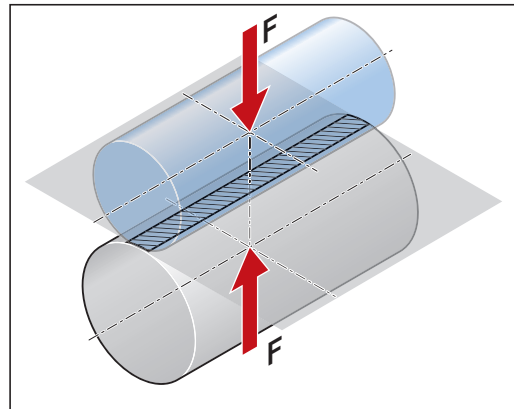


Point contact for ball rolling elements

If balls with different diameters are compared, it becomes apparent that, in the case of larger balls, the deformation and the surface pressure are smaller under the same load. The load-bearing capacity therefore rises as the ball diameter increases.

Line contact in the case of rollers

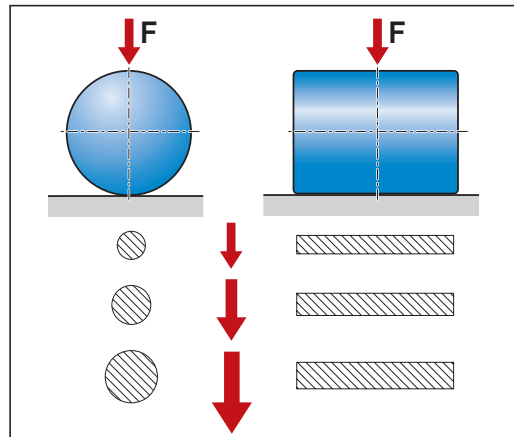
The Hertz theory does not apply to deformation in the case of line contact. If two cylindrical rollers are pressed together with their axes parallel, a line-shaped contact is produced. The resulting contact area is elongated, with the shape and size of the contact area dependent only on the load and the length of the contact line. Elastic deformation during line contact is independent of the roller diameter. At a constant roller diameter, the load-bearing capacity rises with increasing roller length.



Line contact for roller rolling elements

Contact area

Rollers have a larger contact area than balls. This larger contact area enables the rollers to transmit greater forces, leading to greater rigidity. Compared to balls, therefore, smaller sizes can be used to bear the same external load.



Contact area for balls and rollers under increasing load

2 Principles

2.3 Rolling contact

2.3.1 Rolling contact of balls and rollers

2.3.1.2 Ball contact conformity

Running tracks with contact conformity

In the case of rolling ball contact with planar running tracks, the high surface pressure and the absence of guided movement have an unfavorable effect. For these reasons, profiled running tracks offering contact conformity are used. This increases the contact area and reduces the surface pressure accordingly. Higher load-bearing capabilities can therefore be achieved. This also serves to guide the movement of the rolling element.

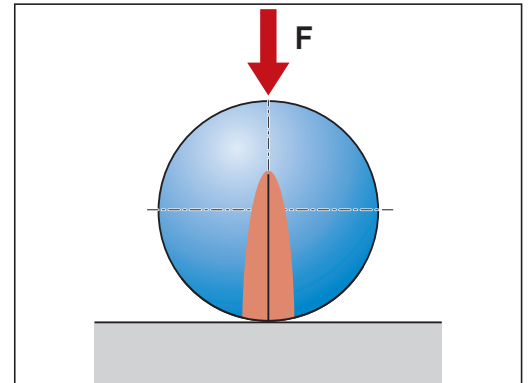
Definition of conformity

Conformity is the ratio of the running track radius to the ball diameter, expressed as a percentage:

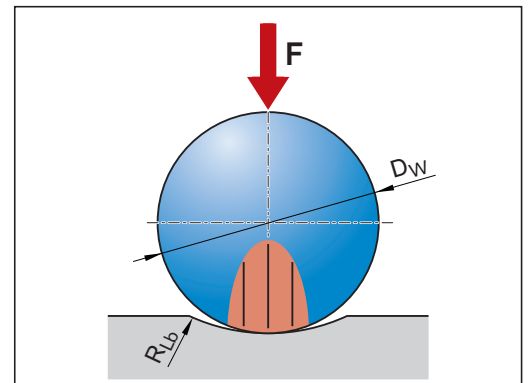
$$(2-1) \quad \kappa = \frac{R_{Lb}}{D_W} \cdot 100\%$$

κ = conformity (%)
 R_{Lb} = running track radius (mm)
 D_W = ball diameter (mm)

A ball on a running track designed for contact conformity will deflect significantly less than a comparable ball on a planar running track. Also, where there is conformity between the ball and the track, the ball will have a longer life than a ball with point contact because of the larger contact area and the resulting distribution of the forces acting on it.



Stress distribution for a contact area without conformity



Stress distribution for a contact area with conformity

2.3.1.3 Logarithmic and cylindrical roller profiles

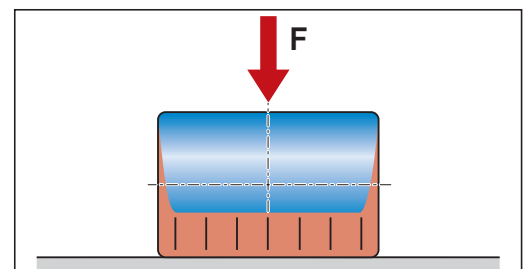
Logarithmic profile

Rolling contact with rollers differs from that with balls. A distinction is made between rollers with cylindrical and logarithmic profiles. Both forms are approximately comparable in terms of their elastic deflection behavior.

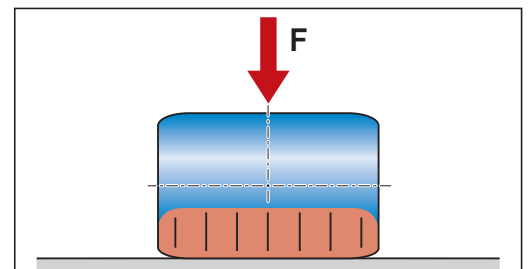
Rollers with logarithmic profiles, however, offer further advantages:

- More even distribution of forces
- Lower peak stresses at the edges
- Correspondingly less edge contact

This results in longer life than with cylindrical rollers. Rexroth therefore uses rollers with logarithmic profiles.



Stress distribution for cylindrical roller profiles



Stress distribution for logarithmic roller profiles

2 Principles

2.3 Rolling contact

2.3.1 Rolling contact of balls and rollers

2.3.1.4 Elastic deflection of balls and rollers

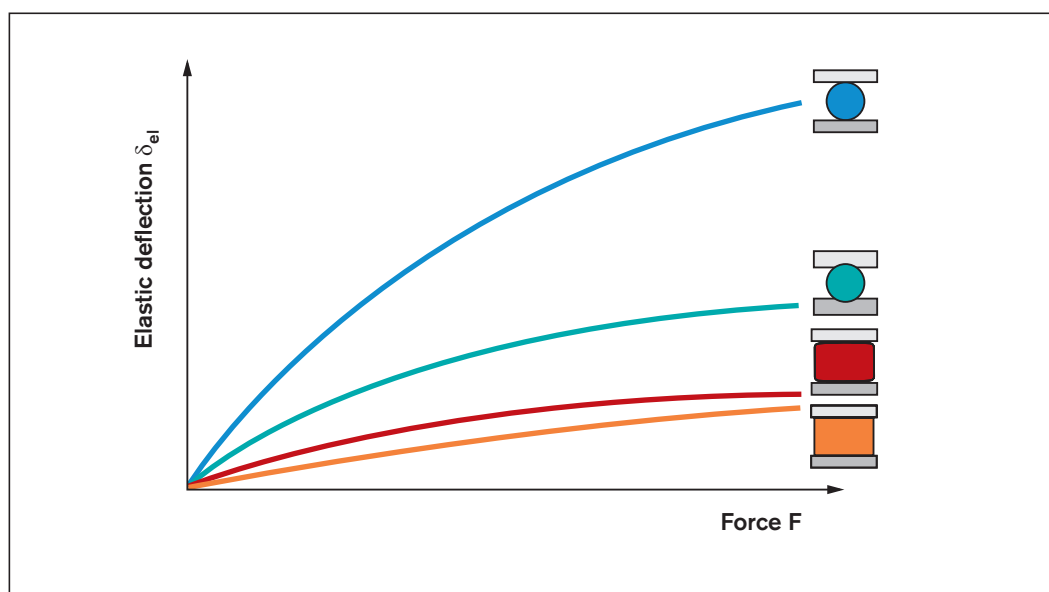
Elastic deflection

Elastic deflection means that no permanent deformation of the parts in contact occurs.

Depending on the type of rolling element and the shape and area of the contacting surfaces, a force acting on the rolling element will lead to different degrees of elastic deflection:

- Rollers deflect less than balls. Rollers have a significantly higher rigidity and a higher load-bearing capacity because of the larger contact area.
- The deflection behavior of rollers with logarithmic profiles and rollers with cylindrical profiles are approximately comparable.
- A ball on a running track with conformity will deflect significantly less than a comparable ball on a track with no conformity.

The graph shows the elastic deflection for the rolling contact conditions described.



Exemplary comparison of elastic deflection in balls and rollers

- Ball and running track with no conformity
- Ball and running track with conformity
- Roller with logarithmic profile
- Roller with cylindrical profile

Assumptions:

- Balls and rollers with the same diameter
- Rollers in standard lengths

2 Principles

2.3 Rolling contact

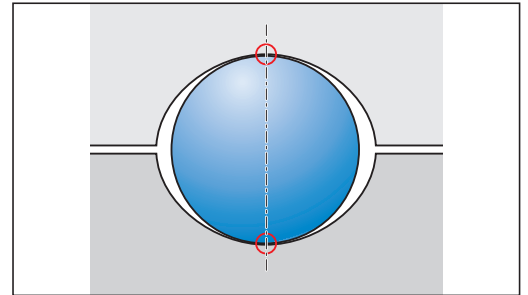
2.3.2 Running track geometry for ball rolling elements

2.3.2.1 Arc-shaped raceways

Profiled running tracks with conformity are used for rolling contact guides with balls. In a rolling contact system, the ball running tracks of the two mating parts between which the ball rolls are designated as raceways. Usually the raceways are designed to have either a circular-arc profile or a Gothic-arch profile.

**Circular-arc raceway
2-point contact**

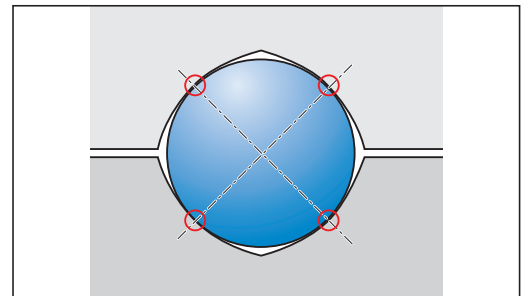
The circular-arc raceway has two running tracks with conformity. This produces a 2-point contact between the running tracks and the rolling element.



Circular-arc raceway with 2-point contact

**Gothic-arch raceway
4-point contact**

In Gothic-arch raceways, the Gothic profile (derived from the pointed arch, a stylistic element in Gothic architecture) produces two running tracks with conformity per side. This results in 4-point contact with the rolling element.



Gothic-arch raceway with 4-point contact

2 Principles

2.3 Rolling contact

2.3.2 Running track geometry for ball rolling elements

2.3.2.2 Differential slip

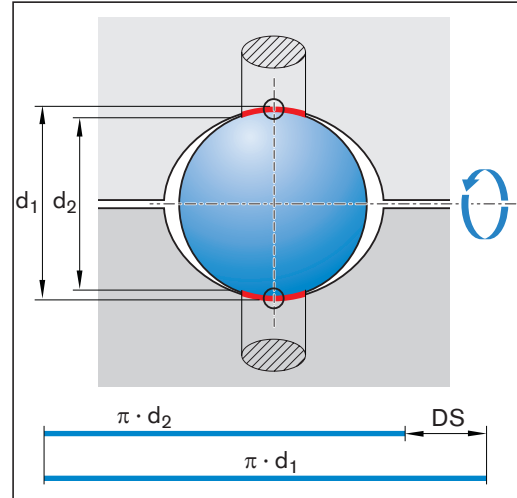
Unlike point contact, because of the curved running tracks with conformity, the ball has a larger, elliptical and similarly curved contact area. The ball therefore rolls in a diameter range of d_1 to d_2 .

The different effective rolling diameters d_1 and d_2 in the contact area result in different rolling speeds, which leads to partial sliding friction. This effect is termed differential slip.

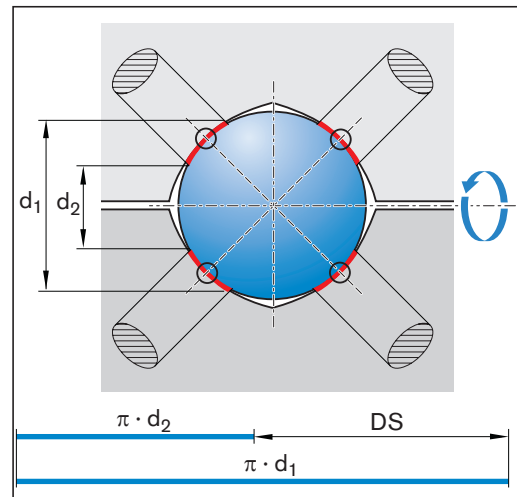
The consequences of differential slip are a higher friction coefficient and hence a higher resistance to movement.

The differential slip is substantially greater in the 4-point contact Gothic-arch raceway than it is in the 2-point contact circular-arc raceway. The friction coefficient is therefore lower with 2-point contact than with 4-point contact.

Rexroth therefore mainly uses 2-point contact systems. Solutions with 4-point contact are generally used where a compact build or very small designs (e.g. miniature ball rail systems) are required. Because the forces are distributed over four contact areas, it is possible to produce linear motion guides with only two raceways, resulting in relatively low-cost systems.



Differential slip (DS) in circular-arc raceways



Differential slip (DS) in Gothic-arch raceways

2 Principles

2.4 Life expectancy

2.4.1 Calculation principles

2.4.1.1 Nominal life

Nominal life L

The nominal life L is the distance that a component can cover before the first signs of fatigue appear on the running tracks or rolling elements. Lundberg and Palmgren have developed a calculation method for predicting the life expectancy of an anti-friction bearing as a function of the loading.

$$(2-2) \quad L = \left(\frac{C}{F} \right)^p$$

- L = nominal life
(100 km for linear guides or 1 million revolutions for ball screw assemblies)
- C = dynamic load capacity (N)
- F = bearing loading and/or sum of external force components acting on the bearing (N)
- p = exponent of the nominal life equation, depending on the type of rolling element (-)

This calculation method is based on the Hertz theory, which enables statements to be made about the maximum surface pressure of two curved bodies. The dynamic load capacities are calculated from this, dependent on the surface factors.

In the case of linear motion guides, the life expectancy is related to the distance traveled and with ball screw drives to the number of revolutions. For both systems the life expectancy calculation is similar to the method given in DIN ISO 281 for rolling bearings. This calculation method is based on a fatigue theory which draws on the alternating shear stress hypothesis.

- p = 3
for linear ball bearings and ball screw assemblies
- p = 10/3
for linear roller bearings

The load capacities resulting from calculation according to the standard are often clearly exceeded by Rexroth in statistically-confirmed endurance tests – a result of their comprehensive system know-how.

Probability of survival

An individual bearing's probability of survival is the probability that the bearing will achieve or exceed a certain service life. The probability of survival is therefore a percentage of a group of

identical bearings that have the same calculated life expectancy when operating under identical conditions.

2 Principles

2.4 Life expectancy

2.4.1 Calculation principles

Nominal life L_{10}

The nominal life L_{10} is understood as being the achievable calculated life expectancy with a probability of survival of 90%. This means that 90% of a sufficiently large quantity of identical bearings

achieve or exceed the theoretical life expectancy before material fatigue occurs.

Modified life expectancy L_{na}

If this probability is too low, the calculated life expectancy must be reduced by a certain factor, this being the life expectancy coefficient a_1 for the

probability of survival. This results in the modified life expectancy L_{na} .

$$(2-3) \quad L_{na} = a_1 \cdot \left(\frac{C}{F}\right)^p$$

$p = 3$
for linear ball bearings and ball screw assemblies
 $p = 10/3$
for linear roller bearings

- L_{na} = modified life expectancy
(100 km for linear guides or 1 million revolutions for ball screw assemblies)
- a_1 = life expectancy coefficient
- C = dynamic load capacity (N)
- F = bearing loading and/or sum of external force components acting on the bearing (N)
- p = exponent of the nominal life equation, depending on the type of rolling element (-)

Probability of survival	(%)	90	95	96	97	98	99
a_1	(-)	1.00	0.62	0.53	0.44	0.33	0.21

2.4.1.2 Dynamic and static load capacities

Standards

Dynamic load capacities are used as a basis for calculating life expectancy.

National and international standards establish the methods for calculating dynamic and static load capacities.

ISO 14728

Static load capacities are required for checking the static load safety factor.

- Profiled rail systems and linear bushings and shafts per ISO 14728 Parts 1 and 2

DIN 69051

Specific details are provided in the corresponding product catalogs. Detailed descriptions of the calculation method are provided in the subsections for the specific guide and drive units.

- Ball screw assemblies per DIN 69051 Part 4

Dynamic load capacity C

The dynamic load capacity C represents the loading at which a sufficiently large number of identical bearings achieves the nominal life expectancy. In the case of ball screw drives and rotating anti-friction bearings, the nominal life expectancy is

1 million revolutions. The dynamic load capacity of linear motion guides, such as profiled rail systems and linear bushings and shafts, is based on a nominal life expectancy of 100 km travel.

Static load capacity C_0

The static load capacity C_0 must be understood as a loading that causes a permanent deformation of the rolling element and the running track, which corresponds to approximately to 0.0001 times the

rolling element's diameter. Experience has shown that deformations of such small magnitude do not adversely affect the smoothness of operation.

2 Principles

2.4 Life expectancy

2.4.1 Calculation principles

Conversion factors for dynamic load capacities

Some linear guide manufacturers base their dynamic load capacity on a nominal life expectancy of 50 km instead of 100 km. This leads to different and, as a rule, higher load capacities

ity values that are not directly comparable. The following conversions are required in order to compare the values:

Ball rolling element factor 1.26

■ For ball rolling elements, multiply the dynamic load capacity C relating to 100 km by a factor of 1.26.

Roller rolling element factor 1.23

■ For roller rolling elements, multiply the dynamic load capacity C based on 100 km by a factor of 1.23.

Derivation of the conversion factors:

Basis 100 km	Basis 50 km
$L = \left(\frac{C_{100}}{F}\right)^p \cdot 100 \text{ km}$	$L = \left(\frac{C_{50}}{F}\right)^p \cdot 50 \text{ km}$
$\Rightarrow \left(\frac{C_{100}}{F}\right)^p \cdot 100 \text{ km} = \left(\frac{C_{50}}{F}\right)^p \cdot 50 \text{ km}$ $\left(\frac{C_{50}}{C_{100}}\right)^p = \frac{100 \text{ km}}{50 \text{ km}}$ $C_{50} = \sqrt[p]{\frac{100 \text{ km}}{50 \text{ km}}} \cdot C_{100}$ $C_{50} = \sqrt[p]{2} \cdot C_{100}$	
For ball rolling elements	For roller rolling elements
$p = 3 \Rightarrow C_{50} = \sqrt[3]{2} \cdot C_{100}$ $C_{50} = 1.26 \cdot C_{100}$	$p = \frac{10}{3} \Rightarrow C_{50} = \sqrt[\frac{10}{3}]{2} \cdot C_{100}$ $C_{50} = 1.23 \cdot C_{100}$

L = nominal life
(100 km for linear guides or 1 million revolutions for ball screw assemblies)

C_{50} = dynamic load capacity at a nominal life expectancy of 50 km (N)

C_{100} = dynamic load capacity at a nominal life expectancy of 100 km (N)

F = bearing loading or sum of external force components acting on the bearing (N)

p = exponent of the nominal life equation:
 $p = 3$ for ball rolling elements
 $p = 10/3$ for roller rolling elements

Non-convertible static load capacities

These manufacturers' static load capacities are also higher than those of Rexroth products. The values cannot be converted because the load

capacity calculations were based on different values from those specified in the standards.

2 Principles

2.4 Life expectancy

2.4.1 Calculation principles

2.4.1.3 Equivalent load on bearing

A linear motion system is subjected to different types of loading during a travel cycle. In order to simplify life expectancy calculations, these loads are summarized into one single load known as the equivalent load on the bearing. The shorter form "equivalent load" may also be used as a synonym.

The equivalent load comprises two aspects, which are described in more detail in the following paragraphs:

- Equivalent static load
- Equivalent dynamic load

Loads summarized in the equivalent load:

- Loads acting in different directions
- Loads acting in different discrete time or travel steps (phases)

Equivalent static load

The equivalent static load has to be determined when loads from several directions and moments simultaneously act on a linear motion system while it is at rest. The calculation formula for the equivalent load differs depending upon the design. Please refer to the corresponding details for the individual products.

Equivalent dynamic load

The equivalent dynamic load is determined when loads alternate frequently during operation. Alternating loads may, for instance, be positive and negative acceleration forces as well as process forces.

Cycle

For calculating the equivalent dynamic load, first of all a representative cycle (cross section) must be established, with the loads, travel distances, speeds and accelerations to be expected. This cycle is divided into n phases in which the loads and speeds are constant. If this is not the case, a mean or equivalent value must be established for the respective phase.

Cycles are distance-dependent for linear motion guides and time-dependent for ball screw drives. A cycle usually consists of a complete travel cycle (forward and back), which is divided into individual time phases.

2 Principles

2.4 Life expectancy

2.4.1 Calculation principles

Determination of the equivalent dynamic load

The equivalent dynamic load for a cycle consisting of different phases is determined as follows: The respective individual loads are multiplied by the distance covered (expressed as a percentage of total distance covered) in the separate phases and thereby converted to an equivalent load.

When calculating with time phases (discrete time steps), changing velocities and speeds must also be factored in.

The procedures for determining cycles and calculating discrete travel and time steps are given below.

Equivalent dynamic load of linear guides with discrete travel steps

Calculation of the equivalent dynamic load for linear motion guides:

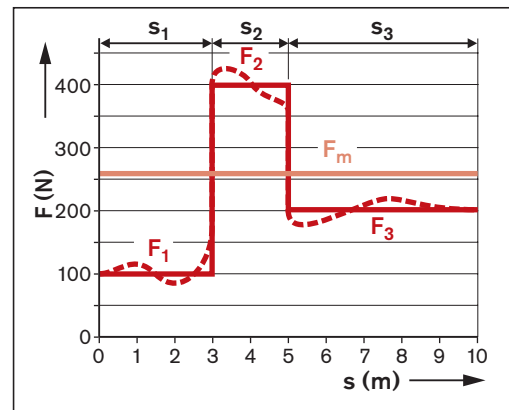
$$(2-4) \quad F_m = \sqrt[p]{|F_1|^p \cdot \frac{q_{s1}}{100\%} + |F_2|^p \cdot \frac{q_{s2}}{100\%} + \dots + |F_n|^p \cdot \frac{q_{sn}}{100\%}}$$

$p = 3$ for linear ball bearings
 $p = 10/3$ for linear roller bearings

F_m = equivalent dynamic load (N)
 $F_1 \dots F_n$ = load in phase 1 ... n (N)
 $q_{s1} \dots q_{sn}$ = discrete travel steps for phases 1 ... n (%)

Distance-dependent cycle

Distance-dependent load cycle (example)



--- Actual force profile
 — Approximated force profile
 — Average force over the entire cycle (equivalent dynamic load F_m)

Cycle for phases 1 to 3 with different loads F_1 to F_3 (simplified illustration without return travel)

Discrete travel steps

Determination of the discrete travel steps: The discrete travel steps q_{sn} in percentages per phase are required in order to calculate the equivalent dynamic load on the bearing.

The entire cycle travel s must therefore be divided into phases with discrete travel steps s_n . A constant load F_n and a constant velocity v_n act during each phase.

Calculation of the discrete travel steps:

$$(2-5) \quad q_{sn} = \frac{s_n}{s} \cdot 100\%$$

$$(2-6) \quad s = s_1 + s_2 + \dots + s_n$$

q_{sn} = discrete travel step in phase n (%)
 $s_1 \dots s_n$ = travel for phase n (mm)
 s = travel for all phases (mm)

2 Principles

2.4 Life expectancy

2.4.1 Calculation principles

Equivalent dynamic load of ball screw drives

Calculation of the equivalent dynamic load for ball screw drives

$$(2-7) \quad F_m = \sqrt[p]{|F_1|^p \cdot \frac{|n_1|}{n_m} \cdot \frac{q_{t1}}{100\%} + |F_2|^p \cdot \frac{|n_2|}{n_m} \cdot \frac{q_{t2}}{100\%} + \dots + |F_n|^p \cdot \frac{|n_n|}{n_m} \cdot \frac{q_{tn}}{100\%}}$$

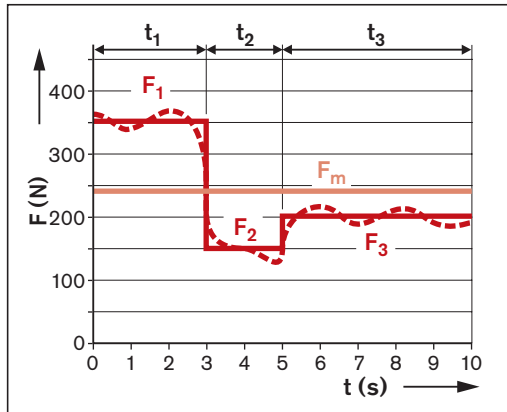
Calculation of the average rotational speed using discrete time steps

$$(2-8) \quad n_m = \frac{|n_1| \cdot q_{t1} + |n_2| \cdot q_{t2} + \dots + |n_n| \cdot q_{tn}}{100\%}$$

- $p = 3$ for ball screw drives
- F_m = equivalent dynamic load (N)
- $F_1 \dots F_n$ = load in phase 1 ... n (N)
- $q_{t1} \dots q_{tn}$ = discrete time step in phases 1 ... n (%)
- $n_1 \dots n_n$ = speed in phase 1 ... n (min^{-1})
- n_m = average speed (min^{-1})

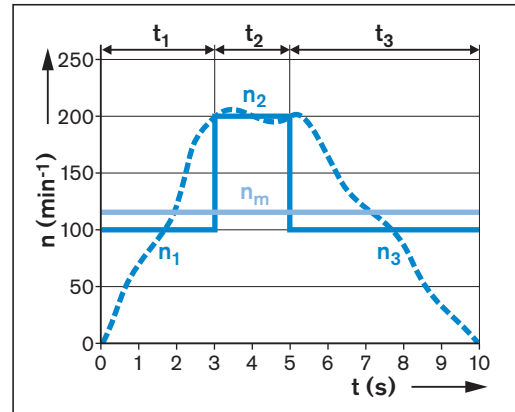
Cycle

Time-dependent load cycle (example)



Cycle for phases 1 to 3 with different loads F_1 to F_3 (simplified illustration without return travel)

Time-dependent speed cycle (example)



Cycle for phases 1 to 3 with different speeds n_1 to n_3 (simplified illustration without return travel)

- Actual force profile
- Approximated force profile
- Average force over the entire cycle (equivalent dynamic load F_m)

- Actual speed profile
- Average speeds in phases 1 to 3
- Average speed over the entire cycle

Discrete time steps

Determination of the discrete time steps: The discrete time steps q_{tn} in percentages per phase are required in order to calculate the equivalent dynamic load on the bearing for ball screw

drives. The entire cycle time t must therefore be divided into phases with discrete time steps t_n . A constant load F_n and a constant speed n_n act during each phase.

Calculation of the discrete time steps

$$(2-9) \quad q_{tn} = \frac{t_n}{t} \cdot 100\%$$

$$(2-10) \quad t = t_1 + t_2 + \dots + t_n$$

- q_{tn} = discrete time step in phase n (%)
- $t_1 \dots t_n$ = time for phase n (s)
- t = time for all phases (s)

2 Principles

2.4 Life expectancy

2.4.1 Calculation principles

2.4.1.4 Static load safety factor

Static load safety factor S_0

The static load safety factor S_0 is required in order to avoid any impermissible permanent deformations of the running tracks and rolling elements. It is the ratio of the static load capacity C_0 to the

maximum load occurring, $F_{0 \max}$, and is always determined using the highest amplitude, even if this is only of very short duration.

$$(2-11) \quad S_0 = \frac{C_0}{F_{0 \max}}$$

S_0 = static load safety factor

C_0 = static load capacity (N)

$F_{0 \max}$ = load (N)

Recommendations for the static load safety factor under different conditions of use

Conditions of use	S_0
Normal conditions of use	1 ... 2
Low impact loads and vibrations	2 ... 4
Moderate impact loads and vibrations	3 ... 5
Heavy impact loads and vibrations	4 ... 6
Unknown load parameters	6 ... 15

Normal conditions of use are defined in section 2.4.2.4.

Irrespective of the static load safety factor, it must be ensured that the maximum permissible loads, as indicated for some linear motion guides, are not exceeded.

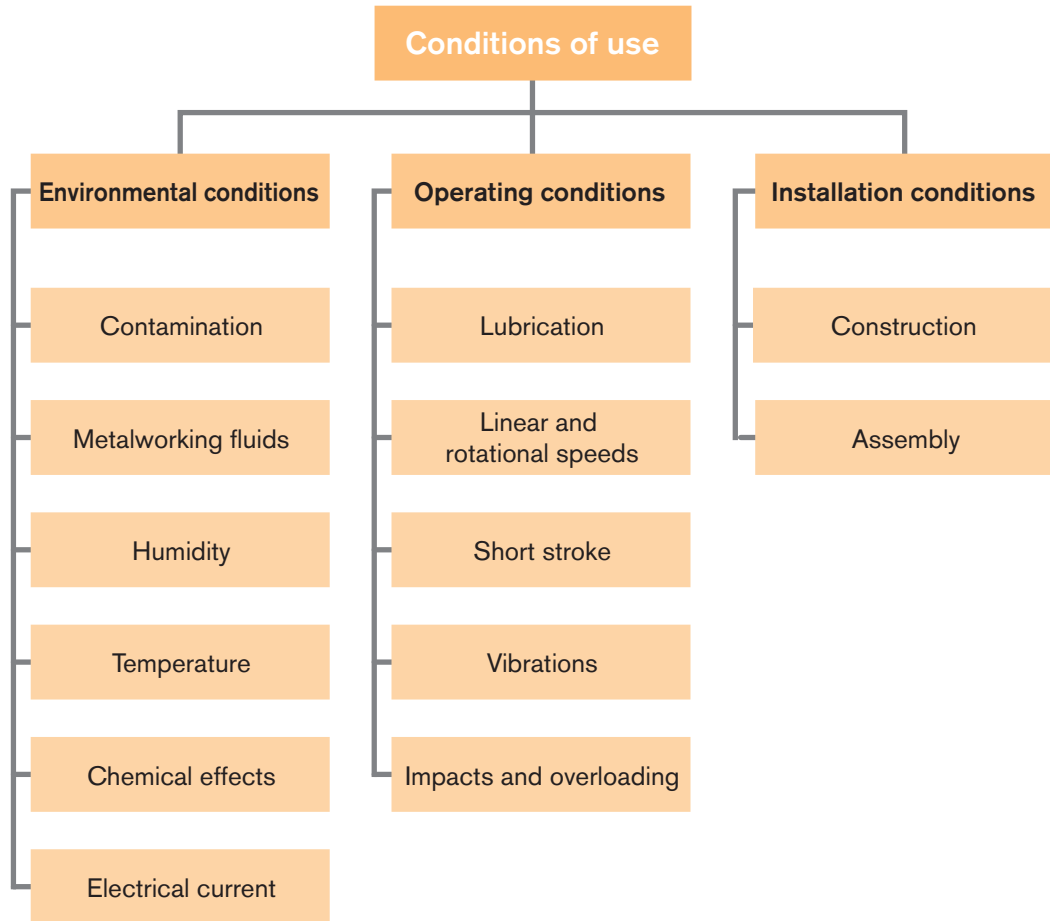
The load-bearing capability of the threaded connections must also be checked. These are frequently weaker than the bearings themselves. The load-bearing capability of linear motion technology components is such that the screws used could be over-stressed.

2 Principles

2.4 Life expectancy

2.4.2 Conditions of use

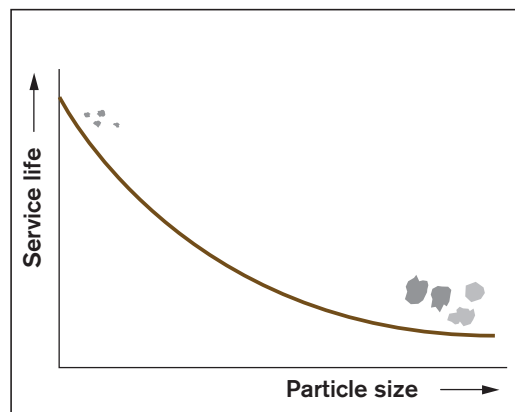
Various conditions affect the life expectancy:



2.4.2.1 Environmental conditions

Contamination

This is understood as being contamination in the operating environment, such as shavings or dust, which can work its way into the products. This contamination leads to the running tracks and bearing surfaces being subject to greater wear and the nominal life perhaps not being achieved. Coarse contamination with shavings from cutting operations can block the rolling elements, resulting in running track damage and fracture of the plastic components. The appropriate seals for the degree of contamination must be selected.



Influence on service life of the size of any dirt particles working their way into the bearing

2 Principles

2.4 Life expectancy

2.4.2 Conditions of use

Metalworking fluids

Metalworking fluids are used in machine tools to cool and lubricate the tool and the workpiece. There are different types of metalworking fluids: non-water-miscible fluids (straight oils), water-miscible fluids (concentrates), and fluids mixed with water (emulsions, solutions).

The guide unit's lubrication is disturbed by the penetration of metalworking fluid, i.e. the lubricant is altered and gradually washed out. The guide unit must therefore be operated with appropriate protection in place.

Aqueous metalworking fluids can also cause corrosion. It has been shown that the water contained in the metalworking fluid evaporates over time and can condense as water on the running tracks and rolling elements. This can lead to premature breakdown because of corroded components.

Preventive measures:

- Use of corrosion-resistant steel
- Hard-chroming of the components
- Reinforcing the seals on the guides
- Adjusting the lubrication
- Execution of scheduled cleaning and lubricating strokes

Humidity

When humidity penetrates into the guide units and drive components, corrosion also occurs. The preventive measures are the same as those for protection from metalworking fluids.

Temperature

There is a permissible operating temperature range for all guide units. The temperature of the elements themselves is ultimately the determining factor, i.e., the incorporated plastic materials, for instance, can lose their mechanical characteristics at inadmissible temperature levels. The maximum operating temperatures are shown in the respective product catalogs and in the following chapters.

When the temperature rises above the upper limit or falls below the lower limit, high stresses can be produced in the components. This can lead to premature system breakdown.

Damage may also be caused as a result of the different heat expansion coefficients of steel and plastic. Plastic deformation, cracking and rupturing of the plastic parts may also lead to premature system breakdown.

When considering the temperature, the whole machine, from bed to attachments, must be taken into account in addition to the linear motion technology components. The different heat expansion coefficients of materials, manufacturing tolerances and any misalignments and temperature gradients in the construction can produce high additional loads as a result of distortive stresses.

Chemical effects

Chemicals can attack the steel and plastic parts of the guide units and/or drive components. The surfaces of the running tracks and the rolling elements are particularly sensitive.

More details of the compatibility of individual chemicals with the guide units can be obtained from Rexroth. If exposure to chemicals cannot be avoided, protective measures must be taken after appropriate consultation with Rexroth.

Electrical current

If electricity flows through the anti-friction bearing elements, this can lead to abrasion and accelerated corrosion. Even an amperage within the mA

range can cause damage to the rolling contact surfaces. This type of damage to the anti-friction bearing is termed ridge formation or ridging.

2 Principles

2.4 Life expectancy

2.4.2 Conditions of use

2.4.2.2 Operating conditions

Lubrication	Insufficient lubrication leads to excessive wear of the running track and rolling element surfaces. Visible signs of this wear include discolorations on the bearing surfaces.	The lubrication guidelines and advice (see section 2.5.4 and the product catalogs) must be followed to prevent unnecessary shortening of the service life.
Linear and rotational speeds	The maximum permissible linear or rotary speed is specified for each product. If these limits are exceeded, the plastic parts in particular can be damaged. All the relevant details for this are provided further on in the chapters on the individual products.	For ball screw drives, resonance, which occurs during operation close to the critical speed, must be avoided because this can destroy the system.
Short stroke	Short-stroke applications are applications in which not all of the rolling elements recirculating within the bearing component arrive in the load-bearing zone during execution of the stroke. The consequences can be premature material fatigue and therefore breakdown of the guide units.	The definition is different for each product and is discussed in the corresponding sub-chapter and in the product catalogs. Short-stroke applications must be taken into account when calculating the life expectancy.
Vibrations	Vibrations in the machine are caused either by the process (operating forces) or by the drive (regulation oscillations and imbalances). Process forces can be, for example, cutting forces in machine tools. Oscillations may be generated by the drive unit when regulating the motor during positioning.	Vibration can lead to contact corrosion, overloading and excessive wear in the affected area. The damaged surfaces can greatly reduce the service life of the components.
Impacts and overloading	Brief, jerky loading peaks can adversely affect the life expectancy of the guide units. They are usually caused by cannoning in the machine or collisions of slides and carriages. This causes high stresses in the machine components. This so-called crash behavior is now taken increasingly into consideration in new machine designs.	Impacts in the dynamic or static state, whose peak loads are higher than the maximum permissible loads can damage the components. Overloading can cause plastic deformation (e.g. dents in the running tracks as a result of massive forces acting on the rolling elements) or fractures.

2 Principles

2.4 Life expectancy

2.4.2 Conditions of use

2.4.2.3 Installation conditions

Construction

The components may be subject to additional preloading if the dimensions of adjoining structures are outside the permissible tolerances for installation. This increases the internal loading, which shortens the life expectancy. This additional loading is often not detectable by increased friction.

The design notes and tolerances indicated in this handbook and the respective product catalogs must therefore be observed.

Mounting

The same applies to incorrect mounting of the components. This can also cause internal stresses. It is therefore essential to follow the guidelines given in the mounting instructions

and the product catalogs. All mounting and assembly work must be performed with care and due attention to cleanliness.

2.4.2.4 Normal conditions of use

Rexroth recommends that all guide and drive units be used under normal environmental, operating and installation conditions.

The following can be regarded as normal conditions of use which will not have any influence on the life expectancy.

Influencing factors	Normal conditions of use
Environmental conditions	
Contamination	No contamination
Metalworking fluids	No exposure to metalworking fluids
Humidity	Use in a dry environment
Temperature	Use at room temperature
Chemical effects	No exposure to chemicals
Electrical current	No electrical current flowing through the components
Operating conditions	
Lubrication	Adequate lubrication
Linear and rotational speeds	Maximum permissible linear or rotational speeds are not exceeded
Short stroke	No short stroke
Vibrations	No vibrations
Impacts and overloading	No impacts
Installation conditions	
Construction	Design notes and guidelines are observed
Mounting	Installation in accordance with mounting instructions





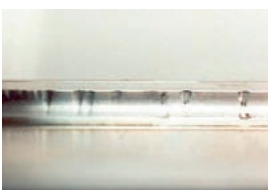

If the environmental, operating and installation conditions differ from those stated above, Rexroth, with its many years of experience, is available for consultation.

The table in section 2.4.3 contains an illustrated overview of the possible effects of the different influencing factors that can cause damage.

2 Principles

2.4 Life expectancy

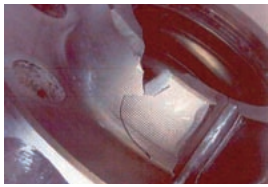
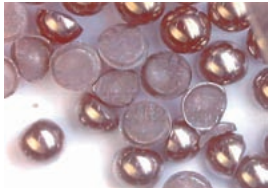




2.4.3 Damage profiles

Damage type	Damage photo	Possible causes of failure	Remedies
Corrosion		<ul style="list-style-type: none"> ■ Unfavorable environmental influences ■ Exposure to metalworking fluids ■ Aggressive media (acids etc.) ■ High humidity (saline mist) 	<ul style="list-style-type: none"> ■ Adapt to suit the environment ■ Use corrosion-protected version ■ Use appropriate sealing systems ■ Use appropriate covers ■ Optimize lubrication
Rolling element blocking		<ul style="list-style-type: none"> ■ Shavings contamination ■ Dust contamination ■ Inadequate lubrication ■ Rolling element fracture ■ Defective recirculation piece 	<ul style="list-style-type: none"> ■ Use appropriate sealing systems ■ Use appropriate covers ■ Ensure adequate lubrication ■ Avoid overloading ■ Check the application
Pronounced darkening		<ul style="list-style-type: none"> ■ Inadequate lubrication (high temperatures) 	<ul style="list-style-type: none"> ■ Optimize lubrication
Pitting Peeling/flaking		<ul style="list-style-type: none"> ■ Rolling element fatigue ■ End of service life 	<ul style="list-style-type: none"> ■ Reduce the loads ■ Use a heavier duty component ■ Check the application
Plastic indentations by rolling elements		<ul style="list-style-type: none"> ■ Static overload 	<ul style="list-style-type: none"> ■ Use a heavier duty component ■ Reduce the loads
Destruction of recirculation zone (e.g. ball runner block)		<ul style="list-style-type: none"> ■ Excessive speeds ■ Collisions ■ Rolling element blocking because of contamination 	<ul style="list-style-type: none"> ■ Reduce the speeds ■ Avoid overloading ■ Avoid collisions ■ Use appropriate sealing systems ■ Use appropriate covers

2 Principles

2.4 Life expectancy

2.4.3 Damage profiles

Damage type	Damage photo	Possible causes of failure	Remedies
Destruction of component body (e.g. ball screw drive)		<ul style="list-style-type: none"> ■ Overloading ■ Collisions ■ Flawed material, manufacturing error 	<ul style="list-style-type: none"> ■ Reduce the loads ■ Avoid collisions ■ Use a heavier duty component
Destruction of rolling elements		<ul style="list-style-type: none"> ■ Overloading ■ Collisions ■ Flawed material, manufacturing error 	<ul style="list-style-type: none"> ■ Avoid overloading ■ Reduce the loads ■ Use a heavier duty component ■ Avoid collisions
Rolling marks on the rolling elements (e.g. balls)		<ul style="list-style-type: none"> ■ Wear ■ Rolling element fatigue ■ End of service life 	<ul style="list-style-type: none"> ■ Optimize lubrication ■ Use a heavier duty component
Fatigue fractures (e.g. ball screw)		<ul style="list-style-type: none"> ■ Rotary bending stresses ■ Vibration stresses ■ Expansion stresses ■ Alternating stresses 	<ul style="list-style-type: none"> ■ Avoid rotary bending (correct any misalignments) ■ Avoid resonance oscillations
Destroyed end caps (e.g. roller runner blocks)		<ul style="list-style-type: none"> ■ Rolling element blocking because of contamination ■ Collisions 	<ul style="list-style-type: none"> ■ Use end seals ■ Use appropriate covers ■ Avoid collisions
Local flattening of the rolling element (e.g. roller)		<ul style="list-style-type: none"> ■ Slip ■ Contamination 	<ul style="list-style-type: none"> ■ Adjust the preload to suit the expected loads and accelerations ■ Use appropriate covers and seals

2 Principles

2.5 System technology

2.5.1 Preload and rigidity

Preload

Preloading increases the rigidity of the overall system. It anticipates the occurrence of elastic deformation of the rolling elements under load, thereby reducing the deflection characteristics of the system as a whole. However, the resistance to movement becomes greater as the preload

increases, and high preloads have a negative effect on the life expectancy. When calculating the nominal life, the preload must be taken into account as an additional load on the bearing.

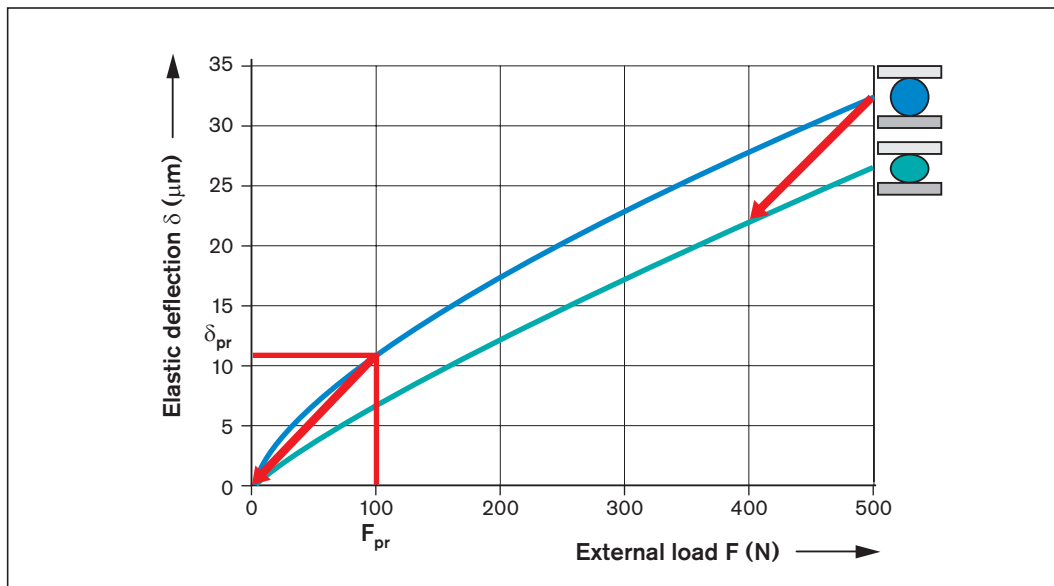
Example:

Deformation of a ball between two flat plates, with or without preloading, according to the Hertz theory.

Ball diameter = 5 mm

Preload force $F_{pr} = 100$ N

The deflection curve for the preloaded ball can be produced by parallel shifting of the curve for the non-preloaded ball.



Effect of preloading on the elastic deflection

- Ball without preload
- Ball with preload
- δ_{pr} Deflection at preload force F_{pr}
- F_{pr} Preload force

2 Principles

2.5 System technology

2.5.2 Friction

Friction coefficient

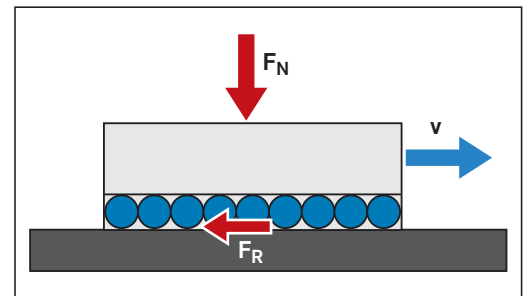
In linear motion technology, the value of the friction coefficient varies according to the system used. The magnitude of the friction force depends primarily on the seals used, the type of rolling

contact, and the loading. The lubrication and the speed also affect the friction.

Friction force

$$(2-12) \quad F_R = \mu \cdot F_N$$

F_R = friction force (N)
 μ = friction coefficient (-)
 F_N = normal force (force perpendicular to the contact area) (N)



Friction force

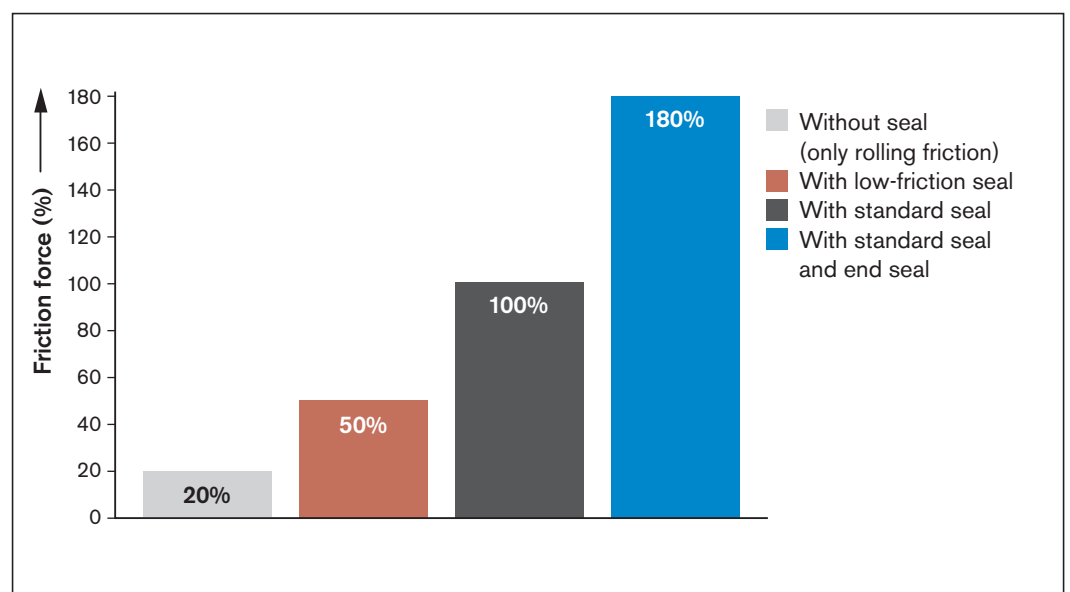
The following table shows the friction coefficients without seals, i.e. the values for rolling friction.

Linear component	Friction coefficient μ without seal	Comment
Linear bushing and shaft	0.001 ... 0.004	Standard linear bushing
Ball rail system	0.002	2-point contact
	0.003	4-point contact
Roller rail system	0.0004	Line contact
Ball screw assembly	0.004	2-point contact
	0.010	4-point contact

Seals and friction forces

Using the ball rail system as an example, the chart below illustrates the effects of different seals on the friction force. It shows that the seal friction

accounts for the largest proportion of the overall friction.



Friction forces of a ball runner block with 2% C preload on a guide rail with mounting hole plugs (example)

2 Principles

2.5 System technology

2.5.3 Sealing

Function of seals Seals prevent dirt, shavings, etc. from working their way into a system and shortening its service life. There are particular types of seals for special applications. The interaction between the seal and the lubricant is described below.

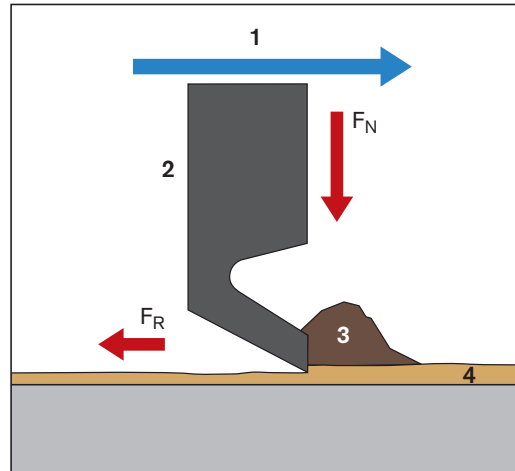
End seals A particular feature of sealing for linear bearings is that wiper seals are used to seal the ends of the components in the direction of motion. Unlike the seals for rotating rolling contact bearings, there is a discharge of lubricant here. The resulting lubricating film ensures that there is little wear on the seals.

Side seals If side seals are present on the linear motion components, these work in exactly the same way as antifriction bearing seals in which the inner side of the seal is separated from the (contaminated) outer side.

Additional seals Depending on the product, additional sealing elements may be required. Auxiliary seals are to be recommended in environments with fine dirt or metal particles and cooling or cutting fluids.

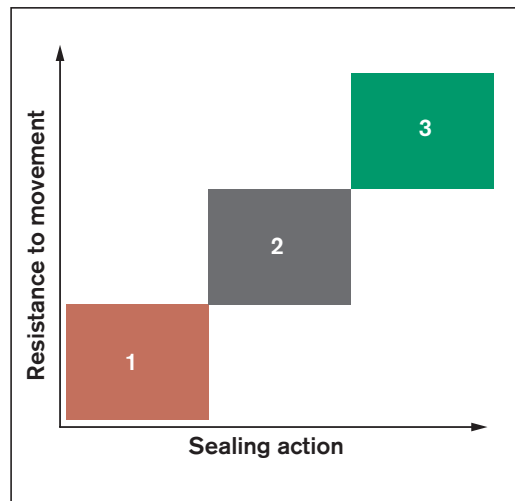
Sealing action The sealing action is influenced by the geometry and material of the seals.

Resistance to movement The chart opposite shows the effect of different varieties of seal on the sealing action and the resistance to movement.



Wiper end seal of a linear motion guide

- 1 Direction of motion
- 2 Inner side of seal
- 3 Contamination
- 4 Lubricant
- F_R Friction force
- F_N Preload force



Relationship between the sealing action and resistance to movement

- 1 Seals with very low friction (low-friction seals)
- 2 Standard seals
- 3 Seals with very good sealing action (reinforced seals)

2 Principles

2.5 System technology

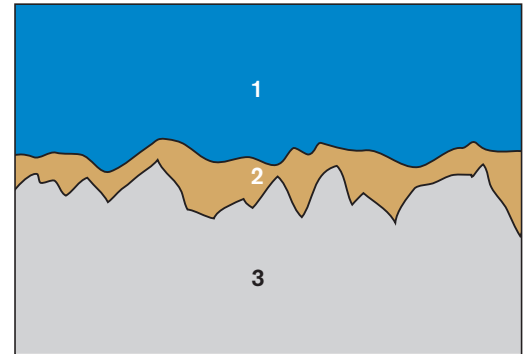
2.5.4 Lubrication

2.5.4.1 Lubrication principles

The lubricant has the task of separating the rolling elements and the running track from each other and thereby minimizing friction and wear. It also prevents corrosion.

Among other things, lubrication prevents wear on the seals. The lubricating film also ensures that the sealing elements slide smoothly.

The choice of lubricant and the specific operating conditions are among the factors determining the service life of linear motion bearings.

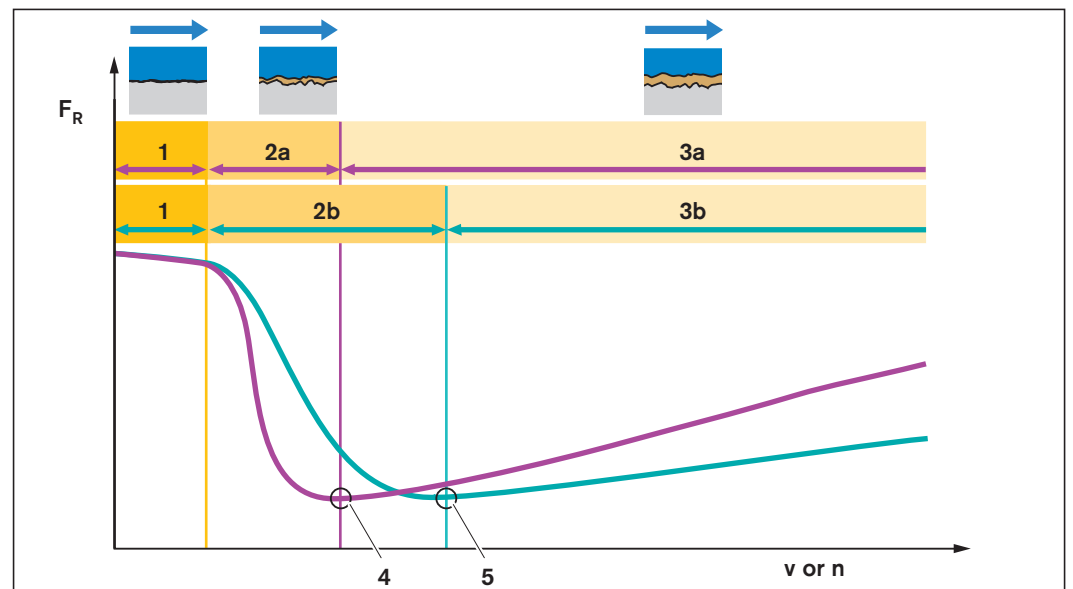


Greatly enlarged representation of the contact area

- 1 Rolling element
- 2 Lubricant
- 3 Running track

Stribeck curve

The Stribeck curve shows the friction force as a function of the viscosity and the speed.



Stribeck curve

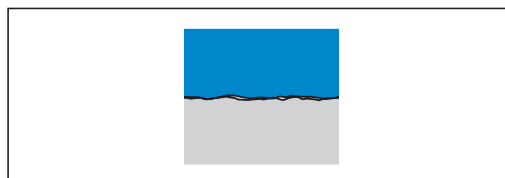
- High viscosity
- Low viscosity
- 1 Boundary lubrication (friction between solid bodies)
- 2a Partial lubrication (mixed friction) at high viscosity
- 2b Partial lubrication (mixed friction) at low viscosity
- 3a Full lubrication (fluid friction) at high viscosity
- 3b Full lubrication (fluid friction) at low viscosity
- 4 Transition point at high viscosity
- 5 Transition point at low viscosity
- F_R Friction coefficient
- v Linear speed
- n Rotary speed

2 Principles

2.5 System technology

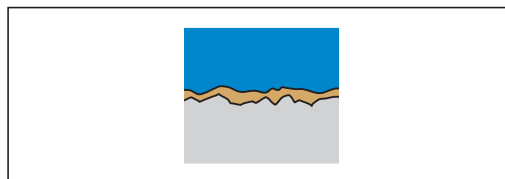
2.5.4 Lubrication

Boundary lubrication At standstill, there is contact between the solid bodies, and solid-body friction prevails. The lubricating film forms as a function of the speed.



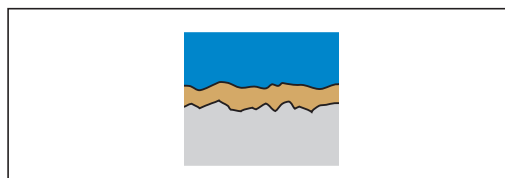
Solid-body friction

Partial lubrication Mixed friction develops during the starting up or running down phase. A thin lubricating film forms, but there is still partial contact between the solid bodies.



Mixed friction

Full lubrication Shortly after the transition point, the optimum condition, fluid friction, is reached. The lubricant's internal friction now determines the increasing friction coefficient as the speed increases further.



Fluid friction

Viscosity Viscosity is a measure of the internal friction of lubricating oils (see also section 2.5.4.2). With low-viscosity oils, the partial lubrication range is greater than with high-viscosity oils. The transition point is only reached at a higher speed. Subsequently, the curve does not rise as steeply, since there is less internal friction at lower viscosity.

2 Principles

2.5 System technology

2.5.4 Lubrication

2.5.4.2 Lubricants

Linear motion components can be lubricated with grease (grease, liquid grease) or oil.

Dry lubricants or lubricants containing solid particles should never be used in Rexroth products.

Grease lubricants

Grease lubricants consist of the basic oil (e.g. a mineral oil base), a thickener (e.g. lithium soap) and various additives (e.g. against corrosion). Because of their thick consistency, grease lubricants help to prevent the penetration of contaminants into the linear motion guides and support the action of the sealing and wiping elements.

Greases provide long-lasting lubrication specifically at the contact point between the friction partners and enable very long lubrication intervals up to lubrication for life.

Liquid grease

Liquid greases have a soft fluid consistency and can be conveyed much more easily than harder types of grease. They are therefore frequently used in central lubrication systems.

The central lubrication system's geometrical characteristics must also be taken into account in addition to the consistency class of the grease.

Consistency NLGI grades

Greases are classified into NLGI grades (National Lubricating Grease Institute) according to their consistency. They are a measure of the lubricant's stiffness.

Consistency classification of lubricants per DIN 51818:

Type	NLGI grade	Consistency	Worked penetration (0.1 mm)	Dynalub
Liquid grease	000	Fluid	445–475	
	00	Moderately fluid	400–430	Dynalub 520
	0	Very soft	355–385	
Grease	1	Soft	310–340	
	2	Moderately soft	265–295	Dynalub 510
	3	Semi-fluid	220–250	
	4	Semi-hard	175–205	
	5	Hard	130–160	
	6	Very hard	85–115	

Grease lubricants from Rexroth are highlighted in the table. For additional specifications, please refer to the "Recommended grease types" section below.

2 Principles

2.5 System technology

2.5.4 Lubrication

Recommended grease types

With its Dynalub greases, Rexroth offers lubricants specially designed for linear motion guides and ball screw drives:

Grease type	Dynalub 510	Dynalub 520
Designation according to DIN 51825 according to DIN 51826	KP2K-20	GP00K-20
NLGI grade according to DIN 51818	NLGI 2	NLGI 00

Oil lubricants

Oil lubricants distribute more evenly than greases and therefore reach the contact points better. They also have a good capacity to dissipate frictional heat. However, it must be borne in mind that lubricating oils are dragged out more easily

than grease. Oil lubricants are classified into various grades, as specified in DIN 51519, according to their viscosity.

Recommended oil types

- Lubricating oil CLP, CGLP per DIN 51517
- Viscosity ISO VG 220 per DIN 51519
- Shell Tonna S 220
- The recommendations in the product catalogs should also be followed.

Preservative oils

Preservative oils for protection against corrosion are not lubricating oils. It is essential to check their compatibility with the lubricant being used.

Lubricant application methods

Depending on the linear motion product, the required lubricant can be applied using a manually operated grease gun, a lubricant dispenser or an appropriate central lubrication system. Oil-air lubrication or oil-mist lubrication is sometimes used for special applications.

Compatibility

Compatibility with any incorporated plastic parts must always be taken into account when using lubricants.

No mixing of lubricants

The lubricants used must always be of the same type.

Special environmental conditions

If the equipment is to be used in areas with special environmental conditions, special testing and possibly a special lubricant may be required.

These environmental conditions are, for instance:

- Clean rooms
- Vacuum
- Food industry
- Direct exposure to metalworking fluids or aggressive chemicals
- Extreme temperatures

In these cases please consult Rexroth.

2 Principles

2.5 System technology

2.5.4 Lubrication

2.5.4.3 Lubrication intervals

To ensure that the linear motion components will function perfectly, the prescribed lubrication intervals and quantities must be observed.

Specific recommendations are given in the Lubrication section of the corresponding product catalogs.

Initial lubrication

The initial (basic) lubrication of the linear motion guides and ball screw drives is of particular importance. Linear motion components must not be put into operation without initial lubrication. The

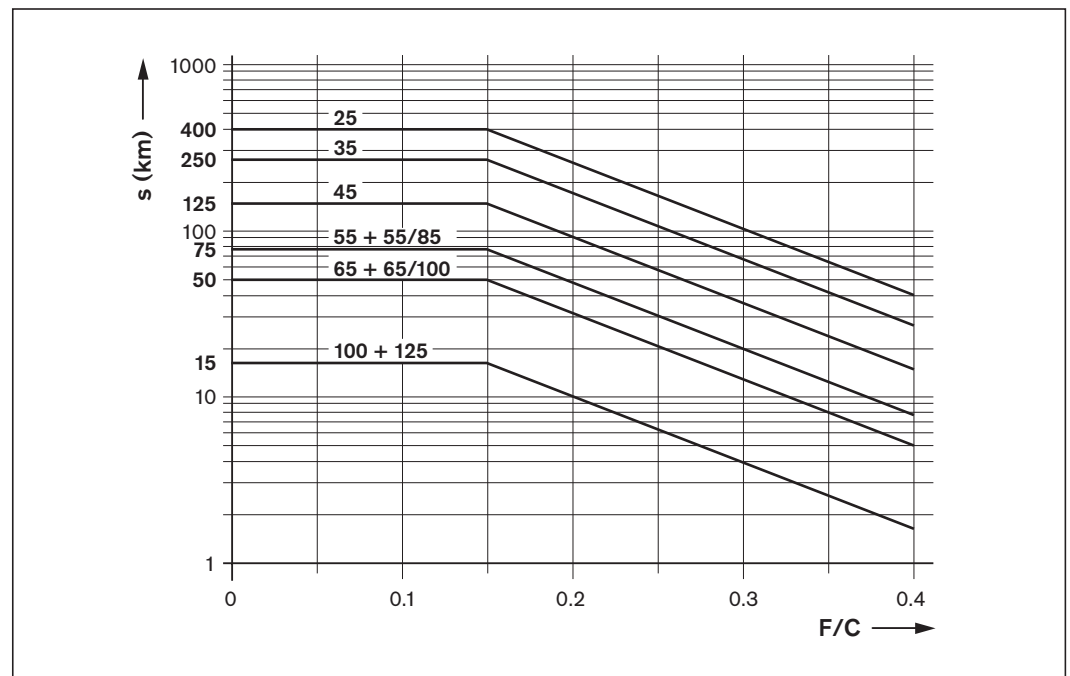
corresponding product catalogs show the lubrication quantities required. If the basic lubrication is applied in-factory before shipment, no initial lubrication by the user is required.

In-service lubrication

The lubrication intervals and quantities for relubrication are also shown in the product catalogs. Shorter lubrication intervals may be required in the case of environmental influences such as contamination, the use of metalworking fluids, vibrations, impact loads, etc.

The lubrication intervals are also load-dependent, i.e. the intervals shorten as the loading increases.

The graph shows an example of the load-dependent lubrication intervals for lubrication with grease.



Lubrication recommendations from the Roller Rail Systems catalog

s = relubrication interval expressed as travel (km)
 C = dynamic load capacity (N)
 F = equivalent dynamic load (N)

2 Principles

2.5 System technology

2.5.5 Accuracy

Geometric accuracy is a decisive production and selection criterion in linear motion technology. The level of accuracy required will depend on the specific application. For optimum results, the level of accuracy to be met must first of all be specified. The higher the level of accuracy

required, the higher the demands will be – both on the linear motion components used and on the overall structure. The various levels of accuracy are defined in different standards.

2.5.5.1 Accuracy levels in guides and drive units

Profiled rail systems, linear bushings and shafts, and ball screw drives are categorized according to accuracy classes or tolerance grades respectively.

Accuracy classes

Profiled rail systems are specified according to accuracy classes. These classes are defined by different levels of maximum tolerances for differences in height and width. They have been standardized in DIN 645 but have been extended

in the meantime with higher accuracy classes as specified by linear motion guide manufacturers. For more detailed information, see Chapter 3.

Tolerance grades

Linear bushings and shafts are specified according to tolerance grades. These grades are standardized under ISO 13012 and ISO 10285. Ball screw drives are also specified according to tolerance grades. In this case, the tolerance

grades specify the permissible travel deviation and travel variation, as defined in DIN 69051-3. Detailed information is provided in the relevant product catalogs.

2.5.5.2 Accuracy types in linear motion systems

Various types of accuracy are specified for linear motion systems:

Absolute accuracy

Absolute accuracy is the discrepancy between an expected target position and the average value for the actual position, resulting from approaching

the target position from different directions (multi-directional motion).

Positioning accuracy

The positioning accuracy is the maximum deviation of the actual position from the target position, in accordance with VDI/DGQ 3441.

Potential influencing factors may be:

- The accuracy of the linear motion unit, the gearbox, the motor and the measuring system
- Ball screw pitch errors
- The play in the system
- The controller or its parameter settings

Repeatability

The repeatability indicates how precisely a linear motion system positions itself when approaching a position repeatedly from the same direction

(unidirectional motion). Repeatability can also be considered as the deviation of the actual position from the target position.

2 Principles

2.6 Product overview

Rexroth's products can be grouped as follows:

- Profiled Rail Systems
- Linear Bushings and Shafts
- Precision Ball Screw Assemblies
- Linear Motion Systems

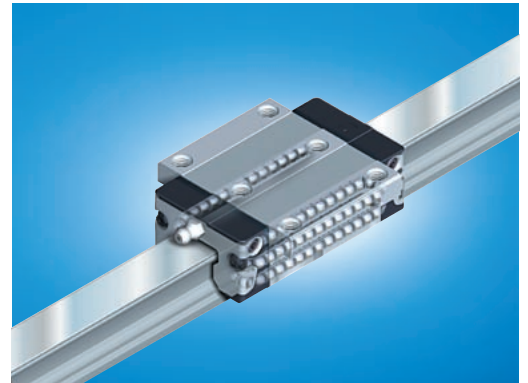
Profiled rail systems

In profiled rail systems, balls, rollers and cam rollers are used as the rolling elements. Because of their high load-bearing capability and their great rigidity they are suitable for almost all tasks requiring precise linear motion. In addition, the guide rails and runner blocks have built-in interchangeability.

This group includes:

- Ball Rail Systems
- Roller Rail Systems
- Cam Roller Guides

Detailed information on Profiled Rail Systems can be found in Chapter 3.

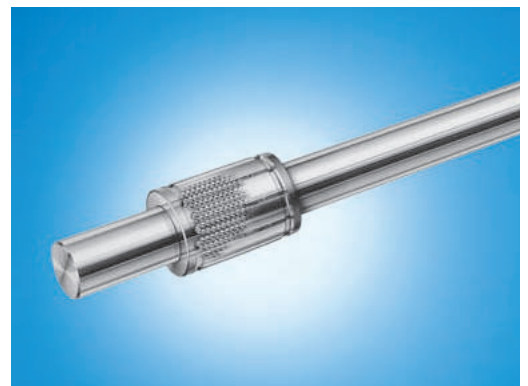


Ball rail system

Linear bushings and shafts

Linear bushings and shafts are rolling-contact linear motion guides. The linear bushings run on hardened and ground precision steel shafts and are available in numerous types, designs and sizes to suit a wide variety of applications. In addition to closed-type linear bushings for self-supporting shafts, open-type bushings are available for high loads and very long guides, with the shaft being supported along its entire length. In comparison to other guides, linear bushings have an additional degree of freedom in the circumferential direction and can compensate for inaccuracies in the mounting base. Linear bushings and shafts are often referred to as round guides.

Detailed information on Linear Bushings and Shafts can be found in Chapter 4.



Linear bushing and shaft

2 Principles

2.6 Product overview

Precision ball screw assemblies

Ball screw assemblies are rolling-contact drive transmission components for converting rotary motion into linear motion. They operate with a high degree of precision and are suitable for high-speed applications. A broad selection of precision screws and zero-backlash preloaded or adjustable-preload single and double nuts is available for all feed, positioning and transport tasks.

Detailed information on Precision Ball Screw Assemblies can be found in Chapter 5.



Precision ball screw assembly

Linear motion systems

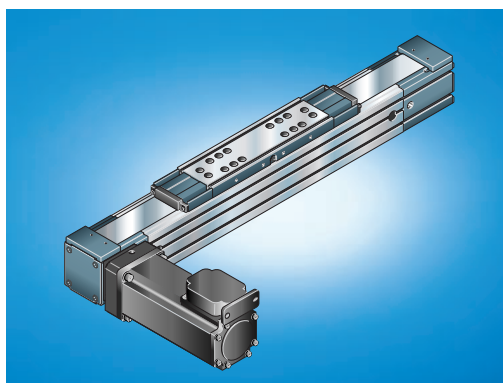
Linear motion systems are ready-to-install systems which essentially comprise a linear guide unit and a drive unit. All of the systems are also available complete with motor, controller, control system and measuring system.

The use of linear motion systems facilitates the design, assembly and commissioning of machines. Individual performance characteristics such as precise movement of loads or fast travel are optimized as necessary for each application. These complete systems provide solutions for a very wide variety of applications.

This group includes:

- Linear Modules
- Compact Modules
- Linear Motion Slides

Detailed information on Linear Motion Systems can be found in Chapter 6.






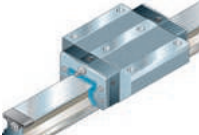
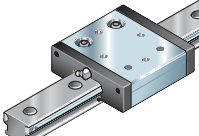
Linear module

3 Profiled rail systems

3.1 Principles

3.1.1 System technology

Profiled rail systems are the guides of choice for applications requiring especially high precision, low maintenance, low wear and low friction, as well as highly accurate positioning. The Rexroth range includes the following profiled rail systems:

Product name	Abbreviation	Section
Ball rail system 	BRS	3.2
Miniature ball rail system 	Mini BRS	3.3
eLINE ball rail system 	eLINE BRS	3.4
Roller rail system 	RRS	3.5
Cam roller guide 	CRG	3.6

3 Profiled rail systems

3.1 Principles

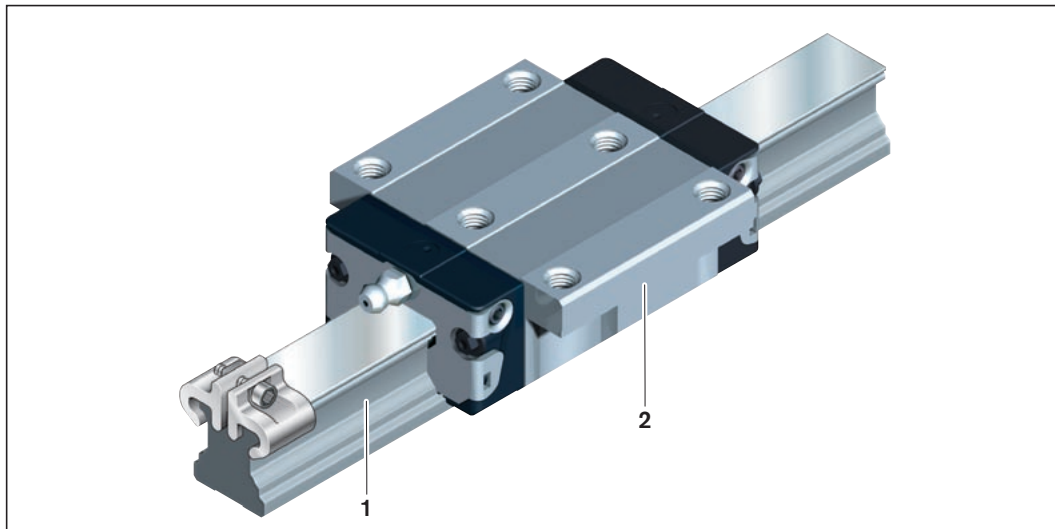
3.1.1 System technology

3.1.1.1 Structural design of a profiled rail system

Runner block and guide rail

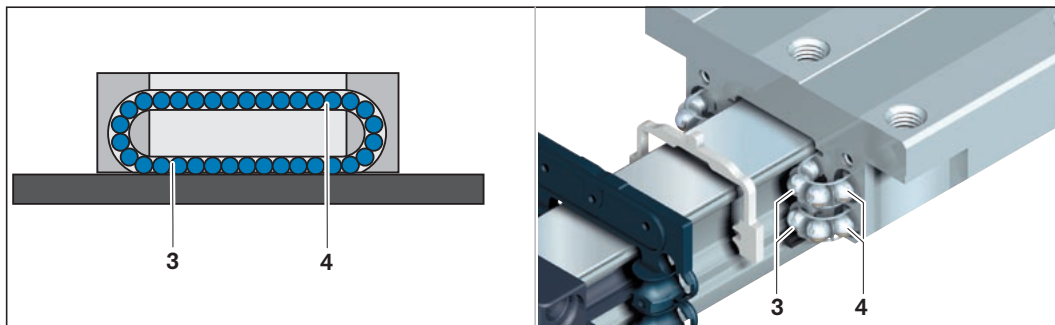
Profiled rail systems consist of a runner block and a guide rail. The runner block comprises several parts. It has one or more rolling element circuits with a load bearing zone and a return zone. In the load-bearing zone the rolling elements transmit the load from the runner block to the rail, and vice

versa. In the return zone the rolling elements are not subjected to loading and are guided around the circuit and back into the load-bearing zone. This recirculation of the rolling elements allows unlimited linear travel.



Profiled rail system (example: ball rail system)

- 1 Guide rail
- 2 Runner block



Rolling element load-bearing zone and return zone in a schematic representation (left) and as implemented in a ball rail system

- 3 Rolling element load-bearing zone
- 4 Rolling element return zone

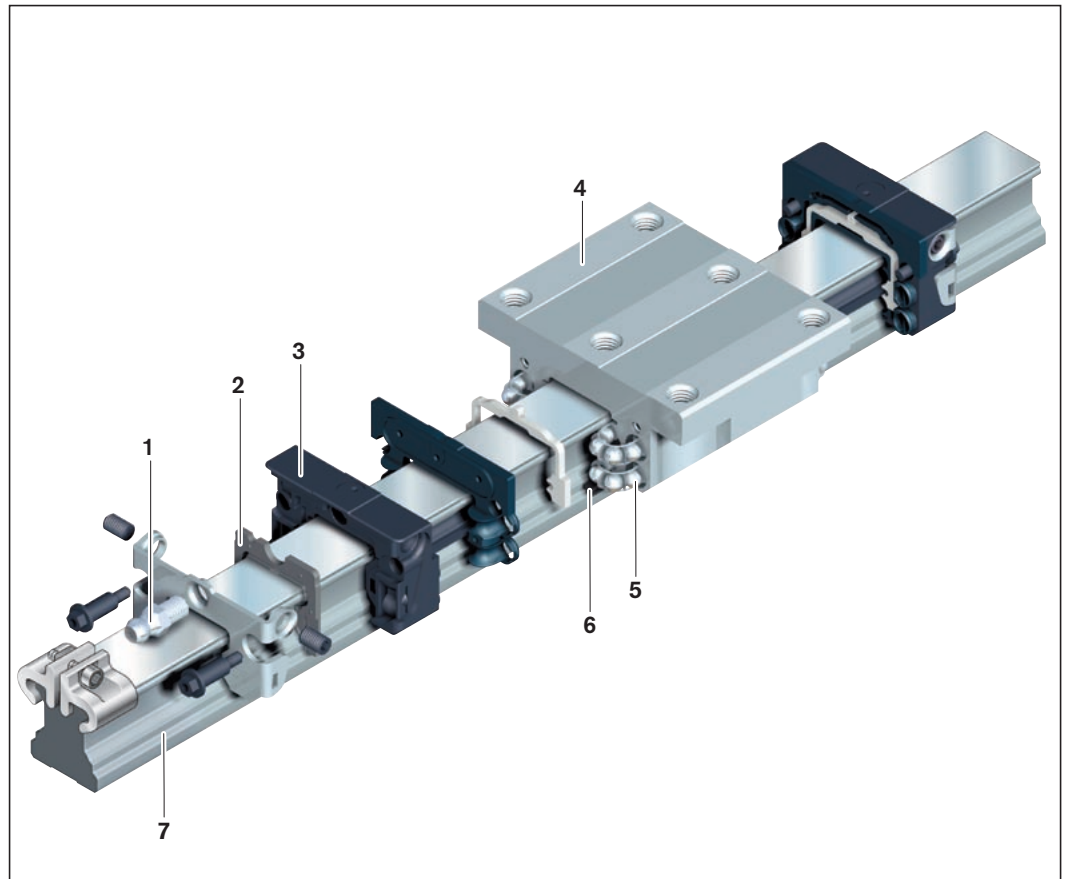
3 Profiled rail systems

3.1 Principles

3.1.1 System technology

A key component of the runner block is the body with its hardened raceways. The rolling elements are normally made from anti-friction bearing steel and are in rolling contact with the runner block and with the rail. The end caps contain recirculation pieces which guide the rolling elements from the load-bearing zone to the return zone, and vice versa. The end caps are also designed to

accommodate sealing elements. A complete seal kit consists of the end wiper seals and the side seals, providing all-around sealing to prevent dirt or dust from working its way into the runner block. Runner blocks are lubricated via lube ports in the end caps to ensure full functionality of the guide. The guide rail has hardened running tracks to match the hardened raceways in the runner block.



Structural design, as illustrated by a latest-generation ball rail system

- 1 Lube port (lube nipple)
- 2 End wiper seal
- 3 End cap
- 4 Runner block body
- 5 Rolling element
- 6 Side seal
- 7 Guide rail

3 Profiled rail systems

3.1 Principles

3.1.1 System technology

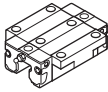
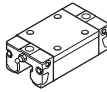
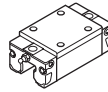
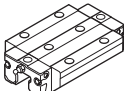
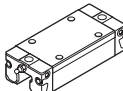
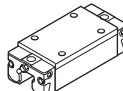
External structure of profiled rail guides

Profiled rail guides are available in a wide variety of designs for use as machine elements. The main design styles and sizes are covered

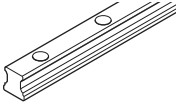
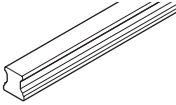
by the DIN 645 standard, which also specifies the main outside dimensions and the connection dimensions.

Design styles

Runner block design styles according to DIN 645-1:

	Series 1	Series 2	Series 3
Design style	Normal 	Slimline 	Slimline High 
	Normal Long 	Slimline Long 	Slimline High Long 

Guide rail design styles according to DIN 645-1:

	Series 1	Series 2
Design style	For mounting from above 	For mounting from below 

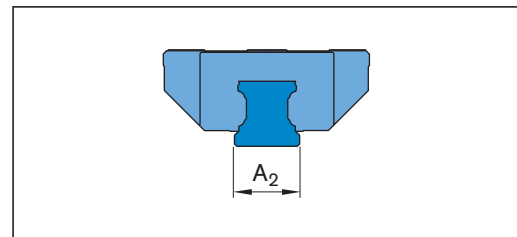
There are, however, many more design styles available than those specified in the standard. Special applications and new machine concepts require specially engineered guides to achieve

maximum performance. Runner block designs today include wide, short and low-profile versions. Guide rails are also available as V-guide rails with a dovetail fit.

Sizes

Size is determined by the width A_2 of the guide rail base, which also determines the dimensions of the runner block.

In wide profiled rail system designations it is the second figure (e.g. 20/40) which denotes the width A_2 of the guide rail base, while the first figure (20/40) refers to the standard sizing system.



Width of guide rail base

Reference standard	DIN 645 Part 2	DIN 645 Part 1
Title of standard	Rolling bearings, profile rail rolling guides – Part 2: Dimensions for series 4	Rolling bearings, profile rail rolling guides – Part 1: Dimensions for series 1 to 3
Profiled rail system	Miniature	Standard
Size	7 9 12 15	15 20 25 30 35 45 55 65

No reference standard							
Profiled rail system	Wide					Heavy duty	
Size	20/40	25/70	35/90	55/85	65/100	100	125

3 Profiled rail systems

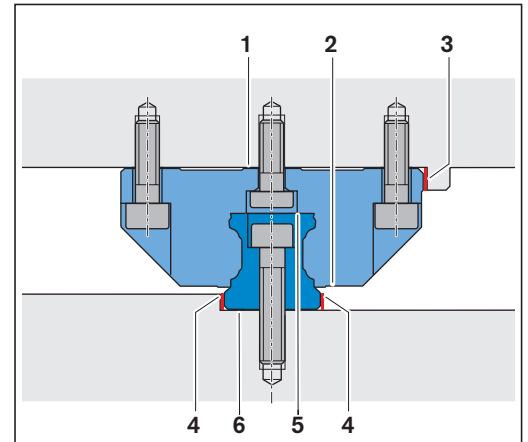
3.1 Principles

3.1.1 System technology

Reference surfaces and edges

Because of their structural design, linear guides have various reference surfaces and edges for alignment with and mounting to adjoining structures. The bases of the runner blocks and guide rails serve as mating surfaces for mounting to the surrounding structure. They have threaded or countersunk holes to receive fixing screws.

The side surfaces serve to transmit forces laterally and to align the components during installation. They are called reference edges. The guide rail has two reference edges that can be used independently of each other. Runner blocks generally have one reference edge which must be taken into consideration during mounting. However, some runner block types can have two or more reference edges.



Reference surfaces and edges

- 1 Base of the runner block
- 2 Head of runner block
- 3 Reference edge on the runner block
- 4 Two reference edges on the rail
- 5 Head of the guide rail
- 6 Base of the guide rail

Coordinate system

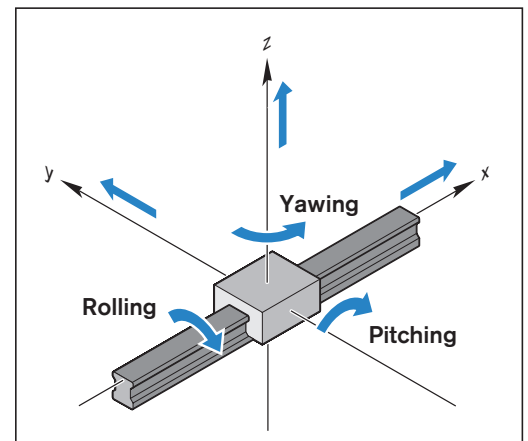
In profiled rail guides, movement or displacement of the runner block is governed by the coordinate system shown in the illustration. This coordinate system has 6 degrees of freedom. The X-axis is the direction of travel. In all other directions, movement is only possible as elastic deflection of the guide unit under load.

Linear degrees of freedom (along the axes):

- Direction of travel (X-axis)
- Lateral movement (Y-axis)
- Lift-off movement (Z-axis)
- Downward movement (Z-axis, negative direction)

Rotational degrees of freedom:

- Rolling (rotation about the X-axis)
- Pitching (rotation about the Y-axis)
- Yawing (rotation about the Z-axis)



Axial and rotational directions of movement

3 Profiled rail systems

3.1 Principles

3.1.1 System technology

Internal structure of profiled rail guides

Manufacturers are free to design the internal structure of profiled rail guides as they wish. The guides produced by the various manufacturers differ in the way rolling contact is achieved.

Specifically, these differences relate to:

- Rolling element shape (ball/roller)
- Rolling element size
- Rolling contact type (2-point/4-point)
- Conformity of ball contact
- Number of rolling element rows (2/4/6)
- Arrangement of rolling element rows (X/O)
- Contact angle

These differences result in different system characteristics in terms of the load capacity, rigidity and friction.

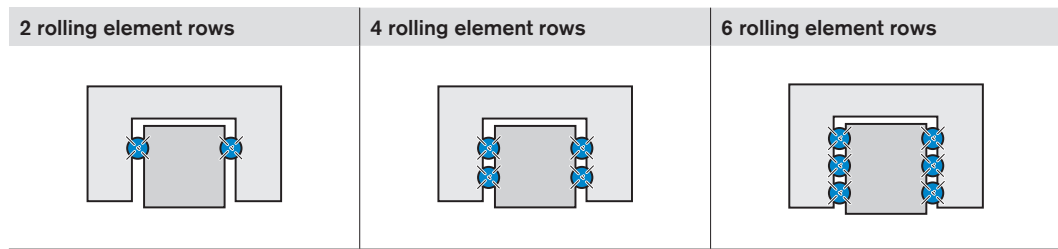
The influences of the rolling element shape and size, rolling contact and conformity were discussed earlier in Chapter 2, section 2.3. This section therefore deals only with the specific characteristics of profiled rail guides.

Number of rolling element rows

The number of load-bearing rolling element rows is a basic distinguishing feature in profiled rail guides. It influences the load capacity, the rigidity behavior and the friction behavior of the profiled rail guide. The more rows a rail guide has, the greater the load capacity and the rigidity will be. However, this statement applies only when all other parameters remain constant, i.e. same rolling element shape and size, same type of rolling contact (2-point or 4-point), same conformity, same arrangement, and same contact angle.

It should also be noted that increasing numbers of rows result in increasingly complex and costly designs.

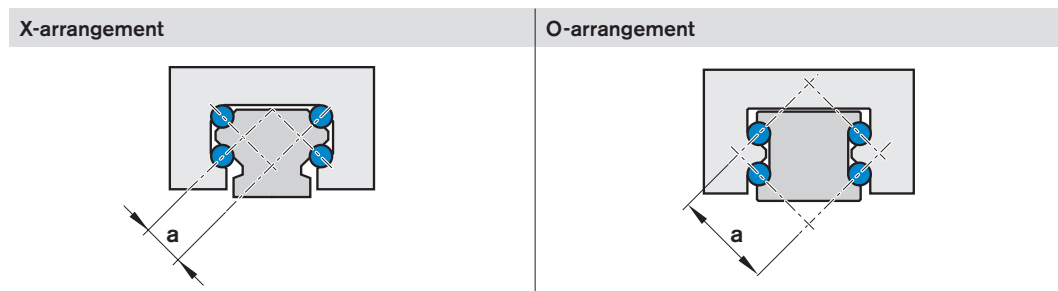
Rexroth uses only 2-row and 4-row designs in its ball rail systems. The roller rail systems have 4 rows. These designs allow a much more even distribution of the load across the rolling element rows than is possible with 6-row profiled rail guides.



Comparison of X- and O-arrangements

Just as in rotary rolling contact bearings, the raceways in profiled rail guides can be arranged in an X- or an O-configuration. The system characteristics of these two arrangements are identical except for their behavior when subjected to a torsional moment. They show no differences in behavior under down loads, lift-off loads and side loads or under longitudinal moments.

Because of its greater leverage (a), the O-arrangement can withstand higher torque forces than the X-arrangement. In same-size systems, the O-arrangement therefore offers higher torsional stiffness. Rexroth's 4-row ball and roller rail systems have an O-arrangement.



3 Profiled rail systems

3.1 Principles

3.1.1 System technology

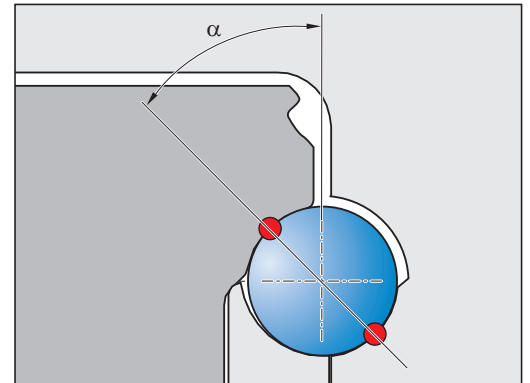
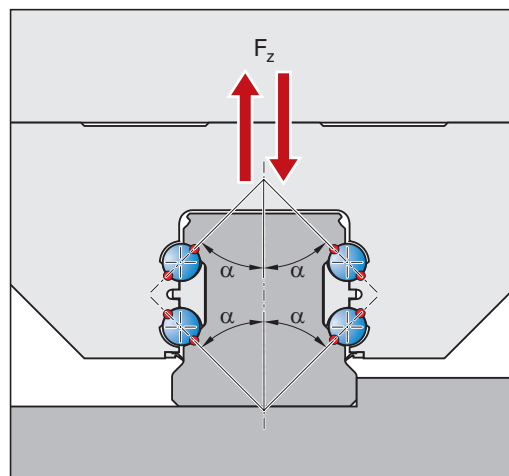
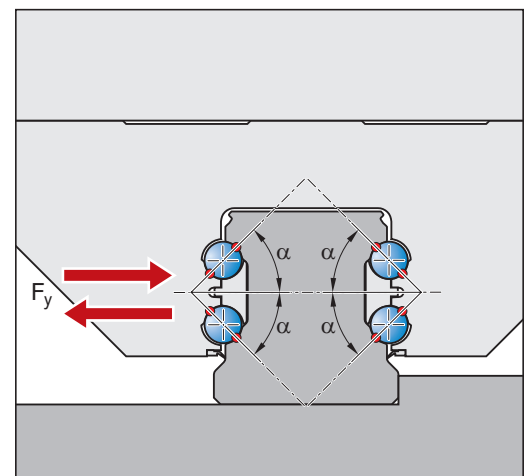
Contact angle

Profiled rail guides should be capable of taking up loads from all directions. The raceways or contact points are therefore arranged at an angle. This corresponds to the nominal contact angle as defined in the ISO 14728 standard for the angle of contact between the rolling-contact partners in profiled rail guides.

ISO 14728 defines the nominal contact angle as follows:

“Angle between the direction of load on the linear bearing and the nominal line of action of the resultant forces transmitted by a bearing raceway member to a rolling element.”

The contact angle is therefore dependent on the direction of loading. It is always indicated for loads in the main directions of loading (Y-axis, Z-axis). In all Rexroth profiled rail systems, the contact angle is 45°.

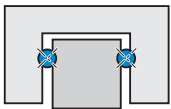
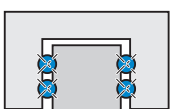
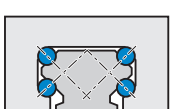
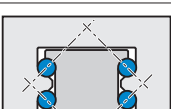
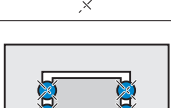
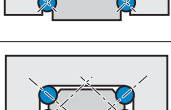
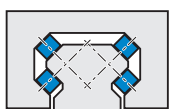
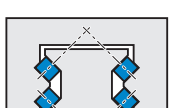
Nominal contact angle α Contact angle α under lift-off or down loadsContact angle α under side loads

3 Profiled rail systems

3.1 Principles

3.1.1 System technology

Number and arrangement of rolling element rows Rolling-bearing profiled rail guides can be differentiated according to the following basic structural criteria.

Profiled rail guide	Number of rolling element rows	Schematic representation	Type of contact	Arrangement of raceways	Rexroth range
Ball rail systems (BRS)	2 rows		4-point contact		■ Miniature ball rail systems ■ eLINE ball rail systems
	4 rows		4-point contact		
	4 rows		2-point contact	X-arrangement	
	4 rows		2-point contact	O-arrangement	■ Ball rail systems
	6 rows		4-point contact		
	6 rows		2-point contact	Combined X-O-arrangement	
Roller rail systems (RRS)	4 rows		Line contact	X-arrangement	
	4 rows		Line contact	O-arrangement	■ Roller rail systems

3 Profiled rail systems

3.1 Principles

3.1.1 System technology

3.1.1.2 Load-bearing capability

Load-bearing capability

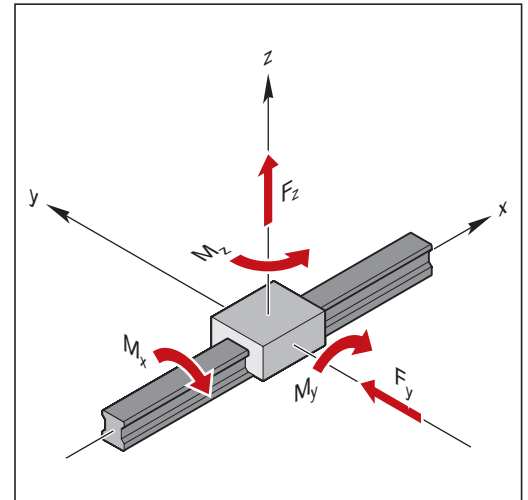
In Rexroth's 2-row and 4-row ball and roller rail systems the raceways are arranged with a 45° angle of contact in relation to the main directions of loading. This results in the same high load capacity of the entire system in all four major planes of load application. The runner blocks can be subjected to forces and to load moments.

Forces in the four major planes of load application:

- Lift-off F_z (positive Z-direction)
- Down load $-F_z$ (negative Z-direction)
- Side load F_y (positive Y-direction)
- Side load $-F_y$ (negative Y-direction)

Moment loads:

- Moment M_x (about the X-axis)
- Moment M_y (about the Y-axis)
- Moment M_z (about the Z-axis)



Major directions of loading

Load capacities

The load-bearing capability of profiled rail systems is described by the static load capacity C_0 and the dynamic load capacity C . These load capacity ratings are key characteristics describing the performance capability of the systems. Rexroth verifies the dynamic load capacities for all of

its products in endurance tests. Its profiled rail systems have the same load capacities in all major planes of load application. The methods for calculating load capacities are defined in the ISO 14728 standard.

Definition of dynamic load capacity C

The radial loading of constant magnitude and direction which a linear rolling bearing can

theoretically endure for a nominal life of 100 km distance traveled (acc. to ISO 14728-1).

Definition of static load capacity C_0

The static load in the direction of loading which results in a permanent overall deformation of approximately 0.0001 times the rolling element diameter at the center of the most heavily loaded rolling element/raceway contact (acc. to ISO 14728-2).

According to ISO 14728-2, this corresponds to a calculated contact stress at the contact point of:

- 4200 to 4600 MPa for ball rail guides
- 4000 MPa for roller rail guides

Dynamic load moments M_t , M_L and static load moments M_{t0} , M_{L0}

The dynamic load moments M_t and M_L and the static load moments M_{t0} and M_{L0} are calculated from the load capacities, the geometry, the number of rolling element rows, the number of load-carrying rolling elements, and the contact angle.

They are crucial factors when the runner blocks are subjected to torsional and longitudinal moment loads.

3 Profiled rail systems

3.1 Principles

3.1.1 System technology

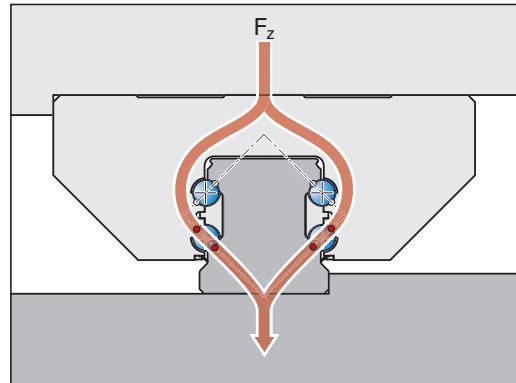
Direction of loading

Runner blocks are normally subjected to loading in four major planes of load application. They may, however, also be subjected to loads acting at any angle between these planes. It should be remembered that the load-bearing capability of the elements will be reduced in such cases. The reasons for this become clear when one considers the flow of forces inside the runner block, as described below.

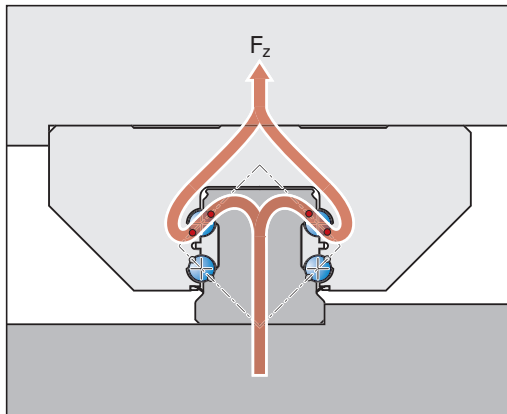
Force flow in the runner block

Force flow inside the runner block for the main directions of loading

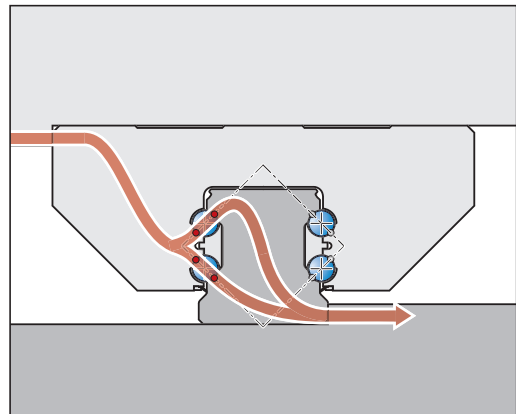
Under down loads, lift-off loads and side loads, the force is transmitted via two rows of rolling elements or via two raceways.



Force flow under a down load F_z



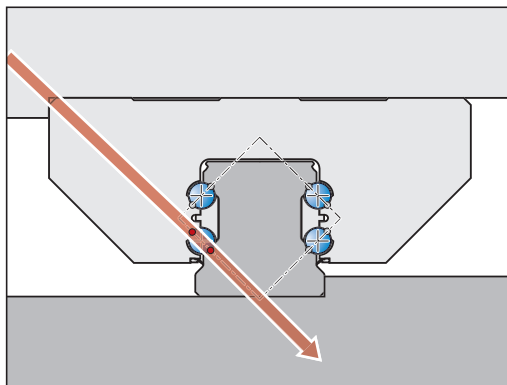
Force flow under a lift-off load F_z



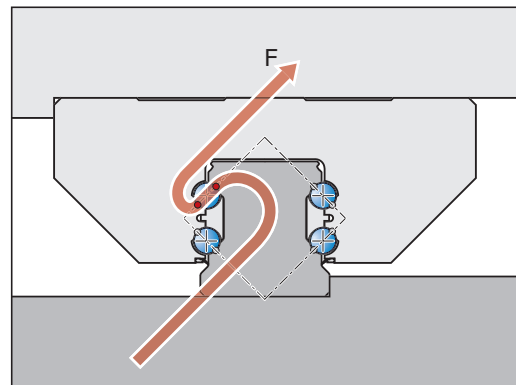
Force flow under a side load F_y

Force flow inside the runner block for a load acting at a 45° angle

The most unfavorable direction of loading in profiled rail guides with a raceway contact angle of 45° is a load acting at an angle of 45°. In this case, the load is carried by only one row of rolling elements or one raceway.



Down load at a 45° angle



Lift-off load at a 45° angle

3 Profiled rail systems

3.1 Principles

3.1.1 System technology

Combined equivalent load on bearing

Since the reference edges and the mounting surfaces can only transmit vertical and horizontal forces, the most unfavorable case occurs when the loads acting in the vertical and horizontal direction are of equal magnitude.

Expressed in mathematical terms, the resultant total load \vec{F}_{res} is obtained by addition of the vertical force vector \vec{F}_z and the horizontal force vector \vec{F}_y :

$$\vec{F}_{\text{res}} = \vec{F}_y + \vec{F}_z$$

The load-bearing capacity of the profiled rail guide depends on the direction of loading. For the nominal life calculation, the factor used to describe the load is therefore not the resultant load \vec{F}_{res} but instead the combined equivalent load on the bearing F_{comb} . This is obtained by adding the absolute values of the vertical force $|F_z|$ and the horizontal force $|F_y|$ acting on the runner block.

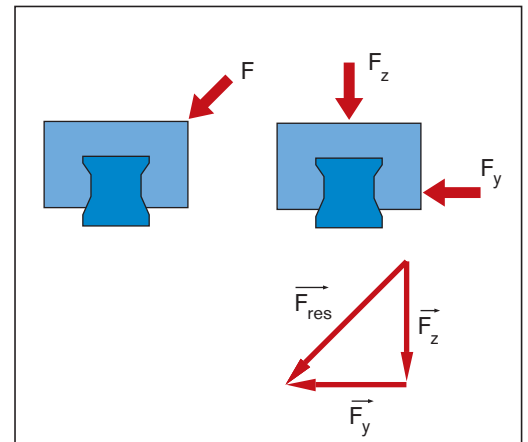
$$(3-1) \quad F_{\text{comb}} = |F_y| + |F_z|$$

If a single load or the load resulting from several forces acts in any direction other than the main directions of loading, then the calculated combined load on the bearing F_{comb} will be greater than the resultant total load $|\vec{F}_{\text{res}}|$ obtained by addition of the force vectors.

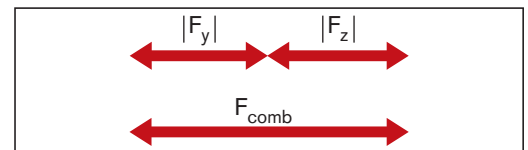
$$F_{\text{comb}} > |\vec{F}_{\text{res}}|$$

Calculating the combined equivalent load on the bearing thus makes allowance for the fact that the load-bearing capability of a profiled rail guide will be reduced when a load is applied at an angle rather than in one of the main directions of loading. For the same load capacity rating, the life expectancy will therefore be shortened due to the higher load on the bearing.

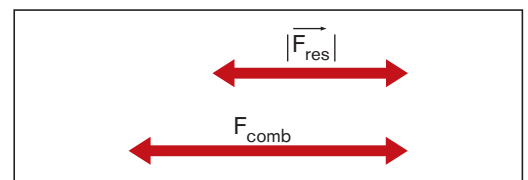
The structural design of Rexroth's profiled rail systems permits a simplified calculation of the combined equivalent load on the bearing F_{comb} using formula (3-1).



Most unfavorable case: Load acting at an angle of 45° or horizontal and vertical loads of equal magnitude



Determination of F_{comb}



Comparison of $|\vec{F}_{\text{res}}|$ and F_{comb} for the most unfavorable case $|F_y| = |F_z|$

3 Profiled rail systems

3.1 Principles

3.1.1 System technology

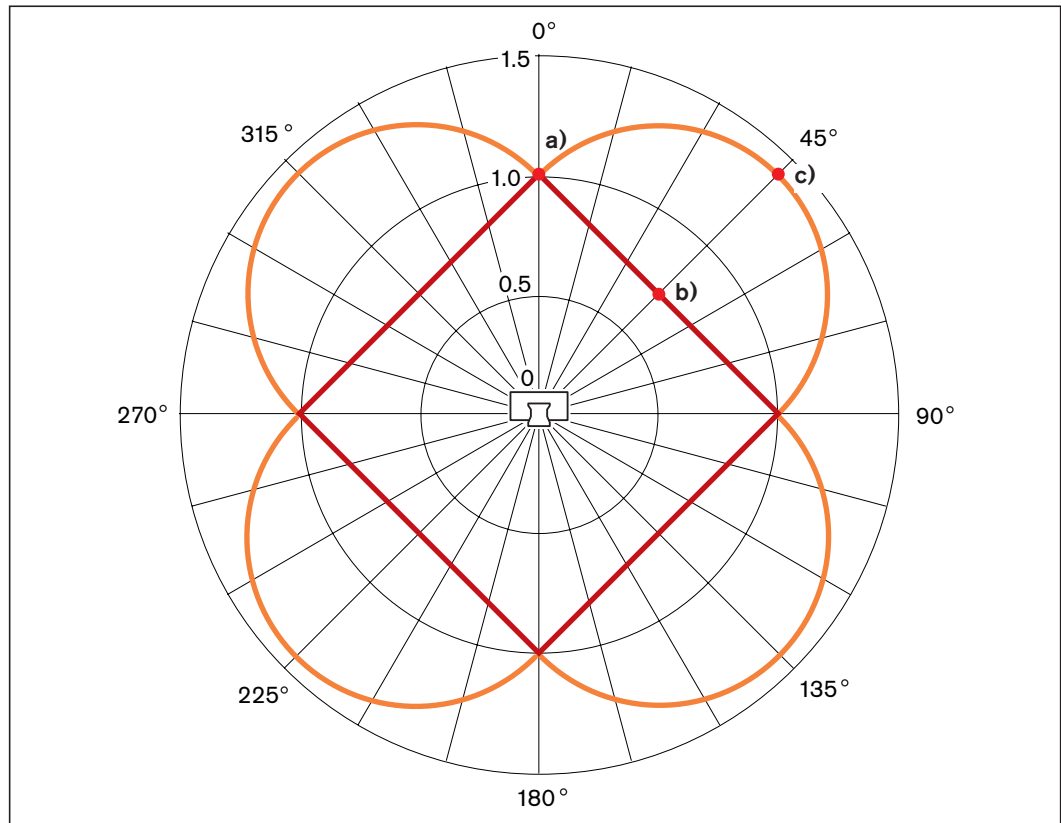
Dependency of nominal life on the direction of loading

The following diagram maps the direction of loading relative to the load or the load capacity, showing the conditions under which the same nominal life will be achieved. The values derive from the

formulas mentioned above for calculating the load on the bearing and the relationship between the load capacity and the load.

$$(3-2) \quad L \sim \left(\frac{C}{F}\right)$$

L = nominal life (km)
 C = load capacity (N)
 F = load (N)



Loads and load capacities for the same nominal life under different directions of loading

— Load
 — Load capacity

Notes explaining the diagram

- a) In all four major directions of loading the value is 1, i.e. the full nominal life will be achieved at loads and load capacities of 100%.
- b) To achieve the same nominal life as in one of the four main directions of loading, a load acting at a 45° angle must not exceed 0.707 times the load acting in one of the four main directions of loading.
- c) Alternatively, to achieve the same nominal life as in one of the four main directions of loading, the load capacity for a load acting at a 45° angle would have to be 1.414 times greater than the load capacity in the main directions of loading. In practice, this means installing a larger profiled rail guide or a greater number of runner blocks in order to increase the load-bearing capability of the system.

3 Profiled rail systems

3.1 Principles

3.1.1 System technology

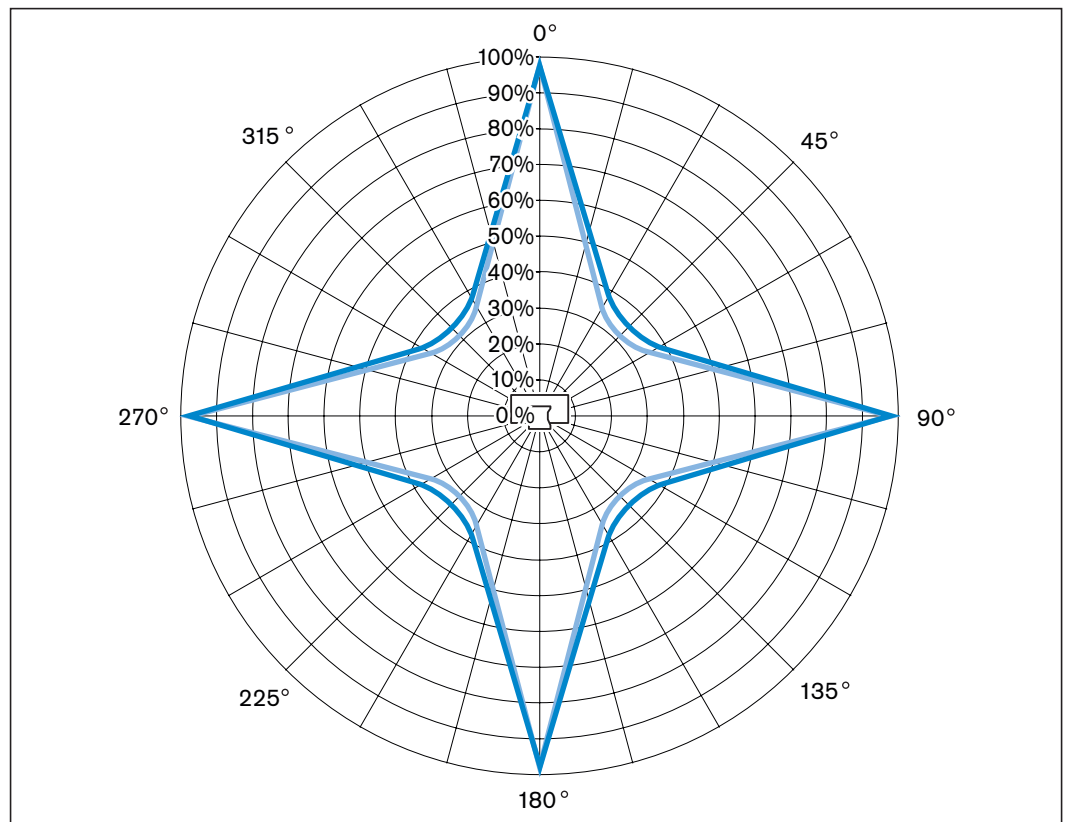
To increase the service life of the guides, they should therefore be installed in an orientation appropriate to the actual direction of loading. If this is not done, the service life may be drastically shortened.

Since the load capacity to load ratio is accounted for in the nominal life calculation by the exponent p , the travel life expectancy will therefore be significantly reduced at a load application angle of 45° .

$$(3-3) \quad L = \left(\frac{C}{F}\right)^p \cdot 10^5 \text{ m}$$

$$p = 3 \quad \text{for ball rail guides}$$

$$p = 10/3 \quad \text{for roller rail guides}$$



Effect of the load direction on the life expectancy (in %)

- Ball rail system
- Roller rail system

Example:

For a load acting at an angle of 45° , the life expectancy of a ball rail system is only 35% of that for a load acting in any of the main loading directions. The life expectancy of a roller rail system may even be as little as 32%.

3 Profiled rail systems

3.1 Principles

3.1.1 System technology

3.1.1.3 Preload

Definition of preload

Runner blocks can be preloaded in order to increase the overall rigidity of the profiled rail guide. The preload anticipates the effects of elastic deflection. It is achieved through widening of the runner block body by using rolling elements (diameter D_W) with a defined oversize d_{OS} .

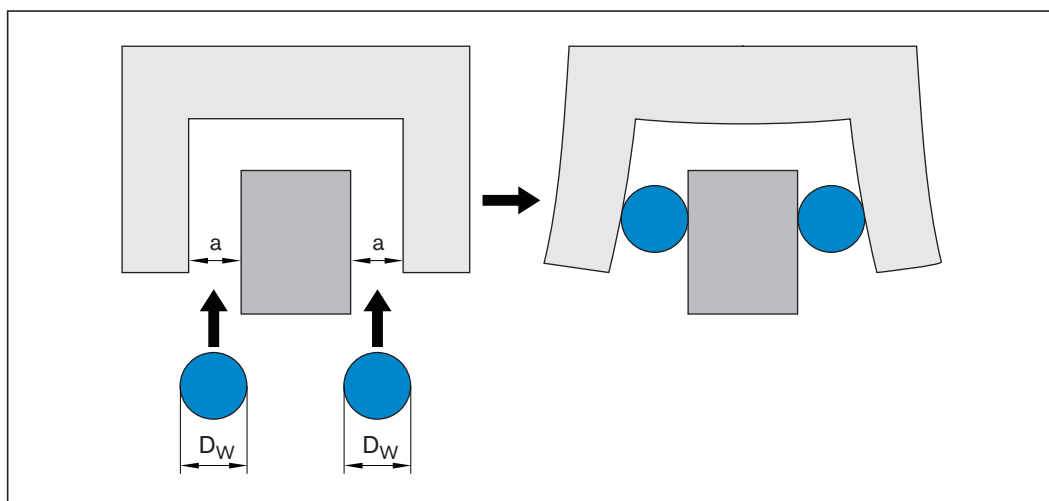
The chosen oversize determines the degree of preload. Preloading causes the flanks of the runner block body to curve outward at the tips. Depending on the linear guide type, versions are available with different degrees of preload or without preload (i.e., with clearance).

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

D_W = rolling element diameter (mm)

a = distance between raceways (mm)

d_{OS} = oversize (mm)



Preloading by inserting oversized rolling elements

3 Profiled rail systems

3.1 Principles

3.1.1 System technology

Preload classes

The degree of preload is classified in relation to the dynamic load capacity C . Depending on their design, runner blocks may be available in up to four preload classes (C0, C1, C2, C3).

The higher the preload, the more rigid the runner block will be.

Selection of the preload class

Preload class Code	Version	Preload class factor X_{pr} (-)	Application area
C0	Without preload (with clearance)	0	For particularly smooth-running guide systems with the lowest possible friction and a minimum of external influences.
C1	Slight preload 0.02 C (2% of C) For roller rail systems (RRS): 0.03 C (3% of C)	0.02 0.03 (RRS)	For zero-clearance guide systems with low external loads and low requirements on overall rigidity.
C2	Medium preload 0.08 C (8% of C)	0.08	For precise guide systems with both high external loading and high demands on overall rigidity; also recommended for single-rail systems. Above-average moment loads can be absorbed without significant elastic deflection.
C3	High preload 0.13 C (13% of C)	0.13	For highly rigid guide systems such as required in precision machine tools or forming/molding machines. Above-average loads and moments can be absorbed with the least possible elastic deflection.

The preload force for a particular preload class can be calculated using the respective preload class factor X_{pr} . This internal loading of the runner block must be taken into account when calculating the life expectancy.

$$(3-4) \quad F_{pr} = X_{pr} \cdot C$$

F_{pr} = preload force of the runner block (N)

X_{pr} = preload class factor (-)

C = dynamic load capacity of the runner block (N)

Example for a size 25 runner block with a load capacity C of 22,800 N and preload class C2:

$$F_{pr} = X_{pr} \cdot C = 0.08 \cdot 22800 \text{ N} = 1824 \text{ N}$$

3 Profiled rail systems

3.1 Principles

3.1.1 System technology

3.1.1.4 Rigidity

Definition of rigidity

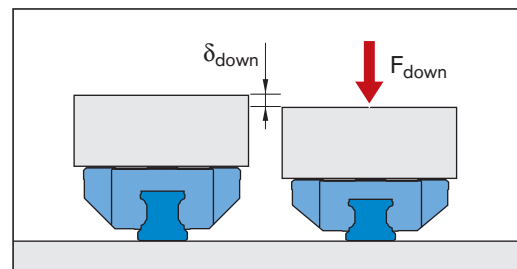
The rigidity of a runner block is defined by the relationship between the external load and the resulting elastic deflection in the direction of loading.

Rigidity is an important criterion for selection of rail guides. Profiled rail systems have different rigidity levels (see selection criteria, section 3.1.2)

according to their type and design. The rigidity of a particular runner block depends on the preload class selected. The higher the preload of the system, the greater the rigidity will be. The rigidity levels of the runner blocks are illustrated as curves in charts (see example on following page).

Down load

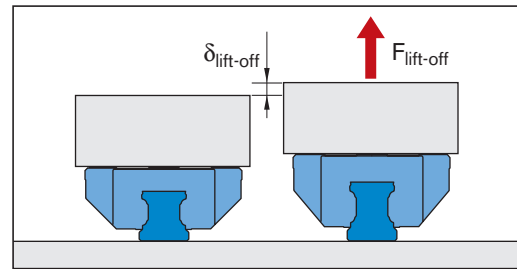
$$(3-5) \quad c_{\text{down}} = \frac{F_{\text{down}}}{\delta_{\text{down}}}$$



Downward load

Lift-off load

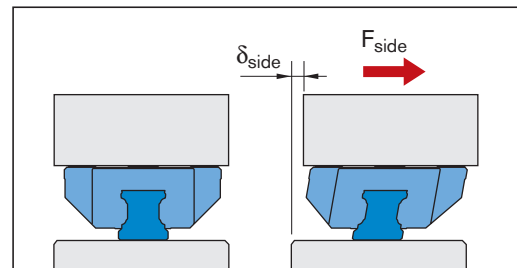
$$(3-6) \quad c_{\text{lift-off}} = \frac{F_{\text{lift-off}}}{\delta_{\text{lift-off}}}$$



Upward load

Side load

$$(3-7) \quad c_{\text{side}} = \frac{F_{\text{side}}}{\delta_{\text{side}}}$$



Sideways load

- F = load resulting from a force (N)
- δ = elastic deflection in the direction of loading (μm)
- c = rigidity in the direction of loading (N/μm)

3 Profiled rail systems

3.1 Principles

3.1.1 System technology

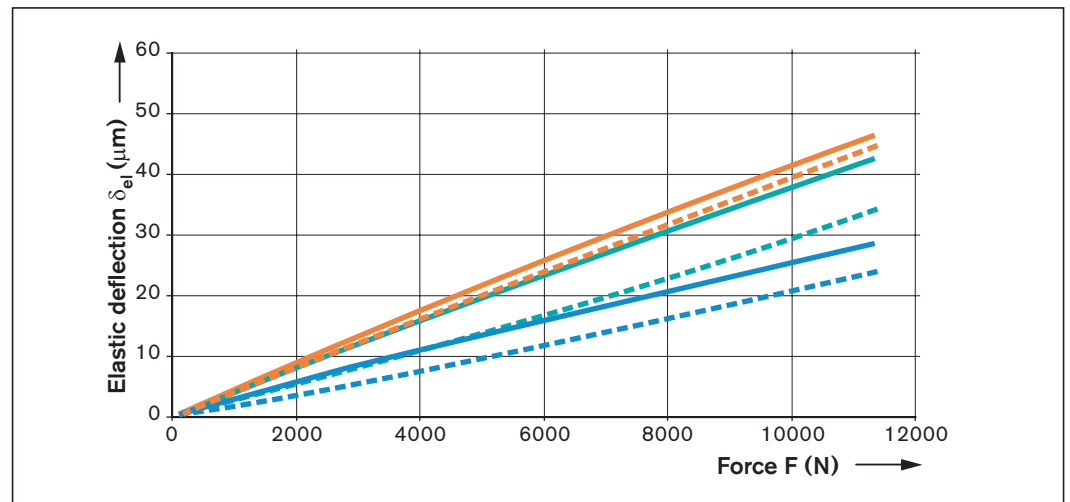
Rigidity charts

Rexroth provides rigidity charts for the various runner block versions and preload classes. When using these charts, the direction of loading must be taken into account. A distinction is made between loads acting in a downward direction (down loads), in an upward direction (lift-off

loads), and from the side. In addition to the deflections resulting from loading in these three main axial directions, runner blocks are also subject to angular deflections resulting from rotational moment loads. Charts for these angular deflections can also be obtained from Rexroth on request.

Deflection under loading in the three main axial directions

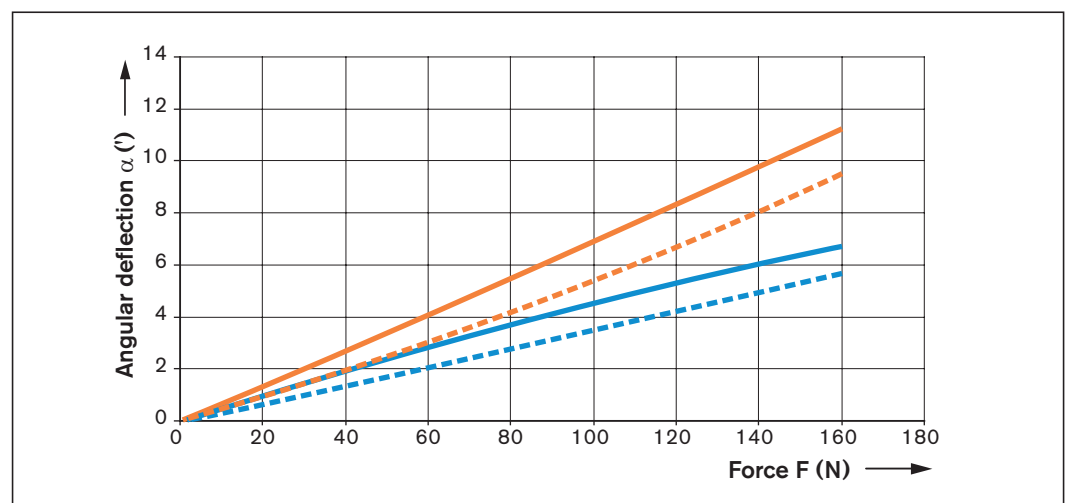
Rigidity chart for the three main directions of loading at preload C1 (0.02 C) and C2 (0.08 C). Example: ball runner block, flanged version, size 25



- C1, down load
- - - C2, down load
- - - C1, lift-off load
- - - C2, lift-off load
- C1, side load
- - - C2, side load

Angular deflection under rotational moment loads

Rigidity chart for angular deflection under rotational moment loads in the rolling and pitching directions at preload C1 (0.02 C) and C2 (0.08 C). Example: ball runner block, flanged version, size 25



- C1, rolling
- - - C2, rolling
- C1, pitching
- - - C2, pitching

3 Profiled rail systems

3.1 Principles

3.1.1 System technology

3.1.1.5 Accuracy

Accuracy classes

Runner blocks and guide rails are specified according to a series of accuracy classes (details of which are given on the following pages). Each accuracy class has associated tolerances defining the maximum permissible deviation.

Height accuracy

The height accuracy specifies the permissible deviation of the guide unit in the Z-axis. The dimension H between the base of the guide rail and the base of the runner block may per-

sibly vary within the tolerances defined for that accuracy class.

Width accuracy

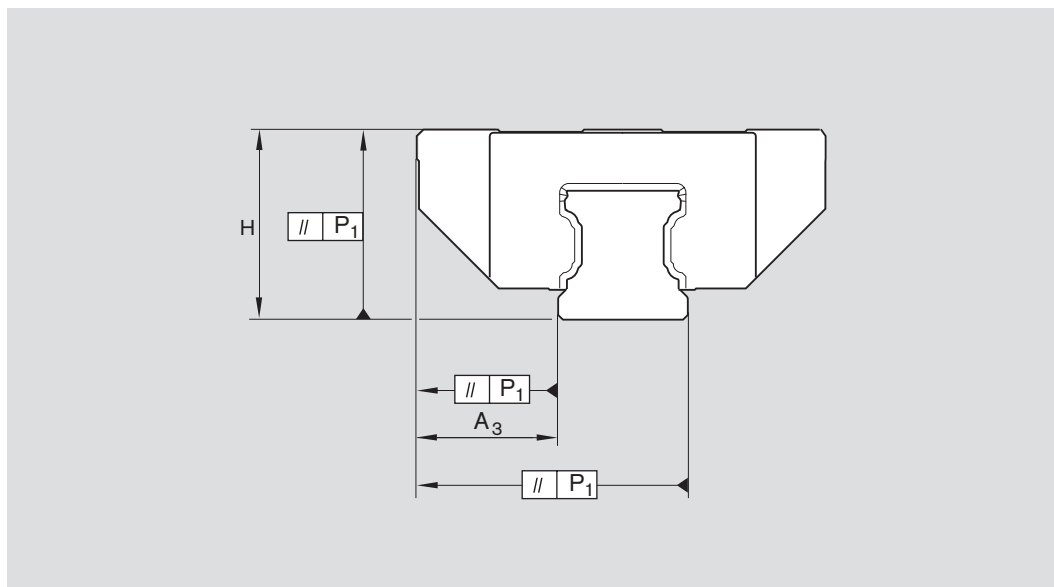
The width accuracy specifies the permissible deviation of the guide unit in the Y-axis. In this case, a tolerance range is defined for the dimension A_3

between the reference plane of the guide rail and the reference plane of the runner block.

Parallelism

Parallelism specifies the permissible deviation from the parallel for pairs of planes in the guide unit. The value P_1 is the tolerance for parallelism

between the reference planes of the runner block and guide rail.



Dimensions H and A_3 and parallelism P_1

H Dimension for height accuracy

A_3 Dimension for width accuracy

P_1 Tolerance for parallelism

3 Profiled rail systems

3.1 Principles

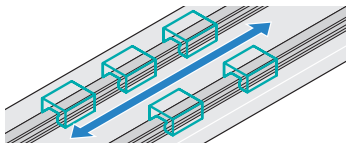
3.1.1 System technology

Tolerances for accuracy classes

The height and width accuracies are specified by tolerances for the dimensions H and A_3 as defined for the various accuracy classes. The accuracy tolerances relate to the manufacturing tolerances of the runner block and the guide rail.

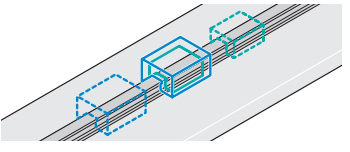
The table below shows the height and width tolerances for profiled rail systems.

Accuracy classes	Tolerances ¹⁾ for dimensions H and A_3 (μm)		Max. difference in dimension H and A_3 on one guide rail (μm) $\Delta H, \Delta A_3$
	H	A_3	
E	± 120	± 70	60
N	± 100	± 40	30
H	± 40	± 20	15
P	± 20	± 10	7
XP ²⁾	± 11	± 8	7
SP	± 10	± 7	5
UP	± 5	± 5	3



Measured at middle of runner block

For any runner block/rail combination at any position on rail



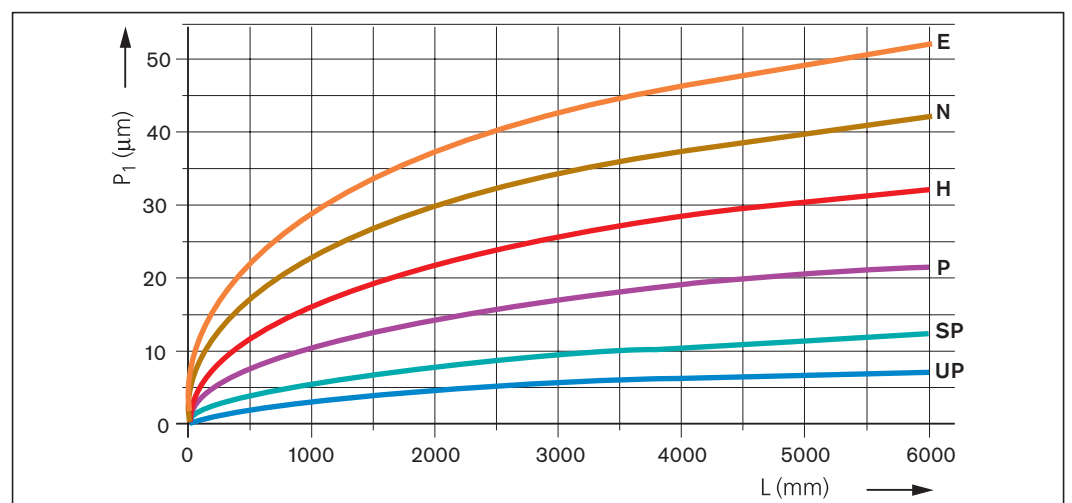
For different runner blocks at same position on rail

1) Tolerances for combinations of guide rails and runner blocks with different accuracy classes on request
 2) Combination of XP runner block and SP guide rail in ball rail systems

Parallelism offset of the rail guide in service

The parallelism offset relates to the manufacturing tolerances of the guide rails. The graph below shows the maximum parallelism offset P_1 when the rail guide is in service as a function of the

guide rail length. These curves assume that the respective rail guides have been mounted under ideal conditions.



Maximum permissible parallelism offset P_1 of the rail guide in service (measured at middle of runner block) as a function of the guide rail length L

3 Profiled rail systems

3.1 Principles





3.1.1 System technology

Accuracy classes

The accuracy classes define the geometric tolerances (i.e. maximum permissible deviations) for rail guides in the directions described above. The original accuracy classes were first specified in DIN 645. Advances in manufacturing techniques

have made it possible to add higher accuracy classes to the ones defined in the standard. The accuracy class for a profiled rail system is selected on the basis of the intended application.

Accuracy classes of Rexroth profiled rail systems:

Profiled rail guide	Accuracy class and description							
	E eLINE	N Normal	H High accuracy	P Precision	XP eXtra precision	SP Super precision	UP Ultra precision	
Ball rail system 	-	✓	✓	✓	✓	✓	✓	
Miniature ball rail system 	-	✓	✓	✓	-	-	-	
eLINE ball rail system 	✓	✓	-	-	-	-	-	
Roller rail system 	-	-	✓	✓	-	✓	✓	

Accuracy class XP applies only to high-precision ball runner blocks with very good travel characteristics. Guide rails are not available in XP. Accuracy classes XP, SP and UP are suitable for high-precision metal-cutting operations, measuring technology, high-precision scanners, electrical discharge machining, etc.

In roller rail systems, guide rails can also be supplied in accuracy class GP (corresponds to SP, but with additional sorting according to height tolerance). Cam roller guides have fixed tolerances.

Precision manufacturing

The guide components are manufactured with such high precision that runner blocks and guide rails can be interchanged without problems. For example, a runner block can be paired with any

guide rail of the same size. Similarly, different runner blocks can also be used on one and the same guide rail. Runner blocks can therefore be ordered separately and combined as required.

3 Profiled rail systems

3.1 Principles

3.1.1 System technology

Selection of the accuracy class

- For a system with several runner blocks spaced at short distances, it is advisable to select a higher accuracy class for the runner blocks than for the guide rail. The runner block tolerances are the deciding factor here, because a configuration with multiple runner blocks may result in preloading of the system.
- If the runner blocks are spaced widely apart, the guide rail should have a higher accuracy class than the runner blocks. In this case, the guide rail tolerances are more important because of possible distortive stresses, especially in systems with several rails installed parallel to one another.

Table showing recommended runner block/guide rail combinations:

Recommended combination		Guide rail accuracy classes					
		E	N	H	P	SP	UP
Runner block accuracy classes	E	✓	✓	–	–	–	–
	N	✓	✓	✓	–	–	–
	H	–	✓	✓	✓	–	–
	P	–	–	✓	✓	✓	–
	XP	–	–	✓	✓	✓	–
	SP	–	–	–	✓	✓	✓
	UP	–	–	–	–	✓	✓

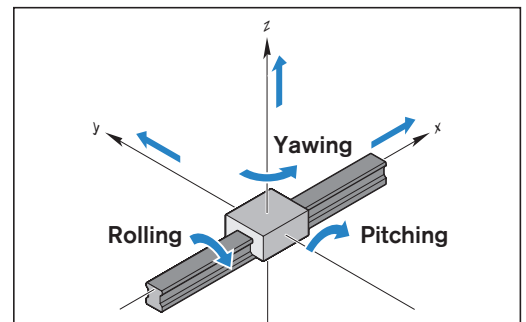
The table below allows preselection according to application areas:

Accuracy classes		Application area				Accuracy
		Handling	Forming operations	Cutting operations	Measuring, testing	
E	eLINE	✓	–	–	–	Low demands on accuracy
N	Normal	✓	–	–	–	
H	High accuracy	✓	✓	✓	✓	
P	Precision	–	✓	✓	✓	
XP	eXtra precision	–	✓	✓	✓	
SP	Super precision	–	–	✓	✓	
UP	Ultra precision	–	–	✓	✓	Extremely high demands on accuracy

3.1.1.6 Travel accuracy

Five degrees of freedom

The travel accuracy of profiled rail systems is described by five of the six possible degrees of freedom. These are the linear degrees of freedom in the Y-axis (width variations) and in the Z-axis (height variations), as well as the rotational degrees of freedom about the X-axis (rolling), Y-axis (pitching) and the Z-axis (yawing). Since the X-axis is the direction in which the guide travels, no accuracy specifications can be made here.



Axial and rotational directions of movement

3 Profiled rail systems

3.1 Principles

3.1.1 System technology

Travel accuracy

Geometric travel accuracy is the term used to describe the actual travel performance of the runner block when in service.

The movement of the rolling elements as they recirculate influences the geometric travel accuracy of the profiled rail system as a whole. A particular phenomenon in this respect is rolling element pulsation. This arises as a result of changes in load distribution as the rolling elements enter the load-bearing zone and the related variation in the

number of rolling elements actually bearing the load. Geometric travel variations due to rolling element pulsation are characterized by the period length of the variations, which is equivalent to two times the rolling element diameter. The geometry of the entry and exit zones guiding the rolling elements into and out of load-bearing zone has a major effect on rolling element pulsation.

Vertical and horizontal waviness

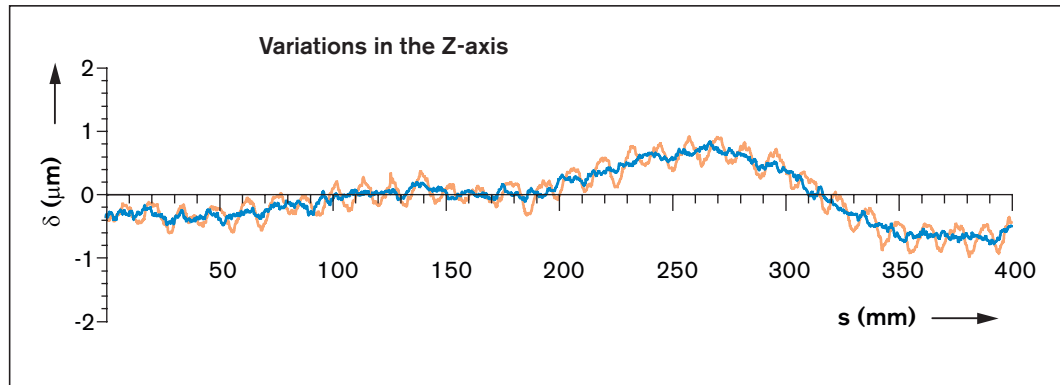
The guide rail also has an influence on geometric travel accuracy. In addition to effects due to the height and width variations described in connection with tolerance classes, screw-fastening of the guide rail can cause local deformations around the mounting holes spaced along the rail. This results in vertical waviness. Horizontal waviness may also occur due to straightness errors in the guide rail, improper mounting, and geometric deficiencies in the adjoining structure.

All of these factors combined – accuracy of the surrounding structures, installation, and the rail and rolling elements themselves – result in geometric travel variations causing the runner block to execute micromovements as it travels along the rail. These micromovements occur both in the rotational degrees of freedom (rolling, pitching, yawing) and in the linear degrees of freedom (height and width variations).

High precision versions

Ball runner blocks are available in high-precision versions with optimized geometry at the transitions between the load-bearing and the return

zone, resulting in especially good travel performance. This optimized geometry is a standard feature in the XP, SP and UP versions.



Comparison of height variations between a conventional ball runner block (—) and a high-precision ball runner block (—). Example: size 35, ball diameter 6.35 mm

δ = height variation (μm)
 s = travel (mm)

3 Profiled rail systems

3.1 Principles

3.1.1 System technology

Entry zone geometry of conventional ball runner blocks

- The balls are guided to the beginning of the entry zone by the ball recirculation track.
- When the distance between the runner block and the rail becomes smaller than the ball diameter, the ball is subjected to loading (preload).
- The preload increases in the entry zone and reaches a maximum in the load-bearing zone. The ball transmits the force from the runner block to the rail.
- As they transition from the entry zone into the load-bearing zone the balls are subjected to pulse-like changes in loading due to the changes in geometry.

Entry zone geometry of high-precision ball runner blocks

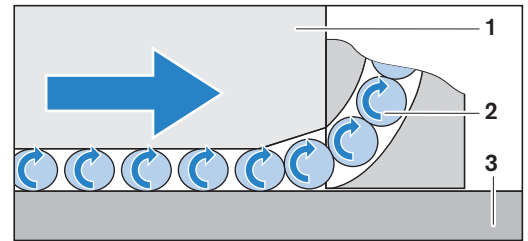
- The balls are guided to the beginning of the entry zone by the ball recirculation track.
- The ball (5) is not loaded as it enters the transition zone because the ball (6) causes the tip of the steel segment (4) to deflect. This deflection is the sum of the compliance of the ball itself and the compliance of the unsupported end of the steel segment.
- As the distance between the steel segment and the rail becomes smaller than the ball diameter, the ball is gradually subjected to loading.
- The preload is smoothly increased until the ball (7) has reached its maximum preload.

A ball deflects the steel segment only as far as necessary to allow the following ball to enter load-free. The ball is no longer guided into the load-bearing zone by a rigid entry channel but by a very smooth flexing curve, which ideally transitions tangentially into the load-bearing zone.

Optimizing the travel accuracy

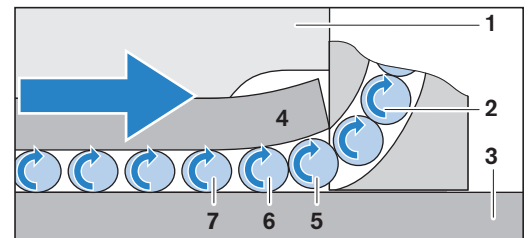
The following measures have a positive effect on the geometric travel performance of profiled rail systems:

- Use of high-precision runner blocks
- Use of runner blocks and guide rails with high accuracy classes
- Use of long runner blocks
- Reducing the tightening torque for the rail mounting screws.
Caution: This may result in a decrease in the transmittable forces and moments.
- Reducing the spacing between guide rail mounting holes in ball rail systems for applications requiring high travel accuracy and low variations in frictional drag



Entry zone in conventional runner blocks

- 1 Runner block
- 2 Rolling element
- 3 Guide rail



Entry zone in high-precision runner blocks

- 1 Runner block
- 2 Rolling element
- 3 Guide rail
- 4 Steel segment

The extremely smooth ball entry behavior and the continuous adjustment of the entry zone in response to the actual load are the great advantages of these high precision ball runner blocks.

- Installing systems with two rails and at least two runner blocks per rail
- Use of wide runner blocks in systems with only one guide rail

Very high travel accuracy can be achieved only if the adjoining structure has been manufactured to close shape and location tolerances and with accurately machined surfaces. Data on the required tolerances can be obtained from Rexroth on request.

3 Profiled rail systems

3.1 Principles

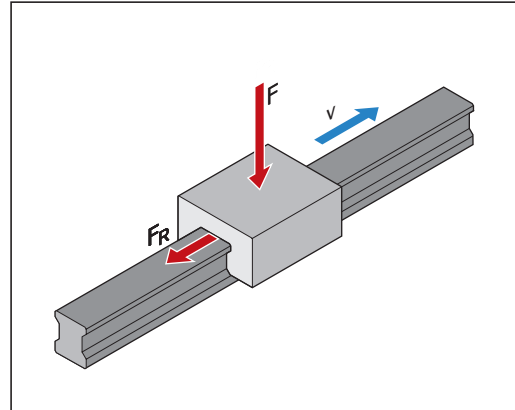
3.1.1 System technology

3.1.1.7 Friction

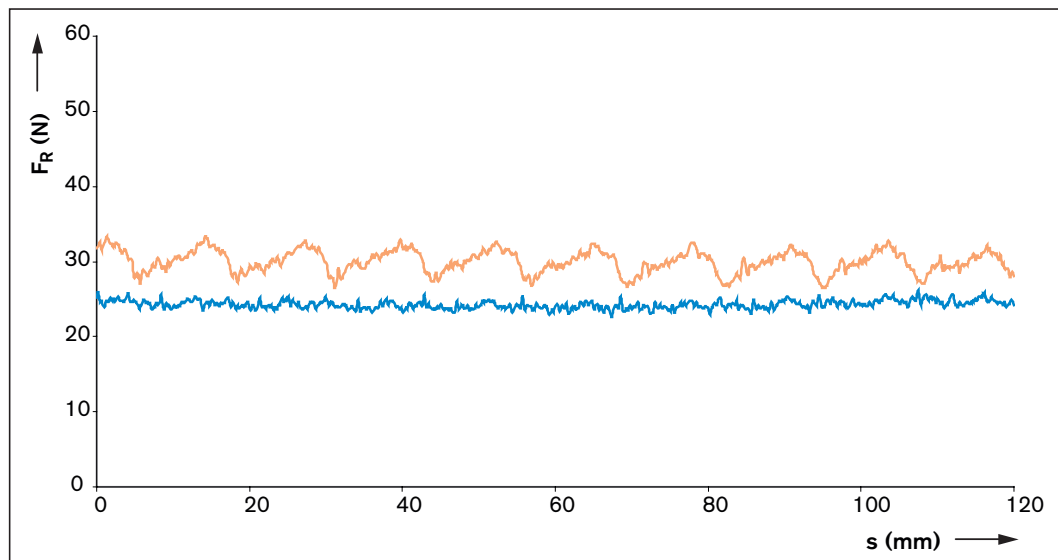
Friction in profiled rail guides

When dimensioning the drive, it is essential to know the level of friction involved. Friction measurements are therefore carried out on all profiled rail systems. The friction values are given in tables in the respective product catalogs. Friction data for special applications can be obtained from Rexroth on request.

The frictional drag of a runner block may vary as it travels along the rail. This is due to the varying number of rolling elements present in the load-bearing zone at any one time. Alternate loading and unloading of the rolling elements as they enter into and exit from the load-bearing zone also causes variations.



Frictional force F_R



Friction profile of a conventional ball runner block (—) and of a high-precision ball runner block (—) under the same load. Example: size 35, ball diameter 6.35 mm

F_R = friction force (N)
 s = travel (mm)

The level of friction in a specific profiled rail system depends on the following factors:

- Load
- Preload
- Sealing
- Travel speed
- Lubricant
- Runner block temperature

Friction factors

The total friction of a runner block is determined by several factors:

- Rolling friction
- Sliding friction
- Lubricant friction
- Friction of the seals

3 Profiled rail systems

3.1 Principles


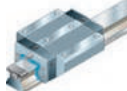
3.1.1 System technology

Rolling friction

Rolling friction is caused by the rolling motion of loaded rolling elements along the raceways. It is influenced by the shape of the rolling element (ball/roller) and by the rolling contact

geometry (2-point/4-point contact, conformity, profiling) – see Chapter 2. The friction coefficient differs according to the type of rolling contact involved.

Friction coefficients of Rexroth ball rail systems and roller rail systems for rolling friction under loads acting in the four main loading directions:

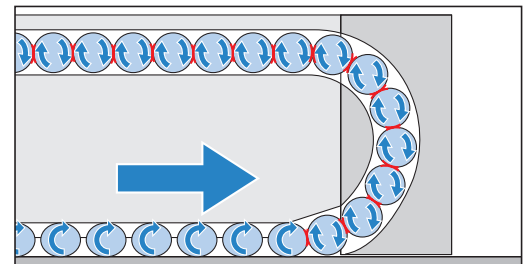
Profiled rail guide	Description of rolling contact	Friction coefficient μ
Ball rail system 	Rolling friction of balls with 2-point contact	0.002
Roller rail system 	Rolling friction of cylindrical rollers	0.0004

The rolling friction of the guide component increases as the load increases. The load may be due either to an external or an internal force. External loads can be machining forces, weight

forces and acceleration forces. Internal loads are caused by the preload or by improper installation.

Sliding friction

Sliding friction occurs between the rolling elements and the plastic components in the recirculation zone and in the return zone (see illustration). The plastic components serve as lateral guides and as recirculation pieces in the end caps. In order to keep the sliding friction between the rolling elements and the plastic parts as low as possible, Rexroth uses only plastics with very good sliding properties.



Sliding friction between rolling elements in the recirculation and return zones

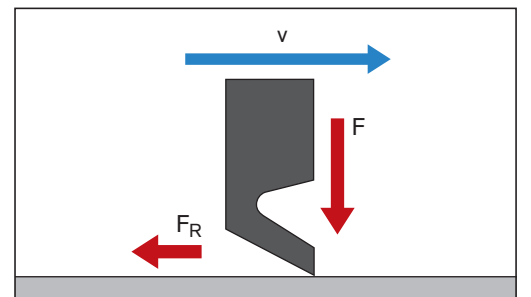
Lubricant friction

Lubricant friction is caused by displacement of the lubricant inside the runner block. The friction level in this case is determined by the properties of the lubricant used.

With fresh lubricant, i.e. at start-up and just after relubrication, the friction coefficient rises briefly. It decreases again after a short running-in period.

Friction of the seals

The end wipers and the side seals also cause friction. These contact-type seals glide along the guide rail when the runner block is in motion, thus increasing the total friction of the linear guide. This type of friction is again increased when additional seal kits and front lube units (available as accessories) are installed. Frictional drag due to seals is highest in new linear guides, but decreases to a constant value after a short running-in phase.



Frictional force F_R due to seals

3 Profiled rail systems

3.1 Principles

3.1.2 Product selection

3.1.2.1 Product selection aids






Profiled rail systems can be pre-selected according to the following criteria:



- Sizes
- Runner block designs
- System characteristics
- Dynamic and static load ratios

Sizes

Various sizes are available, depending on the type of profiled rail system. The individual sizes have different load capacities.

The tables below show the sizes offered by Rexroth (as of 2005):

Profiled rail guide	Sizes	7	9	12	15	20	25	30	32	35	42	45	52	55	65	100	125
			-	-	-	✓	✓	✓	✓	-	✓	-	✓	-	✓	✓	-
Miniature ball rail system		✓	✓	✓	✓	✓	-	-	-	-	-	-	-	-	-	-	
eLINE ball rail system		-	-	-	✓	✓	✓	-	-	-	-	-	-	-	-	-	
Roller rail system		-	-	-	-	-	✓	-	-	✓	-	✓	-	✓	✓	✓	
Cam roller guide		-	-	-	-	✓	✓	-	✓	-	✓	-	✓	-	-	-	

Profiled rail guide	Sizes for wide profiled rail systems	20/40	25/70	35/90	55/85	65/100
		Wide ball rail system		✓	✓	✓
Wide roller rail system		-	-	-	✓	✓

3 Profiled rail systems

3.1 Principles

3.1.2 Product selection



















Runner block designs Each application makes different demands on the profiled rail systems used. There are different runner block designs to meet these different needs. The following coding system is used to identify all ball and roller runner block designs. (The code letters are based on the German product names.)

Cam roller guides are not covered by this system because of their special design.

Definition		Code (example)		
		F	N	S
Width	Flanged (F)	F		
	Slimline (S)			
	Wide (B)			
Length	Normal (N)		N	
	Long (L)			
	Short (K)			
Height	Standard height (S)			S
	High (H)			
	Low-profile (N)			

Identification system for runner blocks

The table below shows which runner block forms are available in each of the profiled rail systems:






Profiled rail guide	Width	F					S						B		
		N	N	L	K	K	N	N	N	L	L	K	K	N	L
Length	Height	Code													
Code	Code	FNS	FNN	FLS	FKS	FKN	SNS	SNN	SNH	SLS	SLH	SKS	SKN	BNN	BLS
Ball rail system															
		✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	-
Miniature ball rail system		-	-	-	-	-	✓	-	-	✓	-	-	-	-	✓
eLINE ball rail system		✓	-	-	-	-	✓	-	-	-	-	-	-	-	-
Roller rail system		✓	-	✓	-	-	-	-	✓	-	✓	-	-	-	✓






3 Profiled rail systems

3.1 Principles

3.1.2 Product selection

System characteristics The demands made on profiled rail systems vary according to the specific application.

Profiled rail guide	Load capacity	Rigidity	Accuracy	Friction	Maximum speed
Ball rail system 	+++	++	+++	+++	10 m/s
Miniature ball rail system 	++	++	++	+++	5 m/s
eLINE ball rail system 	++	+	++	+++	2 m/s
Roller rail system 	+++	+++	+++	++	3 m/s
Cam roller guide 	+	o	+	+++	10 m/s

Profiled rail guide	Short stroke characteristics	Noise characteristics	Lubrication requirement	Costs
Ball rail system 	++	+++	++	++
Miniature ball rail system 	++	++	+++	++
eLINE ball rail system 	++	++	+++	+++
Roller rail system 	+++	++	++	+
Cam roller guide 	++	+++	+++	++

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

3 Profiled rail systems

3.1 Principles

3.1.2 Product selection

Dynamic and static load ratios

The ratio between the load capacity of the runner block and the load applied to it can be used to pre-select the type of linear guide. The dynamic load ratio C/F_{\max} and the static load ratio $C_0/F_{0\max}$ should be chosen as appropriate for the application.

This permits calculation of the required load capacity and selection of the profiled rail guide type, the size and the runner block design using the load capacity tables given in the product catalogs.

$$\text{Dynamic load ratio} = \frac{C}{F_{\max}}$$

C = dynamic load capacity (N)
 F_{\max} = maximum dynamic load on bearing of the most highly loaded runner block (N)

If the static load $F_{0\max}$ is greater than F_{\max} , then:

$$\text{Static load ratio} = \frac{C_0}{F_{0\max}}$$

C_0 = static load capacity (N)
 $F_{0\max}$ = maximum static load on bearing of the most highly loaded runner block (N)
 F_{\max} = maximum dynamic load on bearing of the most highly loaded runner block (N)

If the static load $F_{0\max}$ is smaller than F_{\max} , the static load ratio is determined using the maximum dynamic load on the bearing:

$$\text{Static load ratio} = \frac{C_0}{F_{\max}}$$

Recommended values for load ratios

The table below contains recommendations for load ratios.

The values are offered merely as a rough guide reflecting typical customer requirements (e.g. service life, accuracy, rigidity) by sector and application.

Machine type / Industry sector		C/F_{\max}	$C_0/F_{0\max}$
Machine tools	Application example		
	General	6 ... 9	> 4
	Turning	6 ... 7	> 4
	Milling	6 ... 7	> 4
	Grinding	9 ... 10	> 4
	Engraving	5	> 3
Rubber and plastics processing machinery	Injection molding	8	> 2
Woodworking and wood processing machines	Sawing, milling	5	> 3
Assembly/handling technology and industrial robots	Handling	5	> 3
Oil hydraulics and pneumatics	Raising/lowering	6	> 4

3 Profiled rail systems

3.1 Principles

3.1.2 Product selection

3.1.2.2 Product selection procedure

Many different parameters must be considered to arrive at the optimal choice of profiled rail guide. Though the selection procedure described below is a typical one, it may not apply to all applications. For some applications it may be useful to switch the order of the steps involved. Often, the starting situations will be different. While new-build projects generally give designers full freedom of choice, the range of available options will be restricted at the outset when modifying

existing designs. Also, some types of guide are more commonly used in certain sectors and applications than in others. Another point to be considered at an early stage is the level of accuracy required, as this may eliminate some versions in the first place. It is therefore advisable to run through all the steps once to gain a better idea of the possible options before proceeding to select the product and perform the nominal life calculations.

Procedure	
Step 1	Define the requirements
Step 2	Select an appropriate profiled rail guide
Step 3	Define the layout for the profiled rail guide
Step 4	Define the preload class
Step 5	Perform the calculations
Step 6	Define the accuracy class
Step 7	Define the peripherals
Result	Ordering details with part numbers

Step 1: Define the requirements

When selecting profiled rail systems, the first step is to define the requirements and operating conditions for the application, as shown at right.

Requirements
Stroke length
Speed
Acceleration
Masses
Loads
Accuracy
Rigidity
Installation space
Travel cycles
Required life
Environmental conditions
Operating conditions
Additional functions (position measurement, drive, brakes)

3 Profiled rail systems

3.1 Principles

3.1.2 Product selection

Step 2:**Select an appropriate profiled rail guide**

The next step is to roughly calculate or estimate the expected loads for the individual runner blocks. The appropriate profiled rail system (type, size and runner block design) can then be selected using the load capacities and the selection charts. The load capacities can be found in the respective Rexroth product catalogs. The static and dynamic load ratios (C_0/F_{0max} and C/F_{max}) must also be taken into account here. The selection tables were introduced in section 3.1.2.1. They refer to the sizes, runner block designs, and system characteristics.

Profiled rail guides	Abbreviation	Section
Ball rail system	BRS	3.2
Miniature ball rail system	Mini BRS	3.3
eLINE ball rail system	eLINE BRS	3.4
Roller rail system	RRS	3.5
Cam roller guide	CRG	3.6

Step 3:**Define the layout for the profiled rail guide**

The layout for the profiled rail system now has to be defined (see section 3.1.3). Define the number of runner blocks and guide rails first. Then define the mounting orientation (horizontal, vertical, inclined, wall mounting or overhead mounting). Finally, determine how the guide rails and runner blocks are to be mounted and fastened, keeping the location and use of the reference edges in mind.

Layout parameters	Section
Number of runner blocks and guide rails	3.1.3.1
Mounting orientation	3.1.3.2
Guide rail mounting	3.1.3.3
Runner block mounting	3.1.3.4
Design of the adjoining structure	3.1.3.5

Step 4:**Define the preload class**

The preload class is chosen on the basis of the required rigidity. The rigidity charts should be consulted to check whether the desired rigidity will be achieved. If this check shows that the rigidity will not be high enough, the linear guide must be redimensioned. The tables listing the preload classes according to areas of use and applications can be used as a rough guide here.

	Section
Selection of the preload class	3.1.1.3

3 Profiled rail systems

3.1 Principles

3.1.2 Product selection

**Step 5:
Calculations**

Using the available data, calculate the nominal life and the static load safety factor. If the required values are not met, repeat steps one to four and select a more appropriate profiled rail guide. Rexroth provides a special design calculation service to assist with nominal life calculations.

Procedure for calculations	Section
Define the operating conditions	3.1.5.2
Forces and load moments	3.1.5.3
Combined equivalent load on bearing	3.1.5.4
Taking preload into account	3.1.5.5
Equivalent dynamic load on bearing	3.1.5.6
Nominal life	3.1.5.7
Equivalent static load on bearing	3.1.5.8
Static load safety factor	3.1.5.9

**Step 6:
Define the
accuracy class**

Once the nominal life requirements are fulfilled, the next step is to define the accuracy class. This depends heavily on the area and application in which the linear guide is to be used. Help is provided in the form of selection charts and tables.

	Section
Selection of the accuracy class	3.1.1.5

**Step 7:
Define the
peripherals**

The last step is to define the peripherals for the linear guide. This includes specifying the lubrication system and the in-service lubrication intervals. Adequate protection against life-shortening factors must also be selected, i.e. appropriate sealing and corrosion protection.

Rexroth linear guides can be equipped with additional functionalities such as clamping and braking units, rack and pinion drives, and integrated measuring systems.

Defining the peripherals		Section
Lubrication		3.1.6.1
Sealing		3.1.6.2
Corrosion protection		3.1.6.3
Additional functions	Clamping and braking units	3.1.6.4
	Rack and pinion drive	3.1.6.4
	Integrated measuring system	3.7

**Result:
Ordering details
with part numbers**

After this final step, all the required ordering details are known, including the part numbers for the runner blocks, guide rails and the required accessories.

3 Profiled rail systems

3.1 Principles

3.1.3 Profiled rail system layout

Defining the layout for the profiled rail guide

This section describes how to lay out profiled rail guides. The procedure involves several steps, which are summarized in the table. The different features and characteristics of the layout options are explained using typical installation scenarios.

Notes on the procedure:

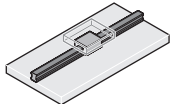
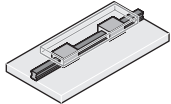
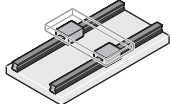
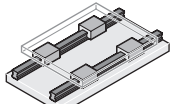
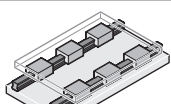
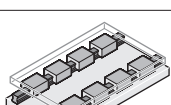
The procedure described below for defining the layout of the profiled rail guides is offered by Rexroth as a suggestion only. The actual procedure will depend heavily on the specific application, and the steps may need to be carried out in a different order.

Procedure	Section
Number of runner blocks and guide rails	3.1.3.1
Mounting orientation of the profiled rail guide	3.1.3.2
Guide rail mounting	3.1.3.3
Runner block mounting	3.1.3.4
Design of the adjoining structure	3.1.3.5

3.1.3.1 Number of runner blocks and guide rails

The number of runner blocks and guide rails used in an application has an influence on the system characteristics of the linear guide. These include the load capacity, rigidity, geometric travel performance, lubrication, and costs. The number of runner blocks installed also makes certain demands on the overall machine design, e.g. the required accuracy of the mounting bases and mating surfaces.

Both the number of runner blocks per guide rail and the number of guide rails themselves may vary. Typical combinations are shown here:

Number of runner blocks and rails	
One runner block – one rail	
Two runner blocks – one rail	
Two runner blocks – two rails	
Four runner blocks – two rails	
Six runner blocks – two rails	
Eight runner blocks – two rails	

3 Profiled rail systems






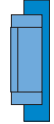




3.1 Principles

3.1.3 Profiled rail system layout

3.1.3.2 Mounting orientation of the profiled rail guide

The mounting orientations are defined below using a combination of one runner block and one rail as an example. The mounting orientation of the linear guide depends on the machine design and affects the lubrication behavior of the profiled

rail system. The lubrication must therefore be optimally adapted to the specific mounting orientation. Lubrication recommendations for all mounting orientations can be found in the respective product catalogs.

Mounting orientation rotated about the X-axis		Mounting orientation rotated about the Y-axis	
Horizontal mounting No rotation about the X-axis		Horizontal mounting No rotation about the Y-axis	
Inclined mounting Rotated 0° ... 90° about the X-axis		Inclined mounting Rotated 0° ... 90° about the Y-axis	
Wall mounting Rotated 90° about the X-axis		Vertical mounting Rotated 90° about the Y-axis	
Inclined mounting Rotated 90° ... 180° about the X-axis		Inclined mounting Rotated 90° ... 180° about the Y-axis	
Top-down mounting Rotated 180° about the X-axis		Top-down mounting Rotated 180° about the Y-axis	

3.1.3.3 Guide rail mounting

Standard guide rails can be bolted into place from above or below. This is done using socket head cap screws per ISO 4762. Recommendations on screw sizes and strength ratings are given in the product catalogs. For examples, see the following pages.

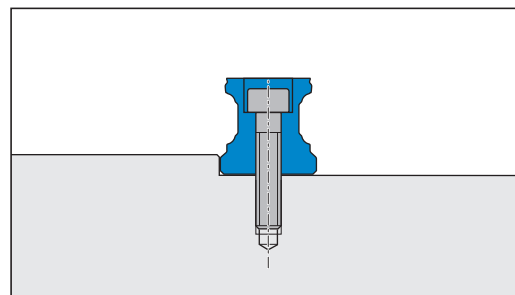
V-guide rails with a dovetail profile can be mounted using pressure pieces or by pressing them directly into the mounting base.

Mounting the guide rail from above

When guide rails are mounted from above, the mounting holes in the rail top have to be closed off. This is the only way to ensure that wiper seals will not be damaged and to prevent any dirt from collecting in the holes.

The holes can be closed off using:

- a cover strip
- plastic mounting hole plugs
- steel mounting hole plugs



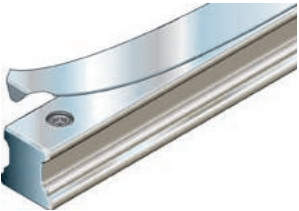
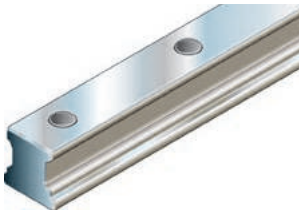
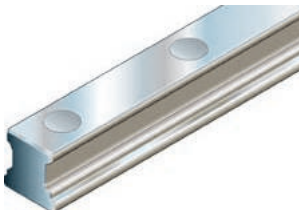
Guide rail mounted from above

3 Profiled rail systems

3.1 Principles

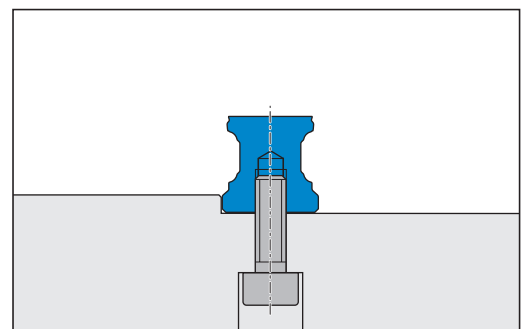
3.1.3 Profiled rail system layout

Options for closing mounting holes

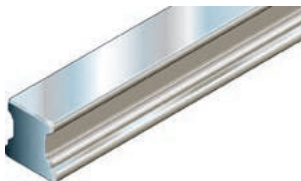
Closure type	Advantages	Disadvantages
Cover strip 	<ul style="list-style-type: none"> ■ A single cover for all holes ■ Reusable ■ Interchangeable ■ Can be retrofitted ■ Fast clip-on mounting ■ Very good sealing action in combination with wiper seals at runner block ends 	<ul style="list-style-type: none"> ■ Strip ends have to be secured ■ Extra space required for securing strip at rail ends
Plastic mounting hole plugs 	<ul style="list-style-type: none"> ■ Low cost ■ No extra space needed at rail ends 	<ul style="list-style-type: none"> ■ Not reusable ■ Each single hole has to be plugged ■ Reduced sealing action of wiper seals at runner block ends ■ Sensitive to mechanical stressing (e.g. impacts)
Steel mounting hole plugs 	<ul style="list-style-type: none"> ■ Very rugged ■ No extra space needed at rail ends ■ Good sealing action in combination with wiper seals at runner block ends ■ Resistant to mechanical stressing (e.g. impacts) 	<ul style="list-style-type: none"> ■ Not reusable ■ Each single hole has to be plugged ■ High mounting effort (special tool required) ■ Very expensive

Mounting the guide rail from below

In this case, the mounting holes do not have to be closed off. In addition, certain sizes of Rexroth rail systems allow the use of stronger screws than permissible for mounting from above. This increases both the rigidity of the system and the permissible side loads.



Guide rail mounted from below

Rails for mounting from below	Advantages	Disadvantages
	<ul style="list-style-type: none"> ■ No covers required ■ No extra space needed at rail ends ■ Very good sealing action in combination with wiper seals at runner block ends 	<ul style="list-style-type: none"> ■ Mounting/removal could be difficult, depending on configuration of adjoining structures ■ Often not possible, due to machine design

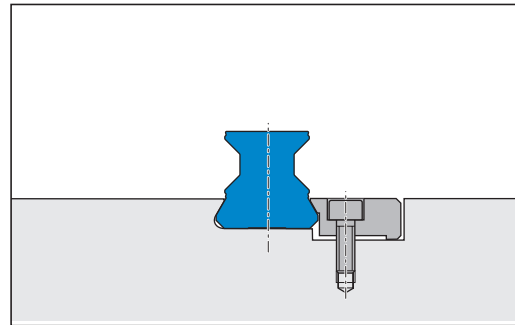
3 Profiled rail systems

3.1 Principles

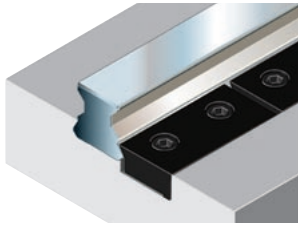
3.1.3 Profiled rail system layout

Mounting of V-guide rail using pressure pieces

Rexroth offers guide rails with a dovetail profile (V-guide rails) for the most commonly used sizes. The rail itself contains no mounting holes. It is mounted by inserting it into a suitably fabricated machine bed and wedged into place from the side using screw-down pressure pieces. V-guide rails with pressure pieces are available for roller rail systems.

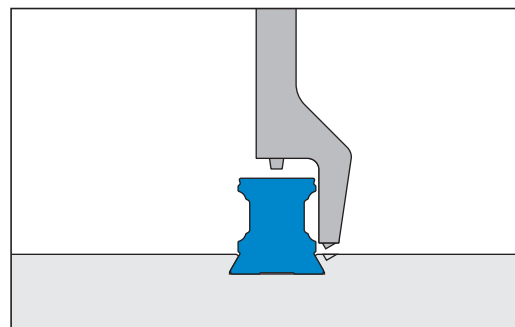


V-guide rail with pressure piece

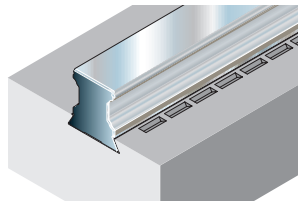
V-guide rail with pressure pieces	Advantages	Disadvantages
	<ul style="list-style-type: none"> ■ No rail cover/plugs required ■ No screws directly in the rail, therefore no waviness ■ No additional lateral retention required ■ No extra space needed at rail ends 	<ul style="list-style-type: none"> ■ Extra space needed at side ■ High effort required to prepare the machine bed and mount the rails

Press-fitting the V-guide rail into mounting base

V-guide rails can also be mounted by pressing (levering) them into the mounting base. The rail is held firmly in place due to plastic deformation of the mounting base. The mounting base must be made from a suitable material, e.g. aluminum. Press-fitting reduces the costs for manufacturing the adjoining structure and for mounting the guide rail. V-guide rails for press-fitting are available for ball rail systems.



V-guide rail mounted by press-fitting

V-guide rail for press-fitting	Advantages	Disadvantages
	<ul style="list-style-type: none"> ■ No cover/plugs required ■ No screw-fasteners ■ Fast mounting ■ Reduced costs ■ No need to bore holes or tap threads in the adjoining structure 	<ul style="list-style-type: none"> ■ Extra space needed at side ■ Requires special mounting tools ■ Reduced load-bearing capability due to aluminum mounting base

3 Profiled rail systems


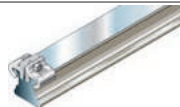
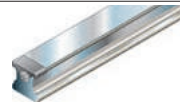

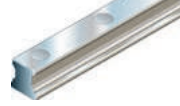
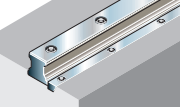

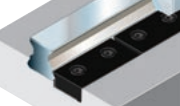
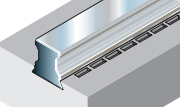
3.1 Principles

3.1.3 Profiled rail system layout

Comparison of mounting options and mounting times

The choice of rail mounting option will depend on the specific application. The following table shows nine different mounting options and the mounting time required in each case. The infor-

mation relates to a size 25 ball guide rail with a rail length of 536 mm, mounted using 9 screws.

Option	Illustration	Description	Mounting time in %
1		Guide rail mounted from above. The mounting holes are not plugged. This option is not recommended by Rexroth. It is shown here merely for the purpose of comparison.	100
2		Guide rail mounted from above and with cover strip. The strip is secured by a clamp at each end.	125
3		Guide rail mounted from above and with cover strip. The strip is secured by screw-down protective caps.	130
4		Guide rail mounted from above. The mounting holes are then closed with plastic plugs.	125
5		Guide rail mounted from above. The mounting holes are then closed with steel plugs, which are pressed in using a special tool.	225
6		Guide rail mounted from above. Additional lateral retention is provided in the form of a wedge profile. This option with open mounting holes is shown merely for the purpose of comparison.	180
7		Guide rail mounted from below.	130
8		V-guide rail mounted using pressure pieces. Since the pressure pieces already provide lateral retention, this option can be compared with option 6. The mounting time is considerably shorter.	130
9		V-guide rail pressed into aluminum mounting base.	115

3 Profiled rail systems

3.1 Principles

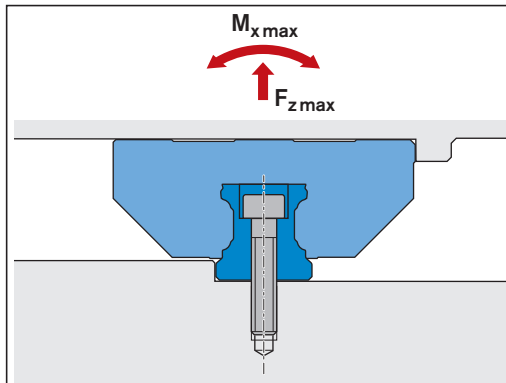
3.1.3 Profiled rail system layout

Load on the screw connections of guide rails

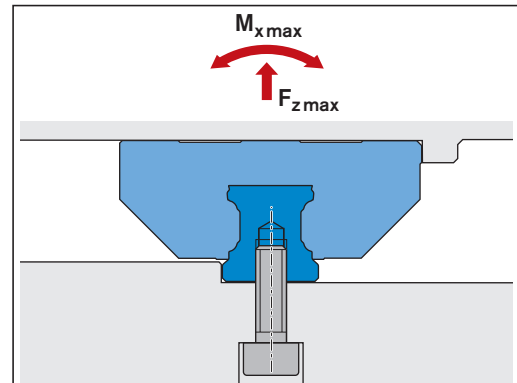
The high performance capability of profiled rail systems may cause the load limits for screw connections as specified in DIN 645-1 to be exceeded. The most critical point is the screw connection between the guide rail and the mounting base. If the lift-off loads F_z or moments M_x are higher than the maximum permissible loads $F_{z,max}$

and moments $M_{x,max}$ shown in the table, the screw connections must be recalculated.

Details of the permissible loads are given in the respective product catalogs. The table shows an extract from the ball rail systems catalog.



Loads on the screw connections for a guide rail mounted from above



Loads on the screw connections for a guide rail mounted from below

The table shows examples of the maximum permissible values for lift-off loads and moments acting on runner blocks in relation to the screw-connections of the guide rails:

Guide rail	Size	Static lift-off loads F_z and moment loads M_x					
		Short runner block		Normal runner block		Long runner block	
		FKS, SKS, FKN, SKN		FNS, SNS, SNH, FNN, SNN		FLS, SLS, SLH	
		$F_{z,max}$ N	$M_{x,max}$ Nm	$F_{z,max}$ N	$M_{x,max}$ Nm	$F_{z,max}$ N	$M_{x,max}$ Nm
Mounted from above	20	10 000	90	11 700	106	13 400	121
	25	14 600	154	17 100	180	19 500	205
	30	–	360	32 400	420	37 100	480
Mounted from below	20	–	128	16 500	149	18 900	170
	25	14 300	150	16 700	170	19 100	200
	30	–	350	31 700	410	36 200	470

Sample values from the ball rail systems product catalog

The values shown in the table apply under the following conditions:

- Mounting screw quality 12.9
- Screws tightened to the specified torque
- Screws lightly oiled
- Rails screwed down to steel or cast iron bases
- Screw-in depth at least 2x thread diameter
- For screws in quality 8.8, an approximation factor of 0.6 can be applied.

If any side loads act on the linear guide, the screw connections must additionally be checked for lateral strength (see section 3.1.3.5).

3 Profiled rail systems

3.1 Principles

3.1.3 Profiled rail system layout

Composite guide rails

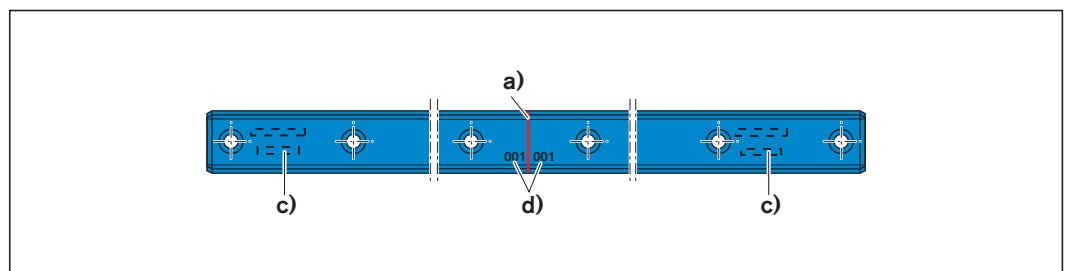
One-piece guide rails can only be supplied up to a certain maximum length. This maximum length depends on the type, size and configuration of the profiled rail system. If longer rail lengths are required, these can be supplied as composite

rails made up of matching rail sections mounted end to end. The end faces of the rail sections are machined to ensure a seamless transition at the joints with full load-bearing capability.

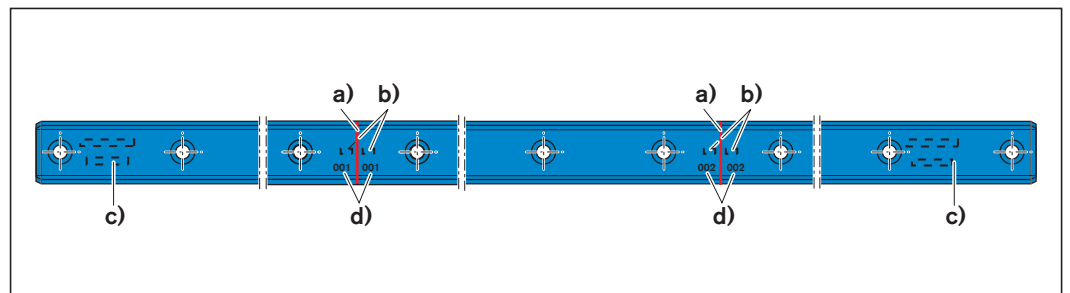
Identification of composite guide rails

In composite rails the full identification code is marked on both the first and the last rail section. The joints (a) are marked with a number (d) identifying the specific joint. They are also marked with

a serial rail number (b) to ensure that the sections can be mounted in the right order.



Guide rail made up of two sections



Guide rail made up of three or more sections

- a) Joint
- b) Serial rail number
- c) Full rail identification on first and last sections
- d) Joint number

3 Profiled rail systems

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3.1.3 Profiled rail system layout

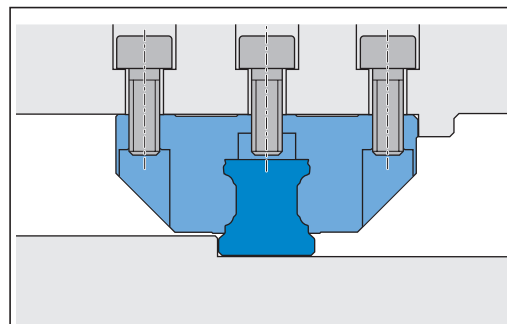
3.1.3.4 Runner block mounting

Just as with the guide rails, the runner block range also offers the possibility of fastening attachments from above or below. However, care should be taken to select the correct runner block

version (see below) for the chosen mounting method. If required, the runner blocks can be additionally secured by pinning.

Top-down fastening of attachments to runner block

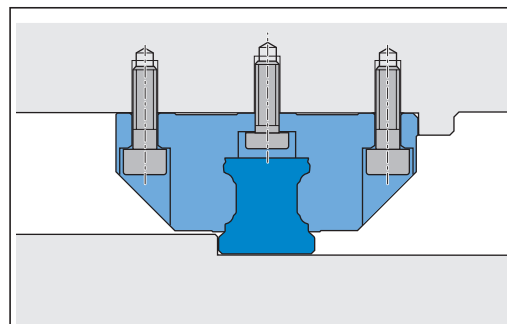
All runner blocks have standardized threaded mounting holes for top-down fastening of attachments. The threads are metric sizes 4, 6 or 9. The number of holes and the mounting hole pattern depend on the type, size and version of runner block.



Example of top-down fastening

Bottom-up fastening of attachments to runner blocks

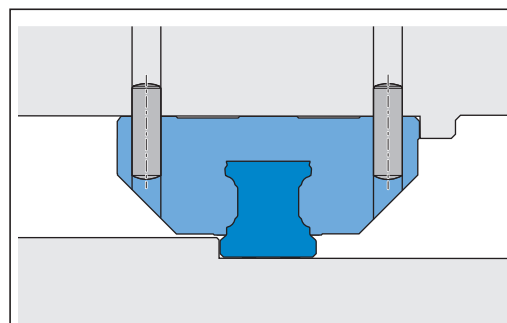
The flanged runner block versions also allow bottom-up fastening of attachments. The mounting hole pattern for this option is standardized. The middle mounting holes will only accommodate socket head cap screws with a low-profile head as specified in DIN 6912.



Example of bottom-up fastening

Pinning of runner blocks

Runner blocks can also be pinned to increase their lateral rigidity. The positions for pin holes are indicated in the respective catalogs. Pre-drilled holes made for production purposes may already exist at these positions. These holes can be bored open. Hardened tapered or straight pins per ISO 8734 can be used for runner block pinning.



Pinning for added security

3 Profiled rail systems

3.1 Principles

3.1.3 Profiled rail system layout

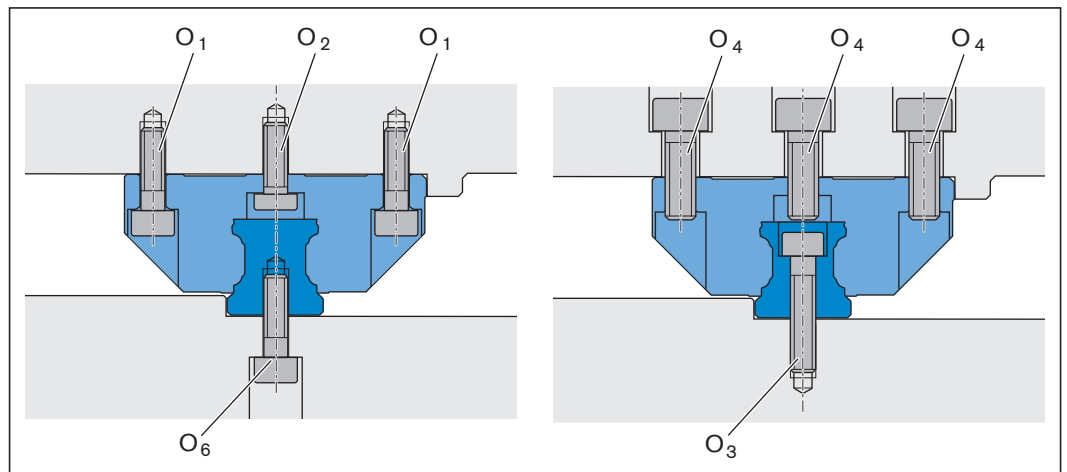
3.1.3.5 Design of the adjoining structure

Permissible side loads without lateral retention

When side loads are to be expected, it is essential to check whether the chosen screw-fasteners will be capable of transmitting these forces. This check can be done using the tables provided in the product catalogs. The tables contain size-related values for permissible side loads without lateral retention. These values are listed

with reference to the dynamic load capacity *C*. If the maximum permissible value is exceeded, reinforcement must be provided in the form of a reference edge, lateral retention or pinning.

The table below shows sample data from the ball rail systems catalog.



Examples of screw-fasteners used to mount runner blocks and guide rails

Example:

A size 25 FNS runner block is to be mounted by top-down fastening using six socket head cap screws (*O*₄, M8 x 20, strength class 12.9, per

ISO 4762). The guide rail is also to be mounted by top-down fastening with socket head cap screws (*O*₃, M6 x 30, strength class 12.9, per ISO 4762).

Size	Screw sizes Runner block			Guide rail	
	<i>O</i> ₁ ISO 4762 4 pcs.	<i>O</i> ₂ DIN 6912 2 pcs.	<i>O</i> ₄ ISO 4762 6 pcs.	<i>O</i> ₃ ISO 4762	<i>O</i> ₆ ISO 4762
20	M5x16	M5x12	M6x16	M5x25	M6x16
25	M6x20	M6x16	M8x20	M6x30	M6x20
30	M8x25	M8x16	M10x20	M8x30	M8x20

Screw strength class	Permissible side load without lateral retention				
	Runner block			Guide rail	
	<i>O</i> ₁	<i>O</i> ₂	<i>O</i> ₄	<i>O</i> ₃	<i>O</i> ₆
8.8	11% C	15% C	23% C	6% C	6% C
12.9	18% C	22% C	35% C	10% C	10% C

Sample values from the ball rail systems product catalog

Result:

The runner block can be subjected to a maximum side load of 0.35 *C* (35% *C*). The screws used to fasten the guide rail can only transmit up to 0.10 *C* (10% *C*) without a reference edge or lateral retention.

Any higher load will require a reference edge, lateral retention or pinning.

3 Profiled rail systems

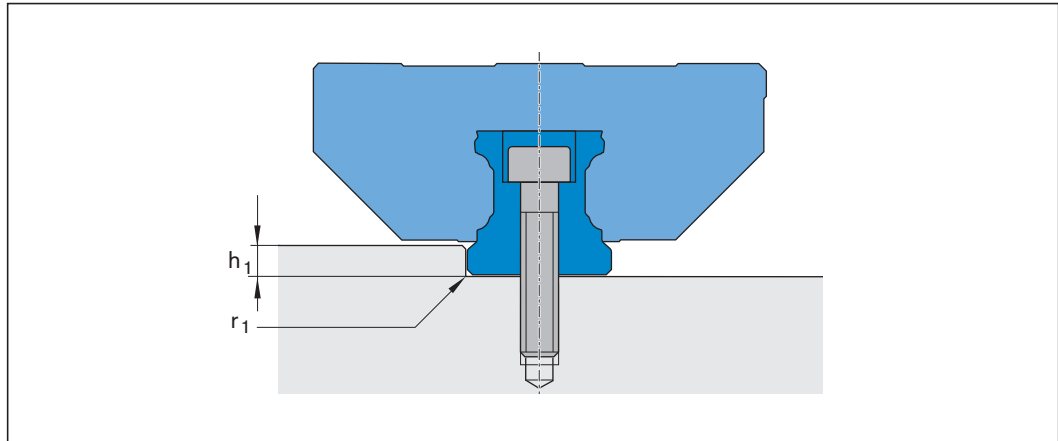
3.1 Principles

3.1.3 Profiled rail system layout

Reference edges

To ensure precise and easy mounting and to transmit high side loads the structure adjoining the guide rail should be provided with a reference edge. The height h_1 and the radius r_1 of the reference edge are given in the product catalogs. A tolerance range with minimum and maximum values is specified for the height of the reference edge. If these tolerances are adhered to, there

will be sufficient clearance between the fixed mounting base and the movable runner block. The radius r_1 is a maximum value, which ensures that the beveled guide rail can be mounted without problems.

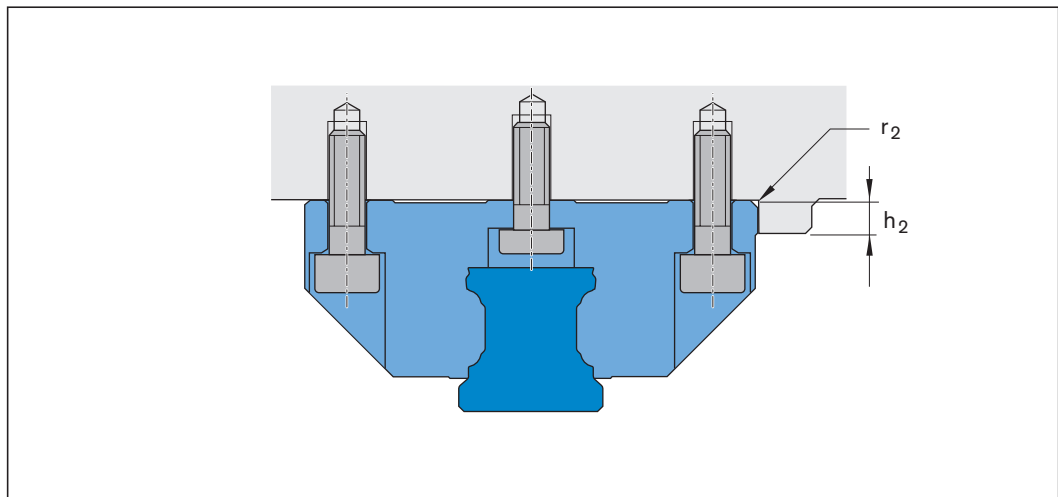
Reference edges in the mounting base

Reference edge in the mounting base

Reference edges in runner block attachments

Reference edges can be just as useful for runner blocks as they are for guide rails. They simplify mounting of the attachment, make installation more precise, and allow higher side loads to be taken up. The height h_2 and the radius r_2 must be

taken into account when designing the adjoining structure. The height h_2 of the reference edge ensures that the side loads can be transmitted without problems. The respective values can be found in the product catalogs.



Reference edge in the runner block attachment

3 Profiled rail systems

3.1 Principles

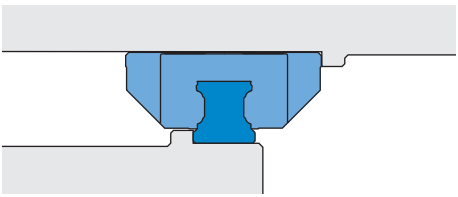
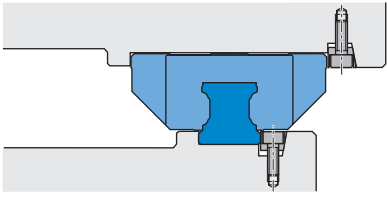
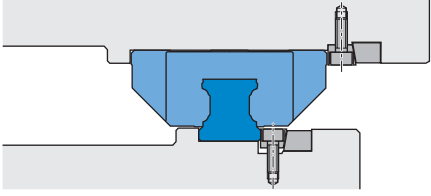
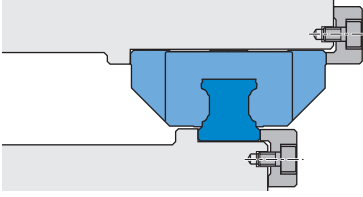
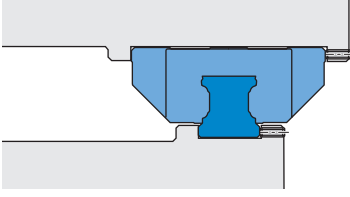
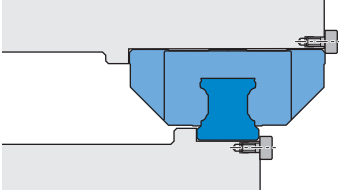
3.1.3 Profiled rail system layout

Lateral retention

Lateral retention permits higher side loads to be transmitted and facilitates precise alignment of the profiled rail guide. In layouts with several guide rails, a reference edge should be provided for the main guide rail. This greatly reduces the effort required to align the other guide rails. The same applies to the runner blocks.

A variety of elements can be used to provide additional lateral retention on the opposite side. These elements are also suitable for taking up side loads. In this case, however, the permissible side forces will depend heavily on the type or design of retaining element chosen. The strength of the element must always be checked.

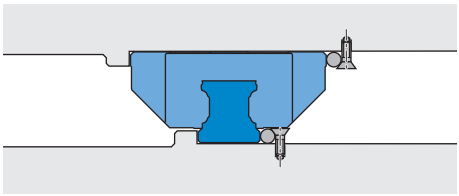
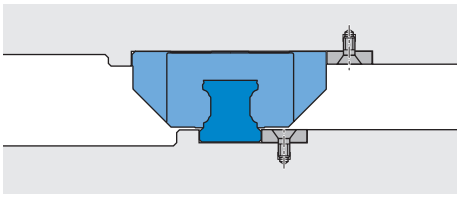
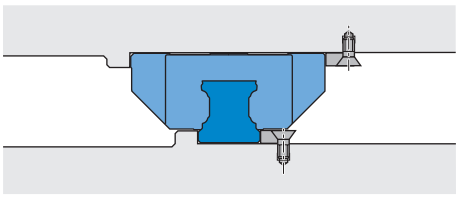
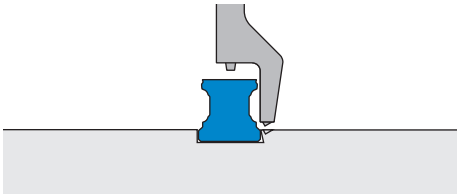
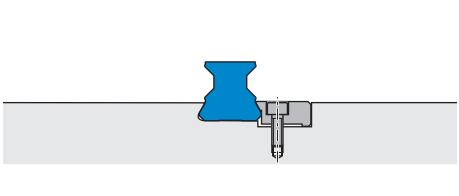
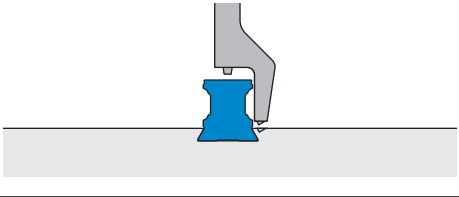
Lateral retention options for guide rails and runner blocks

Option	Type of lateral retention	
1	Reference edge for runner block and guide rail	
2	Wedge profile	
3	Double wedge profile	
4	Clamping strip	
5	Adjusting screw	
6	Clamping screw	

3 Profiled rail systems

3.1 Principles

3.1.3 Profiled rail system layout

Option	Type of lateral retention	
7	Shaft with countersunk screw	
8	Clamping piece with countersunk screw	
9	Clamping piece with tapered countersink	
10	Press-fitted rail	
11	V-guide rail with pressure piece	
12	Press-fitted V-guide rail	

Layout of reference edges and lateral retention elements

When specifying the layout and number of reference edges and/or lateral retention elements for guide rails and runner blocks, the following four factors must be taken into account:

- Load
- Accuracy
- Mounting method
- Geometry

An additional factor is the cost, which should always be checked when selecting the lateral retention option. This includes checking the cost of purchase and manufacturing as well as the expected installation costs.

3 Profiled rail systems

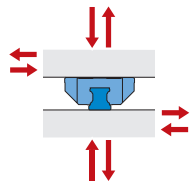
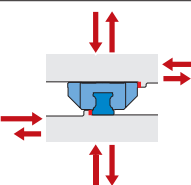
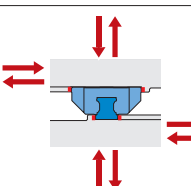
3.1 Principles

3.1.3 Profiled rail system layout

Loads

The magnitude and direction of the loads acting on the linear guide unit determine the number and arrangement of the lateral reference edges. If the permissible values for side loads (see 3.1.3.5) are exceeded, reference edges or additional lateral retention must be provided. Reference edges and lateral retention also increase the rigidity of the system. These reinforcements are therefore recommended for applications involving impacts and vibration.

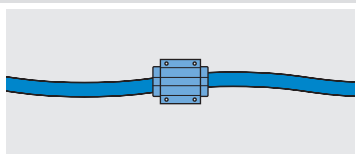
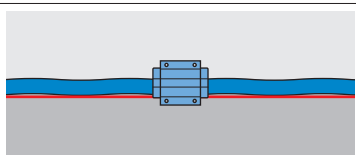
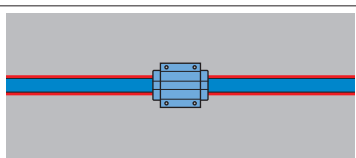
For the transmission of side loads the reference edges should be arranged according to the force flow in the system. Purely vertical loads (lift-off and down loads) have no effect on the layout of the reference edges. The exception here is the V-guide rail, where lateral retention is an integral feature of the mounting method. The examples below illustrate the various load scenarios:

Loads	Reference edges	Features
	<ul style="list-style-type: none"> Without reference edges Mounting of runner block and guide rail without reference edges 	Suitable for transmitting low side loads which can be transmitted via the screw connections.
	<ul style="list-style-type: none"> 2 reference edges Mounting of runner block and guide rail with one reference edge each (on opposite sides) 	Suitable for transmitting high side loads in one direction.
	<ul style="list-style-type: none"> 4 reference edges Mounting of runner block and guide rail with two reference edges each (additional lateral retention in each case) 	Suitable for transmitting alternating side loads. The higher load should be transmitted through the reference edges.

Accuracy

Though the loads themselves might not always require a reference edge, it is often necessary to mount the guide rail with a reference edge,

in order to achieve the required accuracy of the overall installation.

Illustration	Accuracy	Features
	<ul style="list-style-type: none"> Without reference edge Low to high accuracy (depending on mounting method) 	The accuracy is determined by the straightness of the rail. Higher levels of accuracy can be achieved by aligning the rail during installation.
	<ul style="list-style-type: none"> 1 reference edge High accuracy 	The accuracy is determined by the precision with which the rail is pushed against the reference edge during installation and by the straightness of the reference edge.
	<ul style="list-style-type: none"> 2 reference edges Very high accuracy 	The accuracy is determined by the precision of mounting and by the straightness of the reference edge and the lateral retention.

Note: The straightness deviations of the guide rail have been deliberately exaggerated in the illustrations.

3 Profiled rail systems

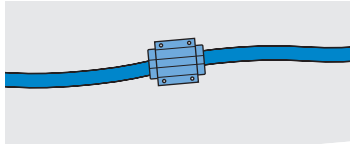
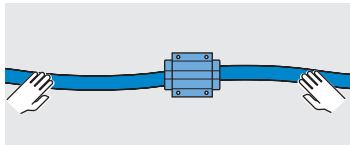
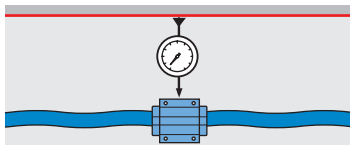
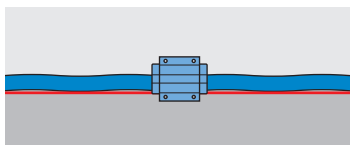
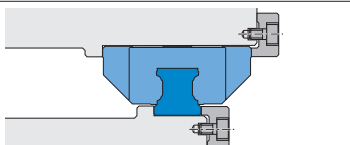
3.1 Principles

3.1.3 Profiled rail system layout

Mounting

As mentioned above, precise mounting can increase the accuracy of the guide. If the guide rail is precisely aligned before tightening the mounting screws, it may be possible to dispense with reference edges.

The mounting requirements must be carefully considered before deciding whether and where reference edges will be needed. The product-specific mounting instructions should be consulted when planning the mounting procedure.

Illustration	Description	Features
	<ul style="list-style-type: none"> ■ No alignment ■ Not permitted! 	No accuracy
	<ul style="list-style-type: none"> ■ Manual alignment ■ Not recommended! 	Low accuracy
	<ul style="list-style-type: none"> ■ Alignment using aids (measuring gauge, mounting runner block) and a reference plane 	Moderate to high accuracy
	<ul style="list-style-type: none"> ■ Alignment by pressing rail against a reference edge, without lateral retention 	High accuracy
	<ul style="list-style-type: none"> ■ Alignment against reference edge, with lateral retention 	Very high accuracy

Note: The straightness deviations of the guide rail have been deliberately exaggerated in the illustrations.

Geometry

Reference edges, lateral retention elements and their mounting accessories all require extra space.

The use of these mounting aids should therefore be checked for compatibility with the machine design.

3 Profiled rail systems

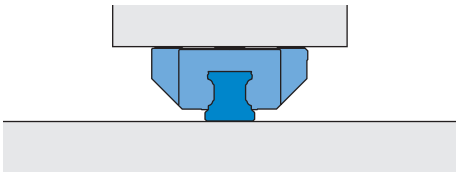
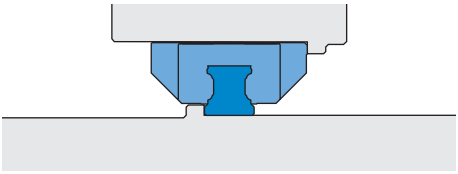
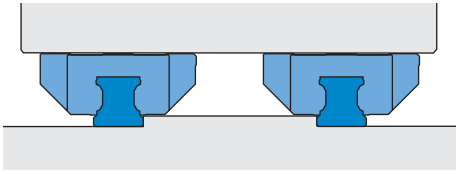
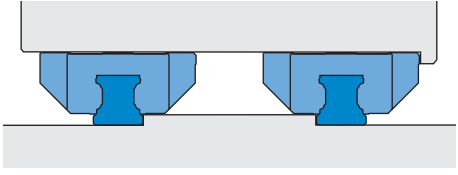
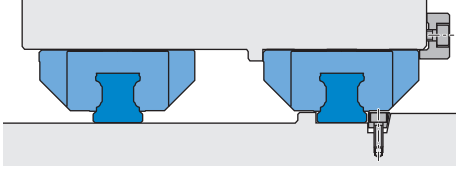
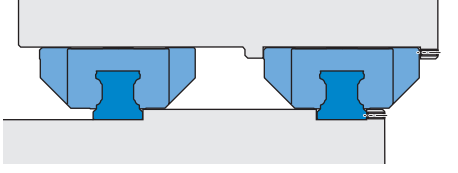
3.1 Principles

3.1.3 Profiled rail system layout

3.1.3.6 Installation scenarios

The following illustrations show typical installation scenarios. These scenarios reflect the layouts, mounting orientations, mounting methods and design criteria described earlier. Most of them relate to applications with 2 rails and 4 runner blocks,

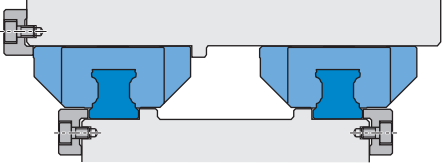
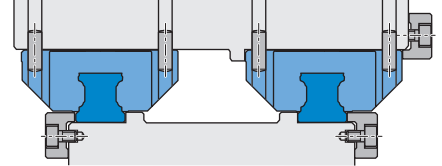
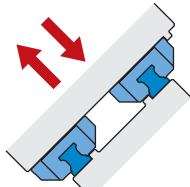
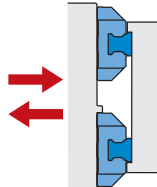
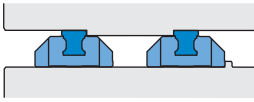
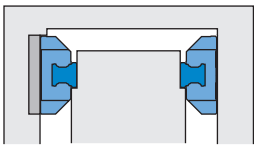

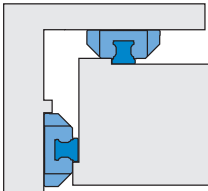
as this is the most commonly used combination. Provided as design aids, these examples give an insight into the great variety of configurations that can be found in practice.

Installation scenario	Description	
1	<ul style="list-style-type: none"> ■ No reference edges ■ For high lift-off and down loads ■ Low side loads possible ■ High mounting effort 	
2	<ul style="list-style-type: none"> ■ Runner block and guide rail with one reference edge each (on opposite sides) ■ Higher side loads from one direction permitted ■ Easy mounting due to reference edges ■ High accuracy 	
3	<ul style="list-style-type: none"> ■ Both guide rails with one reference edge ■ Runner blocks without reference edges 	
4	<ul style="list-style-type: none"> ■ Both guide rails with one reference edge ■ One runner block with reference edge ■ Easy mounting ■ High accuracy ■ Suitable for high side loads from one direction 	
5	<ul style="list-style-type: none"> ■ Runner block and guide rail on one side with reference edge and lateral retention ■ Runner block with reference edge takes up all side loads ■ For high side loads from both directions ■ Easy mounting 	
6	<ul style="list-style-type: none"> ■ First guide rail with reference edge, second with reference edge and lateral retention ■ Runner block with reference edge and lateral retention takes up side loads ■ High accuracy 	

3 Profiled rail systems

3.1 Principles

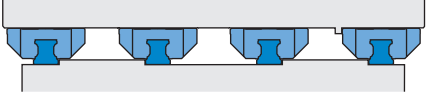
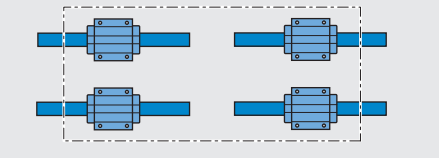
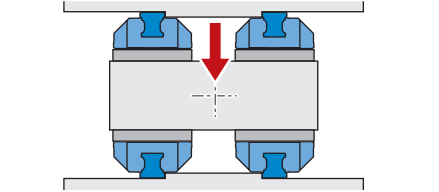
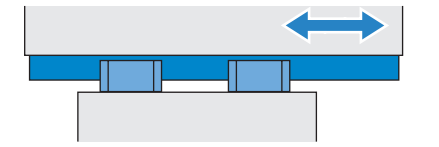
3.1.3 Profiled rail system layout

Installation scenario	Description	
7	<ul style="list-style-type: none"> ■ One runner block and two guide rails braced via reference edges and lateral retention ■ High side loads possible ■ Very high accuracy 	
8	<ul style="list-style-type: none"> ■ Runner blocks and guide rails braced via reference edges and lateral retention ■ Pinning in addition to screw-fasteners for very high side loads ■ Very high accuracy 	
9	<ul style="list-style-type: none"> ■ Inclined installation, rotated 45° about X-axis ■ For very high loads acting at an angle of 45° 	
10	<ul style="list-style-type: none"> ■ Vertical installation, rotated 90° about X-axis (wall mounting) ■ Both guide rails with reference edge ■ For high horizontal loads ■ High accuracy 	
11	<ul style="list-style-type: none"> ■ Top-down installation, rotated 180° about X-axis ■ For mainly vertical loads ■ Higher side loads from one direction possible 	
12	<ul style="list-style-type: none"> ■ For loads acting mainly from above and from the side ■ Low space requirement ■ High mounting effort ■ Intermediate adapter plate required 	
13	<ul style="list-style-type: none"> ■ For mainly horizontal loads ■ Low space requirement ■ High mounting effort ■ Intermediate adapter plate required 	
14	<ul style="list-style-type: none"> ■ "L" layout to take up moment loads ■ High moment load capacity 	

3 Profiled rail systems

3.1 Principles

3.1.3 Profiled rail system layout

Installation scenario	Description	
15	<ul style="list-style-type: none"> ■ Four guide rails to carry extremely heavy loads ■ Very high rigidity 	
16	<ul style="list-style-type: none"> ■ Short travel distances within a relatively large machine ■ Four short rails 	
17	<ul style="list-style-type: none"> ■ Very high rigidity with load acting centrally ■ Very high mounting effort ■ Intermediate adapter plates required 	
18	<ul style="list-style-type: none"> ■ Guide rail travels ■ Runner blocks stationary 	

3 Profiled rail systems

3.1 Principles

3.1.4 Design notes

This section provides information for design engineers on how to plan and install profiled rail systems. The following topics are dealt with in detail:

- Installation tolerances
- Guidelines for economical designs

3.1.4.1 Installation tolerances

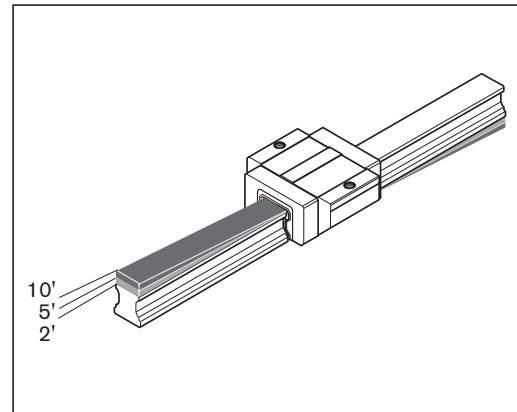
A number of tolerances must be met to ensure that a profiled rail system will deliver full performance. Deviations from the specified values can shorten the life of the guide. As long as the deviations are kept within the tolerance limits, the effect on the service life can generally be neglected.

The installation tolerances relate to:

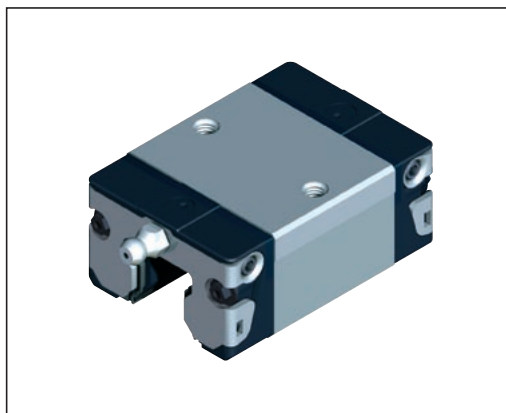
- Vertical offset
- Parallelism offset of mounted rails
- Tolerances for different installation situations
- Surface finish details

Vertical offset

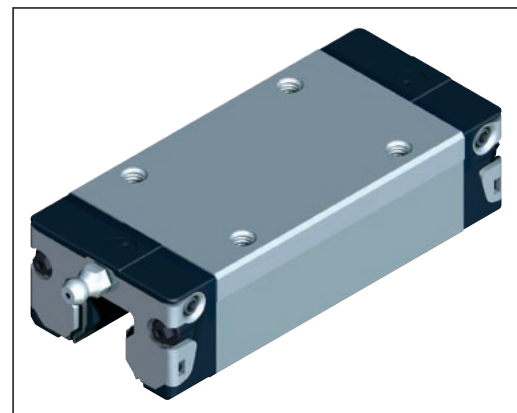
Permissible tolerances are specified for vertical offsets in the longitudinal and transverse directions. These tolerances vary according to the design of the runner block. "Super" runner blocks, which have a self-alignment capability, and runner blocks made from aluminum can compensate for larger errors than runner blocks made from steel. The runner block version (e.g. long or short) also determines the tolerance limits. Specific details can be found in the respective product catalogs.



Self alignment capability in "Super" runner blocks



Short ball runner block (SKS)



Long ball runner block (SLS)

3 Profiled rail systems

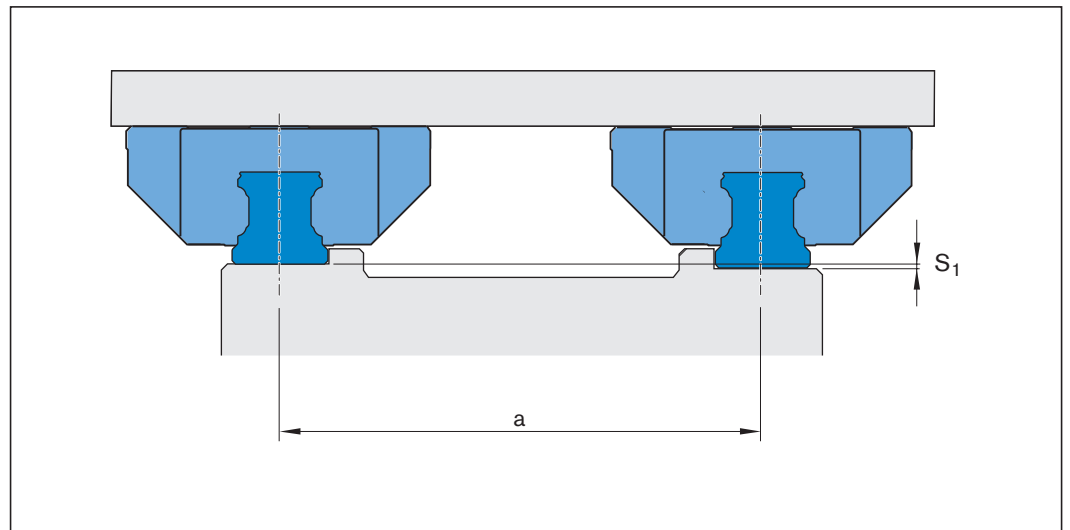
3.1 Principles

3.1.4 Design notes

Permissible vertical offset in the transverse direction S_1

The permissible vertical offset in the transverse direction is calculated from the distance between guide rails "a" and a calculation factor Y, which depends on the preload class (C0, C1, C2, C3) of the runner blocks.

The calculation factor Y also depends on the runner block design (steel, aluminum, "Super" runner block) and on its length. For short runner blocks, the offset can be 20% higher than the permissible value for standard-length runner blocks.



Vertical offset in the transverse direction

$$(3-8) \quad S_1 = a \cdot Y$$

S_1 = permissible vertical offset (mm)
 a = distance between guide rails (mm)
 Y = calculation factor (-)

Example for a Rexroth ball rail system with 4 rows of balls:

Runner block	Length	Calculation factor Y for preload class			
		C0	C1	C2	C3
Steel runner block	Standard/long	$4.3 \cdot 10^{-4}$	$2.8 \cdot 10^{-4}$	$1.7 \cdot 10^{-4}$	$1.2 \cdot 10^{-4}$
	Short	$5.2 \cdot 10^{-4}$	$3.4 \cdot 10^{-4}$	$2.0 \cdot 10^{-4}$	$1.4 \cdot 10^{-4}$
"Super" runner block	Short	$8.0 \cdot 10^{-4}$	$6.0 \cdot 10^{-4}$	–	–
Aluminum runner block	Standard	$7.0 \cdot 10^{-4}$	$5.0 \cdot 10^{-4}$	–	–

3 Profiled rail systems

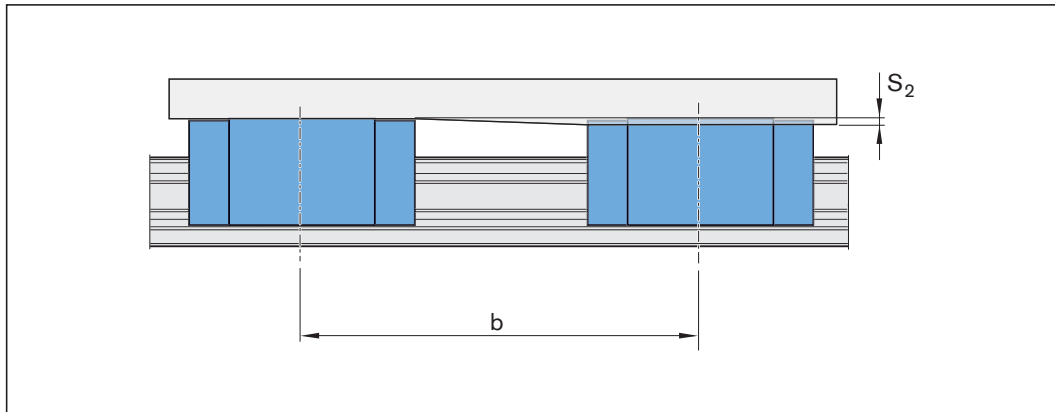
3.1 Principles

3.1.4 Design notes

Permissible vertical offset in the longitudinal direction S_2

The permissible vertical offset in the longitudinal direction is calculated from the distance between runner blocks "b" and a calculation factor X, which depends on the material of the runner blocks (steel/aluminum) and on their length.

The values for long runner blocks are approx. 30% lower and the values for short runner blocks approx. 40% higher than the limits for standard-length runner blocks.



Vertical offset in the longitudinal direction

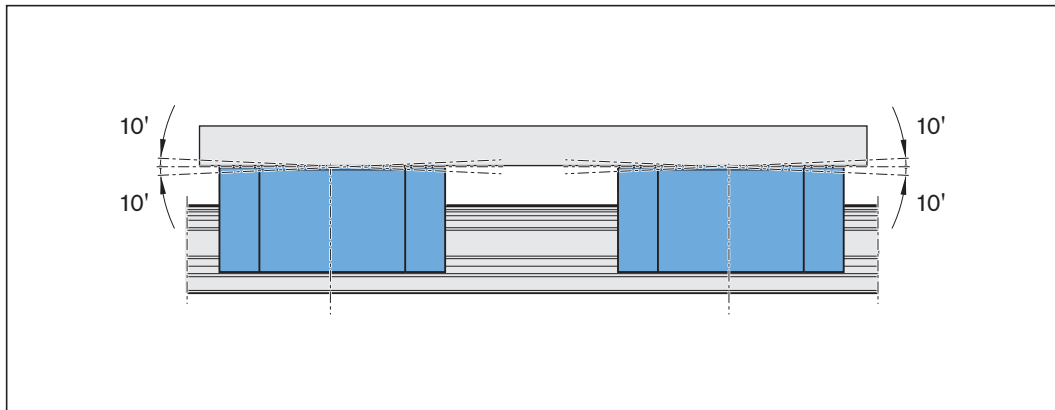
$$(3-9) \quad S_2 = b \cdot X$$

- S_2 = permissible vertical offset (mm)
- b = distance between runner blocks (mm)
- X = calculation factor (-)

Example for a Rexroth ball rail system with 4 rows of balls:

Runner block	Calculation factor X for runner block length		
	Standard	Long	Short
Steel runner block	$4.3 \cdot 10^{-5}$	$3.0 \cdot 10^{-5}$	$6.0 \cdot 10^{-5}$
Aluminum runner block	$6.0 \cdot 10^{-5}$	-	-

"Super" runner blocks can compensate for longitudinal offsets of up to 10' due to their self-alignment capability.



Compensation of offsets in "Super" runner blocks

3 Profiled rail systems

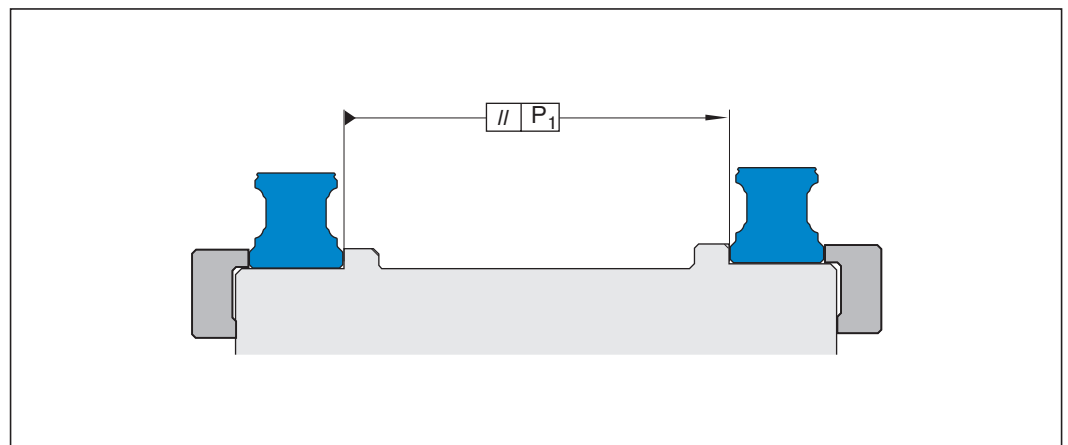
3.1 Principles

3.1.4 Design notes

Parallelism offset of mounted rails

If the guide rails are not aligned parallel to each other, stresses will arise in the guide system, resulting in additional loads and thus shortening the service life. To make sure that this does not happen, the parallelism offset P_1 must be observed. As long as the values specified in the table are met, the effect of parallelism offsets on the service life can generally be neglected. The tolerances depend on the specific installation conditions. In precision installations the adjoining structures are rigid and highly accurate. In

standard installations the adjoining structures are compliant, allowing parallelism offset tolerances up to twice those for precision installations. The parallelism offset values apply to all runner blocks in the standard range and depend on the preload and the material of the runner block. For short runner blocks, the offset can be 20% higher than the permissible value for standard-length runner blocks.



Parallelism of guide rails

Effect of the runner block version on the parallelism offset, using a size 25 ball rail system as an example:

Runner block	Parallelism offset P_1 (mm) for preload class			
	C0	C1	C2	C3
Steel runner block in precision installations	0.019	0.012	0.007	0.005
Short steel runner block in precision installations	0.023	0.014	0.008	0.006
Steel runner block in standard installations	0.038	0.024	0.014	0.010
Short steel runner block in standard installations	0.046	0.029	0.017	0.012
“Super” runner block	0.032	0.023	–	–
Aluminum runner block	0.026	0.017	–	–

Tolerances for different installation situations

If so requested by customers, Rexroth can check the mounting base and attachments to determine whether they meet the accuracy requirements. This check covers the entire installation situation,

including specification of all necessary shape and positional tolerances. The calculated nominal life can only be achieved when these tolerances are observed.

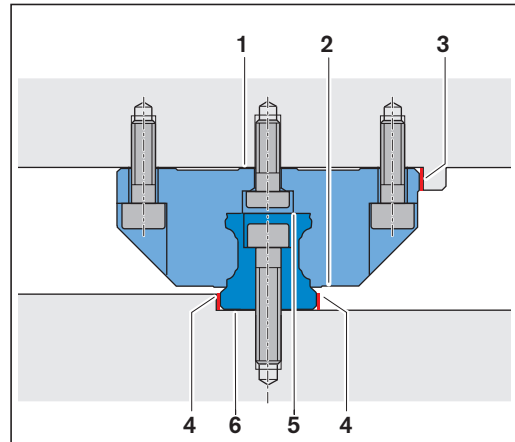
3 Profiled rail systems

3.1 Principles

3.1.4 Design notes

Surface finish

When runner blocks and guide rails are bolted to the adjoining structures, the stressing of the screw-fasteners results in forces which, either alone or in combination with external loads, can cause plastic deformations in the mating surfaces and reference edges and planes (1, 3, 4, 6). To avoid plastic deformation at these points, the surfaces must have a high percentage contact area, i.e. the surface finish must be of appropriately high quality. This is the only way to avoid settling phenomena when the linear guide is in service. The required surface finish for all mating and reference surfaces is specified as a roughness value R_a in μm .



Reference and mating surfaces

- 1 Base of the runner block
- 2 Head of the runner block
- 3 Reference edge for the runner block
- 4 Two reference edges for the guide rail
- 5 Head of the guide rail
- 6 Base of the guide rail

Roughness value for reference and mating surfaces

A roughness average of R_a 0.4 to 2 μm is recommended for the reference and mating surfaces. The recommended roughness values are determined according to the required accuracy and the accuracy class of the profiled rail system.

Accuracy class	Maximum roughness R_a μm
E	2.0
N	1.6
H	0.8
P	0.4
XP	0.4
SP	0.4
UP	0.4

3 Profiled rail systems

3.1 Principles

3.1.4 Design notes

3.1.4.2 Guidelines for economical designs

As accurate as necessary, as inaccurate as possible

When using profiled rail systems it is essential to consider and define all requirements in order to determine the required system characteristics. The maxim here is: as accurate as necessary, as inaccurate as possible.

High accuracy classes only make sense if the adjoining structures have the same high accuracy level. Selecting an appropriate accuracy class right from the beginning can considerably reduce costs when it comes to manufacturing the mounting base and attachments.

Preload only as high as necessary

The higher the preload is, the higher the accuracy requirements on the adjoining structures will be. This should be remembered when choosing the preload class. For high preloads, the permissible vertical offset of the mounting base in the transverse direction is small.

The same applies to parallelism tolerances between two guide rails. High accuracy requirements on the mounting base and the attachments always drive up the manufacturing costs.

Ensure a sufficiently large distance between rails

The smaller the distance between rails installed parallel to each other, the more accurately the rails have to be aligned vertically.

The distance between the guide rails should therefore be appropriately sized.

Short or "Super" runner blocks for larger inaccuracies

If the attachment has low accuracy, short runner blocks or even self-aligning "Super" runner blocks can be used.

Within certain limits, these components can compensate for straightness errors in a guide rail.

Ensure sufficient space for linear guides

The adjoining structures should always be designed to allow optimal installation and subsequent maintenance. The space provided for profiled rail systems should therefore be

appropriately dimensioned. The design should also make allowance for easy replacement of the linear guides.

3 Profiled rail systems

3.1 Principles

3.1.4 Design notes

Use of standard components and preferred lengths

The use of standard components and preferred lengths reduces the cost of purchase. It also speeds up delivery and reduces spare parts complexity.

Same linear guides in all axes

When using linear guides, the same type of profiled rail system should be used per slide or, where possible, per machine or series.

This also reduces the cost of purchase, warehousing, and installation.

Long-term lubrication for cost-efficient operation

When selecting the lubrication system, it is important to consider not only the cost of connecting the system up to the runner block, but also the cost of the additional equipment required.

The use of front lube units or the standard long-term lubrication offered by Rexroth profiled rail systems is therefore recommended.

Eliminating a reference edge by pinning runner blocks

Depending on the application, it may be possible to eliminate a reference edge by pinning the runner blocks.

Design one guide rail as the main rail

In linear guides with two or more rails, one rail should be designed as the main rail. This rail should be fixed against a reference edge. All the other rails can then be aligned relative to the main rail during installation.

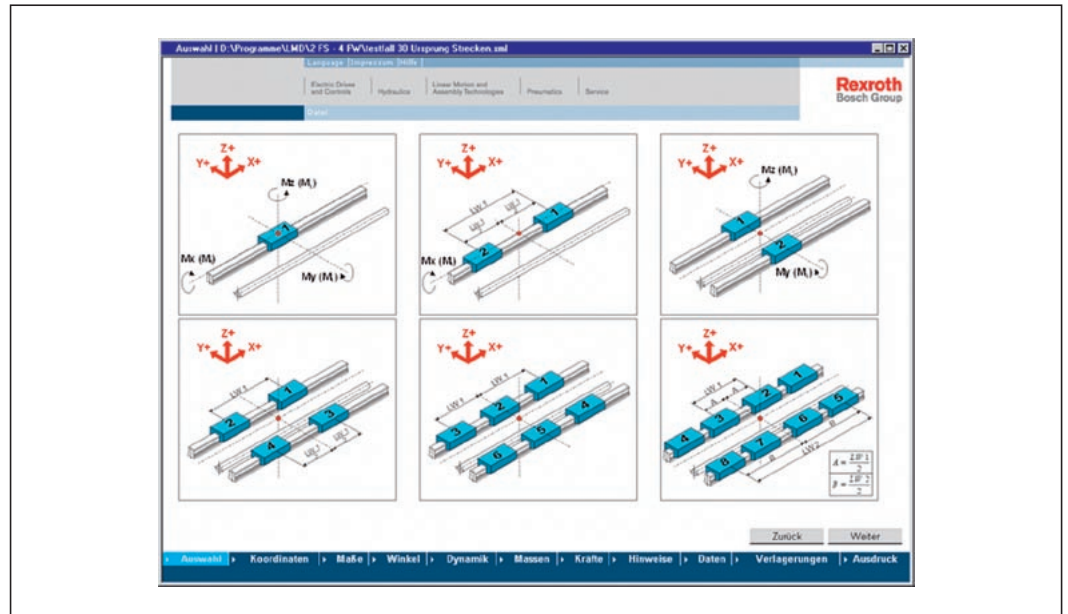
3 Profiled rail systems

3.1 Principles

3.1.5 Calculations

Calculation service

Rexroth provides a design calculation service to assist with nominal life calculations. The travel life expectancy is calculated using a software program called LINEAR MOTION DESIGNER. All the customer has to do is to specify the operating conditions.



Layout selection page from the LINEAR MOTION DESIGNER calculation software

Besides using the LINEAR MOTION DESIGNER program, the nominal life can, of course, be calculated by conventional methods. The manual procedure is described in detail below.

3.1.5.1 Procedure for manual calculations

The nominal life calculation consists of several steps. The calculation principles for determining the nominal life of profiled rail systems will be described first.

The actual calculation of the nominal life in terms of travel or operating hours requires precise determination of the loads acting on the bearings. Finally, the static and dynamic load safety factors have to be checked. The recommended procedure is shown below.

Summary of the procedure:

- Define the operating conditions.
- Calculate the loads due to forces and moments.
- Calculate the combined equivalent load on the bearing.
- Take the preload into account.
- Calculate the equivalent dynamic load on the bearing.
- Calculate the life expectancy.
- Calculate the equivalent static load on the bearing.
- Calculate the static load safety factor.

3 Profiled rail systems

3.1 Principles

3.1.5 Calculations

Detailed procedure, using a linear guide with two rails and four runner blocks as an example:

Input Data	Calculation step	Output Result
<p>Machine design (structure, application)</p> <p>$L_{Wj}, L_{Sj}, L_{yj}, L_{zj}$ $F_{wx,j}, F_{wy,j}, F_{wz,j}$ (F_g, F_p, F_a) $x_{w,j}, y_{w,j}, z_{w,j}$ $(x_S, y_S, z_S, x_p, y_p, z_p)$</p>	<p>Step 1: Define the operating conditions Data on the guide system, layout, dynamic cycle and loads</p> <p style="text-align: center;">↓</p>	<p>$L_{Wj}, L_{Sj}, L_{yj}, L_{zj}, \alpha, \beta,$ $m, x_S, y_S, z_S, F_g, F_a$ $F_p, x_p, y_p, z_p, F_{w,j},$ $F_{wx,j}, F_{wy,j}, F_{wz,j}, x_{w,j},$ $y_{w,j}, z_{w,j}, n, t_n, S_n, v_n,$ $a_n, q_{sn}, q_{tn}, C, C_0,$ $X_{prt}, M_t, M_{t0}, M_L, M_{L0}$</p>
<p>F_{yni}, F_{zni}</p>	<p>Step 2: Calculate the loads due to forces and moments in every phase n for each runner block i in the y- and z-directions</p> <p style="text-align: center;">↓</p>	<p>F_{yni}, F_{zni}</p>
<p>F_{yni}, F_{zni}</p>	<p>Step 3: Calculate the combined equivalent load on the bearing in every phase n and for each runner block i</p> <p style="text-align: center;">↓</p>	<p>$F_{combn i}$</p>
<p>$C, X_{prt}, F_{combn i}$</p>	<p>Step 4: Take the preload into account using the effective equivalent load on the bearing in every phase n for each runner block i</p> <p style="text-align: center;">↓</p>	<p>$F_{prt}, F_{lim}, F_{effni}$</p>
<p>q_{sn}, F_{effni}</p>	<p>Step 5: Calculate the equivalent dynamic load on the bearing for varying loads for each runner block i</p> <p style="text-align: center;">↓</p>	<p>F_{mi}</p>
<p>$C, F_{mi}, S_{stroke}, n_{stroke},$ q_{tn}, v_n, a_1</p>	<p>Step 6: Calculate the nominal or modified life expectancy at constant or varying speed for each runner block i</p> <p style="text-align: center;">↓</p>	<p>v_m L_i, L_{hi}, L_{nai}</p>
<p>F_{0yi}, F_{0zi} $M_{0xi}, M_{0yi}, M_{0zi}$ C_0, M_{t0}, M_{L0}</p>	<p>Step 7: Calculate the equivalent static load on the bearing for combined loads for each runner block i</p> <p style="text-align: center;">↓</p>	<p>$F_{0 comb i}$</p>
<p>C_0 $F_{0 comb i}, F_{max effni}$</p>	<p>Step 8: Calculate the static load safety factor</p>	<p>S_0</p>

3 Profiled rail systems

3.1 Principles

3.1.5 Calculations

3.1.5.2 Define the operating conditions

The following parameters are deciding factors in the nominal life calculation:

Guide system

The characteristic values of the chosen linear guide are required as input data. These are the load capacities, load moments and the preload for the specific profiled rail type, design and size.

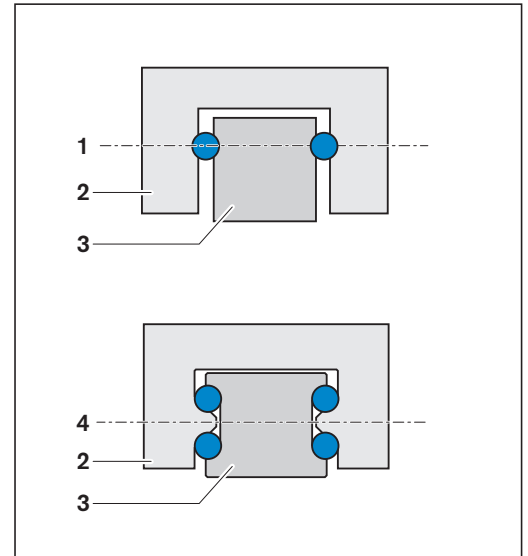
Profiled rail system details		
Parameter		Unit
Dynamic load capacity	C	N
Static load capacity	C_0	N
Dynamic torsional moment load capacity	M_t	Nm
Static torsional moment load capacity	M_{t0}	Nm
Dyn. longitudinal moment load capacity	M_L	Nm
Static longitudinal moment load capacity	M_{L0}	Nm
Preload of the guide	X_{pr}	–

Layout

First, the coordinate system for the selected layout has to be defined. In principle, any coordinate system can be chosen. However, the centroid offset must be included in the calculations. To simplify matters, it is advisable to define the origin of the coordinate system as being the centroid between the runner blocks in the x-direction, the centroid between the guide rails in the y-direction, and the runner block raceway centerline in the z-direction. The raceway centerline is defined differently for 2-row and 4-row profiled rail systems:

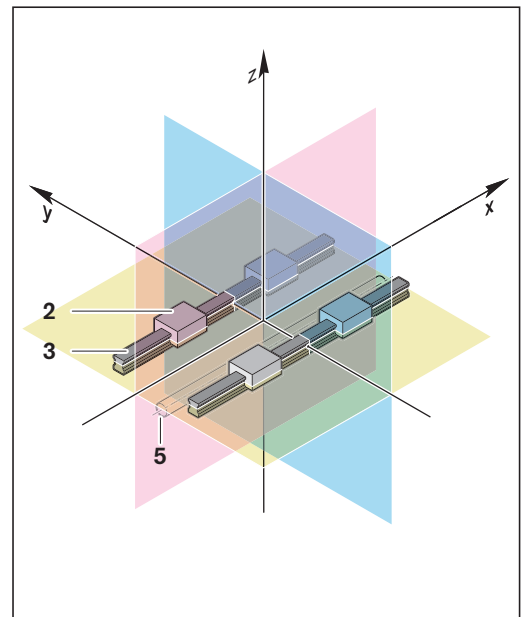
- In 2-row systems the raceway centerline runs through the centers of the rolling element rows (centerline between the contact points on the gothic arch profile of the raceways).
- In 4-row systems the raceway centerline lies between the upper two and lower two raceways (or rows of rolling elements).

All of the following calculation formulas relate to the coordinate system as defined here.



Definition of the raceway centerline for 2-row and 4-row profiled rail systems

- 1 Raceway centerline for a 2-row profiled rail system
- 2 Runner block
- 3 Guide rail
- 4 Raceway centerline for a 4-row profiled rail system
- 5 Drive unit (e.g. ball screw)

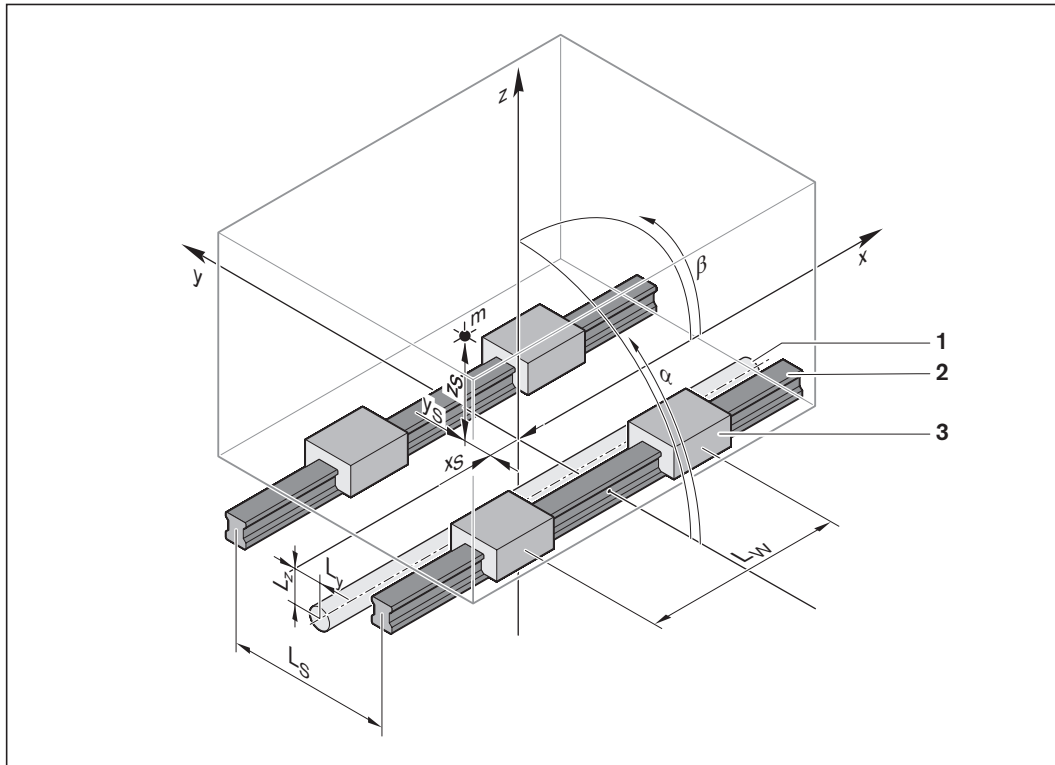


Location of the coordinate system for a linear guide with 2 rails and 4 runner blocks

3 Profiled rail systems

3.1 Principles

3.1.5 Calculations



Layout of a linear guide with 2 rails and 4 runner blocks

- 1 Drive unit (e.g. ball screw)
- 2 Guide rail
- 3 Runner block

The layout of the guide system is described by the centroid distances of the guide rails L_S and the runner blocks L_W .

All forces acting on the system along the x-direction must be applied via the drive unit, e.g. ball screw. Consequently, the location of the drive unit relative to the origin of the coordinate system is included in the calculation as the dimensions L_y and L_z .

If the system is not installed horizontally, the orientation must be specified using the angles α and β . The mass of the carriage assembly and the location of its center of gravity are also required.

Layout details		
Parameter		Unit
Distance between runner blocks	L_W	mm
Distance between guide rails	L_S	mm
Location of drive unit in the y-direction	L_y	mm
Location of drive unit in the z-direction	L_z	mm
Angular location of the guide relative to the X-axis	α	°
Angular location of the guide relative to the Y-axis	β	°
Location of center of gravity in the x-direction	x_S	mm
Location of center of gravity in the y-direction	y_S	mm
Location of center of gravity in the z-direction	z_S	mm
Mass	m	kg

3 Profiled rail systems

3.1 Principles

3.1.5 Calculations

Dynamic cycle

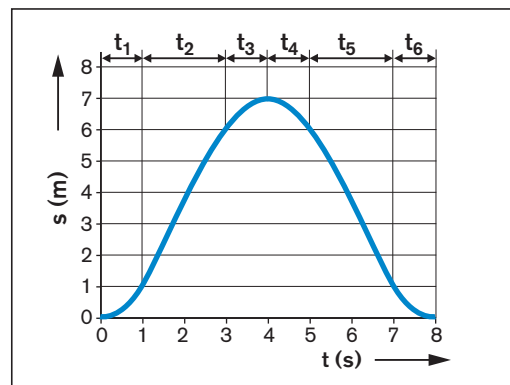
A time-dependent dynamic cycle must be defined for the nominal life calculation. This cycle should be a typical operating cycle as required by the machine user.

The cycle comprises several phases with different travel distances, speeds and accelerations, describing the different steps such as approach, stop, processing and rapid traverse.

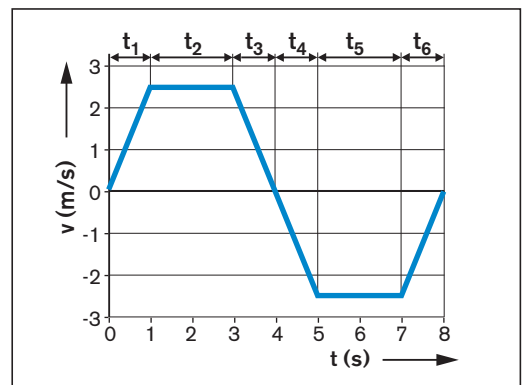
Dynamic cycle details		
Parameter		Unit
Acceleration in phase n	$a_1 \dots a_n$	m/s^2
Travel in phase n	$s_1 \dots s_n$	m
Time in phase n	$t_1 \dots t_n$	s
Speed in phase n	$v_1 \dots v_n$	m/s
Phase	n	-

Example of a dynamic cycle:

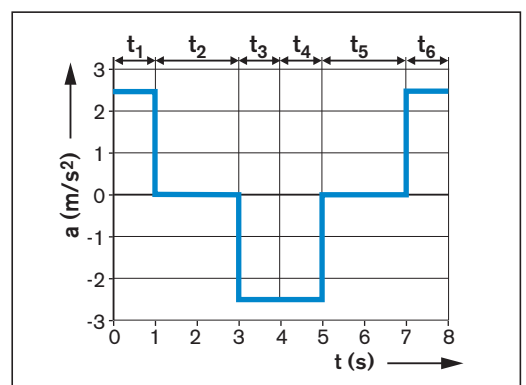
Phase n	Time	Direction of motion	Description
1	0 s to 1 s	Forward	Acceleration
2	1 s to 3 s	Forward	Processing
3	3 s to 4 s	Forward	Deceleration
4	4 s to 5 s	Backward	Acceleration
5	5 s to 7 s	Backward	Return stroke
6	7 s to 8 s	Backward	Deceleration



Travel



Speed



Acceleration

Note: The signs of the parameters travel s , speed v and acceleration a relate to the positive and negative directions of the axes in the chosen coordinate system. A negative value for acceleration does not therefore necessarily mean deceleration, but can mean acceleration in the negative axis direction.

3 Profiled rail systems

3.1 Principles

3.1.5 Calculations

Loads

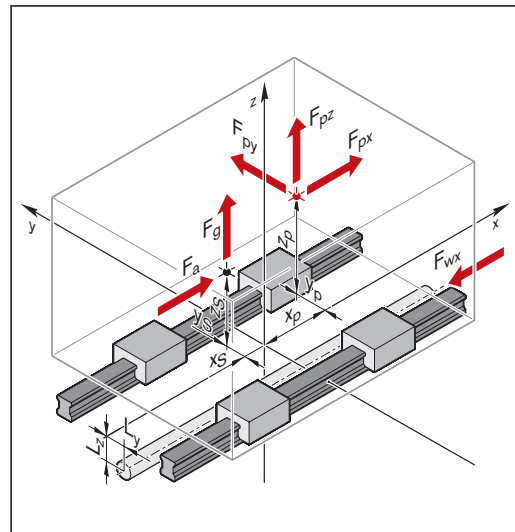
The forces F_w acting on the system are described by their value and direction. They are specified according to their direction, i.e. F_{wx} , F_{wy} and F_{wz} . The force application points are described by the coordinates x_w , y_w and z_w . A load case j is assigned to each force acting in the dynamic cycle. There may be several load cases acting simultaneously within any one phase of the dynamic cycle.

The forces F_w acting on the system result from the weight forces F_g , the acceleration forces F_a , and the process forces F_p . The forces F_g and F_a act at the center of gravity x_s, y_s, z_s and the force F_p at the force application point x_p, y_p, z_p . The different force application points and the force directions must be included in the calculation.

All forces acting on the system in the x-direction must be applied via the drive unit (e.g. ball screw). The drive force is therefore equal in value to F_{wx} , but acts in the opposite direction. It also represents a load on the linear guide.

A load case j is assigned to each load except the drive force. The drive force is not a separate load case because it is the counterforce to the force in the x-direction. It is included in the calculation with the appropriate sign and the dimensions L_y and L_z .

Parameter		Unit
Forces acting in load case j	$F_{wx,j}$	N
	$F_{wy,j}$	
	$F_{wz,j}$	
Coordinates of the force application point in load case j	$x_{w,j}$	mm
	$y_{w,j}$	
	$z_{w,j}$	



Acceleration force F_a , weight force F_g , process forces F_p , drive force F_{wx} , and locations of the force application points

	Effective force	Coordinates of the force application point	Description
Weight forces	$F_{w,j} = F_g$ $F_g = m \cdot g$	$x_{w,j} = x_s$ $y_{w,j} = y_s$ $z_{w,j} = z_s$	The weight forces F_g are calculated from the masses; they act on the associated centers of gravity with the coordinates x_s, y_s and z_s . The masses (and therefore the weight forces) can vary from phase to phase.
Acceleration forces	$F_{w,j} = F_a$ $F_a = -m \cdot a$	$x_{w,j} = x_s$ $y_{w,j} = y_s$ $z_{w,j} = z_s$	The acceleration forces F_a are calculated from the accelerated or decelerated masses m and the specified accelerations a from the dynamic cycle. The forces act at the centers of gravity with the coordinates x_s, y_s, z_s and counter to the direction of acceleration.
Process forces	$F_{w,j} = F_p$	$x_{w,j} = x_p$ $y_{w,j} = y_p$ $z_{w,j} = z_p$	The process forces F_p are calculated from the specific processing operation in the respective phase of the dynamic cycle. These may be, for instance, forces arising during molding/extrusion, forming, machining, etc. The force application points are described by the coordinates x_p, y_p and z_p .

3 Profiled rail systems

3.1 Principles

3.1.5 Calculations

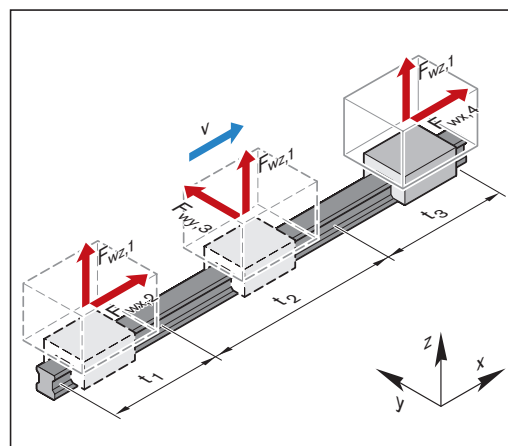
Load details		
Parameter		Unit
Forces acting in load case j	$F_{wx,j}, F_{wy,j}, F_{wz,j}$	N
Weight force	F_g	N
Acceleration force	F_a	N
Process force	F_p	N
Application point of the effective force in load case j	$x_{w,j}, y_{w,j}, z_{w,j}$	mm
Center of gravity	x_s, y_s, z_s	mm

Parameter		Unit
Application point of the process force	x_p, y_p, z_p	mm
Acceleration	a	m/s ²
Acceleration due to gravity (g = 9.81 m/s ²)	g	m/s ²
Mass	m	kg
Load case	j	-
Phase	n	-

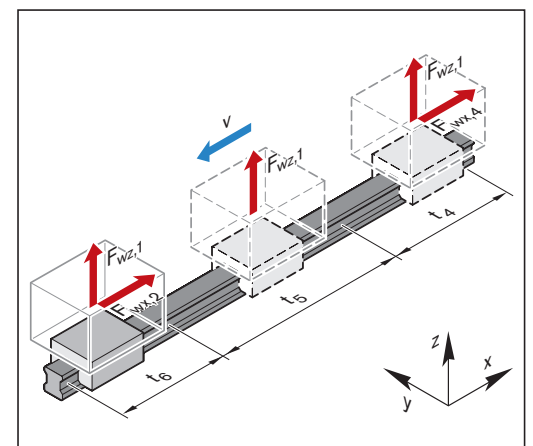
Examples of varying loads:

In the dynamic cycle example introduced above the same weight force F_g acts in all phases. It is calculated from the mass $m = 40$ kg. The forces F_a arise during acceleration and deceleration. In phases 1 and 6 and in phases 3 and 4 the acceleration forces are identical. The forward

stroke is executed in the positive x-direction, and the return stroke in the negative x-direction. During the machining process the force F_p acts in the positive y-direction. This results in 4 load cases for the cycle.



Forward stroke



Return stroke

Load case j	Description	$F_{w,j}$	$x_{w,j}, y_{w,j}, z_{w,j}$	Phase
1	Weight force	$F_{wz,1} = F_g$ $F_{wz,1} = m \cdot g$ $F_{wz,1} = 40 \text{ kg} \cdot (-9.81 \text{ m/s}^2)$ $F_{wz,1} \approx -400 \text{ N}$	$x_{w,1} = x_S = 0 \text{ mm}$ $y_{w,1} = y_S = 100 \text{ mm}$ $z_{w,1} = z_S = 50 \text{ mm}$	1; 2; 3; 4; 5; 6
2	Acceleration force of $a_{1/6} = 2.5 \text{ m/s}^2$	$F_{wx,2} = F_{a1/6}$ $F_{wx,2} = -m \cdot a_{1/6}$ $F_{wx,2} = -40 \text{ kg} \cdot 2.5 \text{ m/s}^2$ $F_{wx,2} = -100 \text{ N}$	$x_{w,2} = x_S = 0 \text{ mm}$ $y_{w,2} = y_S = 100 \text{ mm}$ $z_{w,2} = z_S = 50 \text{ mm}$	1; 6
3	Process force during machining	$F_{wy,3} = F_p = 500 \text{ N}$	$x_{w,3} = x_p = 100 \text{ mm}$ $y_{w,3} = y_p = 200 \text{ mm}$ $z_{w,3} = z_p = 150 \text{ mm}$	2
4	Acceleration force of $a_{3/4} = -2.5 \text{ m/s}^2$	$F_{wx,4} = F_{a3/4}$ $F_{wx,4} = -m \cdot a_{3/4}$ $F_{wx,4} = -40 \text{ kg} \cdot (-2.5 \text{ m/s}^2)$ $F_{wx,4} = +100 \text{ N}$	$x_{w,4} = x_S = 0 \text{ mm}$ $y_{w,4} = y_S = 100 \text{ mm}$ $z_{w,4} = z_S = 50 \text{ mm}$	3; 4

3 Profiled rail systems

3.1 Principles

3.1.5 Calculations

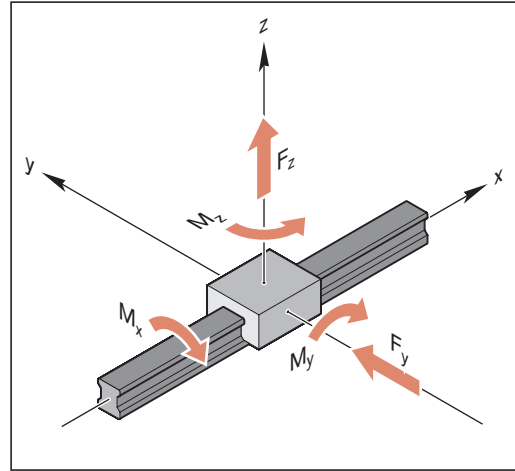
3.1.5.3 Loads due to forces and moments

Forces and moments acting on the runner block

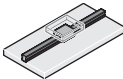
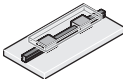
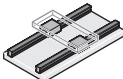
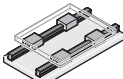
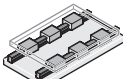
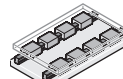
The forces acting on the system are distributed among the runner blocks according to the layout of the system. The loads due to forces and moments resulting from the forces acting on the system have to be calculated for each runner block when performing the nominal life calculation.

All load calculations assume an infinitely rigid mounting base and an infinitely rigid attachment.

The following table shows the layouts that are most commonly used in practice along with the runner block loads that have to be calculated in each case.



Load on a single runner block due to forces and moments

Scenario	Layout	Forces		Moments		
		in z-direction Lift-off/ down force	in y-direction Side load	about the X-axis Torsional moment	about the Y-axis Longitudi- nal moment	about the Z-axis Longitudi- nal moment
1	1 rail 1 runner block 	F_z	F_y	M_x	M_y	M_z
2	1 rail 2 runner blocks 	F_z	F_y	M_x	-	-
3	2 rails 2 runner blocks 	F_z	F_y	-	M_y	M_z
4	2 rails 4 runner blocks 	F_z	F_y	-	-	-
5	2 rails 6 runner blocks 	F_z	F_y	-	-	-
6	2 rails 8 runner blocks 	F_z	F_y	-	-	-

3 Profiled rail systems

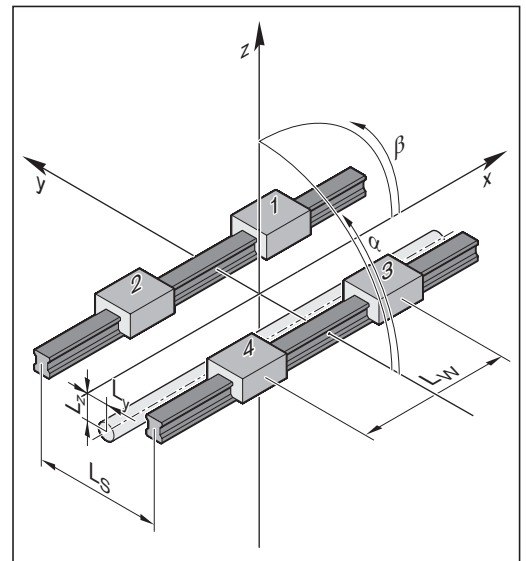
3.1 Principles

3.1.5 Calculations

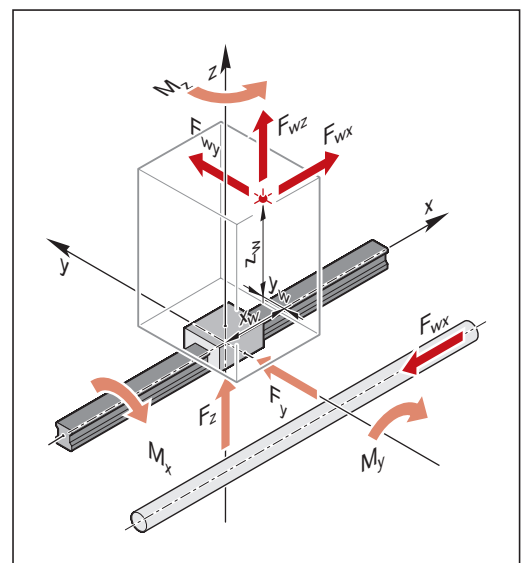
Symbols used in formulas

The calculation formulas for determining the runner block loads are shown on the following pages for the various rail/runner block combinations. The symbols used in these formulas are explained below:

- $F_{wx,j}$ = force acting in the x-direction of load case j (N)
- $F_{wy,j}$ = force acting in the y-direction of load case j (N)
- $F_{wz,j}$ = force acting in the z-direction of load case j (N)
- F_{zni} = force in z-direction on runner block i in phase n (N)
- F_{yni} = force in y-direction on runner block i in phase n (N)
- M_{xni} = moment about the X-axis on runner block i in phase n (Nmm)
- M_{yni} = moment about the Y-axis on runner block i in phase n (Nmm)
- M_{zni} = moment about the Z-axis on runner block i in phase n (Nmm)
- $x_{w,j}$ = x-coordinate of the application point of the effective force (mm)
- $y_{w,j}$ = y-coordinate of the application point of the effective force (mm)
- $z_{w,j}$ = z-coordinate of the application point of the effective force (mm)
- L_S = distance between rails (mm)
- L_W = distance between runner blocks (mm)
- L_y = y-coordinate of the drive unit (mm)
- L_z = z-coordinate of the drive unit (mm)
- α = angular location of the system relative to the X-axis (°)
- β = angular location of the system relative to the Y-axis (°)
- n = phase (-)
- i = runner block (-)
- j = load case (-)
- k = number or final term of the load cases (-)



Geometry of a system with 2 guide rails and 4 runner blocks



Effective forces with force application points and loads due to forces and moments on the runner block in a system with 1 guide rail and 1 runner block

- Forces acting on the system
- Loads arising at the runner block (forces and moments)

3 Profiled rail systems

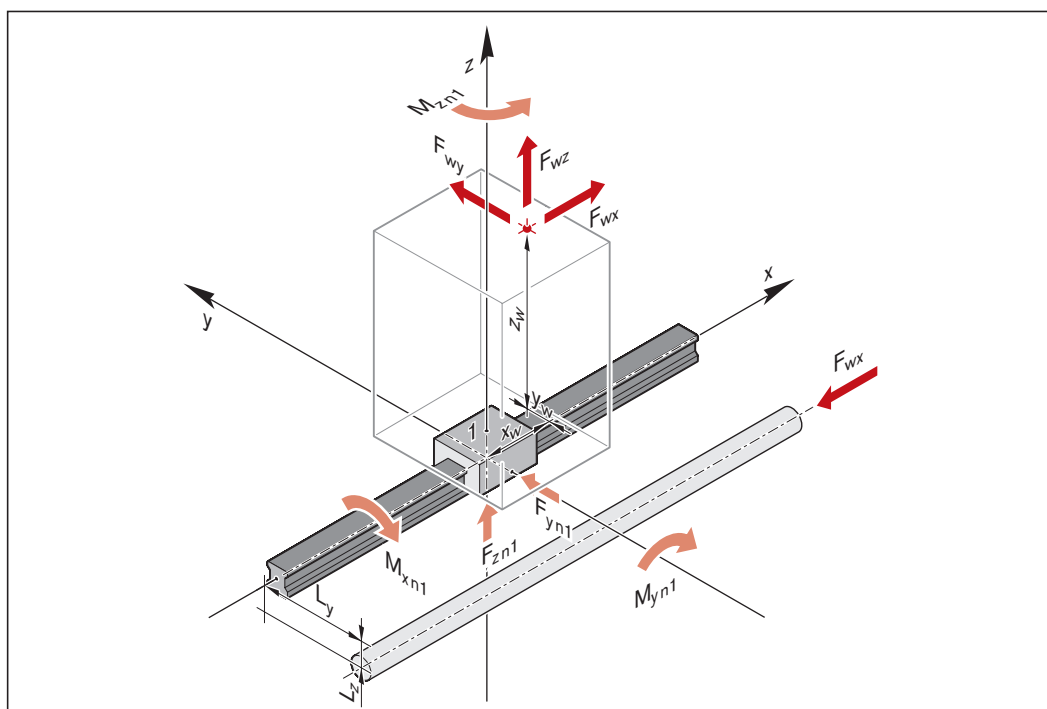
3.1 Principles

3.1.5 Calculations

Application

1 rail

1 runner block



Loads on a system with 1 guide rail and 1 runner block

Runner block i	Load	Formula
1	Force in z-direction	(3-10) $F_{z n1} = \sum_{j=1}^k F_{wz,j}$
1	Force in y-direction	(3-11) $F_{y n1} = \sum_{j=1}^k F_{wy,j}$
1	Moment about X-axis	(3-12) $M_{x n1} = \sum_{j=1}^k (F_{wy,j} \cdot z_{w,j}) - \sum_{j=1}^k (F_{wz,j} \cdot y_{w,j})$
1	Moment about Y-axis	(3-13) $M_{y n1} = \sum_{j=1}^k (F_{wx,j} \cdot (z_{w,j} - L_z)) - \sum_{j=1}^k (F_{wz,j} \cdot x_{w,j})$
1	Moment about Z-axis	(3-14) $M_{z n1} = - \sum_{j=1}^k (F_{wx,j} \cdot (y_{w,j} - L_y)) + \sum_{j=1}^k (F_{wy,j} \cdot x_{w,j})$

3 Profiled rail systems

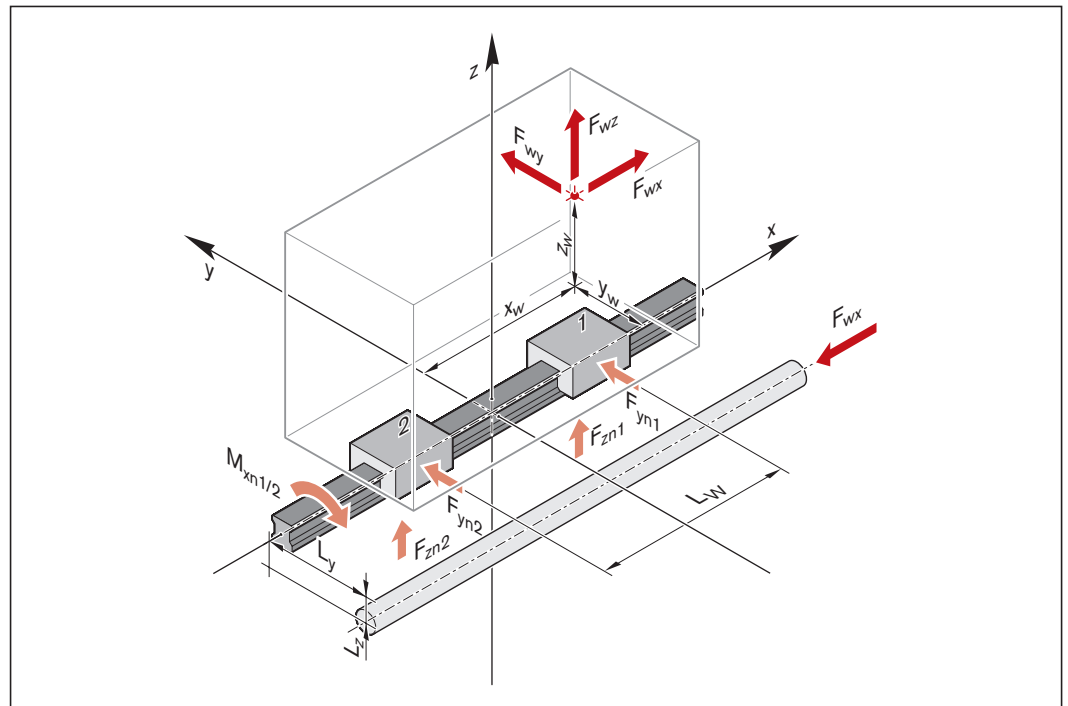
3.1 Principles

3.1.5 Calculations

Application

1 rail

2 runner blocks



Loads on a system with 1 guide rail and 2 runner blocks

Runner block i	Load	Formula
1	Force in z-direction	$(3-15) \quad F_{zn1} = \frac{\sum_{j=1}^k F_{wz,j}}{2} - \frac{\sum_{j=1}^k (F_{wx,j} \cdot (z_{w,j} - L_z)) - \sum_{j=1}^k (F_{wz,j} \cdot x_{w,j})}{L_W}$
2	Force in z-direction	$(3-16) \quad F_{zn2} = \frac{\sum_{j=1}^k F_{wz,j}}{2} + \frac{\sum_{j=1}^k (F_{wx,j} \cdot (z_{w,j} - L_z)) - \sum_{j=1}^k (F_{wz,j} \cdot x_{w,j})}{L_W}$
1	Force in y-direction	$(3-17) \quad F_{yn1} = \frac{\sum_{j=1}^k F_{wy,j}}{2} - \frac{\sum_{j=1}^k (F_{wx,j} \cdot (y_{w,j} - L_y)) - \sum_{j=1}^k (F_{wy,j} \cdot x_{w,j})}{L_W}$
2	Force in y-direction	$(3-18) \quad F_{yn2} = \frac{\sum_{j=1}^k F_{wy,j}}{2} + \frac{\sum_{j=1}^k (F_{wx,j} \cdot (y_{w,j} - L_y)) - \sum_{j=1}^k (F_{wy,j} \cdot x_{w,j})}{L_W}$
1/2	Moment about X-axis	$(3-19) \quad M_{xn1} = M_{xn2} = \frac{\sum_{j=1}^k (F_{wy,j} \cdot z_{w,j}) - \sum_{j=1}^k (F_{wz,j} \cdot y_{w,j})}{2}$

3 Profiled rail systems

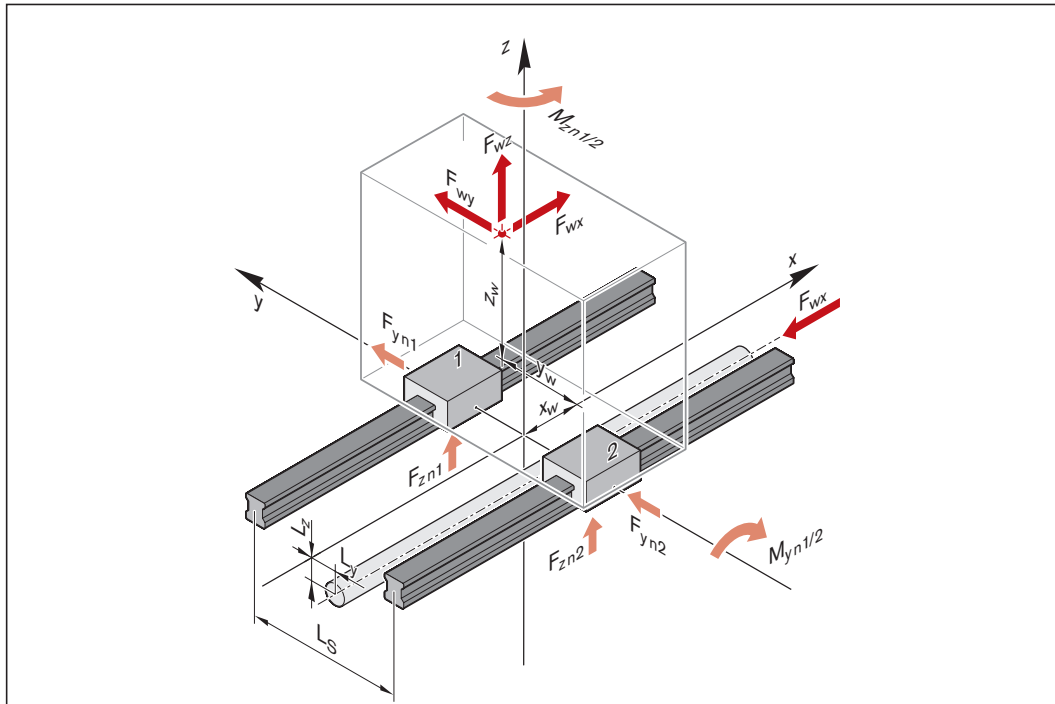
3.1 Principles

3.1.5 Calculations

Application

2 rails

2 runner blocks



Loads on a system with 2 guide rails and 2 runner blocks

Runner block i	Load	Formula
1	Force in z-direction	(3-20) $F_{z n 1} = \frac{\sum_{j=1}^k F_{w z, j}}{2} - \frac{\sum_{j=1}^k (F_{w y, j} \cdot z_{w, j}) - \sum_{j=1}^k (F_{w z, j} \cdot y_{w, j})}{L_s}$
2	Force in z-direction	(3-21) $F_{z n 2} = \frac{\sum_{j=1}^k F_{w z, j}}{2} + \frac{\sum_{j=1}^k (F_{w y, j} \cdot z_{w, j}) - \sum_{j=1}^k (F_{w z, j} \cdot y_{w, j})}{L_s}$
1/2	Force in y-direction	(3-22) $F_{y n 1} = F_{y n 2} = \frac{\sum_{j=1}^k F_{w y, j}}{2}$
1/2	Moment about Y-axis	(3-23) $M_{y n 1} = M_{y n 2} = \frac{\sum_{j=1}^k (F_{w x, j} \cdot (z_{w, j} - L_z)) - \sum_{j=1}^k (F_{w z, j} \cdot x_{w, j})}{2}$
1/2	Moment about Z-axis	(3-24) $M_{z n 1} = M_{z n 2} = \frac{-\sum_{j=1}^k (F_{w x, j} \cdot (y_{w, j} - L_y)) + \sum_{j=1}^k (F_{w y, j} \cdot y_{w, j})}{2}$

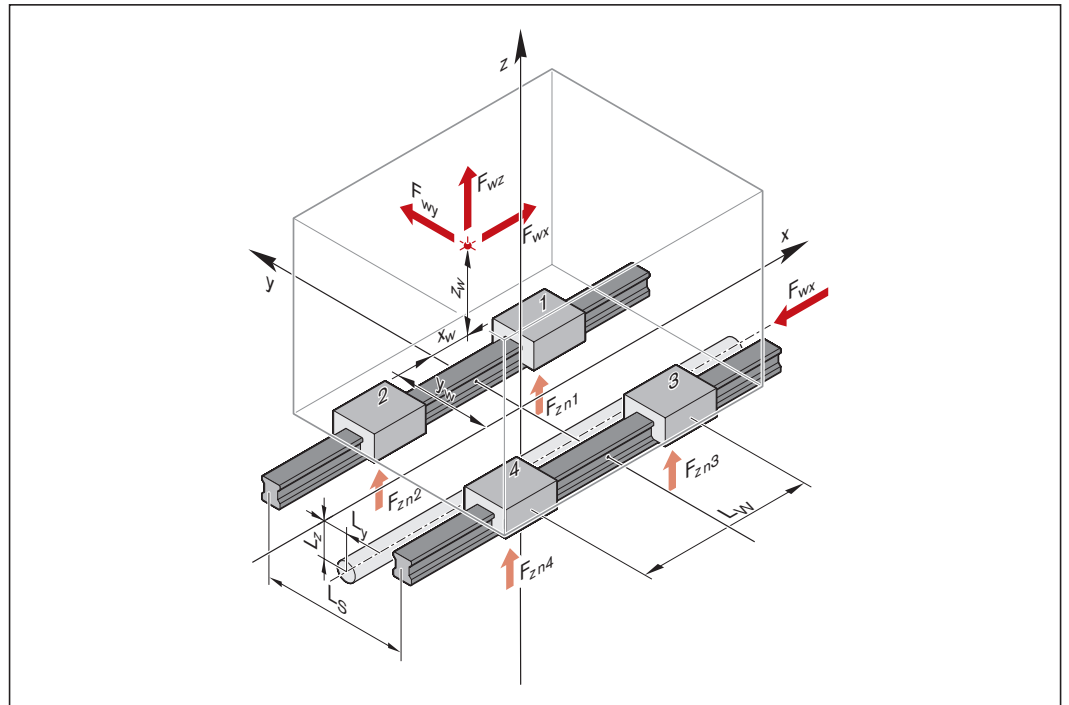
3 Profiled rail systems

3.1 Principles

3.1.5 Calculations

Application**2 rails****4 runner blocks**

Applications with 2 rails and 4 runner blocks are the most common of all layouts. This layout will therefore be described in more detail.



Loads on a system with 2 guide rails and 4 runner blocks

Loads in z-direction

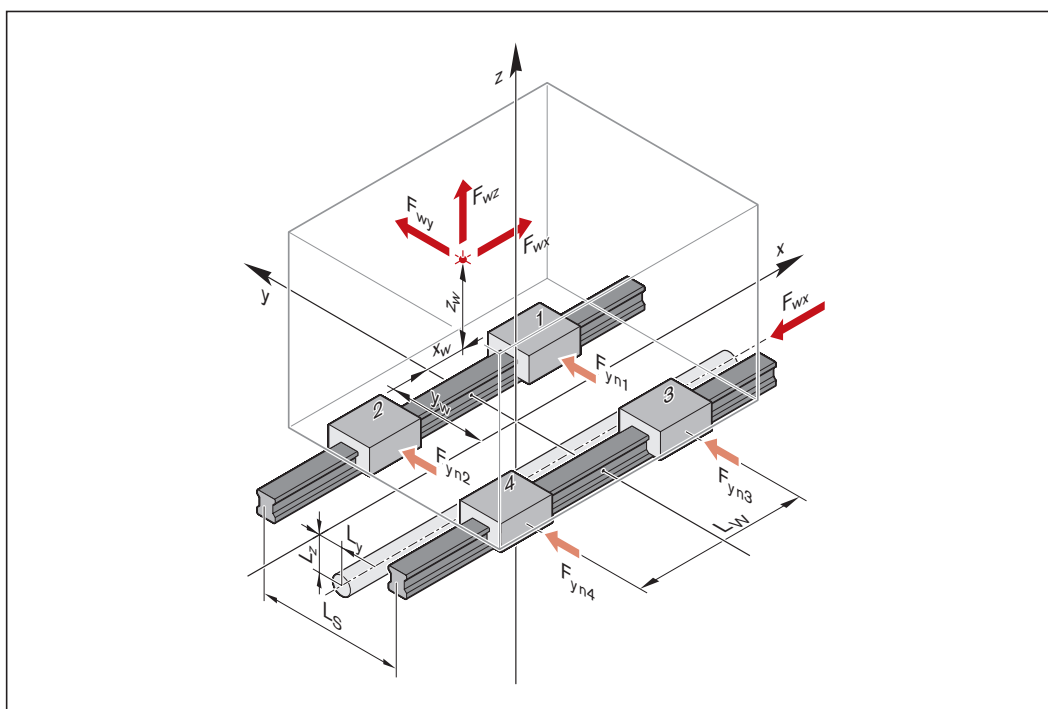
The external loads F_{zn_i} acting on the runner blocks i (1 to 4) in the z -direction in a phase n are calculated using the following formulas.

i	Formula
1	$(3-25) \quad F_{zn1} = \frac{\sum_{j=1}^k F_{wz,j}}{4} + \frac{\sum_{j=1}^k (F_{wz,j} \cdot y_{w,j}) - \sum_{j=1}^k (F_{wy,j} \cdot z_{w,j})}{2 \cdot L_s} + \frac{\sum_{j=1}^k (F_{wz,j} \cdot x_{w,j}) - \sum_{j=1}^k (F_{wx,j} \cdot (z_{w,j} - L_z))}{2 \cdot L_w}$
2	$(3-26) \quad F_{zn2} = \frac{\sum_{j=1}^k F_{wz,j}}{4} + \frac{\sum_{j=1}^k (F_{wz,j} \cdot y_{w,j}) - \sum_{j=1}^k (F_{wy,j} \cdot z_{w,j})}{2 \cdot L_s} + \frac{\sum_{j=1}^k (F_{wx,j} \cdot (z_{w,j} - L_z)) - \sum_{j=1}^k (F_{wz,j} \cdot x_{w,j})}{2 \cdot L_w}$
3	$(3-27) \quad F_{zn3} = \frac{\sum_{j=1}^k F_{wz,j}}{4} + \frac{\sum_{j=1}^k (F_{wy,j} \cdot z_{w,j}) - \sum_{j=1}^k (F_{wz,j} \cdot y_{w,j})}{2 \cdot L_s} + \frac{\sum_{j=1}^k (F_{wz,j} \cdot x_{w,j}) - \sum_{j=1}^k (F_{wx,j} \cdot (z_{w,j} - L_z))}{2 \cdot L_w}$
4	$(3-28) \quad F_{zn4} = \frac{\sum_{j=1}^k F_{wz,j}}{4} + \frac{\sum_{j=1}^k (F_{wy,j} \cdot z_{w,j}) - \sum_{j=1}^k (F_{wz,j} \cdot y_{w,j})}{2 \cdot L_s} + \frac{\sum_{j=1}^k (F_{wx,j} \cdot (z_{w,j} - L_z)) - \sum_{j=1}^k (F_{wz,j} \cdot x_{w,j})}{2 \cdot L_w}$

3 Profiled rail systems

3.1 Principles

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Loads on a system with 2 guide rails and 4 runner blocks

Loads in y-direction

The external loads $F_{y_{ni}}$ acting on the runner blocks i (1 to 4) in the y -direction in a phase n are calculated using the following formulas.

i	Formula
1 3	$(3-29) \quad F_{y_{n1}} = F_{y_{n3}} = \frac{\sum_{j=1}^k F_{wy,j}}{4} + \frac{\sum_{j=1}^k (F_{wy,j} \cdot x_{w,j}) - \sum_{j=1}^k (F_{wx,j} \cdot (y_{w,j} - L_y))}{2 \cdot L_w}$
2 4	$(3-30) \quad F_{y_{n2}} = F_{y_{n4}} = \frac{\sum_{j=1}^k F_{wy,j}}{4} - \frac{\sum_{j=1}^k (F_{wy,j} \cdot x_{w,j}) - \sum_{j=1}^k (F_{wx,j} \cdot (y_{w,j} - L_y))}{2 \cdot L_w}$

3 Profiled rail systems

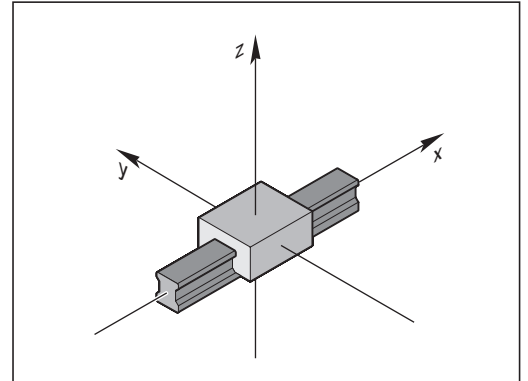
3.1 Principles

3.1.5 Calculations

3.1.5.4 Combined equivalent load on bearing

The loads calculated for the individual runner blocks (forces F_x , F_y , F_z and moments M_x , M_y , M_z) are combined into comparative loads for each runner block. These comparative loads are called the combined equivalent loads on the bearing.

For the calculation of loads on runner blocks, a distinction is made between horizontal loads (y-direction) and vertical loads (z-direction), similar to the distinction between radial and axial forces in rotary anti-friction bearings. For loads acting in a direction other than the main directions described above, the forces must be resolved. Runner blocks can take up moments as well as forces. If several runner blocks are mounted one behind the other at close intervals on a guide rail, this must be taken into account when calculating the load on the bearing.

Main directions of loading

Definition of main load directions

The load cases are therefore as follows:

- Vertical and horizontal forces
- Vertical and horizontal forces combined with moments
- Consideration of closely spaced runner blocks (using contact factor f_C)

Vertical and horizontal forces

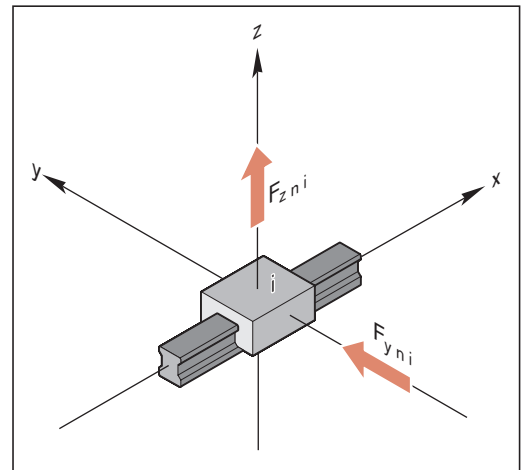
For external loads acting on a runner block i in phase n due to vertical forces F_z and horizontal forces F_y , a comparative load is required for the nominal life calculation. This combined equivalent load on the bearing is the sum of the absolute values of the forces F_z and F_y . The structure of the profiled rail systems allows this simplified calculation.

$$(3-31) \quad F_{\text{comb}ni} = |F_{yni}| + |F_{zni}|$$

$F_{\text{comb}ni}$ = combined equivalent load on bearing for runner block i (N)

F_{zni} = force in z-direction on runner block i during phase n (N)

F_{yni} = force in y-direction on runner block i during phase n (N)



Vertical and horizontal forces

3 Profiled rail systems

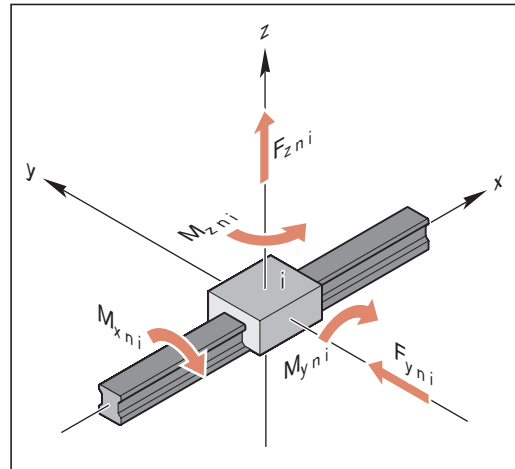
3.1 Principles

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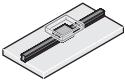
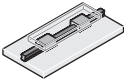
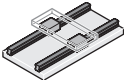
Vertical and horizontal forces combined with moments

For an external load acting on a runner block *i* in phase *n* due to vertical and horizontal forces combined with moments about the X, Y and Z-axes, the combined equivalent load on the bearing is calculated using the formulas given below.

Since it is assumed that the adjoining structure is infinitely rigid, moments can only occur in three specific layouts. The only situation in which moments can be taken up in all directions is a layout with just one runner block (see section 3.1.5.3).



Combination of force and moment loads

Layout	Formula
1 rail 1 runner block 	(3-32) $F_{combni} = F_{yni} + F_{zni} + C \cdot \frac{ M_{xni} }{M_t} + C \cdot \frac{ M_{yni} }{M_L} + C \cdot \frac{ M_{zni} }{M_L}$
1 rail 2 runner blocks 	(3-33) $F_{combni} = F_{yni} + F_{zni} + C \cdot \frac{ M_{xni} }{M_t}$
2 rails 2 runner blocks 	(3-34) $F_{combni} = F_{yni} + F_{zni} + C \cdot \frac{ M_{yni} }{M_L} + C \cdot \frac{ M_{zni} }{M_L}$

- F_{combni} = combined equivalent load on bearing for runner block *i* during phase *n* (N)
- F_{yni} = force in y-direction acting on runner block *i* during phase *n* (N)
- F_{zni} = force in z-direction acting on runner block *i* during phase *n* (N)
- M_{xni} = torsional moment about the X-axis acting on runner block *i* during phase *n* (Nm)
- M_{yni} = longitudinal moment about the Y-axis acting on runner block *i* during phase *n* (Nm)
- M_{zni} = longitudinal moment about the Z-axis acting on runner block *i* during phase *n* (Nm)
- C = dynamic load capacity (N)
- M_t = dynamic torsional moment load capacity (Nm)
- M_L = dynamic longitudinal moment load capacity (Nm)

3 Profiled rail systems

3.1 Principles

3.1.5 Calculations

Closely spaced runner blocks

If runner blocks are mounted on a guide rail one behind the other with a center-to-center distance L_W of less than 1.5 times the runner block length L_{FW} , this is likely to result in an unequal distribution of the load between the runner blocks. The reasons for this are inaccuracies in the mounting surfaces and the manufacturing tolerances of the guide components.

Contact factor

In such a case, the contact factor f_C is included when calculating the load on the bearing. The contact factor depends on the number of closely spaced runner blocks. It is a statistical value. In normal operation, with sufficient space between the runner blocks, the contact factor is $f_C = 1$.

The contact factor f_C can be calculated using the following formula:

$$(3-35) \quad f_C = \frac{i^{0.7}}{i}$$

f_C = contact factor (-)
 i = number of closely spaced runner blocks (-)

Because the contact factor f_C increases the equivalent load on the bearing, the nominal life will be reduced to a greater or lesser extent, depending on the load case.

$$(3-36) \quad F_{\text{comb } n \ i} = \frac{1}{f_C} \cdot (|F_{y \ n \ i}| + |F_{z \ n \ i}|)$$

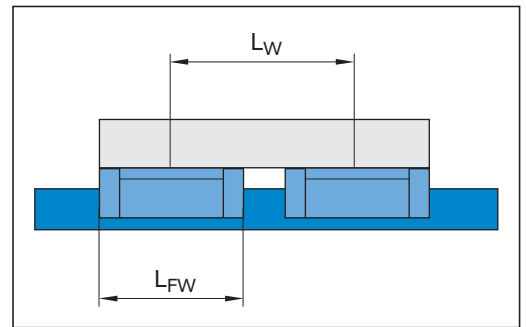
$F_{\text{comb } n \ i}$ = combined equivalent load on bearing for runner block i during phase n (N)

$F_{y \ n \ i}$ = force in y -direction acting on runner block i during phase n (N)

$F_{z \ n \ i}$ = force in z -direction acting on runner block i during phase n (N)

Moment loads

For situations with closely spaced runner blocks, the formulas (3-32) to (3-34) taking account of moment loads are also multiplied with the inverse of f_C .



Runner block length and center-to-center distance between runner blocks

Number of runner blocks	Contact factor f_C
1	1
2	0.81
3	0.72
4	0.66
5	0.62

3 Profiled rail systems

3.1 Principles

3.1.5 Calculations

3.1.5.5 Taking the preload into account

If the profiled rail systems used have a preload, this must be taken into account in the nominal life calculation for certain load cases.

To determine whether the preload will have an effect on the service life, the internal preload force of the runner block must first be calculated.

Preload force

$$(3-37) \quad F_{pr} = X_{pr} \cdot C$$

F_{pr} = internal loading of the runner block due to the preload (preload force) (N)
 X_{pr} = preload class factor (-)
 C = dynamic load capacity (N)

Preload class	Preload class factor X_{pr}
C0	0
C1	0.02
C2	0.08
C3	0.13

Lift-off force

The preload force and the rigidity curve for the runner block can be used to determine the load point at which individual raceways inside the runner block are relieved, i.e. the preload force becomes zero. This effect is known as "lift-off" and represents the limit for the external load. For

profiled rail systems, this point is expressed as the lift-off force F_{lim} . The force differs according to whether the rolling elements are balls or rollers. To simplify the calculation procedure, the lift-off force for ball rail systems is assumed to apply to both ball and roller rail systems.

$$(3-38) \quad F_{lim} = 2.8 \cdot F_{pr}$$

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

Distinction between cases

A distinction therefore has to be made between two cases:

Case 1: $F > F_{lim}$

If the external load, i.e. the combined equivalent load on the bearing in phase n for a runner block i is greater than the lift-off force, then the preload need not be considered when calculating the nominal life.

Preload may be disregarded:

$$F_{comb\ n\ i} > 2.8 \cdot F_{pr}$$

Case 2: $F \leq F_{lim}$

If the external load, i.e. the combined equivalent load on the bearing in phase n for a runner block i is smaller than or equal to 2.8 times the internal preload force, then the preload will have an effect on the nominal life.

Preload must be considered:

$$F_{comb\ n\ i} \leq 2.8 \cdot F_{pr}$$

Effective equivalent load on bearing

$$(3-39) \quad F_{eff\ n\ i} = F_{comb\ n\ i}$$

$F_{comb\ n\ i}$ = combined equivalent load on bearing for runner block i during phase n (N)
 F_{pr} = preload force (N)
 $F_{eff\ n\ i}$ = effective equivalent load on bearing during phase n for runner block i (N)

$$(3-40) \quad F_{eff\ n\ i} = \left(\frac{F_{comb\ n\ i}}{2.8 \cdot F_{pr}} + 1 \right)^{\frac{3}{2}} \cdot F_{pr}$$

3 Profiled rail systems

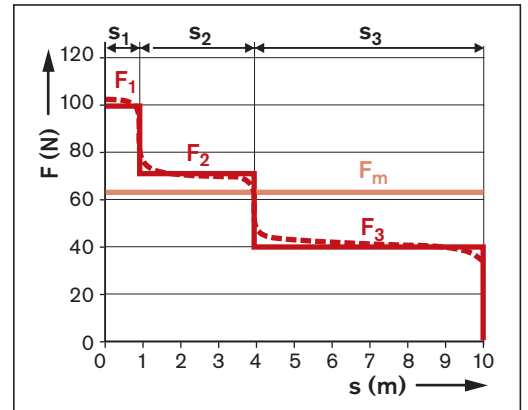
3.1 Principles

3.1.5 Calculations

3.1.5.6 Equivalent dynamic load on bearing

Varying forces

The equivalent dynamic load on the bearing F_m must always be calculated when varying process forces or varying weight forces are involved, or when the system is to be accelerated and decelerated. Calculation of the equivalent dynamic load combines the loads on a runner block i in the individual phases n to produce a resulting comparative load for the entire dynamic cycle. If the load on the bearing varies in steps, the equivalent dynamic load on the bearing is calculated in a manner similar to that for rotary anti-friction bearings.



Example of stepwise variations in loading

- - - Actual force profile
- Effective equivalent load in phases 1 ... n
- Equivalent dynamic load for entire cycle

Stepwise variations in bearing loads

For stepwise variations in loads, the equivalent dynamic load on the bearing is calculated according to discrete travel steps.

Calculation of discrete travel steps for phase n :

$$(3-41) \quad q_{sn} = \frac{s_n}{s} \cdot 100\%$$

$$(3-42) \quad s = s_1 + s_2 + \dots + s_n$$

- $q_{s1} \dots q_{sn}$ = discrete travel steps for phases 1 ... n (%)
- $s_1 \dots s_n$ = travel in phases 1 ... n (mm)
- s = total travel (mm)
- n = phase (-)

For each runner block i and each motion phase n , the individual loads are calculated as described in the preceding sections. These individual loads are then multiplied by the percentages for the discrete travel steps. Finally, formula (3-43) is used to calculate the equivalent load on the runner block throughout the entire motion cycle.

Equivalent dynamic loading of a runner block

$$(3-43) \quad F_{mi} = \sqrt[p]{(F_{eff1i})^p \cdot \frac{q_{s1}}{100\%} + (F_{eff2i})^p \cdot \frac{q_{s2}}{100\%} + \dots + (F_{effni})^p \cdot \frac{q_{sn}}{100\%}}$$

- $p = 3$ for ball rail guides
- $p = 10/3$ for roller rail guides
- F_{mi} = equivalent dynamic load on bearing for runner block i (N)
- $F_{eff1i} \dots F_{effni}$ = effective equivalent load on bearing for runner block i in phases 1 ... n (N)
- $q_{s1} \dots q_{sn}$ = discrete travel steps for phases 1 ... n (%)
- $s_1 \dots s_n$ = travel in phases 1 ... n (mm)
- s = total travel (mm)

3 Profiled rail systems

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3.1.5.7 Life expectancy

Nominal life

The calculated service life which an individual linear motion rolling bearing, or a group of apparently identical linear motion rolling bearings operating under the same conditions, can attain with a 90% probability, with contemporary,

commonly used materials and manufacturing quality under conventional operating conditions (per ISO 14728 Part 1).

The nominal life of a runner block *i* is calculated using the following formula. The result is the expected travel life in meters.

$$(3-44) \quad L_i = \left(\frac{C}{F_i} \right)^p \cdot 10^5 \text{ m}$$

- L_i = nominal life of the runner block *i* (m)
- C = dynamic load capacity (N)
- F_i = load on bearing of the runner block *i* (N)

- p = 3 for ball rail guides
- p = 10/3 for roller rail guides

Distinction according to load case

Depending on the load case, the following forces can be factored into the formula.

Load case	Force F_i	Nominal life	Description	Section
Force loads	$F_{y_{ni}}$ $F_{z_{ni}}$	(3-45) $L_i = \left(\frac{C}{F_{y_{ni}}} \right)^p \cdot 10^5 \text{ m}$	A constant force acting in the main load direction on the runner block <i>i</i>	3.1.5.3
Combined equivalent load on bearing	$F_{comb_{ni}}$	(3-46) $L_i = \left(\frac{C}{F_{comb_{ni}}} \right)^p \cdot 10^5 \text{ m}$	A constant force acting at a certain angle or a constant moment acting on the runner block <i>i</i>	3.1.5.4
Taking preload into account with the effective equivalent load on bearing	$F_{eff_{ni}}$	(3-47) $L_i = \left(\frac{C}{F_{eff_{ni}}} \right)^p \cdot 10^5 \text{ m}$	Effect of preload and a constant load on bearing on runner block <i>i</i>	3.1.5.5
Equivalent dynamic load on bearing	F_{m_i}	(3-48) $L_i = \left(\frac{C}{F_{m_i}} \right)^p \cdot 10^5 \text{ m}$	Varying load on bearing acting in <i>n</i> phases on runner block <i>i</i>	3.1.5.6

Nominal life in operating hours

If the stroke length s_{stroke} and the stroke frequency n_{stroke} are constant throughout the service life, the service life in operating hours can be calculated as follows:

$$(3-49) \quad L_{h_i} = \frac{L_i}{2 \cdot s_{stroke} \cdot n_{stroke} \cdot 60}$$

- L_{h_i} = nominal life (h)
- L_i = nominal life (m)
- s_{stroke} = stroke length (m)
- n_{stroke} = stroke frequency (full cycles per minute) (min^{-1})

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Nominal life in operating hours

Alternatively, the service life in operating hours can be calculated using an average speed v_m . When the speed varies in steps, this average speed is calculated using the discrete time steps q_{tn} of the individual phases.

Formula for calculating the discrete time steps:

$$(3-50) \quad q_{tn} = \frac{t_n}{t} \cdot 100\%$$

$$(3-51) \quad t = t_1 + t_2 + \dots + t_n$$

Formula for calculating the average speed:

$$(3-52) \quad v_m = \frac{|v_1| \cdot q_{t1} + |v_2| \cdot q_{t2} + \dots + |v_n| \cdot q_{tn}}{100\%}$$

Formula for calculating the service life in operating hours using the average speed:

$$(3-53) \quad L_{hi} = \frac{L_i}{60 \cdot v_m}$$

Nominal life calculation limits

According to ISO 14728-1, the nominal life calculation performed using the above formulas will only be valid under the following conditions:

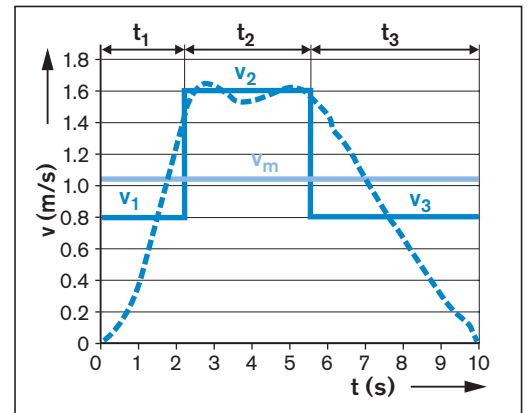
- The load must not exceed a certain level.
- The stroke must not be less than a certain length.
- There must be no major vibrations when the rolling bearing is in service.

Load limits

The loads on the bearing must be in a certain ratio to the static and dynamic load capacities.

Load limits for the nominal life calculation:

- $F \leq 0.5 C$
- $F \leq C_0$



Example of stepwise variations in speed

- Actual speed profile
- Average speed in phases 1 ... 3
- Average speed over the entire cycle

- $q_{t1} \dots q_{tn}$ = discrete time steps for phases 1 ... n (%)
- $t_1 \dots t_n$ = times for phases 1 ... n (s)
- t = total time (s)
- v_m = average speed (m/min)
- $v_1 \dots v_n$ = average speed in phases 1 ... n (m/min)
- L_{hi} = nominal life in operating hours (h)
- L_i = nominal life (m)

- F = load on bearing (N)
- C = dynamic load capacity (N)
- C_0 = static load capacity (N)

3 Profiled rail systems

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Limitation due to short stroke

When the stroke is less than two times the runner block length, not all the rolling elements will undergo rolling contact in the load-bearing zone. These applications are called short-stroke applications.

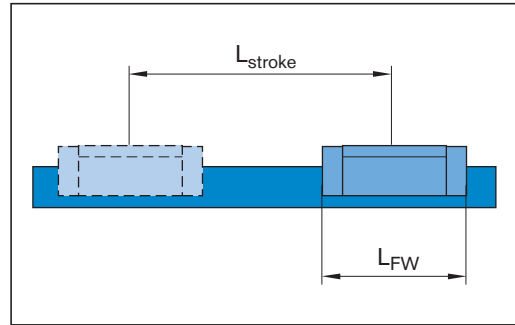
Stroke limit for the nominal life calculation:

$$\blacksquare L_{\text{stroke}} > 2 \cdot L_{\text{FW}}$$

Definition of short stroke:

$$\blacksquare L_{\text{stroke}} < 2 \cdot L_{\text{FW}}$$

If the application is a short stroke one, this must be taken into account in the nominal life calculation. The expected service life will then be shorter than the nominal life. Users should contact Rexroth for assistance in calculating the service life for such applications.



Stroke length and runner block length

L_{FW} = runner block length (mm)
 L_{stroke} = stroke length (mm)

Limitation due to vibrations

Vibrations can considerably shorten the service life of a profiled rail system. If major vibrations are expected in the machine environment, users should contact Rexroth for advice.

Modified life expectancy

The probability that a guide will attain the nominal life L is 90%. If a higher life expectancy is desired, the calculations are performed using the modified life expectancy L_{na} . The nominal life L is then multiplied by the coefficient a_1 . This coefficient stands for the probability of survival of the runner blocks and is shown in the table alongside.

Probability of survival (%)	L_{na}	a_1
90	L_{10a}	1
95	L_{5a}	0.62
96	L_{4a}	0.53
97	L_{3a}	0.44
98	L_{2a}	0.33
99	L_{1a}	0.21

$$(3-54) \quad L_{\text{na}i} = a_1 \cdot \left(\frac{C}{F_i}\right)^p \cdot 10^5 \text{ m}$$

$L_{\text{na}i}$ = modified life expectancy (m)
 C = dynamic load capacity (N)
 F_i = load on bearing of the runner block i (N)
 a_1 = life expectancy coefficient (-)

$p = 3$ for ball rail guides
 $p = 10/3$ for roller rail guides

3 Profiled rail systems

3.1 Principles

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3.1.5.8 Equivalent static load on bearing

High static loads

If the runner block is subjected to high static loads, the equivalent static load on the bearing must be calculated. A static load is a load acting on the runner block while it is at rest, not while it is traveling.

The equivalent static load is distinguished according to its load components:

- Horizontal and vertical forces
- Horizontal and vertical forces combined with moments

The equivalent static load $F_{0\text{comb}i}$ must not exceed the static load capacity C_0 . The equivalent static load is required to determine the static load safety factor, see section 3.1.5.9.

Just as in dynamic load cases, a check must be performed in static load cases to determine whether the preload will have an effect (see section 3.1.5.5). If the preload does have an effect, the effective equivalent load on the bearing is calculated using formula (3-40).

Horizontal and vertical forces

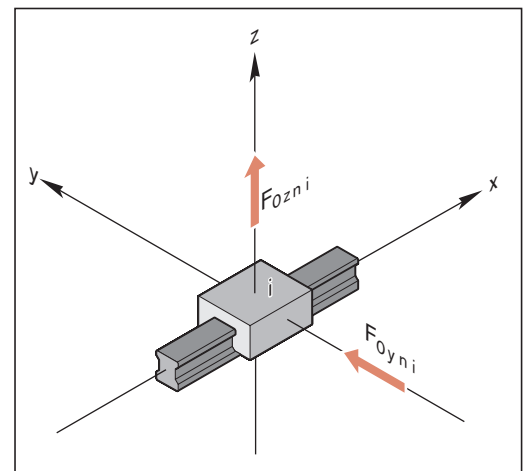
External static loads acting at an angle must be resolved into their horizontal (y-direction) and vertical (z-direction) components. The absolute values of these two components are then added. The structure of the profiled rail systems allows this simplified calculation.

$$(3-55) \quad F_{0\text{comb}i} = |F_{0y_i}| + |F_{0z_i}|$$

$F_{0\text{comb}i}$ = equivalent static load on bearing of the runner block i (N)

F_{0y_i} = static load due to a force in the y-direction acting on runner block i (N)

F_{0z_i} = static load due to a force in the z-direction acting on runner block i (N)



Horizontal and vertical forces

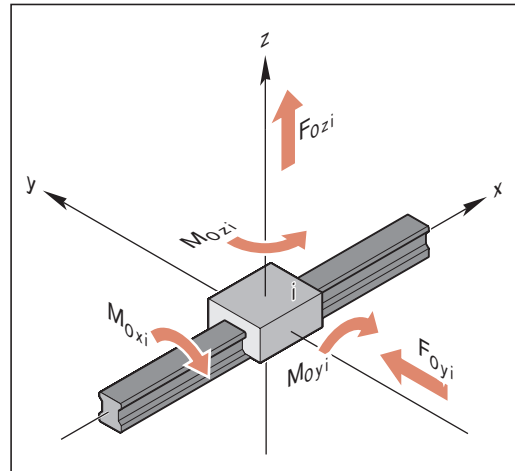
3 Profiled rail systems

3.1 Principles

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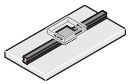
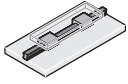
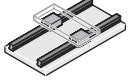
Horizontal and vertical forces combined with moments

The combined external static load on the bearing due to forces and moments about the X, Y and Z-axes occurs only in certain layouts (see section 3.1.5.3).



Combination of force and moment loads

The formulas for the respective layouts of profiled rail systems are shown in the table below.

Layout	Formula
1 rail 1 runner block 	(3-56) $F_{0combi} = F_{0yi} + F_{0zi} + C_0 \cdot \frac{ M_{0xi} }{M_{t0}} + C_0 \cdot \frac{ M_{0yi} }{M_{L0}} + C_0 \cdot \frac{ M_{0zi} }{M_{L0}}$
1 rail 2 runner blocks 	(3-57) $F_{0combi} = F_{0yi} + F_{0zi} + C_0 \cdot \frac{ M_{0xi} }{M_{t0}}$
2 rails 2 runner blocks 	(3-58) $F_{0combi} = F_{0yi} + F_{0zi} + C_0 \cdot \frac{ M_{0yi} }{M_{L0}} + C_0 \cdot \frac{ M_{0zi} }{M_{L0}}$

F_{0combi} = equivalent static load on bearing of the runner block i (N)

F_{0yi} = force in the y-direction acting on runner block i (N)

F_{0zi} = force in the z-direction acting on runner block i (N)

C_0 = static load capacity (N)

M_{0xi} = static torsional moment about the X-axis acting on runner block i (Nm)

M_{0yi} = static longitudinal moment about the Y-axis acting on runner block i (Nm)

M_{0zi} = static longitudinal moment about the Z-axis acting on runner block i (Nm)

M_{t0} = static torsional moment load capacity (Nm)

M_{L0} = static longitudinal moment load capacity (Nm)

3 Profiled rail systems

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3.1.5.9 Static load safety factor

The static load safety factor S_0 is calculated to assure that the rolling elements and the raceways will not be subjected to impermissible loading. This calculation is based on the maximum load on the most heavily loaded runner block. For a static

Case 1: Calculation using the maximum static load on bearing $F_{0\max}$

$$(3-59) \quad S_0 = \frac{C_0}{F_{0\max}} = \frac{C_0}{F_{0\text{combi}}}$$

S_0	= static load safety factor	(-)
C_0	= static load capacity	(N)
$F_{0\max}$	= maximum static load on bearing	(N)
$F_{0\text{combi}}$	= maximum equivalent static load on bearing of the runner block i	(N)

Irrespective of the static load safety factor, it must be ensured that the maximum permissible loads of the profiled rail system are not exceeded. The maximum permissible load is determined by the structural strength of the runner blocks. Values for the maximum permissible loads are indicated in the product catalogs for some of the profiled rail systems.

load, this will be the maximum static load on the bearing $F_{0\max}$, and for a purely dynamic load, the maximum dynamic load on the bearing F_{\max} .

Case 2: Calculation using the maximum dynamic load on bearing F_{\max}

$$(3-60) \quad S_0 = \frac{C_0}{F_{\max}} = \frac{C_0}{F_{\text{eff}ni}}$$

F_{\max}	= maximum dynamic load on bearing	(N)
$F_{\text{eff}ni}$	= maximum dynamic load on bearing in phase n on runner block i	(N)

Recommendations for the static load safety factor:

Conditions of use	S_0
Normal conditions of use ¹⁾	1 ... 2
Low impact loads and vibrations	2 ... 4
Moderate impact loads and vibrations	3 ... 5
Heavy impact loads and vibrations	4 ... 6
Unknown load parameters	6 ... 15

1) Normal conditions of use are defined in Chapter 2, section 2.4.2.4.

3.1.5.10 Example of a nominal life calculation

**Step 1:
Define the operating
conditions**

The system in this calculation example comprises 2 rails and 4 runner blocks. This is a very commonly used layout. The required service life is 10,000 hours. The motion cycle to be calculated involves weight forces at the center of gravity of the mass and a process force F_p . To make the calculation example easier to understand, a simplified dynamic cycle without a return stroke is assumed.

A size 30 ball rail system was chosen as the outcome of the product selection procedure.

Profiled rail system details		Value
Profiled rail guide		Ball rail system
Size		30
Runner block		Flanged, long, standard height (FLS), without ball chain
Part number		R1653 721 20
Preload class		C2
Preload class factor	X_{pr}	0.08
Accuracy class		SP
Dynamic load capacity	C	40 000 N
Static load capacity	C_0	57 800 N

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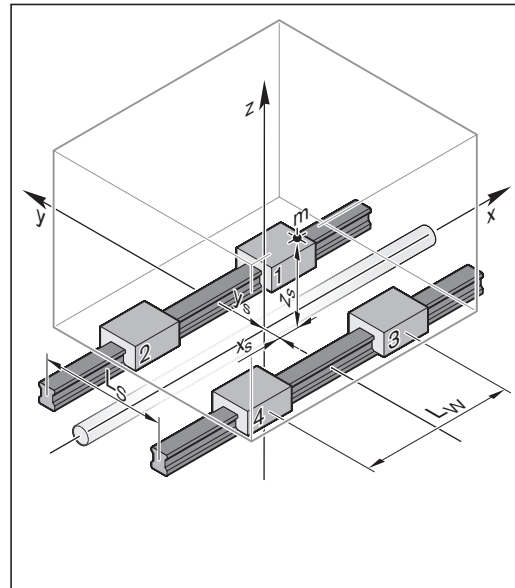
3.1 Principles

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Layout

The chosen layout is an arrangement with 2 rails and 4 runner blocks installed horizontally.

Layout details		
Parameter		Value
Distance between runner blocks	L_W	600 mm
Distance between guide rails	L_S	450 mm
y-coordinate of drive unit	L_y	0 mm
z-coordinate of drive unit	L_z	0 mm
Angular location rel. to X-axis	α	0 °
Angular location rel. to Y-axis	β	0 °
Mass of machine table	m	450 kg
x-coordinate of center of gravity	x_S	300 mm
y-coordinate of center of gravity	y_S	-50 mm
z-coordinate of center of gravity	z_S	250 mm



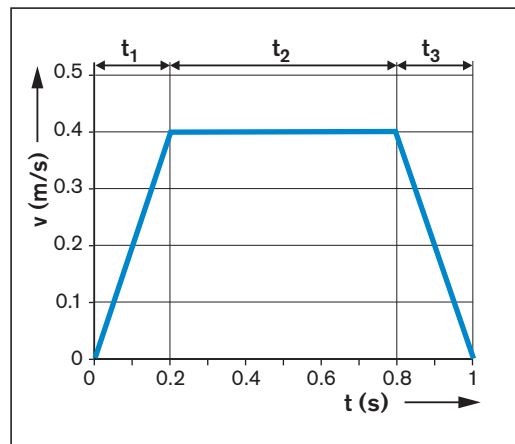
Layout of the guide system and location of the mass center of gravity in the calculation example with 2 rails and 4 runner blocks

Dynamic cycle

The dynamic cycle consists of three phases:

- Phase 1: acceleration
- Phase 2: processing
- Phase 3: deceleration

Dynamic cycle details		
Parameter		Value
Number of phases	n	3
Time for phase 1	t_1	0.2 s
Time for phase 2	t_2	0.6 s
Time for phase 3	t_3	0.2 s
Travel in phase 1	s_1	0.04 m
Travel in phase 2	s_2	0.24 m
Travel in phase 3	s_3	0.04 m
Acceleration in phase 1	a_1	2 m/s ²
Acceleration in phase 2	a_2	0 m/s ²
Acceleration in phase 3	a_3	-2 m/s ²



Calculation example: Speeds in the individual phases

3 Profiled rail systems

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Discrete travel steps Calculate the discrete travel steps q_{sn} using formulas (3-42) and (3-41):

$$s = \sum s_n = s_1 + s_2 + s_3 = 0.04 \text{ m} + 0.24 \text{ m} + 0.04 \text{ m} = 0.32 \text{ m}$$

$$q_{sn} = \frac{s_n}{s} \cdot 100\%$$

$$q_{s1} = \frac{s_1}{s} \cdot 100\% = \frac{0.04 \text{ m}}{0.32 \text{ m}} \cdot 100\% = 12.5\%$$

$$q_{s2} = \frac{s_2}{s} \cdot 100\% = \frac{0.24 \text{ m}}{0.32 \text{ m}} \cdot 100\% = 75\%$$

$$q_{s3} = \frac{s_3}{s} \cdot 100\% = \frac{0.04 \text{ m}}{0.32 \text{ m}} \cdot 100\% = 12.5\%$$

Discrete time steps Calculate the discrete time steps q_{tn} using formulas (3-51) and (3-50):

$$t = \sum t_n = t_1 + t_2 + t_3 = 0.2 \text{ s} + 0.6 \text{ s} + 0.2 \text{ s} = 1 \text{ s}$$

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

$$q_{t1} = \frac{t_1}{t} \cdot 100\% = \frac{0.2 \text{ s}}{1 \text{ s}} \cdot 100\% = 20\%$$

$$q_{t2} = \frac{t_2}{t} \cdot 100\% = \frac{0.6 \text{ s}}{1 \text{ s}} \cdot 100\% = 60\%$$

$$q_{t3} = \frac{t_3}{t} \cdot 100\% = \frac{0.2 \text{ s}}{1 \text{ s}} \cdot 100\% = 20\%$$

Average speeds

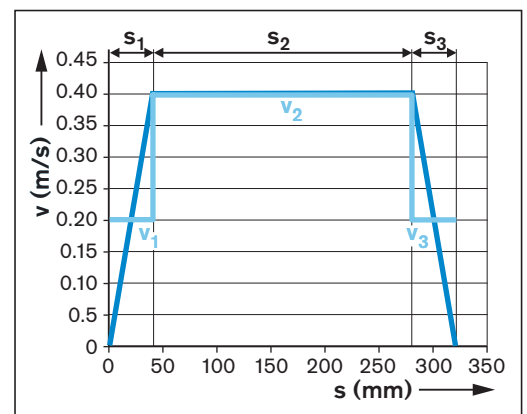
Calculate the average speeds in the individual phases:

$$v_n = \frac{s_n}{t_n}$$

$$v_1 = \frac{s_1}{t_1} = \frac{0.04 \text{ m}}{0.2 \text{ s}} = 0.2 \frac{\text{m}}{\text{s}} = 12 \frac{\text{m}}{\text{min}}$$

$$v_2 = \frac{s_2}{t_2} = \frac{0.24 \text{ m}}{0.6 \text{ s}} = 0.4 \frac{\text{m}}{\text{s}} = 24 \frac{\text{m}}{\text{min}}$$

$$v_3 = \frac{s_3}{t_3} = \frac{0.04 \text{ m}}{0.2 \text{ s}} = 0.2 \frac{\text{m}}{\text{s}} = 12 \frac{\text{m}}{\text{min}}$$



Travel-speed curve

— Actual speed profile
— Average speed

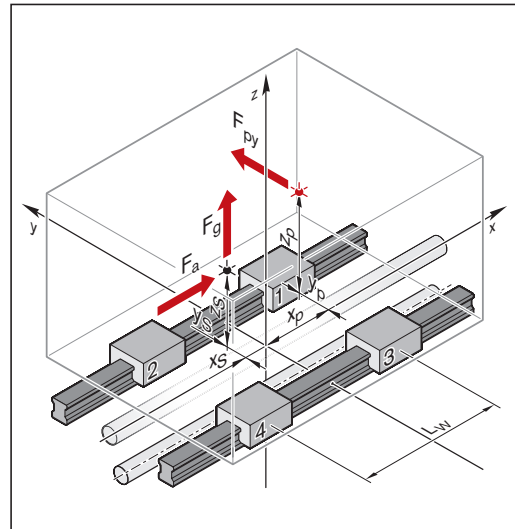
3 Profiled rail systems

3.1 Principles

3.1.5 Calculations

Load

Load details		
Parameter		Value
Mass of machine table	m	450 kg
x-coordinate of center of gravity	x_S	300 mm
y-coordinate of center of gravity	y_S	-50 mm
z-coordinate of center of gravity	z_S	250 mm
Process force in y-direction	F_p	-4500 N
x-coordinate of process force application point	x_p	200 mm
y-coordinate of process force application point	y_p	150 mm
z-coordinate of process force application point	z_p	500 mm



Analyze the effective forces:

Phase n	Effective force
1	Weight force and acceleration force
2	Weight force and process force
3	Weight force and acceleration force

Acceleration force F_a , weight force F_g , process force F_p and coordinates of the force application points

Since the drive unit lies along the origin of the coordinate system, there is no need to include the drive force in the calculation.

Calculate the effective forces:

Effective force	Calculation
Weight force	$F_g = m \cdot g = 450 \text{ kg} \cdot \left(-9.81 \frac{\text{m}}{\text{s}^2}\right) = -4415 \text{ N}$
Acceleration force in positive x-direction during approach. Because of the mass moment of inertia, the force acts in the negative x-direction.	$F_{a1} = (-1) \cdot m \cdot a_1 = 450 \text{ kg} \cdot 2 \frac{\text{m}}{\text{s}^2} = -900 \text{ N}$
Acceleration force in negative x-direction during deceleration. Because of the mass moment of inertia, the force acts in the positive x-direction.	$F_{a3} = (-1) \cdot m \cdot a_3 = (-1) \cdot 450 \text{ kg} \cdot \left(-2 \frac{\text{m}}{\text{s}^2}\right) = 900 \text{ N}$

Analyze and assign the load cases:

Load case j	Description	Effective force $F_{w,j}$	Force application point $x_{w,j}$, $y_{w,j}$, $z_{w,j}$	Phase n
1	Weight force	$F_{wz,1} = F_g = -4415 \text{ N}$	$x_{w,1} = x_S = 300 \text{ mm}$ $y_{w,1} = y_S = -50 \text{ mm}$ $z_{w,1} = z_S = 250 \text{ mm}$	1; 2; 3
2	Acceleration force of $a_1 = 2 \text{ m/s}^2$	$F_{wx,2} = F_{a1} = -900 \text{ N}$	$x_{w,2} = x_S = 300 \text{ mm}$ $y_{w,2} = y_S = -50 \text{ mm}$ $z_{w,2} = z_S = 250 \text{ mm}$	1
3	Process force during machining	$F_{wy,3} = F_p = -4500 \text{ N}$	$x_{w,3} = x_p = 200 \text{ mm}$ $y_{w,3} = y_p = 150 \text{ mm}$ $z_{w,3} = z_p = 500 \text{ mm}$	2
4	Acceleration force of $a_3 = -2.5 \text{ m/s}^2$	$F_{wx,4} = F_{a3} = +900 \text{ N}$	$x_{w,4} = x_S = 300 \text{ mm}$ $y_{w,4} = y_S = -50 \text{ mm}$ $z_{w,4} = z_S = 250 \text{ mm}$	3

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3.1 Principles

3.1.5 Calculations

Step 2:**Calculate the loads due to forces and moments**

For a 2-rail/4-runner block layout, only the forces have to be calculated because – assuming an infinitely rigid adjoining structure – no moments arise at the runner blocks.

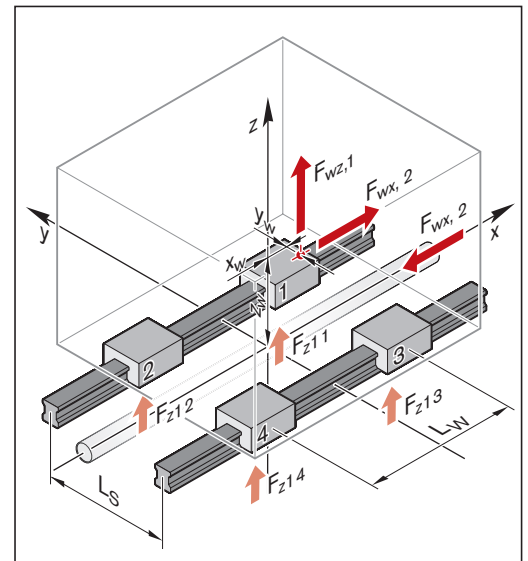
Calculate the forces using the formulas:

- (3-25) to (3-28) for the loads on the runner blocks in the z-direction
- (3-29) to (3-30) for the loads on the runner blocks in the y-direction

Calculate load on bearing per runner block in phase 1

Load on runner blocks due to

- Weight force $F_g = -4415 \text{ N}$
- Acceleration force $F_{a1} = -900 \text{ N}$ during approach



Loads on bearing in the z-direction during phase 1

Load case j	$F_{wx,j}$	$F_{wy,j}$	$F_{wz,j}$	$x_{w,j}$	$y_{w,j}$	$z_{w,j}$
1	-	-	$F_{wz,1} = -4415 \text{ N}$	$x_{w,1} = 300 \text{ mm}$	$y_{w,1} = -50 \text{ mm}$	$z_{w,1} = 250 \text{ mm}$
2	$F_{wx,2} = -900 \text{ N}$	-	-	$x_{w,2} = 300 \text{ mm}$	$y_{w,2} = -50 \text{ mm}$	$z_{w,2} = 250 \text{ mm}$

Lift-off/down loads in the z-direction

$$F_{z11} = \frac{F_{wz,1}}{4} + \frac{F_{wz,1} \cdot y_{w,1}}{2 \cdot L_s} + \frac{(F_{wz,1} \cdot x_{w,1}) - (F_{wx,2} \cdot z_{w,2})}{2 \cdot L_w}$$

$$F_{z11} = \frac{(-4415 \text{ N})}{4} + \frac{(-4415 \text{ N}) \cdot (-50 \text{ mm})}{2 \cdot 450 \text{ mm}} + \frac{((-4415 \text{ N}) \cdot 300 \text{ mm}) - ((-900 \text{ N}) \cdot 250 \text{ mm})}{2 \cdot 600 \text{ mm}} = -1775 \text{ N}$$

$$F_{z12} = \frac{F_{wz,1}}{4} + \frac{F_{wz,1} \cdot y_{w,1}}{2 \cdot L_s} + \frac{(F_{wx,2} \cdot z_{w,2}) - (F_{wz,1} \cdot x_{w,1})}{2 \cdot L_w}$$

$$F_{z12} = \frac{(-4415 \text{ N})}{4} + \frac{(-4415 \text{ N}) \cdot (-50 \text{ mm})}{2 \cdot 450 \text{ mm}} + \frac{((-900 \text{ N}) \cdot 250 \text{ mm}) - ((-4415 \text{ N}) \cdot 300 \text{ mm})}{2 \cdot 600 \text{ mm}} = 58 \text{ N}$$

$$F_{z13} = \frac{F_{wz,1}}{4} + \frac{-(F_{wz,1} \cdot y_{w,1})}{2 \cdot L_s} + \frac{(F_{wz,1} \cdot x_{w,1}) - (F_{wx,2} \cdot z_{w,2})}{2 \cdot L_w}$$

$$F_{z13} = \frac{(-4415 \text{ N})}{4} + \frac{-((-4415 \text{ N}) \cdot (-50 \text{ mm}))}{2 \cdot 450 \text{ mm}} + \frac{((-4415 \text{ N}) \cdot 300 \text{ mm}) - ((-900 \text{ N}) \cdot 250 \text{ mm})}{2 \cdot 600 \text{ mm}} = -2265 \text{ N}$$

$$F_{z14} = \frac{F_{wz,1}}{4} + \frac{-(F_{wz,1} \cdot y_{w,1})}{2 \cdot L_s} + \frac{(F_{wx,2} \cdot z_{w,2}) - (F_{wz,1} \cdot x_{w,1})}{2 \cdot L_w}$$

$$F_{z14} = \frac{(-4415 \text{ N})}{4} + \frac{-((-4415 \text{ N}) \cdot (-50 \text{ mm}))}{2 \cdot 450 \text{ mm}} + \frac{((-900 \text{ N}) \cdot 250 \text{ mm}) - ((-4415 \text{ N}) \cdot 300 \text{ mm})}{2 \cdot 600 \text{ mm}} = -433 \text{ N}$$

3 Profiled rail systems

3.1 Principles

3.1.5 Calculations

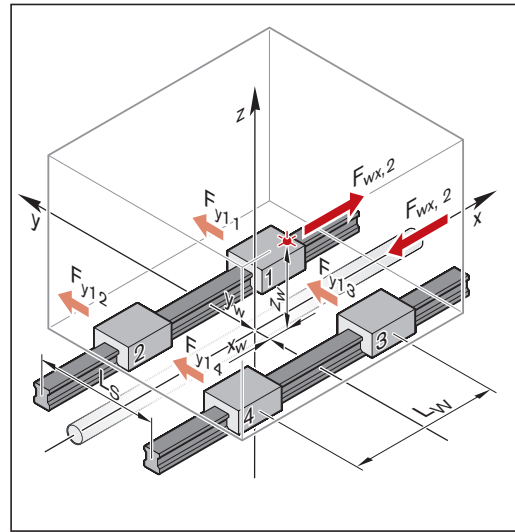
Side loads in the y-direction

$$F_{y11} = \frac{- (F_{wx,2} \cdot y_{w,2})}{2 \cdot L_W} = \frac{-((-900 \text{ N}) \cdot (-50 \text{ mm}))}{2 \cdot 600 \text{ mm}} = -38 \text{ N}$$

$$F_{y12} = - \frac{- (F_{wx,2} \cdot y_{w,2})}{2 \cdot L_W} = \frac{((-900 \text{ N}) \cdot (-50 \text{ mm}))}{2 \cdot 600 \text{ mm}} = 38 \text{ N}$$

$$F_{y13} = \frac{- (F_{wx,2} \cdot y_{w,2})}{2 \cdot L_W} = \frac{-((-900 \text{ N}) \cdot (-50 \text{ mm}))}{2 \cdot 600 \text{ mm}} = -38 \text{ N}$$

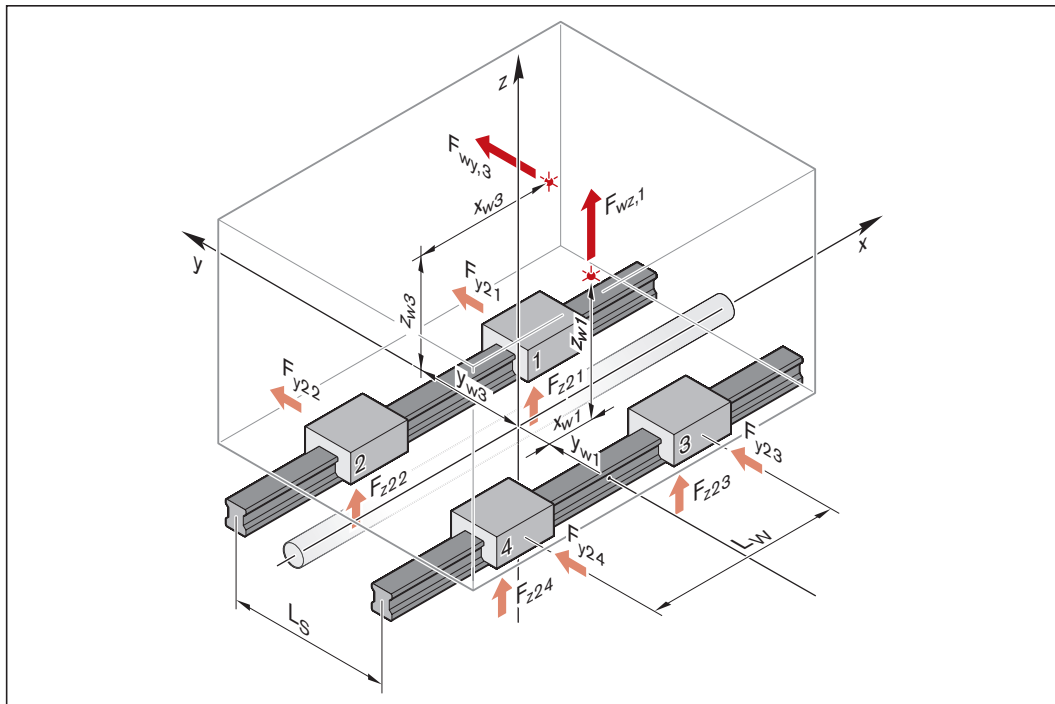
$$F_{y14} = - \frac{- (F_{wx,2} \cdot y_{w,2})}{2 \cdot L_W} = \frac{(-900 \text{ N}) \cdot (-50 \text{ mm})}{2 \cdot 600 \text{ mm}} = 38 \text{ N}$$



Loads on bearing in the y-direction during phase 1

Calculate load on bearing per runner block in phase 2

- Load on runner blocks due to Weight force $F_g = -4415 \text{ N}$
- Process force $F_p = -4500 \text{ N}$ during machining



Loads on bearing in the z- and y-direction during phase 2

Load case j	$F_{wx,j}$	$F_{wy,j}$	$F_{wz,j}$	$x_{w,j}$	$y_{w,j}$	$z_{w,j}$
1	-	-	$F_{wz,1} = -4415 \text{ N}$	$x_{w,1} = 300 \text{ mm}$	$y_{w,1} = -50 \text{ mm}$	$z_{w,1} = 250 \text{ mm}$
3	-	$F_{wy,3} = -4500 \text{ N}$	-	$x_{w,3} = 200 \text{ mm}$	$y_{w,3} = 150 \text{ mm}$	$z_{w,3} = 500 \text{ mm}$

3 Profiled rail systems

3.1 Principles

3.1.5 Calculations

Lift-off/down loads in the z-direction

$$F_{z21} = \frac{F_{wz,1}}{4} + \frac{(F_{wz,1} \cdot y_{w,1}) - (F_{wy,3} \cdot z_{w,3})}{2 \cdot L_S} + \frac{(F_{wz,1} \cdot x_{w,1})}{2 \cdot L_W}$$

$$F_{z21} = \frac{(-4415 \text{ N})}{4} + \frac{(-4415 \text{ N}) \cdot (-50 \text{ mm}) - ((-4500 \text{ N}) \cdot 500 \text{ mm})}{2 \cdot 450 \text{ mm}} + \frac{(-4415 \text{ N}) \cdot 300 \text{ mm}}{2 \cdot 600 \text{ mm}} = 538 \text{ N}$$

$$F_{z22} = \frac{F_{wz,1}}{4} + \frac{(F_{wz,1} \cdot y_{w,1}) - (F_{wy,3} \cdot z_{w,3})}{2 \cdot L_S} + \frac{-(F_{wz,1} \cdot x_{w,1})}{2 \cdot L_W} = 2745 \text{ N}$$

$$F_{z23} = \frac{F_{wz,1}}{4} + \frac{(F_{wy,3} \cdot z_{w,3}) - (F_{wz,1} \cdot y_{w,1})}{2 \cdot L_S} + \frac{(F_{wz,1} \cdot x_{w,1})}{2 \cdot L_W} = -4953 \text{ N}$$

$$F_{z24} = \frac{F_{wz,1}}{4} + \frac{(F_{wy,3} \cdot z_{w,3}) - (F_{wz,1} \cdot y_{w,1})}{2 \cdot L_S} + \frac{-(F_{wz,1} \cdot x_{w,1})}{2 \cdot L_W} = -2745 \text{ N}$$

Side loads in the y-direction

$$F_{y21} = \frac{F_{wy,3}}{4} + \frac{F_{wy,3} \cdot x_{w,3}}{2 \cdot L_W} = \frac{(-4500 \text{ N})}{4} + \frac{(-4500 \text{ N}) \cdot 200 \text{ mm}}{2 \cdot 600 \text{ mm}} = -1875 \text{ N}$$

$$F_{y22} = \frac{F_{wy,3}}{4} - \frac{F_{wy,3} \cdot x_{w,3}}{2 \cdot L_W} = -375 \text{ N}$$

$$F_{y23} = \frac{F_{wy,3}}{4} + \frac{F_{wy,3} \cdot x_{w,3}}{2 \cdot L_W} = -1875 \text{ N}$$

$$F_{y24} = \frac{F_{wy,3}}{4} - \frac{F_{wy,3} \cdot x_{w,3}}{2 \cdot L_W} = -375 \text{ N}$$

3 Profiled rail systems

3.1 Principles

3.1.5 Calculations

Calculate load on bearing per runner block in phase 3

Load on runner blocks due to

■ Weight force $F_g = -4415 \text{ N}$ ■ Acceleration force $F_{a3} = 900 \text{ N}$ during deceleration

Refer to illustrations for phase 1.

Load case j	$F_{wx,j}$	$F_{wy,j}$	$F_{wz,j}$	$x_{w,j}$	$y_{w,j}$	$z_{w,j}$
1	-	-	$F_{wz,1} = -4415 \text{ N}$	$x_{w,1} = 300 \text{ mm}$	$y_{w,1} = -50 \text{ mm}$	$z_{w,1} = 250 \text{ mm}$
4	$F_{wx,4} = 900 \text{ N}$	-	-	$x_{w,4} = 300 \text{ mm}$	$y_{w,4} = -50 \text{ mm}$	$z_{w,4} = 250 \text{ mm}$

Lift-off/down loads in the z-direction

$$F_{z31} = \frac{F_{wz,1}}{4} + \frac{F_{wz,1} \cdot y_{w,1}}{2 \cdot L_S} + \frac{(F_{wz,1} \cdot x_{w,1}) - (F_{wx,4} \cdot z_{w,4})}{2 \cdot L_W}$$

$$F_{z31} = \frac{(-4415 \text{ N})}{4} + \frac{(-4415 \text{ N}) \cdot (-50 \text{ mm})}{2 \cdot 450 \text{ mm}} + \frac{((-4415 \text{ N}) \cdot 300 \text{ mm}) - (900 \text{ N} \cdot 250 \text{ mm})}{2 \cdot 600 \text{ mm}} = -2150 \text{ N}$$

$$F_{z32} = \frac{F_{wz,1}}{4} + \frac{F_{wz,1} \cdot y_{w,1}}{2 \cdot L_S} + \frac{(F_{wx,4} \cdot z_{w,4}) - (F_{wz,1} \cdot x_{w,1})}{2 \cdot L_W} = 433 \text{ N}$$

$$F_{z33} = \frac{F_{wz,1}}{4} + \frac{-(F_{wz,1} \cdot y_{w,1})}{2 \cdot L_S} + \frac{(F_{wz,1} \cdot x_{w,1}) - (F_{wx,4} \cdot z_{w,4})}{2 \cdot L_W} = -2640 \text{ N}$$

$$F_{z34} = \frac{F_{wz,1}}{4} + \frac{-(F_{wz,1} \cdot y_{w,1})}{2 \cdot L_S} + \frac{(F_{wx,4} \cdot z_{w,4}) - (F_{wz,1} \cdot x_{w,1})}{2 \cdot L_W} = -58 \text{ N}$$

Side loads in the y-direction

$$F_{y31} = \frac{-(F_{wx,4} \cdot y_{w,4})}{2 \cdot L_W} = \frac{-(900 \text{ N} \cdot (-50 \text{ mm}))}{2 \cdot 600 \text{ mm}} = 38 \text{ N}$$

$$F_{y32} = -\frac{-(F_{wx,4} \cdot y_{w,4})}{2 \cdot L_W} = -38 \text{ N}$$

$$F_{y33} = \frac{-(F_{wx,4} \cdot y_{w,4})}{2 \cdot L_W} = 38 \text{ N}$$

$$F_{y34} = -\frac{-(F_{wx,4} \cdot y_{w,4})}{2 \cdot L_W} = -38 \text{ N}$$

**Intermediate results:
Loads on bearing
per runner block and
phase**

Runner block i	Phase 1		Phase 2		Phase 3	
	F_{z1i}	F_{y1i}	F_{z2i}	F_{y2i}	F_{z3i}	F_{y3i}
1	-1775 N	-38 N	538 N	-1875 N	-2150 N	38 N
2	58 N	38 N	2745 N	-375 N	433 N	-38 N
3	-2265 N	-38 N	-4953 N	-1875 N	-2640 N	38 N
4	-433 N	38 N	-2745 N	-375 N	-58 N	-38 N

3 Profiled rail systems

3.1 Principles

3.1.5 Calculations

Note

It is also possible to calculate the loads on the runner blocks individually for each load case (weight force, acceleration forces, and process force).

If this is done, then the calculated individual loads have to be added as appropriate for the load case combination in the respective phase. This makes the calculation clearer.

Step 3:
Calculate combined equivalent load on bearing

The combined equivalent load on the bearing is calculated using formula (3-31).

Calculate the combined equivalent load on bearing in phase 1 for runner blocks 1 to 4:

$$F_{\text{comb } 1 \ 1} = |F_{z \ 1 \ 1}| + |F_{y \ 1 \ 1}| = |-1775 \text{ N}| + |-38 \text{ N}| = 1813 \text{ N}$$

$$F_{\text{comb } 1 \ 2} = |F_{z \ 1 \ 2}| + |F_{y \ 1 \ 2}| = |58 \text{ N}| + |38 \text{ N}| = 96 \text{ N}$$

$$F_{\text{comb } 1 \ 3} = |F_{z \ 1 \ 3}| + |F_{y \ 1 \ 3}| = |-2265 \text{ N}| + |-38 \text{ N}| = 2303 \text{ N}$$

$$F_{\text{comb } 1 \ 4} = |F_{z \ 1 \ 4}| + |F_{y \ 1 \ 4}| = |-433 \text{ N}| + |38 \text{ N}| = 471 \text{ N}$$

Calculate the combined equivalent load on bearing in phase 2 and phase 3 for runner blocks 1 to 4:

$$F_{\text{comb } 2 \ 1} = |F_{z \ 2 \ 1}| + |F_{y \ 2 \ 1}| = |538 \text{ N}| + |-1875 \text{ N}| = 2413 \text{ N}$$

...

$$F_{\text{comb } 3 \ 4} = |F_{z \ 3 \ 4}| + |F_{y \ 3 \ 4}| = |-58 \text{ N}| + |-38 \text{ N}| = 96 \text{ N}$$

Intermediate results:
Combined equivalent loads on bearing per runner block and phase

Runner block i	Phase 1 $F_{\text{comb } 1 \ i}$	Phase 2 $F_{\text{comb } 2 \ i}$	Phase 3 $F_{\text{comb } 3 \ i}$
1	1813 N	2413 N	2188 N
2	96 N	3120 N	471 N
3	2303 N	6828 N	2678 N
4	471 N	3120 N	96 N

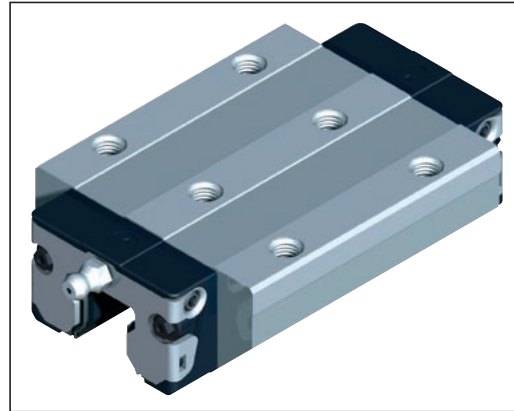
3 Profiled rail systems

3.1 Principles

3.1.5 Calculations

Step 4:
Take the preload into account

- The following ball rail system was selected:
- Size 30, runner block flanged, long, standard height (FLS)
 - Preload class C2 (corresponds to a preload of 8% of dynamic load capacity C, $X_{pr} = 0.08$)
 - Dynamic load capacity $C = 40,000 \text{ N}$



Ball runner block flanged, long, standard height (FLS)

Calculate the preload force F_{pr} for the runner block according to formula (3-37):

$$F_{pr} = X_{pr} \cdot C = 0.08 \cdot 40000 \text{ N} = 3200 \text{ N}$$

To determine whether the preload will have an effect on the service life, the lift-off force F_{lim} must be calculated using formula (3-38):

$$F_{lim} = 2.8 \cdot F_{pr} = 2.8 \cdot 3200 \text{ N} = 8960 \text{ N}$$

Since the combined equivalent load on bearing is smaller than the lift-off force for all calculated runner blocks i and phases n , the preload must be taken into account in the calculations.

$$F_{comb ni} < F_{lim} \text{ for all loads on bearing}$$

For the rest of the calculation procedure, the effective equivalent load on bearing must first be calculated for all runner blocks and all phases using formula (3-40).

Calculate phases 1, 2 and 3 for runner blocks 1 to 4:

$$F_{eff 1 1} = \left(\frac{F_{comb 1 1}}{2.8 \cdot F_{pr}} + 1 \right)^{\frac{3}{2}} \cdot F_{pr} = \left(\frac{1813 \text{ N}}{2.8 \cdot 3200 \text{ N}} + 1 \right)^{\frac{3}{2}} \cdot 3200 \text{ N} = 4219 \text{ N}$$

$$F_{eff 1 2} = \left(\frac{F_{comb 1 2}}{2.8 \cdot F_{pr}} + 1 \right)^{\frac{3}{2}} \cdot F_{pr} = \left(\frac{96 \text{ N}}{2.8 \cdot 3200 \text{ N}} + 1 \right)^{\frac{3}{2}} \cdot 3200 \text{ N} = 3252 \text{ N}$$

...

$$F_{eff 3 4} = \left(\frac{F_{comb 3 4}}{2.8 \cdot F_{pr}} + 1 \right)^{\frac{3}{2}} \cdot F_{pr} = \left(\frac{96 \text{ N}}{2.8 \cdot 3200 \text{ N}} + 1 \right)^{\frac{3}{2}} \cdot 3200 \text{ N} = 3252 \text{ N}$$

Intermediate results:
Effective equivalent load on bearing per runner block and phase

Runner block i	Phase 1		Phase 2		Phase 3	
	$F_{eff 1 i}$		$F_{eff 2 i}$		$F_{eff 3 i}$	
1	4219 N		4576 N		4441 N	
2	3252 N		5009 N		3456 N	
3	4510 N		7485 N		4737 N	
4	3456 N		5009 N		3252 N	

3 Profiled rail systems

3.1 Principles

3.1.5 Calculations

**Step 5:
Calculate the
equivalent dynamic
load on the bearing**

For the nominal life calculation, the equivalent dynamic load on the bearing F_m is calculated according to the discrete travel steps q_{sn} using formula 3-43).

Discrete travel steps from the dynamic cycle:

Phase n	Discrete travel step q_{sn}
1	12.5%
2	75.0%
3	12.5%

$$F_{m i} = \sqrt[3]{(F_{\text{eff } 1 i})^3 \cdot \frac{q_{s1}}{100\%} + (F_{\text{eff } 2 i})^3 \cdot \frac{q_{s2}}{100\%} + (F_{\text{eff } 3 i})^3 \cdot \frac{q_{s3}}{100\%}}$$

$$F_{m 1} = \sqrt[3]{(4219 \text{ N})^3 \cdot \frac{12.5\%}{100\%} + (4576 \text{ N})^3 \cdot \frac{75\%}{100\%} + (4441 \text{ N})^3 \cdot \frac{12.5\%}{100\%}} = 4518 \text{ N}$$

...

$$F_{m 4} = \sqrt[3]{(3456 \text{ N})^3 \cdot \frac{12.5\%}{100\%} + (5009 \text{ N})^3 \cdot \frac{75\%}{100\%} + (3252 \text{ N})^3 \cdot \frac{12.5\%}{100\%}} = 4698 \text{ N}$$

**Intermediate results:
Equivalent dynamic
load on bearing**

Runner block i	Equivalent dynamic load on bearing $F_{m i}$
1	4518 N
2	4698 N
3	6974 N
4	4698 N

This shows that runner block 3 is the most heavily loaded one. The nominal life calculation therefore concentrates on runner block 3.

3 Profiled rail systems

3.1 Principles

3.1.5 Calculations

**Step 6:
Calculate the
nominal life**

The service life must now be checked to see whether it will meet the required 10,000 operating hours.

According to formula (3-48), the nominal life in meters for a size 30 ball runner block with a dynamic load capacity of 40,000 N is:

$$L_i = \left(\frac{C}{F_{mi}} \right)^3 \cdot 10^5 \text{ m}$$

$$L_3 = \left(\frac{C}{F_{m3}} \right)^3 \cdot 10^5 \text{ m} = \left(\frac{40000 \text{ N}}{6974 \text{ N}} \right)^3 \cdot 10^5 \text{ m} = 18868000 \text{ m}$$

Discrete time steps and average speeds in the individual phases:

Phase n	Discrete time step	Average speed
	q_{tn}	v_n
1	20%	12 m/min
2	60%	24 m/min
3	20%	12 m/min

Calculate the average speed over the entire cycle using formula (3-52):

$$v_m = \frac{|v_1| \cdot q_{t1} + |v_2| \cdot q_{t2} + \dots + |v_n| \cdot q_{tn}}{100\%}$$

$$v_m = \frac{12 \frac{\text{m}}{\text{min}} \cdot 20\% + 24 \frac{\text{m}}{\text{min}} \cdot 60\% + 12 \frac{\text{m}}{\text{min}} \cdot 20\%}{100\%} = 19.2 \frac{\text{m}}{\text{min}}$$

Because of the varying speed, the nominal life in operating hours is calculated using formula (3-53):

$$L_{hi} = \frac{L_i}{60 \cdot v_m}$$

$$L_{h3} = \frac{L_3}{60 \cdot v_m} = \frac{18868000 \text{ m}}{60 \frac{\text{min}}{\text{h}} \cdot 19.2 \frac{\text{m}}{\text{min}}} = 16379 \text{ h}$$

Results: Nominal life

Runner block i	Nominal life L_i	Nominal life L_{hi}
1	69397000 m	60241 h
2	61722000 m	53578 h
3	18868000 m	16379 h
4	61722000 m	53578 h

For the most heavily loaded runner block 3, the nominal life is 16,379 operating hours. The service life requirement of 10,000 operating hours is therefore satisfied.

3 Profiled rail systems

3.1 Principles

3.1.5 Calculations

Step 7:
Calculate the equivalent static load on the bearing

The maximum load F_{\max} occurs in phase 2 at runner block 3. In order to calculate the static load safety factor, the load due to the preload must also be taken into account.

$$F_{\max} = F_{\text{eff } 2 \ 3} = 7485 \text{ N}$$

Step 8:
Calculate the static load safety factor

According to formula (3-60), the static load safety factor for $C_0 = 57,800 \text{ N}$ is:

$$S_0 = \frac{C_0}{F_{\max}}$$
$$S_0 = \frac{57800 \text{ N}}{7485 \text{ N}} = 7.72$$

3 Profiled rail systems

3.1 Principles

3.1.6 Defining the peripherals

An extensive range of standard parts, special add-ons and accessories is available for profiled rail systems, allowing them to be adapted to each specific application. This offer includes recommendations for designing the lubrication system and lubrication intervals, as well as accessories to simplify installation and maintenance of the system. There are special seals and seal kits to ensure consistently good performance in environmental conditions which might otherwise shorten

the life of the equipment. For humid environments, profiled rail systems can be supplied in special corrosion-resistant versions. In addition to their main purpose, the guides can also be equipped with items such as clamping and braking units, drive units, and measuring systems to extend their range of functionalities.

Rexroth offers the following solutions for enhancing profiled rail guides:

Solutions		Section
Lubrication		3.1.6.1
Sealing		3.1.6.2
Corrosion protection		3.1.6.3
Additional functions	Clamping and braking units	3.1.6.4
	Rack and pinion drives	3.1.6.4
	Integrated measuring system	3.7

Note:

Not all solutions are available for all types and sizes of profiled rail systems. The choice of peripherals should be considered as early as

the product selection stage. The availability of parts/versions and the relevant details can be found in the respective product catalogs.

3.1.6.1 Lubrication

Operating conditions

When selecting the lubricant, the following factors must be considered:

- Loads
- Speed
- Stroke length
- Temperature
- Humidity
- Exposure to metalworking fluids
- Dirt/shavings

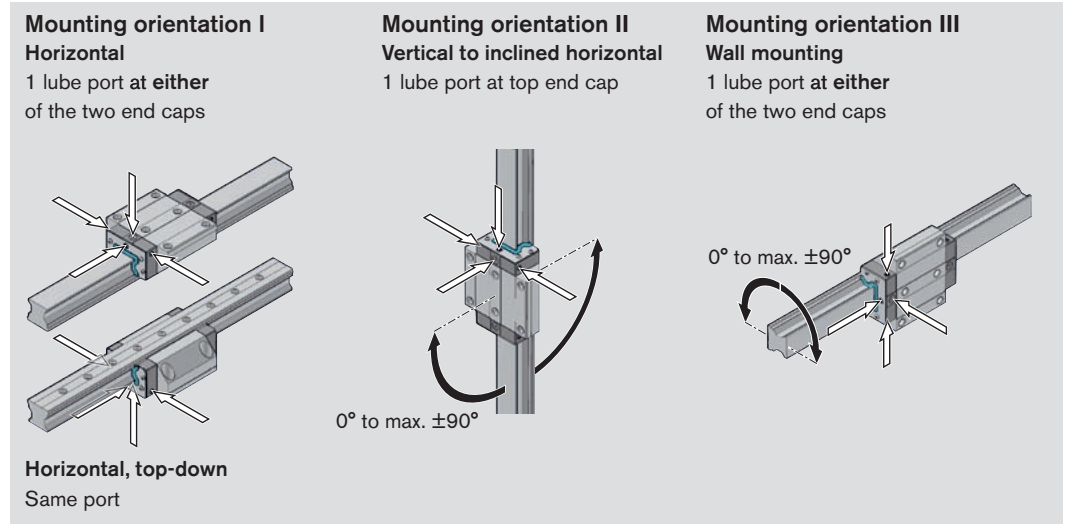
If linear guides are to be used in extreme operating conditions, e.g. with high exposure to metalworking fluids, this must be taken into account when specifying the lubricant and the lubricant quantities. Metalworking fluids that have worked their way into a runner block can wash out the lubricant. This can be avoided by shortening the lubrication cycles.

3 Profiled rail systems

3.1 Principles

3.1.6 Defining the peripherals

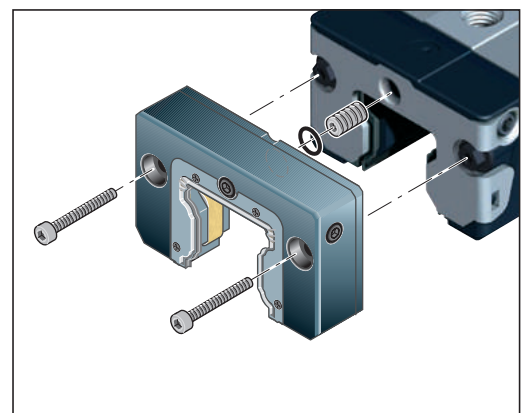
Mounting orientation If oil lubricants are used, the mounting orientation will affect the distribution of the lubricant within the system. The number and location of the lube ports and the lubricant quantities must be specified as appropriate for the chosen mounting orientation. Details can be found in the Rexroth product catalogs.



Information on mounting orientations and lube ports for normal-stroke applications (Examples taken from the roller rail systems product catalog; full details are given in the catalog)

Lubrication intervals The required lubrication intervals vary according to the conditions of use and the lubricant used. Lubrication intervals also depend on the loads applied. Details can be found in the Rexroth product catalogs.

Front lube units Front lube units are fastened to the end faces of the runner block. They supply the rolling contact points and the seals with lubricant. Specially designed lube distribution ducts ensure that the lubricant is applied where it is needed: directly to the raceways and to the guide rail surface. This prolongs the relubrication intervals. The runner block should be pre-lubricated with grease. The front lube units are filled with oil. Under normal loads, they allow travel distances of up to 10,000 km without relubrication. The maximum operating temperature for front lube units is 60°C .



Front lube unit for ball rail systems

3 Profiled rail systems

3.1 Principles

3.1.6 Defining the peripherals

3.1.6.2 Sealing

The rolling elements and the raceways in the runner block must be protected from foreign particles. Dirt or shavings can considerably shorten the life of the guide. Rexroth offers a broad range

of accessories in addition to the standard sealing options. Users can therefore put together the sealing system that best suits their needs.

Version	Seal types	Degree of contamination
Integrated seals	Low-friction seals	
	Standard (universal) seals	
	Reinforced seals	
Additional external seals	End seals (standard feature in roller rail systems)	
	Viton seals	
	Seal kit	
	Bellows	

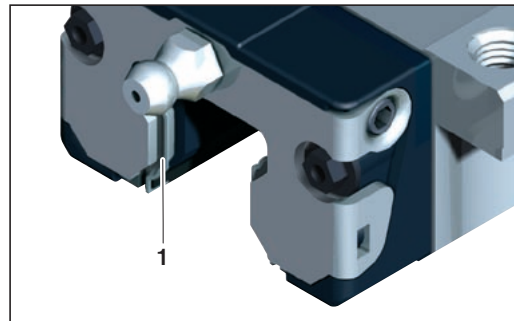
Wiper seals

Wiper seals (1) are seals integrated in the end caps. They are a standard feature in all runner blocks. Wiper seals come in different types: standard (universal) seals, low-friction seals, and reinforced (heavy-duty) seals.

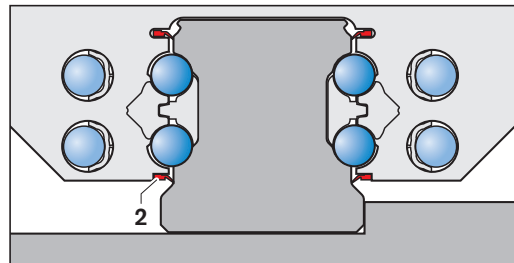
Universal seals are the standard option for Rexroth runner blocks. Designed for applications in normal operating environments, they offer good sealing action and low resistance to movement.

Low-friction seals are the solution for applications requiring especially smooth running in environments with little or no contamination, where sealing action is of secondary importance.

Reinforced seals were designed for operation in extreme environmental conditions. With their excellent sealing action, they provide highly effective protection.



Integrated wiper seals in a ball runner block



Integrated side seals in a ball runner block

Side seals

Most of the profiled rail systems come standard with two or four side seals (2), providing lateral protection (perpendicular to the direction of travel) for the runner block internals.

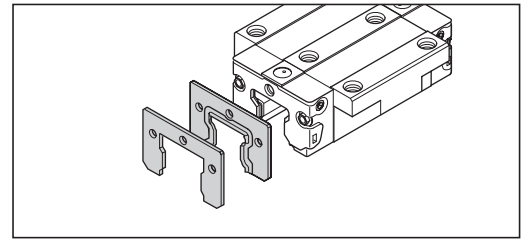
3 Profiled rail systems

3.1 Principles

3.1.6 Defining the peripherals

End seals

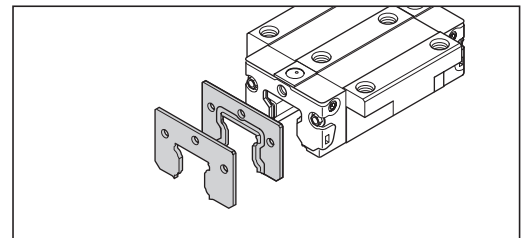
External end seals provide effective protection for the runner block, preventing dirt or liquids from working their way in. End seals are attached to the end face of the runner block. Depending on the type of profiled rail system, these seals may be of one-piece or two-piece design. They are standard accessories in roller rail systems.



Two-piece end seal

Viton seals

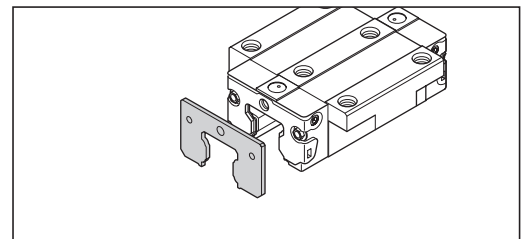
Viton seals are an additional option for external sealing. They offer even better sealing action than the end seals. However, the Viton material causes significantly higher friction. Viton seals are chemically resistant and can withstand high temperatures.



Viton seal

Metal scrapers

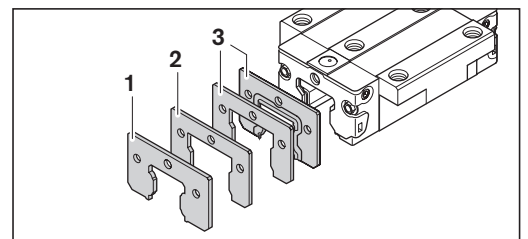
Metal scrapers provide added protection against coarse particles. Made from stainless spring steel, they are designed to ensure effective removal of shavings and coarse contamination.



Metal scraper

Seal kit

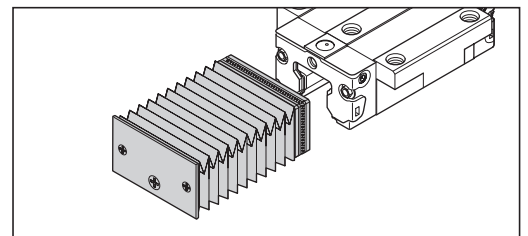
The seal kit is intended for applications involving a combination of coarse and fine dirt and exposure to fluids. It consists of a metal scraper (1), a reinforcing plate (2) and a two-piece end seal (3).



Seal kit

Bellows

Bellows come in a variety of designs. They can be delivered with or without a lubrication plate. Heat-resistant bellows are also available. These are metallized on one side, making them resistant to individual sparks, welding splatter or hot shavings. They are designed for an operating temperature of 100°C, but can withstand brief temperature peaks of up to 200°C.



Bellows

3 Profiled rail systems

3.1 Principles

3.1.6 Defining the peripherals

3.1.6.3 Corrosion protection

The corrosion protection options for the profiled rail systems range are listed below.

The product catalogs indicate which type of corrosion protection is available for each particular guide.

Corrosion protection type	Description
Aluminum	Runner block made from aluminum
Resist NR	Runner block made from corrosion-resistant steel
Resist NR II	Runner block and guide rail made from corrosion-resistant steel
Resist CR	Runner block and guide rail made from hard chrome plated steel

Aluminum

In this version, the runner block body is made of aluminum. The runner blocks have the same dynamic load capacity (100% C) as the standard steel version, but their load-bearing capability is

reduced due to the lower strength of aluminum. Rexroth therefore indicates the maximum permissible load F_{max} for these designs. Aluminum runner blocks offer weight savings up to 60%.

Resist NR

The runner block body is made from corrosion-resistant steel. All other steel parts are identical to those used in standard runner blocks. This design

provides high corrosion resistance with the same dynamic and static load capacities (100% C and 100% C_0) as the standard version.

Resist NR II

In this version, all steel parts are made from corrosion-resistant steel, thus offering especially good protection against corrosion. The disadvantage of this design is that the load capacities are reduced (65% C) relative to the standard version because the runner block and guide rail raceways and the rolling elements are not made from anti-friction bearing steel. Profiled rail systems in Resist NR II are specifically intended for use in applications involving aqueous media, very dilute acids, alkalis or salt solutions. They are particu-

larly suitable for environments with a relative humidity of over 70% and temperatures above 30°C, such as cleaning lines, surface treatment and pickling lines, steam degreasing equipment, and refrigeration systems. Since they have built-in corrosion protection, they are also ideal for use in cleanrooms and under vacuum. Other application areas include printed circuit board assembly, and the pharmaceuticals and food industries.

Resist CR

In this version, both the runner block body and the guide rail are hard chrome plated with a matt silver finish. Their corrosion resistance is correspondingly high. Resist CR rail guides offer the same dynamic and static load capacities (100% C and 100% C_0) as the standard versions.

The thickness of the hard chrome layer on the runner blocks and guide rails increases the outside dimensions, resulting in different tolerances for the accuracy classes.

3 Profiled rail systems

3.1 Principles

3.1.6 Defining the peripherals

3.1.6.4 Additional functions

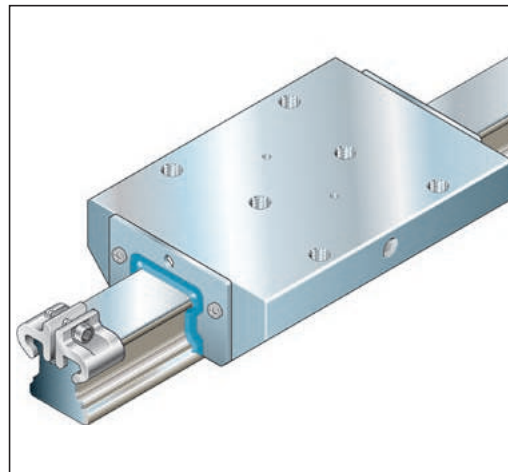
Clamping and braking units

Among its range of accessories, Rexroth also offers clamping and braking units. The clamping units serve to prevent linear guides from moving when they are at rest. The braking units were designed to bring moving linear guides to a standstill and keep them stationary during rest phases. These elements have no guidance function.

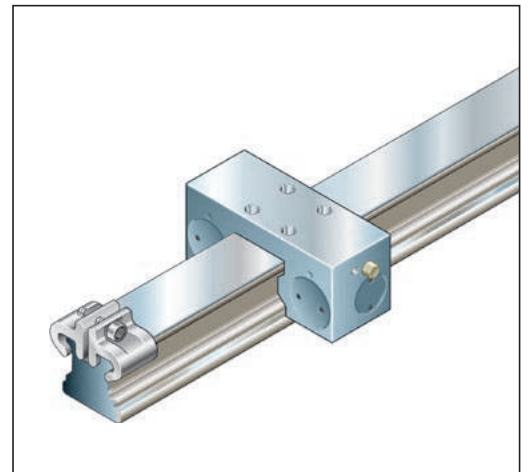
Note:
The braking units are not safety brakes!

Clamping and braking units are available in the following versions:

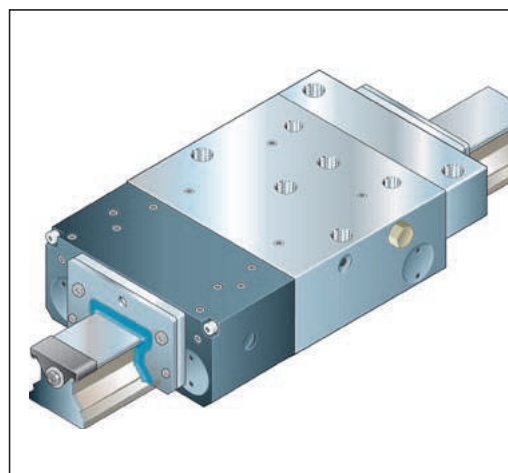
- Hydraulic clamping units
- Pneumatic clamping units
- Electrical clamping units
- Hydraulic clamping and braking units
- Pneumatic clamping and braking units
- Manual clamping units



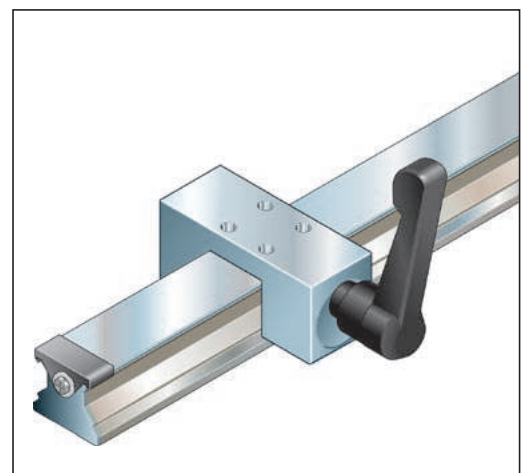
Hydraulic clamping unit
on a ball rail



Pneumatic clamping unit
on a ball rail



Pneumatic clamping and braking unit
on a roller rail



Manual clamping unit
on a roller rail

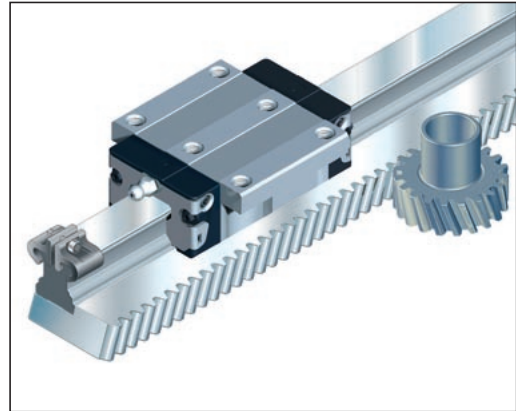
3 Profiled rail systems

3.1 Principles

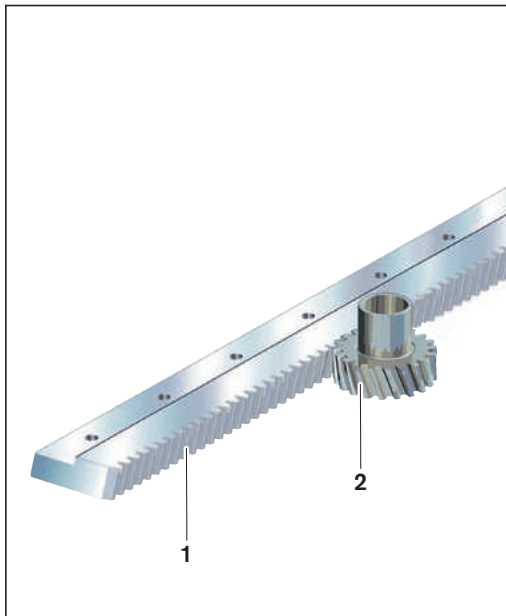
3.1.6 Defining the peripherals

Rack and pinion drives

Gear racks and pinions are space-saving solutions for driving linear motion guides. To ensure long life, Rexroth uses proven high grade steel materials with inductively hardened tooting for these components. In addition to the purely mechanical elements, all attachments such as gear reducers, motors and controllers are also available. The helical tooting allows high forces to be transmitted within a small space and with low noise generation.

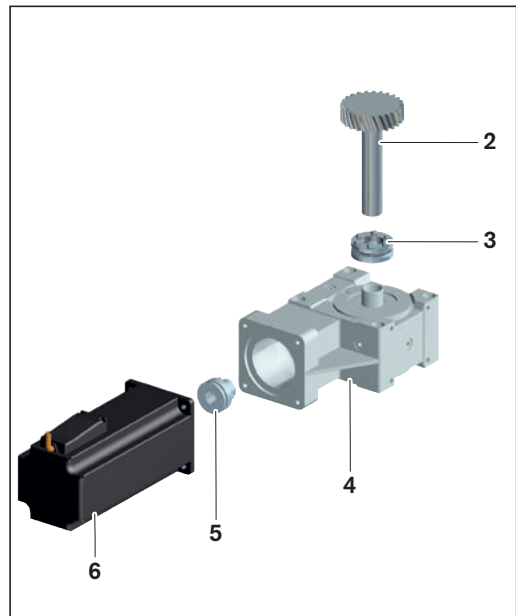


Ball rail system with rack and pinion drive



Rack and pinion drive components

- 1 Gear rack
- 2 Pinion (various gear ratios possible)
- 3 Shrink-fit disk



- 4 Gear reducer
- 5 Coupling
- 6 Motor

The gear rack can be lined up with the rail and bolted directly with it to the machine bed. This significantly reduces the mounting effort. Rack and pinion drives must be adequately lubricated. Permanent lubrication is therefore recommended, preferably by means of a felt wheel connected to a central lubrication system.

Advantages of rack and pinion drives:

- Long guideway lengths
- High travel speeds
- In multiple-carriage applications, each axis can be moved separately
- Reduced mass in Z-axis, by designing applications with traveling rails (stationary motor and runner block, traveling rail and gear rack)
- High rigidity, especially over long strokes

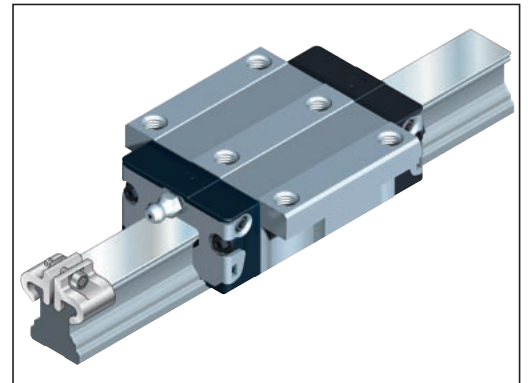
3 Profiled rail systems

3.2 Ball rail systems

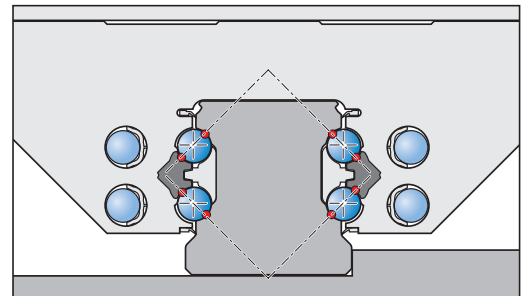
3.2.1 System characteristics

A ball rail system (BRS) consists of a guide rail and runner blocks. The BRS has 4 rows of balls in an O-arrangement with a contact angle of 45° . The balls are in 2-point contact with the rail and the runner block (see illustration).

The guide rail has four running tracks along which one or more runner blocks can travel. The guide rail can be bolted into place from above or below. V-guide rails are pressed into the mounting base. Depending on the requirements, the runner block has either through-bores or threaded holes for direct mounting to the adjoining structure. Ball runner blocks are available in various sizes, designs and preload classes, thus covering a wide range of applications. The ball rail system is the most versatile of all the profiled rail systems. It is offered in many different versions (see section 3.2.3.1).



Ball rail system BRS



Four rows of balls in an O-arrangement

Features

- High load capacities in all four major planes of load application
- High system rigidity
- Limitless interchangeability due to precision manufacturing
- Smooth running performance
- Zero-clearance movement
- Excellent high-speed characteristics
- Easy-to-achieve precision
- Very good travel accuracy with HP series runner blocks
- Long-term zero maintenance
- Minimum quantity lubrication system with integrated reservoir for oil lubrication (depending on version)
- Lube ports on all sides
- Optional ball chain
- Broad range of accessories for industry-specific solutions (seals, wipers/scrapers)
- High dynamic characteristics with high-speed runner blocks
- Optimum installation error compensation with super runner block
- Integrated, inductive and wear-free measuring system as an option
- Runner blocks in rust- and acid-resistant steel to EN 10088 available
- Up to 60% weight saving with aluminum runner block

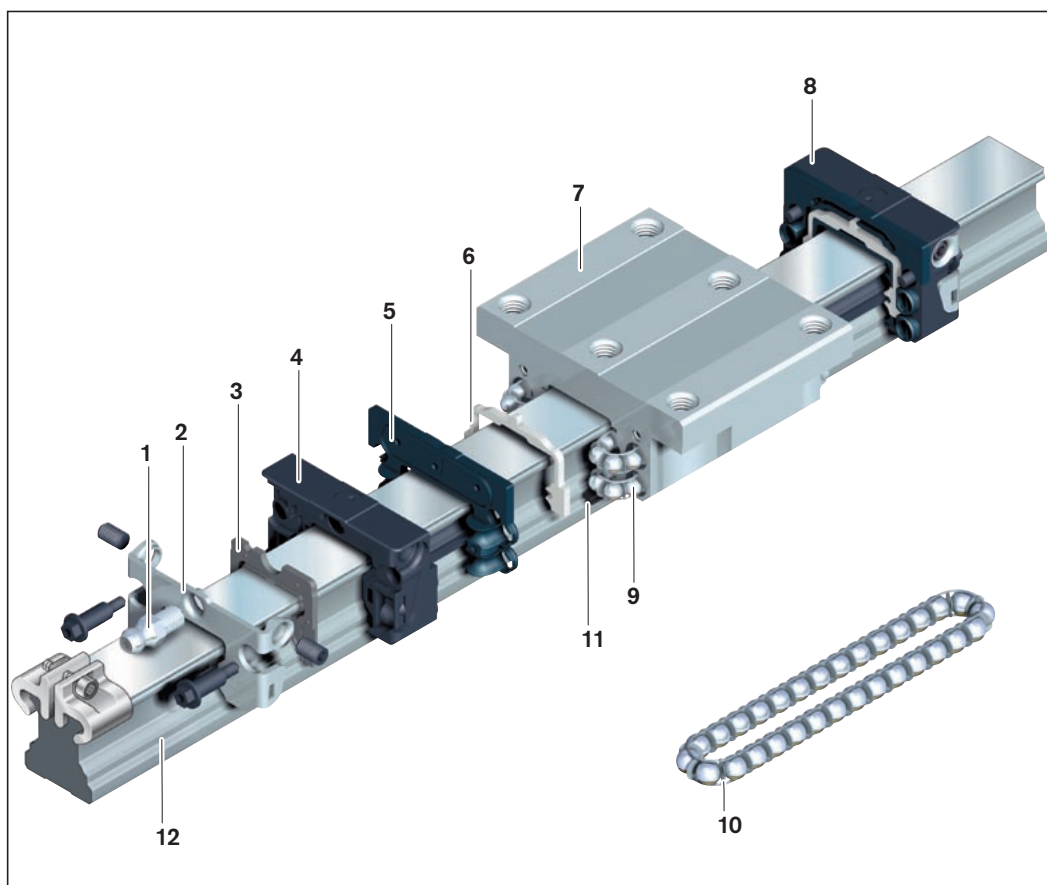
Product data

Product data	Values	
Speed	v_{\max}	Up to 10 m/s
Acceleration	a_{\max}	Up to 500 m/s ²
Temperature resistance	t_{\max}	100°C brief peaks, 80°C in continuous operation
Preload classes	4	C0, C1, C2, C3
Rigidity		Sample rigidity charts in product catalog, charts for other types on request
Accuracy classes	6	N, H, P, XP, SP, UP
Sizes	11	15, 20, 25, 30, 35, 45, 55, 65 20/40, 25/70, 35/90

3 Profiled rail systems

3.2 Ball rail systems

3.2.2 Structural design



Structural design as implemented in a latest-generation ball rail system

- 1 Lube port (lube nipple)
- 2 Threaded plate
- 3 Sealing plate (wiper seal)
- 4 Ball guide (part of end cap)
- 5 Recirculation plate (part of end cap)
- 6 Lubrication insert
- 7 Runner block body
- 8 End cap
- 9 Balls (rolling elements)
- 10 Ball chain (optional)
- 11 Side seal
- 12 Guide rail

**Runner blocks,
general description**

Ball runner blocks are pre-lubricated before shipment. The customer is free to decide which lubricant to use in his application. Either grease or oil lubrication is possible. Corrosion-protected versions are available in Resist CR, Resist NR and Resist NR II.

For additional sealing of the runner block there are end seals, Viton seals, metal scrapers, seal kits and bellows.

3 Profiled rail systems

3.2 Ball rail systems

3.2.2 Structural design

Runner block body

Depending on the version, the runner block body can be made from heat-treated steel or anti-friction bearing steel. For special applications, especially for use in industrial robots, the body is made from aluminum. The aluminum version offers weight savings up to 60% compared to the steel version. Each runner block has a lateral reference edge (1). This edge mates with the adjoining structure. It permits precise alignment during installation and serves to transmit side loads.

Steel inserts

The runner blocks have two hardened steel inserts (2) made from anti-friction bearing steel. These inserts transmit the load from the runner block body to the balls.

Recirculation sleeves

Each of the four ball return bores in the runner block body is lined with a sleeve (3). This sleeve ensures good, low-friction recirculation of the balls inside the runner block. It also acts as a guide for the optional ball chain.

End cap

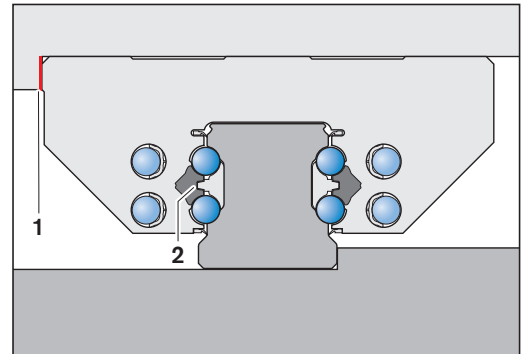
The end cap (4) consists of the recirculation plate (6), the lubrication insert (7), the ball guide (5), the sealing plate, and the threaded plate.

Recirculation plate

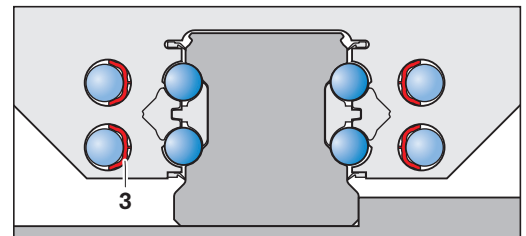
The recirculation plate (6) has specially designed lube ducts which conduct the lubricant directly to the lubrication insert, thus ensuring optimal lubrication results. It is thanks to this particular feature that long maintenance intervals or even lubrication for life can be achieved. The recirculation plate also picks up and redirects the balls inside the runner blocks.

Lubrication insert

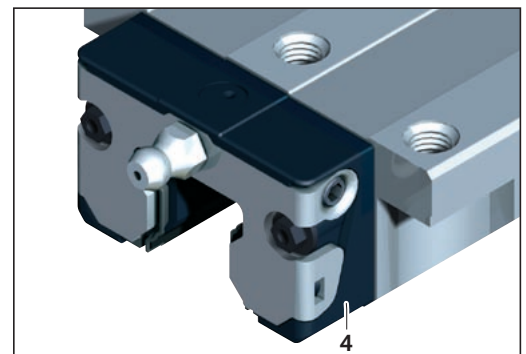
The lubrication insert (7) is made from open-pored polyurethane foam. This foam soaks up the lubricant and releases it to the passing balls. The lubrication insert has been designed to allow lubrication with either oil or grease.



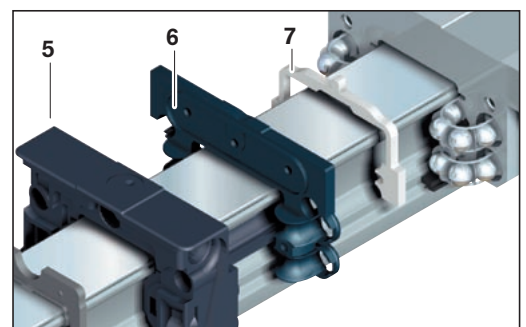
Steel inserts and reference edge in a runner block



Recirculation sleeves



Complete end cap



Recirculation plate with lubrication insert

3 Profiled rail systems

3.2 Ball rail systems

3.2.2 Structural design

Ball guide

The ball guide is fixed in place by the recirculation plate. The balls are redirected in the space between these two parts. The ball guide also serves to retain the balls in the load-bearing raceway of the runner block when it is not mounted on the rail.

Lube ports

The ball guide also contains lube ports. The lube nipples or fittings of a central lubrication system can be inserted into these lube ports. The ports are located on the end face and at both sides. This allows lubrication from any of three directions without the need for an adapter. Lubrication from the top is also possible, by opening a pre-drilled hole. The hole can be punched open using a heated, pointed metal tool to allow lubrication through the machine table. An O-ring seals the interface to the machine table. High-profile runner blocks require an adapter to compensate for the height difference between the end cap and the runner block body.

Threaded plate

The threaded plate (1) has two functions: it accommodates lube nipples and protects the end cap assembly. It is made from stainless steel.

Sealing plate

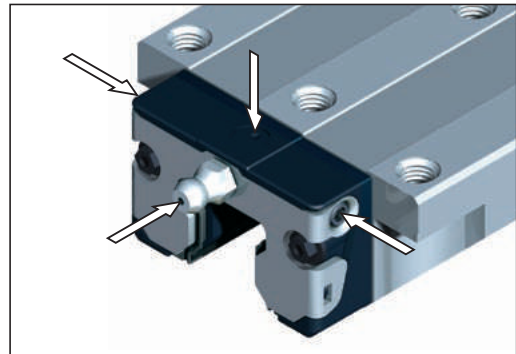
The sealing plate (2) on the end face protects internal runner block components from dirt particles, shavings and liquids. It also prevents the lubricant from being dragged out. Optimized sealing lip geometry results in minimal friction. Sealing plates are available with a standard seal, low-friction seal, or a reinforced seal.

Side seals

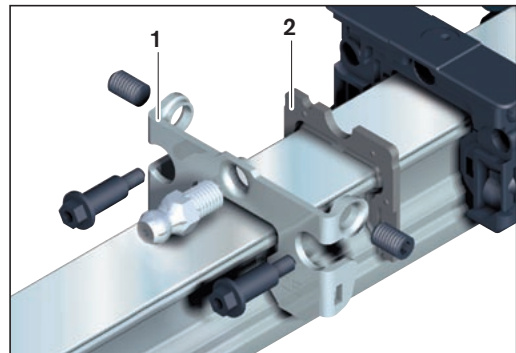
Lateral sealing strips provide additional protection, keeping dirt and shavings out of the load-bearing zones. Each runner block has four of these side seals (3).

Balls

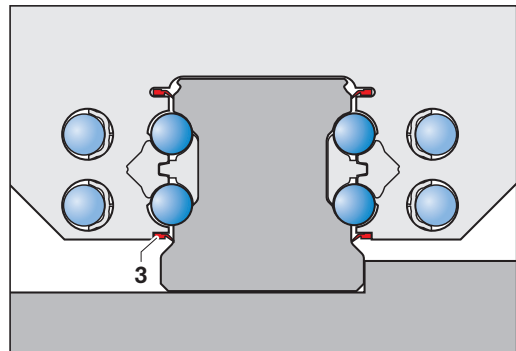
The rolling elements are balls. Normally, these are made from anti-friction bearing steel, grade 100Cr6. Stainless steel balls are used for runner blocks that will be operating in extremely hostile environments requiring corrosion-resistant elements. High-speed runner blocks have special ceramic balls. Because of their lightweight design, these balls deliver excellent dynamic performance. Ceramic balls are also good electrical insulators.



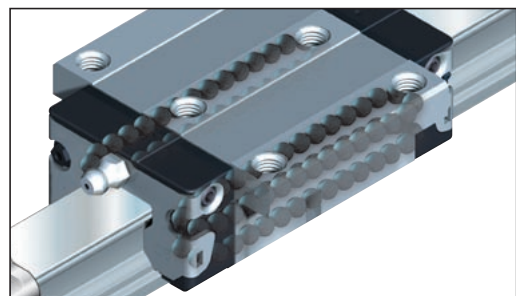
Lube ports in the end cap



Sealing plate and threaded plate



Integrated side seals in a ball runner block



Ceramic balls for high-speed runner blocks

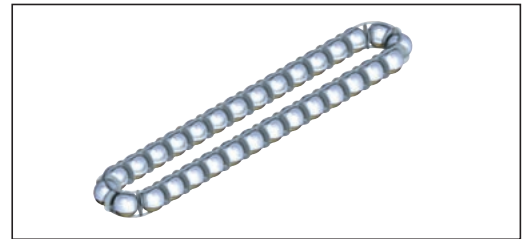
3 Profiled rail systems

3.2 Ball rail systems

3.2.2 Structural design

Ball chain

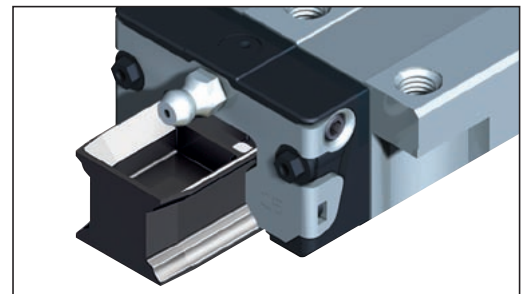
Runner blocks can also be equipped with a ball chain. The ball chain prevents the balls from bumping into each other and ensures smoother travel. This reduces the noise level. Runner blocks with ball chains have fewer load-bearing balls, which may result in lower load capacities.



Ball chain

Transport and mounting arbor

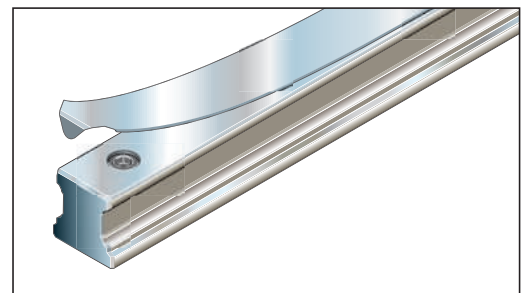
Ball runner blocks are mounted on an arbor for shipment. This arbor protects the balls from damage during transport and makes it easier to mount the runner block to and remove it from the guide rail.



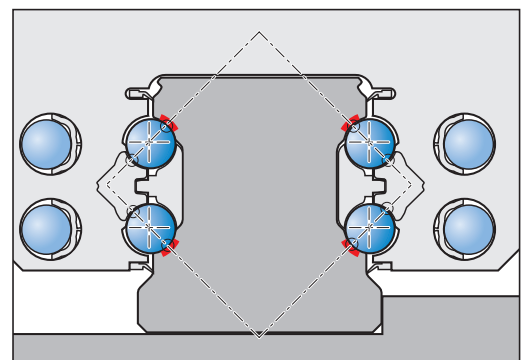
Transport and mounting arbor

Guide rail

The guide rail is made from heat-treated steel. This steel was specially designed to meet linear motion requirements and therefore offers optimal system characteristics. The four ground running tracks have a circular-arc profile with conformity. This geometry ensures ideal running performance and can also compensate to a certain extent for misalignments. The running tracks are inductively hardened and precision-ground. Rexroth guide rails are also available in hard chrome plated (Resist CR) or in corrosion-resistant steel (Resist NR II) versions. These rails can be used in environments with aggressive media, such as dilute acids, alkalis or salt solutions. Depending on the size, one-piece rails can be delivered in lengths up to 6 m. If longer lengths are required, several rails can be fitted end to end to produce a composite rail. Guide rails can be bolted into place from above or below. V-guide rails are installed by pressing them into the mounting base.



Guide rail for mounting from above (with cover strip to seal off the mounting holes)



Guide rail with four running tracks (circular-arc profile with conformity)














3 Profiled rail systems

3.2 Ball rail systems

3.2.3 Product selection guide

3.2.3.1 Versions

Rexroth offers many different designs and versions to meet the needs of a broad range of applications:

Runner block formats	Width	F					S							B
	Length	N	N	L	K	K	N	N	N	L	L	K	K	N
	Height	S	N	S	S	N	S	N	H	S	H	S	N	N
	Code	FNS	FNN	FLS	FKS	FKN	SNS	SNN	SNH	SLS	SLH	SKS	SKN	BNN
														

- Runner blocks with or without ball chain
- Super runner blocks with self-alignment capability
- Aluminum runner block
- High-speed runner blocks
- High-precision ball runner blocks
- Corrosion-resistant ball guide rails

3.2.3.2 Application areas

Ball rail systems are used in a wide variety of industries and applications. Typical examples include:

Industry sector	Applications
Metal-cutting machine tools	<ul style="list-style-type: none"> ■ Machining centers ■ Lathes and turning machines ■ Drilling machines ■ Milling machines ■ Grinding machines ■ Nibbling machines ■ Planing machines ■ Electrical discharge machines ■ Laser/light/photo beam machine tools
Assembly/handling technology and industrial robots	<ul style="list-style-type: none"> ■ Assembly equipment ■ Assembly robots ■ Multi-purpose industrial robots ■ Gripping and clamping equipment
Woodworking and wood processing machines	<ul style="list-style-type: none"> ■ Belt saws ■ Circular saws ■ Planing machines ■ Drilling machines ■ Mortising machines ■ Sanding machines ■ Slitters
Rubber and plastics processing machinery	<ul style="list-style-type: none"> ■ Calendering machines ■ Rolling mills ■ Extruders ■ Blow molding machines ■ Injection molding machines
Food industry	<ul style="list-style-type: none"> ■ Filling machines ■ Molding machines ■ Confectionary technology
Printing and paper industry	<ul style="list-style-type: none"> ■ Paper and pulp machines ■ Cutters for paper and cellulose ■ Packaging machines ■ Winders/rewinders ■ Printing machines ■ Paper converting machines
Automotive industry	<ul style="list-style-type: none"> ■ Car production lines ■ Welding systems
Forming and stamping machine tools	<ul style="list-style-type: none"> ■ Bending machines ■ Straightening/leveling machines ■ Presses ■ Wire bending machines

3 Profiled rail systems

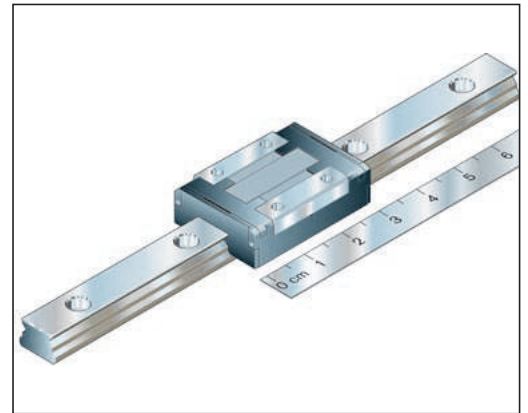
3.3 Miniature ball rail systems

3.3.1 System characteristics

There is a growing demand for miniaturization. This is driven by various factors:

- Complex parts are being redesigned with ever smaller dimensions
- Need for compact and highly precise equipment to perform pick-and-place operations within a small space
- Lower masses of moved parts mean lower moment loads and mass moments of inertia
- Smaller installation spaces to make room for new technologies

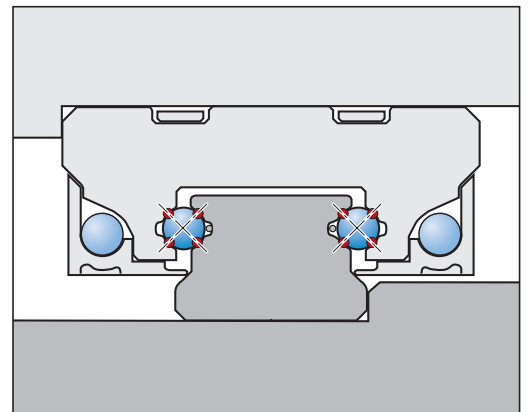
Rexroth has responded to this trend by developing miniature ball rail systems. These systems have two rows of balls with a contact angle of 45°. The rolling elements have 4-point contact.



Miniature ball rail system

Features

- Extremely compact design with high load-bearing capability
- Same load capacities in all four major planes of load application
- High load capacities in all load directions, including moments about all axes, due to the use of largest possible ball sizes
- Smooth running thanks to optimized ball recirculation and guidance
- Low-friction seals ensure low friction despite 4-point contact
- Limitless interchangeability due to precision manufacturing
- Zero-maintenance for a travel life of at least 5,000 km at:
 - $F < 10\% C$
 - $v_m = 0.65 \text{ m/s}$
 - Stroke = 90 mm
 - Low-friction seals
- Cleanroom certification (class 10 to US Fed. Std. 209E)
- High permissible travel speed and acceleration
- Easy mounting due to ball retention



2 rows of balls with 4-point contact

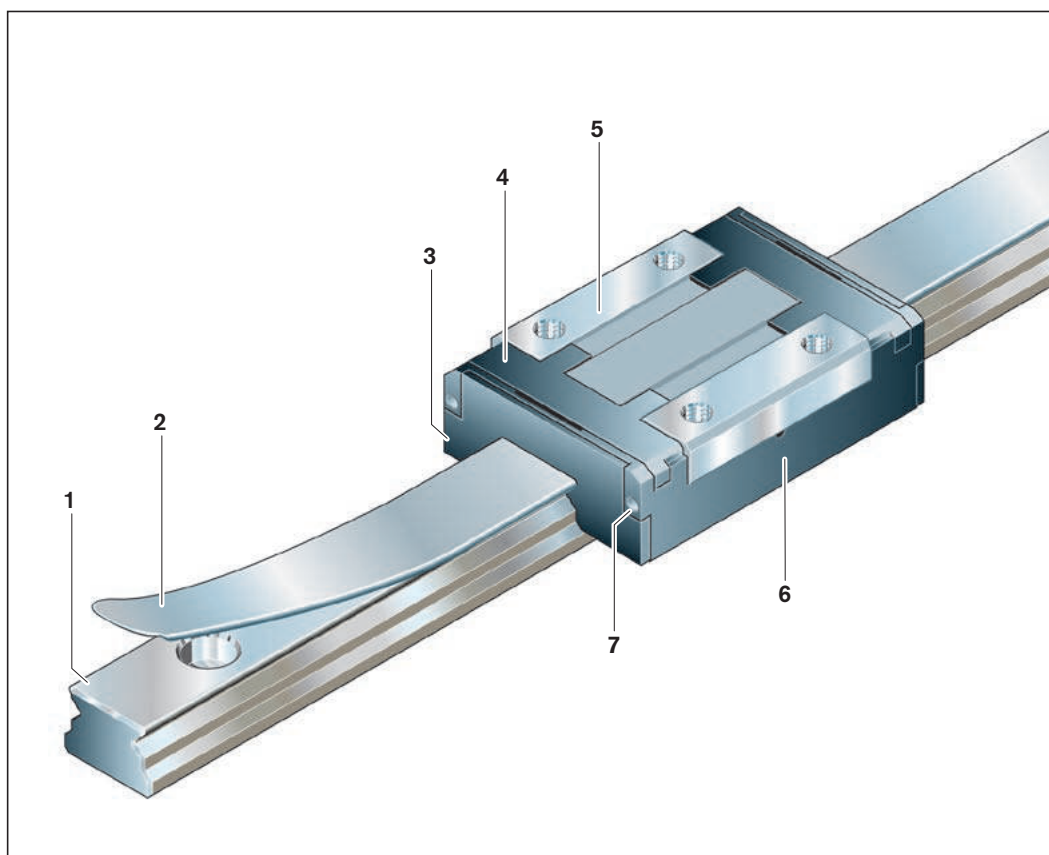
Product data

Product data	Values	
Speed	v_{\max}	Up to 5 m/s
Acceleration	a_{\max}	Up to 250 m/s ²
Temperature resistance	t_{\max}	100°C brief peaks, 80°C in continuous operation
Preload classes	2	Clearance, preload
Rigidity		Rigidity charts for lift-off and down loads in product catalog
Accuracy classes	3	N, H, P
Sizes	8	7, 9, 12, 15, 20 9B, 12B, 15B

3 Profiled rail systems

3.3 Miniature ball rail systems

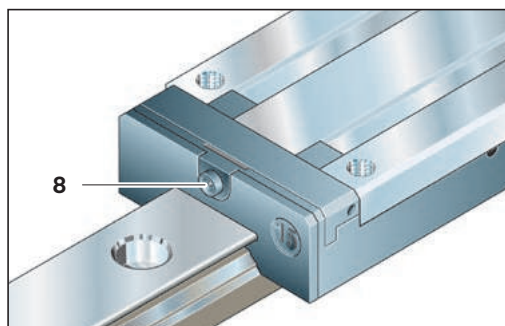
3.3.2 Structural design



Miniature ball rail system, size 12

- | | |
|---------------------------|---------------------------|
| 1 Guide rail | 5 Runner block body |
| 2 Cover strip (rail seal) | 6 Housing |
| 3 Wiper seal | 7 Lube holes |
| 4 Ball recirculation | 8 Funnel-type lube nipple |

All steel parts in miniature ball rail systems are made from corrosion-resistant steel per ISO 683-17/EN 10088. This material assures resistance to aggressive media and preserves the appearance of the system throughout its useful life.



Miniature runner block, size 15, with funnel-type lube nipple

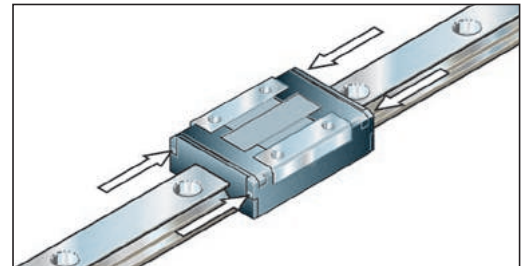
3 Profiled rail systems

3.3 Miniature ball rail systems

3.3.2 Structural design

**Runner blocks,
general description**

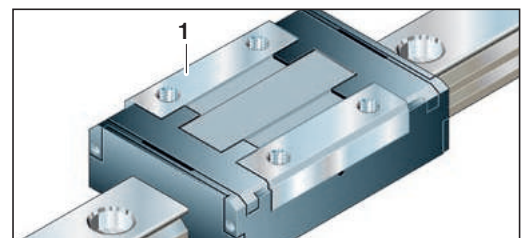
The standard versions of these runner blocks have dimensions per DIN 645-2. Wide and long versions are also available. The runner blocks can be ordered with or without basic lubrication, thus allowing lubrication with grease or oil, as required. Lubrication holes are provided for in-service lubrication of the runner blocks. From size 15 and up, miniature ball rail systems have an additional lube hole at the side, as well as the lube nipple on the end face.

**Lube holes or
lube nipples**

Lube holes (example: size 12)

Runner block body

The runner block body (1) has four threads at the top for fastening to the adjoining structure. The steel part is hardened throughout and is also corrosion-resistant. Reference edges on both sides facilitate mounting to the surrounding machine structures.



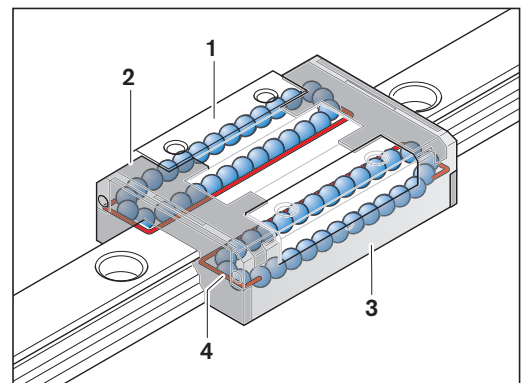
Runner block body

Ball recirculation

The runner block body (1) has a ball recirculation assembly (2) at each end. Each row of balls recirculates outward through a lateral return bore. The ball recirculation assembly consists of a recirculation piece and a sealing plate which contains the lube ducts.

Housing

The recirculating balls are protected and guided by a channel formed by the housing (3) and the body. The housing encloses the body and provides a mounting surface for all attachments.



Ball recirculation assembly with housing and retaining wire

Retaining wire

A retaining wire (4) reliably retains the balls inside the load-bearing raceways of the runner block. This simplifies handling during mounting and removal of the runner block and thus reduces the installation time.

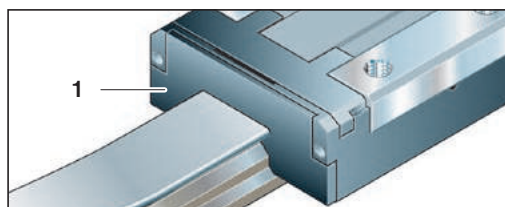
3 Profiled rail systems

3.3 Miniature ball rail systems

3.3.2 Structural design

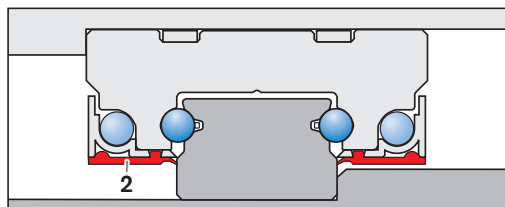
Balls The balls used in the runner block are made from corrosion-resistant and specially hardened steel. This wear-resistant material is ideal for use in miniature ball rail systems.

Wiper seals Miniature runner blocks come standard with low-friction wiper seals (1). It is, however, also possible to install a standard seal with excellent wiping properties.



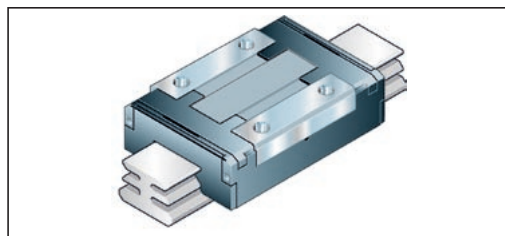
Wiper seal

Side seals Some runner block sizes can be fitted with a standard seal and a two-piece side seal (2) on the underside of the runner block, resulting in a fully sealed design.



Side seals (example: size 15)

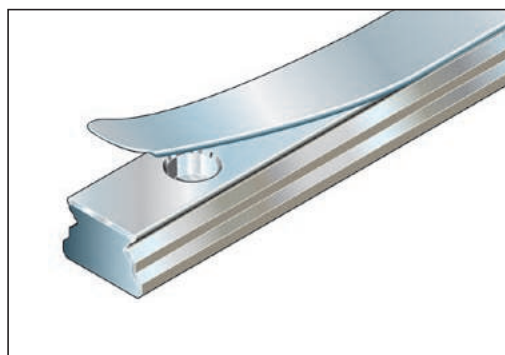
Transport and mounting arbor All miniature runner blocks are delivered mounted on an arbor. This arbor protects the balls from damage during transport and makes it easier to install and remove the runner block.



Miniature runner block, size 12, on mounting arbor

Guide rail Standard miniature guide rails are fully hardened and have through-holes for mounting from above. Mounting hole cover strips are available for size 9 and larger rails. Guide rails can also be supplied in versions for mounting from below. Rexroth offers these guide rails in one-piece lengths up to 2 m. Larger rail lengths are made up of matching rail sections mounted end to end.

Cover strip The cover strip is made from stainless steel and simply clips on to the rail. It prevents dirt from collecting in the rail's mounting holes and interacts with the seals to provide optimal sealing action.






Guide rail for mounting from above (with cover strip to seal off the mounting holes)

3 Profiled rail systems

3.3 Miniature ball rail systems

3.3.3 Product selection guide

3.3.3.1 Versions

Runner block formats			
Width	S		B
Length	N	L	N
Height	S	S	S
Code	SNS	SLS	BNS
			

3.3.3.2 Application areas

Examples of the broad range of applications for miniature ball rail systems:

Industry sector	Applications	
Textile technology	<ul style="list-style-type: none"> ■ Spinning machines ■ Yarn doubling machines ■ Auxiliary equipment 	<ul style="list-style-type: none"> ■ Weaving machines ■ Textile finishing machines ■ Knitting and hosiery machines
Rubber and plastics processing machinery	<ul style="list-style-type: none"> ■ Demolders for plastics processing machines ■ Extruders 	<ul style="list-style-type: none"> ■ Blow molding machines ■ Injection molding machines
Assembly/handling technology and industrial robots	<ul style="list-style-type: none"> ■ Assembly equipment ■ Assembly robots ■ Multi-purpose industrial robots 	<ul style="list-style-type: none"> ■ Gripping and clamping equipment ■ Small parts handling ■ Industrial robots
Medical technology	<ul style="list-style-type: none"> ■ Microscopes ■ Diagnostic equipment 	<ul style="list-style-type: none"> ■ Analytical instruments
Electrical/electronics industry	<ul style="list-style-type: none"> ■ Microelectronics ■ Semiconductor manufacturing 	<ul style="list-style-type: none"> ■ Computer peripherals ■ Automated PCB assembly systems
Food and packaging industries	<ul style="list-style-type: none"> ■ Cleanroom applications ■ Labeling machines 	<ul style="list-style-type: none"> ■ Filling machines ■ Boxing machines
Printing and paper industry	<ul style="list-style-type: none"> ■ Cutters for paper and cellulose 	<ul style="list-style-type: none"> ■ Paper converting machines ■ Bookbinding machines
Precision machine tools	<ul style="list-style-type: none"> ■ Measuring machines 	<ul style="list-style-type: none"> ■ Small tools

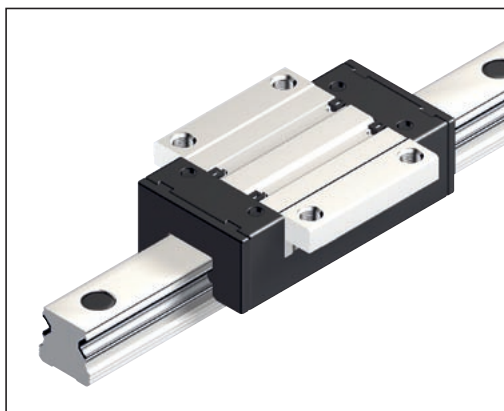
3 Profiled rail systems

3.4 eLINE ball rail systems

3.4.1 System characteristics

Rexroth's eLINE range of ball rail systems was developed especially for light machinery and for handling and positioning movements where the main emphasis is on economy and durability. eLINE guides offer an excellent price-performance ratio.

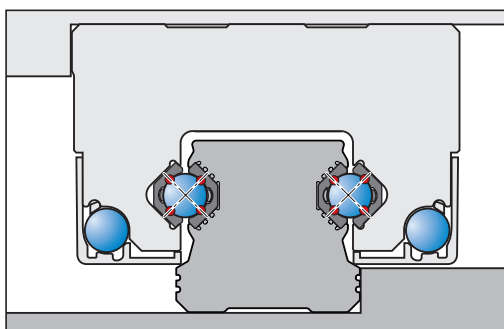
The eLINE ball rail system has two rows of balls with 4-point contact. The balls have a contact angle of 45°. The runner block body and the guide rail profile are made of aluminum. The load-bearing capability is achieved by means of hardened steel inserts in the rail and the runner block.



eLINE ball rail system

Features

- Low weight
- Compact design
- Same load capacities in all four major planes of load application
- Low accuracy requirements on the mating surfaces in the adjoining structure
- Significantly better corrosion resistance compared with the steel versions
- Limitless interchangeability due to precision manufacturing
- Large balls make this profiled rail system insensitive to dirt
- Optional front lube unit with sealing function
- Available in the most common DIN sizes and formats



2 rows of balls with 4-point contact

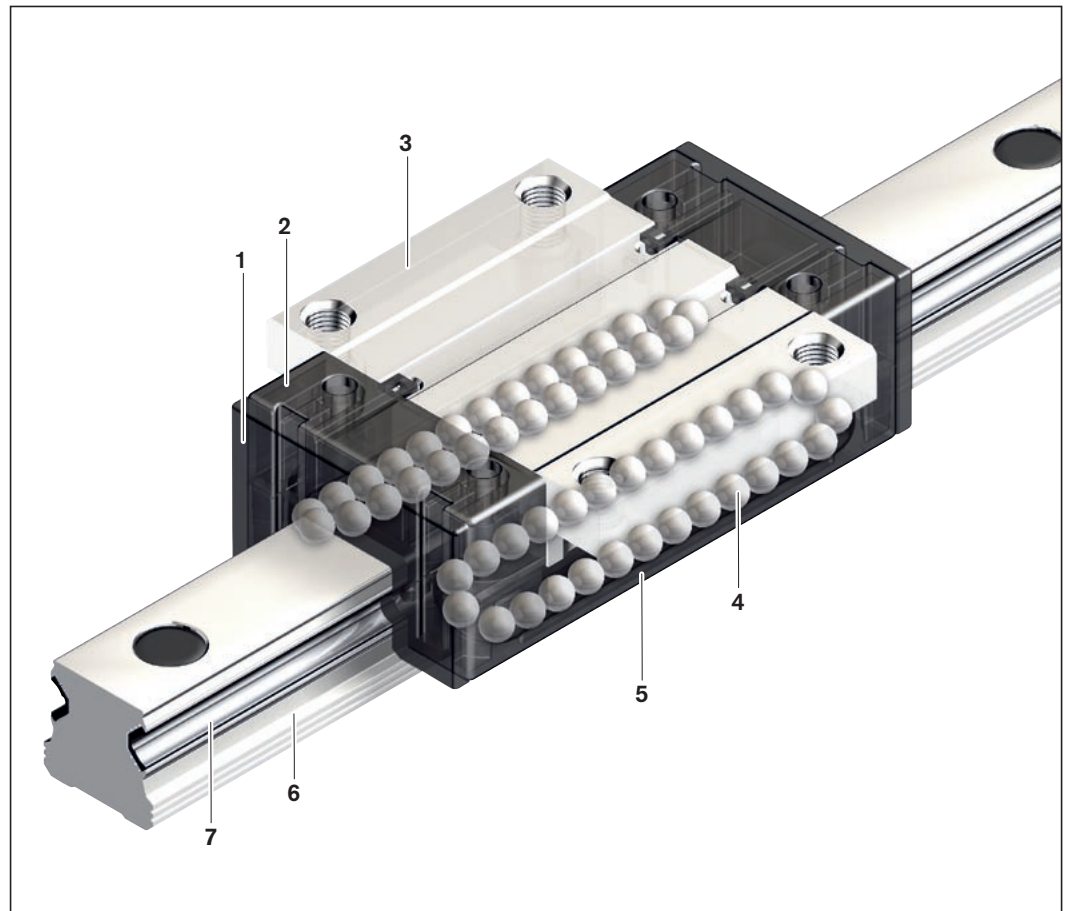
Product data

Product data	Values	
Speed	v_{\max}	Up to 2 m/s
Acceleration	a_{\max}	Up to 30 m/s ²
Temperature resistance	t_{\max}	60°C in continuous operation
Preload classes	2	C0, C1
Rigidity		Rigidity charts on request
Accuracy classes	2	E, N
Sizes	3	15, 20, 25

3 Profiled rail systems

3.4 eLINE ball rail systems

3.4.2 Structural design



Components of the eLINE ball rail system

- | | |
|--|------------------------------|
| 1 Seal unit (wiper seal) | 4 Row of balls |
| 2 End cap | 5 Housing |
| 3 Runner block body with integrated steel insert | 6 Guide rail body |
| | 7 Steel insert in guide rail |

3 Profiled rail systems

3.4 eLINE ball rail systems

3.4.2 Structural design

Runner blocks, general description

The runner blocks are available in flanged and slimline designs. They are pre-lubricated in-factory, therefore provided with long-term lubrication. A front lube unit with sealing function is available as an accessory, allowing the runner blocks to be relubricated with oil when in service.

Runner block body

The runner block body (1) is made from aluminum. This considerably reduces the overall weight of the runner block. The runner blocks can be fastened from above or below (depending on the version) using four screws.

Steel inserts

Hardened steel inserts (2) are integrated in the runner blocks as raceways for the balls.

Retaining plate

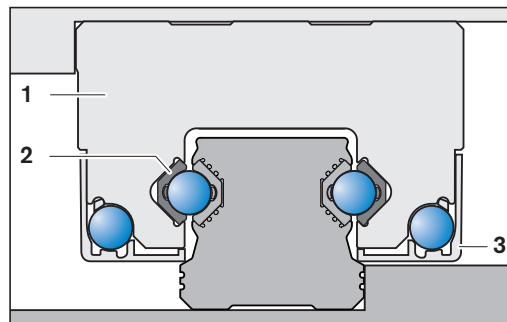
The steel balls are prevented from falling out by a retaining plate. Thanks to these retaining plates the runner blocks can be easily removed from the rail, which significantly simplifies mounting and disassembly.

Housing

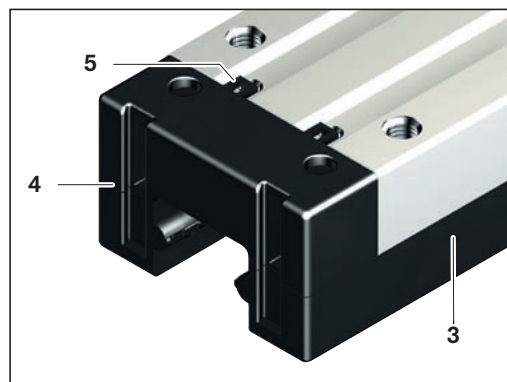
The balls are guided out of the load-bearing zone into the recirculation zone by two lateral housing parts (3), which also form the recirculation channel in combination with the aluminum body. They protect the balls from external influences.

End caps

The end caps (4) also form part of the ball recirculation geometry. They have clip fasteners (5) for easy fastening to the aluminum body and safe retention of the plastic internals. In addition, the end caps accommodate the seal unit or the front lube unit with sealing function.



Steel inserts in the runner block



End caps with clip fasteners

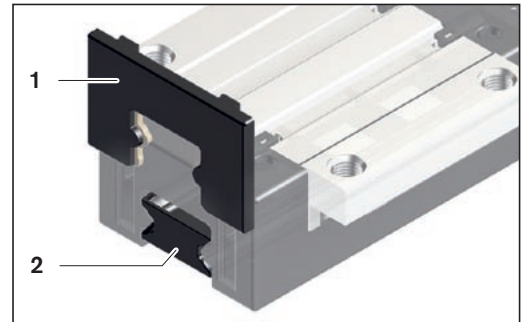
3 Profiled rail systems

3.4 eLINE ball rail systems

3.4.2 Structural design

Seal unit

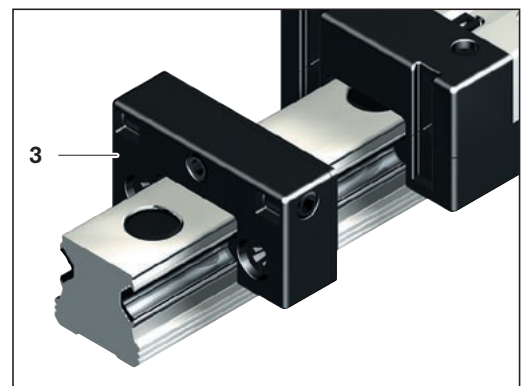
Each runner block comes with two seal units (1) mounted at the end faces. These seal units can be pulled out and replaced with lube units with sealing function. Seal units are two-piece components consisting of a holder and a foam insert.



eLINE runner block with seal unit

Lube unit with sealing function

The lube unit (3) allows in-service lubrication of the runner block while providing an added sealing function. These lube units are available as accessories.



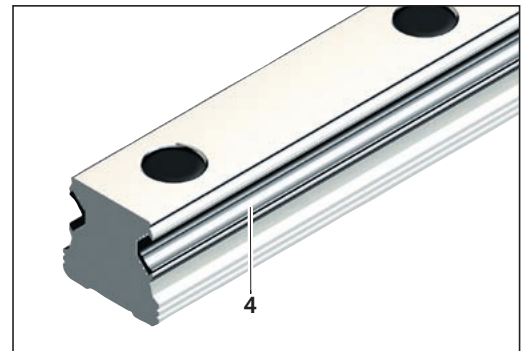
Lube unit with sealing function as an accessory

Transport and mounting arbor

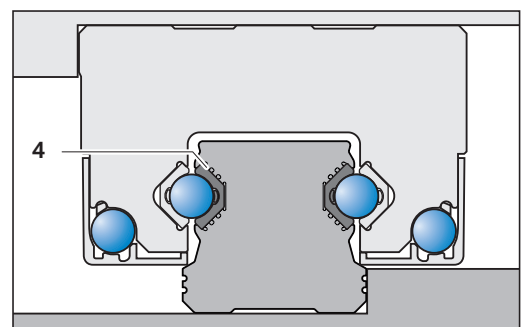
The runner blocks are delivered mounted on an arbor (2). This arbor prevents any loss of balls while the runner block is being transported and facilitates mounting of the runner block to the rail.

Guide rail

eLINE guide rails consist of a rail body and two steel inserts (4). The steel inserts are the running tracks for the balls and are therefore made from hardened steel. The rail is bolted to the mounting base from above or below. The mounting holes can then be closed with plastic plugs.



eLINE guide rail (with plastic mounting hole plugs)





Steel inserts in the eLINE guide rail

3 Profiled rail systems

3.4 eLINE ball rail systems

3.4.3 Product selection guide

3.4.3.1 Versions

Runner block formats		
Width	F	S
Length	N	N
Height	S	S
Code	FNS	SNS
		

3.4.3.2 Application areas

eLINE ball rail systems were designed for use in applications calling for good performance at an affordable price. Typical areas are:

Industry sector	Applications
Building services technology Tradeshow and shop furnishings	<ul style="list-style-type: none"> ■ Door and window technology ■ Furniture elements ■ Interior design
Assembly/handling technology and industrial robots	<ul style="list-style-type: none"> ■ Assembly equipment ■ Assembly lines ■ Positioning units ■ Manual displacement systems ■ Gripping and clamping equipment
Special-purpose machines	<ul style="list-style-type: none"> ■ Jigs and fixtures ■ Light machinery ■ Machine enclosures
DIY	<ul style="list-style-type: none"> ■ Various DIY projects

3.4.3.3 Simplified calculations

The application areas for eLINE ball rail systems are different from those of the ball rail systems mentioned earlier. The nominal life calculation for the eLINE range has therefore been simplified. The load on the bearing P_{act} is calculated using a number of factors: the coefficient for the operating condition k_f , and the coefficients for loads due to torsional moments k_t and longitudinal moments k_L . The required size can be selected using the calculated load on the bearing P_{act} and a maximum load on the bearing P_{max} . The outcome of the calculation is the travel life in kilometers, according to the type of lubrication used.

If required, the exact nominal life can be calculated in the usual way, using the equivalent load on bearing and the load capacity (see section 3.1.5). Because of the weight-optimized design of eLINE ball rail systems, the maximum permissible forces F_{max} and the maximum permissible torsional moments $M_{t max}$ and longitudinal moments $M_{L max}$ must not be exceeded.

Details of the full and the simplified nominal life calculation can be found in the eLINE ball rail systems catalog.

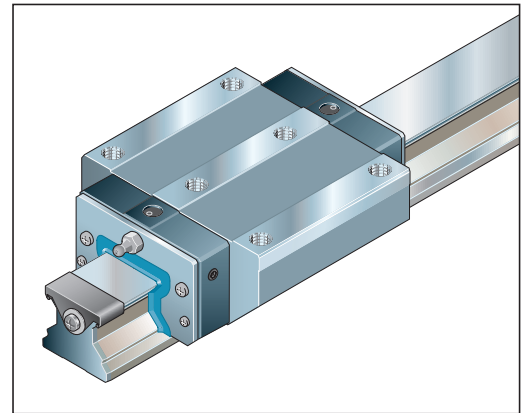
3 Profiled rail systems

3.5 Roller rail systems

3.5.1 System characteristics

Rexroth roller rail systems were designed especially for applications in machine tools and industrial robots. Available in various accuracy classes, these linear guides have exceptionally high load-bearing capacity and rigidity.

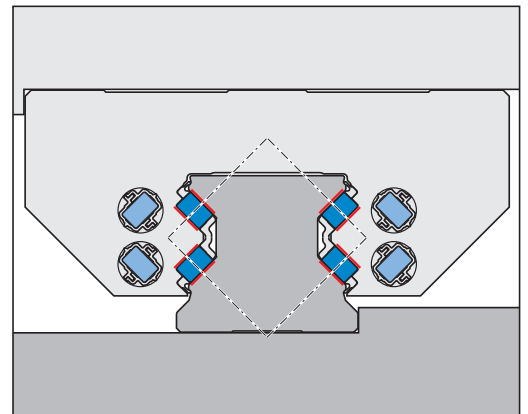
Roller rail systems have four roller bearing circuits in an O-arrangement. The rollers are in line-contact with the raceways at a contact angle of 45°.



Roller rail system

Features

- Rolling elements: rollers
- Very high static load capacities
- Very high dynamic load capacities
- High static torque capacity
- Very high rigidity in all major planes of load application
- Very good travel performance even under extremely high loads
- Unlimited interchangeability due to precision manufacturing
- Integrated all-round sealing
- Broad range of accessories for special sectoral solutions (seals, wipers/scrapers)
- Integrated, inductive and wear-free measuring system as an option
- Clamping and braking units available



Four rows of rollers in an O-arrangement

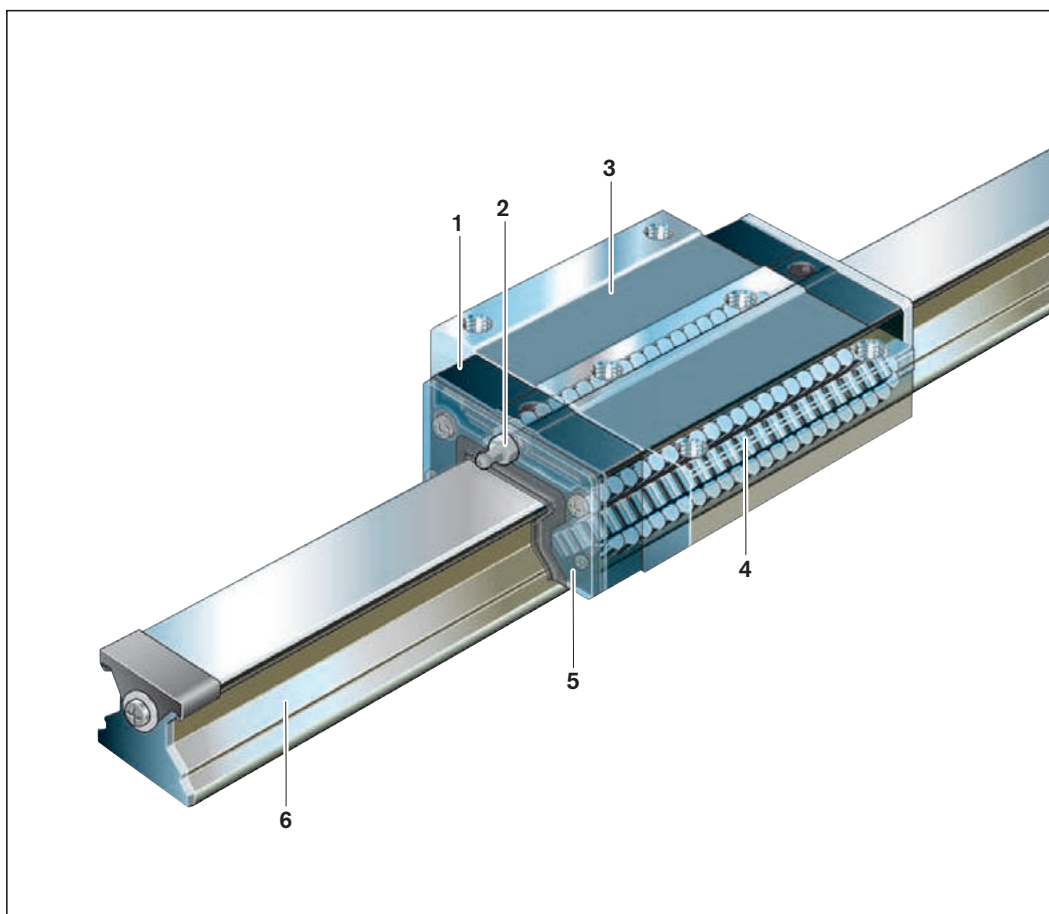
Product data

Product data	Values	
Speed	v_{\max}	Up to 3 m/s
Acceleration	a_{\max}	Up to 50 m/s ²
Temperature resistance	t_{\max}	100°C brief peaks, 80°C in continuous operation
Preload classes	4	C2, C3 (C1 on request)
Rigidity		Rigidity charts showing lift-off/down/side loads for all types in product catalog
Accuracy classes	4	H, P, SP, UP
Sizes	9	25, 35, 45, 55, 65, 100, 125 55/85, 65/100

3 Profiled rail systems

3.5 Roller rail systems

3.5.2 Structural design



Roller rail system

- 1 End cap
- 2 Lube port (lube nipple)
- 3 Runner block body
- 4 Cylindrical rollers (rolling elements)
- 5 End seal
- 6 Guide rail

**Runner blocks,
general description**

Roller runner blocks are oiled prior to shipment to protect them from corrosion. They can be lubricated with oil or grease. Rexroth offers special runner blocks for minimum-quantity lubrication with oil, for wall mounting, and for lubrication from above. A front lube unit is available to provide long-term lubrication. For optimal sealing, the guide can be equipped with accessories (e.g. Viton seal, metal scraper, bellows).

3 Profiled rail systems

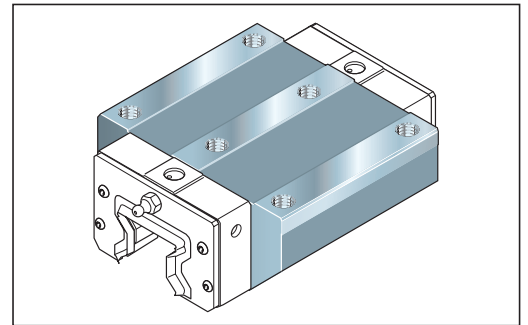
3.5 Roller rail systems

3.5.2 Structural design

Runner block body

Rexroth runner block bodies are available in four versions according to DIN 645-1 and in various special versions. All runner block bodies are made from anti-friction bearing steel. Depending on their design, same-size runner blocks may have different load capacities and rigidities.

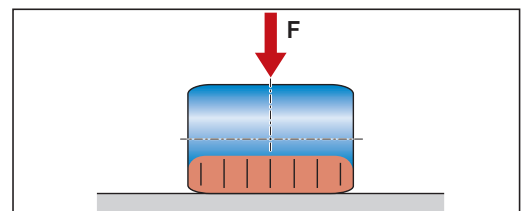
The runner block bodies contain four raceways and recirculation bores for the rolling elements and have threaded holes for mounting to the adjoining structure.



Runner block body

Logarithmic roller

The rolling elements in the runner block bodies are rollers with a logarithmic profile. This profile prevents excessive stresses at the edges and helps to prolong the life of the runner block. The rollers are made from grade 100Cr6 anti-friction bearing steel.



Stress distribution in a logarithmic roller profile

Roller guidance

The rollers are guided by recirculation pieces and a frame. To keep the friction as low as possible, all of the roller guidance parts are made from a plastic material with excellent sliding friction properties.

Recirculation piece (roller pick-up)

The recirculation piece guides the rollers from the load-bearing zone to the return zone. In the O-arrangement used by Rexroth the rolling elements are recirculated cross-wise. Integrated lube ducts ensure that all raceways are reliably lubricated.

Frame

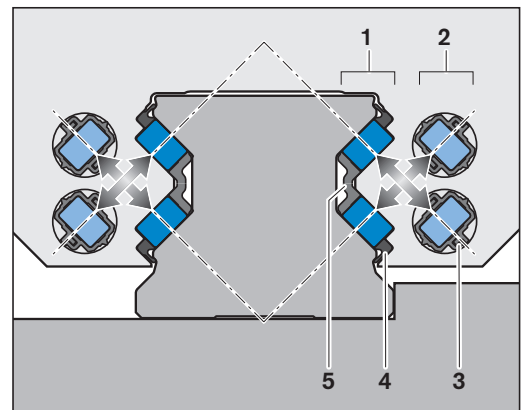
The main purpose of the frame is to provide lateral guidance for the rollers in the runner block load-bearing zones. The frames also retain the rollers and prevent them from dropping out during mounting and removal of the runner blocks.

Side seals

Two integrated side seals per frame protect the rolling elements and the raceways from dirt.

Return channel

The recirculation zone of the roller runner blocks essentially consists of return channels. As with the roller guidance parts, the return channels are made from a plastic material with good sliding properties. Lateral pockets ensure adequate lubricant transport and reduce friction.



Roller guidance and return channels

- 1 Roller load-bearing zone
- 2 Roller recirculation zone
- 3 Return channels
- 4 Side seals (4x)
- 5 Frame

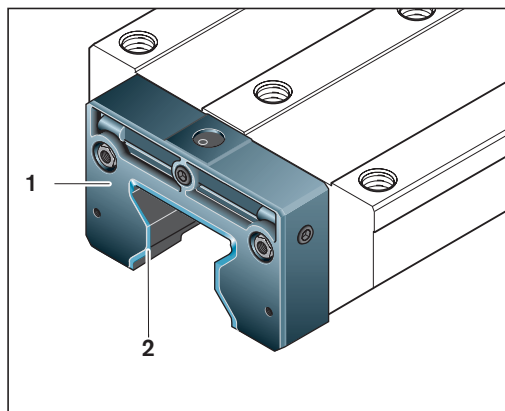
3 Profiled rail systems

3.5 Roller rail systems

3.5.2 Structural design

End cap

The end cap (1) is designed to protect the internal plastic components from dirt and to distribute the lubricant. The lube ports are closed by set screws. Runner blocks with standard black end caps can be used for both grease and oil lubrication. Runner blocks with gray end caps are used for minimum-quantity lubrication with oil. For wall mounting, the Rexroth range includes a runner block with two lube ports on the end face of each end cap for separate lubrication of each raceway. Runner blocks with aluminum end caps offer added protection from coarse dirt and hot shavings. A version with adapter for lubrication from the top rounds out this varied range of runner blocks.



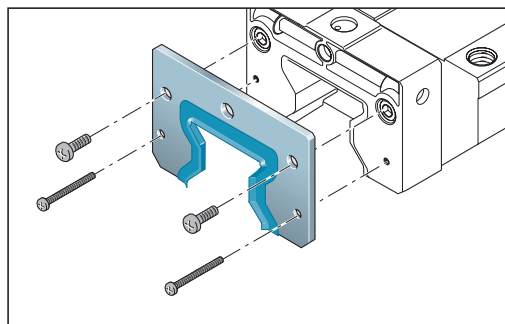
End cap with integrated sealing plate

Sealing plate

The integrated sealing plate (2) prevents dirt from working its way into the runner block and keeps the lubricant inside it. To ensure good sealing action with low friction, the lip of the wiper seal has elastic properties and is slightly tensioned against the guide rail.

End seals

All roller rail runner blocks are delivered with additional, rugged external seals, known as end seals. Combined with the internal sealing plate, this results in an effective sealing system that will perform well in all standard conditions of use. The end seal consists of an elastic sealing lip formed in place on a metal plate.



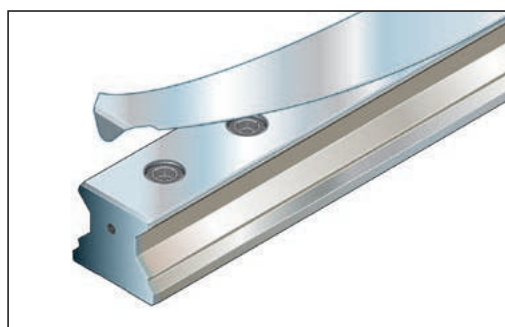
End seal

Transport and mounting arbor

All roller runner blocks are mounted on an arbor for shipment. The arbor prevents the rollers from falling out during transport and facilitates mounting and removal of the runner block.

Guide rail

The guide rail is made from heat-treated steel and has four hardened running tracks. Rexroth offers guide rails for mounting from above and below. For guide rails mounted from above, there are various options for sealing the mounting holes. V-guide rails with a dovetail fit can be installed on the mounting base using pressure pieces.



Guide rail with cover strip

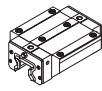
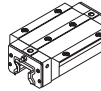
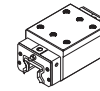
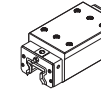
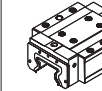
3 Profiled rail systems

3.5 Roller rail systems

3.5.3 Product selection guide

3.5.3.1 Versions

Rexroth offers different designs and versions to meet the needs of a broad range of applications:

Runner block formats					
Width	F		S		B
Length	N	L	N	L	N
Height	S	S	H	H	N
Code	FNS	FLS	SNH	SLH	BNN
					

- Roller rail systems with Resist CR
- Wide roller rail systems
- Heavy duty roller rail systems
- Runner blocks for wall mounting
- Runner blocks with aluminum end caps
- Runner blocks for central oil lubrication systems (minimum-quantity lubrication)

3.5.3.2 Application areas

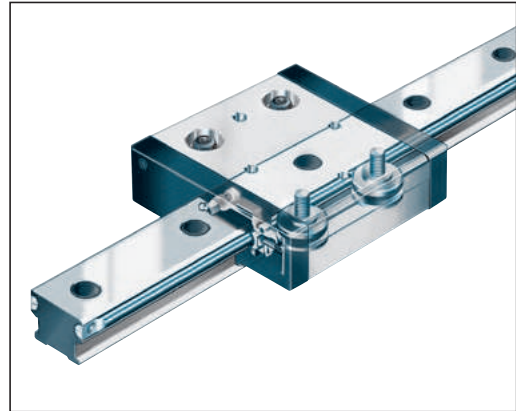
Industry sector	Applications	
Metal-cutting machine tools	<ul style="list-style-type: none"> ■ Machining centers ■ Lathes and turning machines ■ Drilling machines ■ Milling machines 	<ul style="list-style-type: none"> ■ Grinding machines ■ Nibbling machines ■ Electrical discharge machines ■ Laser/light/photo beam machine tools
Forming and stamping machine tools	<ul style="list-style-type: none"> ■ Bending machines ■ Straightening/leveling machines ■ Presses 	<ul style="list-style-type: none"> ■ Wire bending machines ■ Wire drawing machines
Rubber and plastics processing machinery	<ul style="list-style-type: none"> ■ Injection molding machines ■ Extruders 	<ul style="list-style-type: none"> ■ Calendering machines ■ Blow molding machines
Automotive industry	<ul style="list-style-type: none"> ■ Car production lines ■ Welding systems 	<ul style="list-style-type: none"> ■ Pressing and stamping lines ■ Paintshop systems
Paper and printing machines	<ul style="list-style-type: none"> ■ Paper winders/unwinders ■ Printing machines 	<ul style="list-style-type: none"> ■ Cutters
Assembly/handling technology, industrial robots	<ul style="list-style-type: none"> ■ Heavy duty equipment ■ Cable and tape reelers 	<ul style="list-style-type: none"> ■ Palletizer robots
Steel industry rolling mills	<ul style="list-style-type: none"> ■ Roll adjustment 	<ul style="list-style-type: none"> ■ Coilers/uncoilers
Welding technology	<ul style="list-style-type: none"> ■ Automatic welders ■ Hot welding equipment 	<ul style="list-style-type: none"> ■ Friction welding equipment
Food and packaging industries	<ul style="list-style-type: none"> ■ Palletizers ■ Molding machines 	<ul style="list-style-type: none"> ■ Cutters
Woodworking and wood processing machines	<ul style="list-style-type: none"> ■ Heavy duty wood processing equipment 	<ul style="list-style-type: none"> ■ Sawing machines

3 Profiled rail systems

3.6 Cam roller guides

3.6.1 System characteristics

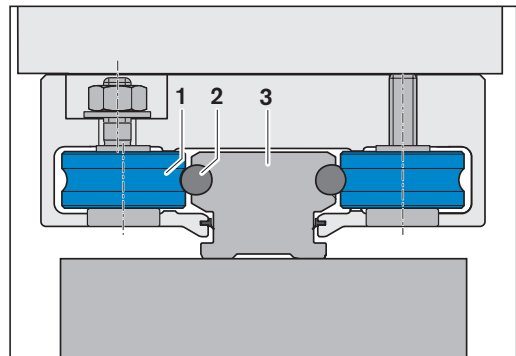
Rexroth cam roller guides were developed primarily for handling and automation applications. They differ from typical profiled rail systems because the rolling elements do not circulate between the runner blocks and the guide rails. The main components of cam roller guides are the cam rollers (1), which are mounted on ball bearings. Made from steel, the cam rollers guide the runner blocks along the running tracks (2) in the guide rail (3).



Cam roller guide

Features

- High load-bearing capability in all four major planes of load application
- High moment load capacity about all axes
- Very high permissible speed
- Compact dimensions
- Very low weight
- Easy mounting
- Low friction
- Low-noise operation
- Complete guide unit
- Rugged design
- Interchangeable elements readily available
- Low demands on accuracy of the adjoining structures



Cam roller guide – schematic of operating principle

Product data

Product data	Values	
Speed	v_{\max}	10 m/s
Acceleration	a_{\max}	50 m/s ² Higher acceleration possible, provided slip is avoided
Temperature resistance	t_{\max}	8°C
Preload classes	none	Adjustable via eccentric spigot/pins
Rigidity		Depends on eccentric spigot/pin adjustment
Accuracy classes	1	One accuracy class, higher accuracies available on request
Sizes	5	20, 25, 32, 42, 52

Higher rigidity can be achieved by adjusting the eccentric spigot/pin to increase the preload force.

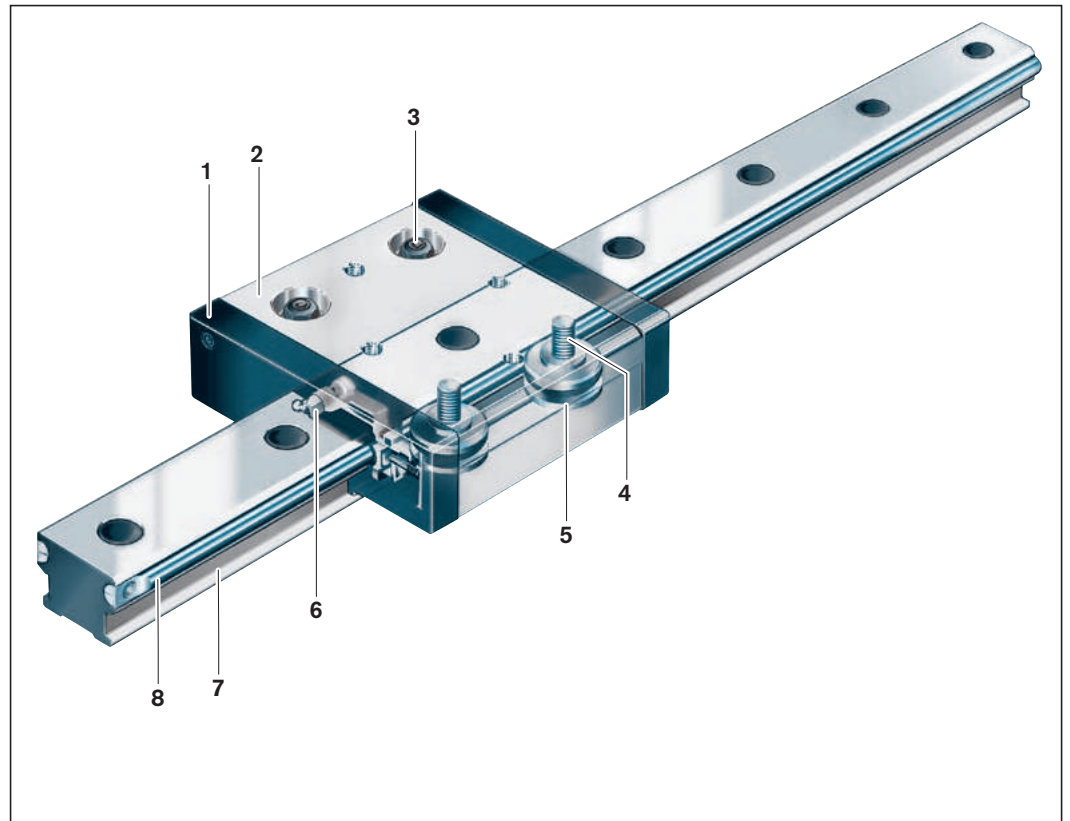
Unlike all other profiled rail systems, cam roller guides have no accuracy classes. All sizes have the same accuracy values (see product catalog).

3 Profiled rail systems

3.6 Cam roller guides

3.6.2 Structural design

Cam roller guides are available in many different versions. They generally consist of at least one guide rail and at least one runner block with cam rollers. The structural design is illustrated below, using a standard cam roller runner block as an example. This is the most commonly used design. Further designs are shown in section 3.6.3.1.



Components of the cam roller guide

- | | |
|-------------------------------|---------------------------|
| 1 Oil applicator/wiper unit | 5 Cam roller |
| 2 Runner block body | 6 Lube port (lube nipple) |
| 3 Eccentric cam roller spigot | 7 Guide rail body |
| 4 Central cam roller spigot | 8 Precision steel shaft |

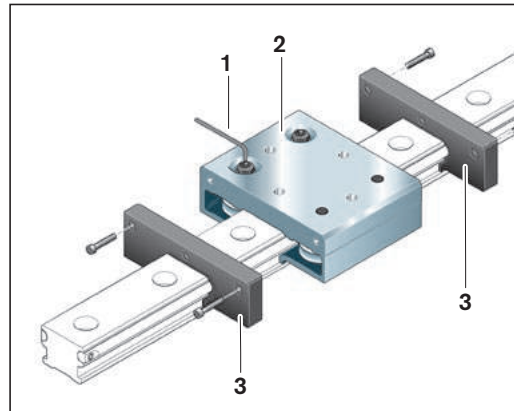
3 Profiled rail systems

3.6 Cam roller guides

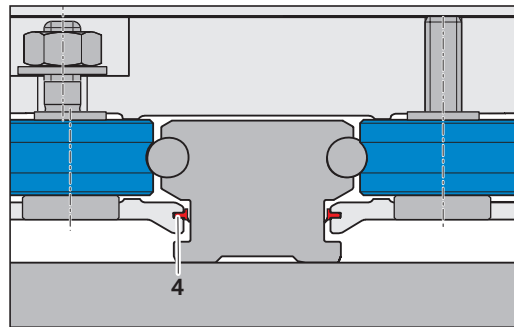
3.6.2 Structural design

Runner block

The cam rollers have eccentric bearings and the runner blocks can be adjusted by means of eccentric spigots (1). The runner block body (2) is made of aluminum. It has an oil applicator and wiper unit with a large oil reservoir (3) at each end. This ensures long maintenance intervals and possibly even lubrication for life. Lube nipples can be mounted at either end for in-service lubrication. In addition to the oil applicator and wiper units, the runner blocks have side seals (4) to seal them off on the underside. The runner blocks can be fastened to the surrounding structure with screws.



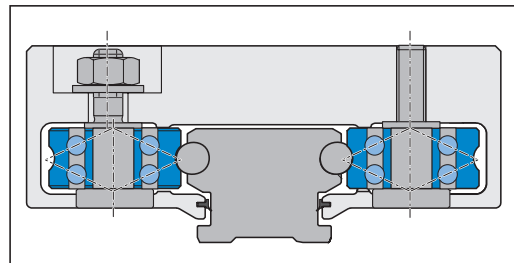
Components of the cam roller runner block



Side seals

Cam rollers

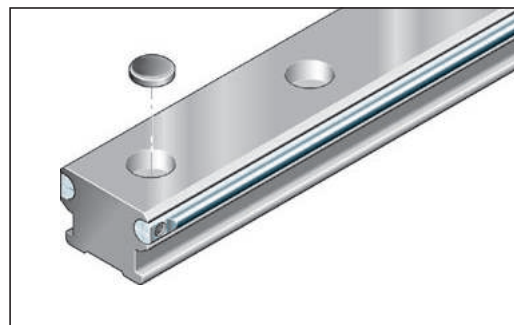
The cam rollers are mounted on two-row angular-contact thrust ball bearings. These ball bearings are sealed and lubricated for life. The inner and outer raceways are made from anti-friction bearing steel. The outer raceway is in rolling contact with the precision steel shaft in the guide rail. The cam rollers installed in the runner blocks are also available as separate parts.



Cam rollers with two-row angular-contact thrust ball bearings

Guide rail

The guide rail body is made of anodized aluminum. One or two hardened, corrosion-resistant precision steel shafts are integrated in the guide rail as running tracks. The cam rollers run along these shafts. Guide rails are available in different versions to meet different requirements. The rail can be mounted from above or from below. In the latter case, sliding blocks are used to anchor the screw-fasteners. If rails are mounted from above, the mounting holes can be closed with plugs.



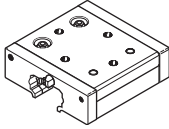
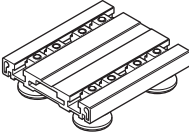
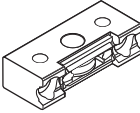
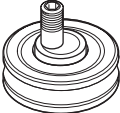
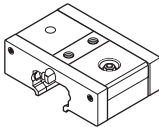
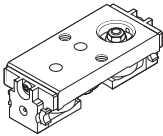
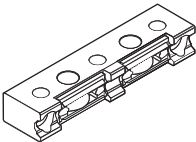
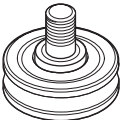
Standard guide rail with mounting hole plug

3 Profiled rail systems

3.6 Cam roller guides

3.6.3 Product selection guide

3.6.3.1 Versions

Versions			
Runner block	Profile runner block	Single cassette	Cam roller with eccentric spigot
			
Super runner block	U-type runner block	Double cassette	Cam roller with central spigot
			

Super runner block

The main feature of the super runner block is its ability to compensate for misalignments.

Super runner blocks have eccentric cam roller spigots and oil applicator units, just like the standard runner blocks.

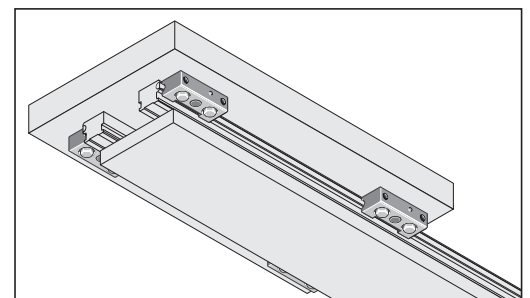
Profile runner block

The profile runner block has two T-slots for easy mounting of customer-built attachments using screws and sliding blocks. It is adjusted to zero clearance before shipment. Long maintenance

cycles can be achieved with this type of runner block as well, by installing oil applicator/wiper units.

Single cassette

Both single and double cassettes offer many opportunities for building customized solutions quickly and effectively. A complete guide unit consists of at least two double or four single cassettes. The cassettes have integrated lubrication units, which assure long travel life while providing a wiper function. The preferred lubricant for the cassettes is grease.

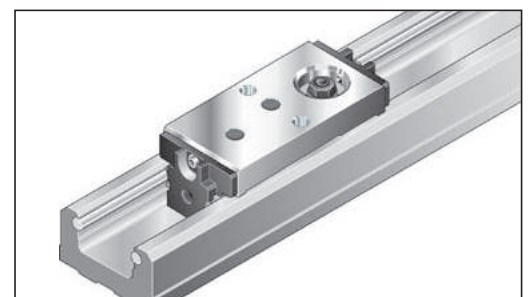


Guide unit with four single cassettes and two standard half-rails (example)

Double cassette

U-type runner block

U-type runner blocks run in a U-shaped rail with running tracks on the inside of the "U". Thanks to this geometry, the compact U-type runner block is protected by the guide rail. The benefit here is that users may be able to dispense with protective covers, if the application permits this.



U-type cam roller guide

3 Profiled rail systems

3.6 Cam roller guides

3.6.3 Versions, application areas, different calculation procedure

3.6.3.2 Application areas

Cam roller guides are used in the following industries, mainly in applications with low loads and high speeds. They are often installed in auxiliary equipment serving the main production machines.

Industry sector
Assembly/handling/industrial robots
Food and packaging industries
Metal-cutting machine tools
Printing and paper industry
Electrical/electronics industry
Rubber and plastics processing machinery
Machinery for building materials, ceramics and glass
Conveyor systems
Precision machine tools
Forming and stamping machine tools

3.6.3.3 Different calculation procedure

The nominal life calculation for cam roller guides differs from that of other profiled rail systems. The static load capacity $C_{0,y,z}$ and the dynamic load capacity $C_{y,z}$ are calculated using the load ratings of the integrated angular-contact thrust ball bearings instead of the rolling contact between the running track and the cam roller. These load capacities result in static load moment capacities $M_{0,x,y,z}$ and dynamic load moment capacities $M_{x,y,z}$ for calculation of the moment loads.

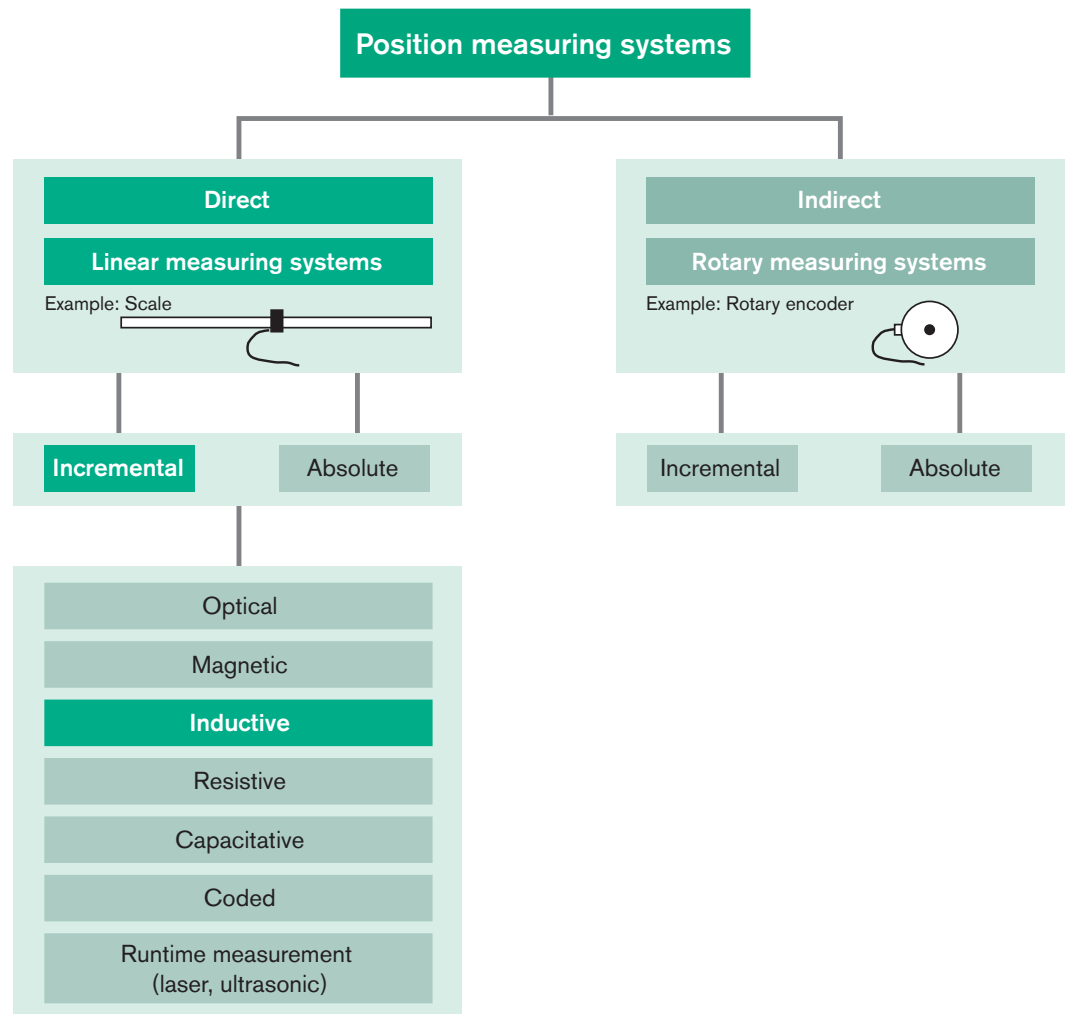
In addition, there are load limits for effective dynamic forces $F_{\max,y,z}$, static forces $F_{0,\max,y,z}$, and for dynamic moments $M_{\max,x,y,z}$ and static moments $M_{0,\max,x,y,z}$. This takes account of the strength of the rail and the runner block, the load-bearing capability of the cam rollers and of the screw connections. The load capacities and maximum permissible loads are given in the product catalog for all versions and sizes.

3 Profiled rail systems

3.7 Integrated measuring system

3.7.1 Position measuring systems principles

A variety of different principles can be used to measure linear travel. In order to understand the advantages and disadvantages of the different systems, certain terms have to be explained first.



Direct and indirect position measuring systems

There are direct and indirect position measuring systems. In direct position measuring systems, the linear displacement is measured using a scale, which must be as long as the entire travel distance. By contrast, indirect position measuring

systems measure linear displacement by means of changes in the angular position of the drive. The Rexroth measuring system is a direct linear measuring system.

Incremental and absolute position measuring systems

A distinction is made between incremental and absolute systems. Incremental position measuring systems measure only changes in the distance traveled. A homing cycle has to be performed to determine the absolute position. The system that Rexroth uses is an incremental one.

In absolute systems the precise position is always known immediately after system start-up. No referencing is required to find the zero point.

3 Profiled rail systems

3.7 Integrated measuring system

3.7.1 Position measuring system principles

Operating principles of position measuring systems

Another distinguishing feature of position measuring systems is the design of the sensors used. It is useful to have a system which supplies electrical measurement signals. The measuring principle

in such systems is usually optical, magnetic or inductive. Rexroth uses an inductive system. To allow comparison with the other systems, the individual operating principles are explained below.

Optical position measuring systems

Optical beam paths can be used in combination with apertures of defined sizes or graduated scales with coded or incremental, light-permeable areas or grids to generate analog signals. Using

counting and evaluation units, these signals can be processed to generate digital position measurement signals.

Magnetic position measuring systems

Magnetic measuring systems are based on travel-dependent influencing of magnetic effects in suitable sensors, e.g. Hall sensors.

Inductive measuring systems

This method involves travel-dependent influencing of electromagnetic inductance by reciprocal displacement of AC-powered coil systems and iron cores (plunger-type and yoke-type armatures).

Advantages and disadvantages of the position measuring systems

Position measuring systems	Advantages	Disadvantages
Optical systems	<ul style="list-style-type: none"> ■ High resolution and linearity 	<ul style="list-style-type: none"> ■ Very difficult to integrate ■ Sensitive to dirt
Magnetic systems	<ul style="list-style-type: none"> ■ Easy application (add-on element) 	<ul style="list-style-type: none"> ■ Sensitive to magnetizable particles (metal shavings, abraded particles) ■ Sensitive to extreme static magnetic fields (linear motor)
Inductive systems	<ul style="list-style-type: none"> ■ Insensitive to dirt and shavings ■ Easy to integrate (non-magnetic strip) ■ Insensitive to magnetic fields (linear motor) ■ Non-contacting 	<ul style="list-style-type: none"> ■ Relatively high power consumption

Position measuring system	Compatibility with coolants	Insensitivity to shavings	Dry processing	Space requirement	Mounting
Optical	o	o	++	+	++
Magnetic	+	o	++	+++	+++
Inductive	+++	+++	+++	+++	+++

Position measuring system	Retrofitability	Design	Accuracy class	Resolution Repeatability
Optical	o	o	+++	+++
Magnetic	++	+++	+	+++
Inductive	+++	+++	++	++

+++ Very good

++ Good

+ Satisfactory

o Adequate

3 Profiled rail systems

3.7 Integrated measuring system

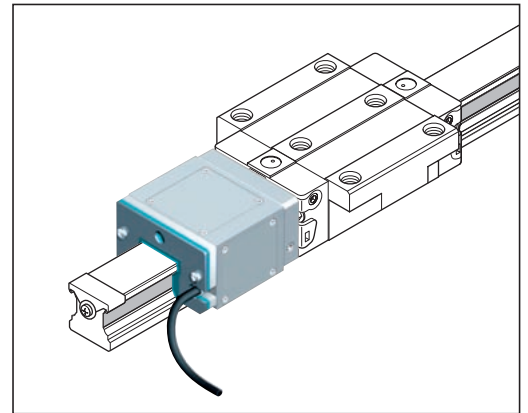
3.7.2 System characteristics

Rexroth ball rail and roller rail systems can be supplied with an integrated, inductive, direct linear measuring system. This development combines the guiding and measuring functions in one unit and opens up new opportunities in machine design.

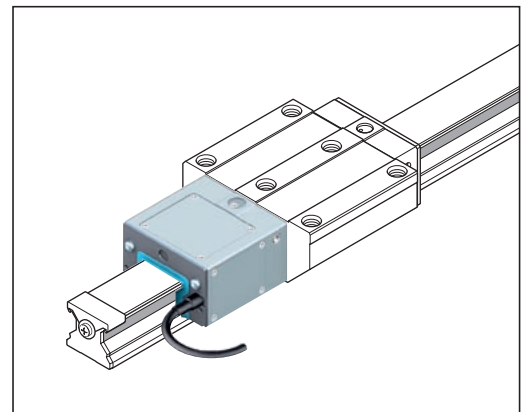
The integrated measuring system consists of a scanner (sensor) mounted on the runner block and a highly precise scale integrated in the rail. It is the only linear measuring system that combines extremely rugged design with the precision of optical systems.

Rexroth's integrated measuring system has the following features:

- Direct linear measuring system
- Non-contacting, inductive scanning system
- Maintenance-free, virtually no aging
- Combined guidance and measurement in one unit
- Insensitive to magnetic fields
- Interchangeable
- High shock and vibration resistance
- No measuring inaccuracies due to deviations in parallelism
- Several sensor units can be mounted on one rail
- Resistant to water, oil, metalworking fluids, dust, shavings, etc. (protection class IP67)
- No space required for external measuring system
- Easy retrofitting (compatible with standard guidance components)
- No costs for external mounting
- Insensitive to any kind of contamination
- High accuracy and resolution
- Allows high travel speeds
- Integrated reference mark, also distance-coded



Ball rail system with integrated measuring system



Roller rail system with integrated measuring system

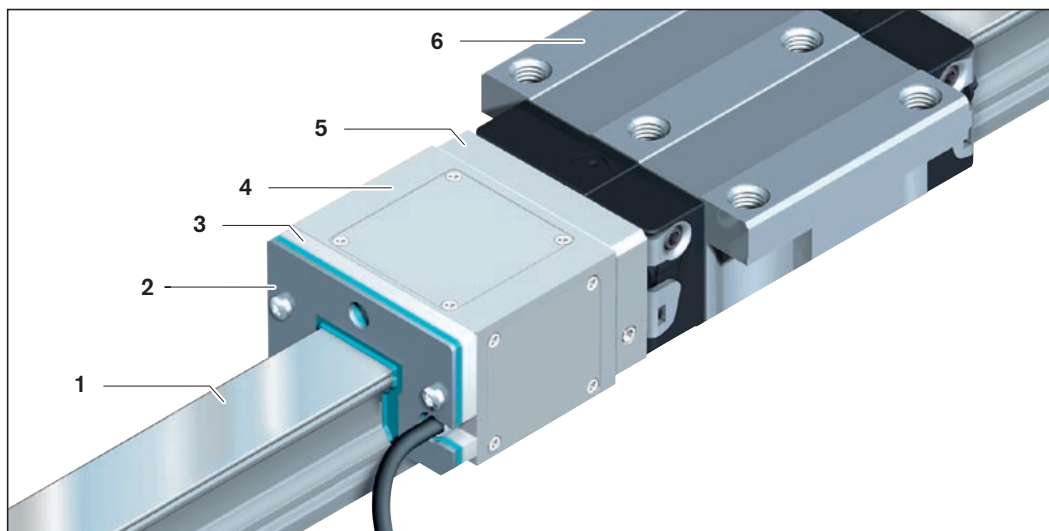
3 Profiled rail systems

3.7 Integrated measuring system

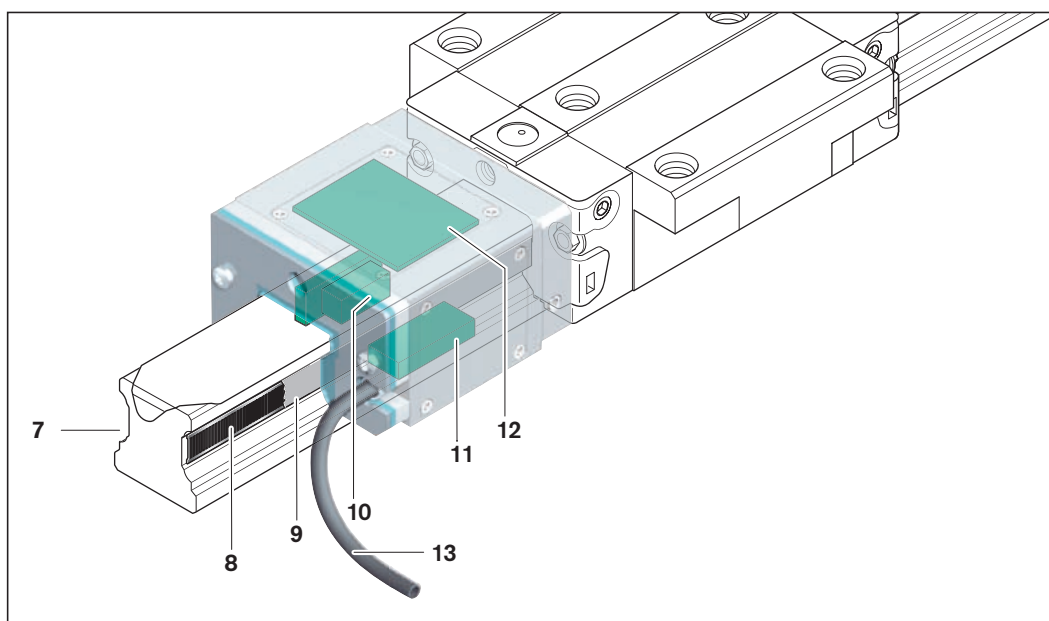
3.7.3 Structural design

The direct inductive linear measuring system consists of a scanner, a scale and reference marks integrated in a ball or roller rail system. The scanner with sensors is mounted on the runner block.

As it travels over them, it evaluates the scale and the reference marks integrated in the rail.



Components of the inductive measuring system



Functional elements of the integrated measuring system

- | | | | |
|---|--|----|--|
| 1 | Guide rail with scale | 8 | Scale |
| 2 | End seal | 9 | Scale protection: laser-welded stainless steel strip |
| 3 | Support plate | 10 | Reference sensor |
| 4 | Scanner | 11 | Measuring sensor |
| 5 | Adapter with intermediate plate | 12 | Evaluation electronics |
| 6 | Runner block | 13 | Cable and connector |
| 7 | Reference marks (on opposite side of the guide rail) | | |

3 Profiled rail systems

3.7 Integrated measuring system

3.7.3 Structural design

3.7.3.1 Components functions

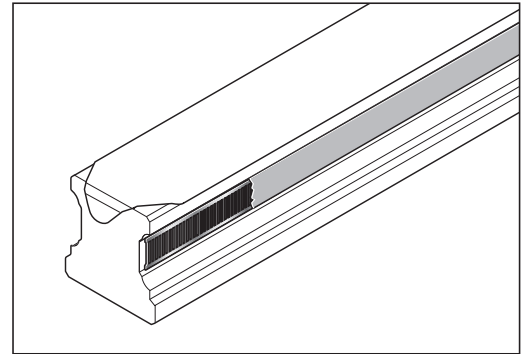
Guide rail

The scale and the reference marks for the measuring system are integrated in the guide rail. All tuning work has therefore been completed at the factory prior to shipment. The use does not

incur any extra installation costs. Moreover, the system does not take up any additional space.

Scale

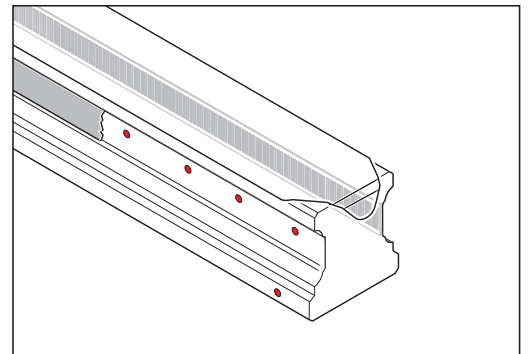
The scale is integrated in the side of the guide rail. It is a non-magnetic, high-precision, graduated steel strip with a pitch of $1,000\ \mu\text{m}$, which is joined to the rail by welding. A rust-proof laser-welded stainless steel strip protects the scale from contamination.

Scale protection

Scale

Reference marks

Reference marks are machined into the rail on the side opposite the scale. These reference marks are holes drilled at defined positions. They, too, are protected by a stainless steel strip. To avoid any confusion between the rail sides, the side with the reference marks is designated by a hole drilled into the reference edge. Alternatively, a single reference mark can be provided at a position specified by the user.



Reference marks

Mounting

The guide rail is mounted from above or below. The mounting holes can be sealed by a cover strip or with plugs.

3 Profiled rail systems

3.7 Integrated measuring system

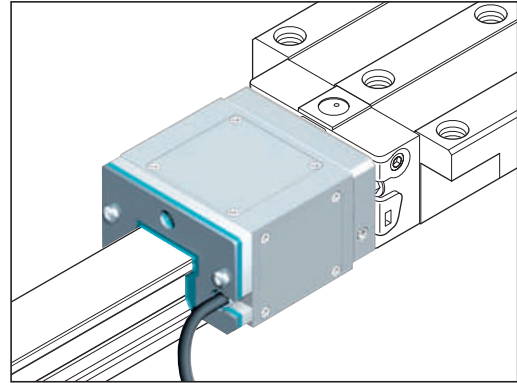
3.7.3 Structural design

Scanner on the runner block

The basic design of the scanner is explained here, using a ball runner block as an example.

The scanner for roller runner blocks differs only in minor details.

The scanner is mounted to one end cap of the runner block and has the same width and height as the cap. The mounting hole pattern of the runner block remains unchanged. The advantage of this is that the runner block can be mounted to the adjoining structure in the same way as a runner block without measuring system. The scanner contains the non-contacting sensor system and the unit for recognizing the reference marks. It also contains the required electronics. The scanner can deliver either analog or digital signals, as required.



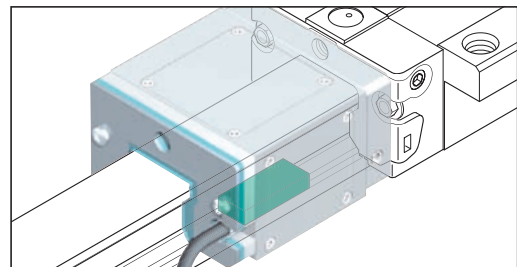
Sensor housing with adapter

Scanner housing

The aluminum scanner housing accommodates all of the other scanner components. It is fastened to the runner block via an adapter, thus forming one unit with it. The adapter allows the scanner to be replaced without having to remove the runner block from the rail. Together with the three housing covers, it protects the electronic and mechanical components against dirt and impact loads. The covers are tightly screwed down on the housing and may not be opened by the user.

Measuring sensor

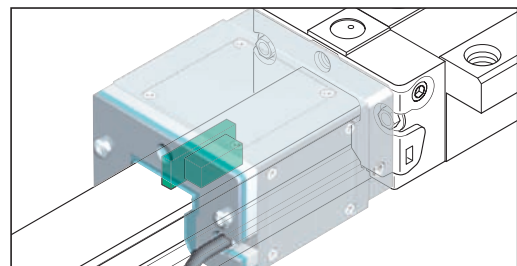
The non-contacting measuring sensor which scans the scale in the guide rail is located on one side of the scanner. It consists of a large number of transmitter and receiver coils (see function principle, section 3.7.3.2) and is protected by a sensor mount made from aluminum.



Measuring sensor

Reference sensor

The reference sensor is located on the opposite side of the scanner. This sensor scans the reference marks on the guide rail, thus allowing the absolute position of the runner block on the rail to be detected. The reference sensor is also fixed to a protective sensor mount. Both sensor mounts are fastened to the scanner housing with screws.



Reference sensor

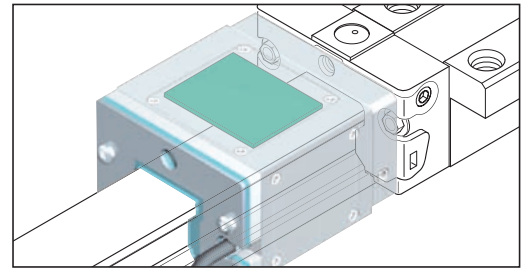
3 Profiled rail systems

3.7 Integrated measuring system

3.7.3 Structural design

Evaluation electronics

The evaluation electronics (a printed circuit board) includes all the necessary electronic circuits to generate, process, calibrate and transmit the signal. The interpolation function is already integrated in the printed circuit board for the digital version.



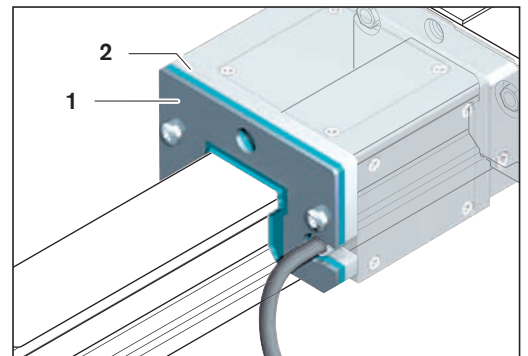
Printed circuit board

Potting compound

To achieve protection class IP67, the above components are installed in the scanner housing and the cavities are then filled with potting compound. This renders the system insensitive to water, oil and metalworking fluids.

End seal

To provide added protection, an end seal (1) is fastened to the scanner with screws. This prevents water, oil, metalworking fluids, shavings and dust from working their way into the scanner from the end face. Side seals protect the underside of the scanner.



End seal and support plate

Support plate

The support plate (2) is mounted between the end seal and the scanner. It has a clearance of 0.1 mm to the guide rail and prevents the sensor from touching the rail in the event of strong vibrations and impact loads.

Connector Cable

The connector and cable connect the scanner to the control system. A choice of connectors and cables is offered to meet different requirements and control system designs.

3 Profiled rail systems

3.7 Integrated measuring system

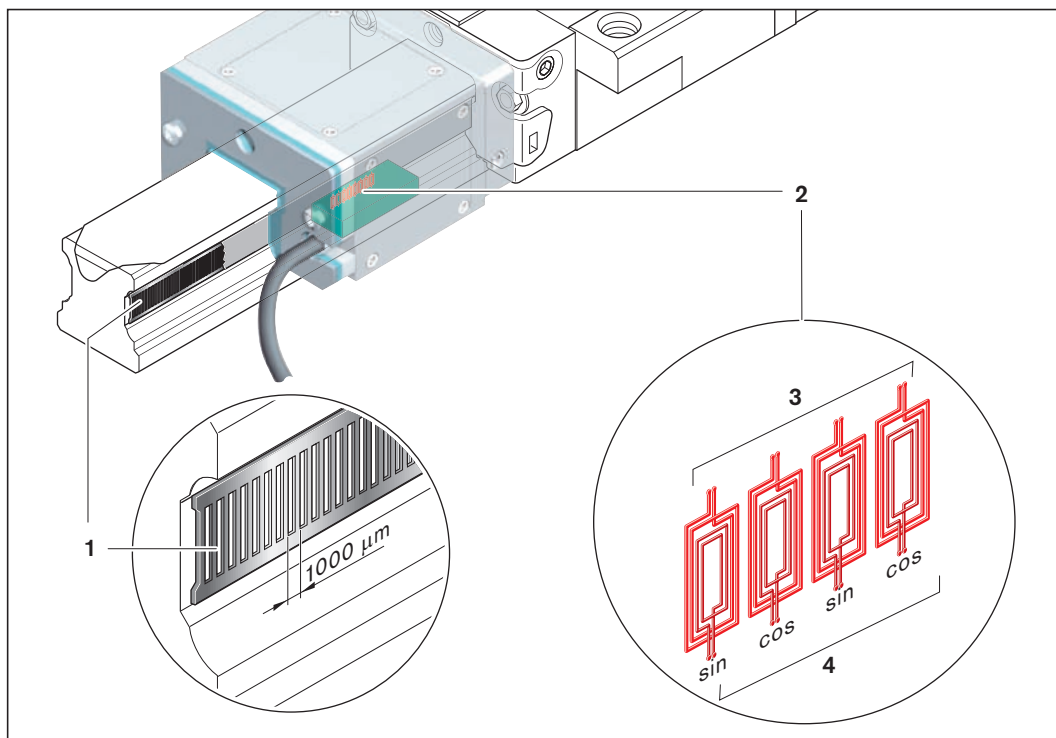
3.7.3 Structural design

3.7.3.2 Function description of the inductive sensors

Function of the measuring sensor and scale

The integrated measuring system works like a transformer. The scanner features an array of coils

which are protected against mechanical damage and electromagnetic interference.



Function principle of the scale and measuring sensor

- 1 Scale (graduated steel strip with recesses)
- 2 Coils in the measuring sensor of the scanner
- 3 Primary coils
- 4 Secondary coils

The scale consists of a steel strip with recesses that have been made at equal distances using a precision process. The magnetic resistance (reluctance) of the individual magnetic fields between the primary and secondary coils varies as a function of their position relative to the scale.

Voltages induced in the secondary coils are further processed and transmitted as signals with a 90° phase shift. The evaluation electronics then determine the exact position and direction of motion from these signals.

3 Profiled rail systems

3.7 Integrated measuring system

3.7.3 Structural design

Function of the reference sensor and reference marks

When scanned, the scale itself delivers only ascending or descending numerical values (incremental signals). This incremental measuring method does not allow the absolute position of the measuring system to be detected.

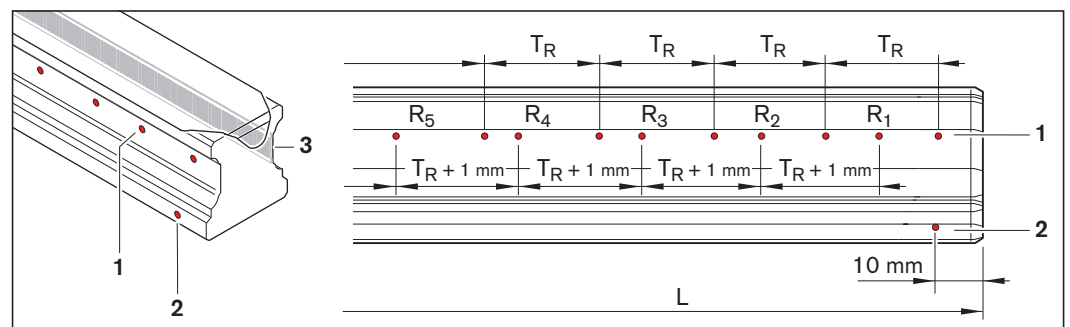
An additional reference is needed to determine the absolute position of the runner block on the rail. This can be provided in one of two ways:

- distance-coded reference marks
- single reference mark

Distance-coded reference marks

This reference marks are holes that are machined into the guide rail on the side opposite the scale and sealed to protect them from contamination. The distance coded reference marks supply a reference mark signal and are also protected by a tightly welded stainless steel strip. The side with reference marks is designated by a hole drilled

into the reference edge of the guide rail. The coding ensures that an absolute positioning signal is available as soon as the sensor on the runner block has passed two reference marks.



Distance-coded reference marks

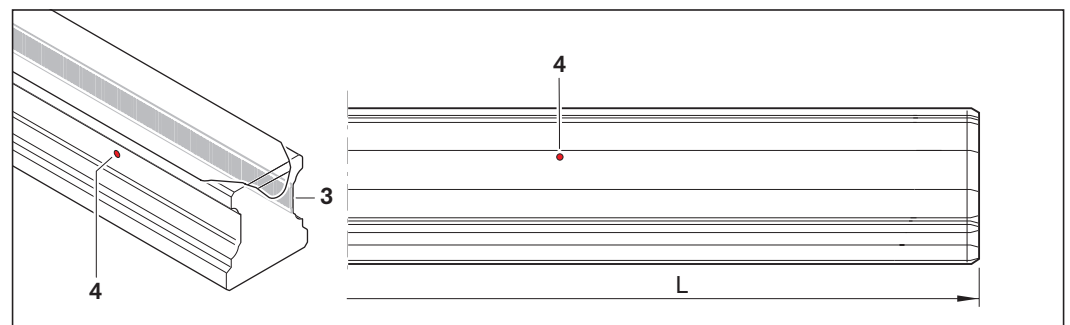
- 1 Distance-coded reference marks
- 2 Hole identifying the reference mark side
- 3 Scale on the opposite side

- $R_1 \dots R_n$ Reference marks
- T_R Distance between reference marks
- L Rail length

Single, absolute reference mark

A single, absolute reference mark is a hole that is machined into the guide rail on the side opposite the scale. It is closed with a brass pin to protect it from contamination and damage. This sensor must travel past this reference mark to detect the

position. The user is free to define the location of the reference mark anywhere within the measuring range.



Single, absolute reference mark

- 3 Scale on the opposite side
- 4 Single, absolute reference mark (at any desired position)

3 Profiled rail systems

3.7 Integrated measuring system

3.7.4 Electronics

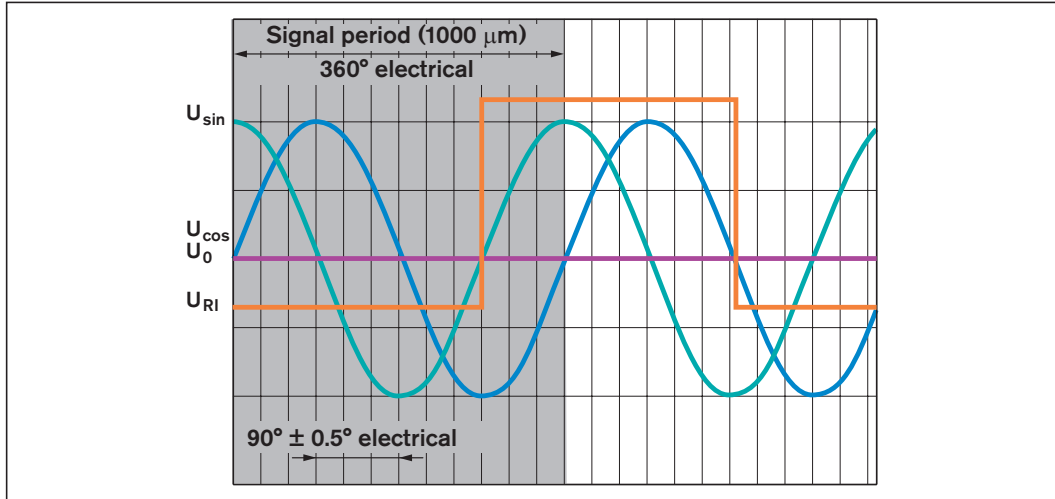
The electronic circuitry integrated in the scanner has both analog and digital functions, i.e. either analog or digital signals can be emitted,

as required. Both signal outputs are equipped with evaluation electronics in real-time mode so that highly dynamic drives can be served.

Analog signal shape

The non-contacting relative motion between the scanner and the scale generates sinusoidal

voltage signals (1 Vpp), supplied directly by the sensor during scanning.



Sinusoidal analog signals

U_{sin}/U_{cos} Sinusoidal voltage signals
 U₀ Passage through zero
 U₀ = 2.5 V ± 0.5 V

U_{RI} Reference mark signal

Interpolation

Intermediate values are obtained by offsetting the sine signals against the cosine signals. This process is called interpolation. Since the interpo-

lation unit is integrated in the scanner, no external interpolation unit is required.

Resolution

The measuring system can have different resolution rates, depending on the interpolation factor. The resolution determines the smallest possible measurable change in position of the measuring system. A scale pitch of 1000 μm and 4-edge evaluation of the signals in the control system results in the following resolution rates.

Integrated interpolation factors after 4-edge evaluation:

$$\text{Resolution} = \frac{\text{scale pitch}}{\text{evaluation} \cdot \text{factor}}$$

Factor	Calculation	Resolution
25x	$\frac{1000 \mu\text{m}}{4 \cdot 25} = 10 \mu\text{m}$	10 μm
50x	$\frac{1000 \mu\text{m}}{4 \cdot 50} = 5 \mu\text{m}$	5 μm
256x	$\frac{1000 \mu\text{m}}{4 \cdot 256} = 0.976 \mu\text{m} \approx 1 \mu\text{m}$	1 μm
1024x	$\frac{1000 \mu\text{m}}{4 \cdot 1024} = 0.244 \mu\text{m} \approx 0.25 \mu\text{m}$	0.25 μm

3 Profiled rail systems

3.7 Integrated measuring system

3.7.4 Electronics

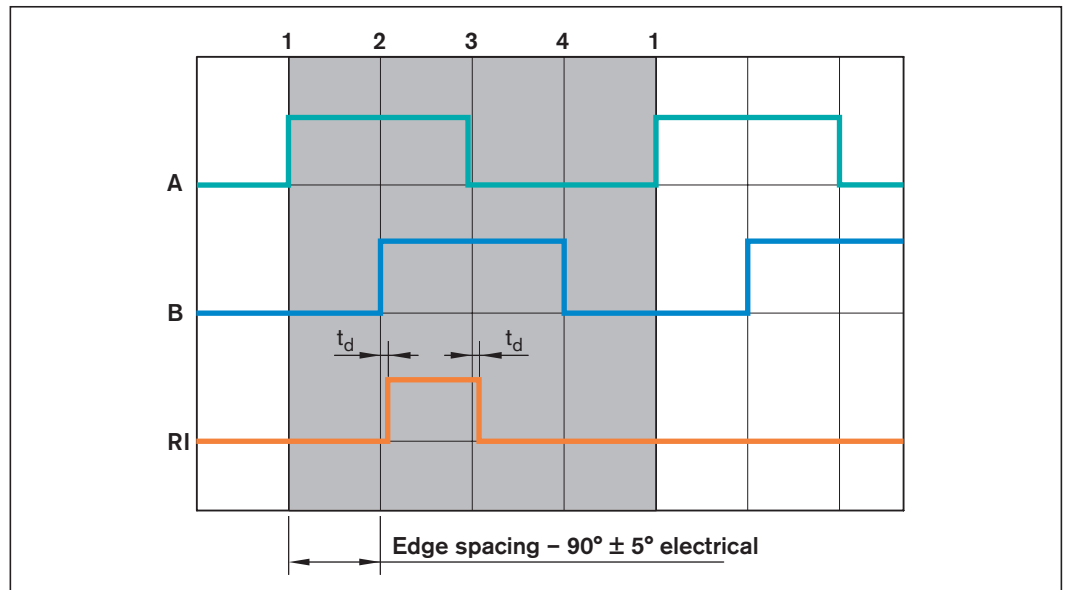
Interpolation accuracy

The interpolation accuracy is identical for all resolution rates and is $\pm 3 \mu\text{m}$.

Digital signals

After interpolation, the unit provides square-wave output signals (TTL signals). Square-wave output signals are digital signals.

As described above, the resolution rates for the TTL signals are $0.25 \mu\text{m}$, $1 \mu\text{m}$, $5 \mu\text{m}$ and $10 \mu\text{m}$.



Digital square-wave signals

- A/B Incremental TTL square-wave signals
- RI Reference mark signals
- t_d Time delay $|t_d| < 0.1 \mu\text{s}$

Repeatability

The different resolutions rates determine the repeatability of the system. The repeatability is the accuracy with which one and the same point can be repeatedly measured.

Resolution with TTL signal	μm	0.25	1	5	10
Repeatability	μm	2	2	5	10

3 Profiled rail systems

3.7 Integrated measuring system

3.7.5 Product selection guide

3.7.5.1 Accuracy of the measuring system

The accuracy of the measuring system is determined by the accuracy of the scale pitch and the guideway, as well as the accuracy of the interpolation. The sum of the deviations is summarized in the term system accuracy.

The system accuracy is the maximum deviation from the mean of any position over a measuring distance of 1 m, expressed in $\pm a$ (μm).

Scale pitch + guideway accuracy

Four different scale pitch and guideway accuracies are available. These values are guaranteed at an ambient temperature of 20°C.

The pitch accuracy is selected by stating the appropriate code in the guide rail part number. A detailed pitch accuracy report can be provided on request.

Scale pitch and guideway accuracy

 $\pm 3 \mu\text{m}$

 $\pm 5 \mu\text{m}$

 $\pm 10 \mu\text{m}$

 $\pm 30 \mu\text{m}$

Interpolation accuracy

The standard interpolation accuracy is $\pm 3 \mu\text{m}$.

Interpolation accuracy

 $\pm 3 \mu\text{m}$

System accuracy

The system accuracies are as follows:

Scale pitch and guideway accuracy	Interpolation accuracy	System accuracy (sum)
$\pm 3 \mu\text{m}$	$\pm 3 \mu\text{m}$	$\pm 6 \mu\text{m}$
$\pm 5 \mu\text{m}$	$\pm 3 \mu\text{m}$	$\pm 8 \mu\text{m}$
$\pm 10 \mu\text{m}$	$\pm 3 \mu\text{m}$	$\pm 13 \mu\text{m}$
$\pm 30 \mu\text{m}$	$\pm 3 \mu\text{m}$	$\pm 33 \mu\text{m}$

3.7.5.2 Application areas

Industry sector	Applications
Woodworking and wood processing machines	<ul style="list-style-type: none"> ■ Sawing machines ■ Handling equipment
Electrical/electronics industry	<ul style="list-style-type: none"> ■ PCB assembly machines
Metal-cutting machine tools	<ul style="list-style-type: none"> ■ Machining centers ■ Grinding machines ■ Milling machines
Forming and stamping machine tools	<ul style="list-style-type: none"> ■ Pressing, stamping
Textile technology	<ul style="list-style-type: none"> ■ Textile machines
Printing and paper industry	<ul style="list-style-type: none"> ■ Printing cylinder machines
Welding technology	<ul style="list-style-type: none"> ■ Laser welding lines

4 Linear bushings and shafts

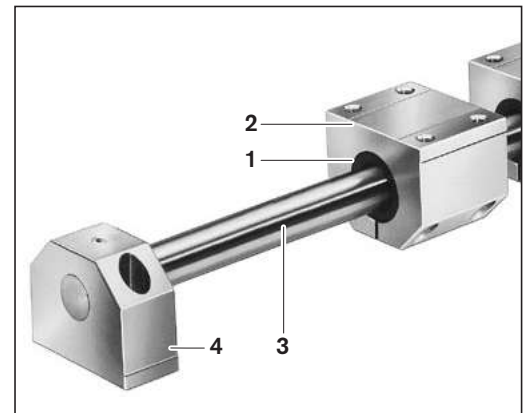
4.1 Principles

4.1.1 System technology

Linear bushing guideways offer economical solutions for executing linear movements. Available in a great variety of designs, they can be used in many different industrial applications.

A linear bushing guideway consists of:

- One or more linear bushings (1, 5)
- One or more precision steel shafts (3) for guiding the bushings
- A housing (2) for connecting the bushings to the adjacent structure
- Shaft support blocks (4) or shaft support rails for holding the precision steel shafts



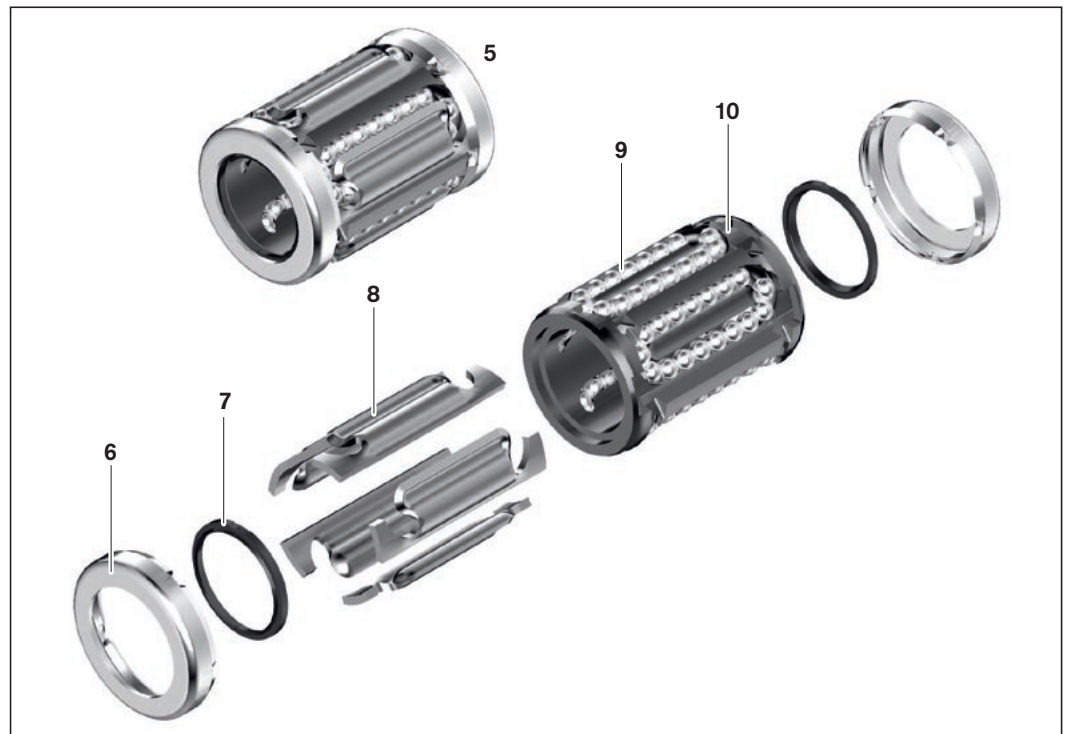
Main elements of a linear bushing guideway

- | | |
|------------------|-----------------------|
| 1 Linear bushing | 3 Steel shaft |
| 2 Housing | 4 Shaft support block |

4.1.1.1 Structural design of a linear bushing

Linear bushings comprise:

- A steel sleeve or several segmental steel load-bearing plates
- A steel or plastic ball retainer
- Balls made from anti-friction bearing steel
- Possibly, steel holding rings and seals, depending on the design



Example: Compact linear bushing

- | | |
|---------------------------|---------------------------------------|
| 5 Complete linear bushing | 8 Segmental steel load-bearing plates |
| 6 Holding ring | 9 Balls |
| 7 Seal | 10 Ball retainer |

4 Linear bushings and shafts

4.1 Principles

4.1.1 System technology

Ball retainer

The rows of balls circulate in closed circuits in the ball retainer. In the load-bearing zone (2), the balls rest directly on the shaft. At the end of the load-bearing zone, the balls are raised and conducted through the return zone (1) without any contact with the shaft. The ball retainer is not subjected to any external forces in this process.

Steel sleeve
Steel load-bearing plates
Balls

The steel sleeve or segmental load-bearing plates (3) transmit the forces applied from outside to the balls. Because of the high surface pressure at the point contact between the ball and the shaft or between the ball and the guiding surface, the individual components are hardened to at least 60 HRC. Corrosion-resistant steel shafts are hardened to 54 HRC.

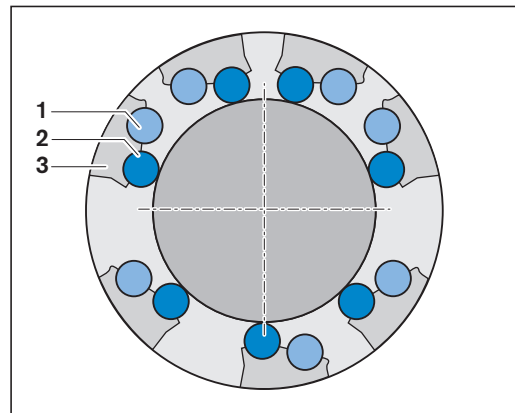
Seals
Holding rings

The seals protect the linear bushings from contamination and the holding rings keep the steel load-bearing plates in the desired position.

Main dimensions

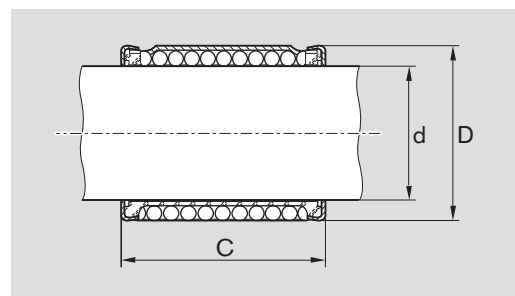
A linear bushing's main dimensions are described by:

- The shaft diameter d (bushing size)
- The outside diameter D
- Length C of the linear bushing



Ball recirculation in a linear bushing

- 1 Non-loaded row of balls
- 2 Load-bearing row of balls
- 3 Segmental steel load-bearing plate



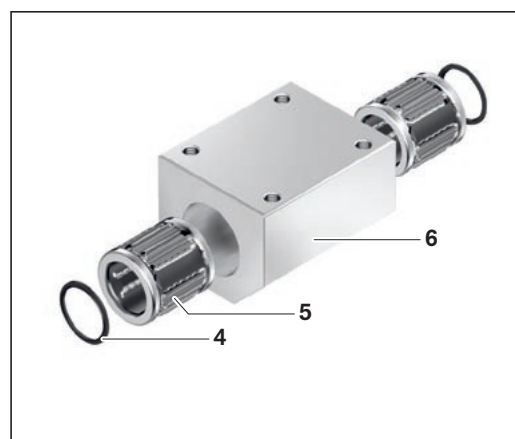
Main dimensions of a linear bushing

4.1.1.2 Structural design of a linear set

Linear bushings must be installed in housings for connection to the adjacent structure. Although this can be done with customer-built housings, which have to be specially designed and manufactured, it is generally simpler and cheaper to use complete standardized bushing units. Rexroth offers such units in the form of linear sets. These can easily be fixed to the adjacent structure.

Linear sets consist of:

- A housing with holes or threading for connection to the customer's application
- One or two linear bushings
- Seals



Elements of a linear set

- 4 Seal
- 5 Linear bushing
- 6 Housing

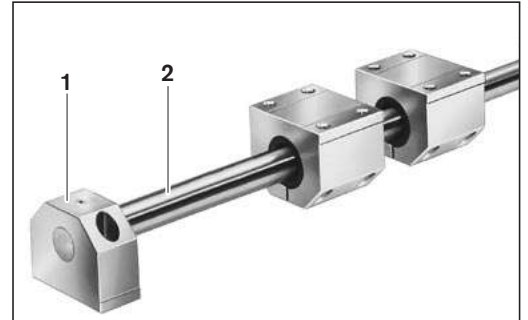
4 Linear bushings and shafts

4.1 Principles

4.1.1 System technology

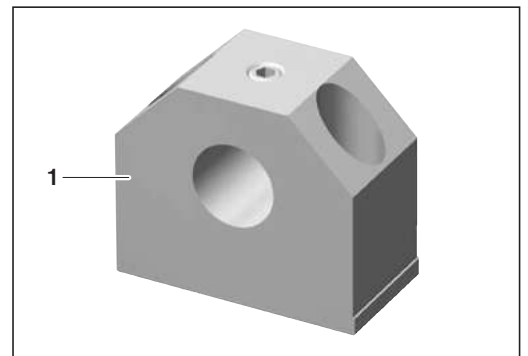
4.1.1.3 Structural design of shafts, shaft support blocks and shaft support rails

Precision steel shafts Precision steel shafts (2) are available as solid and tubular shafts. There are shaft support blocks (1) or shaft support rails (3) for holding the shafts. Just like the linear sets, these standardized units can significantly reduce installation time. No expensive joining structure is required because the shaft is simply fastened by screwing down the block or rail.



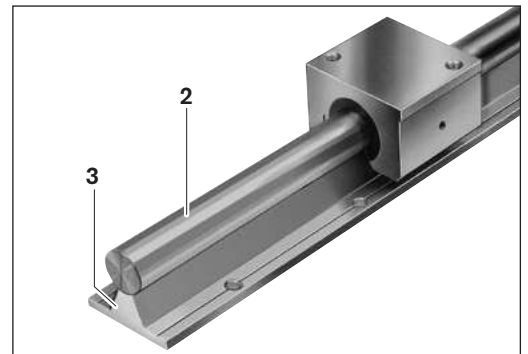
Linear bushing guideway with shaft support blocks

Shaft support blocks The precision steel shafts (2) can be fastened using shaft support blocks (1), without the need for any further processing. The shaft is slid into the bore in the shaft support block and fixed with a set screw.



Example: Aluminum shaft support block with machined reference edge

Shaft support rails The use of shaft support rails (3) prevents shaft deflection. To fasten the shaft to the shaft support rail, radial threaded holes must be made in the shaft along its entire length to accommodate the fixing screws for joining the two parts.



Linear bushing guideway with shaft support rail

- 1 Shaft support block
- 2 Precision steel shaft
- 3 Shaft support rail

4.1.1.4 Standards

Linear bushings and shafts are linear ball bearings whose boundary dimensions, tolerances and definitions are specified in ISO 10285.

ISO 13012 describes the accessories for linear ball bearings. These accessories are bearing housings, shafts, shaft support blocks and shaft support rails.









4 Linear bushings and shafts

4.1 Principles

4.1.1 System technology

4.1.1.5 Type designations and forms of linear bushings

There are different designs, versions and sizes and different sealing systems for each type of linear bushing. The wide variety of available linear bushings provides ample choice for each individual application. The table below gives an overview of Rexroth's linear bushings.

Type of linear bushing		Construction forms	Sealings	Versions	Characteristics
Compact linear bushing eLINE linear bushing See section 4.2.1		<ul style="list-style-type: none"> ■ Closed-type 	<ul style="list-style-type: none"> ■ Without seals ■ Integral wiper seals ■ Separate seals 	<ul style="list-style-type: none"> ■ Also available in corrosion-resistant version ¹⁾ 	<ul style="list-style-type: none"> ■ Compact design ■ No additional axial retention necessary ■ Low-cost
Super linear bushing A See section 4.2.2		<ul style="list-style-type: none"> ■ Closed-type ■ Open-type 	<ul style="list-style-type: none"> ■ Without seals ■ Integral wiper seals ■ Separate seals 		<ul style="list-style-type: none"> ■ Self-aligning
Super linear bushing B See section 4.2.2		<ul style="list-style-type: none"> ■ Closed-type ■ Open-type 	<ul style="list-style-type: none"> ■ Without seals ■ Integral wiper seals ■ Separate seals 		<ul style="list-style-type: none"> ■ Without self-alignment
Standard linear bushing See section 4.2.3		<ul style="list-style-type: none"> ■ Closed-type ■ Adjustable ■ Open-type 	<ul style="list-style-type: none"> ■ Without seals ■ Integral wiper seals 	<ul style="list-style-type: none"> ■ Also available in corrosion-resistant version ¹⁾ 	<ul style="list-style-type: none"> ■ Exceptionally robust ■ For high temperatures ■ All-metal design
Segmental linear bushing See section 4.2.4		<ul style="list-style-type: none"> ■ Closed-type 	<ul style="list-style-type: none"> ■ Without seals ■ Separate seals 	<ul style="list-style-type: none"> ■ Also available in corrosion-resistant version ¹⁾ 	<ul style="list-style-type: none"> ■ Low-cost
Super linear bushing H See section 4.2.5		<ul style="list-style-type: none"> ■ Closed-type ■ Open-type 	<ul style="list-style-type: none"> ■ Without seals ■ Integral wiper seals ■ Fully sealed 		<ul style="list-style-type: none"> ■ For high loads ■ Self-aligning
Super linear bushing SH See section 4.2.5		<ul style="list-style-type: none"> ■ Closed-type ■ Open-type 	<ul style="list-style-type: none"> ■ Without seals ■ Integral wiper seals ■ Fully sealed 		<ul style="list-style-type: none"> ■ For very high loads ■ Self-aligning
Radial linear bushing See section 4.2.6		<ul style="list-style-type: none"> ■ Open-type 	<ul style="list-style-type: none"> ■ Without seals ■ Fully sealed 		<ul style="list-style-type: none"> ■ For high loads ■ For high rigidity
Torque-resistant linear bushing See section 4.2.7		<ul style="list-style-type: none"> ■ Closed-type 	<ul style="list-style-type: none"> ■ Without seals ■ Separate seals 	<ul style="list-style-type: none"> ■ With 1, 2 or 4 ball guide grooves 	<ul style="list-style-type: none"> ■ For torque transmission
Linear bushing for combined linear and rotary motion See section 4.2.8		<ul style="list-style-type: none"> ■ Closed-type 	<ul style="list-style-type: none"> ■ Integral wiper seals 	<ul style="list-style-type: none"> ■ With deep-groove ball bearings or needle bearings 	<ul style="list-style-type: none"> ■ For combined linear and rotary motion

1) Corrosion-resistant to EN 10088

4 Linear bushings and shafts

4.1 Principles

4.1.1 System technology

Bushing designs

Depending on the type of linear bushing, either a closed, an open or an adjustable form can be chosen. Open-type linear bushings must be employed when using shaft support rails to prevent shaft deflection.

For standard linear bushings, there is an adjustable version for setting the radial clearance. For all other linear bushings, the radial clearance can be adjusted by the choice of fit between the shaft and the bore.



Closed-type linear bushing



Open-type linear bushing



Adjustable linear bushing

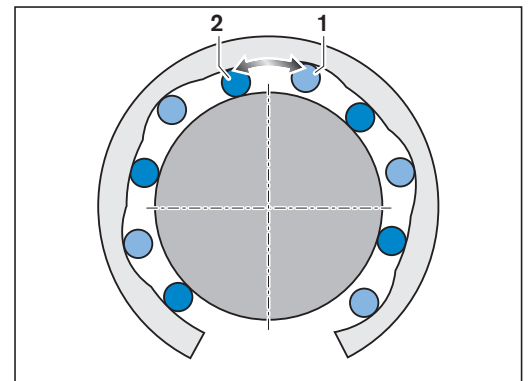
Ball recirculation

The type of ball recirculation is an important distinguishing feature of linear bushings. This has a direct effect on the linear bushing's load capacity and its overall dimensions.

Tangential recirculation

In tangential recirculation, the balls are returned to the load-bearing zone from the side. These linear bushings are distinguished by their small space requirement (small outside diameter). This group comprises:

- Compact and eLINE linear bushings
- Super linear bushings
- Standard linear bushings
- Segmental linear bushings
- Torque-resistant linear bushings
- Linear bushings for combined linear and rotary motion



Standard linear bushing with tangential recirculation for a 30mm diameter shaft, load capacity $C = 2890 \text{ N}$

Radial recirculation

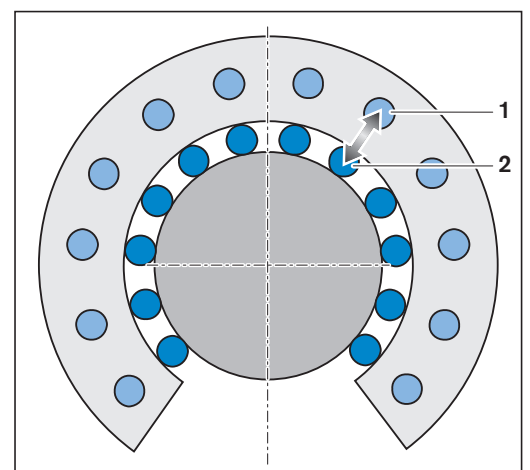
In radial recirculation, the ball return channel is located above the load-bearing zone.

This construction principle permits a larger number of load-bearing rows of balls for the same shaft diameter and therefore higher load capacities. This group comprises:

- Radial linear bushings

Further distinguishing features

In addition to these distinguishing features, there are different kinds of sealing systems, and some linear bushings also come in corrosion-resistant versions.



Linear bushing with radial recirculation for a 30mm diameter shaft, load capacity $C = 8500 \text{ N}$

- 1 Non-loaded row of balls
- 2 Load-bearing row of balls

4 Linear bushings and shafts

4.1 Principles

4.1.2 Product selection

4.1.2.1 Linear bushing applications

Linear bushings can be used in many areas. They are better suited than other linear guides for the following applications:

- For self-supporting guides, i.e. supported at the ends only
- To compensate for unevenness in the mounting base, e.g. unmachined welded constructions (through the degree of freedom in the circumferential direction and linear bushings with a rocker effect)
- For maintenance-free guides (linear bushings require hardly any lubrication)
- For linear guides requiring low friction
- For integrated versions (the linear bushing is pressed into the part to be moved instead of screwed down on the outside)
- For corrosive environments (corrosion-resistant versions)
- For the food processing, chemical, pharmaceutical and medical industries and other sectors where aggressive media are used for cleaning
- In extremely harsh environments, e.g. brick and cement factories, woodworking (robust standard linear bushings, all-steel version without any plastic components)
- At high temperatures far above 100 °C, e.g. foundries (all-metal version of the standard linear bushing)
- For applications under vacuum (linear bushings without plastic components)
- For combined linear and rotary motion (linear bushings with installed ball or needle bearing)
- For rotary applications (guides with rotational symmetry)
- For concealed routing of sensor cables, compressed air hoses, etc., through tubular shafts
- For easy attachment of peripherals to shaft ends with customer-specific machining
- For extremely long guides with composite shafts, allowing travel across the joints under full load









4 Linear bushings and shafts

4.1 Principles

4.1.2 Product selection

4.1.2.2 Linear bushing characteristics and technical data

There is a wide variety of different requirements for linear bushings and shafts. The following tables provide assistance in choosing linear bushings.

Requirements	Type of linear bushing							
	Compact/ eLINE	Super A/B	Standard	Seg- mental	Super H/SH	Radial	Torque- resistant	Comb. lin- ear and ro- tary motion
								
Frequency of use	+++	+++	++	++	+	+	+	+
Low costs	+++	++	++	+++	+	+	+	+
Especially easy installation	+++	++	++	++	+	+	+	++
Very compact design	+++	+	+	+++	+	o	+	+
Corr.-resistant version available	+++	o	+++	+++	o	o	o	o
High loads	+	++	+	+	+++	+++	++	+
Self-alignment	o	+++ ¹⁾	o	o	+++	o	o	o
Especially smooth running	++	+++	++	++	+	++	++	++
High temperature > 100 °C	o	o	+++	o	o	o	o	o
Heavy contamination	o	o	+++	o	o	o	o	o
Damp/wet environment	++	o	+++	++	++	o	o	o
With aqueous metalworking fluids	++	o	+++	++	o	o	o	o
Suitability for vacuum	o	o	+++	o	o	o	o	o
Torque transmission	o	o	o	o	o	o	+++	o
Comb. linear and rotary motion	o	o	o	o	o	o	o	+++









1) Super A only

+++ Very good

++ Good

+ Satisfactory

o Adequate (not recommended)

Technical data		Type of linear bushing							
		Compact/ eLINE	Super A/B	Standard	Seg- mental	Super H/SH	Radial	Torque- resistant	Comb. lin- ear and ro- tary motion
									
Dyn. load capacity C_{max} ¹⁾	N	5680	12 060	21 000	3870	23 500	54 800	9 250 ⁴⁾	21 000
Diameter d	mm	8 to 50	10 to 50	3 to 80	12 to 40	20 to 60	30 to 80	12 to 50	5 to 80
Friction coefficient μ ²⁾	–	0.001 to 0.004	0.001 to 0.004	0.001 to 0.0025	0.001 to 0.004	0.001 to 0.004	0.001 to 0.002	0.001 to 0.004	0.001 to 0.0025
Velocity v_{max}	m/s	5	3	2.5	3	5	2	3	2.5
Acceleration a_{max}	m/s ²	150	150	100	150	150	50	150	100
Operating temperature	°C	–10 to 100 ³⁾							

1) The load capacity is dependent on the direction of loading. See section 4.1.3.1.

2) Values without seals. The friction coefficient is lowest under high load. Under low loads, it can be even higher than the value shown.

3) Standard linear bushings without seals can also be used at temperatures above 100 °C.

4) The value applies to torque-resistant linear bushings with 1 or 2 ball guide grooves. The version with 4 ball guide grooves has a capacity of up to 36600 N.

4 Linear bushings and shafts

4.1 Principles

4.1.2 Product selection

4.1.2.3 Application parameters

The following application requirements must be defined before beginning linear bushing product selection:

- Necessary rigidity
- Dimensions
- Loads
- Direction of loading
- Customer-built housing or ready-to-install linear set
- Velocity range
- Rotary motion (degrees of freedom)
- Ambient conditions (dirt, humidity, etc.)
- Price

These parameters are necessary to determine the appropriate linear bushings for the application.

4.1.2.4 Selection of appropriate linear bushings

Deflection

When deflection must be avoided, high rigidity is required and shaft support rails and the appropriate open-type linear bushings are necessary. If deflection is less important, more economical closed-type linear bushings (possibly with shaft support blocks) can be used.

Vertical space requirement

The available construction height and the height of the load carried will limit the size and the choice of linear bushing.

Direction of loading

If the direction of loading is different from the main direction of loading, the load capacities of the linear bushing will be reduced. The deviation from the main direction has to be taken into account by applying a reduction factor in the design calculations.



Shaft support rails

4 Linear bushings and shafts

4.1 Principles

4.1.2 Product selection

Adjacent structure

It is generally best to use ready-to-install linear sets because they save time and money during design and installation. When installing linear bushings in customer-built housings, please follow the recommendations in the product catalog.

Travel speed

Not all types of linear bushings can be used when the velocity required is greater than 2 m/s.

Preventing rotary motion

If rotary motion must be prevented in linear bushing guideways with only one shaft, torque-resistant linear bushings must be used.

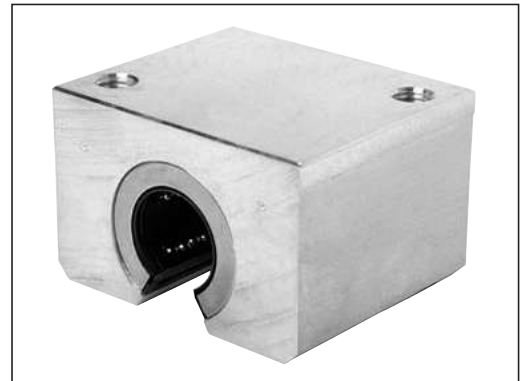
Environmental conditions

Some types of linear bushings are better suited than others for particular environmental conditions because of their design. There are different sealing systems or corrosion-resistant versions to cater for different applications.

Initial selection

An initial selection of appropriate linear bushings can be made by analyzing these parameters. The tables in sections 4.1.1.5 and 4.1.2.2 also provide assistance in this preselection process. Normally, several types of linear bushings may be suitable for a particular application. If the type of linear bushing cannot be determined because of special conditions, you can always rely on Rexroth's many years of experience.

After selecting the type of linear bushing, the next step is to perform the design calculations.



Linear set



Torque-resistant linear bushing

4 Linear bushings and shafts

4.1 Principles

4.1.3 Design notes

To achieve trouble-free operation of linear bushing guides it is essential to follow the advice given below.

4.1.3.1 Influence of the direction of loading on the load capacity

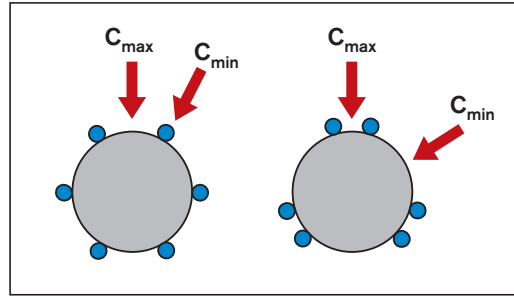
Effective load capacity

In linear bushings, the direction of loading determines the effective load capacity. This depends on the orientation of the direction of load application relative to the position of the rows of balls.

Dynamic load capacity C
Static load capacity C₀

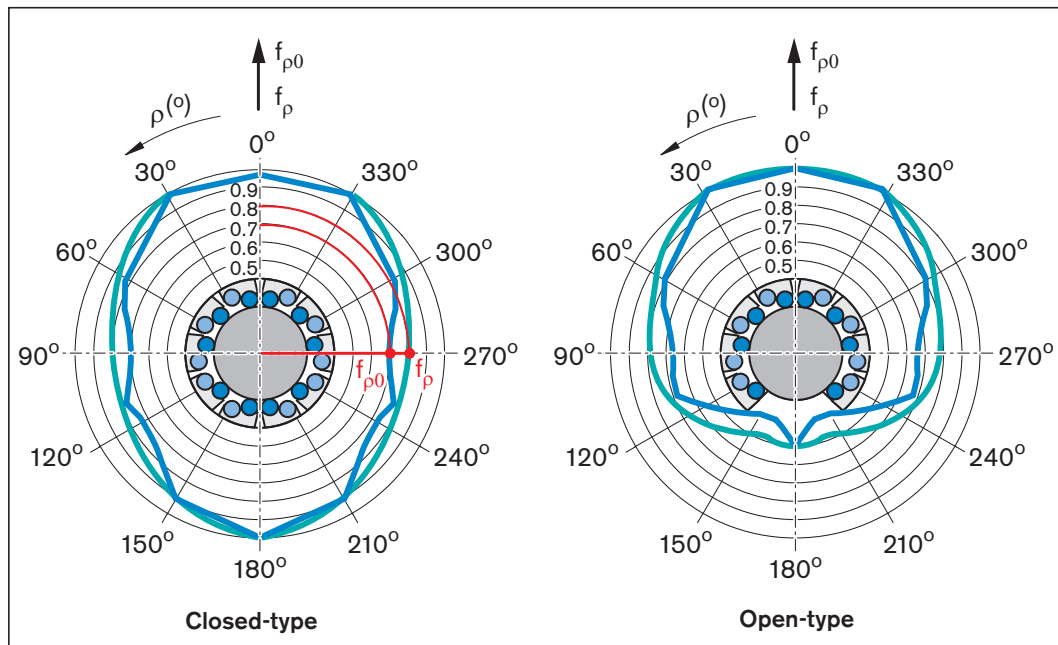
For each direction of load application, the maximum load capacities from the product catalog are multiplied by the factor f_p (dynamic load capacity C) or f_{p0} (static load capacity C_0) in order to obtain the effective load capacity. Directions of load application for which the maximum load capacity C_{max} applies are called main directions of loading. The load capacity can be optimally utilized by correctly aligning the components during installation. If aligned installation is impossible or if the direction of loading is not defined, the minimum load capacities C_{min} apply. The corresponding load capacities are given in the Rexroth product catalog.

Main direction of loading



Examples of maximum and minimum effective load capacities depending on the arrangement of the rows of balls

The following example shows the effect of the direction of loading and the related f_p and f_{p0} factors on a closed-type and an open-type Super Linear Bushing SH.



Super Linear Bushing SH, 20 to 25mm diameter shafts

— Factor f_p for dynamic load cases

— Factor f_{p0} for static load cases

Sample reading: For a direction of loading of $\rho = 270^\circ$, the dynamic load capacity C must be multiplied by $f_p = 0.8$ for closed-type Super

Linear Bushings SH with 20 to 25mm diameter shafts. The static load capacity C_0 must be multiplied by a factor f_{p0} of 0.68.

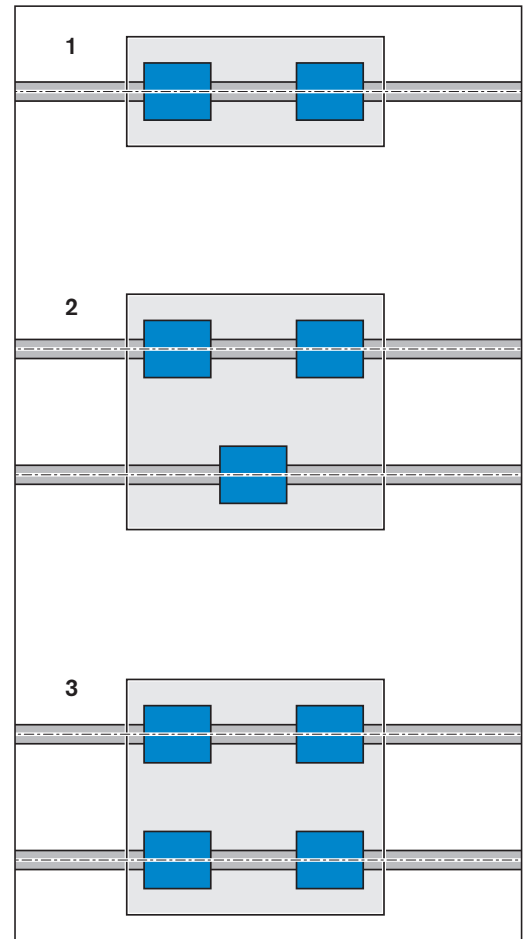
4 Linear bushings and shafts

4.1 Principles

4.1.3 Design notes

4.1.3.2 Design measures

Number of linear bushings	In guideways with only one shaft, two linear bushings should be used. If there are two shafts, at least one of the shafts should be fitted with two linear bushings.
Parallelism in the case of two shafts	High demands are placed on the accuracy of the spacing between two shafts and their respective linear bushings and on the parallelism of the shafts and the linear bushings. Inaccuracies lead to distortive stresses or overloading and therefore reduce the service life of the linear bushing guide. The product catalog gives recommended values for the maximum difference in spacing, including parallelism offsets.
Operating temperatures	Linear bushings and shafts are essentially suitable for a temperature range of -10 °C to $+100\text{ °C}$. For standard linear bushings with integral wiper seals and radial linear bushings with separate end seals, the maximum operating temperature is reduced to $+80\text{ °C}$ (with brief peaks up to $+100\text{ °C}$). Higher temperatures are permitted for unsealed standard linear bushings. However, temperatures over $+100\text{ °C}$ result in a reduction in load capacity. At operating temperatures below freezing point, the formation of ice must be avoided.
Shaft deflection	In assemblies with rigid adjacent structures (housings, etc.) and longer distances between shaft support points, shaft deflection and the resultant pressure between the bushing edge and the shaft reduces the service life of self-supporting guides. The exceptions are Super Linear Bushings A, H and SH with inertial error compensation up to 0.5° . Please refer to the notes in section 4.1.4.7 when calculating shaft deflection.
Corrosion-resistant versions	Corrosion-resistant steels are steels to EN 10088. In very critical, corrosive environments, the parts must be checked under operating conditions. Appropriate preserving oils and lubricants must be used.



Installation recommendations for guideways:

- 1 One shaft and two linear bushings
- 2 Two shafts and three linear bushings
- 3 Two shafts and four linear bushings

4 Linear bushings and shafts

4.1 Principles

4.1.3 Design notes

Radial clearance

For all linear bushings, the radial clearance can be adjusted by the choice of shaft and bore tolerances (except for closed-type standard linear bushings). It depends on the nominal diameter and the choice of fit.

For the normal radial clearance, the housing bore is machined to tolerance class H7. For guides with reduced radial clearance, the bores are produced in tolerance classes K7 or K6. Tolerance classes M7 and M6 are appropriate for light preloads. Preloading can reduce the theoretical life of the linear bushings and the shafts.

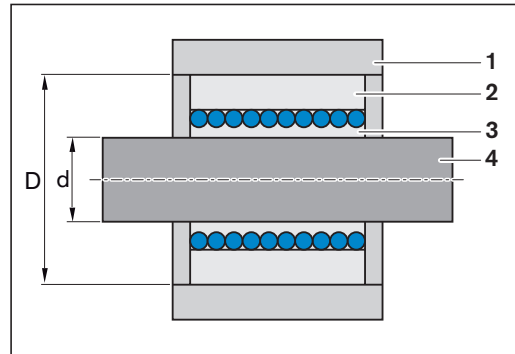
Radial clearance values for linear bushings and linear sets are given in the product catalog. These values were determined statistically and correspond to the values to be expected in practice.

Zero-clearance guideways

For zero-clearance guides, the linear bushing's radial clearance must be reduced by means of an adjusting screw in the housing until a slight resistance is felt when the shaft is turned. In applications subject to vibrations, the adjusting screw must be appropriately secured.

Preloading

If negative clearance (preloading) is required, we recommend that zero clearance should first be established using a dummy shaft whose diameter is smaller by the amount of the desired preload than the actual guide shaft on which the linear bushing is to run. For closed-type standard linear bushings, the relevant adjustable version should be used if the radial clearance needs to be set.

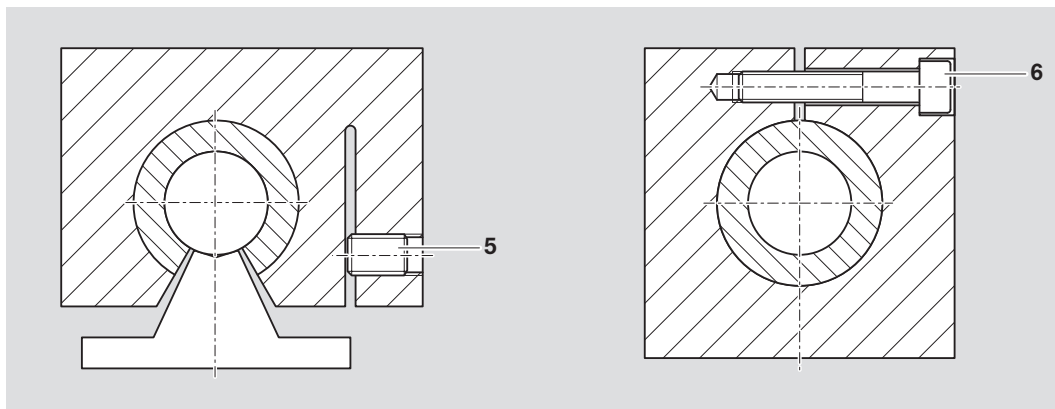


Definition of radial clearance

Operating clearance	Tolerance class	
	Shaft d	Housing bore D
Clearance according to product catalog	h7	H7
Transition range	h7	K7
Slight preload	h7	M7

Selection of the fit

- 1 Housing
- 2 Linear bushing
- 3 Radial clearance
- 4 Shaft
- 5 Adjusting set screw
- 6 Adjusting screw



Examples for adjustable linear bushings

4 Linear bushings and shafts

4.1 Principles

4.1.3 Design notes

Vertical dimensions The product catalog provides the tolerance values for the heights of the linear sets. These tolerance values were determined statistically and correspond to the values to be expected in practice.

Installation in customer-built housings

When installing in a customer-built housing, the edge of the housing bore must be chamfered. Small linear bushings (except for Compact and eLINE linear bushings from size 12 up) can be mounted by hand. For linear bushings with larger diameters and for Compact and eLINE linear bushings, Rexroth recommends the use of a mounting tool. Under no circumstances should pressure be exerted on the wiper seals and steel holding rings (Standard linear bushings) because this might damage the ball retainers.

Mounting tool

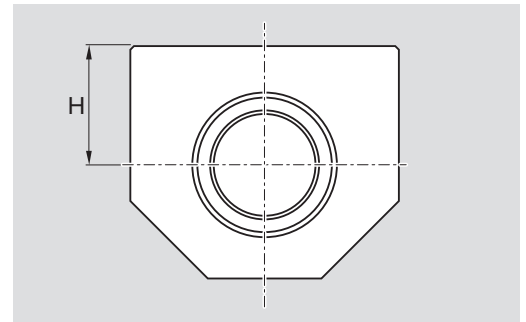
If a Compact or an eLINE linear bushing is slightly skewed on entering the housing bore, it will align itself as it is inserted further. Removal and re-alignment are unnecessary.

Retention

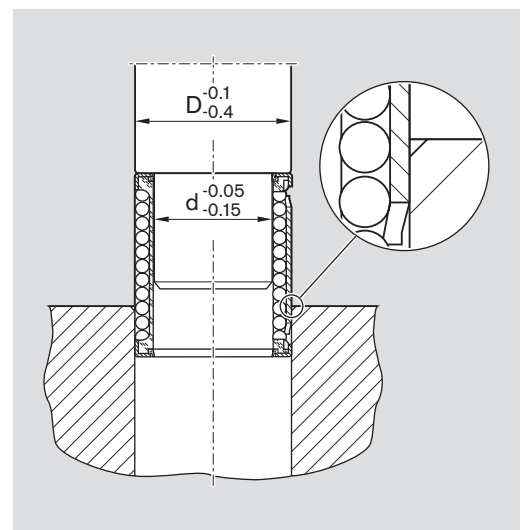
For retention of linear bushings in the housing, there are various aspects to be considered depending on the type of linear bushing. The product catalog provides the corresponding advice for each type of linear bushing under the heading "Customer-built housing."

Sliding the bushing onto the shaft

Rexroth's precision steel shafts are supplied with chamfered shaft ends. The chamfering is required for sliding the linear bushing onto the shaft and protects the linear bushing's end seals from damage. The linear bushing must not be misaligned when sliding it onto the shaft. Hammer blows can damage the linear bushing's sleeve, holding rings or ball retainer. Linear bushings with seals should not be pushed over sharp edges on the shaft as this can cause damage to the lips of the seals.



Vertical dimension



Installation in a customer-built housing using a mounting tool; the edge of the housing bore must be chamfered.

4 Linear bushings and shafts

4.1 Principles

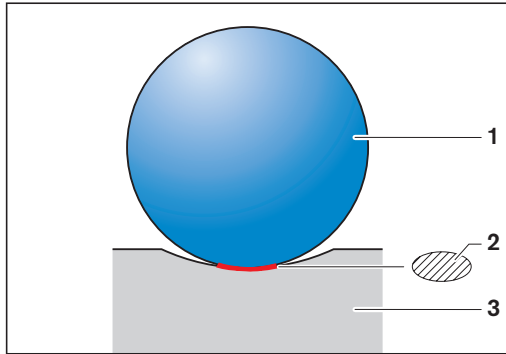
4.1.3 Design notes

4.1.3.3 Lubrication

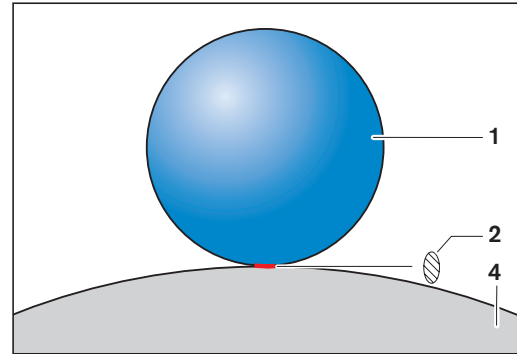
Low lubrication requirement

Linear bushings are preferably lubricated with grease (Dynalub). Grease helps to seal the linear bushing and adheres to its inside surfaces. In-service lubrication is only required at long intervals.

Unlike ball rail systems and ball screw drives, linear bushings and shafts require significantly less lubricant because of the smaller contact area between the rolling elements and the running track.



Large contact area in ball rail systems and ball screw drives

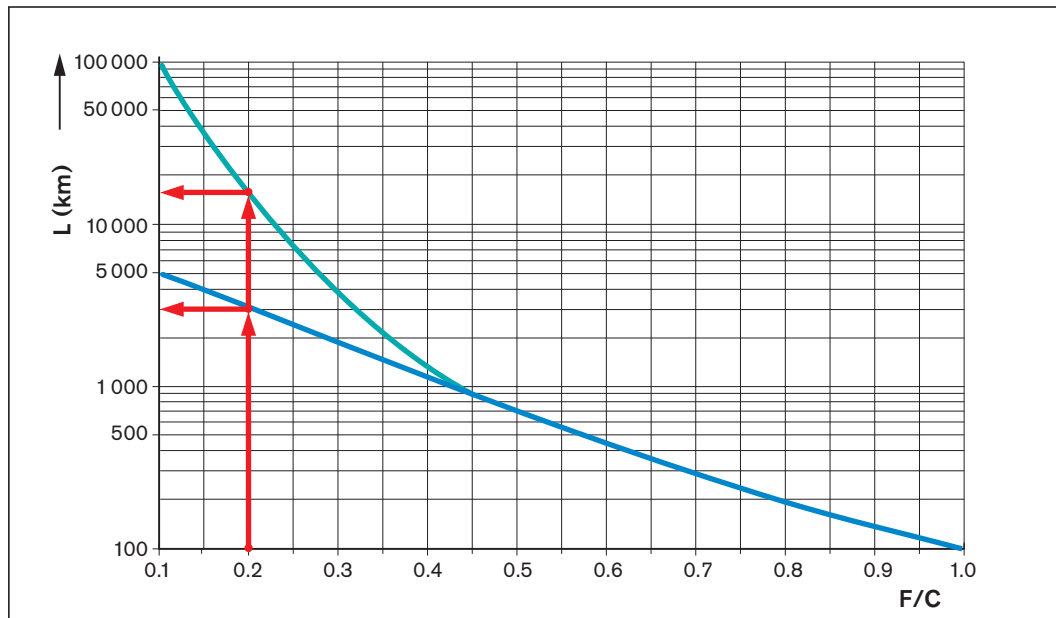


Small contact area in linear bushings and shafts

- 1 Ball
- 2 Contact area

- 3 Running track
- 4 Shaft

Recommended values for load-dependent in-service lubrication



Recommended values for load-dependent in-service lubrication, taking an eLINE linear bushing as an example

- With initial lubrication (no in-service lubrication)
- With regular in-service lubrication

F/C = load/dynamic load capacity (-)
L = service life (km)

Example with in-service lubrication

If an eLINE linear bushing is loaded with 20% of the dynamic load capacity, its service life under test conditions is 3000 km with initial lubrication

only. With regular in-service lubrication, a service life of 15000 km can be achieved.

4 Linear bushings and shafts

4.1 Principles

4.1.3 Design notes

Lubrication intervals

In order to achieve long lubrication intervals, it is essential to perform the initial lubrication carefully and regularly check the lubrication status. The lubricants, quantities and lubrication intervals are influenced by many factors, for instance:

- Loading
- Travel speed
- Motion sequence
- Temperature

The following factors reduce the lubrication intervals:

- High loads
- High speeds
- Short strokes
- Low resistance to aging in the lubricant

General lubrication principles can be found in Chapter 2, section 2.2.5.4.

4 Linear bushings and shafts

4.1 Principles

4.1.4 Calculations

4.1.4.1 Nominal life

Nominal life calculation basis

The basis for calculating the nominal life is the dynamic load capacity. The dynamic load capacity is determined based on a distance traveled of 100 km. If 50 km are used as the basis, the C values in the tables in the product catalog must be multiplied by 1.26 (see Chapter 2, section 2.4.1.2). Calculations in accordance with ISO 14728 are only valid under the following conditions:

- $F \leq 0.5 C$
- $F \leq C_0$

- F = load (N)
- C = dynamic load capacity (N)
- C_0 = static load capacity (N)

Extended nominal life calculation

When using shafts with a hardness of less than 60 HRC at operating temperatures over 100 °C or in short stroke applications, the calculations may deviate from the specifications given in ISO 14728.

For these applications, the following formulas should be used:

(4-1)
$$L = \left(\frac{C}{F_m} \cdot f_H \cdot f_t \cdot f_w \right)^3 \cdot 10^5 \text{ m}$$

(4-2)
$$L_h = \frac{L}{2 \cdot s \cdot n \cdot 60}$$

- L = nominal life (m)
- L_h = nominal life (h)
- C = dynamic load capacity (N)
- F_m = equivalent dynamic load (N)
- f_H = shaft hardness factor (-)
- f_t = temperature factor (only relevant at temperatures above 100 °C) (-)
- f_w = short stroke factor (for Segmental, Compact, eLINE and Super linear bushings only) (-)
- s = length of stroke (m)
- n = stroke repetition rate (full cycles) (min^{-1})

Shaft hardness factor

Shaft hardness plays an important role in calculating the nominal life. The effect of the shaft's hardness is taken into account in the nominal life formula via the hardness factor f_H . In shafts with a minimum hardness of 60 HRC, the f_H value is 1, i.e. the service life of the linear bushing guide is not limited by the shaft's hardness. All Rexroth shafts made from heat-treatable steels have a minimum hardness of 60 HRC. Rexroth's corrosion-resistant steel shafts have a minimum hardness of 54 HRC. This corresponds to a hardness factor of $f_H = 0.68$ (see example). In the case of customer-fabricated shafts, the factor must be taken into account as shown in the diagram opposite.

Example:
For a shaft made from X46Cr13 with a minimum hardness of 54 HRC, the hardness factor is $f_H = 0.68$.

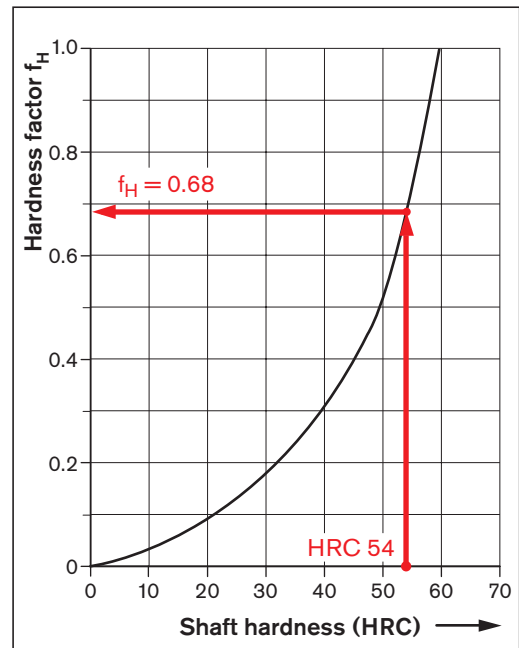


Chart for hardness factor f_H

4 Linear bushings and shafts

4.1 Principles

4.1.4 Calculations

Temperature factor

High operating temperatures cause permanent changes in the microstructure of the hardened steel. This reduces the hardness of the material. The resulting reduction in load capacity is taken into account by the temperature factor f_t .

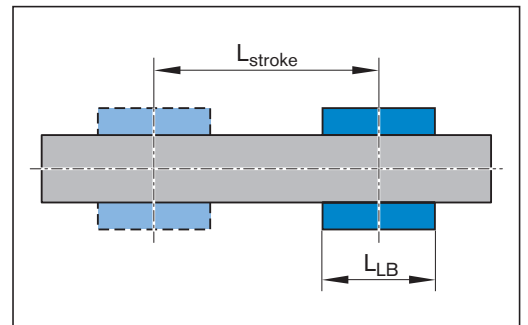
Bearing temperature	100 °C	125 °C	150 °C	175 °C	200 °C
Temperature factor f_t	1	0.92	0.85	0.77	0.70

Short stroke factor

For linear bushings, a short-stroke application is defined as an application where the stroke length L_{stroke} is less than three times the length of the linear bushing L_{LB} .

In short-stroke applications, the life of the shaft is shorter than that of Segmental, Compact, eLINE and Super linear bushings. The load capacities C for these types of linear bushings must be multiplied by the short stroke factor f_W . The product catalog contains charts for determining the f_W factor for each of these linear bushing types.

If the application does not involve short strokes or in the case of other linear bushing types, this problem can be ignored, i.e. the short stroke factor is $f_W = 1$.



Stroke length and linear bushing length

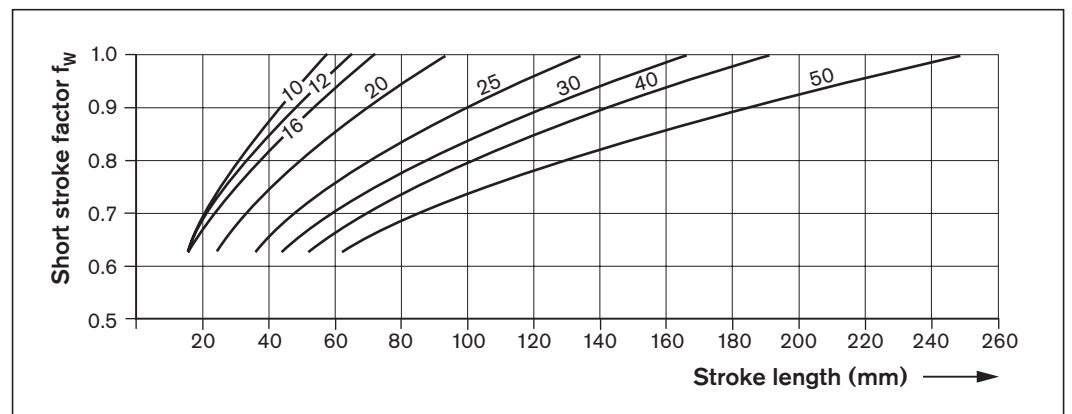


Chart for determining the short stroke factor using Super linear bushings A and B (sizes 10 to 50) as an example

4 Linear bushings and shafts

4.1 Principles

4.1.4 Calculations

Required load capacity

For design and product selection purposes (see section 4.1.2) the required load capacity can be calculated using the following formula:

$$(4-3) \quad C = \frac{F_m}{f_H \cdot f_t \cdot f_w \cdot f_L}$$

- C = dynamic load capacity (N)
- F_m = equivalent dynamic load (N)
- f_H = shaft hardness factor (-)
- f_t = temperature factor (-)
- f_w = short stroke factor (for Segmental, Compact, eLINE and Super linear bushings only) (-)
- f_L = life expectancy factor (dependent on the required life) (-)

This value can be used to determine the size and type of linear bushing guide. The actual nominal life calculation can be performed once the linear bushing type and the load capacity have been determined.

Effect of the required life

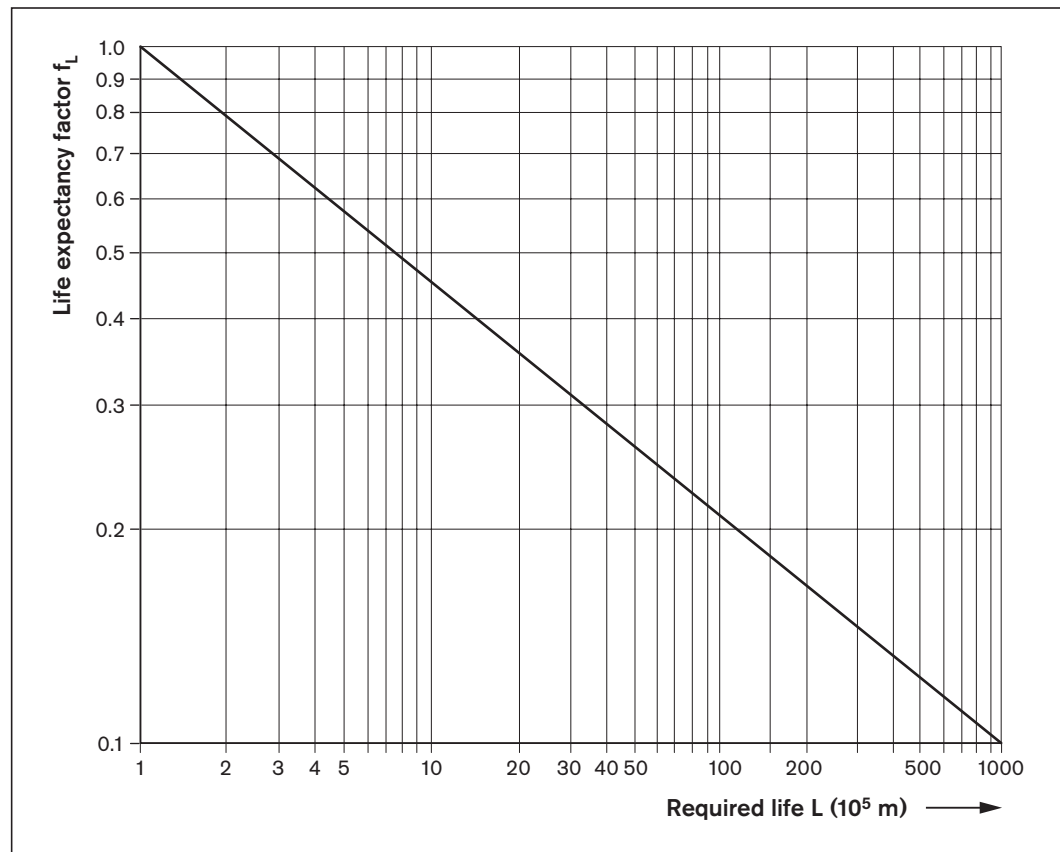


Chart for the life expectancy factor f_L

4 Linear bushings and shafts

4.1 Principles

4.1.4 Calculations

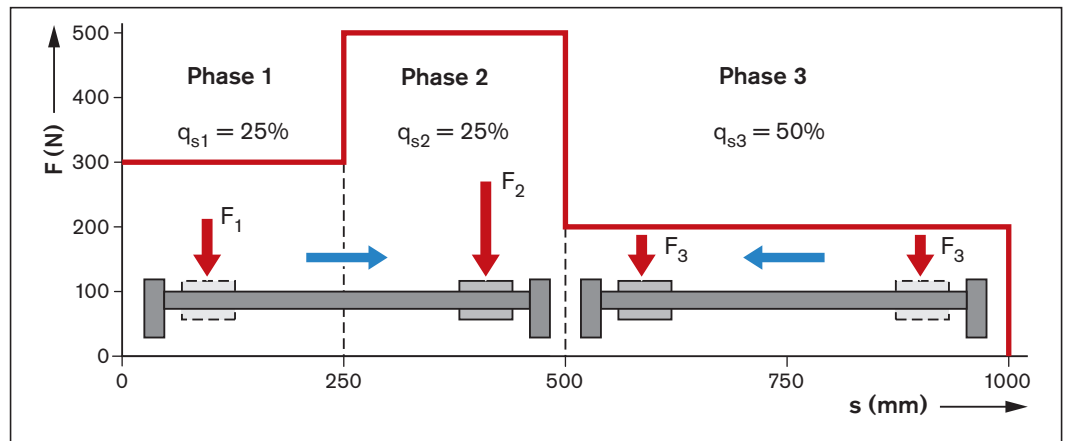
4.1.4.2 Equivalent dynamic load on bearing

Varying bearing loads for the same direction of loading

When there are varying bearing loads in the same direction of loading, the equivalent dynamic load F_m is calculated as follows, using formula (4-4):

F_m	= equivalent dynamic load	(N)
$F_1 \dots F_n$	= discrete dynamic load steps	(N)
$q_{s1} \dots q_{sn}$	= discrete travel steps for $F_1 \dots F_n$	(%)
n	= number of phases	(-)
s	= distance traveled	(mm)

$$(4-4) \quad F_m = \sqrt[3]{|F_1|^3 \cdot \frac{q_{s1}}{100\%} + |F_2|^3 \cdot \frac{q_{s2}}{100\%} + \dots + |F_n|^3 \cdot \frac{q_{sn}}{100\%}}$$



Example of varying individual loads and the corresponding discrete travel steps over a cycle

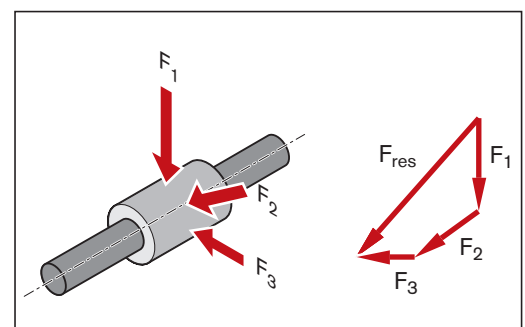
4.1.4.3 Resulting load

Forces from different directions

If the linear bushing is affected by several forces from different load directions at the same time, the resulting load must be determined from all the forces present.

$$(4-5) \quad \vec{F}_{res} = \vec{F}_1 + \vec{F}_2 + \vec{F}_3 + \dots + \vec{F}_k$$

$\vec{F}_1 \dots \vec{F}_k$	= individual loads from different directions	(N)
\vec{F}_{res}	= resultant load	(N)
k	= number of forces from different directions	(-)



Forces from different directions (example)

Then the angle of the resultant load is calculated. With this angle, the factor f_p is determined from the load direction chart. If the load direction cannot be determined, all further calculations must be based on the minimum load capacity (see also section 4.1.3.1).

4 Linear bushings and shafts

4.1 Principles

4.1.4 Calculations

4.1.4.4 Varying bearing loads from varying load directions

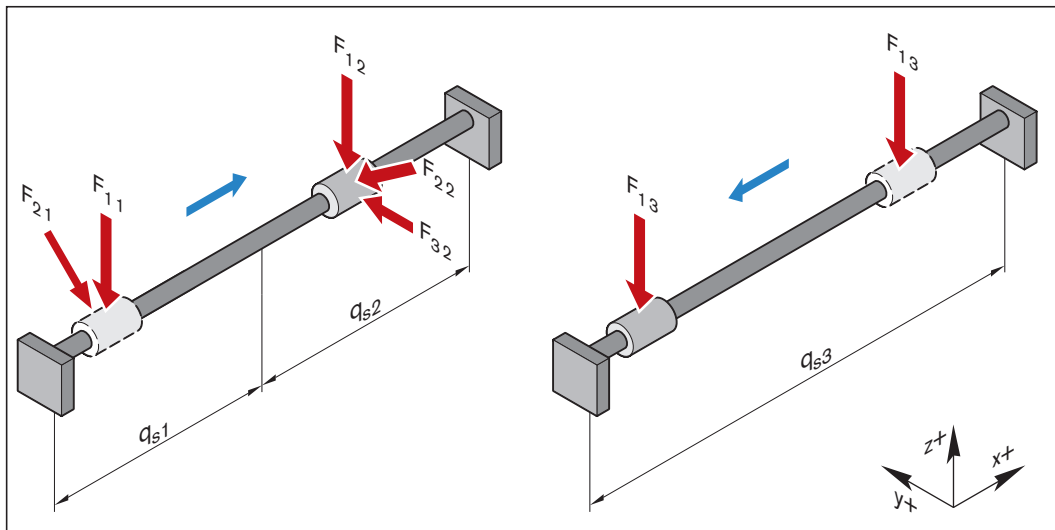
Varying bearing loads and load directions When there are both varying bearing loads and varying load directions, the equivalent dynamic load F_m is calculated as follows:

The resultant load for each phase must be determined from all the forces present (see section 4.1.4.3).

For each individual phase n:

$$(4-6) \quad \vec{F}_{res\ n} = \vec{F}_{1\ n} + \vec{F}_{2\ n} + \vec{F}_{3\ n} + \dots + \vec{F}_{k\ n}$$

- $\vec{F}_{res\ n}$ = resultant load during phase n (N)
- $\vec{F}_{1\ n} \dots \vec{F}_{k\ n}$ = individual loads from different directions in phase 1 ... n (N)
- k = number of different load directions during phase n (-)
- n = number of phases with different bearing loads (-)



Example of varying individual loads from different directions and the corresponding discrete travel steps over a cycle

The resultant loads for all the individual phases are then used to calculate the equivalent dynamic load (see section 4.1.4.2).

$$(4-7) \quad F_m = \sqrt[3]{|F_{res1}|^3 \cdot \frac{q_{s1}}{100\%} + |F_{res2}|^3 \cdot \frac{q_{s2}}{100\%} + \dots + |F_{resn}|^3 \cdot \frac{q_{sn}}{100\%}}$$

The equivalent dynamic load F_m is subsequently used to calculate the life as shown in section 4.1.4.1.

- F_m = equivalent dynamic load (N)
- $F_{res1} \dots F_{resn}$ = resultant load in phases 1 ... n (N)
- $q_{s1} \dots q_{sn}$ = discrete travel steps for $F_{res1} \dots F_{resn}$ (%)

4 Linear bushings and shafts

4.1 Principles

4.1.4 Calculations

4.1.4.5 Torque considerations for torque-resistant linear bushings

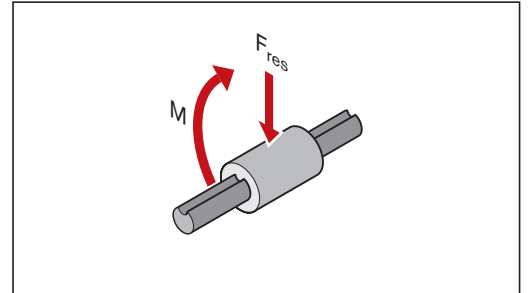
Combined load

If the bushing is simultaneously subjected to a radial load and a torque (moment about the travel axis), the equivalent total load has to be determined.

$$(4-8) \quad F_{\text{comb}} = F_{\text{res}} + \frac{C \cdot M}{M_t}$$

F_{comb}	= equivalent total load	(N)
F_{res}	= resultant radial load	(N)
C	= dynamic load capacity	(N)
M	= torque	(Nm)
M_t	= load moment	(Nm)

For n phases, the equivalent dynamic bearing load F_m is calculated from the equivalent total loads for the individual phases and the corresponding discrete travel steps as shown in the formula (4-7).



Combined radial load and torque

Pure torque load

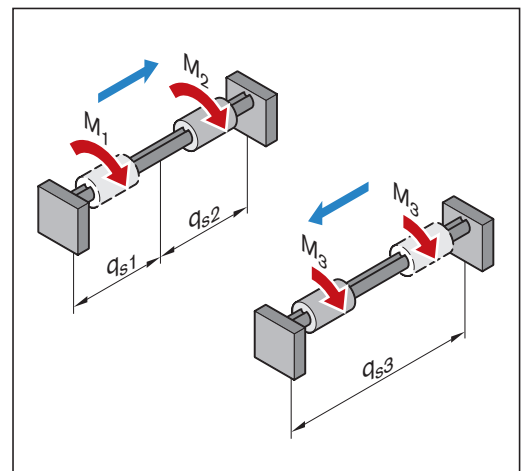
In the case of purely torque loads, the life is calculated using the formula (4-9):

$$(4-9) \quad L = \left(\frac{M_t}{M_m} \right)^3 \cdot 10^5 \text{ m}$$

L	= nominal life	(m)
M_m	= equivalent dynamic torque	(Nm)
M_t	= load moment	(Nm)

The equivalent dynamic torque M_m in the case of varying individual torque loads is calculated using the formula (4-10):

$$(4-10) \quad M_m = \sqrt[3]{|M_1|^3 \cdot \frac{q_{s1}}{100\%} + |M_2|^3 \cdot \frac{q_{s2}}{100\%} + \dots + |M_n|^3 \cdot \frac{q_{sn}}{100\%}}$$



Varying individual torque loads and corresponding discrete travel steps (example)

The hardness factor f_H and the temperature factor f_t are set to 1 when calculating torque-resistant linear bushings because only shafts with a hardness of 60 HRC may be used and the operating temperature may not exceed 100 °C.

M_m	= equivalent dynamic torque	(Nm)
$M_1 \dots M_n$	= individual torque steps	(Nm)
$q_{s1} \dots q_{sn}$	= discrete travel steps for $M_1 \dots M_n$	(%)

4 Linear bushings and shafts

4.1 Principles

4.1.4 Calculations

4.1.4.6 Static load safety factor

The static load safety factor gives the margin of safety against inadmissible permanent deformation on rolling elements and raceways. It is calculated using the formula (4-11):

$$(4-11) \quad S_0 = \frac{C_0}{F_{0max}}$$

S_0 = static load safety factor (-)
 C_0 = static load capacity (N)
 F_{0max} = maximum load (N)

4.1.4.7 Shaft deflection

When steel shafts are used as guideways for linear bushings the shaft deflection must be kept within certain limits to avoid any reduction in their functionality and service life. For Super linear bushings A, H and SH, there are no reductions in load capacity or service life up to a shaft deflection of 0.5°.

To facilitate calculation of the deflection, the product catalog contains tables listing the following details:

- The most commonly occurring bending load cases with the associated deflection equations
- The formulas for calculating the shaft inclination in the linear bushing ($\tan \alpha$) and the permissible shaft inclination values
- The values for $E \cdot I$, dependent on the shaft diameter

Deflection at $x = b$:

$$f = \frac{F \cdot l_W^3}{3 \cdot E \cdot I} \cdot \frac{a^2 \cdot b^2}{l_W^4}$$

Maximum deflection:

$$f_m = f \cdot \frac{l_W + b}{3 \cdot b} \cdot \sqrt{\frac{l_W + b}{3 \cdot a}}$$

Shaft inclination at $x = b$:

$$\tan \alpha = \frac{F \cdot a}{6 \cdot E \cdot I \cdot l_W} \cdot (3b^2 - l_W^2 + a^2)$$

Example of the details in the product catalog for the deflection curve in a typical load case with the corresponding formulas for deflection and shaft inclination

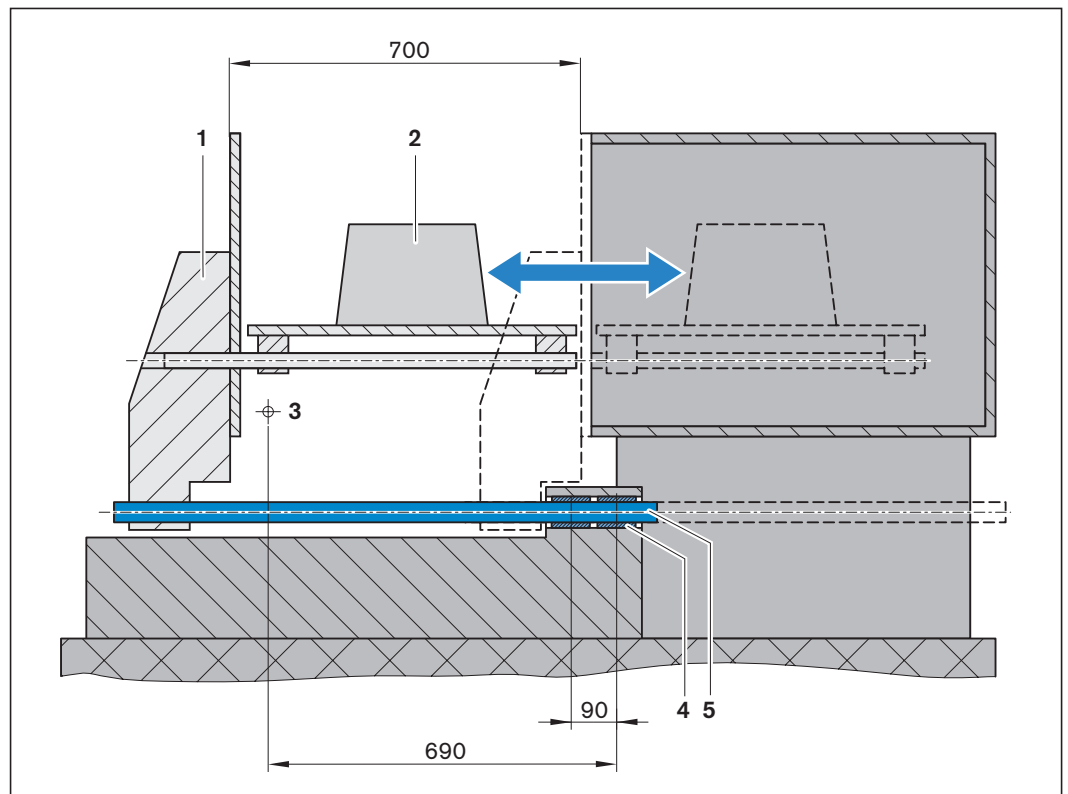
F = individual load (N)
 f = deflection at the load application point (mm)
 f_m = maximum deflection (mm)
 α = shaft inclination at load application point (°)
 l_W = shaft length (mm)
 E = elasticity modulus (N/mm²)
 I = planar moment of inertia (mm⁴)
 a, b = lever arm lengths between load and supports (mm)

4 Linear bushings and shafts

4.1 Principles

4.1.5 Calculation example

Curing oven



Calculation example for a curing oven

- | | |
|---------------------------------------|--------------------------|
| 1 Slide (m = 30 kg) | 4 Linear bushings |
| 2 Load (m = 50 kg) | 5 Precision steel shafts |
| 3 Center of gravity of slide and load | |

Technical data of the application

A slide with a dead weight of 30 kg carries a 50kg load. Two parallel shafts are used to guide the slide into the oven. The shafts are fastened to the slide. The linear bushings are stationary and mounted to the oven.

- Load acting on the two shafts (weight of slide and load): 80 kg
- Two support points per shaft approx. 90 mm apart
- A maximum of 100 °C heat radiation can be expected in the bearing area.
- Manual operation with approx. 6 strokes per hour (0.1/minute)
- Stroke length: 700 mm
- Centre of gravity approx. 690 mm away from the rear bushing centerline
- Use of precision steel shafts with a hardness of at least 60 HRC
- The required service life is at least 10 years under 24 hours a day operation.

Necessary calculations

- Design calculations for the bearings incl. determination of the shaft diameter and selection of the appropriate linear bushing and shaft combination
- Nominal life calculation
- Checking the operating safety using the static load safety factor

4 Linear bushings and shafts

4.1 Principles

4.1.5 Calculation example

Bearing design calculation (stationary application)

Load per shaft:

$$F = 0.5 \cdot m \cdot g = 0.5 \cdot (30 + 50) \text{ kg} \cdot 9.81 \text{ m/s}^2 \\ = 392 \text{ N} \approx 400 \text{ N}$$

$$F_B = F \cdot \frac{l_W}{a} = 400 \text{ N} \cdot \frac{690 \text{ mm}}{90 \text{ mm}} = 3067 \text{ N} \\ \approx 3100 \text{ N}$$

$$F_A = F_B - F = 3100 \text{ N} - 400 \text{ N} = 2700 \text{ N}$$

For calculating the nominal life, the maximum load F_B is taken as the equivalent dynamic total load F_m .

Determination of the linear bushing size

For a shaft hardness of 60 HRC, the reading from the chart for hardness factor f_H in section 4.1.4.1, sub-section "Shaft hardness factor", gives a value of $f_H = 1$.

The life expectancy factor f_L is calculated from the required service life:

$$L_h = 10 \cdot 365 \cdot 24 \text{ h} = 87600 \text{ h}$$

$$L = L_h \cdot 2 \cdot s \cdot n \cdot 60 = 87600 \text{ h} \cdot 2 \cdot 0.7 \text{ m} \cdot 0.1 \text{ min}^{-1} \cdot 60 \text{ min/h}$$

$$L = 7.36 \cdot 10^5 \text{ m}$$

According to the chart in section 4.1.4.1, sub-section "Effect of the required life", the life expectancy factor is $f_L = 0.50$. The maximum load on linear bushing B is $F = 3100 \text{ N}$.

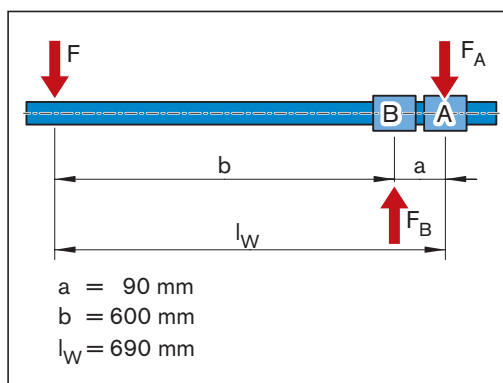
The required load capacity C is calculated using the formula (4-3):

$$C = \frac{F}{f_H \cdot f_t \cdot f_L \cdot f_w} = \frac{3100 \text{ N}}{1 \cdot 1 \cdot 0.50 \cdot 1} = 6200 \text{ N}$$

A Super linear bushing A is selected because of the expected shaft deflection. The product catalog is then consulted to determine which of these linear bushings will meet the following conditions for a maximum load $F_B = 3100 \text{ N}$:

- $F < C_{0 \text{ min}}$
- $F < 0.5 C_{\text{min}}$

This results in the selection of a Super linear bushing A with a shaft diameter of 40_{h7} and $C_{\text{min}} = 8240 \text{ N}$ and $C_{0 \text{ min}} = 4350 \text{ N}$.



Total load acting on one linear bushing

According to the table in section 4.1.4.1, sub-section "Temperature factor", the temperature factor f_t for $100 \text{ }^\circ\text{C}$ is $f_t = 1$.

4 Linear bushings and shafts

4.1 Principles

4.1.5 Calculation example

Shaft deflection

The following formula applies for calculating the shaft deflection:

$$f = \frac{F \cdot b^2 \cdot l_w}{3 \cdot E \cdot I}$$

$$\tan \alpha = \frac{F \cdot a \cdot b}{3 \cdot E \cdot I}$$

According to the product catalog, the result for a shaft diameter of 40_{h7} is the value $E \cdot I = 2.64 \cdot 10^{10} \text{ Nmm}^2$.

Calculated shaft inclination without clearance:

$$f = \frac{400 \text{ N} \cdot (600 \text{ mm})^2 \cdot 690 \text{ mm}}{3 \cdot 2.64 \cdot 10^{10} \text{ Nmm}^2}$$

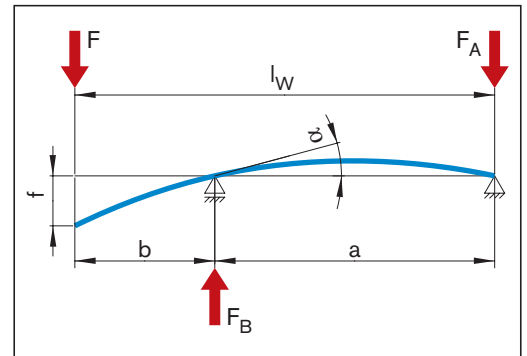
$$f = 1.25 \text{ mm}$$

Shaft inclination in the linear bushing:

$$\tan \alpha = \frac{400 \text{ N} \cdot 600 \text{ mm} \cdot 90 \text{ mm}}{3 \cdot 2.64 \cdot 10^{10} \text{ Nmm}^2} = 2.73 \cdot 10^{-4}$$

$$\alpha = 0.016^\circ (\approx 1')$$

The permissible inclination of the Super linear bushing version A without reduction of the load capacity is 0.5° (resp. $30'$).



Load case

Service life

Nominal life in meters according to formula (4-1):

$$L = \left(\frac{C}{F} \cdot f_H \cdot f_t \cdot f_w \right)^3 \cdot 10^5 \text{ m}$$

$$L = \left(\frac{8240 \text{ N}}{3100 \text{ N}} \cdot 1 \cdot 1 \cdot 1 \right)^3 \cdot 10^5 \text{ m}$$

$$L = 18.78 \cdot 10^5 \text{ m}$$

Nominal life in hours according to formula (4-2):

$$L_h = \frac{L}{2 \cdot s \cdot n \cdot 60} = \frac{18.78 \cdot 10^5 \text{ m}}{2 \cdot 0.70 \text{ m} \cdot 0.1 \text{ min}^{-1} \cdot 60}$$

$$L_h = 223571 \text{ h}$$

Static load safety factor

Static load safety factor according to formula (4-11):

$$S_0 = \frac{C_0}{F_{0\max}} = \frac{C_0}{F_B} = \frac{4350 \text{ N}}{3100 \text{ N}} = 1.40$$

Note: The nominal life calculation is performed for the above-mentioned maximum load $F = 3100 \text{ N}$. This simplification provides an additional safety margin for the service life. To calculate the nominal life precisely, a dynamic cycle with the corresponding loads in the individual phases must be determined. The procedure for determining the operating conditions is described in detail in Chapter 3, section 3.1.5.2.

Taking into account all the stated parameters and assuming 24-hour operation throughout the year, in an ideal case, the guideway's nominal life is 26 years.

In this case, under normal conditions of use, a static load safety factor of $S_0 = 1.4$ is sufficient.

4 Linear bushings and shafts

4.2 Linear bushings

4.2.1 Compact and eLINE linear bushings

Compact design

Compact and eLINE linear bushings are characterized by their small dimensions, comparable to those of sliding bearings. They consist of a plastic ball retainer with 5 or 6 closed ball circuits. The hardened segmental steel plates with ball tracks for the load-bearing zones are integrated into the ball retainer and transmit the applied forces. The individual components are fixed by two metal holding rings that accommodate the end wiper seals.

Structural design**Axial retention**

In the linear bushings for shaft diameters 12 mm to 50 mm, the metal holding rings (1) are oversized. With these types of linear bushings, this avoids the otherwise usual requirement for additional axial retention in the receiving bore. In the linear bushings for shaft diameters 8 mm and 10 mm, the diameter of the plastic outer sleeve (2) is oversized to ensure axial retention. For applications with vibrations and/or higher accelerations, additional retention is required. Compact linear bushings are also available in corrosion-resistant versions or as eLINE bushings with reduced radial clearance.

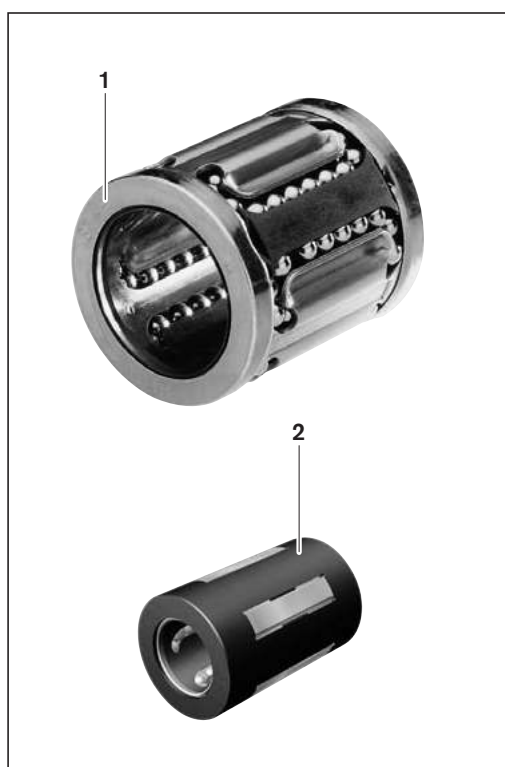
Advantages

The use of Compact and eLINE linear bushings offers the following advantages:

- A very economical linear bushing for general requirements
- Small overall dimensions for particularly compact assemblies
- High load capacities and long service life due to hardened segmental steel plates with ball conformity in the running track
- High travel speed (5 m/s)
- With integral wiper seals, separate end seals, or without seals
- Easy installation: Simply press bushing in; no additional retention required
- Many pockets acting as lubricant reservoirs for extended lubrication intervals or lubrication for life
- Pre-lubricated for life at the factory (eLINE linear bushings)
- Also available with reduced radial clearance guides (eLINE linear bushings)
- Also available in corrosion-resistant versions for applications in the medical, chemical and food industries
- Linear sets available with aluminum or corrosion-resistant steel housing



Compact linear bushing and shaft



Axial retention for different sizes of Compact linear bushings

- 1 Metal holding ring (Compact linear bushing for shaft diameters 12 mm to 50 mm)
- 2 Plastic outer sleeve (Compact linear bushings for shaft diameters 8 mm and 10 mm)

4 Linear bushings and shafts

4.2 Linear bushings

4.2.2 Super linear bushings A and B

**Smooth running
Long service life**

Super linear bushings A and B have steel inserts with ground ball tracks and optimized ball entry zone geometry for especially smooth running and long life.

Structural design

They consist of a plastic ball retainer with 5 or 6 closed ball circuits, depending on the diameter. The hardened segmental steel plates with ground ball tracks are integrated into the plastic outer sleeve. These segments are the load-bearing elements. The plastic outer sleeve forms a closed shell that covers the balls in the return tracks.

Sealing

Super linear bushings are sealed either with integral wiper seals or separate end seals. If separate end seals are used, these are held in place by a metal case. The case is oversized and can also be used for axial retention of the linear bushing. The integral wiper seals are floating seals (1), which provide very good sealing even when the linear bushing rocks. This significantly prolongs the bushing's life.

**Super linear
bushing A
Self-alignment**

Because of the slightly crowned outer contours of the segmental steel plates (2), the Super linear bushing A can compensate for alignment errors of up to 0.5° between the shaft and the receiving bore. Alignment errors are caused by:

- Manufacturing inaccuracies
- Mounting errors
- Shaft deflection

**Angular
self-adjustment**

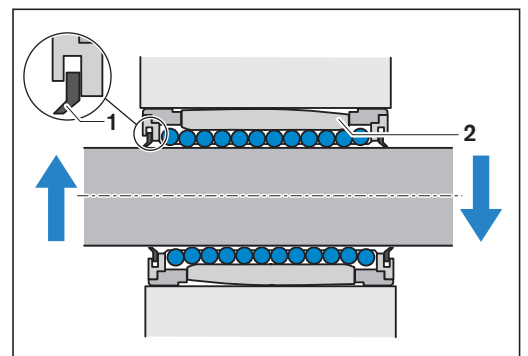
Angular self-adjustment ensures that the balls enter the load-bearing zone smoothly and that the load is distributed evenly across the whole row of balls and over the full travel stroke. This also prevents any critical pressure between the bushing edge and the shaft due to distortive stresses. There is no reduction in load capacity or service life and the bushing runs exceptionally smoothly. However, the self-aligning feature cannot compensate for parallelism offsets between the shafts in a table guide.

Rocking effect

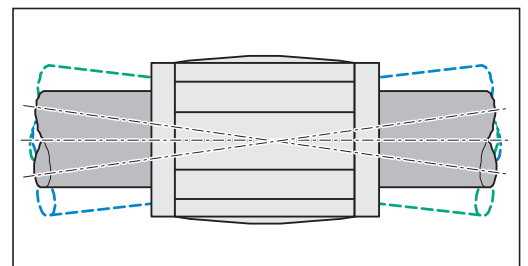
The travel profile opposite shows a comparison with a conventional linear bushing. The example is based on a load of 800 N and an alignment error of approx. $8'$ (0.13°), due to shaft deflection. Self-alignment may cause a slight rocking effect. Two Super linear bushings must therefore be used on at least one of the guideway's shafts.



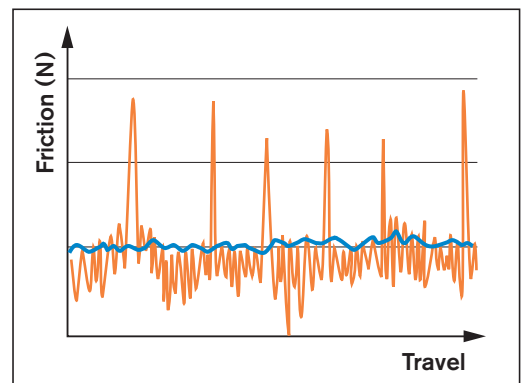
Super linear bushing A



Floating seals and self-alignment



Alignment error and rocking effect



Travel profiles recorded at the same speed of travel

- Conventional linear bushing, size 20
- Super linear bushing A, size 20

4 Linear bushings and shafts

4.2 Linear bushings

4.2.2 Super linear bushings A and B

Super linear bushing B

Super linear bushings B have no self-alignment function. They are the solution for applications in which only one linear bushing is used per shaft and the linear bushing may not rock on the shaft.

Super linear bushings A and B

Since Super linear bushings A and B provide exceptional conformity between the balls and the ground ball tracks, higher load capacities are possible compared to Standard linear bushings of the same dimensions.

Advantages

The use of Super linear bushings A or B offers the following advantages:

- An economical linear bushing for demanding applications
- Very good, low-noise running performance
- High dynamic load capacity
- High acceleration and high travel speed thanks to good ball guidance and wear-resistant ball retainer
- High rigidity
- With integral wiper seals, separate end seals, or without seals
- Identical installation dimensions ensure interchangeability with Super linear bushings H and SH and also with Standard linear bushings
- Linear sets available with aluminum or corrosion-resistant steel housing



Super linear bushing B without self-alignment

4 Linear bushings and shafts

4.2 Linear bushings

4.2.3 Standard linear bushings

Robust linear bushings for use under harsh conditions

Standard linear bushings are so called because they were the first to be developed. Their precision and therefore their load-bearing capacity and travel performance have been constantly improved through continuous further development. In the meantime, since other types of linear bushings with sometimes significantly higher performances have been developed, Standard linear bushings are now primarily used for applications in harsh environments. Typical applications are found in the woodworking industry, foundries and cement factories.

Structural design

The outer sleeve of Standard linear bushings is hardened and ground. The shaft diameter determines the number of rows of balls. Standard linear bushings can have from 4 to 6 rows. The balls run in closed circuits inside the sheet steel ball retainer. The ball retainer is fixed by holding or sealing rings in grooves in the outer sleeve. The steel ball retainer makes Standard linear bushings especially robust.

Construction forms

Standard linear bushings are available in three forms: closed-type, adjustable and open-type. For use at very high temperatures, there is also an unsealed version.

Closed-type Standard linear bushings are also available in versions made entirely from corrosion-resistant steel and are used, among other areas, in the medical, chemical and food industries. When long stroke lengths are required, open-type Standard linear bushings are used with shaft support rails. The shaft support rail prevents the shaft from deflecting.

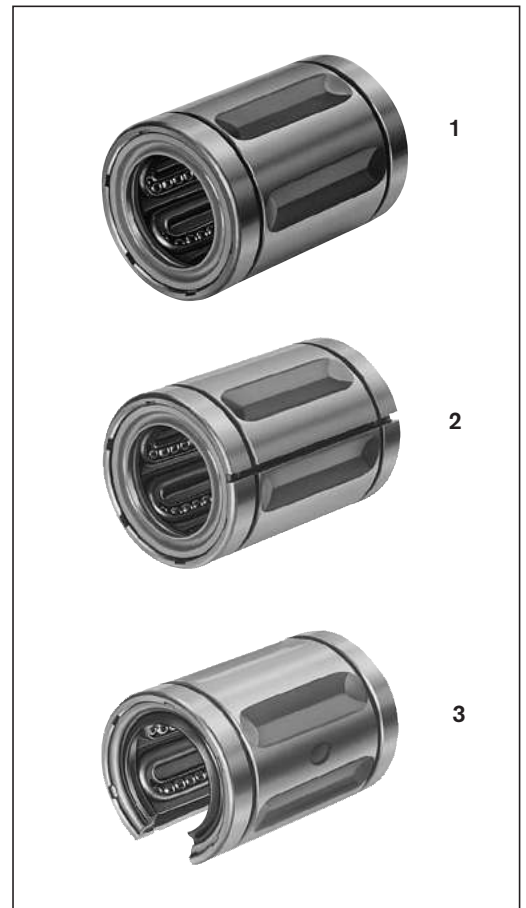
Advantages

The use of Standard linear bushings offers the following advantages:

- Long service life
- High precision
- Low friction
- Robust all-metal design
- Suitable for temperature ranges above 100 °C or vacuum applications
- Many pockets acting as lubricant reservoirs for extended lubrication intervals or lubrication for life
- With integral wiper seals or without seals
- Various flanged versions available
- Linear sets available with cast iron housing



Adjustable Standard linear bushing



Forms of Standard linear bushings

- 1 Closed-type Standard linear bushing
- 2 Adjustable Standard linear bushing
- 3 Open-type Standard linear bushing

4 Linear bushings and shafts

4.2 Linear bushings

4.2.4 Segmental linear bushings

Shortest linear bushing type

Segmental linear bushings are the shortest types of linear bushing. They consist of a plastic ball retainer with hardened segmental steel plates that are fixed in the ball retainer by two plastic rings. Either the wiper sealing ring or a metal case with integral sealing ring are used for axial retention.

Structural design**Corrosion-resistant version**

For applications in corrosive conditions or where there are heavy requirements on cleanliness, such as, for instance, in the food industry, semiconductor manufacture or the medical equipment industry, these linear bushings are also available in corrosion-resistant versions.

Advantages

The use of Segmental linear bushings offers the following advantages:

- Low-noise operation
- Low weight
- As linear sets with reinforced plastic housings, they represent a low-cost solution for general requirements.
- The shortest type of linear bushing due to its design, it provides the longest stroke for a given shaft length.
- With separate seals or without seals



Segmental linear bushing

4 Linear bushings and shafts

4.2 Linear bushings

4.2.5 Super linear bushings H and SH

Higher number of load-bearing ball rows

Super linear bushings H and SH differ from types A and B in their higher number of load-bearing rows of balls. Depending on the particular version, they can have up to 12 rows. The bushings are sealed with integral double acting wiper seals. In the open-type versions, as with the other types of linear bushings, additional longitudinal seals are installed along the sides of the opening. These linear bushings are retained by means of a locating pin or a screw through a lateral hole. In Super linear bushings type H, the hole can also be used for lubrication.

Structural design**Dynamic load capacity**

Because of the high number of ball circuits, the dynamic load capacities are almost double those of Super linear bushings A and B.

Self-alignment

The higher load capacities allow particularly heavy weights to be moved with full self-alignment.

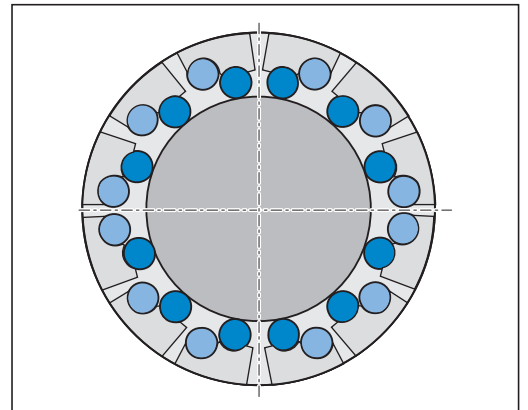
Advantages

The use of Super linear bushings H or SH offers the following advantages:

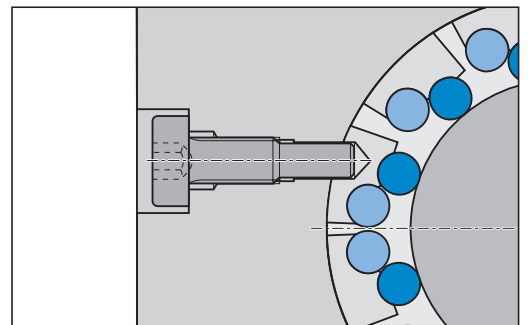
- Highly accurate linear bushings for moving heavy weights
- High load capacities and long life
- High travel speed (5 m/s) and acceleration due to the wear-resistant ball retainer
- Automatic compensation for alignment errors or shaft deflection up to 0.5°
- Smooth ball running characteristics
- High rigidity
- With integral wiper seals or without seals
- Optional side seals for open-type linear bushings
- Steel load-bearing plates with ground ball tracks and backs for the highest precision
- In-service lubrication possible via lube hole or pockets
- Linear sets available with aluminum housing



Super linear bushing H



Super linear bushing SH with 12 rows of balls



Retention via lateral locating screw

4 Linear bushings and shafts

4.2 Linear bushings

4.2.6 Radial linear bushings

Radial ball recirculation

In Radial linear bushings, the balls are recirculated outwards through radially arranged return raceways. This more than doubles the number of load-bearing rows of balls compared to conventional linear bushings and extends the load-bearing zones, which results in very high load capacities and high rigidity.

High load capacities and rigidity**Structural design**

Radial linear bushings are available only as open-type versions. They consist of a hardened and ground steel sleeve with a plastic ball retainer in which 12 closed ball circuits and two retaining rings are integrated. The load-bearing balls are guided in several tracks in the plastic ball retainer, directed radially outwards and then conducted back to the load-bearing zone via ball return bores. Because of their characteristics, these linear bushings are suitable for use in machine tools, numerous special machines, and transfer and automation systems.

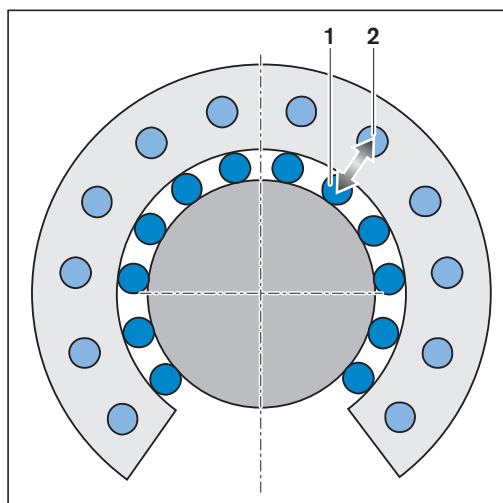
Advantages

The use of radial linear bushings offers the following advantages:

- Highly precise linear bushings
- Suitable for moving very heavy weights
- Very high load capacities
- Very high rigidity
- Very smooth operation
- With integral wiper seals and side seals (fully sealed), with separate end seals, or without seals
- Heavy-duty version with a degree of freedom in the circumferential direction
- For applications where other linear guides might be susceptible to distortive stresses because of inaccuracies in supporting structures
- Linear sets available with steel housing
- Radial Compact sets available for highly compact designs



Radial linear bushing



Principle of radial ball circulation

- 1 Load-bearing balls
- 2 Non-loaded balls in the return zone

4 Linear bushings and shafts

4.2 Linear bushings

4.2.7 Torque-resistant linear bushings

Torque-resistant linear guides

A torque-resistant linear bushing can absorb a torque around the travel axis. Torque-resistant linear bushings were developed as machine elements enabling true, i.e. torsionally stiff, linear motion with only one shaft. This compact design meets the requirements of many equipment and special machinery construction sectors. The basic structure corresponds to that of Super linear bushings B, but with one or two rows of balls at a lower level (2). The shaft (1) has one or two ball guide grooves in which the lower-level ball rows are guided and can transmit torque forces. The guide grooves in the shaft and the relevant steel inserts (3) in the linear bushing have a Gothic profile. This profile forms a 4-point contact with the balls and enables the transmission of moment loads in both directions around the travel axis.

Structural design**Gothic profile****Adjusting for zero clearance**

An adjustment screw (4) is inserted into the receiving housing and engages with a counter-sunk area on the steel insert (3) belonging to the lower-level row of balls. The adjustment screw allows the torque-resistant linear bushing to be adjusted to zero clearance. A locking nut (5) secures the adjustment screw.

Versions

For special applications, torque-resistant linear bushings can be provided with up to four ground ball guide grooves. These are capable of transferring higher torques. Another version is the Torque-resistant Compact linear bushing. The compact linear bushing version has a smaller outside diameter and is fitted with a more compact adjusting and locking screw. The compact design makes insertion into sleeves particularly easy.

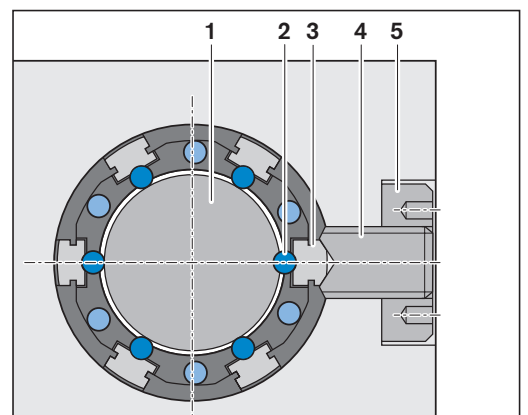
Advantages

The use of Torque-resistant linear bushings offers the following advantages:

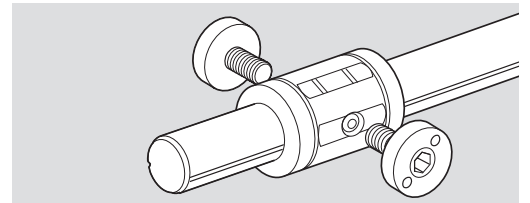
- Axially and radially true linear motion with only one shaft
- Large choice of formats
- With separate seals or without seals
- Various flanged versions available
- With steel or aluminum housings in various formats



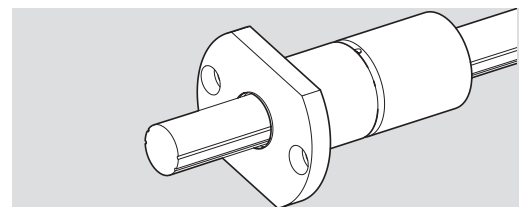
Torque-resistant linear bushing



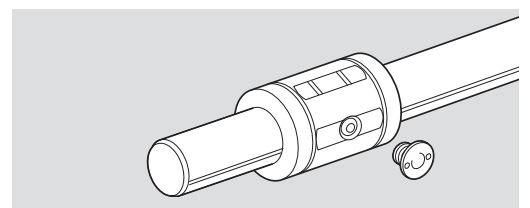
Structure of a Torque-resistant linear bushing and shaft with one ball guide groove (represented as a linear set)



Torque-resistant linear bushing and shaft with two ball guide grooves



Version with four ball guide grooves and flange



Torque-resistant Compact linear bushing

4 Linear bushings and shafts

4.2 Linear bushings

4.2.8 Linear bushings for combined linear and rotary motion

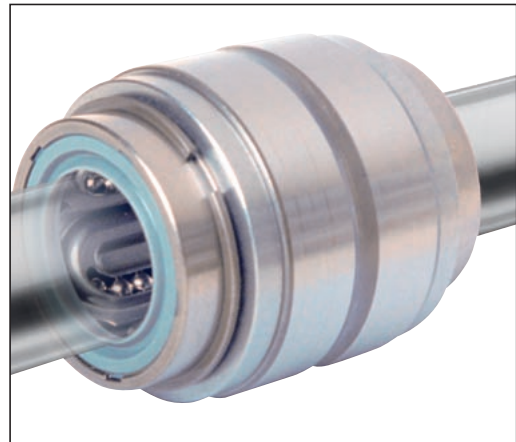
Deep-groove ball bearings**Needle bearings**

Linear bushings for combined linear and rotary motion are supplied with deep-groove ball bearings or needle bearings. They are suitable for applications involving both linear and rotary motion.

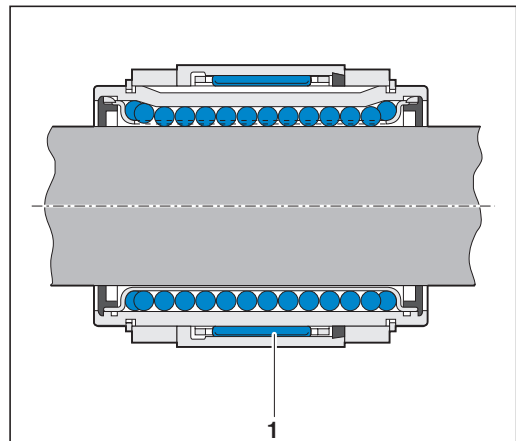
Advantages

The use of this type of linear bushings offers the following advantages:

- Precise guidance with high load capacities
- High rotational speed and low friction
- Suitable for linear applications with additional rotary motion
- Suitable for gripping and swiveling functions
- Suitable for winding applications



Linear bushing for combined linear and rotary motion



Structural design of a linear bushing for combined linear and rotary motion with needle bearings (1)

4 Linear bushings and shafts

4.3 Linear sets

Complete bearing units

Linear sets are complete bearing units consisting of a housing with one or two linear bushings. They are available in many different configurations. Because of their rationalized construction and fabrication, linear sets offer users significant cost advantages over customer-built designs. The housings can be easily aligned during mounting, thereby avoiding distortive stresses on the linear bushings.

Cost advantages

The high precision ensures the linear bushings' operational reliability and makes the units fully interchangeable.

Interchangeability



Linear bushing guide with two linear sets

Type of linear bushing	Designs	Versions
Compact linear bushing eLINE linear bushing		<ul style="list-style-type: none"> ■ Adjustable/non-adjustable ■ Normal/corrosion-resistant ■ Single/tandem
Super linear bushing A Super linear bushing B	 	<ul style="list-style-type: none"> ■ Cast iron/aluminum housing ■ Adjustable/non-adjustable ■ With side opening ■ With flange ■ Single/tandem (aluminum only)
Standard linear bushing		<ul style="list-style-type: none"> ■ Closed/open type ■ Adjustable/non-adjustable ■ With side opening ■ With flange
Segmental linear bushing		<ul style="list-style-type: none"> ■ Normal/corrosion-resistant
Super linear bushing H Super linear bushing SH		<ul style="list-style-type: none"> ■ Closed/open type ■ Adjustable/non-adjustable ■ With side opening
Radial linear bushing		<ul style="list-style-type: none"> ■ Adjustable/non-adjustable ■ With side opening ■ Radial Compact sets
Torque-resistant linear bushing	 	<ul style="list-style-type: none"> ■ With 1, 2 or 4 ball guide grooves ■ Single/tandem ■ Steel/aluminum ■ Housing/sleeve ■ Sleeve with flange

4 Linear bushings and shafts

4.3 Linear sets

Version	Applications
Closed type	For high-precision guidance with extreme ease of mounting. Version with fixed working bore diameter.
Adjustable	For use when zero clearance or preload is required. The desired radial clearance is established by means of an adjusting screw. These Linear Sets are adjusted to zero clearance before delivery.
Open type	For long guideways when the shafts must be supported and high rigidity is required.
Open type, adjustable	For use when zero clearance or preload is required. The desired radial clearance is established by means of an adjusting screw. These Linear Sets are adjusted to zero clearance before delivery.
With side opening	Handles forces from all directions without reduction of load capacity.
With side opening, adjustable	For use when zero clearance or preload is required. The desired radial clearance is established by means of an adjusting screw. These Linear Sets are adjusted to zero clearance before delivery.
Corrosion-resistant	Housing in corrosion and acid-resistant chrome-nickel steel for use in the food, semi-conductor, medical, pharmaceutical and chemical industries.
Flanged type	This element was developed as a complement to the linear set series for use in applications requiring the shaft to be arranged at right angles to the mounting base.
Tandem type	Linear set with two linear bushings for heavy loads.

High rigidity

Linear sets provide high rigidity regardless of the load direction. Just as with linear bushings, for linear sets also the effect of the load direction on load capacity must be taken into account. The high precision ensures the linear bushings' operational reliability and makes the units fully interchangeable.

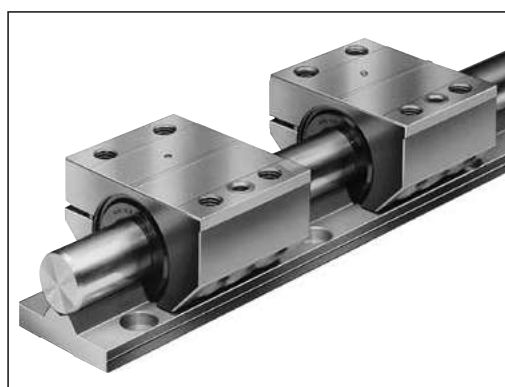
Radial Compact set

Unlike linear sets, in radial compact sets the housing and the linear bushing are integrated into one element. The ball recirculation principle is the same as that of radial linear bushings.

Advantages

Radial Compact sets offer decisive advantages over linear sets with radial linear bushings:

- Lower height and narrower width due to compact design
- Lower weight
- Increased dimensional accuracy and closer tolerances: The radial clearance is 50% smaller and the height tolerance is up to 25% less.
- Fully sealed with a sealing ring and a side seal along the opening
- Reference edge for easier mounting
- Predrilled holes for locating pins

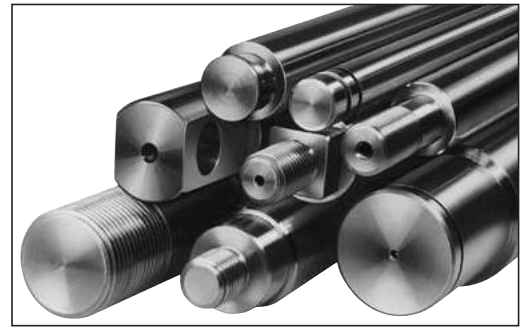


Linear bushing guide with two Radial Compact sets



4 Linear bushings and shafts

4.4 Precision steel shafts

Precision steel shafts are available in metric diameters with various tolerances, as solid shafts and tubular shafts, made from heat-treated steel, corrosion-resistant steel or hard chrome plated steel. Rexroth supplies precision steel shafts cut to customer-specified lengths with chamfering at both ends, or machined to customers' drawings or specifications.



Precision steel shafts

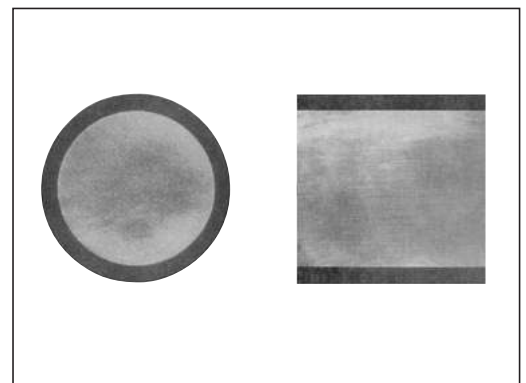
Types	Versions
Solid shaft 	<ul style="list-style-type: none"> ■ Heat-treated steel ■ Corrosion-resistant steel X46Cr13 ■ Corrosion-resistant steel X90CrMoV18 ■ Hard chrome plated
Tubular shaft 	<ul style="list-style-type: none"> ■ Heat-treated steel ■ Hard chrome plated

Tolerance classes

The diameters of the precision steel shafts are machined to tolerance classes h6 and h7.

Hardness

The shafts are induction hardened and smoothed by centerless grinding. The depth of hardening is 0.4 to 3.2 mm depending on the shaft diameter. The surface hardness and depth of hardness are extremely uniform, both in the axial and in the circumferential direction. This is the reason for the excellent dimensional consistency and the long service life of the precision steel shafts. The photographs opposite show a cross-section and a longitudinal section through a hardened and ground precision steel shaft. The hardened surface zone has been made visible by polishing and caustic etching.



Cross-section and longitudinal section through a hardened steel shaft

Deflection

When steel shafts are used as round guideways for linear bushings, shaft deflection must be taken into account (see section 4.1.4.7). This occurs because of the operating loads. Deflection must be kept within certain limits in order to avoid any reduction in the functionality and service life of the assemblies.

Chamfering

Steel shafts intended for use as round guideways for linear bushings must be chamfered at the ends to prevent damage to the ball retainers or wiper seals when the linear bushing is being pushed onto the shaft.

Applications

In addition to their use as guide shafts for linear bushings, precision steel shafts are well-proven in many other applications, for instance as rollers, pistons or axles.

4 Linear bushings and shafts

4.5 Shaft support rails

Ready-to-install elements

Shaft support rails are ready-to-install elements with high dimensional accuracy.

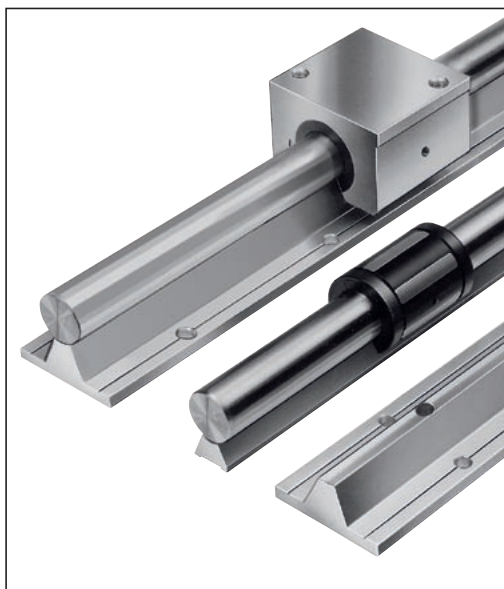
Rigidity

Shaft support rails are designed for use with open-type linear bushings. They prevent the shaft from bending and increase the rigidity of the overall system.

Advantages

The use of shaft support rails offers the following advantages:

- Prevention of shaft deflection
- Improving the performance of linear motion guideways
- An additional degree of freedom in the circumferential direction compared to profiled rail systems
- Saving on complex and costly customer-built designs
- For applications where other linear guides might be susceptible to distortive stresses because of inaccuracies in supporting structures
- All shaft support rails are also suitable for use with corrosion-resistant shafts.



Shaft support rails

Suitability	Designs	Characteristics	Designs	Characteristics
For Super and Standard linear bushings		<ul style="list-style-type: none"> ■ Low profile ■ Especially economical ■ With flange ■ Material: aluminum 		<ul style="list-style-type: none"> ■ Especially economical ■ For high loads ■ Flangeless ■ Material: aluminum
		<ul style="list-style-type: none"> ■ Suitable for aluminum profile systems ■ Low profile ■ Especially economical ■ With flange ■ Material: aluminum 		<ul style="list-style-type: none"> ■ With reference edge ■ High precision ■ For high loads ■ Flangeless ■ Material: steel
		<ul style="list-style-type: none"> ■ Low profile ■ High precision ■ For high loads ■ With flange ■ Material: aluminum 		<ul style="list-style-type: none"> ■ For side fitting ■ High precision ■ For high loads ■ Material: aluminum
		<ul style="list-style-type: none"> ■ High profile ■ High precision ■ For high loads ■ With flange ■ Material: aluminum 		
For Radial linear bushings		<ul style="list-style-type: none"> ■ Low profile ■ High precision ■ For high loads ■ With flange ■ Material: steel 		<ul style="list-style-type: none"> ■ For side fitting ■ High precision ■ For high loads ■ Material: steel
For Radial Compact sets		<ul style="list-style-type: none"> ■ With reference edge ■ Low profile ■ High precision ■ For high loads ■ With flange ■ Material: steel 		

4 Linear bushings and shafts

4.6 Shaft support blocks

For closed-type linear sets






In guides with closed-type linear sets the guide shafts are fixed at the ends. Precision shaft support blocks have been specially developed for this purpose.

Rigidity

Shaft support blocks from Rexroth provide significant cost advantages over customer-built designs. They are also highly rigid. The individual elements have a high degree of dimensional accuracy and are therefore interchangeable. The shaft support blocks are designed for easy mounting and fast alignment. Where especially high precision is required, shaft support blocks with reference edges are the best choice.

Interchangeability

Shaft support block

Designs	Versions/special features
	<ul style="list-style-type: none"> ■ Spheroidal graphite cast iron
	<ul style="list-style-type: none"> ■ Aluminum ■ Rigid shaft mounting due to extra-wide design ■ With reference edge
	<ul style="list-style-type: none"> ■ Flanged ■ Gray cast iron
	<ul style="list-style-type: none"> ■ Aluminum ■ For linear sets with Compact linear bushings ■ For particularly space-saving constructions
	<ul style="list-style-type: none"> ■ Corrosion-resistant chrome-nickel steel ■ For use in the food, semiconductor, pharmaceutical and chemical industries ■ For particularly space-saving constructions

5 Ball screw drives

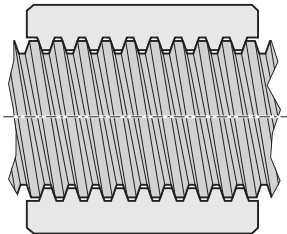
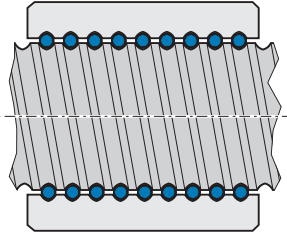
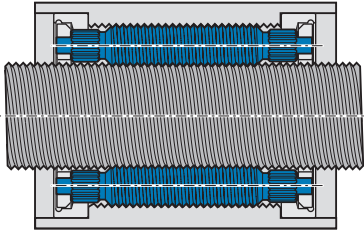
5.1 Principles

5.1.1 System technology

Screw drive overview

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

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

Screw drive type	Description
Acme screw drive 	<ul style="list-style-type: none"> ■ Screw drive with sliding contact between the screw and the nut
Ball screw drive 	<ul style="list-style-type: none"> ■ Screw drive with rolling contact between the screw, rolling elements and nut ■ Rolling elements: balls
Planetary roller screw drive 	<ul style="list-style-type: none"> ■ Screw drive with integral planetary gear ■ Screw drive with rolling contact between the screw and the rolling elements and between the rolling elements and the nut ■ Rolling elements: planetary rollers

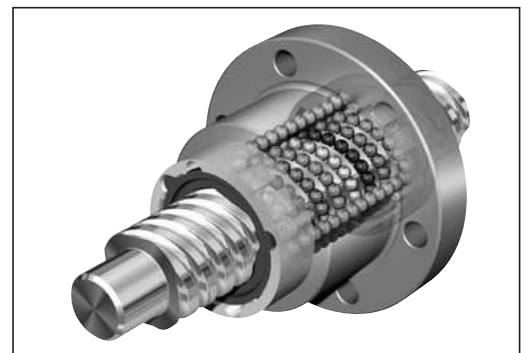
In linear motion technology, ball screw drives are the most commonly used option. In the following sections, ball screw drives are dealt with in more detail.

DIN standard

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

DIN 69051 Part 1 defines ball screw drives as follows:

An assembly comprising a ball screw shaft and a ball nut and which is capable of converting rotary motion into linear motion and vice versa. The rolling elements of the assembly are balls.



Screw and nut of a ball screw assembly

5 Ball screw drives

5.1 Principles

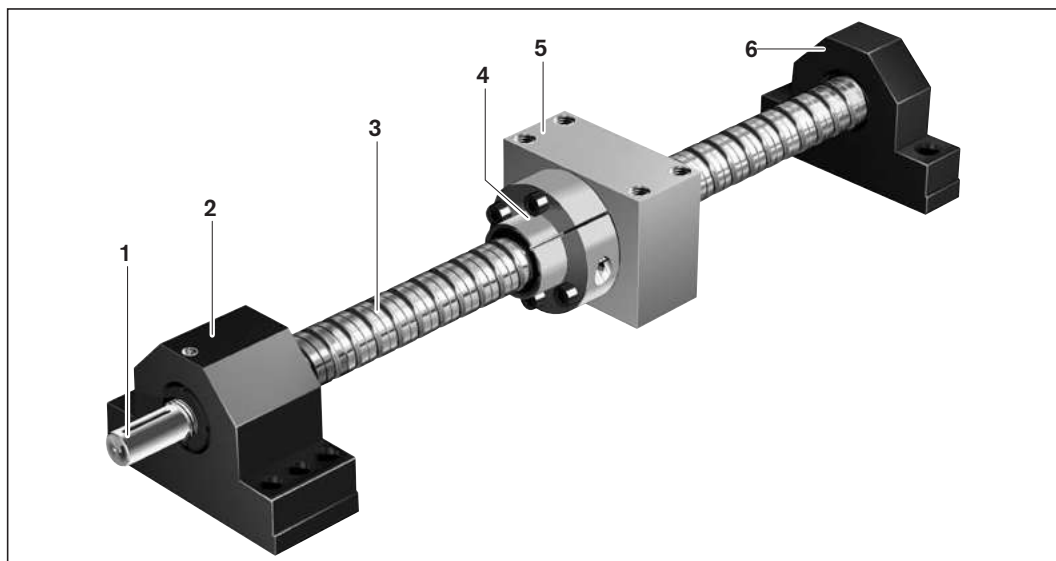
5.1.1 System technology

5.1.1.1 Structural design of a ball screw assembly

Ball screw assemblies generally consist of the following components:

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

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



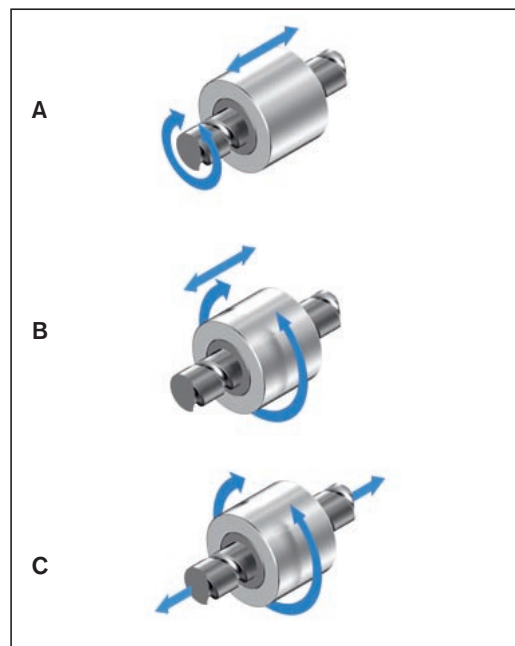
Structural design of a ball screw assembly

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

Operating principle

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

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



Operating principles of ball screw assemblies

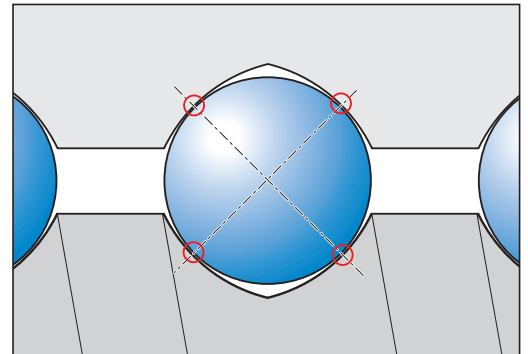
5 Ball screw drives

5.1 Principles

5.1.1 System technology

Screw

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

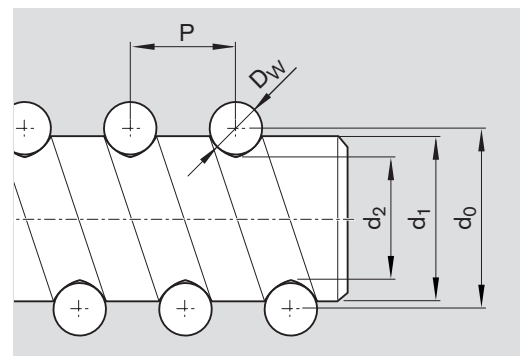


Gothic profile of the ball tracks in the screw and nut and contact points on the rolling elements

Screw dimensions

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

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



Screw geometry

Screw sizes

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

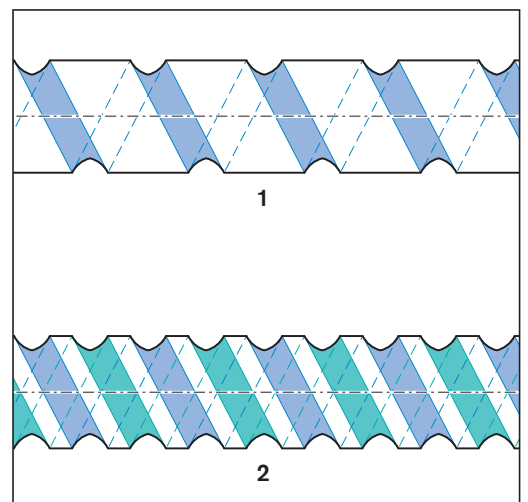
Example: 32 x 5R x 3.5 for a screw with a nominal diameter of 32 mm, lead of 5 mm, right-hand thread, and a ball diameter of 3.5 mm.

Multi-start screws

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

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

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



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

5 Ball screw drives

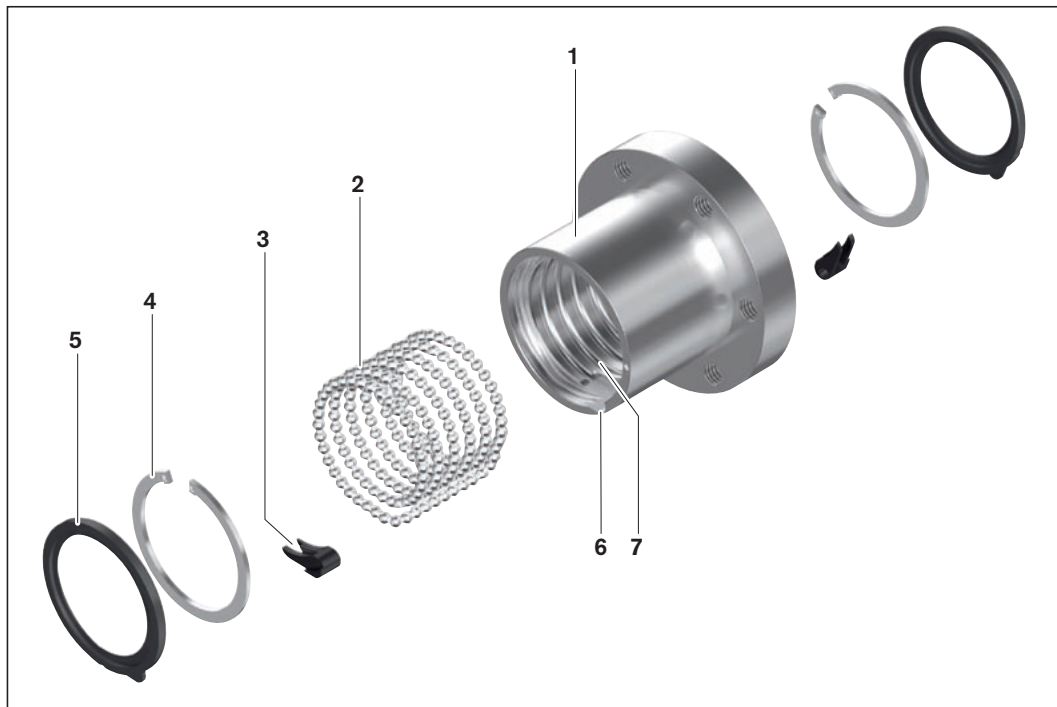
5.1 Principles

5.1.1 System technology

Ball nut

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

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

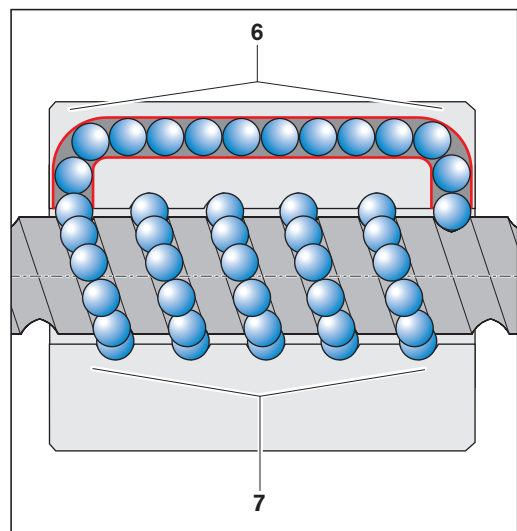


Structural design of a ball nut

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

Rolling element circuit

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



Principle of rolling element recirculation

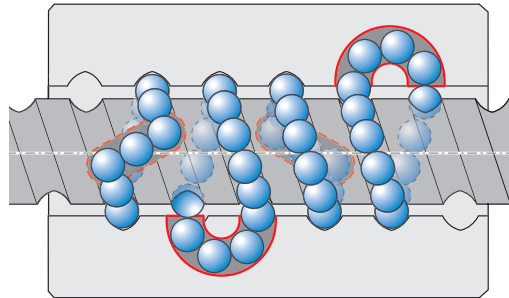
5 Ball screw drives

5.1 Principles

5.1.1 System technology

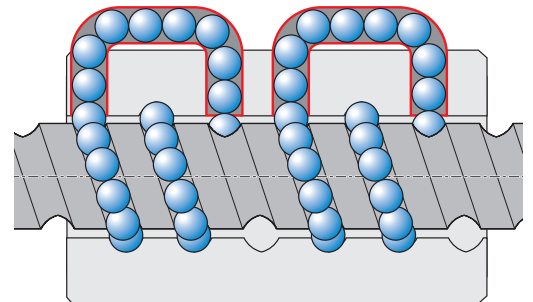
Recirculation systems

Single-turn recirculation



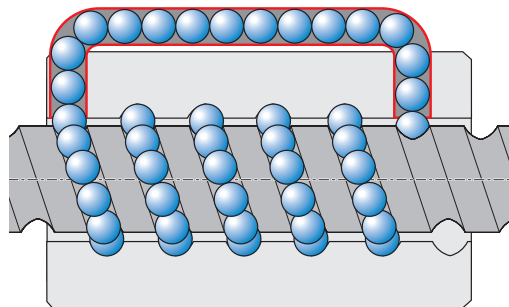
The single-turn recirculation system recirculates the balls from just one ball track turn. A recirculating piece inserted into the nut body guides the balls over the screw shoulder and into the neighboring thread turn.

Multiple-tube recirculation



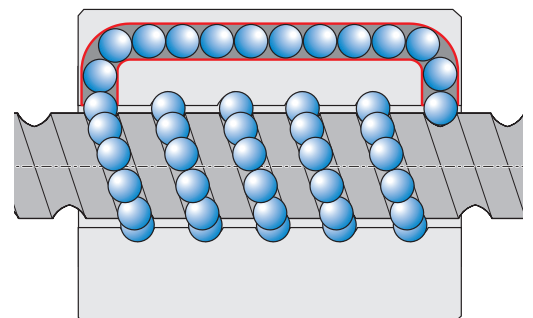
In this particular recirculation system, the balls are brought back to the threading by tubes spanning two or three ball track turns. A nut with several ball circuits will have several recirculation tubes.

Single-tube full recirculation



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

Integrated single-bore full recirculation



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

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

The advantages of this system are:

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




5 Ball screw drives

5.1 Principles

5.1.1 System technology

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




Nut forms

Nut form	Fastening to the adjacent structure and further system characteristics
<p>Screw-in nut</p> 	<ul style="list-style-type: none"> ■ Screw-in nuts are inserted directly into a mating thread in the adjacent structure.
<p>Cylindrical nut</p> 	<ul style="list-style-type: none"> ■ Cylindrical nuts are used for compact constructions. ■ The torque is transferred by a key in the nut body. ■ The cylindrical nut is axially secured by a ring nut.
<p>Flanged nut</p> 	<ul style="list-style-type: none"> ■ The flange is used to bolt the nut to the adjacent structure. ■ The position of the flange (at the end or in the center) depends on the nut series.

Flanged nuts

The different flange shapes on the flanged nuts enable fixing even in restricted spaces. The number of flange holes is adapted to the load

rating and/or the potential load on the respective ball nut.

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

5 Ball screw drives

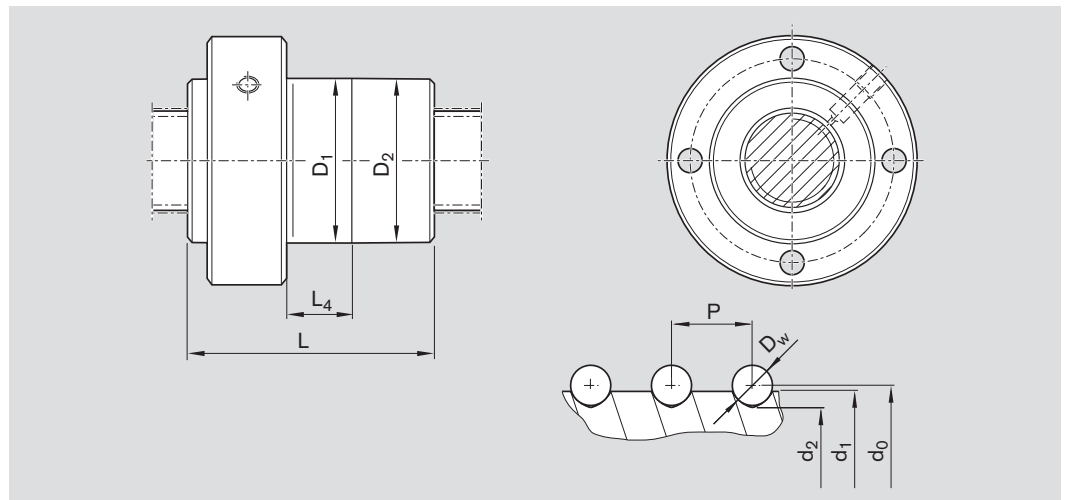
5.1 Principles

5.1.1 System technology

Nut dimensions

The dimensions of the nuts are specified using the same geometric parameters as for the screw.

The number of ball track turns around the screw is also given.



Ball nut geometry

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

Nut sizes

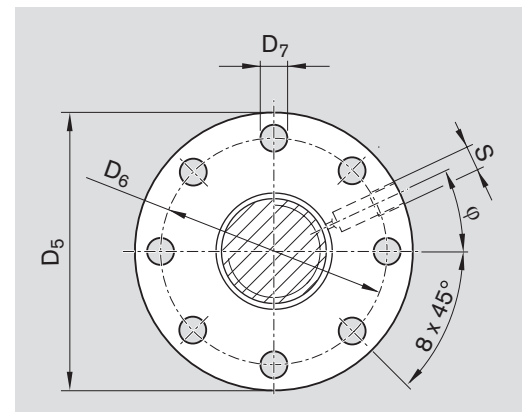
The nominal screw diameter d_0 , lead P , thread direction, ball diameter D_W and the number of ball track turns form the nut's size designation: $d_0 \times P \times D_W - i$

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

Flanged nut connection dimensions

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

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



Example of the mounting hole pattern for a flanged ball nut

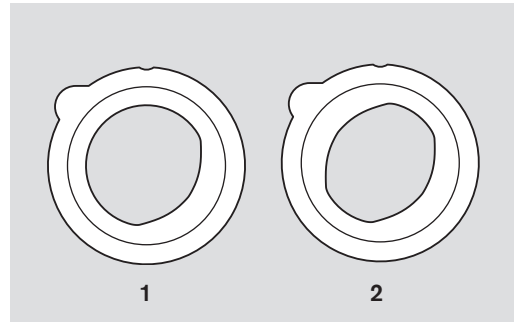
5 Ball screw drives

5.1 Principles

5.1.1 System technology

Sealing system

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

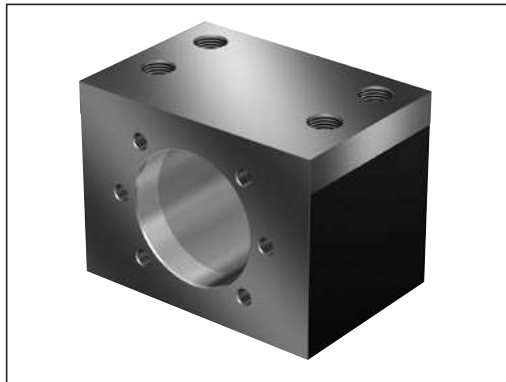


Seals for single-start (1) and two-start (2) screws

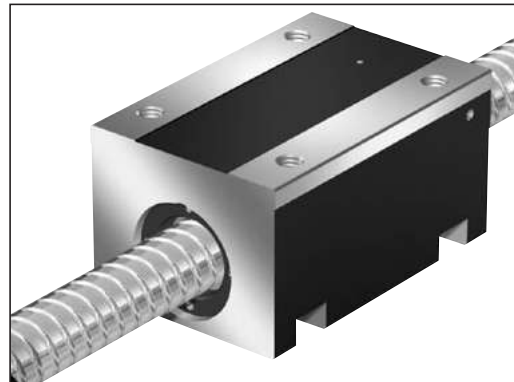
Nut housings

Rexroth provides nut housings for easy and low-cost fastening of the nut to the adjacent structure. Nut housings are precision components that can be installed with a minimum of effort. They elimi-

nate the need for customer-built mounting brackets or expensive processing of cast iron parts.



Nut housing for flanged nut



Nut housing for cylindrical nut

End bearings

Normally, a fixed-floating bearing combination is selected for the screw's end fixity. Generally the fixed bearing is on the drive side. Rexroth offers matched bearing-pillow block units that eliminate the need for costly customer-built designs and

the search for suitable bearings (see also section 5.1.4.2). Today, preloaded angular-contact thrust ball bearings are generally used for fixed bearings. For the floating bearings, deep groove ball bearings will suffice in most cases.

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

5 Ball screw drives

5.1 Principles

5.1.1 System technology

5.1.1.2 Load ratings

Load-bearing capability

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

Load ratings

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

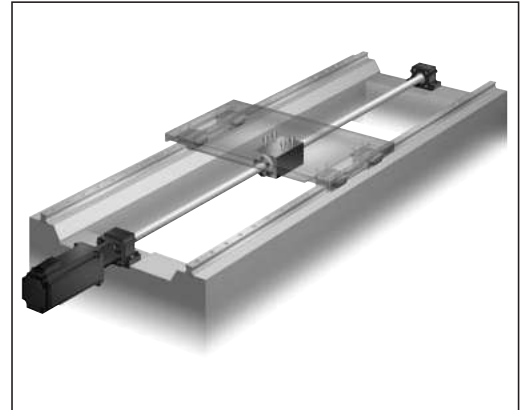
Depending on the conditions of use, the screw (buckling risk) and the end bearings can limit the permissible loads or affect the choice of product. Explanations regarding buckling and the end bearings can be found in section 5.1.3.

Definition of the dynamic load rating C

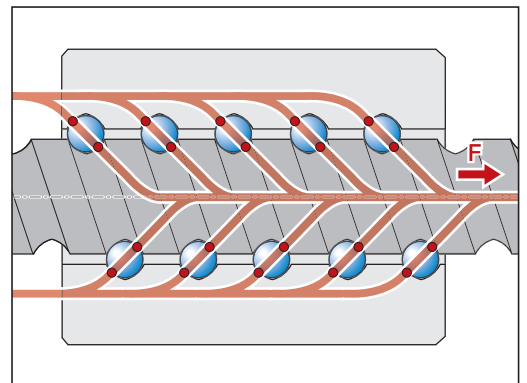
The axial force of constant magnitude and direction under which a ball screw can theoretically achieve a nominal service life of one million revolutions.

Definition of the static load rating C_0

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



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



Flow of forces in the ball screw assembly

5 Ball screw drives

5.1 Principles

5.1.1 System technology

5.1.1.3 Preload

Zero backlash Rigidity

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

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

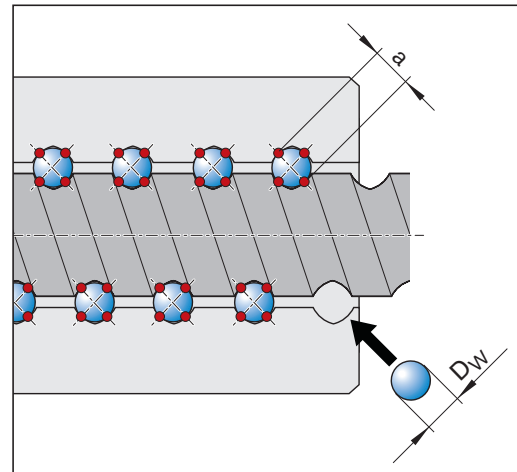
- Ball size selection
- Adjustable-preload single nut
- Double nut
- Shifted single nut

Preloading by ball size selection

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

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

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

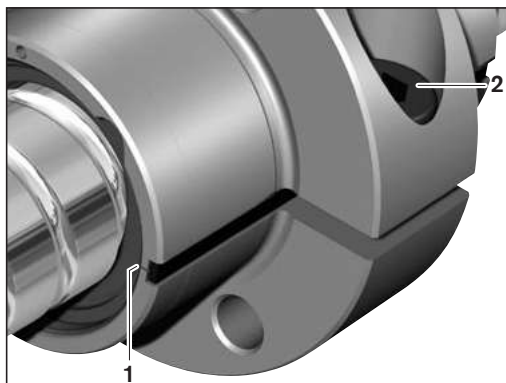


Preloading by inserting oversized balls

Adjustable-preload single nut

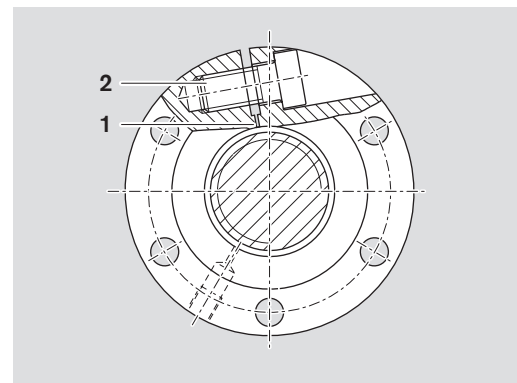
With this type of nut, the preload is adjusted via the effective diameter of the nut raceway. The nut body is provided with a narrow slot, whose width (and therefore the preload) is adjusted with an ad-

justing screw. The balls run under a slight external preload, which induces four-point contact. The advantage of this system is the infinitely variable adjustability of the preload.



Slot and adjusting screw on an adjustable-preload single nut

- 1 Slot
- 2 Adjusting screw



Preload adjustable via a tangential adjusting screw

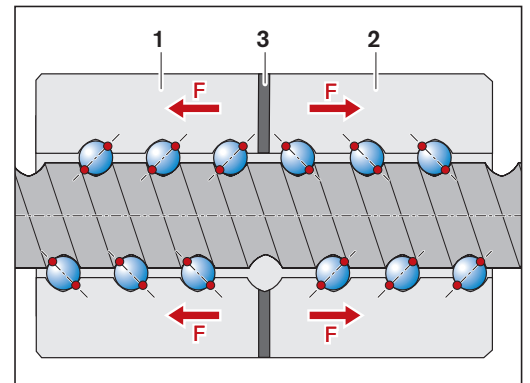
5 Ball screw drives

5.1 Principles

5.1.1 System technology

Double nut

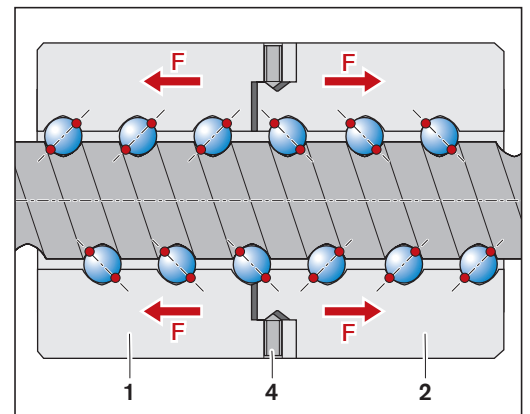
In this type of preload generation, two single nuts are tensioned against each other to a defined level and then secured. This produces a two-point contact combined with an O-arrangement. The operating force is transferred by either of the two nut halves, depending on the direction of the applied load.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



Double nut with spacer ring

Double nut for machine tools

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

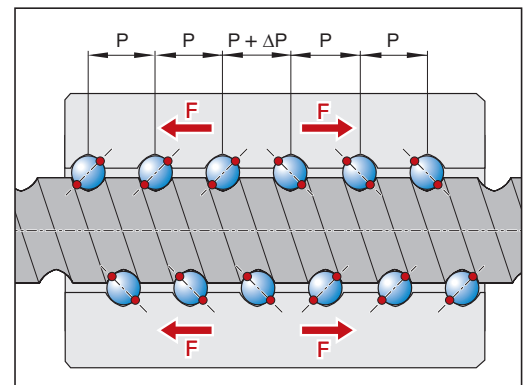


Double nut from the machine tool series

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

Shifted single nut

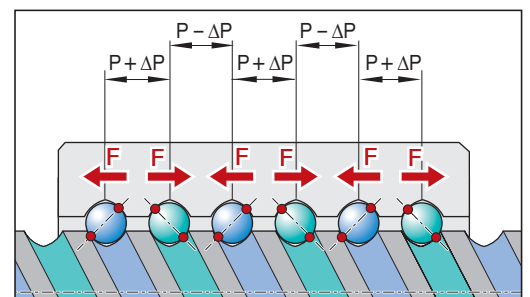
With shifting, the operating principle of the double nut is applied to a single nut. A specific lead offset ΔP is produced in the thread at a point half-way along the nut. The ball track turns before and after the lead offset are thereby tensioned against each other. Just as with double nuts, a two-point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



Shifting within a ball track

Shifting within a ball track turn

Shifting between threads



Shifting between threads

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

5 Ball screw drives

5.1 Principles

5.1.1 System technology

5.1.1.4 Rigidity

Definition of rigidity

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

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

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

Rigidity of a ball screw assembly

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

Definition of the overall axial rigidity R_{tot}

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

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

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

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

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

Rigidity of the bearing R_{aL}

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

Rigidity of the nut unit R_{nu}

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

Rigidity of the screw R_S

The rigidity of the screw depends on the screw's cross-sectional area, the screw length, the position of the nut unit on the screw and the type of bearing used. The rigidity of the screw is calculated according to DIN 69051 Part 6. The following two installation examples illustrate the method for

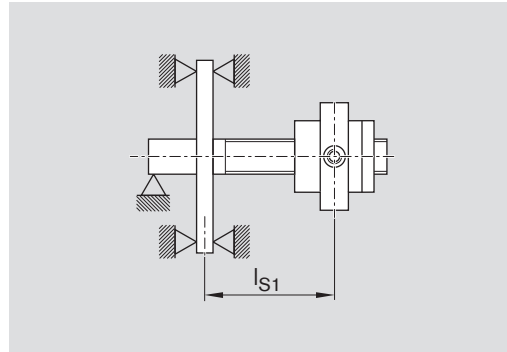
calculating the screw rigidity. The product catalogs contain corresponding tables which can be used to compare the screw and nut rigidities at the pre-selection stage. The screw rigidity details relate to a screw length of 1 meter.

5 Ball screw drives

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Example 1: Ball screw shaft fixed at one end



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

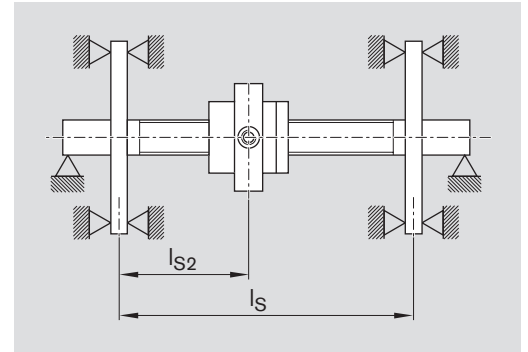
$$(5-3) \quad R_{S1} = \frac{\pi \cdot (d_0 - D_W \cdot \cos \alpha)^2 \cdot E}{4 \cdot l_{S1} \cdot 10^3} \left(\frac{\text{N}}{\mu\text{m}} \right)$$

By inserting the values for the material ($E = 210,000 \text{ N/mm}^2$) and the ball track geometry ($\alpha = 45^\circ$) and combining the dimensionless values we obtain the following simplified formula:

$$(5-4) \quad R_{S1} = 165 \cdot \frac{(d_0 - 0.71 \cdot D_W)^2}{l_{S1}} \left(\frac{\text{N}}{\mu\text{m}} \right)$$

R_{S1} = rigidity of screw with shaft fixed at one end (N/ μm)
 R_{S2} = rigidity of screw with shaft fixed at both ends (N/ μm)
 E = elasticity modulus (N/ mm^2)
 d_0 = nominal diameter (mm)
 D_W = ball diameter (mm)

Example 2: Ball screw shaft fixed at both ends



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

$$(5-5) \quad R_{S2} = \frac{\pi \cdot (d_0 - D_W \cdot \cos \alpha)^2 \cdot E}{4 \cdot l_{S2} \cdot 10^3} \cdot \frac{l_S}{l_S - l_{S2}} \left(\frac{\text{N}}{\mu\text{m}} \right)$$

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

$$(5-6) \quad R_{S2} = 165 \cdot \frac{(d_0 - 0.71 \cdot D_W)^2}{l_{S2}} \cdot \frac{l_S}{l_S - l_{S2}} \left(\frac{\text{N}}{\mu\text{m}} \right)$$

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

$$(5-7) \quad R_{S2\text{min}} = 660 \cdot \frac{(d_0 - 0.71 \cdot D_W)^2}{l_S} \left(\frac{\text{N}}{\mu\text{m}} \right)$$

l_{S1} = distance between bearing and nut (mm)
 l_{S2} = distance between bearing and nut (mm)
 l_S = distance between bearing and bearing (mm)
 α = contact angle between the ball and the raceway ($^\circ$)

5 Ball screw drives

5.1 Principles

5.1.1 System technology

5.1.1.5 Accuracy

Standard

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

The most important terms, tolerances and tests are explained in the following paragraphs:

- Travel deviations and variations
- Run-outs and location deviations
- Drag torque variations

Travel deviations and variations

Even with the most advanced production techniques, it is impossible to produce a ball screw with no deviations. The amount of travel deviation is evaluated according to a series of tolerance

grades. The evaluation is performed in three steps that are explained below, with reference to the chart on the opposite page.

Travel compensation

If required, a target value for the travel deviation is determined before beginning the actual evaluation process. This target travel deviation, known as the travel compensation c , is the desired deviation

from the nominal lead within the useful travel. It is determined by the user and depends on the conditions of use and the specific application. The standard value for travel compensation is zero.

Evaluation over the entire useful length

The travel deviation is evaluated first of all over the entire useful length. The actual travel deviation is recorded over the useful travel l_u . Since the actual travel deviation is difficult to evaluate, the mean actual travel deviation is determined as the geometric mean of the measurements recorded over the useful travel. The difference between the travel compensation value and the mean actual travel deviation at the end of the useful travel constitutes the tolerance for mean actual travel deviation e_p , which gives an indication of the screw's average precision over the useful travel l_u . However, a screw might exhibit widely varying accuracy errors that virtually canceled each other out over the useful travel. The tolerance for mean actual travel deviation e_p would then indicate a high level of precision, even

though the screw displayed significant errors. For this reason, the bandwidth of travel variation around the mean actual travel deviation must also be analyzed. To do this, two lines are drawn parallel to the line for mean actual travel deviation to form an "envelope" enclosing the actual travel deviation curve. The distance between these parallel lines is called the permissible travel variation within the useful travel and is denoted by the symbol v_{up} . This is the bandwidth for travel variations.

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

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

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

Evaluation over a reference length

A second evaluation step is performed for a reference length of 300 mm. In this case, the value v_{300p} for the specified tolerance grade may not be exceeded at any point on the screw within the 300mm length.

The tolerance for travel variations within 300 mm of travel is verified for positioning ball screws and for transport ball screws.

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

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

5 Ball screw drives

5.1 Principles

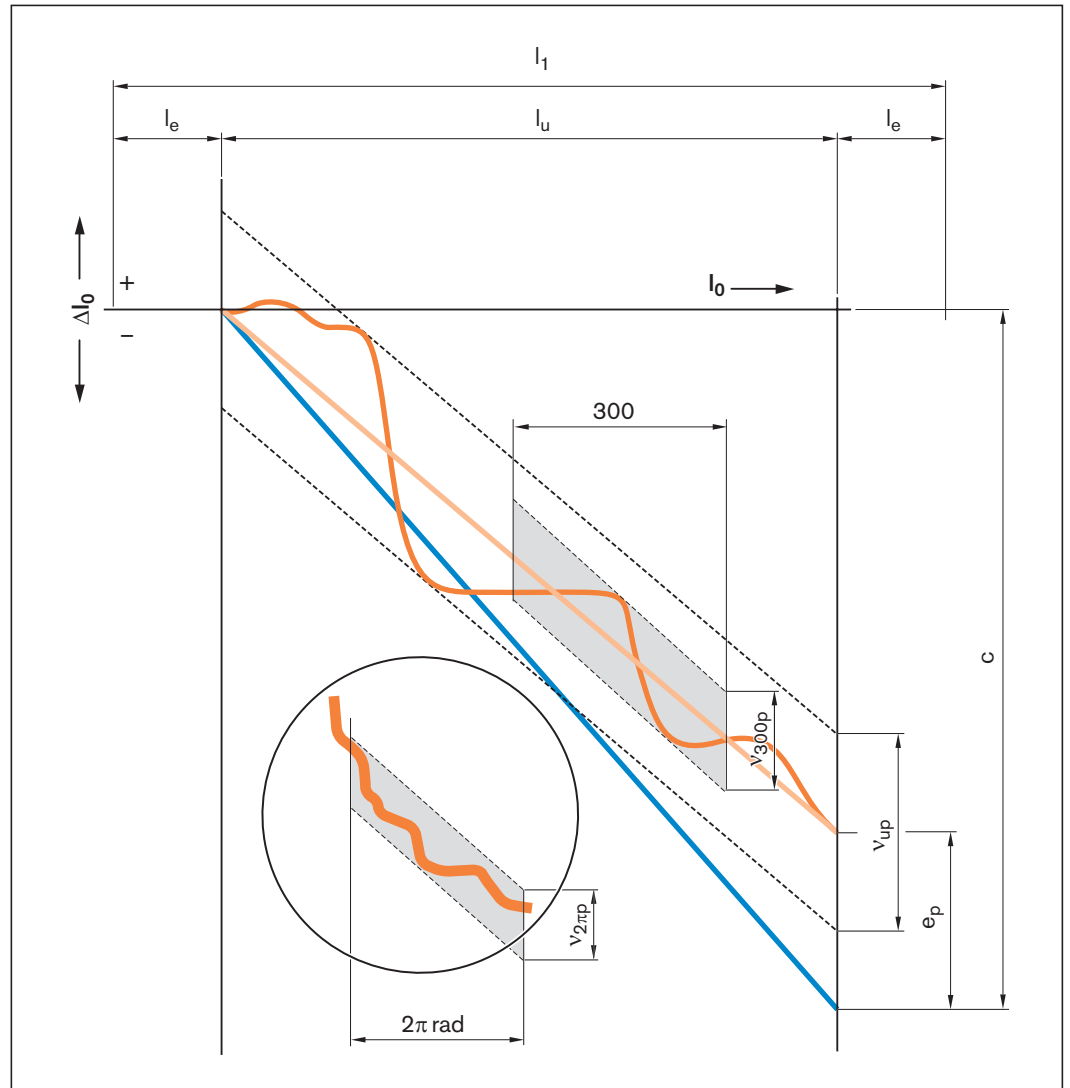
5.1.1 System technology

Evaluation per revolution

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

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

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



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

- Travel compensation (target travel deviation)
- Actual travel deviation
- Mean actual travel deviation
- - Permissible travel variation (tolerance "envelope")
- l_0 = nominal travel
- l_1 = thread length
- Δl_0 = travel deviation
- l_e = excess travel (non-usable length)
- l_u = useful travel
- c = travel compensation for useful travel (standard: $c = 0$)
- e_p = tolerance for mean actual travel deviation
- v_{up} = permissible travel variation within useful travel l_u
- v_{300p} = permissible travel deviation within 300 mm travel
- $v_{2\pi p}$ = permissible travel deviation within one revolution (2π rad)

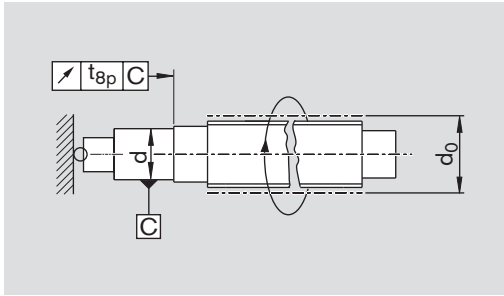
5 Ball screw drives

5.1 Principles

5.1.1 System technology

Run-outs and location deviations (geometric accuracy)

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



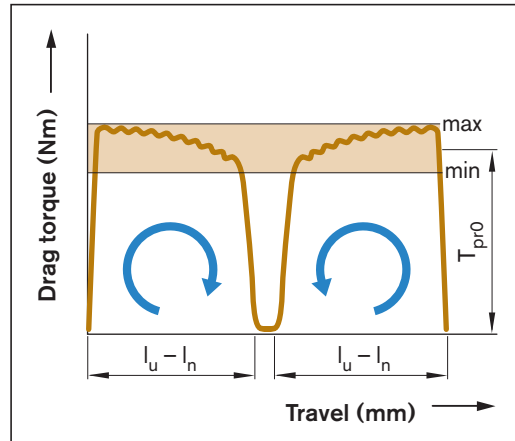
Example: Axial run-out t_{8p} of the shaft (bearing) face of the ball screw shaft in relation to the bearing diameter

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

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

Drag torque variations

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



Qualitative representation of the dynamic drag torque

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

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

5.1.1.6 Dynamic drag torque

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

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

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

5 Ball screw drives

5.1 Principles

5.1.1 System technology

5.1.1.7 Characteristic speed and maximum linear speed

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

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

$$d_0 \cdot n \leq 150000 \text{ mm/min}$$

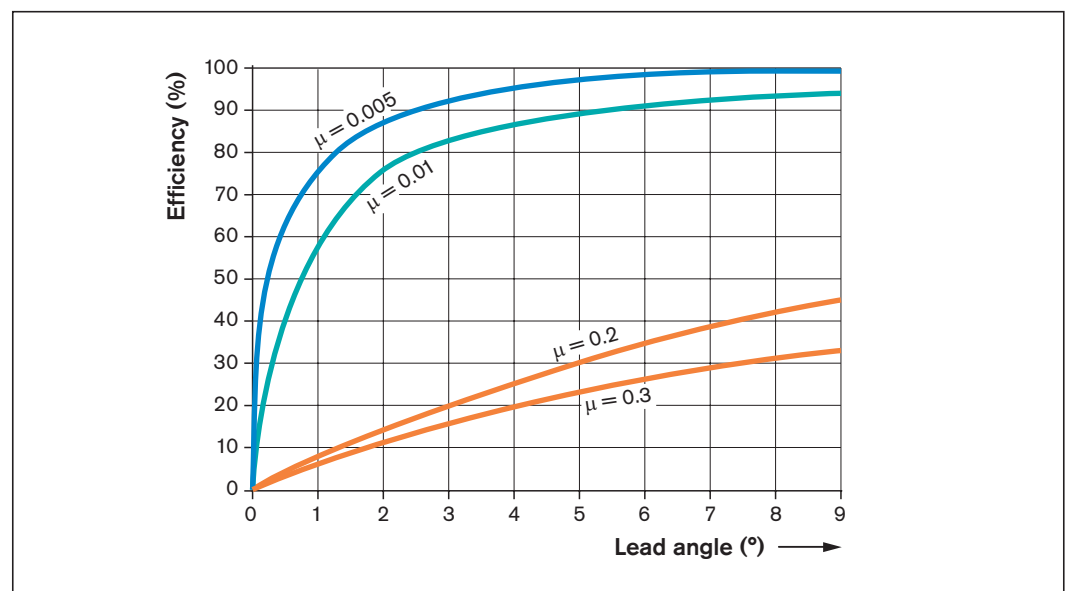
$$(5-9) \quad v_{\max} = \frac{(d_0 \cdot n) \cdot P}{d_0} = \frac{150000 \frac{\text{mm}}{\text{min}} \cdot P}{d_0}$$

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

5.1.1.8 Mechanical efficiency

Due to the use of rolling elements, sliding friction is largely avoided (with the exception of contact seals, for example). Ball screw assemblies therefore have a very high degree of mechanical

efficiency. As a result, they can be put to very effective use as powerful machine components for a wide variety of applications.



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

- Ball screw assembly with two-point contact
 - Ball screw assembly with four-point contact
 - Acme screw
- μ = friction coefficient

5 Ball screw drives

5.1 Principles

5.1.1 System technology

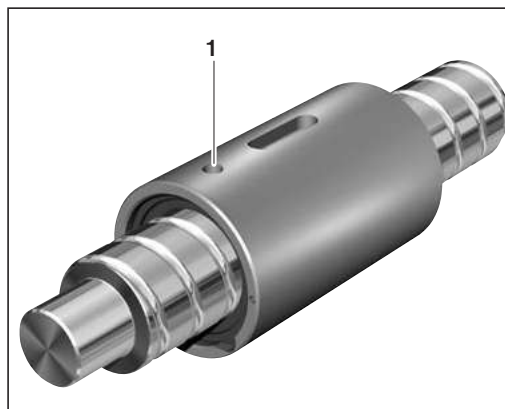
5.1.1.9 Lubrication

Lube port

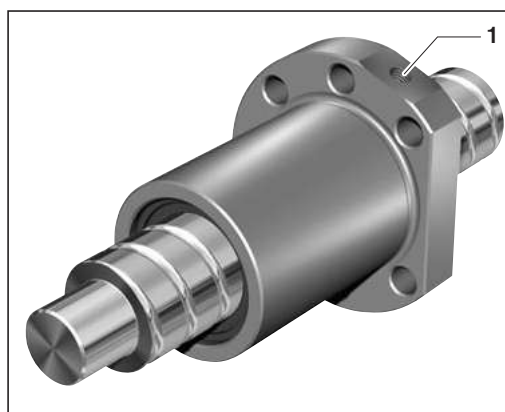
Just like every other type of rolling bearing, ball screws must be adequately lubricated. They can be lubricated with oil or grease. Normally, the ball nut is lubricated via a lube port (1). In the case of flanged nuts, the lube port is situated on the flange. The lubricant quantities depend on the size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

Short stroke

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



Cylindrical ball nut



Flanged ball nut

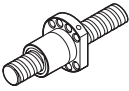
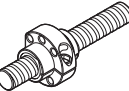
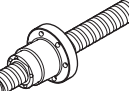
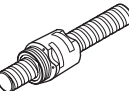
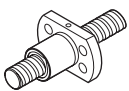
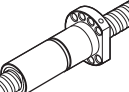
5 Ball screw drives

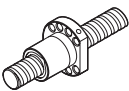
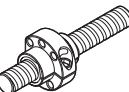
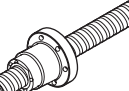
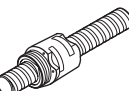
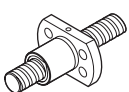
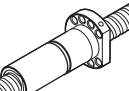
5.1 Principles

5.1.2 Product selection

5.1.2.1 Guide to choosing the right product

System characteristics

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

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

+++ Very good

++ Good

+ Satisfactory

o Adequate

1) at the same linear speed

5 Ball screw drives

5.1 Principles

5.1.2 Product selection

5.1.2.2 Product selection procedure

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

bearings and the drive unit must also be checked by performing the necessary calculations.

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

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

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

5.1.2.3 Pre-selection

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

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

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

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

5 Ball screw drives

5.1 Principles

5.1.3 Calculations

5.1.3.1 Defining the requirements

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

The required parameters relate to:

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

Ball screw drive

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

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

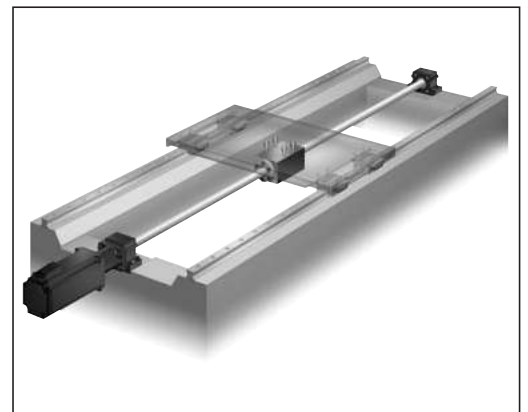


Example: Ball screw assembly with flanged nut from the Standard series, as determined at the pre-selection stage

Application layout

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

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



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

5 Ball screw drives

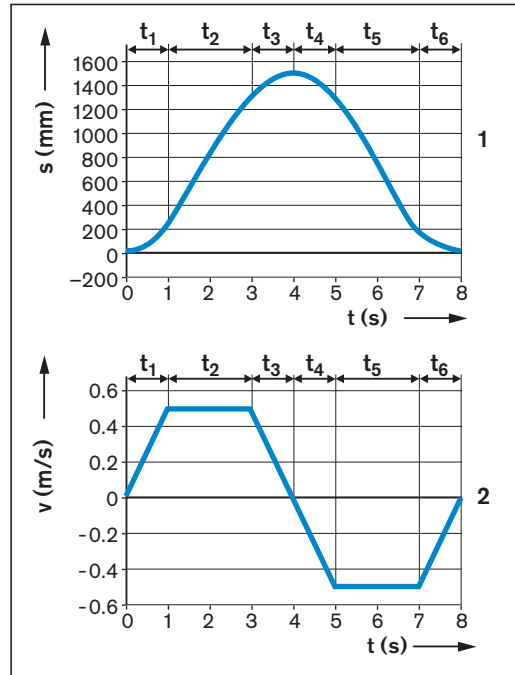
5.1 Principles

5.1.3 Calculations

Dynamic cycle

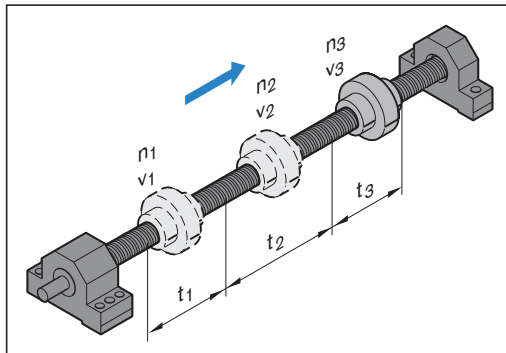
The next step is to determine a reference cycle for the application. This cycle represents the expected dynamic motion sequences and forms the basis for calculating the nominal life. Cycles consist of several phases representing the individual operating steps that the ball screw drive is to perform (e.g.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

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

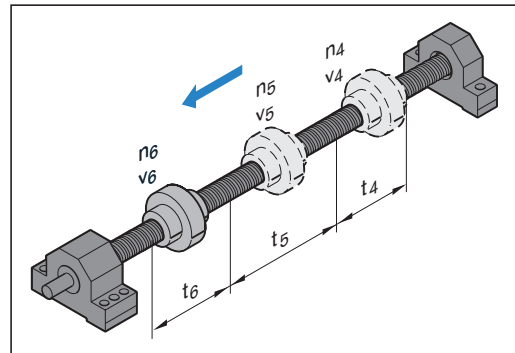


Example of a simple dynamic cycle

- 1 Travel-time curve
- 2 Speed-time curve



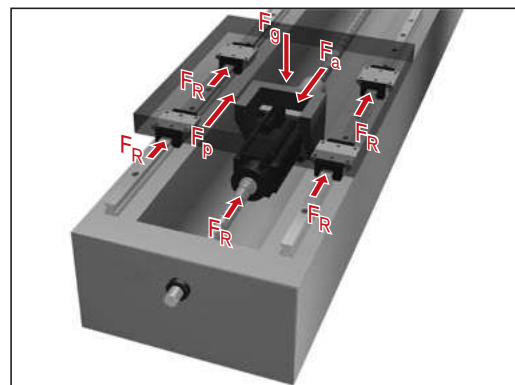
Example of a simple dynamic cycle: forward stroke



Example of a simple dynamic cycle: return stroke

Load scenario

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



Example showing an axially effective process force F_p

5 Ball screw drives

5.1 Principles

5.1.3 Calculations

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

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

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

5.1.3.2 Life expectancy

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

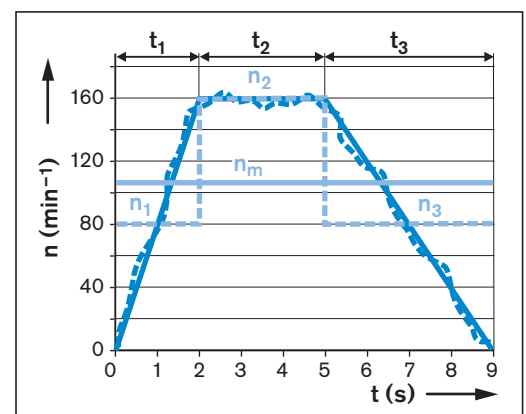
For applications with varying speed and load parameters, the average rotary speed n_m and the equivalent (average) dynamic load F_m must be calculated first. The nominal life calculation is then performed using these average values.

Average rotary speed

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

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

n_m = average rotary speed (min⁻¹)
 $n_1 \dots n_n$ = rotary speed in phases 1 ... n (min⁻¹)
 $q_{t1} \dots q_{tn}$ = discrete time steps in phases 1 ... n (%)



Example of a simple rotary speed cycle

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

5 Ball screw drives

5.1 Principles

5.1.3 Calculations

Taking preload into account

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

$$(5-12) \quad F_{pr} = X_{pr} \cdot C$$

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

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

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

$$(5-13) \quad F_{lim} = 2.8 \cdot F_{pr}$$

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

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

A distinction therefore has to be made between two cases:

Case 1: $F > F_{lim}$

If the external axial force acting on the ball screw assembly in phase n is greater than the lift-off force, then the preload need not be considered when calculating the nominal life.

Preload may be disregarded:

$$F_n > 2.8 \cdot F_{pr}$$

$$(5-14) \quad F_{eff n} = |F_n|$$

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

Case 2: $F \leq F_{lim}$

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

Preload must be considered:

$$F_n \leq 2.8 \cdot F_{pr}$$

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

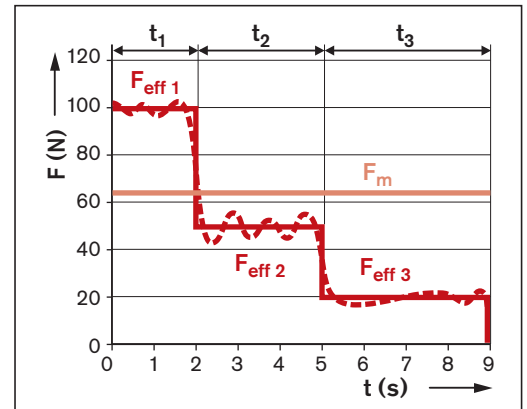
5 Ball screw drives

5.1 Principles

5.1.3 Calculations

Equivalent dynamic axial load

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



Example of a simple load cycle at constant rotary speed

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

Equivalent dynamic axial load at constant speed:

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

Equivalent dynamic axial load at varying speed:

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

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

5 Ball screw drives

5.1 Principles

5.1.3 Calculations

Nominal life

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

Nominal life in revolutions

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

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

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

Nominal life in hours

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

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

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

Machine operating hours

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

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

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

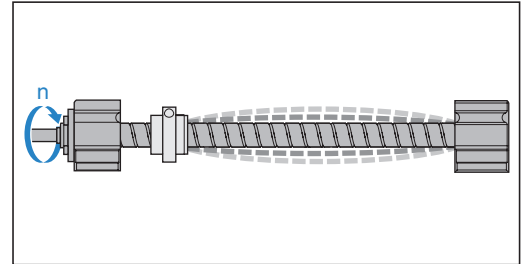
5 Ball screw drives

5.1 Principles

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5.1.3.3 Critical speed

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



Bending vibrations (screw whip)

Critical speed n_k

The critical speed n_k depends on:

- the type of end bearings, coefficient f_{nk}
- the screw's core diameter d_2
- the critical screw length l_n , i.e. the maximum unsupported screw length.

In the case of ball nuts with backlash, the critical screw length is the same as the bearing-to-bearing length l_1 . In preloaded systems, the position of the ball nut is taken into account.

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

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

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

Driven nuts

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

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

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

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

End fixity	Coefficient f_{nk}
<p>fixed-fixed</p>	27.4
<p>fixed-floating</p>	18.9
<p>floating-floating</p>	12.1
<p>fixed-free</p>	4.3

5 Ball screw drives

5.1 Principles

5.1.3 Calculations

5.1.3.4 Permissible axial load on screw (buckling load)

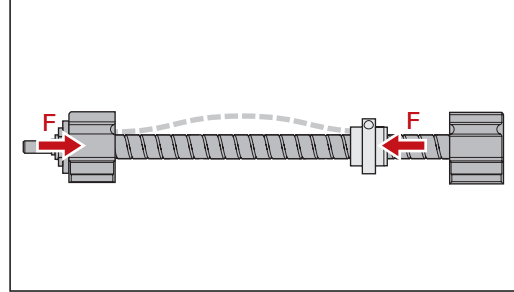
Buckling

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

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

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

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



Buckling caused by axial loading

Permissible axial screw load F_k

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

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

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

The following measures can be taken to avoid buckling:

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

End fixity	Coefficient f_{Fk}
<p>fixed-fixed</p>	40.6
<p>fixed-floating</p>	20.4
<p>floating-floating</p>	10.2
<p>fixed-free</p>	2.6

5 Ball screw drives

5.1 Principles

5.1.3 Calculations

5.1.3.5 End bearings

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

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

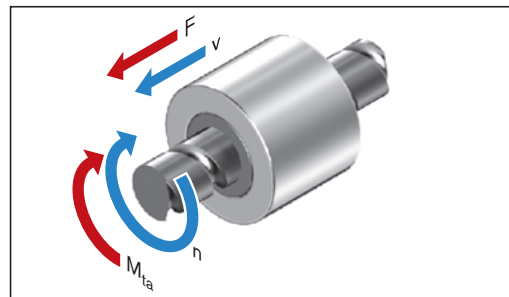
5.1.3.6 Drive torque and drive power

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

Definitions of drive torque and transmitted torque

Drive torque M_{ta} :

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



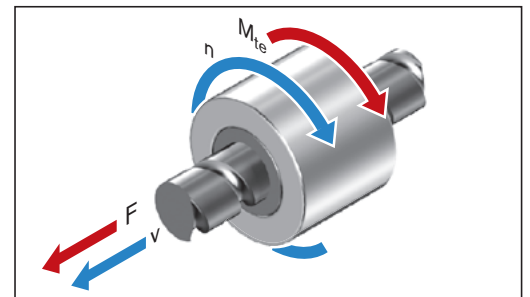
Conversion of rotary to linear motion

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

M_{ta} = drive torque (Nm)
 M_{te} = transmitted torque (Nm)
 F = operating load (N)
 P = lead (mm)

Transmitted torque M_{te} :

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



Conversion of linear to rotary motion

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

η = mechanical efficiency (-)
 $\eta \approx 0.9$ for drive torque
 $\eta' \approx 0.8$ for transmitted torque

Drive power

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

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

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

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

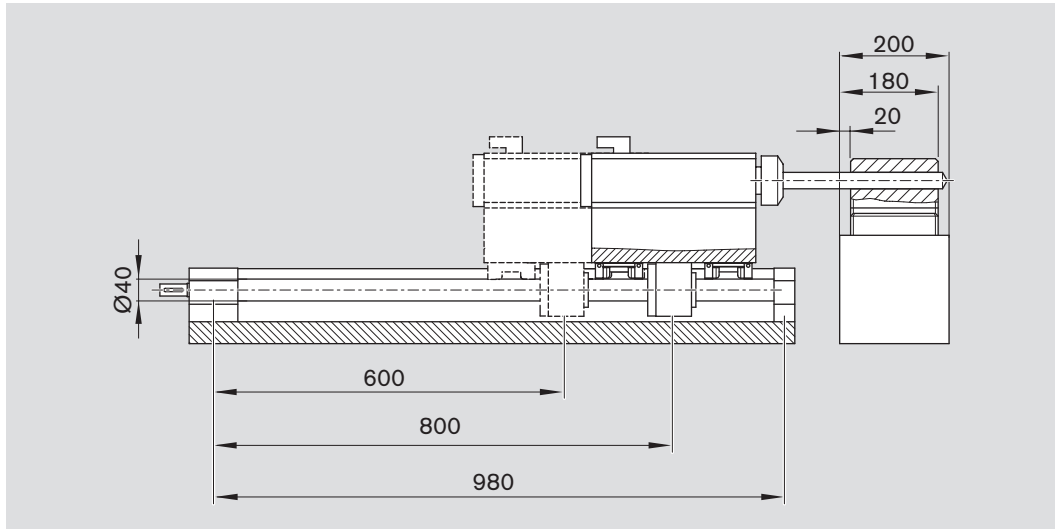
5 Ball screw drives

5.1 Principles

5.1.3 Calculations

5.1.3.7 Calculation example

Drilling station



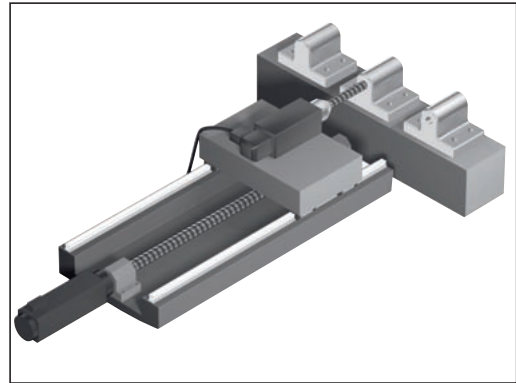
Dimensions for calculating the ball screw drive

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

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

Technical data of the application

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



Calculation example for a drilling station

Ball screw assembly

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

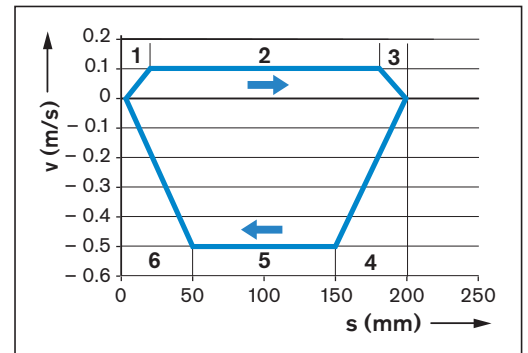
5 Ball screw drives

5.1 Principles

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Dynamic cycle

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

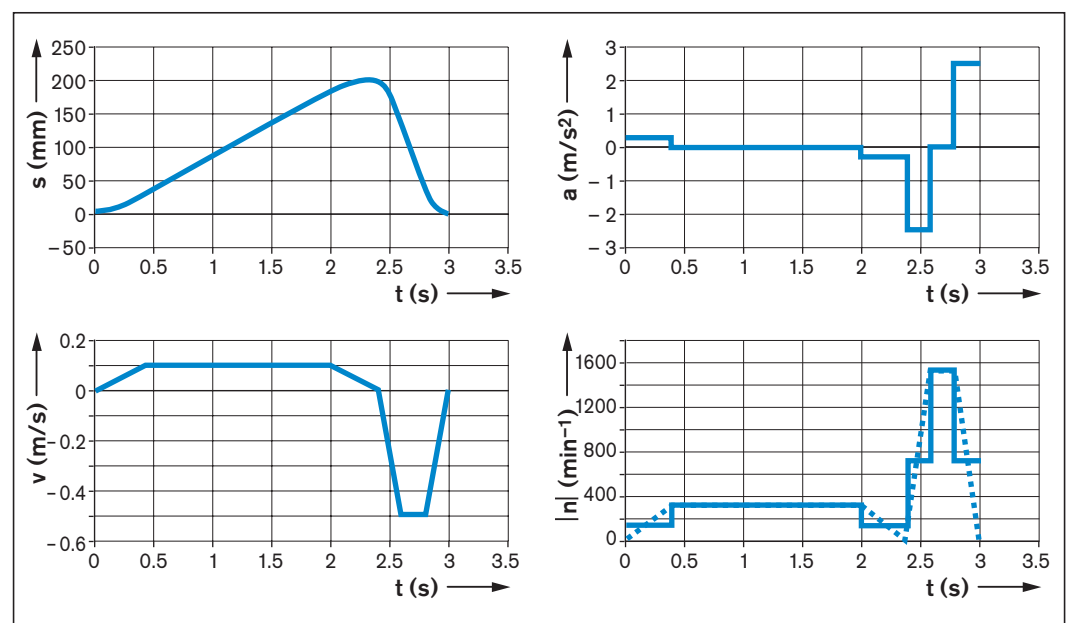


Reference cycle

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

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

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



Kinematic data for the reference cycle

5 Ball screw drives

5.1 Principles

5.1.3 Calculations

Discrete time steps

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

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

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

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

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

...

Loads

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

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

$$F_n = F_{an} + F_{Rn} + F_{pn}$$

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

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

...

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

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

5 Ball screw drives

5.1 Principles

5.1.3 Calculations

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

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

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

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

Taking preload into account

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

The following ball screw has been selected:

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

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

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

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

In all phases F_n is $< F_{lim}$.

The preload must therefore be taken into account in the nominal life calculation.

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

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

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

$$F_{eff2} = \left(\frac{|F_2|}{2.8 \cdot F_{pr}} + 1 \right)^{\frac{3}{2}} \cdot F_{pr} = \left(\frac{4650 \text{ N}}{2.8 \cdot 1895 \text{ N}} + 1 \right)^{\frac{3}{2}} \cdot 1895 \text{ N} = 4871 \text{ N}$$

...

**Intermediate results:
effective axial load**

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

5 Ball screw drives

5.1 Principles

5.1.3 Calculations

Equivalent dynamic axial load

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

F_m according to formula (5-17):

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

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

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

Nominal life in revolutions

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

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

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

Nominal life in hours

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

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

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

Machine operating hours

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

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

$$L_{h \text{ machine}} = 43096 \text{ h} \cdot \frac{100\%}{50\%} = 86191 \text{ h}$$

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

5 Ball screw drives

5.1 Principles

5.1.3 Calculations

Critical speed

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

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

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

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

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

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

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

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

Permissible axial screw load

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

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

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

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

$$F_k = 416023 \text{ N}$$

The permissible axial load should be calculated with a safety factor of at least 2. Since this calculation example concerns a reference cycle that

$$F_{k\text{perm}} = \frac{416023 \text{ N}}{6} = 69337 \text{ N}$$

The two checks performed therefore show that neither the critical speed nor buckling represent a problem for the chosen ball screw assembly.

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

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

takes no disruptions into account (e.g. tool wear, tool breakage or collision in fast approach mode), a safety factor of 6 is chosen here.

The ball screw assembly can therefore be safely used in this application.

5 Ball screw drives

5.1 Principles

5.1.4 Design notes

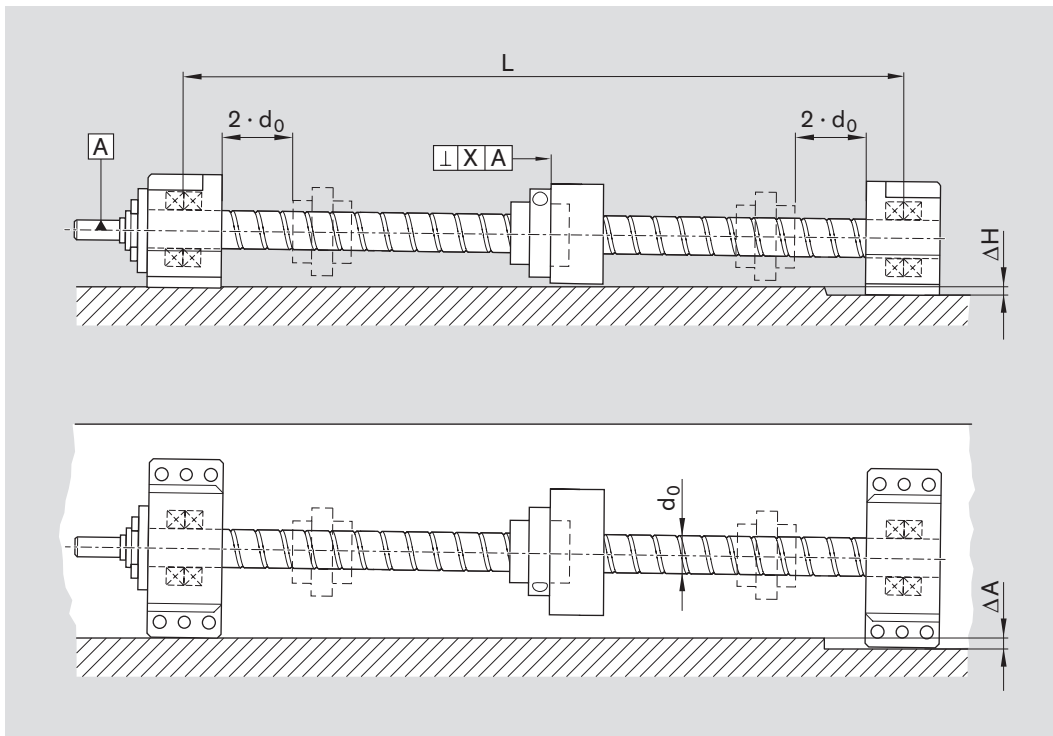
To ensure that a ball screw drive can actually achieve the calculated service life and performance, its system-related requirements and limitations must be taken into account at the design stage. Screw drives are not suitable for transferring radial forces and torques that may be caused

by misalignments during installation. The following sections illustrate the most important principles for achieving designs that will be compatible with the ball screw system and its requirements.

5.1.4.1 Adjoining structures and installation tolerances

When using ball screw drives, the specified installation tolerances must be observed when designing and building the adjoining structures. The first basic principle is: The higher the ball screw drive's precision and preload, the more accurate the adjoining structures must be. This

applies in particular to applications in which the nut travels right up to the end bearings since, in this area, the risk of distortive stresses and therefore of additional loads is very high.



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

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

5 Ball screw drives

5.1 Principles

5.1.4 Design notes

Installation tolerances

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

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

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

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

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

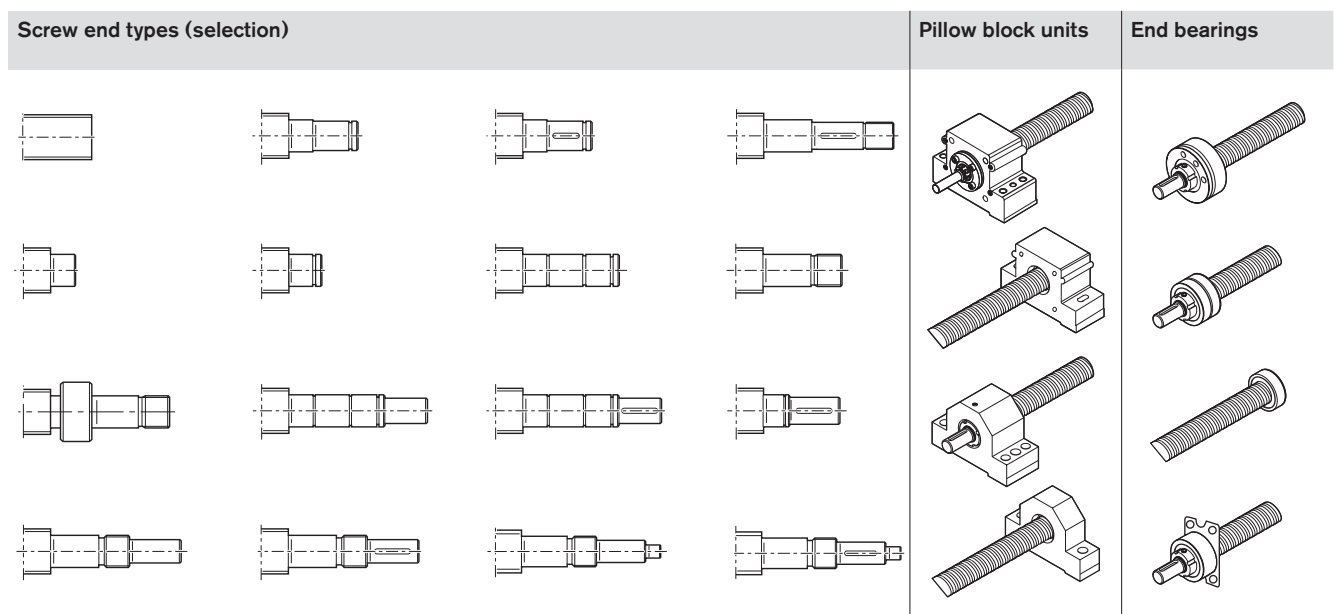
Preload	X mm	ΔH mm	ΔA mm
Backlash	0.10	0.10	0.10
2% of C	0.08	0.08	0.08
5% of C	0.05	0.05	0.05
7% of C	0.02	0.02	0.02
10% of C	0.02	0.02	0.02

5.1.4.2 Guidelines for economical constructions

Use of standard elements

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

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



5 Ball screw drives

5.1 Principles

5.1.4 Design notes

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

- It is not necessary to use a precision screw if a direct position measuring system is available on the travel axis. The required positioning accuracy can be achieved using the position measuring system and the electronic controls.
- High accuracy can also be achieved by one-time tuning even without a direct measurement system, if the lead deviations are entered in the control system (see section 5.1.1.5).
- Only use a preloaded system when necessary. The higher the preload, the higher the requirements will be for the accuracy of the adjoining structures.
- Check whether cylindrical nuts can be used. Cylindrical nuts make it possible to build more compact structures. No fixing holes have to be drilled for this type of nut.
- Use screw-in nuts whenever possible.

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

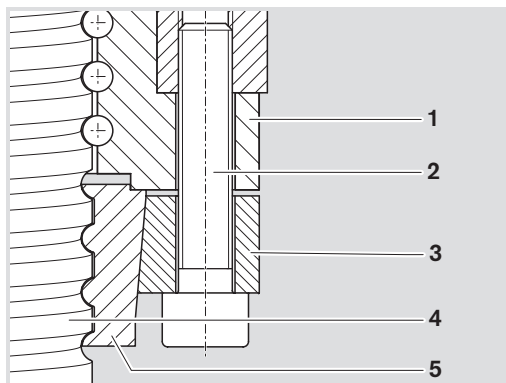
- Only center the nut in the housing if necessary. Centering increases the processing costs and makes higher demands on manufacturing tolerances.
- Whenever possible, use standardized nuts. Common ball nut sizes and types, as specified in the DIN 69051 standard, are usually more affordable. Rexroth identifies the nut types belonging to this category in its product catalogs. Using such nuts also offers the advantage of ready availability if a replacement is required.
- Dimension the ball screw drive correctly. The more well-founded the design data is, the more accurate the calculations will be, which helps to avoid expensive oversizing of the assembly.

5.1.4.3 Safety nuts for vertical applications

In vertical applications, it should be remembered that a failure of the ball nut could result in uncontrolled dropping of the entire carriage. A safety nut can be used to protect against such crashes. Safety nuts are profiled with a negative profile to that of the screw. In normal service, there is a clearance between the two profiles. In the event of failure of the ball nut, the profiles on the safety nut and the screw come into contact, acting as a

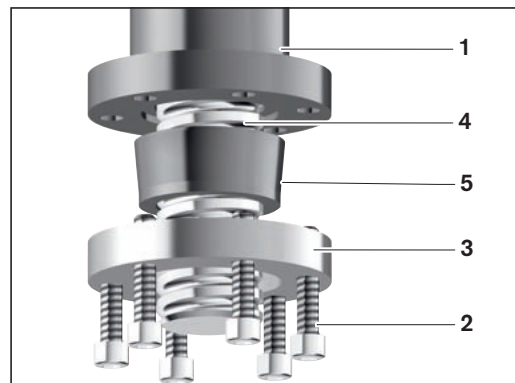
jam to prevent the ball nut and the carriage assembly from falling.

The safety nut is fastened to the ball nut by means of a clamping ring. For vertical applications it must always be fitted below the ball nut, as this is the only position that can ensure the safety function.



Structural design of a safety nut

- 1 Ball nut
- 2 Socket head cap screws
- 3 Clamping ring



Mounting configuration with safety nut

- 4 Screw
- 5 Safety nut

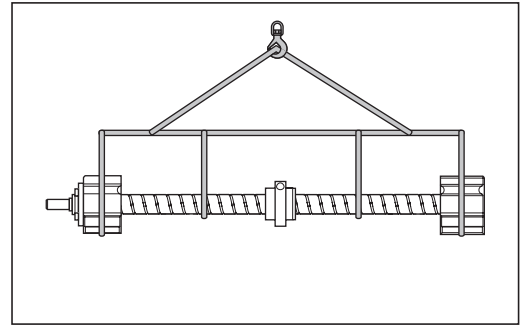
5 Ball screw drives

5.1 Principles

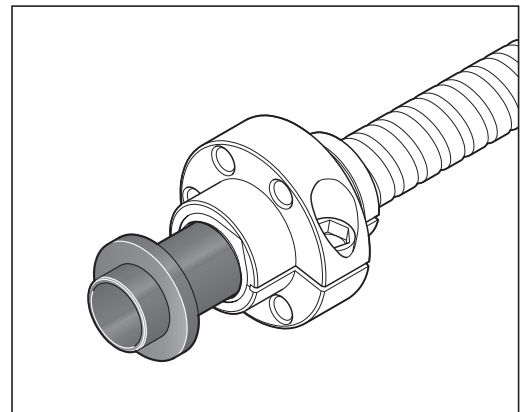
5.1.5 Mounting instructions

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

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



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



Mounting arbor

5 Ball screw drives

5.2 Ball nuts

5.2.1 Single nuts

5.2.1.1 System characteristics

Most common ball nuts

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

High performance Compact design**Ball nut series**

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

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

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

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

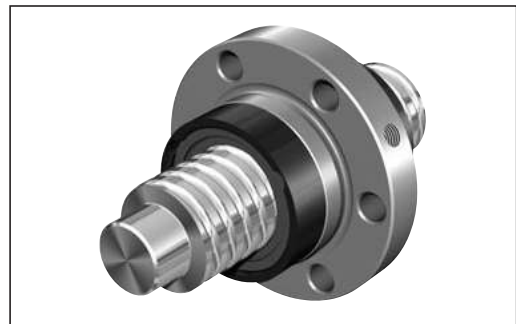
Single nuts in the Speed series are characterized by their high maximum permissible linear speeds. This is due to the fact that their leads are equal to or greater than the nominal diameter.

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

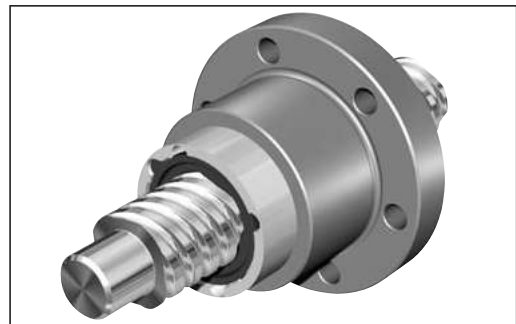
Examples of single nuts from the Rexroth range are shown in the illustrations at right.



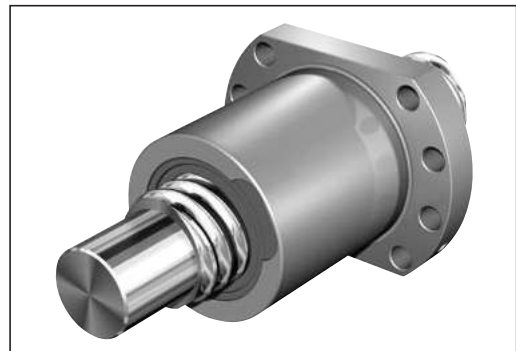
Cylindrical single nut from the Standard series



Single nut from the ECOplus series with plastic recirculation caps



Single nut from the Speed series with plastic recirculation caps



Single nut from the Machine Tool series

5.2.1.2 Application areas

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

5 Ball screw drives

5.2 Ball nuts

5.2.2 Standard series single nuts

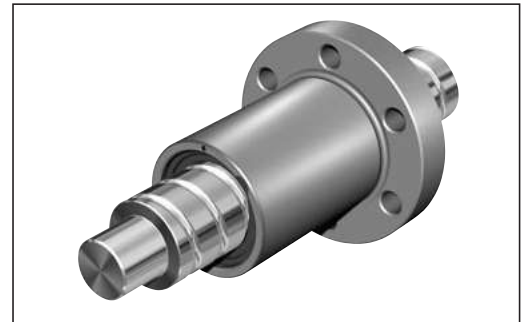
5.2.2.1 System characteristics

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

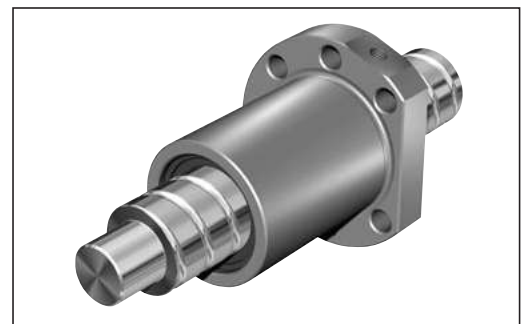
Standardized design Single nuts in the Standard series are available with mounting dimensions per DIN 69051 part 5 or with Rexroth mounting dimensions. Matching nut housings and end bearings are also available in several versions for all Standard single nuts.

5.2.2.2 Application areas

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



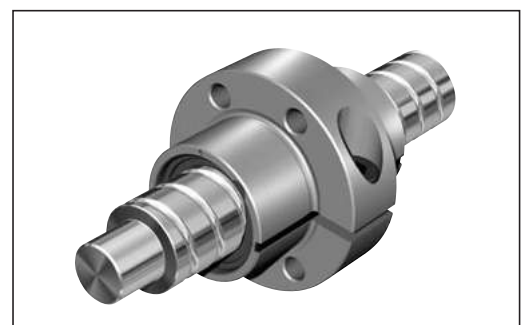
Standard series flanged nut



Standard series flanged nut with single flat



Standard series cylindrical nut



Standard series adjustable preload nut

5 Ball screw drives

5.2 Ball nuts

5.2.3 Miniature series single nuts

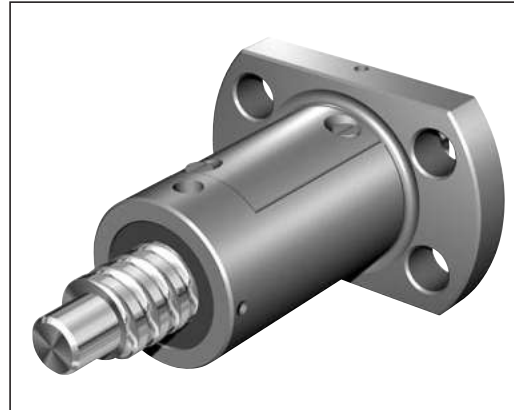
5.2.3.1 System characteristics

Nominal diameter less than 12 mm

Miniature ball screw assemblies are conventionally understood to be systems with a nominal diameter of less than 12 mm. Miniaturized nut geometries are achieved through the use of optimized recirculation systems and very small balls. These ball screws are usually not preloaded or only very slightly preloaded to ensure the smoothest possible travel.

Low preload

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



Miniature series flanged nut

5.2.3.2 Application areas

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

Typical application areas are:

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

5 Ball screw drives

5.2 Ball nuts

5.2.4 eLINE series single nuts

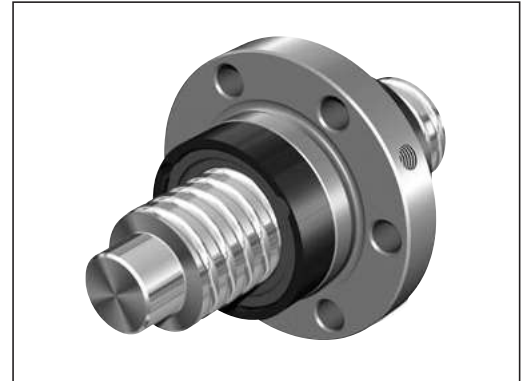
5.2.4.1 System characteristics

Economical

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

Without preload

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



eLINE series flanged nut with recirculation caps

5.2.4.2 Application areas

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

Typical application areas are:

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



eLINE series screw-in nut

5 Ball screw drives

5.2 Ball nuts

5.2.5 Double nuts

5.2.5.1 System characteristics

Preload
Two-point contact

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

Modular construction

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

Double nut series

Rexroth offers double nuts in two series:

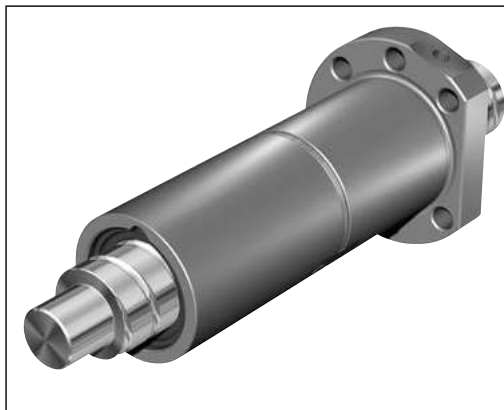
- Standard series
- Machine Tool series

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

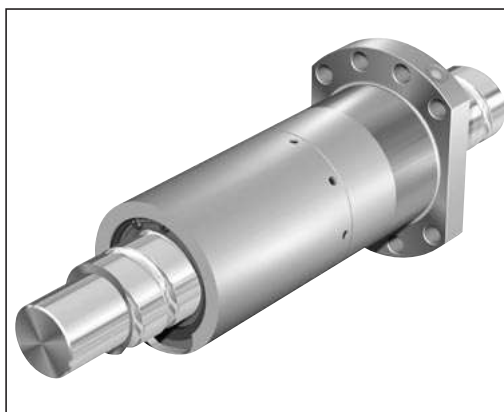
The illustrations at right show two typical double nuts.

5.2.5.2 Application areas

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



Standard series double nut



Machine Tool series double nut

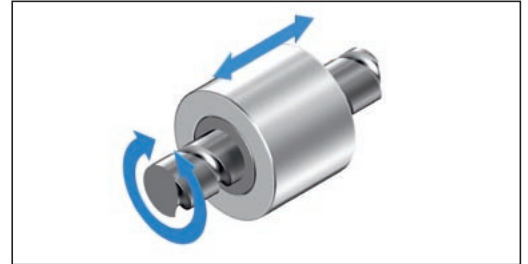
5 Ball screw drives

5.3 Drive units

5.3.1 Drive units with driven screw

5.3.1.1 System characteristics

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



Operating principle of the driven screw

Rexroth offers drive units with driven screw in two forms:

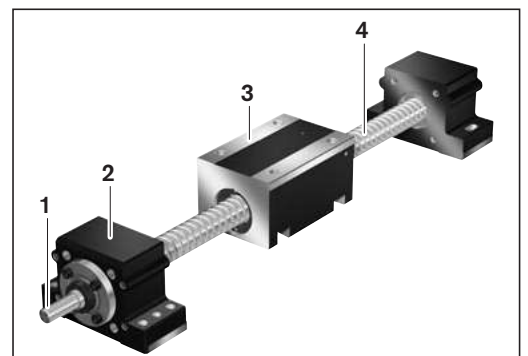
- Open drive unit AOK
- Drive unit AGK with enclosure and sealing strip

Open drive units (AOK)

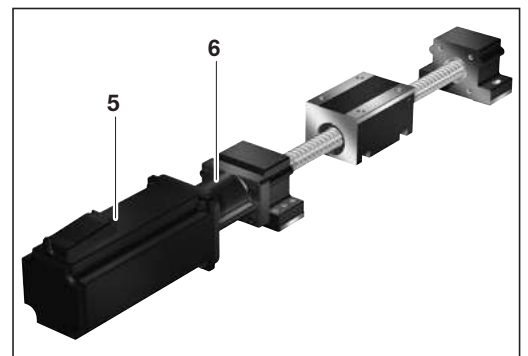
The AOK drive unit is the classic ball screw assembly with pillow blocks and pre-assembled ball nut enclosure. A motor and gear unit can be supplied along with the unit, if so requested by the customer.

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

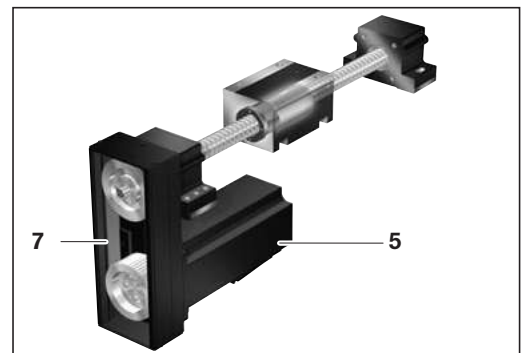
The illustrations at right show AOK drive units with various motor attachment options.



AOK drive unit



AOK drive unit with motor mount, coupling and motor



AOK drive unit with side drive timing belt and motor

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

5 Ball screw drives

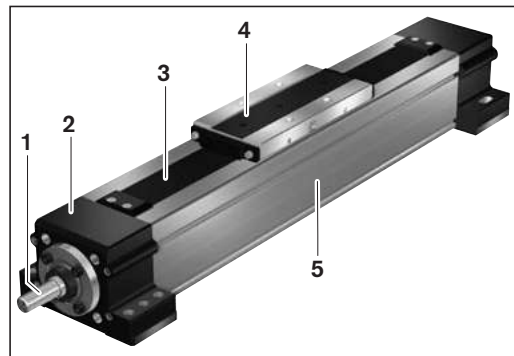
5.3 Drive units

5.3.1 Drive units with driven screw

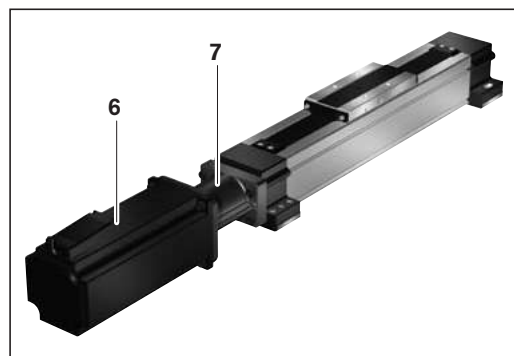
Closed drive unit (AGK)

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

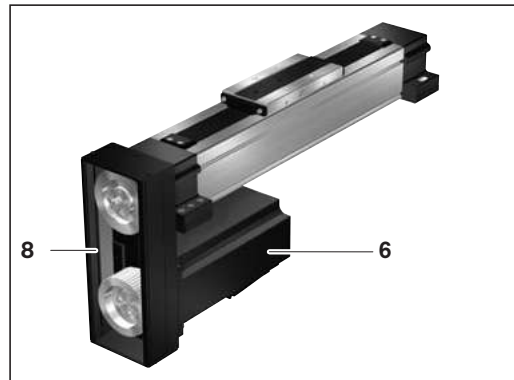
The illustrations at right show AGK drive units with various motor attachment options.



AGK drive unit



AGK drive unit with motor mount, coupling and motor

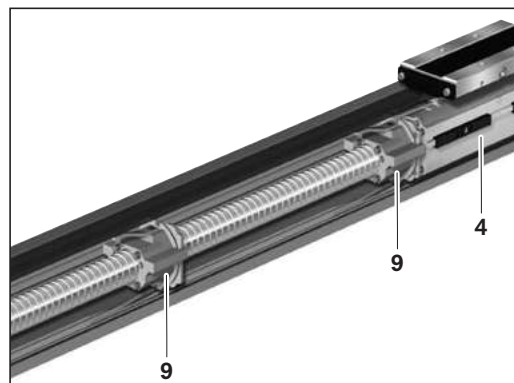


AGK drive unit with side drive timing belt and motor

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

Screw support (SS)

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



AGK drive unit with screw supports

5 Ball screw drives

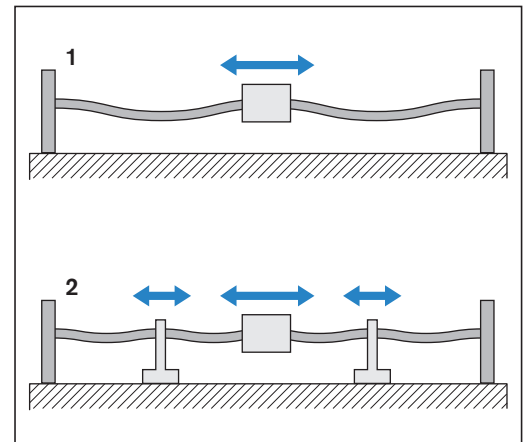
5.3 Drive units

5.3.1 Drive units with driven screw

Operating principle of the screw support

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

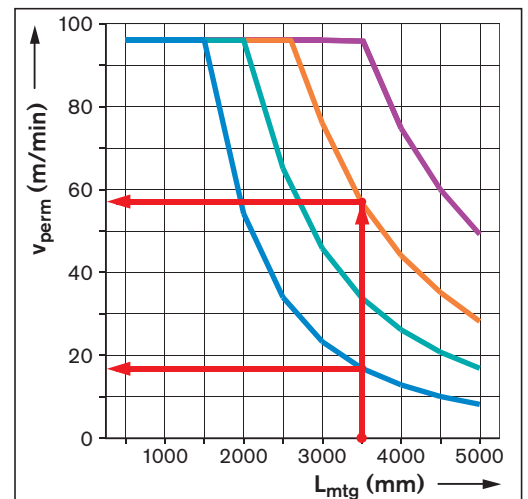
- 1 Ball screw assembly without screw supports
- 2 Ball screw assembly with one traveling screw support on each side of the nut



Operating principle of the screw support

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

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



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

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

5.3.1.2 Application areas

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

5 Ball screw drives

5.3 Drive units

5.3.2 Drive units with driven nut

5.3.2.1. System characteristics

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

Advantages

This concept provides marked advantages over the driven screw design:

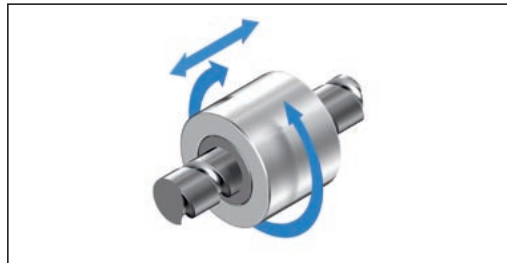
- Since the screw does not rotate, the attainable rotary speed is no longer limited by the critical screw speed. However, even with stationary screws, the natural frequency must still be taken into account. On the other hand, resonance (that is, when the rotational frequency of the nut is the same as the first order frequency) is far less critical than in systems with driven screws because of the significantly lower energy introduced by the driven nut. This is due to the optimized radial and axial run-out of Rexroth's driven nuts. The systems can theoretically be operated up to the maximum rotary speed as determined by the characteristic speed (see section 5.1.1.7). Nevertheless, the resonance range must be traversed as rapidly as possible to avoid unnecessary stresses on the system.
- The screw does not have to be set into rotation. This reduces the system's overall mass moment of inertia.

Rexroth offers drive units with driven nut in two forms:

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

FAR drive unit with belt-driven nut

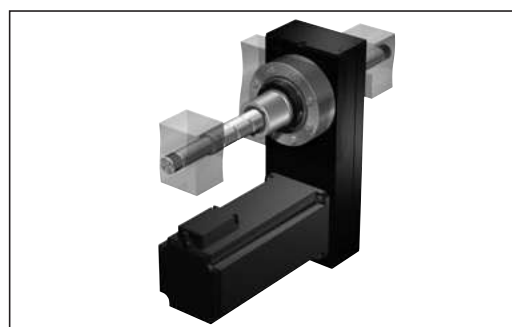
In drive units with FAR belt-driven nuts, the ball nut is driven by the motor via a toothed belt. FAR drive units are available as complete func-



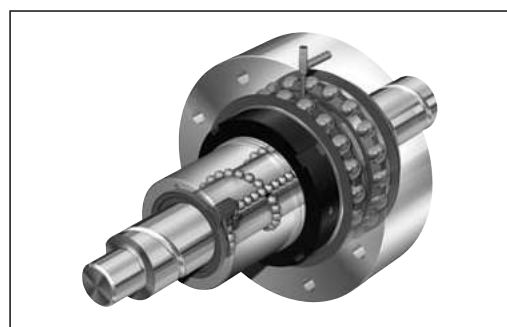
Operating principle of a driven nut

- The end fixity for the non-rotating screw can be of a simpler and therefore more economical design.
- Since the screw is stationary, it can be stretched (tensioned) with relatively little effort. This makes it possible to compensate for length variations due to temperature fluctuations.
- Thermal influences can also be compensated for by using a hollow screw with a cooling system.

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



FAR drive unit with side drive timing belt



Ball nut of a FAR drive unit

5 Ball screw drives

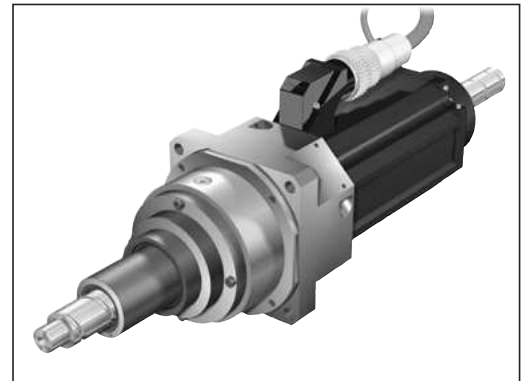
5.3 Drive units

5.3.2 Drive units with driven nut

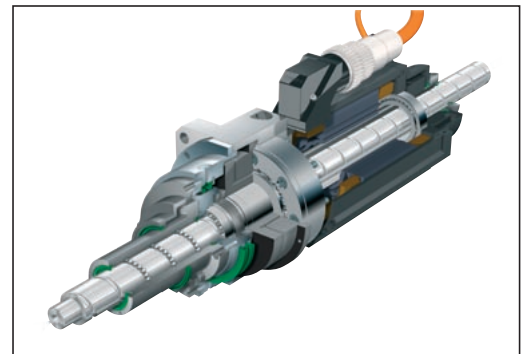
MHS drive unit with hollow shaft motor

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

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



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

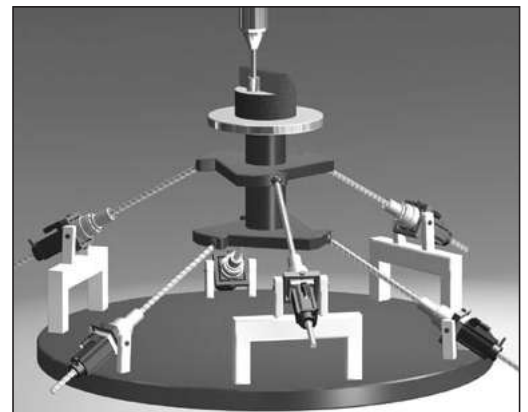


Structural design of the MHS drive unit

5.3.2.2 Application areas

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

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



Model of a hexapod with six MHS drive units

6 Linear motion systems

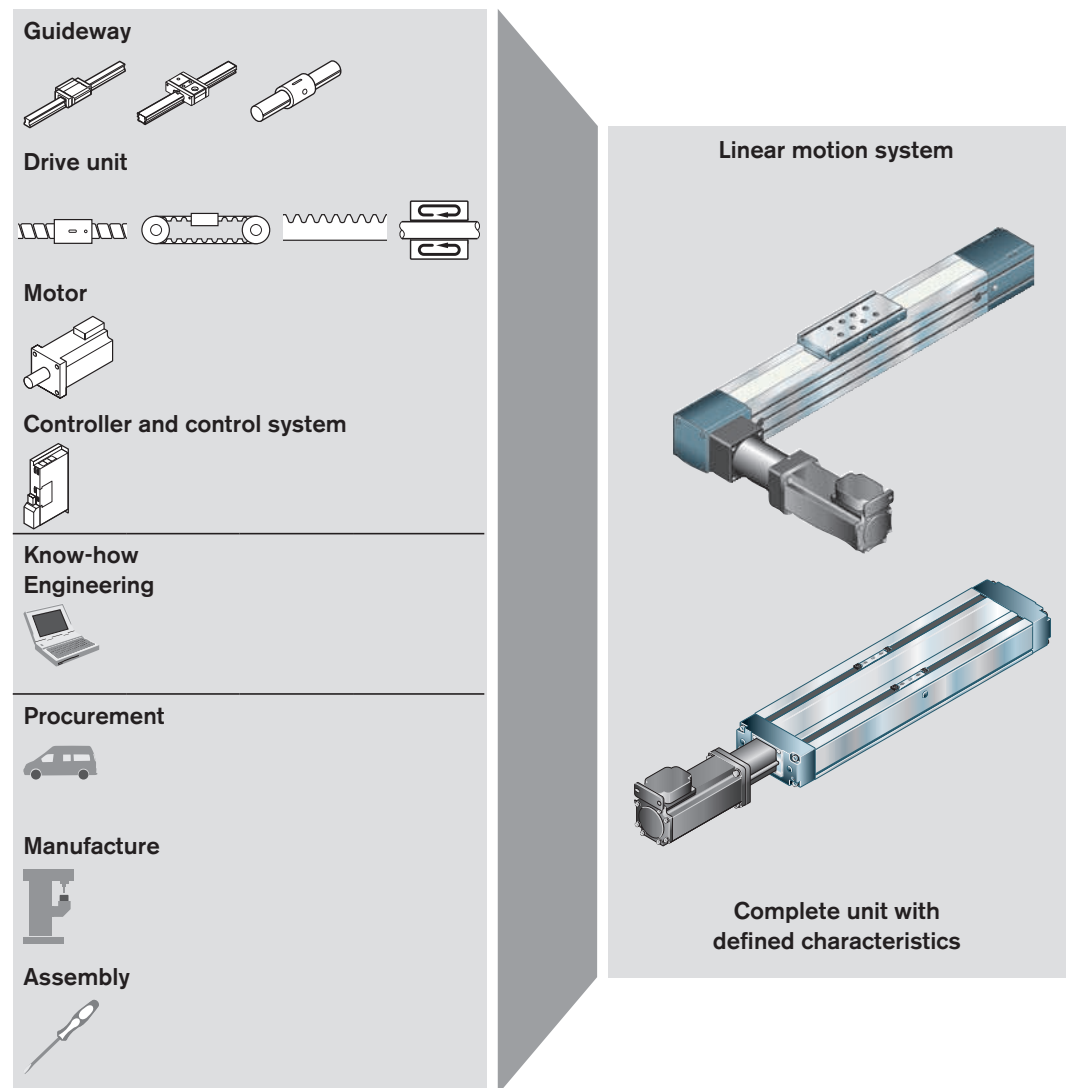
6.1 Principles

6.1.1 System technology

Innovative complete solutions

Linear motion systems are precise, ready-to-install guidance and drive systems that combine high performance with compact dimensions. Available in a wide variety of configurations, they can be used in many different industrial sectors.

Machinery and equipment can often be built more rapidly, more easily, and more cost-efficiently using standardized linear motion systems. Design, project engineering, manufacturing and logistics are all significantly simplified.

**Advantages**

Rexroth's linear motion systems offer many advantages:

- Complete product range for virtually any application
- Multiple drive options
- Versatile design allowing multi-axis combinations adapted for use with Rexroth's profile construction system
- Lengths up to 12 meters possible
- All linear motion systems can be supplied complete with motor, drive amplifier and control system.
- Scalable, customizable systems
- Reduced design and manufacturing effort
- Cost-efficient adaptations to individual customer requirements
- Extensive range of accessories
- Highly experienced technical sales and development team available for consultation

6 Linear motion systems

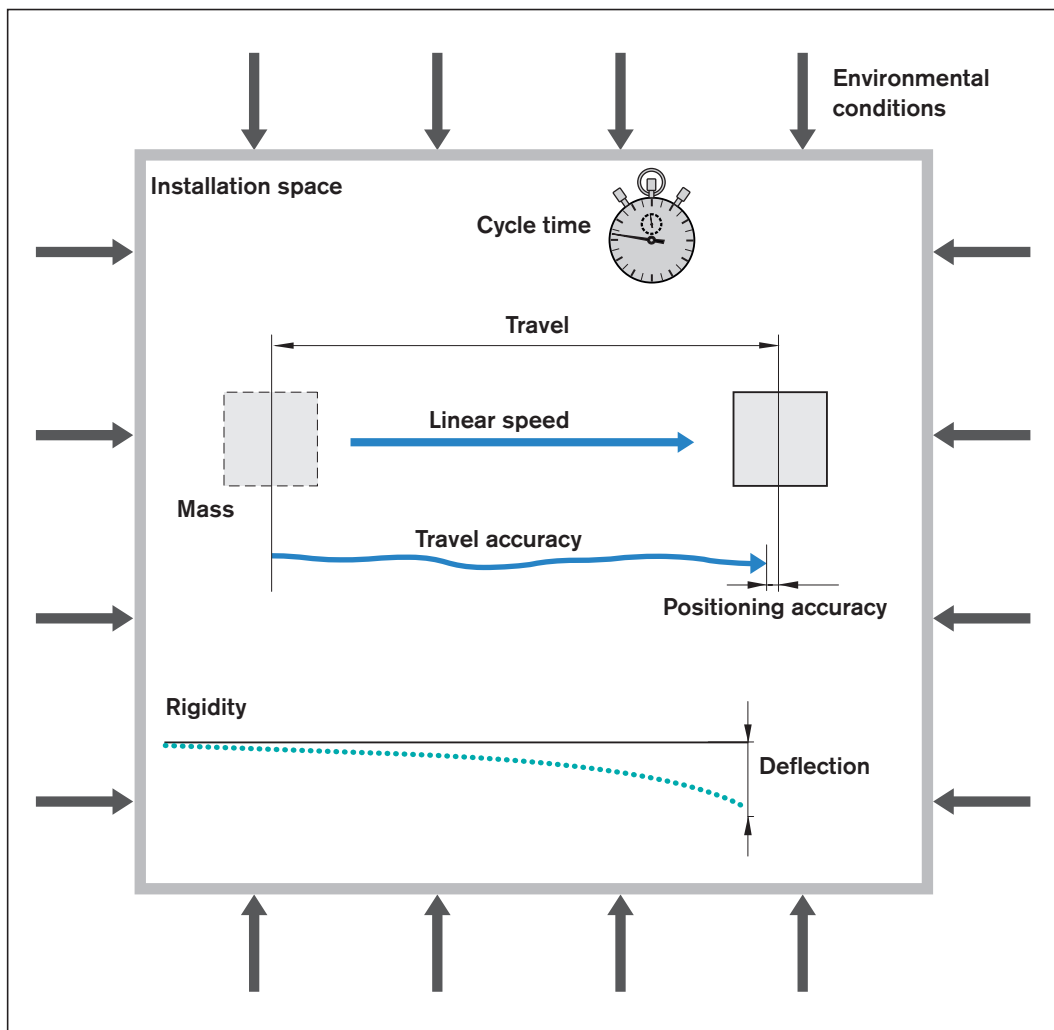
6.1 Principles

6.1.1 System technology

Customer applications for linear motion systems

A typical customer application for linear motion systems is illustrated below. A mass is to be moved over a certain distance within a defined time.

Important parameters for this are the installation space available, the prevailing environmental conditions and the accuracy and rigidity required.



Parameters for using a linear motion system

Application areas

Essentially, a linear motion system can always be used whenever a linear movement is to be automated. However, not all linear motion systems are suitable for all areas of application.

The application areas can be defined according to the tasks to be performed or by industrial sectors. Typical tasks for linear motion systems are:

- Handling (pick and place)
- Assembly
- Measurement tasks
- Processing/machining

Linear motion systems can be used in every sector. They are particularly widespread in the following areas:

- Electronics and semi-conductor manufacturing
- Medical technology and pharmaceuticals industry
- General factory automation
- Woodworking
- Food and packaging industries

6 Linear motion systems

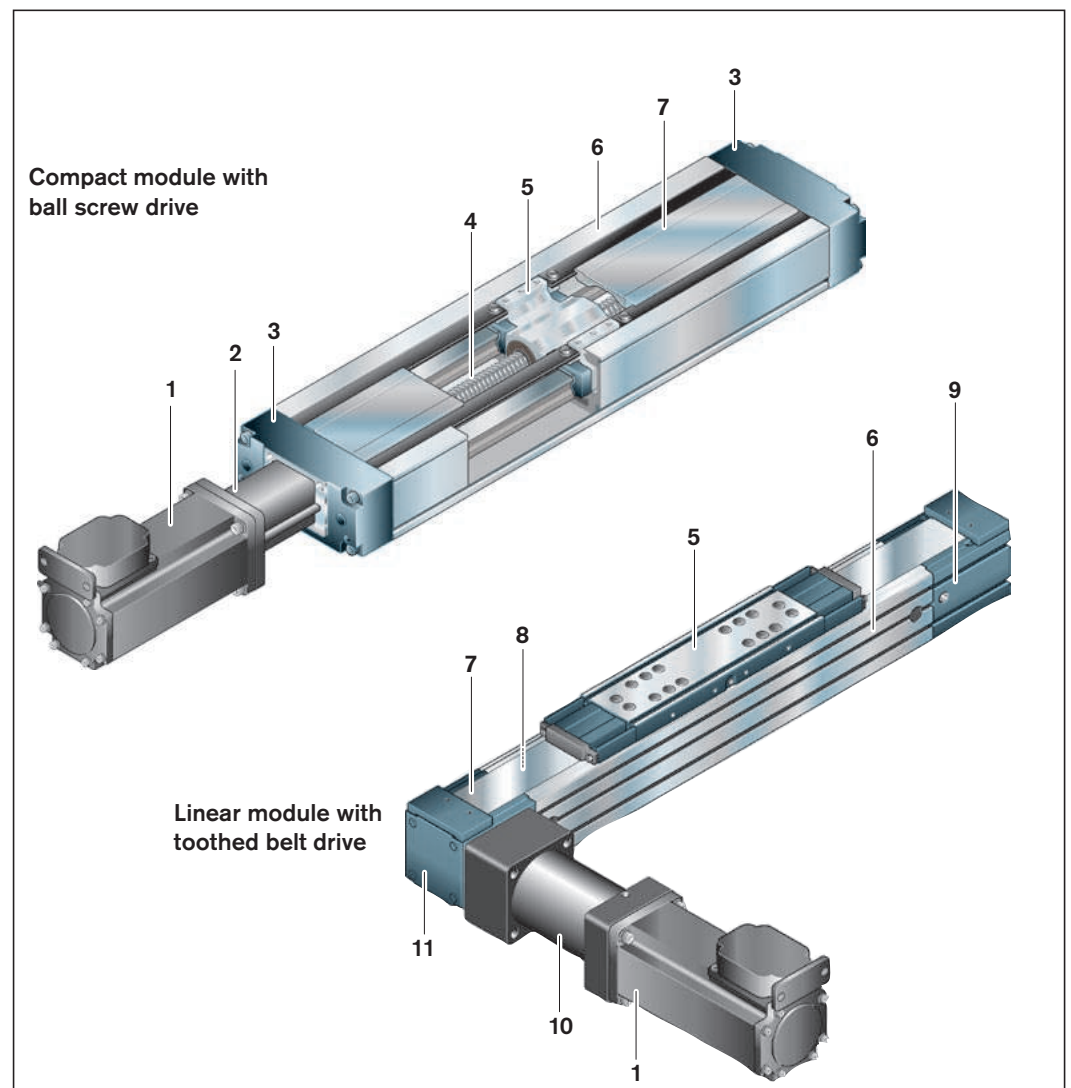
6.1 Principles

6.1.1 System technology

6.1.1.1 Basic structural design of linear motion systems

Linear motion systems always have the same basic structure. They consist of the following components:

- Load-bearing profile (frame) with guideway (6)
- Carriage with runner blocks (5)
- End blocks with bearings (3) or drive end enclosure (11) and tension end enclosure (9)
- Drive unit, i.e. ball screw drive (4), toothed belt drive (8), linear motor, etc.
- Cover (7), e.g. cover plate, sealing strip
- AC servo motor, three-phase motor or stepping motor (1) attached either directly via a motor mount with coupling (2) or via a gear unit (10), with a controller and control unit
- Switches, socket and plug, cable duct
- Optional components such as screw supports, connection plates, clamping fixtures, position measuring systems, etc.



Structural design of linear motion systems

6 Linear motion systems

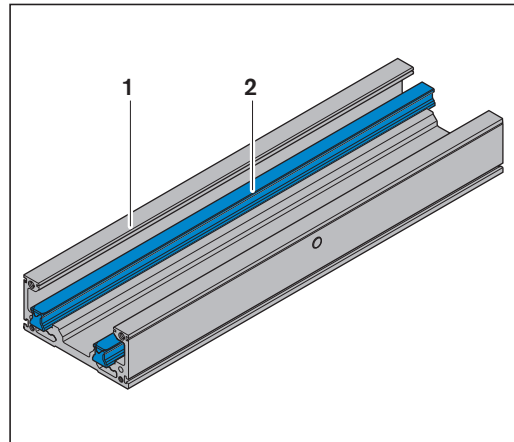
6.1 Principles

6.1.1 System technology

Frame with linear guideway

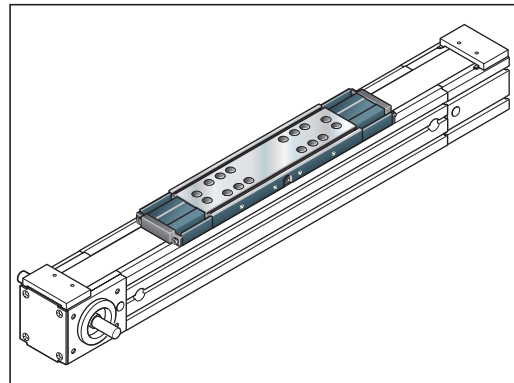
The guideway assembly consists of a load-bearing profile as the frame (1) and the linear guides (2).

In most cases, the frame is fastened to the customer's mounting base via clamping fixtures. It usually consists of an anodized aluminum extrusion profile, making the linear motion system highly rigid. The anodized coating enhances the frame's visual appearance and protects the profile from scratches and corrosion. The TKK ball rail tables are also available with steel base plates, which offer even better rigidity and accuracy than the aluminum base plates. In PSK precision modules, the frame simultaneously serves as a U-shaped guide rail and is therefore always made from steel.



Frame with guide rails

The actual guidance element in the linear motion system is the guide rail. This is fixed to the frame. The guide rail is either bolted down on the frame, staked into it as a dovetail profile, or integrated into the frame. In the case of cam roller guides, the guide shaft is pressed into the frame. The various guideway types are described in more detail in section 6.1.1.3.



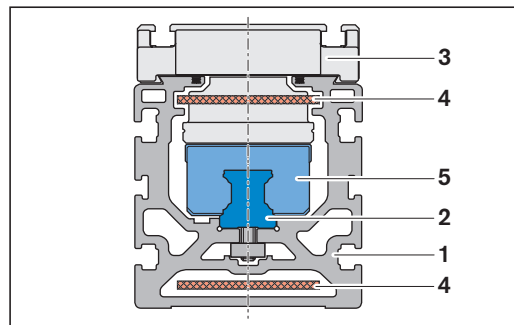
Carriage assembly

Carriage assembly

The carriage assembly generally consists of a compact aluminum profile with integrated or screw-fastened runner blocks. Customer-built attachments are usually mounted on the carriage. The carriage is fastened to the drive unit of the linear motion system. When the motor applies a drive torque, the drive unit sets the carriage in motion. Typical drive units are ball screw drives or toothed belt drives.

Lubrication via the carriage

All linear motion systems are designed for one-point lubrication with grease. (Cam roller guides are lubricated with oil.) The guideways are lubricated via the carriage, either from the side (e.g., by hand) or from above via a customer-built lube system.



Cross-section of a linear module with toothed belt drive

- 1 Frame
- 2 Guide rail
- 3 Carriage
- 4 Toothed belt
- 5 Runner block

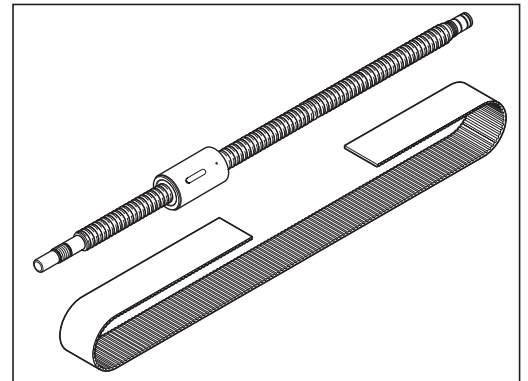
6 Linear motion systems

6.1 Principles

6.1.1 System technology

Drive unit of a linear motion system

The drive unit of a linear motion system contains force-generating and force-transmitting elements with the associated bearings. The various drive unit versions are described in more detail in section 6.1.1.4.

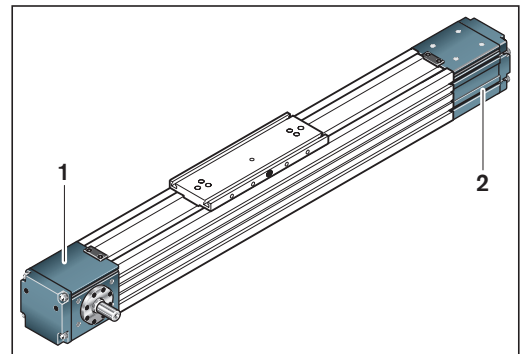


Ball screw drive and toothed belt drive versions

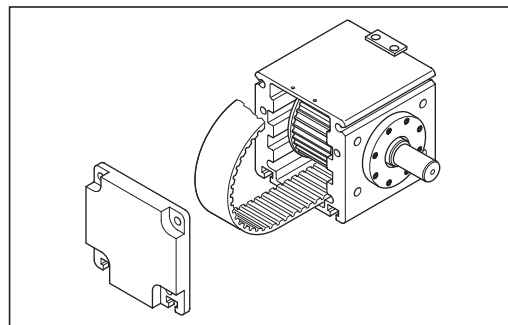
Toothed belt drive

Belt-driven linear modules have a drive end enclosure (1) and a tension end enclosure (2). The main component in the drive end enclosure is a pulley mounted on rolling bearings. The toothed pulley transfers the motor's drive torque to the toothed belt.

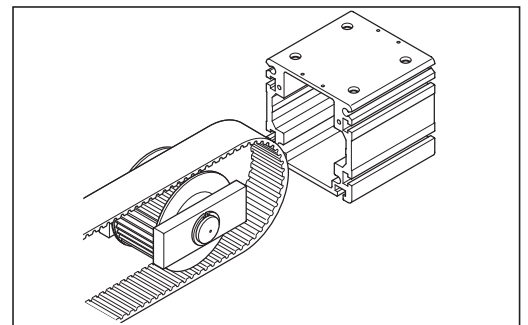
In the tension end enclosure, the toothed belt is wrapped around a second bearing-mounted pulley for the return motion. The belt can also be tensioned here by adjusting the position of the pulley.



End enclosures



Drive end enclosure (1)



Tension end enclosure (2)

6 Linear motion systems

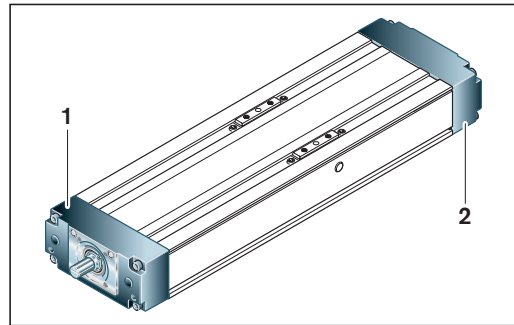
6.1 Principles

6.1.1 System technology

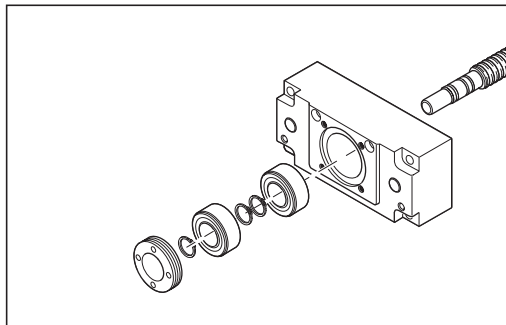
Ball screw drive

In linear motion systems with ball screw drive, the end enclosures are called end blocks. They accommodate the ball screw drive's end bearings. The screw shaft journal protrudes from one of the two end blocks to allow connection of the screw to the motor.

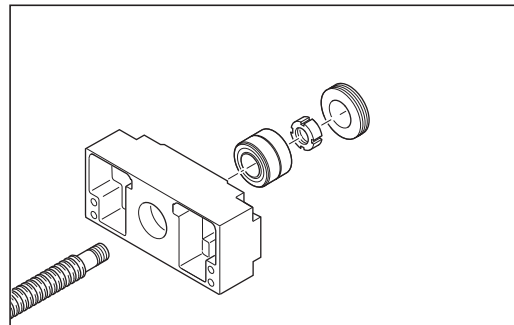
- 1 Drive end block
- 2 Idler end block



End blocks



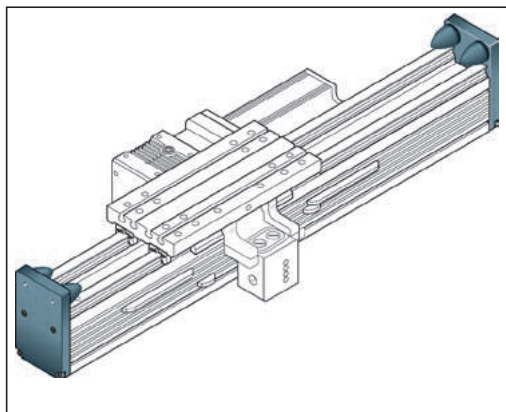
End block assembly with fixed bearing (1)



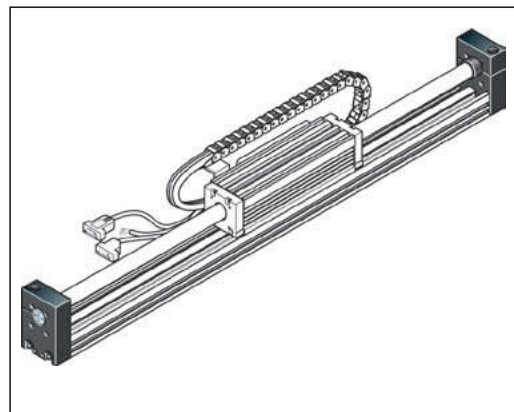
End block assembly with floating bearing (2)

**Linear motor
Rack and pinion drive**

In linear motion systems with a linear motor or rack and pinion drive, the end blocks serve both as end covers for the frame and as stops to prevent the carriage from overshooting the end of the frame.



End blocks with buffers on a linear module with rack and pinion drive



End blocks with buffers on a linear module with linear motor

6 Linear motion systems

6.1 Principles

6.1.1 System technology

Motors

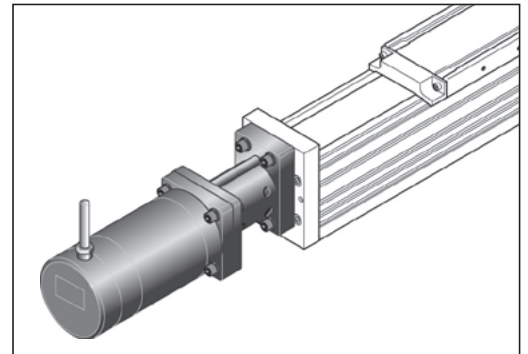
Linear motion systems with ball screw drive, toothed belt drive or rack and pinion drive are driven by motors. Rexroth offers a broad range of AC servo motors, three-phase motors and stepping motors. Depending on the application and the chosen combination of linear motion system and motor, the systems are driven either directly via a motor mount and coupling or indirectly via a gear unit. Timing belt side drives or planetary gears are used as gear units. A special form is a planetary gear unit that is integrated into the pulley in the drive end enclosure.

Gear units

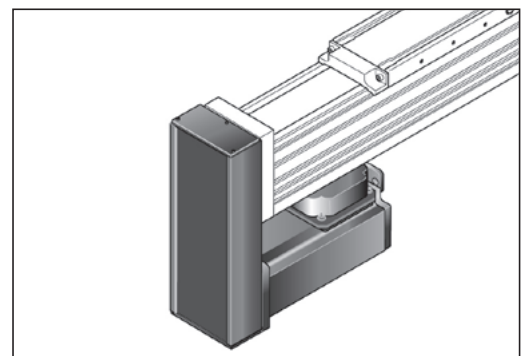
A locating feature and fastening thread are provided to facilitate the attachment of the motor or gear unit. A coupling transfers the drive torque stress-free to the linear motion system's drive shaft. Linear modules with rack and pinion drive are connected to the motor via a worm gear.

By using selectable gear ratios, the customer can adjust the drive torque to the specific application requirements and achieve the best match between the external load and the motor's moment of inertia. This is particularly important for optimizing the drive control loop and for obtaining highly dynamic drives.

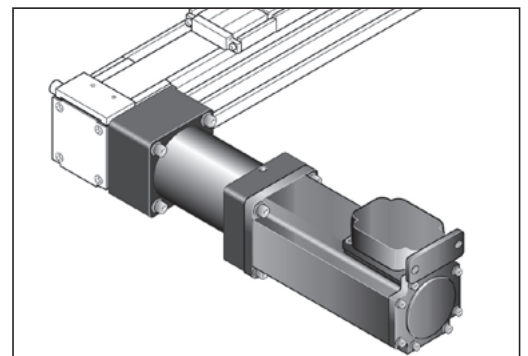
If a timing belt side drive is used, the overall length of the linear motion system can also be reduced compared to a configuration with direct motor attachment.



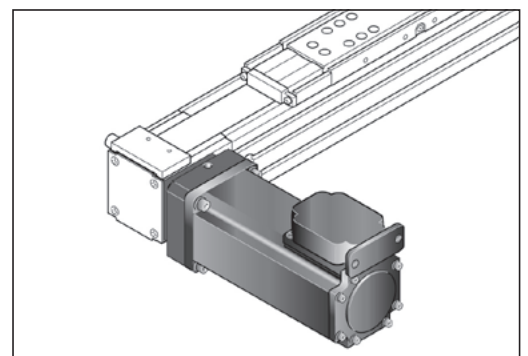
Motor mount with coupling and stepping motor



Gear unit as a timing belt side drive with servo motor



Gear unit as a planetary gear with servo motor



Integrated planetary gear and servo motor

6 Linear motion systems

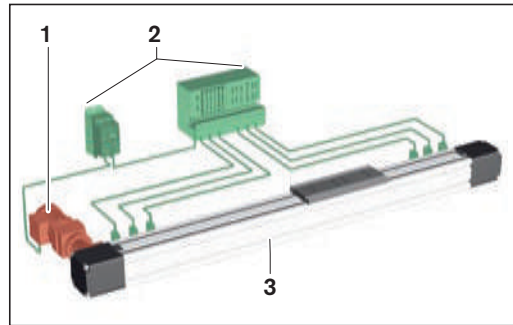
6.1 Principles

6.1.1 System technology

Controllers and control units

Controllers and control units are available for all motor options. The complete unit, i.e. the linear motion system, motor, controller and control unit, can therefore be sourced directly from Rexroth.

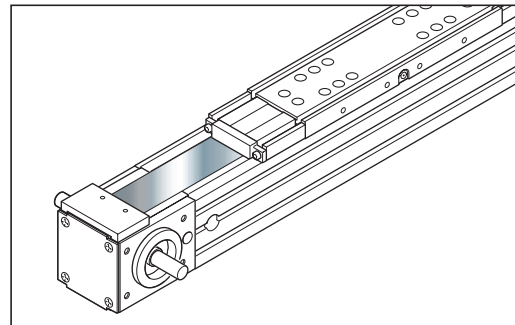
- 1 Motor
- 2 Controller and control unit
- 3 Linear motion system



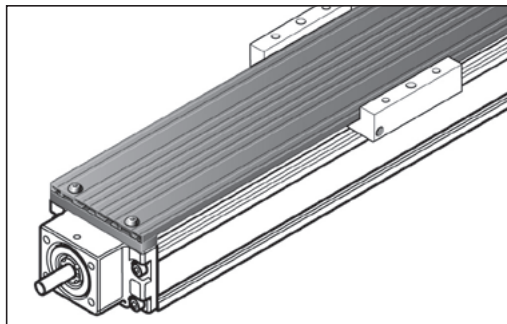
Linear motion system complete with control unit, controller and motor

Cover

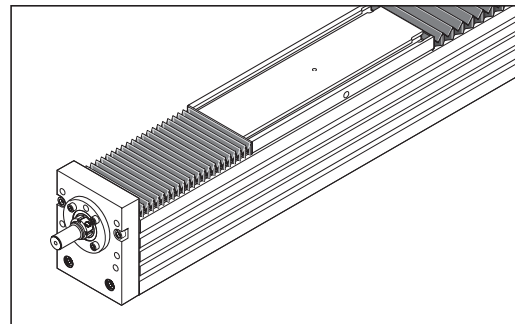
Some linear motion systems come standard with a cover to protect them from contamination. A cover can also be installed as an option in other linear motion systems. The cover may be designed as a sealing strip, cover plate or bellows, as appropriate for the type of system.



Sealing strip



Sheet metal cover plate



Bellows

Measuring systems

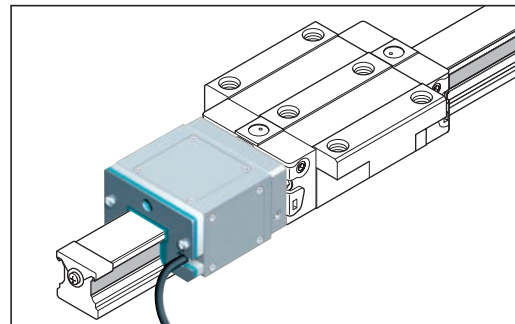
Linear motion systems can be fitted with position measuring systems. The choice of measurement principle will depend on the type of linear motion system used. Available options are:

- Optical systems
- Magnetic systems
- Inductive systems

Measuring systems can also be supplied as:

- Rotary systems (rotary encoders)
- Linear systems (e.g. integrated measuring system from Rexroth, glass scale)

All measuring systems can either be integrated or mounted externally, depending on the system design.



Integrated inductive measuring system on the ball rail system

6 Linear motion systems

6.1 Principles

6.1.1 System technology

Switching systems

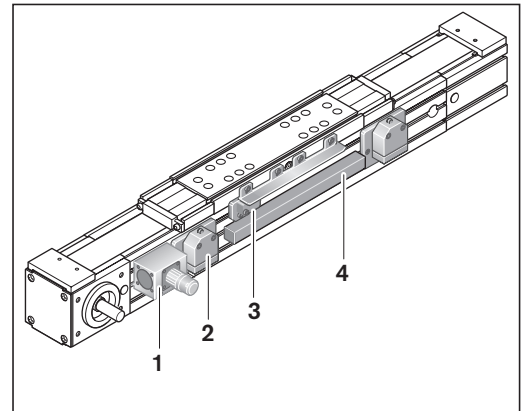
There are various switching systems available for linear motion systems. These can be used as limit switches or reference switches. Normally, the switches used on linear modules are either mechanical (2) or inductive (3). Compact modules are equipped with magnetic field sensors (Hall or Reed sensors).

Socket and plug

The switch wiring can be grouped and routed through a socket and plug. As a result, only one cable is needed for connection to the controller.

Cable duct

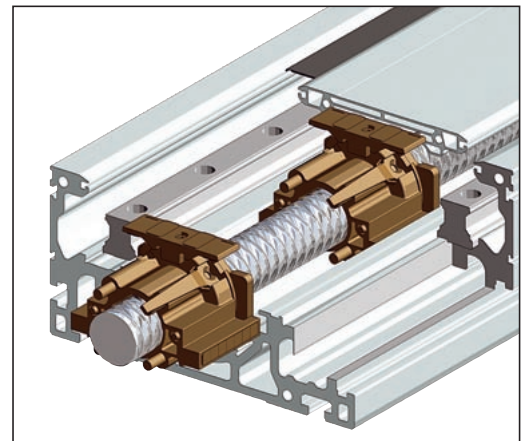
A side-mounted cable duct (4) serves to protect the switch cables (see section 6.8.4).



Switching system

Screw support

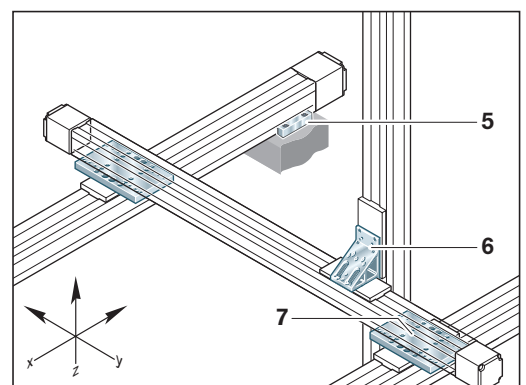
Linear motion systems with ball screw drive can be equipped with screw supports as an option. Screw supports make it possible to increase the stroke length or to achieve a significant increase in the maximum permissible speed while maintaining the same stroke length. The maximum permissible rotary speed is determined by the screw's critical speed.



Compact module with screw supports

Connection elements

For compact modules, connection plates (7) with the same T-slot design as the Rexroth construction profiles are available for connecting additional modules or for mounting of customer-built attachments. This enables the attachment of components to be standardized. For linear and compact modules, there are also connection brackets (6) for building X-Y-Z combinations. Clamping fixtures (5) can be used to fasten the linear motion systems to the mounting base.



Connection elements

- 1 Plug
- 2 Mechanical switch
- 3 Proximity switch
- 4 Cable duct

- 5 Clamping fixture
- 6 Connection bracket
- 7 Connection plate

6 Linear motion systems

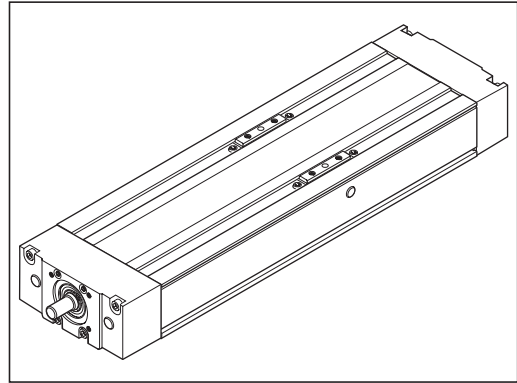
6.1 Principles

6.1.1 System technology

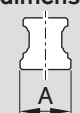

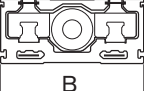
6.1.1.2 Type and size designations

For easy differentiation of the many versions of linear motion systems, Rexroth uses a simple identification system comprising a type and a size designation. The type designation consists of three letters, which define the type of system, guideway and drive unit used. This is followed by the size designation, which consists of the size of the linear guideway and the width of the frame.

The table below illustrates the coding system used for the type and size designations of Rexroth linear motion systems, using a compact module as an example. (The code letters are based on the German product names.)



Example: Compact module CKK 20-145

Designation	Type			Size	
	C	K	K	20 -	145
Example: Compact Module					
System					
	Linear Module, closed type (M)				
	Linear Module, open type (L)				
	Compact Module (C)	C			
	Precision Module (P)				
	Ball Rail Table (T)				
	Linear Motion Slide (S)				
Guideway					
	Ball rail system (K)		K		
	Integrated ball rail system (S)				
	Cam roller guide (L)				
	Linear bushing and shaft, closed type (G)				
	Linear bushing and shaft, open type (O)				
Drive unit					
	Ball screw drive (K)				K
	Toothed belt drive (R)				
	Linear motor (L)				
	Pneumatic drive (P)				
	Rack and pinion drive (Z)				
	Without drive (O)				
Guideway dimension					
	Rail width for ball rail systems (Example: A = 20 mm)			20 -	
	Shaft diameter for cam roller guides Shaft diameter for linear bushings and shafts				
Frame dimension					
	Width of the frame or the base plate (Example: B = 145 mm)				145

Identification system for Rexroth linear motion systems

6 Linear motion systems

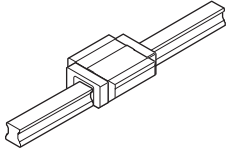
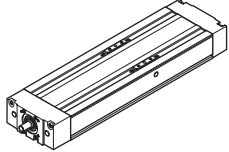
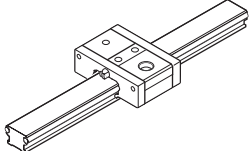
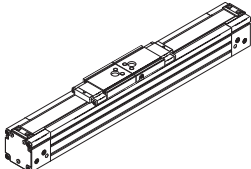
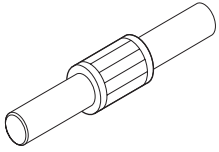
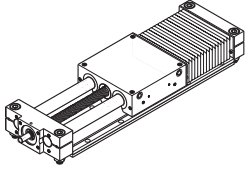
6.1 Principles

6.1.1.3 Guideway types

Rexroth uses three different kinds of guideways in its linear motion systems. Each of these offers different advantages, allowing the most appropriate guideway to be selected for the specific application.

Detailed descriptions of the individual guideway types can be found in the related sections of the handbook (Chapter 3, section 3.2: Ball Rail Systems; Chapter 3, section 3.6: Cam Roller Guides; Chapter 4: Linear Bushings and Shafts).

Linear guideway types

Guideway	Example	Characteristics
	Compact module CKK 	<ul style="list-style-type: none"> ■ High rigidity ■ High precision ■ Comes standard with 2% C preload ■ Travel speeds up to 5 m/s possible
	Linear module MLR 	<ul style="list-style-type: none"> ■ Low noise level ■ High travel speeds up to 10 m/s possible
	Linear motion slide SOK 	<ul style="list-style-type: none"> ■ Smooth running ■ Insensitive to dirt ■ Robust (particularly the closed type)

The guideways are always mounted to the frame. They are connected by screw fasteners and/or staking of the rail or shaft into the frame (ball rail system or cam roller guide). In the case of linear bushings and shafts, the shafts are fastened to shaft support rails or to shaft support blocks at the shaft ends (see Chapter 4).

The application requirements for rigidity and precision are important criteria for selecting the correct linear guideway. The rigidity of the overall system depends on the type and number of guideways installed.

6 Linear motion systems

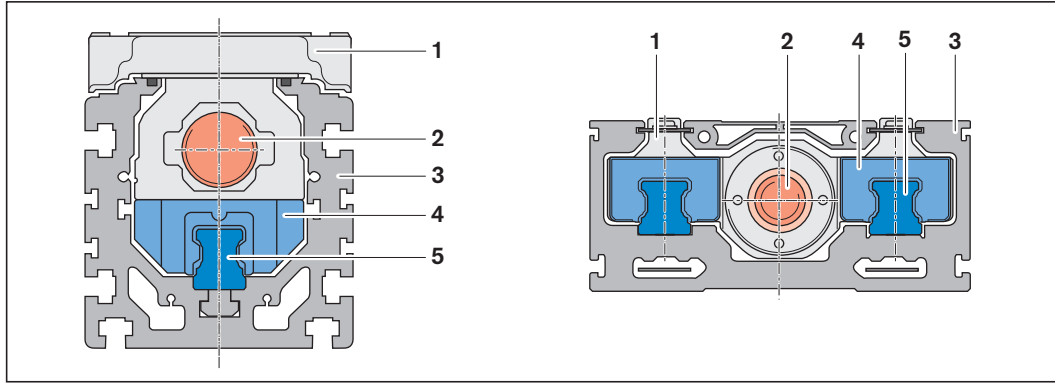
6.1 Principles

6.1.1 System technology

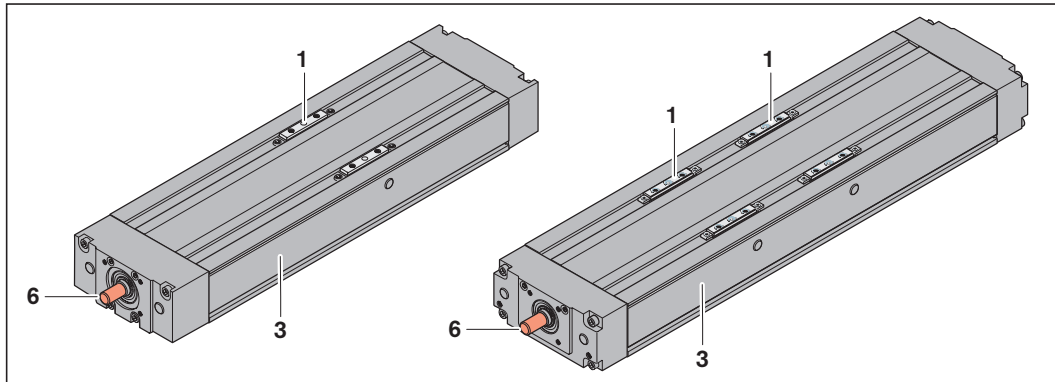
Number of guideways

In addition to the choice of guideway type, the number of guideways installed is also an important factor determining the linear motion system's

overall rigidity. The number of guide rails or shafts as well as the number of carriages may vary.



Linear module MKK with one rail guide (left) and compact module CKK with two rail guides



Compact module CKK with one carriage per rail (left) and with two carriages per rail

- 1 Carriage
- 2 Ball screw drive
- 3 Frame
- 4 Runner block
- 5 Guide rail
- 6 Screw journal for ball screw drive

Load capacities and moments

In linear modules and compact modules with the same rail size the load capacities and moments will differ according to:

- Configuration with one or two rails
- One or more carriages

The table below gives a comparison of the load capacities and moments for two different linear motion systems:

- Linear module MKK with one carriage running on a guide rail with two runner blocks
 - Compact module CKK with two carriages running on two guide rails with four runner blocks
- The rail width is the same in both cases.

Module	Number of guide rails	Number of runner blocks per rail	Dynamic load capacity C of the guideway	Dynamic moments	
				Torsional moment M_t	Longitudinal moment M_L
MKK 15-65	1	2	12 670 N	120 Nm	449 Nm
CKK 15-110	2	2	25 340 N	835 Nm	1 075 Nm

6 Linear motion systems


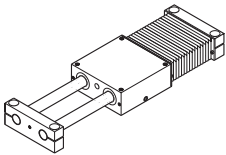

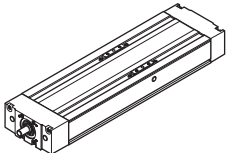

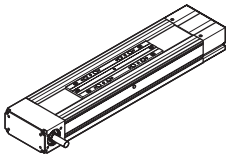
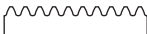
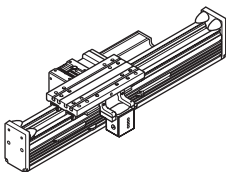
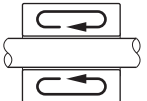
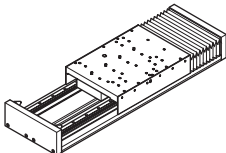

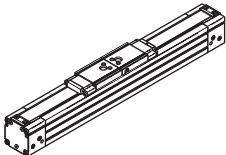
6.1 Principles

6.1.1 System technology

6.1.1.4 Drive unit types

The characteristics of linear motion systems are essentially determined by the type of drive unit used.

Rexroth offers a variety of drive unit options to cover many different applications.

Drive unit	Example	Characteristics
	Linear motion slide SGO 	<ul style="list-style-type: none"> ■ Manual movement ■ Robust linear motion system
	Compact module CKK 	<ul style="list-style-type: none"> ■ High rigidity in the direction of travel ■ High thrust force ■ Repeatability ± 0.005 mm (zero backlash) ■ Travel speeds up to 1.6 m/s
	Compact module CKR 	<ul style="list-style-type: none"> ■ High travel speeds up to 5 m/s (MLR module: up to 10 m/s) ■ Low rigidity in the direction of travel ■ Repeatability ± 0.1 mm
	Linear module MKZ 	<ul style="list-style-type: none"> ■ Allows long guideways lengths ■ Travel speeds up to 5 m/s ■ Allows applications with multiple, independent carriages ■ Low noise
	Ball rail table TKL 	<ul style="list-style-type: none"> ■ High travel speeds up to 8 m/s and high acceleration rates ■ Short cycle times ■ High positioning accuracy and repeatability ■ Allows applications with multiple, independent carriages ■ Virtually no down-time due to low number of wear parts ■ Maintenance-free linear motor ■ Low noise
	Linear module MKP 	<ul style="list-style-type: none"> ■ No motor required ■ Travel to fixed end positions (no intermediate positions) ■ Travel speeds up to 2 m/s

6 Linear motion systems

6.1 Principles

6.1.1 System technology

Drive unit types and system characteristics

The type of drive unit chosen determines the dynamics (speed and acceleration) of the system and its positioning accuracy and repeatability. The duty cycle is influenced by the kinematic requirements.

Due to their different designs, the drive units each have their own specific system characteristics. Linear motion systems therefore cover a broad range of customer applications. The structural design of the individual drive units is described in the following sections.

Ball screw

In linear motion systems, ball screw drives are primarily used with zero-backlash cylindrical nuts. They are therefore particularly suitable for feed tasks requiring high precision. Ball screw drives are discussed in detail in Chapter 5.

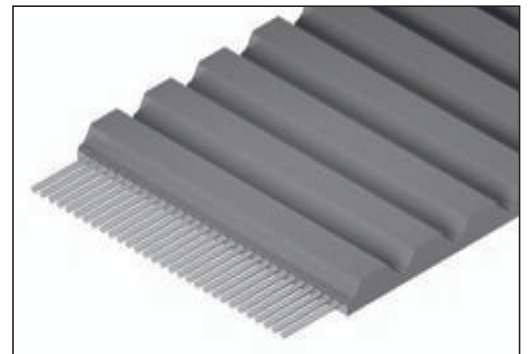


Screw and nut of a ball screw drive

Toothed belt

Toothed belt drives are particularly suitable for highly dynamic applications. Toothed belts for linear motion systems consist of abrasion-resistant polyurethane (PU) reinforced with high-strength steel cords.

The toothed belt is fastened to each end of the carriage by means of clamps. The drive end enclosure contains a pulley, through which the motor's drive torque is applied and transferred to the belt. At the tension end, the toothed belt is wrapped around another pulley and tensioned to ensure smooth operation.



Toothed belt

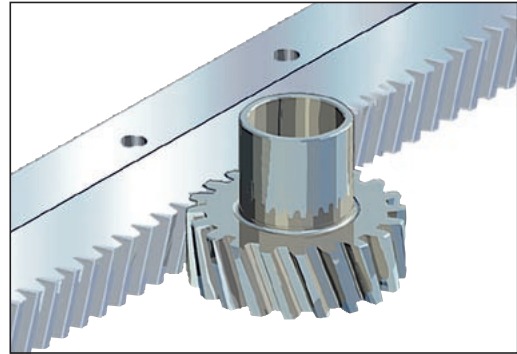
6 Linear motion systems

6.1 Principles

6.1.1 System technology

Rack and pinion

In the case of rack and pinion drives, a helical-cut gear rack is mounted to the side of the frame. The pinion, worm gear and motor are mounted on the carriage. This makes it possible to move several carriages independently of each other and to build systems with long travel distances and high-speed motion. The low-backlash worm gear makes the rack and pinion drive highly rigid. The helical-cut teeth reduce the noise generated by the drive.



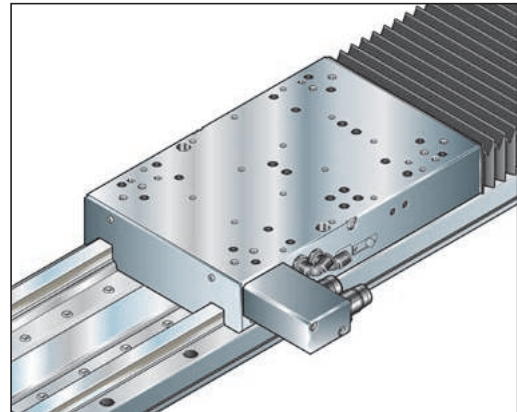
Rack and pinion drive

Linear motor

A linear motor is an electric motor that produces linear motion instead of rotary motion. The thrust force is generated directly at the moved part (carriage). No additional drive element is required to transform and transmit the thrust. Linear motors are therefore also referred to as direct linear drives.

The key components of a linear motor are the carriage (primary element) and the secondary element with permanent magnets. Rexroth offers the secondary element in three versions: In linear modules LKL and MKL, a round thrust rod is connected to the frame via the end blocks. In the TKL ball rail table, the flat permanent magnet and, in the case of the CKL compact module, the U-shaped permanent magnet is bolted directly to the frame. The basic principle is the same for all versions and is described in section 6.8.2.2.

Because the thrust is generated directly, no additional mechanical parts are required to convert rotary motion to linear motion. This means that there is no backlash and no mechanical compliance in the drive train, which results in unparalleled positioning accuracy and repeatability throughout the life of the drive. Since linear motors have no internal moving parts such as those contained in rotary drive systems, they are not subject to wear and require no maintenance.



TKL ball rail table with linear motor

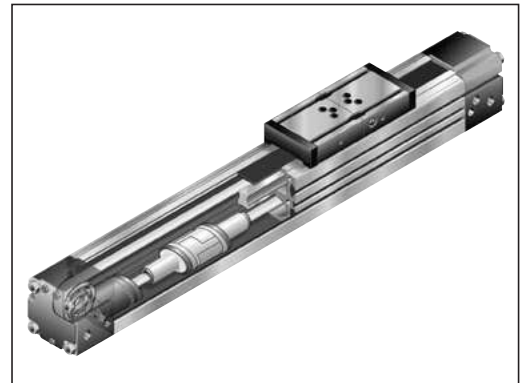
6 Linear motion systems

6.1 Principles

6.1.1 System technology

Pneumatic drive






Pneumatic drives operate by means of a piston guided in a closed cylinder. Compressed air can be applied to one or both ends of the linear module and is supplied to the cylinder via internal air ducts. The compressed air moves the piston. A belt fastened to the piston and guided around pulleys in the end enclosures transmits the piston movement, i.e. the driving force, to the carriage. In pneumatic drives, the carriage can only travel full strokes, from end position to end position, or up to a shock absorber as a mechanical stop. Unlike all the other drive types, travel to intermediate positions is not possible.



Linear module MKP

Overview

The following table summarizes the characteristics of the various drives:

Drive unit	Requirements				
	Thrust	Rigidity	Speed	Precision ¹⁾	Noise characteristics
Ball screw 	+++	+++	+	+++	++
Toothed belt 	++	+	+++	+	++
Rack and pinion 	+++	+++	++	++	++
Linear motor 	++	+++	+++	+++	+++
Pneumatic 	+	0	+	0	++

1) Depending on the measuring system used, its accuracy, and the control system

+++ Very good
 ++ Good
 + Satisfactory
 0 Adequate

6 Linear motion systems

6.1 Principles

6.1.2 Product selection

6.1.2.1 Application parameters

A suitable linear motion system can be pre-selected by analyzing and defining the following application parameters:

- Stroke length
- Installation space
- Load
- Cycle times and linear speed
- Repeatability and positioning accuracy
- Rigidity (including the mounting base)
- Environmental conditions

Often, several linear motion systems may be suitable for a particular application. In this case, or if special conditions make it difficult to determine which linear motion system should be used,

customers can always call on Rexroth's many years of experience. After pre-selecting the linear motion system, the next step is to perform the design calculations.

Stroke length

The overall length of a linear motion system is calculated from the stroke length. The maximum achievable stroke length of a linear motion system will depend on the type of drive used. With ball screw drives, the problem of critical speed becomes a factor beyond a certain length. For longer lengths, toothed belt drives are more suitable.

Some linear motion systems with ball screw drives are available with screw supports. These shorten the free screw length and support the cover plate, which is also often present. This can significantly increase the permissible rotary speed and the stroke length.

Installation space

When selecting a linear motion system, care must be taken to ensure sufficient installation space (height, width and length). Depending on the configuration, the attachment of a motor can significantly increase the length or width of the linear motion system.

Load

The size and type of the linear motion system is restricted by the load it will be required to carry. Two-rail compact modules have higher load capacities than one-rail linear modules with the same size of linear guide. This is also true for linear motion systems with several carriages. In general, loads of up to approximately 20% of the dynamic load and moment capacities (C , M_t , M_L) have proven to be acceptable. The following limits should not be exceeded:

- Permissible drive torque
- Maximum permissible speed
- Maximum permissible forces and moments
- Maximum permissible deflection

The permissible drive torque, the maximum permissible speed, and the permissible forces and moments are determined by the linear guides and drive units used. The product catalogs provide details of the dynamic characteristic values and the maximum permissible forces and moments.

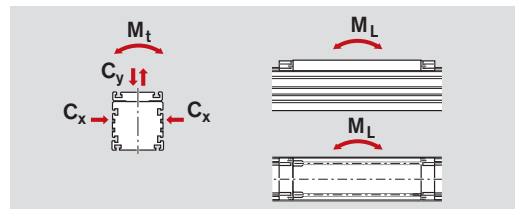


Illustration of dynamic load and moment specifications

6 Linear motion systems

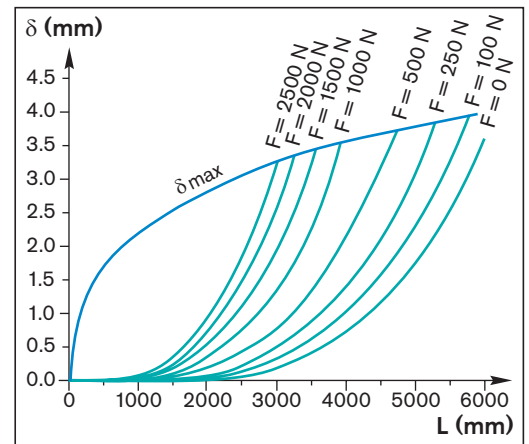
6.1 Principles

6.1.2 Product selection

Deflection

Linear motion systems are frequently not fully supported but installed as unsupported structures, i.e. they are fastened to the adjoining customer-built structure at points close to the ends of the axes. In the unsupported configuration, deflection of the linear motion system must be taken into account. If the maximum permissible deflection δ_{\max} is exceeded, additional support must be provided for the linear motion system. The maximum permissible deflection δ_{\max} depends on the length of the linear axis and the load F . The corresponding charts in the product catalogs help in determining the maximum permissible deflection.

Regardless of the permissible deflection of the linear motion system, the required system accuracy must also be taken into account. If necessary, the system must be supported at several points. The same applies when high demands are made on the system dynamics. Some versions, such as ball rail tables or precision modules, generally require to be mounted fully supported on a rigid customer-built structure to achieve the required precision.



Sample chart showing the deflection for a linear module MKK 20-80

- L = module length (mm)
- δ = actual deflection (mm)
- δ_{\max} = maximum permissible deflection (mm)
- Maximum permissible deflection depending on the length of the linear motion system
- Deflection under the given force

Cycle times and speeds

The required cycle times, and therefore the required speeds, essentially determine the type of linear motion system to be used. Both the linear guide and the drive unit influence the maximum permissible speed.

Linear motion systems with cam roller guide and toothed belt drive or linear motor are particularly suitable for very high speeds.

Repeatability and positioning accuracy

Where good repeatability and positioning accuracy are required, ball rail tables or precision modules are an especially good choice. In this case, ball screws or linear motors are used as

the drive. These provide clear advantages over toothed belt drives because of their high axial rigidity.

Rigidity

The rigidity perpendicular to the direction of travel is known as the radial rigidity. A linear motion system's radial rigidity is essentially determined by the guide unit, the geometry of the frame and the mounting base. Ball rail systems are considerably more rigid than cam roller guides or linear bushing and shaft systems.

An appropriately designed mounting base can support the linear motion system in such a way that the application's intrinsic rigidity will be significantly increased.

Environmental conditions

Some linear motion systems are more suitable than others for particular environmental conditions because of their design. In dirty or dusty environments, linear motion systems with cam roller guides or with additional covers are advantageous.

The following cover options are available:

- Without cover
- Steel sealing strip
- Polyurethane sealing strip
- Bellows
- Gap-type sealing

6 Linear motion systems

6.1 Principles

6.1.2 Product selection

6.1.2.2 Product selection aids

Rexroth offers users a broad range of linear motion systems in various designs and sizes. The selection criteria stated in section 6.1.2.1 enable

a suitable linear motion system to be found for almost any conceivable application.

Combinability of guides and drives

Since all of the possible guides and drives can be combined, a linear motion system can be configured to match the specific requirements of any application. Sections 6.1.1.3 and 6.1.1.4 contain tables summarizing the characteristics of the different guide and drive options.

However, because of the special demands on linear motion systems, not all combinations are sensible or cost-efficient. The following summary shows all the combinations offered by Rexroth:

System	Guide unit			Drive unit					
	Ball rail system	Cam roller guide	Linear bushing and shaft	Without drive	Ball screw	Toothed belt	Rack and pinion	Pneumatic	Linear motor
Linear module	✓	✓	-	✓	✓	✓	✓	✓	✓
Compact module	✓	-	-	✓	✓	✓	-	-	✓
Precision module	✓	-	-	✓	✓	-	-	-	-
Ball rail table	✓	-	-	✓	✓	-	-	-	✓
Linear motion slide	-	-	✓	✓	✓	-	-	-	-

Generally, several different linear motion systems may represent a technically optimal solution for the linear motion requirements, so that economic and visual design aspects can also be consid-

ered. The following summary shows the various linear motion systems with the most important and logical selection criteria.

Linear motion system		Load capacity	Rigidity	Precision	Cover	Maximum speed	Maximum length
Linear module	MKK	++	++	++	+++	1.6 m/s	6000 mm
	MKR	++	++	+	+++	5.0 m/s	12000 mm
	MLR	+	+	+	+++	10.0 m/s	10000 mm
	MKZ	+++	+++	++	+	2.2 m/s	6000 mm
	MKP	+	+	0	+++	2.0 m/s	5600 mm
	MKL	+	++	++	+++	5.0 m/s	2000 mm
	LKL	+	+	++	++	5.0 m/s	2000 mm
Compact module	CKK	+++	+++	++	++	1.6 m/s	5500 mm
	CKR	+++	+++	+	++	5.0 m/s	10000 mm
	CKL	+	+	++	+	5.0 m/s	2800 mm
Precision module	PSK	++	++	+++	+++	1.6 m/s	940 mm
Ball rail table	TKK	+++	++	+++	++	1.6 m/s	2860 mm
	TKL	+++	++	+++	++	8.0 m/s	4000 mm
Linear motion slide	SGO	++	+	0	++	Without drive	5300 mm
	SOO	++	++	0	++	Without drive	5300 mm
	SGK	++	+	+	++	1.4 m/s	4000 mm
	SOK	++	++	+	++	1.4 m/s	4000 mm

+++ Very good

++ Good

+ Satisfactory

0 Adequate

6 Linear motion systems

6.1 Principles

6.1.2 Product selection

6.1.2.3 Motor, controller and control system

In addition to selecting the linear motion system itself, the necessary peripherals must also be considered. This involves specifying and calculating the lubrication system, including in-service lubrication intervals, and, in particular, specifying the motor, controller and control system to be used. For all linear motion systems, Rexroth offers a large number of motors with controllers and control systems adapted to particular environments and applications. For each type and each size of linear motion system, there is always a choice of several possible motors to cover different speed and drive torque requirements.

Depending on the application, the most appropriate solution may be a servo motor, a three-phase motor or a stepping motor. Linear motors are a special case, as the motor is already integrated into the linear motion system.

The associated controllers and control systems enable the control loop to be optimally parameterized to match the linear motion system to the specific customer application. More information on motor selection can be found in section 6.8 "Electrical components."

6.1.2.4 Conditions of use

When selecting a linear motion system, the conditions of use for the customer's application must also be taken into account. It is essential to check whether the linear motion system can be used in the given environment, whether there are any constraints, or whether another linear motion system must be selected.

The following environmental factors in particular must be checked:

- Dust, dirt, shavings, etc.
- Temperature
- Installation conditions and available space
- Shocks and vibrations
- Lubrication
- Special conditions of use

Dust, dirt, shavings, etc.

Dust, dirt and shavings, etc. are crucial factors affecting the life expectancy of a linear motion system. Depending on the type and the quantity of foreign particles that a linear motion system will be exposed to, appropriate covers must be provided to ensure trouble-free operation and prolong the life of the guide and drive components.

Temperature

Essentially, it is possible to operate linear motion systems within a temperature range of 0°C to 40°C. The actual permissible operating temperature can be limited by the guide system and the lubricant used.

6 Linear motion systems

6.1 Principles

6.1.2 Product selection

Installation conditions and available space

The installation conditions (e.g. required operating space, layout) often determine the type and size of linear motion system to be used. The specific application determines whether a linear motion system can be installed unsupported or must be supported. Some linear motion systems must be fully supported in order to ensure trouble-free operation and the required precision.

The space available can also affect the choice of linear motion system and motor attachment. The various drive configurations offered make it possible to adapt the systems flexibly to the given spatial conditions. Attaching the motor via a motor mount and coupling (direct drive) extends the overall length of the linear motion system. Using a timing belt side drive enables the motor to be installed alongside, above or below the linear motion system. The available installation space can be optimally exploited by selecting the appropriate motor attachment configuration.

Shocks and vibrations

Shocks and vibrations can have an adverse effect on a linear motion system's life expectancy. These effects can be minimized by selecting the appropriate linear motion system and sizing it accordingly.

Lubrication

The service life of a linear motion system also depends on adequate lubrication of the guideway and, where applicable, the ball screw assembly. The system can be lubricated manually via the lube nipples provided. Many linear motion systems also offer the possibility of one-point lubrication via the carriage. The advantage here is that a central lubrication system and lubricant dispenser

can be used to ensure an adequate supply of lubricant to the system while it is in operation.

All of the rotary anti-friction bearings built into the linear motion systems (e.g. screw end bearings in the end enclosures) are lubricated for life and will not require in-service lubrication under normal conditions of use.

Special conditions of use

Special conditions of use for applications in clean rooms or under vacuum, for example, with corresponding application-specific parameters, are accounted for by specially designed linear motion systems. Chemical effects and aggressive media (metalworking fluids, solvents, vapor, etc.)

also place particular demands on linear motion systems. The use of specially adapted lubricants and chemically resistant materials may be necessary here.

Normal conditions of use

Normal conditions of use are considered to be:

- Use at room temperature
- No exceptionally high levels of contamination
- No exceptional shock and vibration loads
- Adequate lubrication with an appropriate lubricant

6 Linear motion systems

6.1 Principles

6.1.3 Design notes

6.1.3.1 General design notes for linear motion systems

When designing machines, equipment and installations with linear motion systems, potential problems during later operation can be avoided by taking a number of basic precautions:

- | | |
|----------------------------------|---|
| Drive unit | <ul style="list-style-type: none"> ■ The maximum torque and speed of the motor must not exceed the limits for the linear motion system and the components used, such as the coupling. ■ The attached motor may project into the work zone of neighboring systems. The linear motion system and its surroundings should therefore be checked for possible interfering edges. |
| Lubrication | <ul style="list-style-type: none"> ■ Ensure compliance with the recommended lubrication intervals and the mounting, start-up and maintenance instructions. ■ For short-stroke applications, make sure to schedule lubrication strokes. More information on short-stroke applications is provided in Chapter 2, section 2.4.2.2. |
| Risk of buckling | <ul style="list-style-type: none"> ■ For vertically installed linear motion systems with a ball screw drive, the screw's fixed bearing must be at the top to avoid the risk of the screw buckling. |
| Dust protection | <ul style="list-style-type: none"> ■ Wherever possible, install the axis rotated through 180° (overhead mounting, carriage pointing downward). This will protect the linear motion system's mechanical parts from dust. |
| Special conditions of use | <ul style="list-style-type: none"> ■ Any special conditions of use and possible effects on materials should be discussed with Rexroth in advance (see section 6.1.2.4). |
| Mounting orientations | <ul style="list-style-type: none"> ■ Linear motion systems with a linear motor (MKL, LKL, TKL, CKL) should preferably be installed horizontally. ■ When installing linear motion systems in a vertical position, a braking device or counterweight must be provided to prevent the carriage from dropping if the motor has no brake (as is the case with a linear motor). The carriage must also be secured appropriately during transport. |

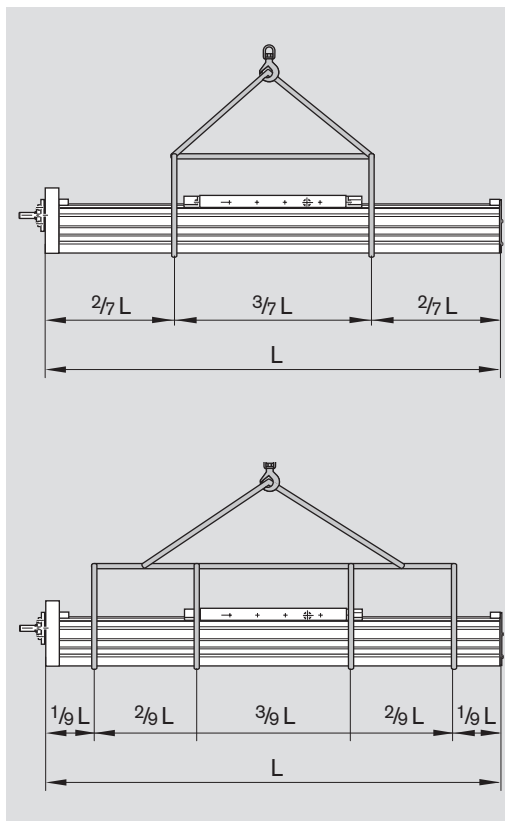
6 Linear motion systems

6.1 Principles

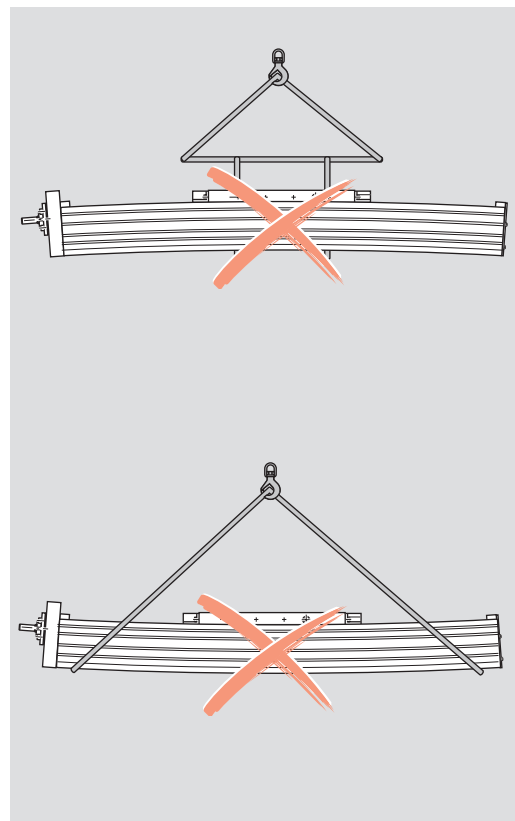
6.1.3 Design notes

Transport

- Especially with long, and therefore heavy, linear motion systems there is a risk of significant deflection when lifting them, which could result in permanent deformation. Appropriate care must therefore be taken when handling the systems. Always use suitable lifting equipment. The lifting equipment must not damage the linear motion system and must minimize deflection.



Recommendations for lifting linear motion systems



Strictly avoid any such lifting arrangements

6 Linear motion systems

6.1 Principles

6.1.3 Design notes

6.1.3.2 Fastening linear motion systems to the mounting base

When fastening linear motion systems to the mounting base, the following aspects must be taken into account because they have a significant effect on the system characteristics (e.g. service life, precision).

Accuracy

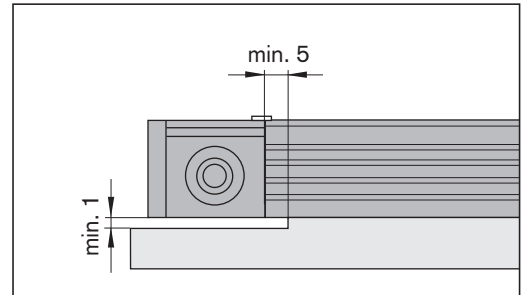
- In the worst case, accuracy errors in the linear motion system and the mounting base may have a cumulative effect. If the foundation or the mounting base does not have the required accuracy, even very precise linear motion systems may not achieve the required overall system precision.

Fastening options

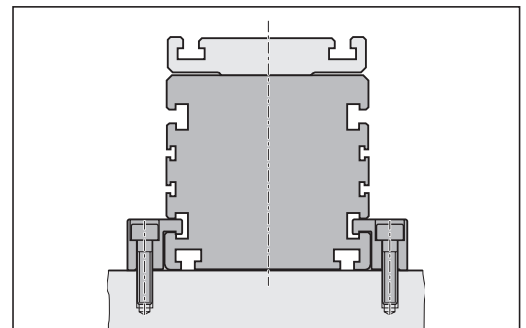
- Linear motion systems may not be supported at the end enclosures or the end blocks. The frame is the main load-bearing structure.
- Standardized clamping fixtures, sliding blocks, threaded anchor strips, connection plates and brackets allow easy fastening of the linear motion systems to the mounting base or, in the case of multi-axis motion systems, easy connection of the individual linear components to one another. Rexroth offers these mounting accessories in versions suitably adapted for each linear motion system, thereby facilitating the design of these interfaces.
- The recommended number of clamping fixtures per linear motion system should always be used.

Reference edge

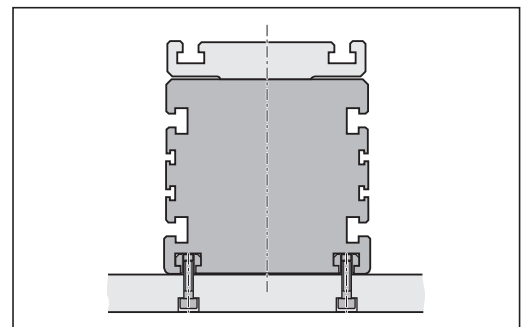
- In CKL compact modules, PSK precision modules and TKK ball rail tables, a reference edge is provided on the side of the frame profile to facilitate alignment.



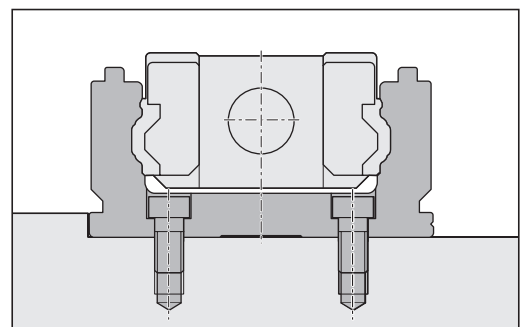
Minimum clearances between the end enclosure and the mounting base



Fastening with clamping fixtures



Fastening with sliding blocks



PSK precision module with reference edge

6 Linear motion systems

6.1 Principles

6.1.4 Calculations

The basic calculations for customized configuration of linear motion systems are:

- Calculation of the external loads acting on the linear motion system and the resulting nominal life
- Motor design calculations, including cycle times
- Deflection (optional)

Additional calculations may be required, depending on the application.

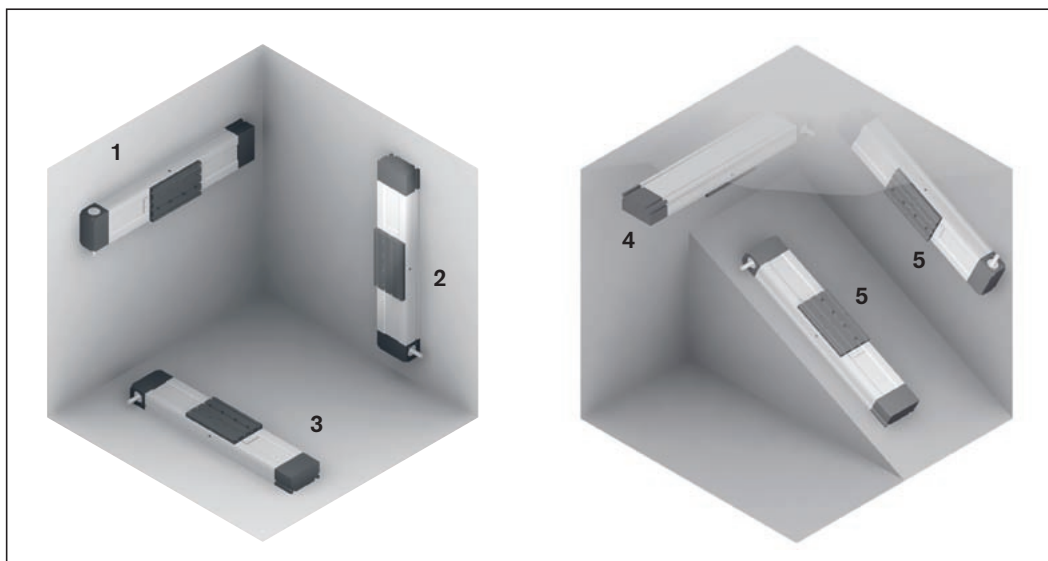
6.1.4.1 External loads and nominal life calculation

Generally, the nominal life can be calculated according to the methods explained in Chapter 3 "Profiled rail systems." The permissible load capacities given in the product catalogs relate to the carriage. The calculations are therefore performed as they would be for an individual runner block. In general, external loads of up to approximately 20% of the characteristic dynamic values (C , M_t and M_L) have proven to be acceptable. The mounting orientation must also be taken into account when determining the external loads:

- 1 Wall mounting
- 2 Vertical mounting
- 3 Horizontal mounting
- 4 Overhead mounting
- 5 Inclined mounting

The coordinate systems for linear motion systems and the positive and negative directions of travel must be appropriately determined for the application.

In linear motion systems with ball screw drive, the nominal life of the guide, the ball screw drive and, where applicable, the fixed bearing must be calculated.



Mounting orientations

6 Linear motion systems

6.1 Principles

6.1.4 Calculations

6.1.4.2 Motor design calculations, including cycle times

The motor always plays an important role when designing customer applications. Linear motion systems with ball rail guides can carry high loads. However, these must also be moved. Design calculations must therefore be performed for the motor as well as for the linear motor system. The product catalogs give indications for performing preliminary design calculations. For precise calculations, Rexroth offers a design calculation service.

Motor types

The design calculations for synchronous, three-phase and stepping motors depend very much on the type of motor used. Different aspects must be taken into consideration for each motor type. As an example, the following pages describe how to perform a rough calculation for a linear motion system with a ball screw drive.

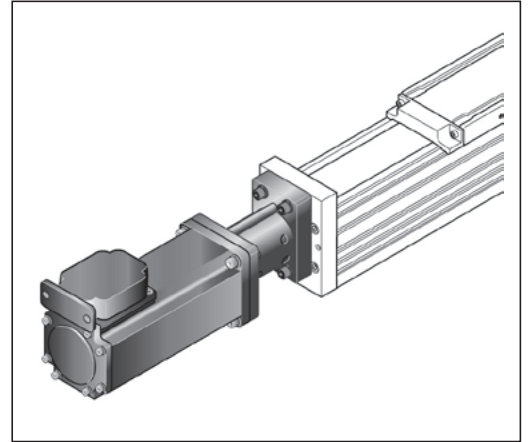
Drive torque Mass moment of inertia

The procedure for checking the drive torque and the mass moment of inertia is, however, the same for all motor types. The values for the linear motion system are taken from the product catalog or calculated and then used to select the motor.

Drive types

The design calculations for linear motion systems with toothed belt drive are similar to those for systems with a ball screw drive. The feed constant and the gear transmission ratio are included in the calculation. Preliminary design details can be found in the product catalogs.

When selecting and dimensioning the drive unit, a distinction has to be made between systems with rack and pinion drive, pneumatic drive or a linear motor and systems with ball screw drive or toothed belt drive. The differences are described in the product catalogs.



MKK linear module with a servo motor drive

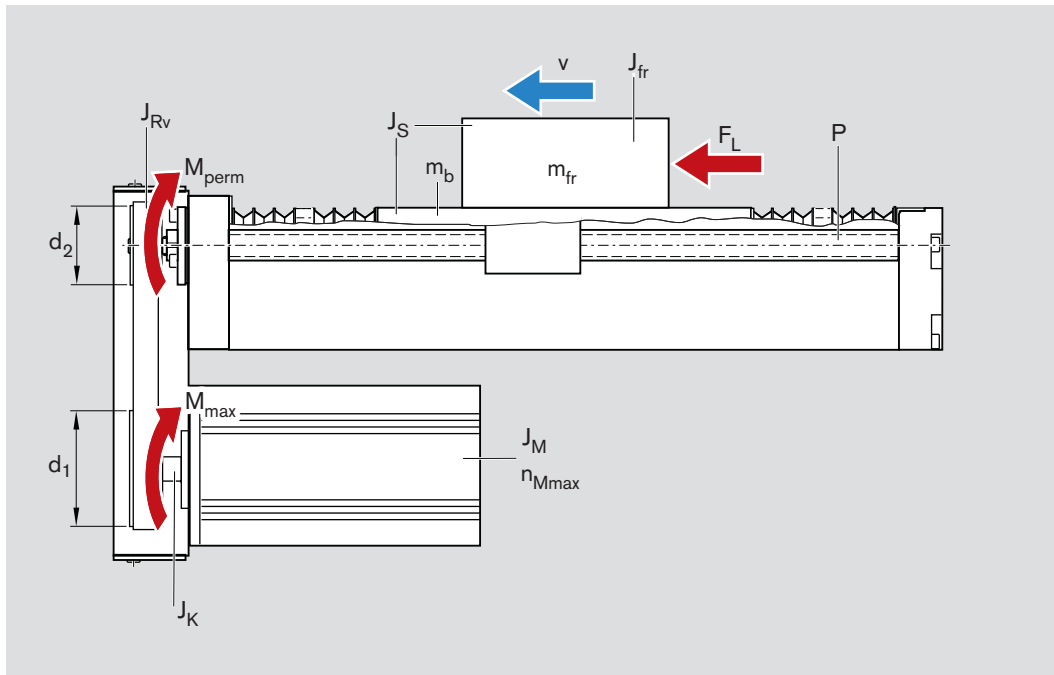
6 Linear motion systems

6.1 Principles

6.1.4 Calculations

Systems with ball screw drive

The following calculation principles can be used for rough selection and sizing of linear motion systems with ball screw drive. A precise calculation for the complete drive (motor and controller), in particular the thermal aspects, is only possible when the motion sequence, including pause times, the feed forces and the environmental conditions are known.

Calculation parameters

a	= acceleration	(m/s ²)	M_B	= maximum acceleration torque of motor	(Nm)
d_1	= diameter of driving sprocket (on motor shaft)	(mm)	M_G	= weight moment	(Nm)
d_2	= diameter of driven sprocket (on screw shaft)	(mm)	M_L	= load moment	(Nm)
F_L	= thrust	(N)	M_{max}	= maximum motor torque	(Nm)
i	= transmission ratio	(-)	M_R	= friction torque	(Nm)
J_{Br}	= mass moment of inertia of motor brake	(kgm ²)	M_{perm}	= permissible system drive torque	(Nm)
J_{fr}	= mass moment of inertia of external load	(kgm ²)	m_b	= moved mass (carriage)	(kg)
J_{tot}	= total reduced mass moment of inertia on motor journal	(kgm ²)	m_{fr}	= external load	(kg)
J_K	= mass moment of inertia of coupling (motor side)	(kgm ²)	m_{tot}	= total mass (with linear motion system)	(kg)
J_M	= mass moment of inertia of motor	(kgm ²)	m_{lin}	= total linearly moved mass	(kg)
J_{Rv}	= mass moment of inertia of timing belt side drive	(kgm ²)	n_1	= speed of motor	(min ⁻¹)
J_S	= mass moment of inertia of system with external load	(kgm ²)	n_2	= speed of screw	(min ⁻¹)
			n_{Mmax}	= maximum motor speed	(min ⁻¹)
			P	= screw lead	(mm)
			S	= safety factor	(-)
			s_B	= acceleration travel	(m)
			t_B	= acceleration time	(s)
			v	= maximum linear speed (as required, or limited by mechanics)	(m/min)

6 Linear motion systems

6.1 Principles

6.1.4 Calculations

Servo motor acceleration characteristics

Horizontal mounting orientation:

$$(6-1) \quad M_B = 0.8 \cdot M_{\max} - M_R \pm M_L$$

Vertical mounting orientation:

$$(6-2) \quad M_B = 0.8 \cdot M_{\max} - M_R - M_G \pm M_L$$

$$(6-3) \quad M_L = \frac{1.592 \cdot 10^4}{i} \cdot F_L \cdot P$$

The calculations for stepping motors are performed in a similar manner, but with certain constraints. Rexroth provides assistance on request.

Conditions:

$$M_{\max} \leq M_{\text{perm}}$$

M_L must be added if acting in the direction of movement, subtracted if acting counter to the direction of movement.

Weight moment (vertical mounting orientation)

$$(6-4) \quad M_G = \frac{1.561 \cdot 10^3}{i} \cdot m_{\text{lin}} \cdot P$$

Base plate fixed, carriage travels:

$$(6-5) \quad m_{\text{lin}} = m_b + m_{\text{fr}}$$

Carriage fixed, base plate travels:

$$(6-6) \quad m_{\text{lin}} = m_{\text{tot}} - m_b$$

Conditions:

$$S \cdot M_G < M_{\text{brake}}$$

$$S \geq 2 \text{ (recommended)}$$

m_{tot} according to the calculation in the respective linear motion system product catalog, plus weight of motor, coupling or timing belt side drive, trailing cables

Mass moment of inertia of system with external load

Motor attachment via motor mount and coupling:

$$(6-7) \quad J_{\text{fr}} = J_S + J_K + J_{\text{Br}}$$

Conditions:

$$\text{for handling} \quad J_{\text{fr}} < 6 \cdot J_M$$

$$\text{for processing} \quad J_{\text{fr}} < 1.5 \cdot J_M$$

J_S from the respective linear motion system product catalog

Motor attachment via timing belt side drive:

$$(6-8) \quad J_{\text{fr}} = \frac{J_S}{i^2} + J_{\text{Rv}} + J_{\text{Br}}$$

Mass moment of inertia on motor journal

$$(6-9) \quad J_{\text{tot}} = \frac{J_S}{i^2} + J_M + J_K + J_{\text{Rv}} + J_{\text{Br}}$$

J_S from the respective linear motion system product catalog

J_M see motor data in the product catalog

6 Linear motion systems

6.1 Principles

6.1.4 Calculations

Speed

$$(6-10) \quad n_1 = \frac{i \cdot v}{P} \cdot 1000$$

$$(6-11) \quad i = \frac{d_2}{d_1} = \frac{n_1}{n_2}$$

Conditions:

$$n_1 \leq n_{Mmax}$$

v from the respective linear motion system product catalog

Acceleration time

$$(6-12) \quad t_B = J_{tot} \cdot \frac{n_1 \cdot 0.10472}{M_B}$$

Acceleration

$$(6-13) \quad a = \frac{v}{t_B \cdot 60}$$

Acceleration travel

$$(6-14) \quad s_B = 0.5 \cdot a \cdot t_B^2$$

6.1.4.3 Deflection

Unsupported installation

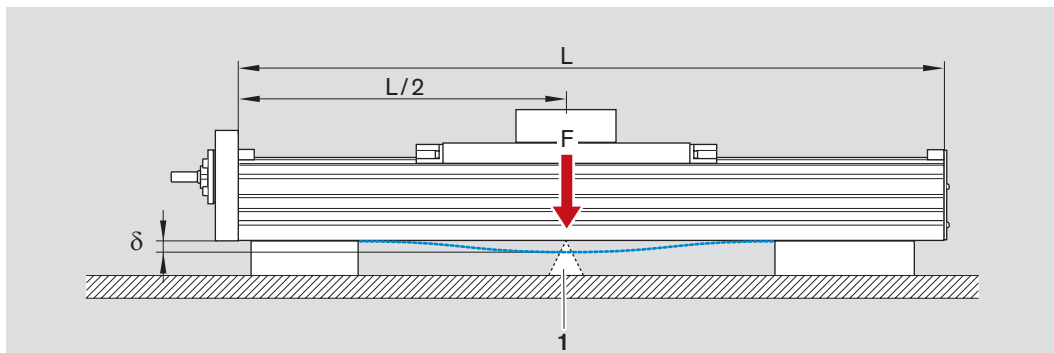
A particular feature of linear and compact modules is that they can be installed without supports. Deflection must, however, be taken into consideration, because it limits the possible load.

The maximum permissible deflection δ_{max} depends on:

- the external load F,
- the length L,
- the rigidity of the linear motion system,
- the rigidity of the mounting base and the bearings.

The maximum permissible deflection δ_{max} must not be exceeded. If the deflection is too great or if high system dynamics are required, supports must be provided every 300 to 600 mm (1).

The deflection can affect the life expectancy. Normally, this is not taken into account when calculating the nominal life.



Deflection δ in an unsupported installation

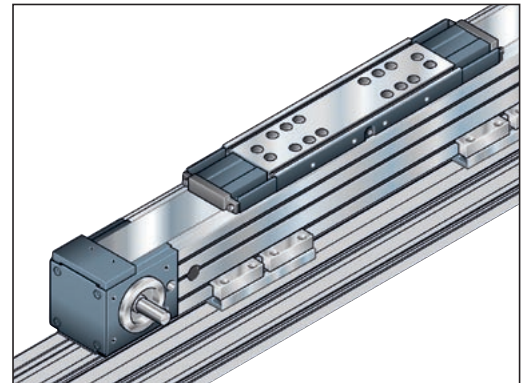
6 Linear motion systems

6.2 Linear modules






6.2.1 System characteristics

All drive types

Available in many different versions, linear modules can be used in a wide variety of applications. All of the drive types in the Rexroth range are used in linear modules. The guideways can be either ball rail systems or cam roller guides.

**Ball rail system
Cam roller guide**

Linear module MKR with toothed belt drive on a Rexroth construction profile

Drive unit		Guideway	
		Ball rail system	Cam roller guide
Ball screw		MKK	–
Toothed belt		MKR	MLR
Rack and pinion		MKZ	–
Pneumatic		MKP	–
Linear motor		MKL/LKL	–

Linear modules are recognizable by their almost square cross-section. Most modules are equipped with one ball rail guideway. The MKZ and one MKR size have two ball rail systems, and the

MLR series has one cam roller guideway. The versions with two rails are particularly suitable for very high moment loads.

Motor attachment

In MKK modules, the motor is attached via a motor mount and coupling, and in MKR modules via a gear reducer. Timing belt side drives can be

fitted to all linear modules with ball screw drive. Toothed belt driven linear modules are also available with a gear unit.

Features

- All linear modules can be delivered in any desired length, i.e. each module can be cut to length with millimeter accuracy to suit the customer's application.
- All linear module types are generally available in lengths up to 6 m, the actual length depending on the size and the drive type. Modules with toothed belt drive (MKR) can even be manufactured in lengths of up to 12 m.
- The largest linear module sizes can move masses of up to 1000 kg.
- MLR modules with cam roller guide can travel at speeds of up to 10 m/s.
- MKK linear modules with ball screw drive achieve a repeatability of up to 0.005 mm and a positioning accuracy of up to 0.01 mm.
- If required, the linear modules can be equipped with direct measuring systems. Direct travel measurement improves the positioning accuracy.
- Depending on the type and size, linear modules can be covered with a steel or polyurethane sealing strip or with bellows to protect the guideway and the drive unit from dirt.
- Carriages are available in different lengths and versions (with threaded holes or T-slots).

6 Linear motion systems

6.2 Linear modules

6.2.2 Linear modules MKK with ball rail system and ball screw drive

High precision and high load capacities

The ball screw drive in MKK linear modules enables a very high level of positioning accuracy with simultaneously high load capacities when adjusted to zero backlash. Through the special design of the ball screw end bearings, MKK linear modules can achieve higher travel speeds than normal for linear motion systems of this kind.

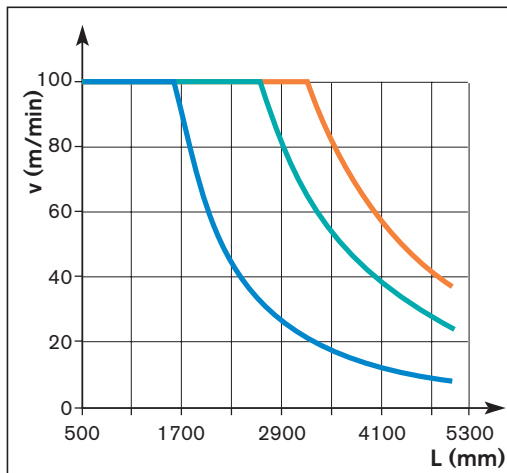
The travel speed and the module length are restricted by the ball screw's critical speed (see Chapter 5, section 5.1.3.3).

Screw supports

One size of the MKK linear module series can be fitted with screw supports. This significantly extends the permissible length and enables travel at high speed over the entire length.

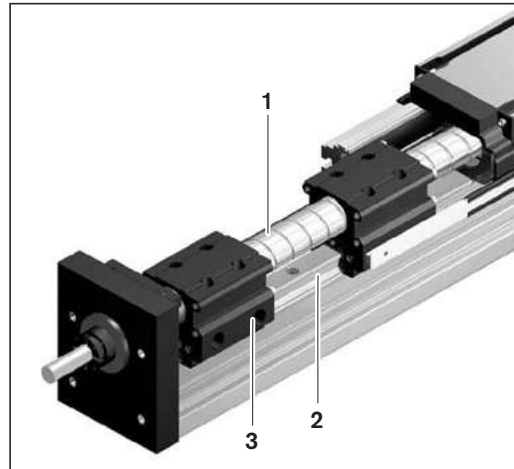
Example

The example below, based on the data for an MKK 25-110 linear module, clearly shows the effect of the screw supports (SS) on the length. Travel at maximum speed can be achieved over significantly longer strokes.

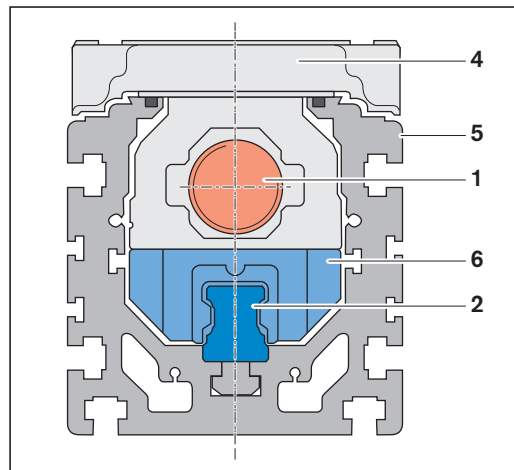


Permissible speeds with and without screw supports for a linear module MKK 25-110

- L = module length (mm)
- v = travel speed (m/min)
- Permissible speed without SS
- Permissible travel speed with 1 SS (on either side of the carriage)
- Permissible travel speed with 2 SS (on either side of the carriage)



Linear module MKK 25-110 with screw supports



Structural design of a linear module MKK

- 1 Ball screw
- 2 Guide rail
- 3 Screw support (only in one MKK size)
- 4 Carriage
- 5 Frame
- 6 Runner block

6 Linear motion systems

6.2 Linear modules

6.2.3 Linear modules MKR/MLR with ball rail system/cam roller guide and toothed belt drive

High speeds

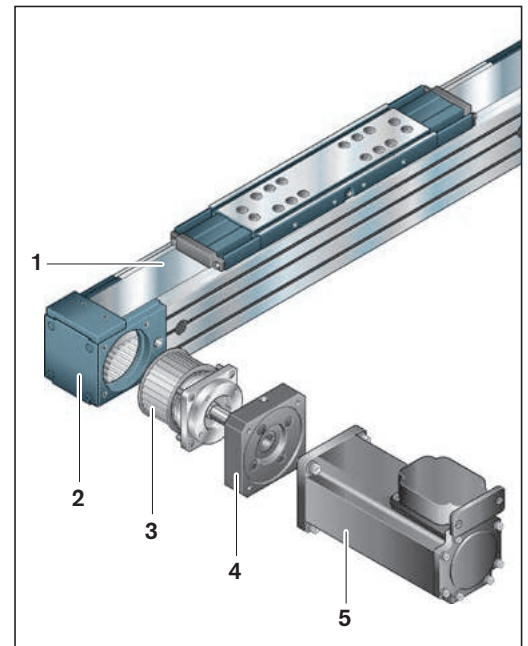
MKR and MLR linear modules are particularly suitable for applications with high travel speeds because of the toothed belt drive. They can be supplied with a separate gear unit or with an integrated gear reducer. In the case of integrated gear reducers, the planetary gears are located in the module's belt pulley (3), ensuring very compact construction. Different gear transmission ratios allow optimum matching of the external load and the motor's inertia. This results in a highly dynamic drive.

Gear reducer**Sealing system**

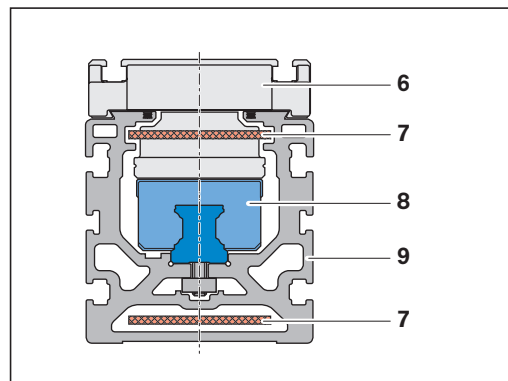
The guideway is sealed off against dirt by gap-type sealing and by the fact that the toothed belt runs inside the frame. This sealing system is maintenance-free.

Sealing strip

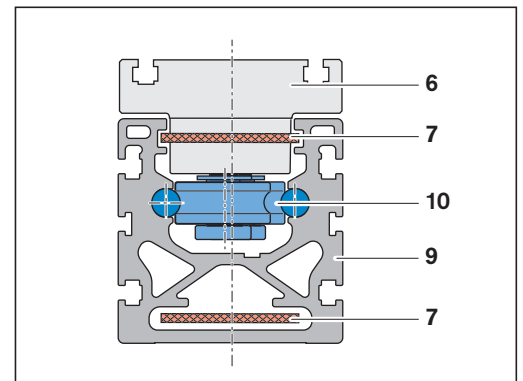
In very dirty environments, the MKR module can be fitted with a sealing strip to provide additional protection.



Linear module MKR with sealing strip



Linear module MKR with ball rail system



Linear module MLR with cam roller guide

- 1 Sealing strip
- 2 Drive end block
- 3 Belt sprocket with integrated gear reducer
- 4 Motor mount
- 5 Motor

- 6 Carriage
- 7 Toothed belt
- 8 Ball rail system
- 9 Frame
- 10 Cam roller guide

6 Linear motion systems

6.2 Linear modules

6.2.4 Linear modules MKR/MKZ with two ball rail systems and toothed belt/rack and pinion drive

High torque load capability

The MKR and MKZ types of two-rail linear modules are particularly suitable for high torque loads. Because of the frame's high planar moment of inertia and the spacing between the two guide rails, these modules can be operated with significantly higher loads than one-rail linear modules.

Two guide rails

High speeds

MKR linear modules with toothed belt drive are suitable for high-speed and material handling applications.

Heavy loads

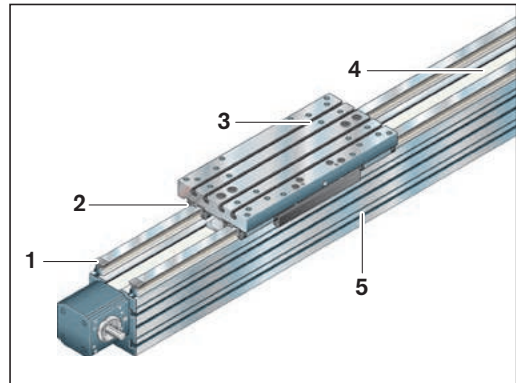
The MKZ linear module with rack and pinion drive is available for moving heavy loads over long travel distances. For vertical applications in particular, it allows large masses to be lifted safely. In vertical installations, there is also the possibility of having the frame profile travel. In this case, the carriage, including the gear unit and motor, is stationary.

Vertical operation

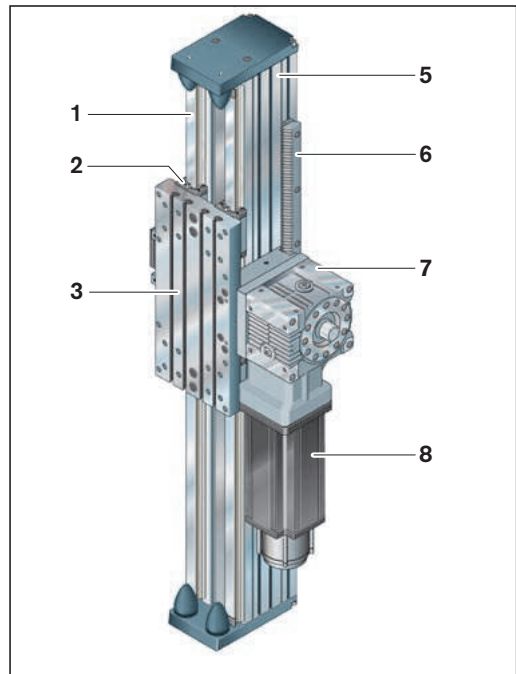
Multiple-carriage applications

Unlike all the other drive versions, in linear modules with rack and pinion drive the motor is not connected to the frame or to one of the end blocks, but instead is fastened directly to the carriage. This makes it possible to move several carriages independently of each other and to build systems with long travel distances and high-speed motion.

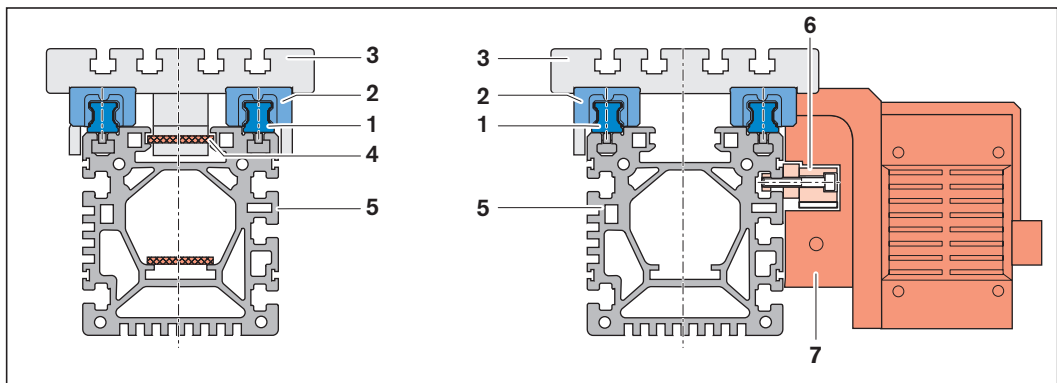
- 1 Guide rail
- 2 Runner block
- 3 Carriage
- 4 Toothed belt
- 5 Frame
- 6 Gear rack
- 7 Gear reducer
- 8 Motor



Linear module MKR with toothed belt drive



Linear module MKZ V (for vertical installation) with rack and pinion drive



Linear module MKR (left) with two ball rail systems, and linear module MKZ with two ball rail systems

6 Linear motion systems

6.2 Linear modules

6.2.5 Linear modules MKP with ball rail system and pneumatic drive

Pneumatic drive

In MKP linear modules, the carriages are driven purely by pneumatic power, i.e. with compressed air. An electric motor is not required.

Travel between end positions

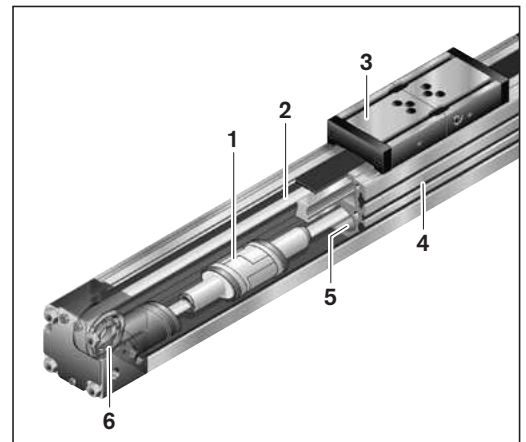
In pneumatic drives, the carriage can only travel full strokes, from end position to end position, or up to a shock absorber as a mechanical stop. Unlike all the other drive types, travel to intermediate positions is not possible.

Shock absorbers

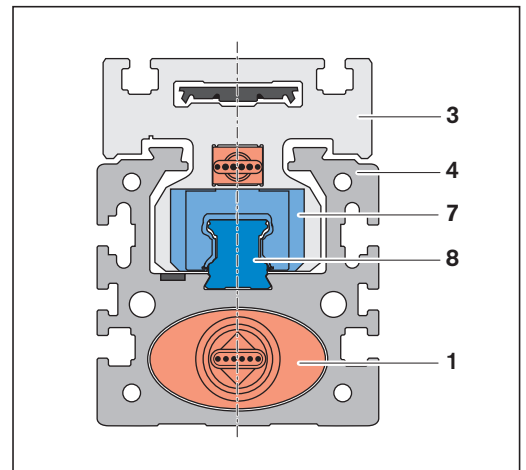
The shock absorbers and adjustable pneumatic end position cushioning allow impact-free braking even from high speeds.

Design calculations

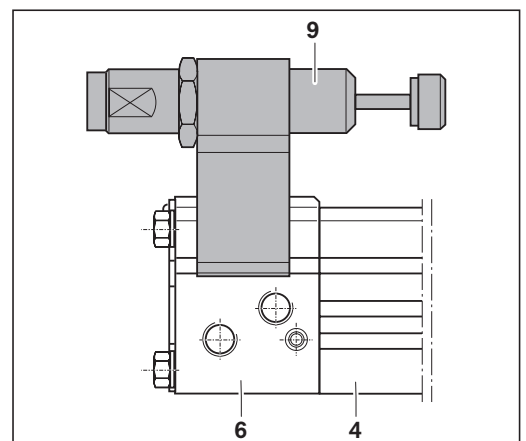
The design calculations for MKP linear modules differ from those for the other modules. The special features of these calculations are explained in the product catalog.



Linear module MKP with pneumatic drive



Structural design of linear module MKP



Shock absorber for linear module MKP

- 1 Double-acting piston of the pneumatic drive
- 2 Belt
- 3 Carriage
- 4 Frame
- 5 Cylinder integrated into the frame
- 6 End block with pulley
- 7 Runner block
- 8 Guide rail
- 9 Shock absorber

6 Linear motion systems

6.2 Linear modules

6.2.6 Linear modules MKL and LKL with ball rail systems and linear motor

High dynamics with low-noise generation

Linear modules with linear motor are characterized particularly by their high speed range, high dynamics, and low noise generation.

The MKL and LKL linear modules are complete linear units. With these models, there is no need to purchase additional components, such as a motor or coupling. In the closed-type MKL linear modules both the guideway and the drive unit are located inside the frame and are also protected by a sealing strip. In the open-type LKL linear modules, the guide and drive unit are exposed. If protection is required, bellows are available.

Wear- and maintenance-free motor

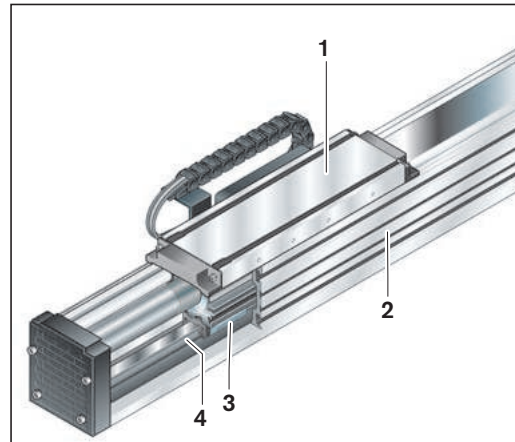
Since the linear motor has no internal moving parts, there is no wear. The motor requires no maintenance. Only the ball rail system requires servicing.

The secondary element of the motor is designed as a thrust rod, so the rail guide is not subjected to additional stressing by magnetic forces.

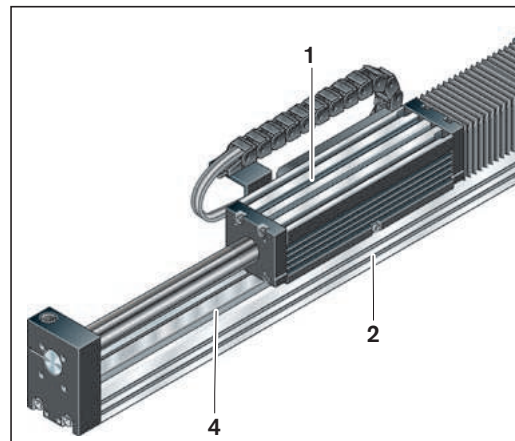
Applications

MKL and LKL linear modules are ideal for highly dynamic positioning of small, equally distributed loads. This can reduce cycle times, particularly in production lines, allowing significantly higher productivity rates to be achieved. These modules can be used for many different tasks in factory automation systems, medical and biomedical equipment, scanning and printing systems, and in the electronics and packaging industries.

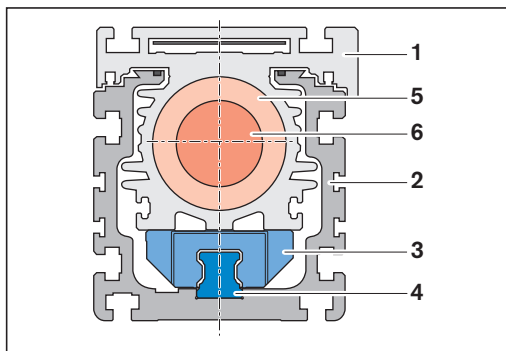
Modules with linear motors are unsuitable for processing ferromagnetic materials because there is a risk that shavings will be attracted by the permanent magnet of the secondary element.



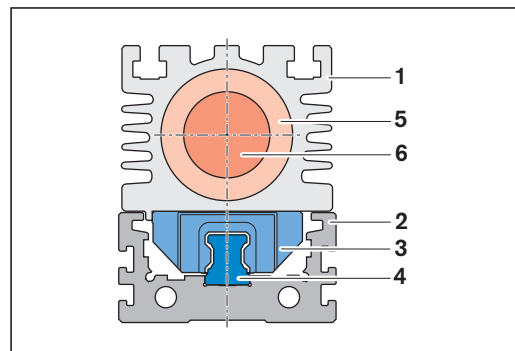
Closed-type linear module MKL



Open-type linear module LKL



Closed version of the linear module: MKL



Open version of the linear module: LKL

- 1 Carriage
- 2 Frame
- 3 Runner block
- 4 Guide rail

- 5 Motor winding (primary element)
- 6 Permanent magnet thrust rod (secondary element)

6 Linear motion systems

6.2 Linear modules

6.2.7 Connection elements for linear modules

Combination of MKK, MKR and MLR

The robotic erector system for Rexroth linear modules helps users to install and attach linear modules more easily and to connect linear modules to one another.

Connection plates, connection brackets

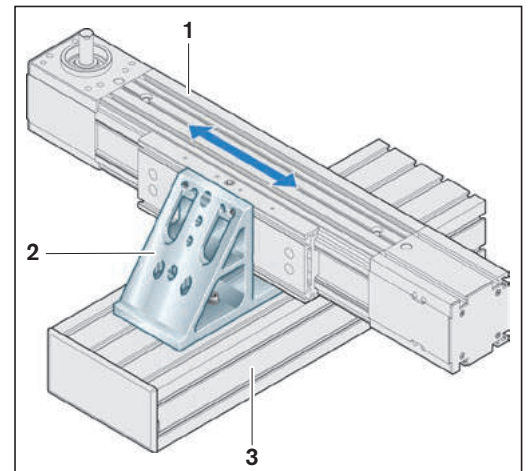
Modules with ball screw drive and toothed belt drive can thus be combined. The basic elements (plates and brackets) have been designed to allow modules to be connected to other modules of the same size or one size larger or smaller. Connecting shafts allow two linear modules with toothed belt drive to operate in parallel.

Connection shafts**Basic mechanical elements range**

The connecting elements also allow quick and easy adaptation to the profiles and frames in Rexroth's basic mechanical elements (BME) range. Linear modules can also be connected to other linear motion systems such as compact modules or ball rail tables.

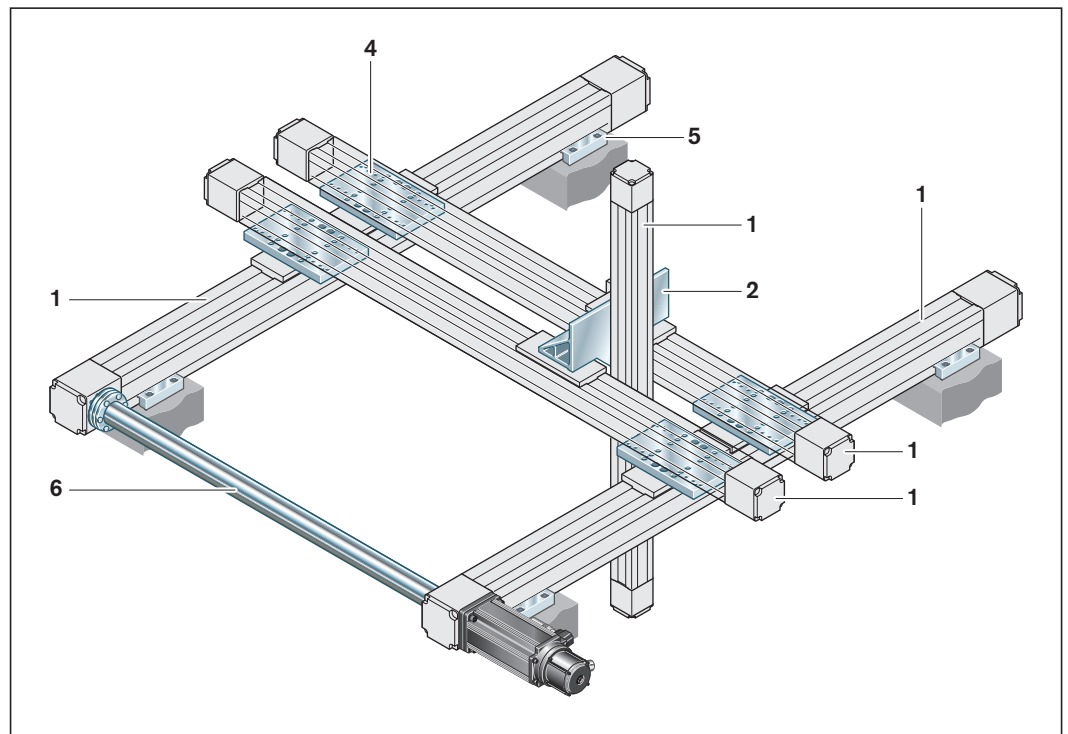
Features

Since the connecting elements are standardized, mass-produced components, they help users to cut their own design and manufacturing costs and to respond flexibly to different linear motion requirements and applications.



Linear module fixed to BME profile via a connection bracket; stationary carriage, moving frame

- 1 Linear module
- 2 Connection bracket
- 3 BME profile
- 4 Connection plate
- 5 Clamping fixture
- 6 Connecting shaft



Linear module connection and combination possibilities

6 Linear motion systems

6.3 Compact modules

6.3.1 System characteristics

High power density Compact design

Compact modules are characterized by their high power density. All compact module types are equipped with two ball rail systems and optionally with one or two carriages. Two carriages enable even higher loads to be carried.

Low profile

Compact modules are recognizable by their relatively flat construction. The ratio of width to height is approximately 2:1 for all types and sizes. In comparison to linear modules with the same size of ball rails, the profile cross-section is smaller although higher loads are possible.

Small profile cross-section

Rexroth offers the following versions in the Compact module range:

- CKK compact module with ball screw drive
- CKR compact module with toothed belt drive
- CKL compact module with linear motor

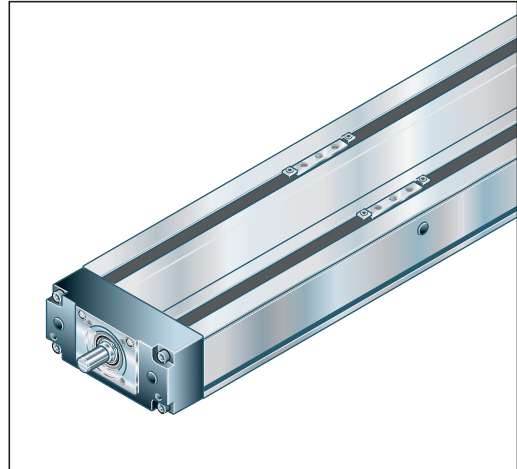
The CKK and CKR versions are closed-type modules, while the CKL series is open.

Motor attachment

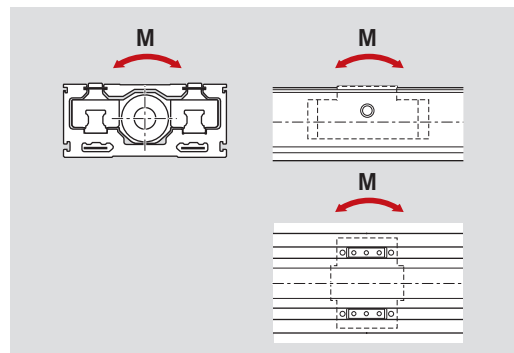
In the CKK and CKR series, the motor is attached via a motor mount and coupling. A timing belt side drive can also be attached to compact modules of the CKK type. Toothed belt-driven compact modules are available with a separate gear unit or an integrated gear reducer.

Features

- All compact modules can be delivered in any desired length to suit the customer's wishes.
- CKR compact modules are available in lengths up to 10 m, depending on the size. CKK modules have a maximum possible length of 5.5 m. CKL modules are available up to 2.8 m.
- Compact modules are particularly suitable for very high torsional and longitudinal moments. The versions with two carriages per guide rail can withstand particularly high longitudinal moment loads.
- The largest versions of compact modules can move masses up to 200 kg.
- CKK compact modules with ball screw drive achieve a repeatability of up to 0.005 mm and a positioning accuracy of up to 0.01 mm.
- For high positioning accuracy, compact modules can be equipped with linear encoders.



Compact module CKK with ball screw drive



Torsional and longitudinal moments

6 Linear motion systems

6.3 Compact modules

6.3.2 Compact modules CKK with ball rail systems and ball screw drive

Higher speeds

Through the special design of the ball screw end bearings, CKK compact modules can achieve higher travel speeds than normal for linear motion systems of this kind. However, the speed and the module length will always be restricted by the ball screw drive's critical speed.

Two carriages

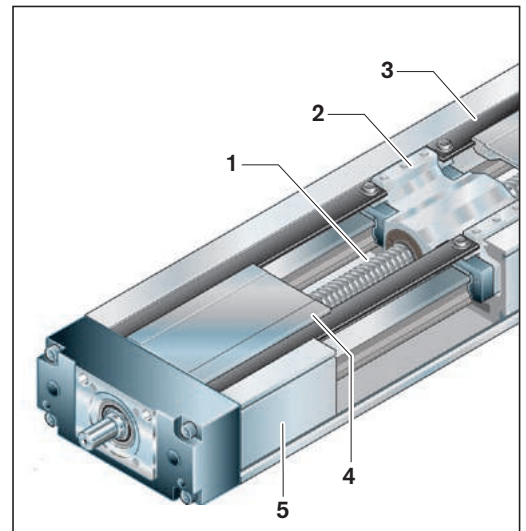
Users can choose between modules with one or two carriages, as appropriate for the application and the load. Versions with two carriages are used for particularly high loads.

Sealing system

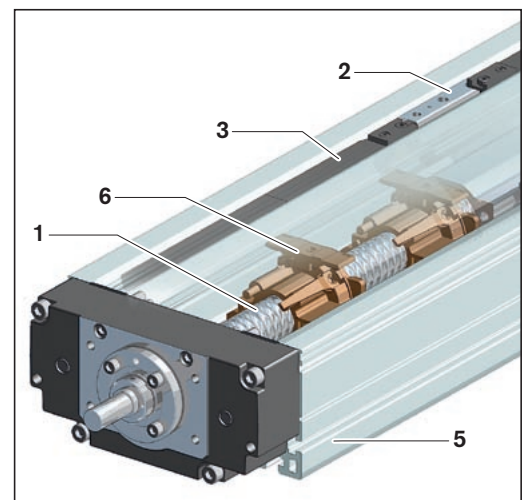
The drive unit and the guideways are protected by an aluminum cover and two gap-type seals made from polyurethane strip.

Screw supports

One size of the CKK compact module series can be fitted with screw supports. This enables high travel speeds to be achieved over the entire length.

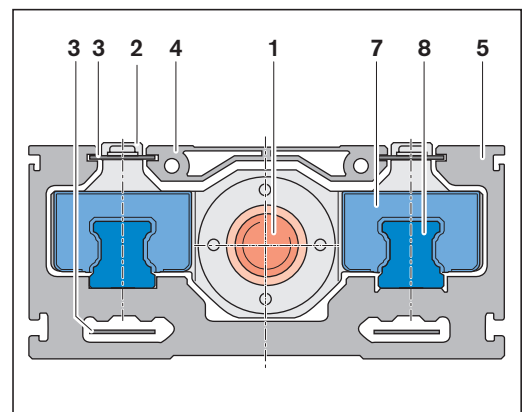


Compact module CKK



Compact module CKK 25-200 with screw supports

- 1 Ball screw drive
- 2 Carriage
- 3 PU strip (gap-type seal)
- 4 Aluminum cover
- 5 Frame
- 6 Screw support
- 7 Integrated runner blocks
- 8 Guide rail



Structural design of compact module CKK

6 Linear motion systems

6.3 Compact modules

6.3.3 Compact modules CKR with ball rail systems and toothed belt drive

Gear unit

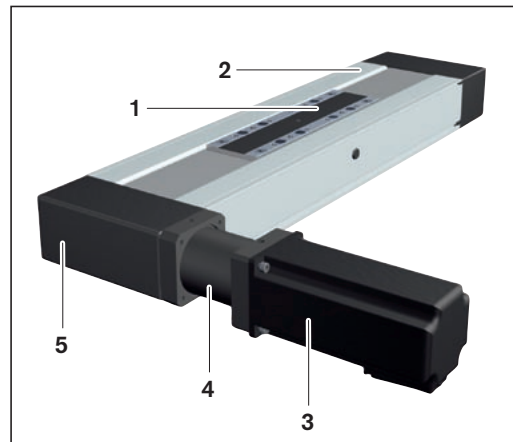
CKR compact modules can be supplied with a separate gear unit. Different gear transmission ratios allow optimum matching of the moved mass and the motor's inertia. This results in a highly dynamic drive with travel speeds of up to 5 m/s.

Carriage

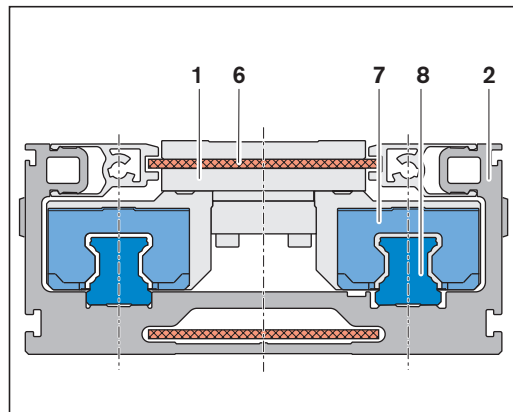
A long or short carriage can be used, as appropriate for the load to be carried. A long carriage contains two runner blocks per guide rail and is therefore suitable for high loads. A short carriage contains one runner block per guide rail.

Sealing system

The guideway is sealed off against dirt by gap-type sealing and by the fact that the toothed belt runs inside the frame. This sealing system is maintenance-free.



Compact module CKR with toothed belt drive



Structural design of compact module CKR

- 1 Carriage
- 2 Frame
- 3 Motor
- 4 Gear unit
- 5 Drive end block
- 6 Toothed belt
- 7 Runner block
- 8 Guide rail

6 Linear motion systems

6.3 Compact modules

6.3.4 Compact modules CKL with ball rail systems and linear motor

Linear encoder

CKL compact modules have an ironless linear motor and a precision optical linear encoder.

Reference edge

A machined reference edge enables quick and precise mounting. Because of the secondary element's U-shape, no additional magnetic forces act on the carriage or the rail guide.

Backlash-free, wear-free and maintenance-free drive

Because no mechanisms are required for converting rotary to linear motion, the drive system in CKL compact modules is backlash-free, wear-free and maintenance-free. The only component requiring maintenance is the ball rail system, which can be lubricated easily by one-point lubrication.

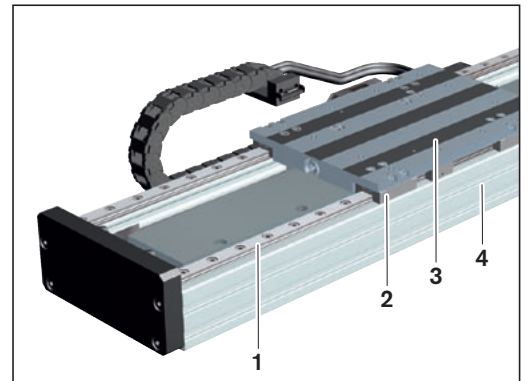
Applications

CKL compact modules are particularly suitable for applications with high requirements on:

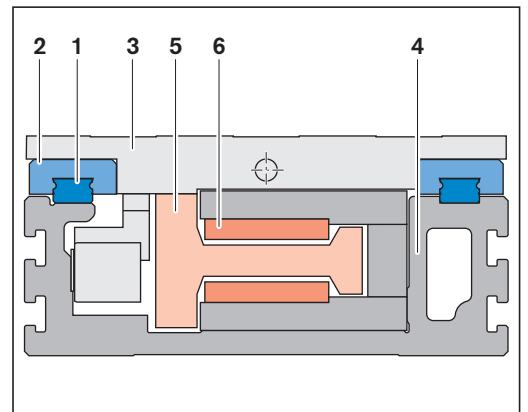
- speed (up to 5 m/s)
- acceleration (up to 250 m/s²)
- travel accuracy
- positioning accuracy
- fast cycle times

CKL compact modules are therefore an ideal choice for handling applications. They can reduce cycle times, particularly in production lines, allowing significantly higher productivity rates to be achieved.

Modules with linear motors are unsuitable for processing ferromagnetic materials because there is a risk that shavings will be attracted by the permanent magnet of the secondary element.



Compact module CKL with two ball rail systems and linear motor



Structural design of compact module CKL

- 1 Guide rail
- 2 Runner block
- 3 Carriage
- 4 Frame
- 5 Primary element
- 6 Secondary element (permanent magnet)

6 Linear motion systems

6.3 Compact modules

6.3.5 Connection elements and Easy-2-Combine automation system

Standardized connection elements

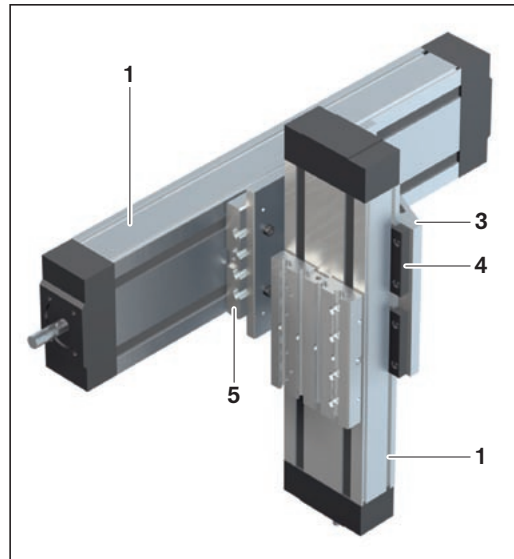
Because of their identical external dimensions, CKK and CKR modules can be combined via brackets and plates. They can also be connected to other linear motion systems, such as linear modules and ball rail tables, and can be adapted to the profiles and frames of Rexroth's basic mechanical elements (BME) range.

Easy-2-Combine

The connection elements for compact modules also allow them to be adapted to the Rexroth Easy-2-Combine automation system. This modular system for multi-axis automation solutions from Rexroth comprises components for:

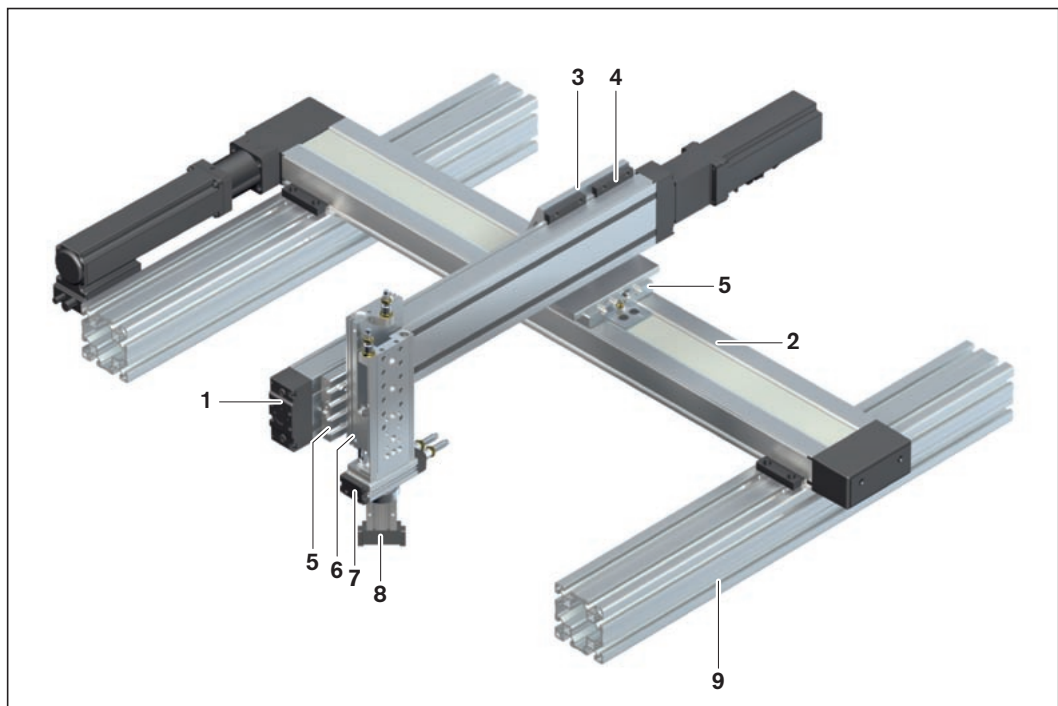
- linear motion (MSC mini-slides)
- rotary motion (RCM rotary compact modules)
- gripping functions (GSP grippers)

Standardized mounting interfaces ensure positive- and force-locking connections between the handling modules, eliminating the need for complex and expensive customer-built adapter plates.



Compact module CKK mounted to the carriage of a second CKK via a bracket and connection plate

- | | |
|----------------------|-----------------------------|
| 1 Compact module CKK | 6 Mini-slide MSC |
| 2 Compact module CKR | 7 Rotary compact module RCM |
| 3 Connection bracket | 8 Gripper GSP |
| 4 Clamping fixture | 9 BME profile |
| 5 Connection plate | |



Module combination using the Easy-2-Combine automation system

6 Linear motion systems

6.4 Precision modules

6.4.1 System characteristics

High rigidity
High precision
Extremely compact design

Precision modules with ball rail systems and ball screw drive are extremely compact, highly rigid and precise linear motion systems. They are particularly suitable for drive, transport and positioning tasks.

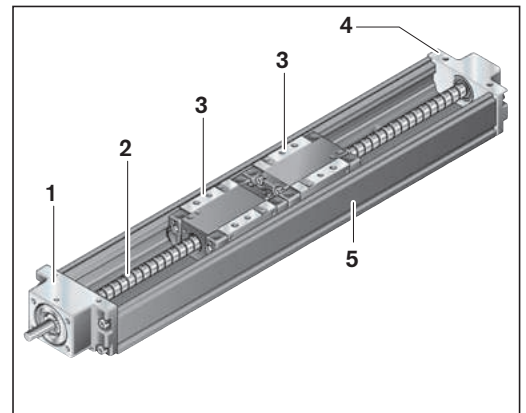
Applications

PSK precision modules are ready-to-install, highly integrated solutions for applications in areas such as the electronics industry, medical technology, packaging machinery, and factory automation.

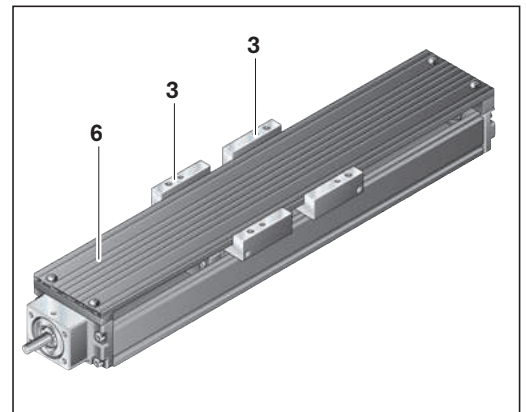
Structural design

PSK precision modules comprise the following components:

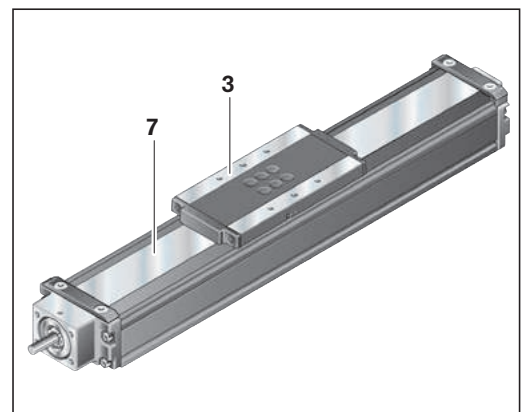
- Extremely compact and rigid precision steel profile frame (5) with reference edge and integrated guideway geometry.
- Precision ball screw drive (2) with zero-backlash nut system
- Aluminum end enclosures with bearings, one designed as a fixed bearing (1), the other as a floating bearing (4)
- Carriages (3) in various designs, made from steel or aluminum with integrated ball runner blocks; one or more carriages, depending on the application
- Optional aluminum cover plate (6) or stainless steel sealing strip (7) to protect the internal elements



Precision module PSK without cover and with two carriages



Precision module PSK with cover plate and two carriages



Precision module PSK with sealing strip and one carriage

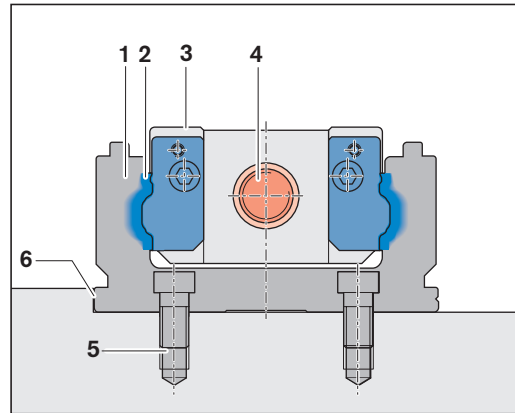
6 Linear motion systems

6.4 Precision modules

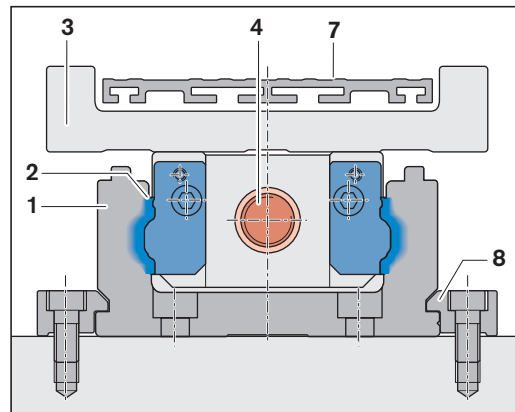
6.4.1 System characteristics

- Mounting of customer-built attachments** The carriages have tapped bores and pin holes for mounting of customer-built attachments.
- Fastening to the mounting base** The machined reference edge on the frame simplifies installation on the mounting base. The reference edge enables rapid mounting and easy alignment of the axis.
- Precision modules can be fixed in place either with screw-fasteners in the frame itself or with external clamping fixtures. Screw-fasteners are used in PSK modules without cover or with cover plate. Clamping fixtures are suitable for all of the cover options:
- Without cover
 - With cover plate
 - With sealing strip
- Motor attachment** The motor is attached via a motor mount and coupling or via a timing belt side drive.
- Features**
- PSK precision modules achieve a repeatability of up to 0.005 mm, a positioning accuracy of up to 0.01 mm, and a guidance accuracy of up to 0.005 mm.
 - All precision modules are available in finely graduated length increments. The maximum length is 940 mm.
 - The largest size can move loads of up to 800 kg.
 - The maximum speed is 1.6 m/s.
 - A machined reference edge on the side of the frame enables rapid mounting and easy alignment of the axis.

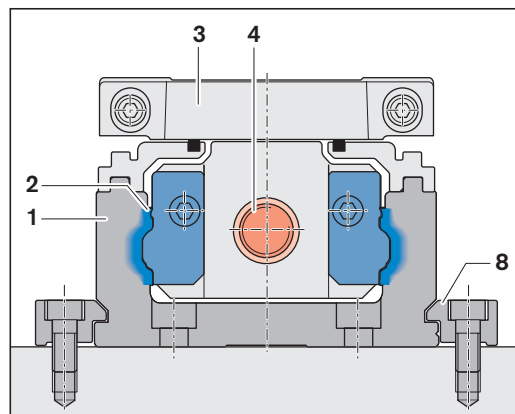
- 1 Steel profile (frame)
- 2 Guideway running track (integrated into the steel profile)
- 3 Carriage with integrated ball runner blocks
- 4 Ball screw drive
- 5 Mounting screws
- 6 Reference edge
- 7 Cover plate
- 8 Clamping fixture



PSK without cover, fastened by screwing the frame directly onto the mounting base



PSK module with cover plate, fastened with clamping fixtures



PSK with sealing strip (the sealing strip cannot be seen in the cross-sectional view), fastened with clamping fixtures

6 Linear motion systems

6.5 Ball rail tables

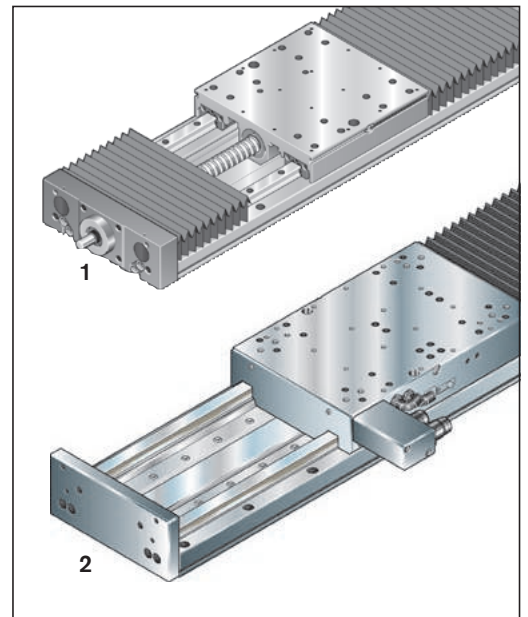
6.5.1 System characteristics

High torsional and longitudinal moments

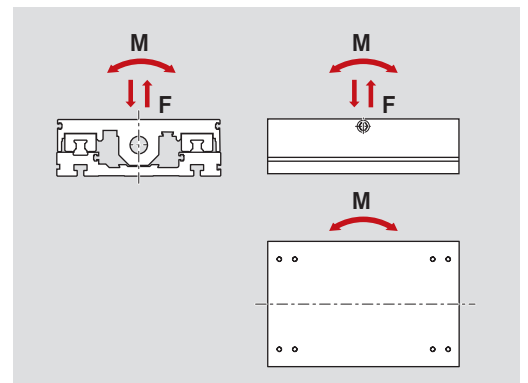
TKK and TKL ball rail tables are equipped with two ball rail systems that can resist very high torsional moments thanks to the wide spacing between the rails. With two runner blocks per guide rail, they can also withstand high longitudinal moments. Because of the four long runner blocks built into the carriage and the large table plate, ball rail tables can handle high forces as well, provided that they are fully supported. Due to machining of the frame and the use of a high precision ball screw (TKK) or linear motor (TKL) as the drive, very high levels of repeatability, positioning and guidance accuracy can be achieved.

High loads**High repeatability, positioning and guidance accuracy****Features**

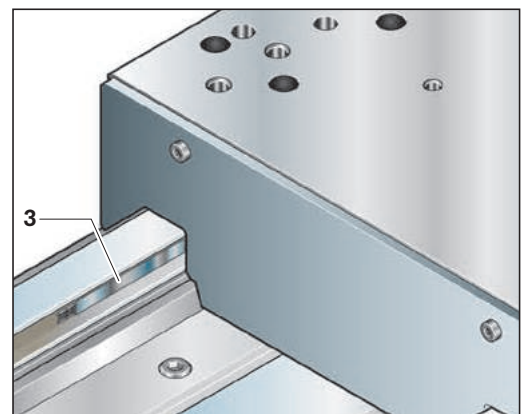
- All ball rail tables can be supplied in 60 or 80 mm length increments, depending on the mounting hole spacing in the rails.
- The maximum length is 2.86 m for the TKK type and 4 m for the TKL type.
- TKK ball rail tables have a maximum travel speed of 1.6 m/s. The TKL can be operated at speeds of up to 8 m/s.
- Ball rail tables are particularly suitable for handling very high payloads F and torsional and longitudinal moments M .
- The largest sizes can move loads of up to 2500 kg.
- TKK and TKL tables achieve a repeatability of up to 0.005 mm, a positioning accuracy of up to 0.01 mm, and a guidance accuracy of up to 0.007 mm.
- High-quality, oil and moisture-resistant, welded bellows protect the internal elements.
- TKK ball rail tables can be equipped with rotary encoders and/or a linear position measuring system. TKL ball rail tables have an integrated measuring system.
- A machined reference edge on the side of the base plate enables rapid mounting and easy alignment of the axis.



Ball rail tables TKK (1) and TKL (2)



Torsional and longitudinal moments, vertical forces



TKL with integrated position measuring system

- 1 Ball rail table TKK with ball screw drive
- 2 Ball rail table TKL with linear motor
- 3 Scale

6 Linear motion systems

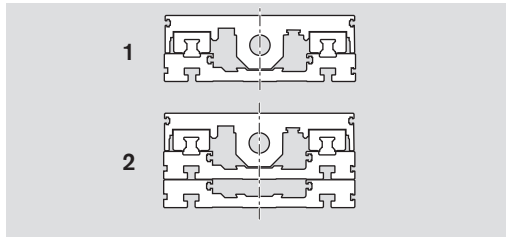
6.5 Ball rail tables

6.5.2 Ball rail tables TKK with ball rail systems and ball screw drive

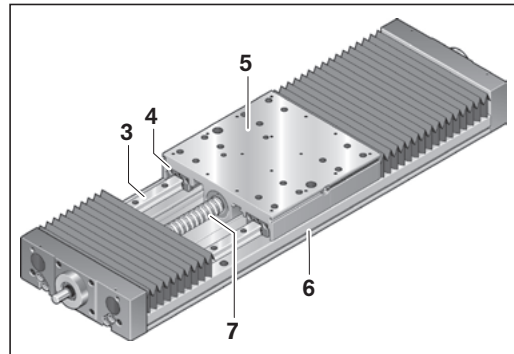
The base plate of TKK ball rail tables consists of a machined aluminum or steel profile.

Increasing the rigidity To increase the rigidity, TKK versions with an aluminum base plate can be reinforced with a second base plate mounted underneath the first (2).

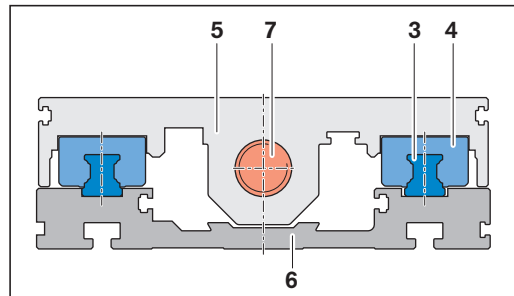
The rigidity can also be increased with a steel base plate. The steel version is not only more rigid but also more accurate.



Versions with one or two base plates



Ball rail tables TKK with two ball rail systems and ball screw drive



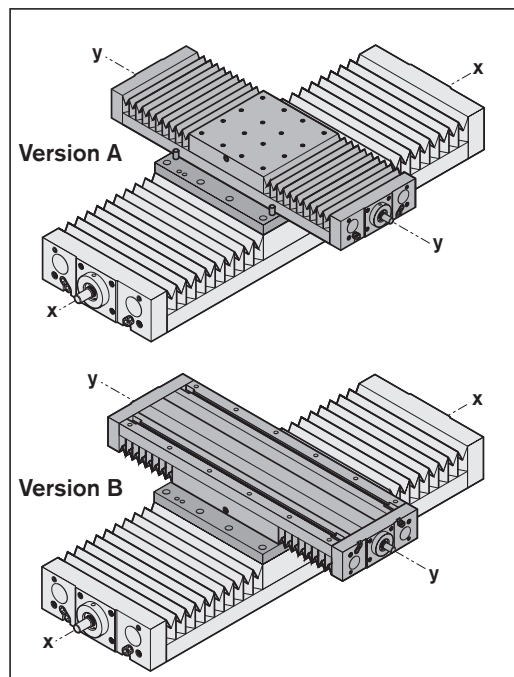
Structural design of a ball rail table TKK

Motor attachment The motor can be attached either directly via a motor mount and coupling or via a timing belt side drive. When a timing belt side drive is used, the motor no longer lies in the same axis as the module but is installed below, above or alongside the drive end enclosure. The linear motion system's overall length is therefore shorter than with motor attachment via motor mount and coupling.

- | | |
|----------------------------|----------------|
| 1 Low-profile TKK version | 4 Runner block |
| 2 High-profile TKK version | 5 Carriage |
| 3 Guide rail | 6 Base plate |
| | 7 Ball screw |

Maintenance The only maintenance required is lubricating the runner blocks and the ball screw assembly. This can be done by one-point lubrication using either of the easily accessible lube ports located on each side of the carriage.

Two-axis units Cross-plates can be used to combine TKK ball tables into X-Y units (two-axis units). The connection system has been designed to allow tables to be combined with others of the same size or one size larger or smaller. Either the base plate (version A) or the carriage (version B) of the Y-axis can be fixed to the carriage of the X-axis. In a two-axis unit, the inaccuracies of the individual axes and that of the cross plate have a cumulative effect. The elastic deflection of the components in the Y-axis also have to be taken into account, because this axis is not fully supported. Use of the high-profile version (2) can significantly reduce elastic deflection.



Two-axis units connected by cross-plates

6 Linear motion systems

6.5 Ball rail tables

6.5.3 Ball rail tables TKL with ball rail systems and linear motor

Speed
Acceleration
Positioning accuracy
Short stroke

TKL ball rail tables are particularly suitable for applications requiring high travel speed, high acceleration, and very good positioning accuracy. High-speed positioning tasks or high-acceleration short-stroke movements in quick succession can be performed without difficulty, even in applications with very high demands on positioning accuracy.

Applications

Application areas for TKL ball rail tables include:

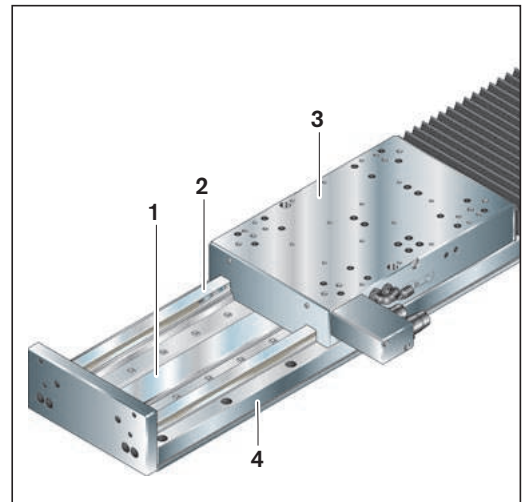
- Transfer lines
- Machining centers
- Handling systems
- Textile machines
- Packaging machines
- Testing equipment

Measuring system

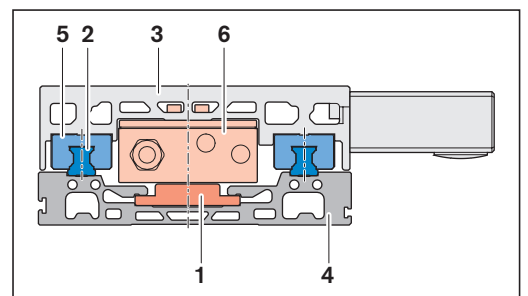
The high positioning accuracy is due to a precise, distance-coded measuring system. This system is largely insensitive to temperature effects, since the scale is fixed to the base plate.

Maintenance

The runner blocks can be easily lubricated via a central lube port. The linear motor itself is maintenance-free.



Ball rail table TKL with two ball rail systems, linear motor and integrated measurement system



Structural design of a ball rail table TKL

- 1 Secondary element (permanent magnet)
- 2 Guide rail
- 3 Carriage
- 4 Base plate
- 5 Runner block
- 6 Primary element

6 Linear motion systems

6.6 Linear motion slides

6.6.1 System characteristics

Linear bushings

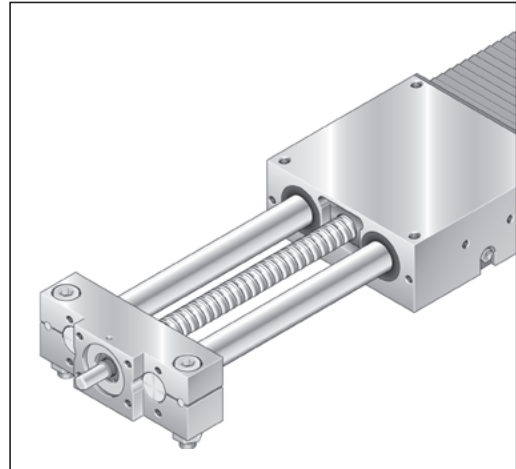
The guide unit in linear motion slides consists of shaft-mounted linear bushings that ensure smooth operation and long life. Linear motion slides are an economical solution for many application areas.

Versions

Closed-type linear motion slides are suitable for use as self-supporting units. Here, the two ends of the guideway are connected to the mounting base by means of shaft support blocks. Open-type linear motion slides are fastened to the mounting base via shaft support rails.

Features

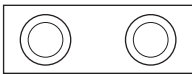
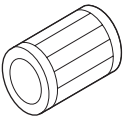

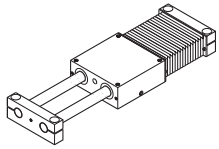
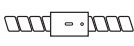
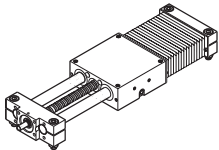
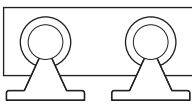
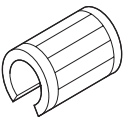

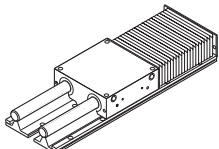
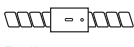
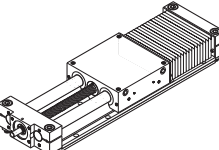
- Linear motion slides SGK and SOK with ball screw drive achieve a repeatability of up to 0.005 mm and a positioning accuracy of up to 0.01 mm.
- Linear motion slides can be delivered in any desired length to suit the customer's wishes. The maximum length is up to 5.3 m, depending on the size and the drive unit used.
- The largest size can move loads of up to 1000 kg.
- High-quality, oil and moisture-resistant, welded bellows on both sides of the carriage protect the internal elements.



Closed-type linear motion slide SGK with ball screw drive

Overview

The following table shows all the available versions. More information on linear bushings and shafts can be found in Chapter 4.

Version	Guideway	Drive unit	Designation	Linear motion slides
 Closed-type	 Closed-type Super linear bushing	 Without drive	SGO	
		 Ball screw	SGK	
 Open-type	 Open-type Super linear bushing	 Without drive	SOO	
		 Ball screw	SOK	

6 Linear motion systems

6.7 Cartesian motion systems

6.7.1 System characteristics

Cartesian Motion System CMS

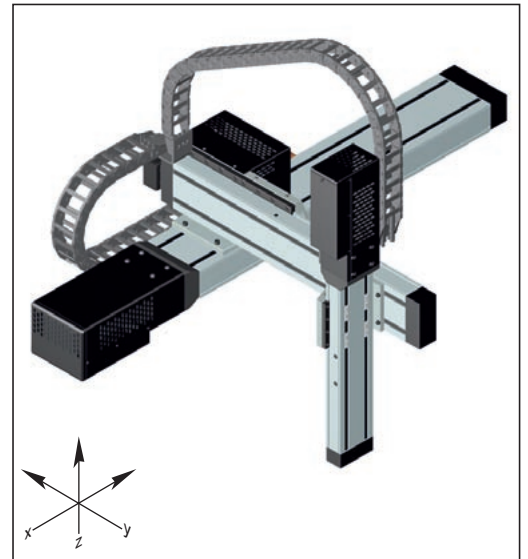
Multiple axes Flexibility Programmability

No custom-designed systems

Motor Controller Connection elements

Cartesian Motion Systems CMS carry the concept of linear motion systems a step further to provide complete plug-and-play solutions. In principle, they can be classed as linear robots because they offer multiple-axis capability, design flexibility and programmability.

Linear motion systems help users to design and build mechanical machine functions significantly faster and more cost-efficiently by eliminating the need for custom-designed systems. Cartesian motion systems take this strategy even further. They combine standard linear motion systems with a motor and controller, connection elements, cables, and accessories, and are delivered as pre-assembled, pre-configured motion systems that already meet the user's specific requirements.



3-axis motion system

Features

- The customer no longer needs to integrate a wide variety of linear components into his design. Instead, he receives a complete, fully adapted CMS with one, two or three axes.
- The user can select exactly the combination he needs from a wide range of axis configurations and then receives a pre-programmed solution with the necessary visualization for the operator interface. All it takes to customize the system to the application is to enter the appropriate positioning data sets.
- For customers, this results in substantial time savings, because the entire planning phase is significantly shorter, and since the system is shipped as a complete unit, much less time is needed to install it.
- The mechanical elements of the system are robust compact modules, assuring smooth operation, high load capacities and high rigidity thanks to their two integrated, zero-clearance ball rail systems and ball screw drive with zero-backlash nut system.
- The system can be easily installed on the mounting base using clamping fixtures and connection plates that are also compatible with Rexroth's basic mechanical elements range.
- Attachments can be precisely aligned and securely fastened using the tapped bores and pin holes in the carriage.
- Any length can be selected up to the respective maximum axis length.
- The maximum payload is limited by the axis configuration, the mounting orientation and the axis length.
- Internal elements and motors are protected by covers.
- The motor and servo controller combination has been specially optimized for the CMS.
- The servo controller is pre-parameterized in-factory and is available with Profibus DB, CANopen, SERCOS and DeviceNet interfaces.
- The system comes standard with matching software to ensure rapid start-up and safe operation.
- Compared to systems built using individual compact modules, the Cartesian motion system saves space by integrating the motors directly into the modules. There is no motor mount and coupling between the module and the motor.

6 Linear motion systems

6.7 Cartesian motion systems

6.7.2 Basic structure of the CMS

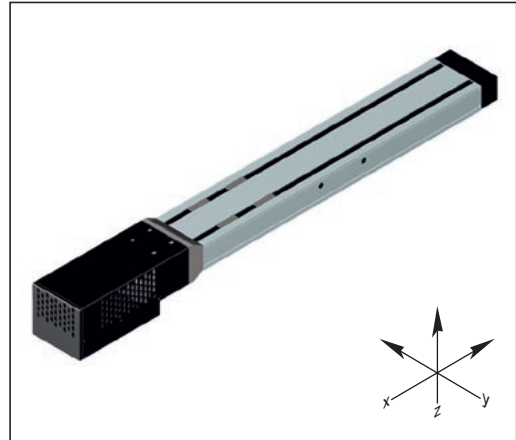
CMS Cartesian motion systems are built using linear motion systems from the CKK Compact module series.

1, 2 or 3-axis systems They are designed as 1, 2 or 3-axis systems. The Y-axis is always one size smaller than the X-axis, and the Z-axis is one size smaller than the Y-axis.

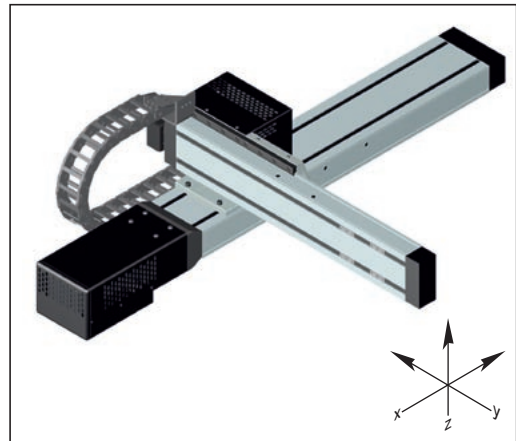
In Cartesian motion systems CMS, the mechanical guide elements and the electric drive, including the drive amplifier and optional control unit, are integrated into an axis system. The pre-configured system reduces the effort required by the user for project planning, on-site integration and operation.

The compact drive amplifier answers the trend toward applications in increasingly smaller cells, as it is suitable for both central and decentralized installation.

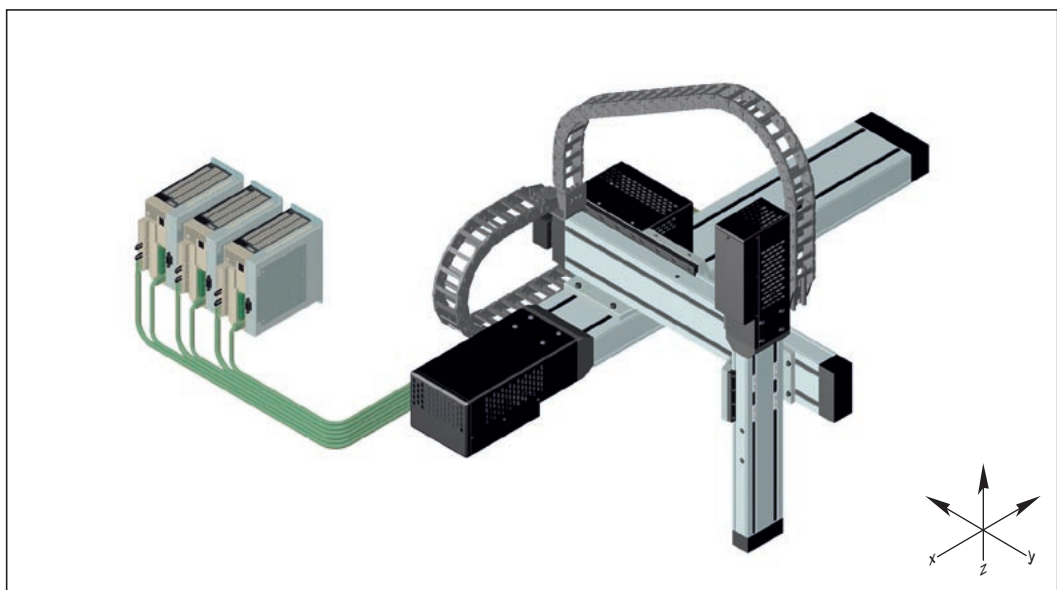
Multi-axis solutions can also be ordered with attractively priced control units.



1-axis system



2-axis system



3-axis system with servo controller for each axis

6 Linear motion systems

6.8 Electrical components

6.8.1 Overview

Prime movers

Various types of prime movers are used with linear motion systems:

- Servo, stepping or three phase motor (for linear motion systems with ball screw, toothed belt or rack and pinion drive)
- Linear motors
- Pneumatic drive (compressed air)
- Handwheel (for linear motion systems with ball screw or toothed belt drive)

Motor

The most commonly used type is a motor attached to the drive unit. The motor is crucial when it comes to designing customer applications using linear motion systems because it directly influences the performance data.

Linear modules, compact modules and ball rail tables from Rexroth are also available in versions with an integrated linear motor. Linear modules can be designed with a pneumatic drive as well.

Handwheels are only used for very simple applications.

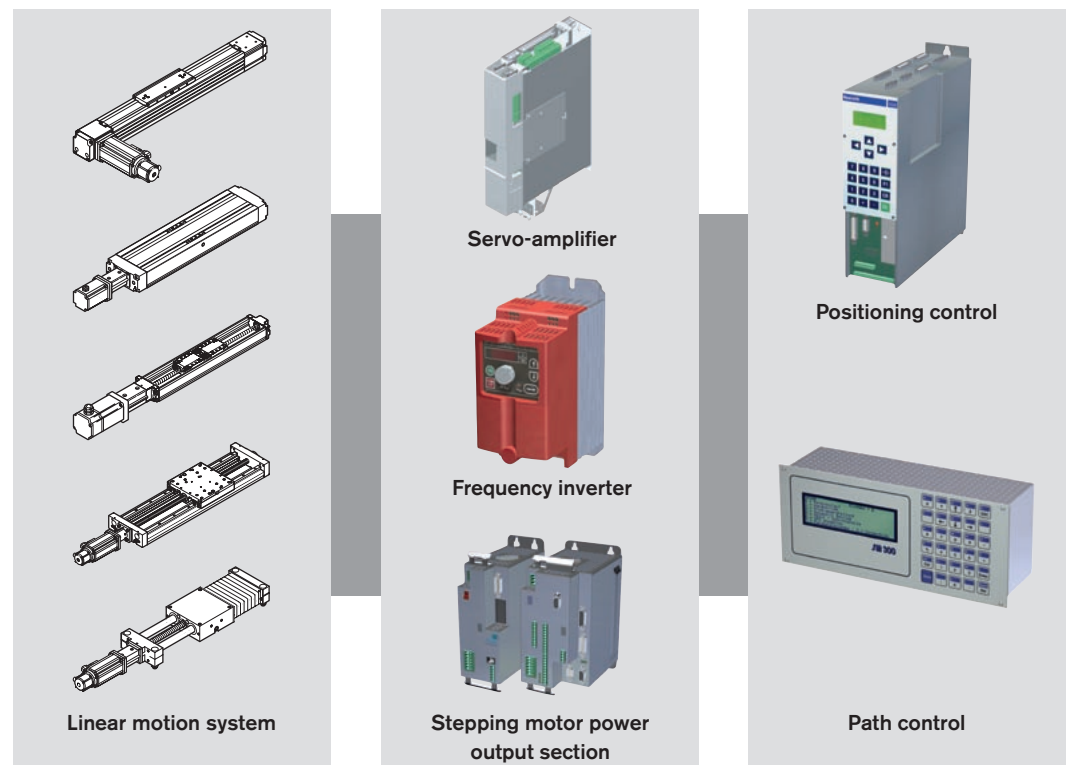
Controller

If a linear motion system has an electric drive, a controller, and optionally a control unit, has to be connected to the motor. The control unit is programmed with the desired travel profile for the linear motion system. The controller and the drive amplifier convert the data from the control unit into corresponding signals for the motor.

In addition to the electrical components in the drive train, switches and sensors are also used in linear motion systems either as limit or reference switches.

Control unit

Switches Sensors



6 Linear motion systems

6.8 Electrical components

6.8.2 Motors

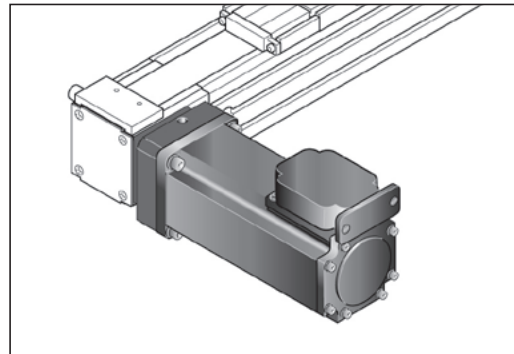
- Servo motors**
- Stepping motors**
- Three-phase motors**

Linear motor

Depending on the application, a linear motion system may be equipped with a servo motor, stepping motor or three-phase motor.

Linear motors are a special case among the servo motors. The linear motor takes the place of the electro-mechanical drive in the linear motion system. It performs no rotary movements, but only linear movements. Linear motors are therefore also referred to as direct linear drives.

The following table shows the main characteristics and their weighting for the various motor types.



Servo motor attached to a linear module with toothed belt drive

Motor	Torque	Speed	Dynamics	Accuracy	Noise characteristics	Handling	Costs
Servo motor (incl. linear motor)	+++	+++	+++	+++	+++	++	+
Three-phase motor	+	+++	+	0	+++	+++	+++
Stepping motor	++	+	++	++	0	++	+++

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

The table can be used to compare motors with one another. However, there are a number of other important points that must also be considered when selecting and dimensioning a motor. To determine the right motor for a linear motion system, the controller and the control unit must be taken into account, since only a sensible combination of these components can ensure that the drive will deliver optimal performance.

The motor design calculation procedure is shown in abbreviated form in the calculation example in section 6.1.4.3.

The operating principle and characteristics of the different motors are described on the following pages. The application areas for each motor type are listed in the table below.

Motor	Uses
Servo motor	<ul style="list-style-type: none"> ■ Highly dynamic and precise positioning applications ■ Diverse applications in all industries and sectors ■ Suitable even for complex motion sequences and cycles ■ High peak torques ■ Synchronous and interpolation modes possible ■ Varying speeds and travel parameters
Three-phase motor	<ul style="list-style-type: none"> ■ Simple positioning tasks with no precision requirements (e.g. switch-controlled) ■ Reversing mode ■ Travel at constant motor speed
Stepping motor	<ul style="list-style-type: none"> ■ Adjustment tasks ■ Transport and positioning tasks with low travel speed and few stations ■ Simple machining processes such as cutting and sawing ■ Not under impact loads

6 Linear motion systems

6.8 Electrical components

6.8.2 Motors

6.8.2.1 Servo motors

Rotary encoder

Servo motors for Rexroth linear motion systems are three-phase synchronous motors with a system for determining the current angular position of the rotor (rotation angle covered in relation to a starting position). The motor must be capable of rotating in both directions. The motor feedback signal is provided by a rotary encoder.

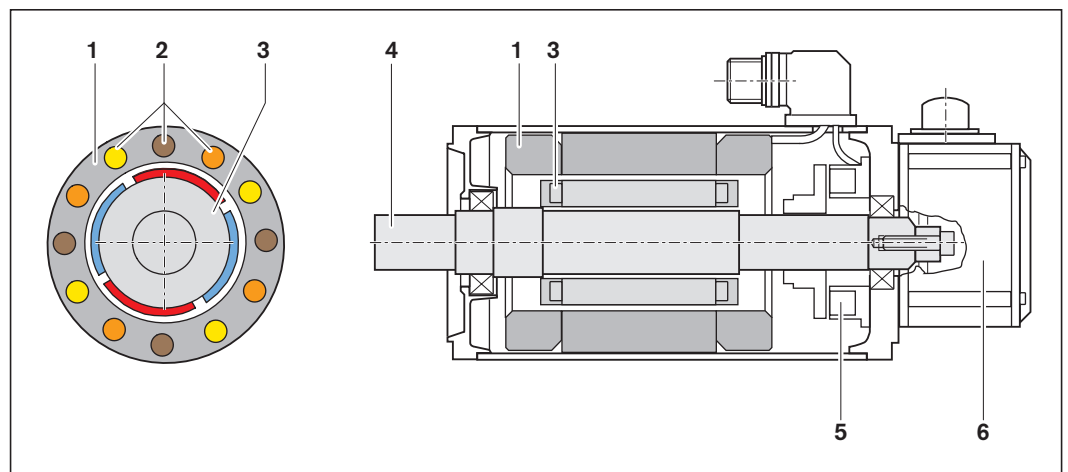
The rotary encoder continuously transmits the current motor position to the drive controller, which regulates the motor speed and positions the motor. The drive controller compares the signal from the rotary encoder with the target value supplied by the control unit. If there is a deviation, the motor is turned in the appropriate direction to reduce the deviation. The drive controller controls the motor windings via a power output section.

**Absolute encoder
Incremental encoder**

Servo motors can be operated with either absolute or incremental encoders. Both versions are used in Rexroth servo motors.



Rexroth servo motor



Structural design of a servo motor

- | | |
|------------------------------------|-----------------------------------|
| 1 Stator with three-phase windings | 4 Rotor shaft |
| 2 Windings (3 phases) | 5 Holding brake |
| 3 Rotor with permanent magnets | 6 Rotary encoder (motor feedback) |

Features

- Servo motors are characterized by their exceptionally low rotor inertia and high power density.
- Servo motors can deliver high peak torques over a wide speed range.
- Because of their low rotor inertia, servo motors are highly dynamic, i.e., they achieve high acceleration rates.
- Servo-drives have very good synchronization capabilities.
- Servo motors are maintenance-free and highly reliable.

6 Linear motion systems

6.8 Electrical components

6.8.2 Motors

6.8.2.2 Linear motors

Primary element
Secondary element

Rexroth linear motors are essentially servo motors in an “unrolled” form. They consist of a primary element with current flowing through it (comparable to the stator in a rotary motor) and a secondary element (comparable to the rotor in a rotary motor). Because they produce linear motion directly, linear motors require no mechanisms, such as a ball screw drive, to convert rotary motion into linear motion.

Linear motors can be designed in different forms:

- Round (MKL and LKL linear modules)
- Flat (TKL ball rail table)
- U-shaped (CKL compact module)

Depending on how the primary and secondary elements are arranged, the magnetic forces in the system can be cancelled out. The drive then exerts no additional radial forces on the guideway. This is the case in the MKL, LKL and CKL modules.

Features

Advantages arising from this drive principle are:

- No mechanical resonance points or compliances
- No backlash
- No wear
- High acceleration capability

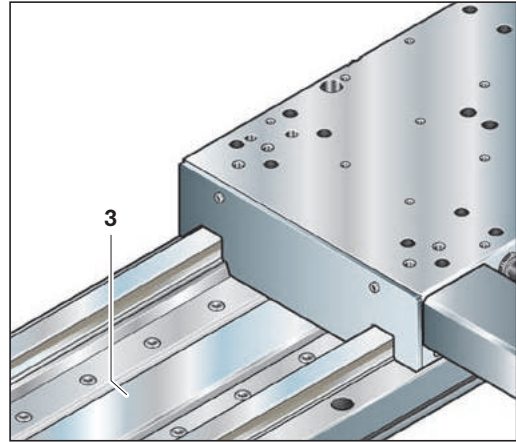
On the other hand, it is not possible use a gear reducer to adapt the travel speed and thrust.

Rotary motor:

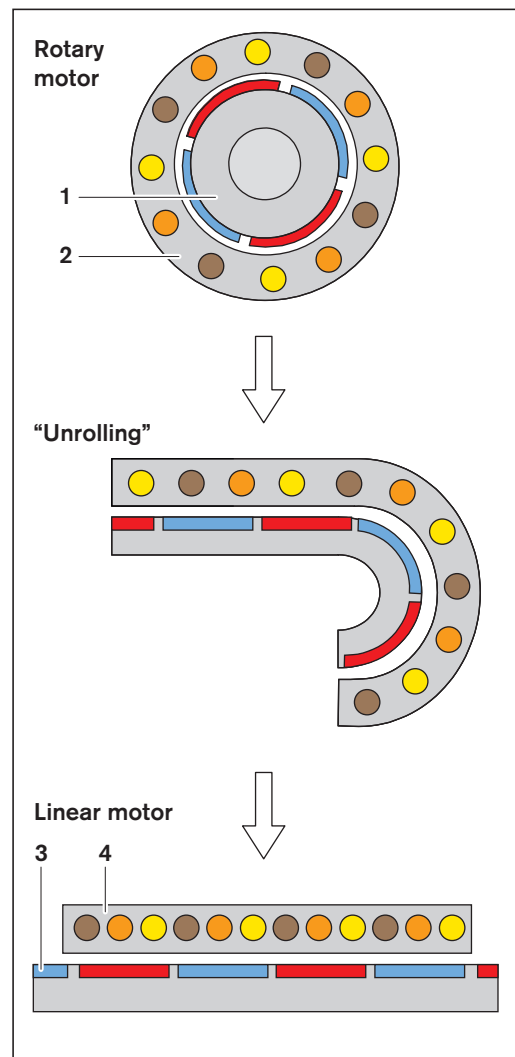
- 1 Rotor with permanent magnets
- 2 Stator with three-phase windings

Linear motor:

- 3 Secondary element (permanent magnets)
- 4 Primary element with three-phase windings



Ball rail table with linear motor



From three-phase motor to linear motor

6 Linear motion systems

6.8 Electrical components

6.8.2 Motors

6.8.2.3 Three-phase motors

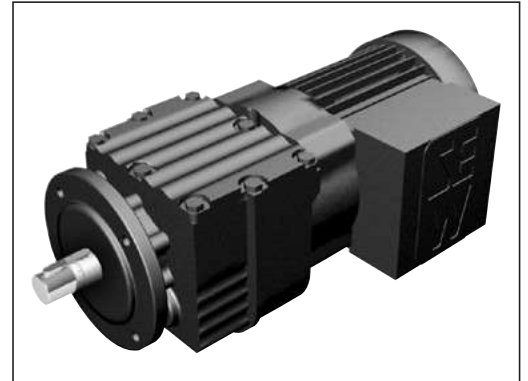
Rugged

The three-phase motors used in Rexroth linear motion systems are asynchronous three-phase motors. They are used as drives in countless applications. Their rugged design has been proven a million times over and they are extremely easy to put into operation. Because of these good characteristics, this motor type has been standardized internationally and is produced in large quantities throughout the world.

Features

Three-phase motors are relatively easy to control and commission.

- Because of their high rotor inertia, the dynamic performance tends to be on the poor side.
- Three-phase motors are maintenance-free and highly reliable.
- They generate very little noise.
- Motor-gear reducer combinations (three-phase gear motors) are available with spur, worm or bevel gearing.



Three-phase gear motor from SEW-EURODRIVE

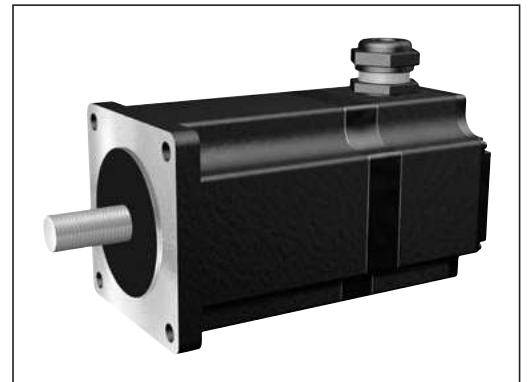
6.8.2.4 Stepping motors

**Economical
Rugged**

Stepping motors are economical, rugged drives. They execute precise rotary movements in steps, as commanded by a positioning control. Stepping motors convert electrical pulses into corresponding analog angles or travel steps, or convert a pulse frequency into a feed value. The motor transmits the travel or speed information coming from the control system directly to the driven system. It does not supply any feedback to the control system. Because stepping motors (provided that they are not overloaded) precisely follow the externally applied field, they can be operated without any position feedback sensors, i.e. without rotary encoders. Therefore, unlike servo motors, which always require a closed-loop position control, stepping motors can be operated in open-loop control.

Features

- In stepping motors, the actual position is not monitored. Nevertheless, a high level of precision is possible, if they are dimensioned correctly.
- Because of their low rotor inertia, stepping motors deliver good dynamic performance, i.e. they can achieve high acceleration rates.
- Stepping motors are maintenance-free and highly reliable.
- Stepping motors can be put into operation quickly and easily, because no control parameters need to be set.
- The torque begins to drop off even at speeds less than 1000 min^{-1} . For high rotary speeds, the torque drop-off must therefore be taken into account.



Stepping motor from Berger Lahr

6 Linear motion systems

6.8 Electrical components

6.8.3 Controllers and control systems

Controllers

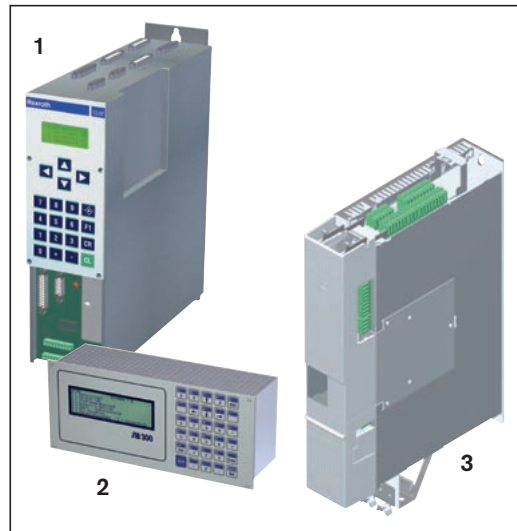
The control system for a linear motion system is programmed with individual positions or with a path including the speeds to be traveled and the accelerations. The commands from the control system are then converted by a drive controller into corresponding signals for the motor. The controller simultaneously monitors the motor's operation.

Adaptation to the motor

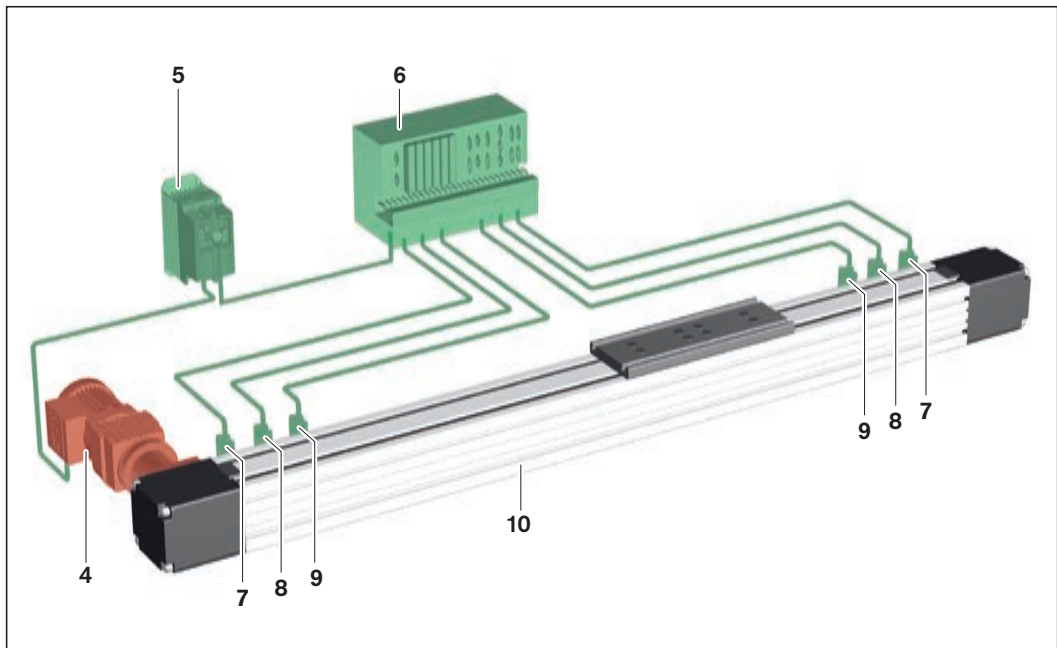
Depending on the application, a servo, three-phase or stepping motor may be used. The open and closed loop control technology must be adapted to the chosen motor.

**Sensors
Actuators**

Additional sensors and actuators may have to be installed to control and monitor the motor (see section 6.8.4).



Control components



Linear motion system with sensors, control system, controller and three-phase motor

- | | |
|----------------------------------|--------------------------------|
| 1 Positioning control | 6 PLC (control system) |
| 2 Path control | 7 Limit switch |
| 3 Controller | 8 Creep mode cutoff switch |
| 4 Three-phase motor | 9 Rapid traverse cutoff switch |
| 5 Frequency inverter (regulator) | 10 Linear module |

6 Linear motion systems

6.8 Electrical components

6.8.3 Controllers and control systems

6.8.3.1 Servo controllers

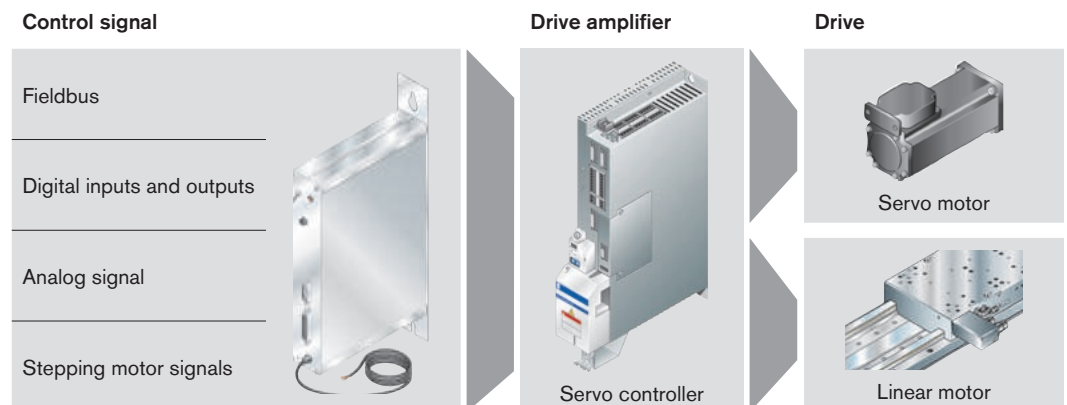
Rexroth's compact controllers contain all the necessary supply and control electronics. Standardized interfaces enable connection to various control systems and permit different operating modes.

Interfaces and bus systems

- **Analog interface**
The servo controller receives analog $\pm 10V$ speed command values from an NC control system. The servo controller in turn transmits incremental or absolute actual position values back to the NC control system, thus closing the position control loop.
- **Stepping motor interface**
The servo controller operates in position loop mode with stepping motor emulation. The position command values are transmitted incrementally from the control system to the controller.
- **Positioning interface**
Up to 64 positioning sets are stored in the servo controller. The drive operates in position-controlled mode and travels to the target position, following the values defined in the positioning sets. The positioning sets are selected and triggered by means of digital PLC inputs. The drive status is signaled back to the master PLC system via digital PLC outputs.

To allow communication between the control system and the controller, different computer cards with specific computing capabilities are available for all versions.

- **SERCOS drive bus**
The SERCOS interface is an internationally standardized real time communications system in which serial data is transmitted via a noise-immune, fiber optic ring. Digital command and actual values are exchanged between the servo controller and the CNC control system in cycles at precisely equal time intervals.
- **Standardized fieldbuses**
When a standardized serial fieldbus interface is used for communication, the servo controller receives digital command values from a control system. In turn, the servo controller transmits status and diagnostic data in digital data packets back to the control system. The following fieldbus interfaces are available:
 - Profibus DP
 - Interbus-S
 - CANopen
 - DeviceNet

**Target position
Actual position**

In addition to converting the control signals into commands for the motor, the controller monitors whether the target position (signaled by the control system) agrees with the actual position of the rotary encoder on the motor and adjusts this if necessary.

6 Linear motion systems

6.8 Electrical components

6.8.3 Controllers and control systems

6.8.3.2 Frequency inverters

Three-phase asynchronous motor

A three-phase (asynchronous) motor's stator voltage and frequency can be varied infinitely by using a frequency inverter. This turns a standard asynchronous motor into a speed-controlled drive system.

Applications

The application areas lie primarily in transport and very simple positioning tasks with few stations and in simple machining processes such as cutting and sawing.



Frequency inverter from SEW-EURODRIVE

6.8.3.3 Positioning control

**Servo motors
Stepping motors**

Positioning controls can be used with servo motors and stepping motors. This type of control is used in linear motion systems when motion sequences are to be determined simply by entering programming sets or when input/output signals are to be monitored and adjusted.

**Positioning of
up to four axes**

Rexroth's positioning controls are used to position up to four axes in a wide variety of applications. They can be combined with the highly dynamic digital servo controllers to produce a powerful and economical control and drive system.

Applications

The application areas cover many sectors, such as:

- Packaging machines
- Linear and X-Y gantries
- Woodworking machines
- Traveling cutters
- Feed and discharge units



Rexroth positioning control

6 Linear motion systems

6.8 Electrical components

6.8.3 Controllers and control systems

6.8.3.4 Path control

Servo motors

A path control unit is used with servo motors when a predefined path is to be traveled at prescribed speeds and accelerations. The path can comprise simultaneous movements in several axes.

24 axes
16 kinematic configurations

Special, high-performance, Windows-based control systems from Rexroth can coordinate up to 24 axes with utmost precision and enable parallel operation of up to 16 kinematic configurations in highly demanding tasks.

Applications

Typical applications for a path control unit are:

- Laser processing
- Water jet cutting
- Milling
- Application of adhesives



Path control unit from SM Motion Control

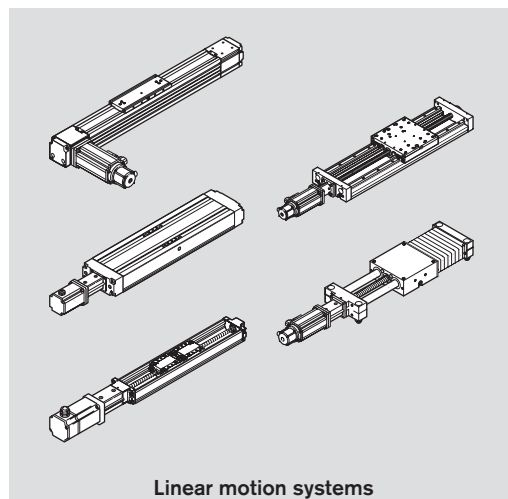
6.8.3.5 Control cabinet solutions

Package of individual components

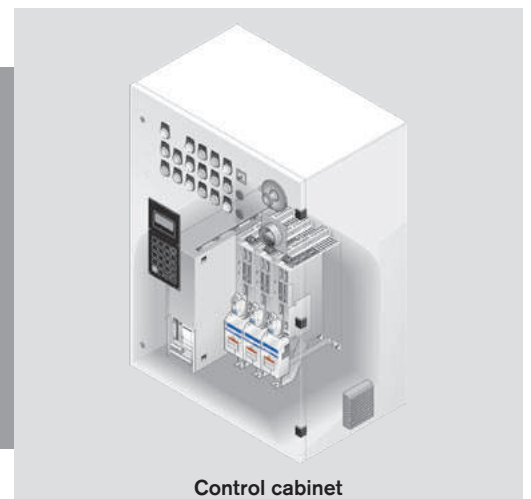
With a package consisting of individual components (linear motion system with motor, drive amplifier, control unit, etc.) the customer has to wire up all the electrical components and commission the system himself.

Complete solution

With a complete solution (linear motion system with motor and control cabinet), all the electrical components are already wired up, built into a control cabinet and ready for operation. All the customer needs to do is to start up the system on site.



Linear motion systems



Control cabinet

6 Linear motion systems

6.8 Electrical components

6.8.4 Switches and sensors

Functions

Switches and sensors are used in linear motion systems to fulfill two important functions:

- Limit switches (not safety switches in the sense of DIN EN 60204-1 or VDE 0113)
- Reference switches

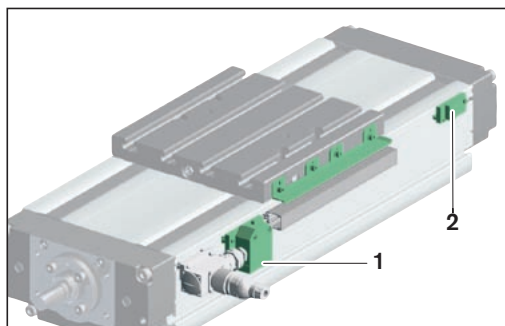
Limit switches

Limit switches immediately interrupt the power supply to the drive when the switch is activated. This is to prevent the carriage from traveling further than intended and thereby damaging components in the linear motion system or in the peripherals.

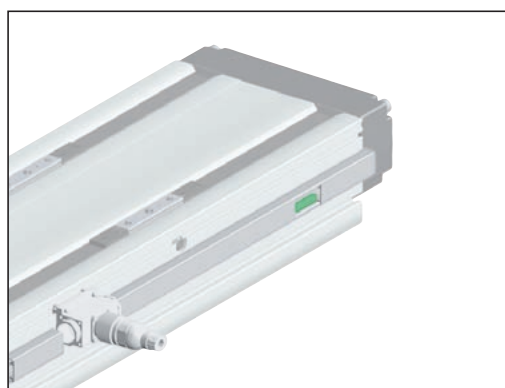
Reference switches

Reference switches inform the controller of the carriage's position in the linear motion system. Motors with incremental encoders require these switches for commissioning purposes and after every interruption to the power supply, if the linear axis does not have a linear measurement system to provide the position reference. With three-phase motors, proximity switches may be used for positioning. In this case, the switch has a direct effect on the linear motion system's precision.

Various types of switch can be installed, as stated below, depending on the type of linear motion system.



Mechanical switch (1) and proximity switch (2) as limit switches



Hall-type sensor as a reference switch

Linear motion system	Switch type			
	Mechanical switch	Proximity switch	Hall sensor	Reed sensor
Linear module	✓	✓	-	-
Compact module	✓ ¹⁾	✓ ¹⁾	✓	✓
Precision module	-	-	✓	✓
Ball rail table	✓	✓	-	-
Linear motion slide	✓	✓	-	-

1) Only one size of compact module can be fitted with this type of switch.

6 Linear motion systems

6.8 Electrical components

6.8.4 Switches and sensors

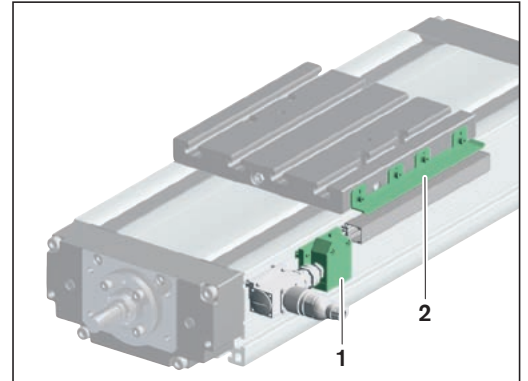
6.8.4.1 Mechanical switches

Limit switches

Mechanical changeover switches are used as limit switches. The switches used as standard by Rexroth are not safety limit switches, however, because they have no positive-opening contacts.

Repeatability

The repeatability of the switch activation point is ± 0.05 mm. The switch is activated by a cam which trips an electro-mechanical switching element. In continuous operation, i.e. when the switch is repeatedly activated at short intervals, an approach speed of 1 m/s must not be exceeded. When the switch is activated infrequently, the speed can be significantly higher without adversely affecting the switch's service life.



Mechanical switch (1) with switching cam (2)

6.8.4.2 Proximity switches

**Reference switches
Limit switches**

Inductive (proximity) switches are used primarily as reference switches. They can, however, also be used as limit switches. Proximity switches are available with normally closed (NC) and normally open (NO) types of contact (both PNP and NPN).

Repeatability

The switch activation point can be traveled to repeatedly with an accuracy of 5% of the switching distance. In Rexroth linear motion systems with a switching distance of 2 mm, an accuracy of less than 0.1 mm can be achieved. Because the switching is electronic and non-contacting, there is no mechanical wear.

Proximity switches can be used up to the maximum speed of the linear motion system concerned.



Proximity switch

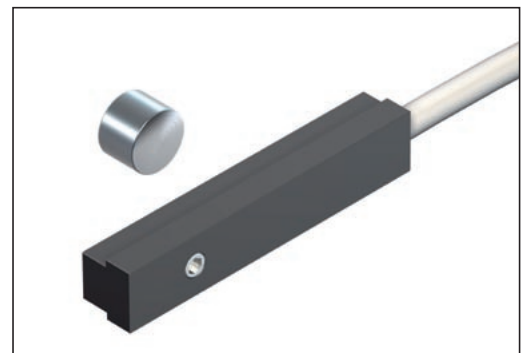
6.8.4.3 Hall sensors

Reference switches

Hall-type sensors are used as reference switches. These are magnetic field sensors equipped with PNP NC and NO contacts.

Repeatability

The switch activation point can be traveled to repeatedly with an accuracy of 0.5% of the switching distance. In a CKK module with a switching distance of approx. 7 to 8 mm, the accuracy is therefore approx. 0.04 mm. Hall-type sensors are suitable for speeds of up to 2 m/s.



Hall sensor

6 Linear motion systems

6.8 Electrical components

6.8.4 Switches and sensors

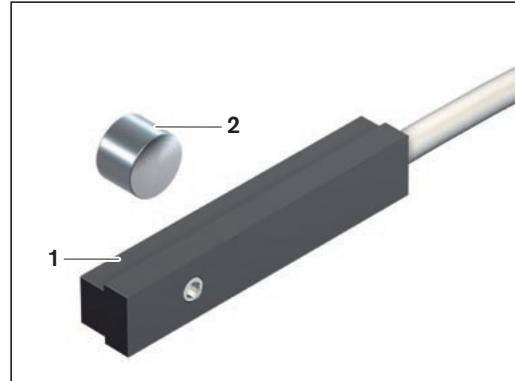
6.8.4.4 Reed sensors

Limit switches

Reed sensors are used as limit switches. They are magnetic field sensors and act as changeover switches. Because of their design they have two switching points and are therefore not suitable for use as reference switches.

Repeatability

The repeatability of the switch activation point is approx. 0.1 mm.
Reed sensors are suitable for speeds of up to 2 m/s.



Reed sensor

- 1 Sensor, mounted on the frame
- 2 Magnet, mounted on the carriage

6.8.4.5 Switch mounting arrangements

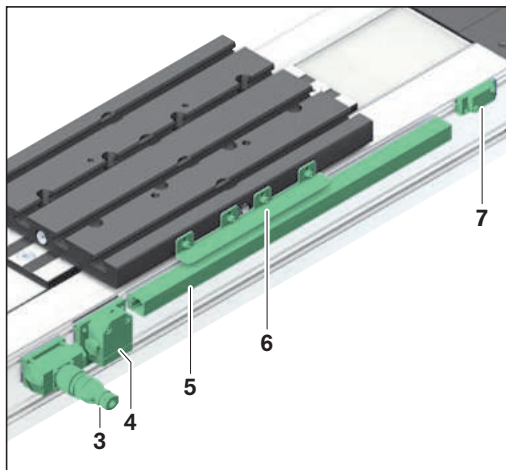
Slots in the frame profiles

All linear motion systems are designed so that switches can be attached. Special slots are provided in the frame profiles for mounting of the switches. Cable ducts, sockets and plugs are also available for wiring up the switches and connecting them to the controllers.

Cable ducts, sockets and plugs

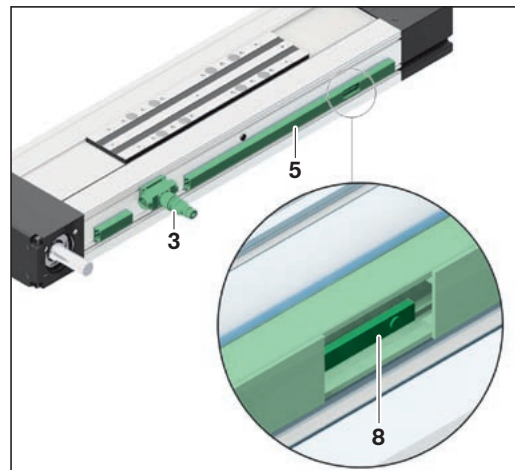
Either mechanical and proximity switch combinations or magnetic field sensor combinations (Hall and Reed sensors) are used.

Most linear motion systems are designed for use with only one of these switch categories. A few modules can be fitted with all of the options.



Mechanical switch, switching cam and proximity switch on a CKR compact module

- 3 Socket and plug
- 4 Mechanical switch
- 5 Cable duct



Magnetic field sensor on a CKR compact module

- 6 Switching cam
- 7 Proximity switch
- 8 Hall or Reed sensor

7 Appendix

7.1 Bosch Rexroth AG: The Drive & Control Company

7.1.1 A strong partner worldwide

Bosch Rexroth AG, part of the Bosch Group, achieved sales of approximately 4.9 billion Euro in 2006 with over 29,800 employees. Under the brand name of Rexroth the company offers all drive and control technologies – from mechanics, hydraulics and pneumatics through to electronics – along with the related services. The global player, represented in over 80 countries, is a strong partner to over 500,000 customers, supplying them with an extensive range of components and systems for industrial and factory automation as well as mobile applications.

7.1.2 Linear motion and assembly technologies

Linear motion products are used in all areas of automated manufacturing requiring precision movement and high load-bearing capacity. Designed as profiled rail systems, linear bushings and shafts, ball screw assemblies or linear modules, they perform crucial functions as interfaces between stationary and moving machine parts. The assembly technologies offered range from basic mechanical elements to modules for manual production systems, transfer systems and modular chain conveyors for transporting parts between machine tools in assembly or packaging lines. With its components and systems Rexroth covers the entire spectrum of linear motion and assembly technology applications.

The range also caters to special requirements and demanding applications. Linear bushings, for example, also come in miniature versions, fulfilling the needs of many machine and system manufacturers for high functionality combined with compact design. Linear motion systems are offered in corrosion-resistant steel versions for applications requiring a high degree of cleanliness and corrosion-resistance, as are commonly found in the food and chemical industries.

7 Appendix

7.1 Bosch Rexroth AG: The Drive & Control Company

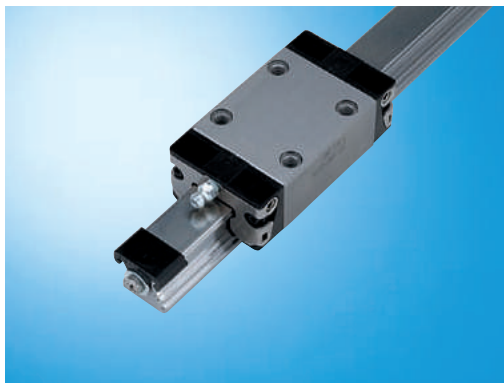
7.1.2 Linear motion and assembly technologies

7.1.2.1 Linear motion technology

Ball rail systems

**One rail system
– many runner blocks**

A complete system for linear guides with ball bearings, which provides the user with infinite possibilities for combining guide rails and runner blocks. Ball rail systems are distinguished in all accuracy classes by their high load capacity and high rigidity, making them suitable for almost all tasks demanding precise linear motion.



Roller rail systems

**Accurate movement
of heavy loads**

Roller rail systems make it easy to handle even the heaviest loads with extremely little effort. With high rigidity central to their design principle to meet the needs of powerful machine tools and robots, linear guides with roller bearings are available in various accuracy and preload classes.



Linear bushings and shafts

**Over 1,000 designs
and variations**

Linear bushing guides can be put together from over 1,000 designs and variations to meet all demands and applications, which means that users can select exactly what they require for every task.



7 Appendix

7.1 Bosch Rexroth AG: The Drive & Control Company

7.1.2 Linear motion and assembly technologies

7.1.2.1 Linear motion technology

Precision ball screw assemblies**Thrust through movement**

Precision ball screw assemblies work with high accuracy and speed and are also available for fast delivery. A broad selection of precision screws, zero backlash or adjustable-preload single nuts in a variety of designs, and double nuts, caters to all feed, positioning and transport requirements. Drive units combine ball screw assemblies with end bearings, housings and screw supports, and can also be supplied with driven nuts.

**Linear motion systems****Installation made easy**

The compact modules are the most advanced of today's linear motion systems. As pre-assembled modules they can be easily integrated into machine constructions without any of the effort normally required to align and match up the guide and the drive unit. This ease of installation applies to all of the linear motion systems, irrespective of their design. Connection elements simplify assembly even further. Individual performance characteristics such as "precise movement of loads" or "fast travel" are optimized as necessary for each application.



7 Appendix

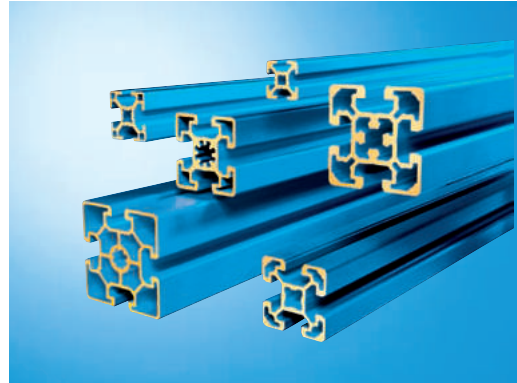
7.1 Bosch Rexroth AG: The Drive & Control Company

7.1.2 Linear motion and assembly technologies

7.1.2.2 Assembly technology

Basic mechanical elements**A solid basis for production**

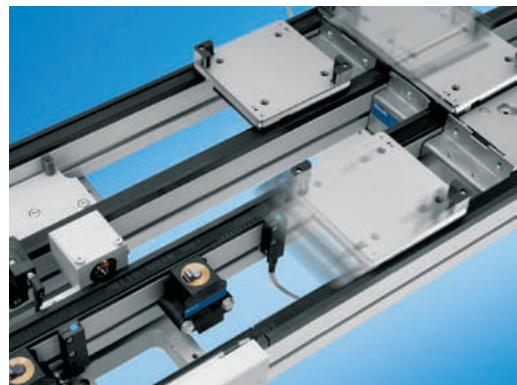
It doesn't matter whether you want to build frames, protective enclosures, machine fixtures, workstations, or an entire production line: Our aluminum profile system, with over 100 profile cross sections and the widest range of accessories worldwide, will give you absolute freedom during construction, while the strong 10 mm T-slot offers maximum security. Quick and easy assembly ensures your economic success.

**Manual production systems****Flexible and economical**

Our manual production systems add efficiency to your production process: with individually adapted, ergonomically designed workstations including material supply, process linking and extensive accessories – available optionally in ESD design. Everything is perfectly matched to each other, and can be combined and configured in minutes with the free planning and calculation software MPScalc.

**Material and information flow technology****Quick and efficient**

Shorter and shorter innovation and product cycle times demand production systems that can quickly adapt to changing demands. Using Rexroth's transfer and identification systems you can make economical, future-oriented solutions a reality, whether you have small, precise, or large and heavy products to manufacture.



7 Appendix

7.1 Bosch Rexroth AG: The Drive & Control Company

7.1.2 Linear motion and assembly technologies

7.1.2.2 Assembly technology

Modular chain conveyor systems

Powerful and cost effective

Whether for linking machine tools, transporting food products or in the packaging industry, VarioFlow and VarioFlow S chain conveyors are fast, reliable and economical transport systems for use in a wide variety of industries. New momentum for your production.



7.2 Glossary

Abrasion: The removal of material by cutting, rubbing or particle impingement.

Actuators: The operative elements in a control circuit. They act as regulators in the control loop, by converting electronic signals (e.g. commands from the control system) into mechanical movement (e.g. of solenoid valves).

Circular-arc profile: One of the possible forms of running tracks in rolling bearing guides and ball screw assemblies. In contrast to the gothic arch (see also: gothic profile) the circular-arc profile comprises a single track per side. This produces a 2-point contact between the running tracks and the rolling element.

Conformity: In rolling contact between balls and raceways, conformity of the surfaces is achieved by giving the raceways a circular shape. Conformity increases the contact area and reduces the surface pressure compared with rolling contact without conformity. This also serves to guide the movement of the rolling element.

Consistency: A measure of the plasticity (or stiffness) of grease lubricants. The consistency is determined in accordance with ISO 2137 using a penetrometer, which measures the depth of penetration of a standard cone dropped into a grease sample. DIN 51818 permits correlation of the cone penetration depth recorded in accordance with ISO 2137 to a consistency class in accordance with the NLGI classification (National Lubricating Grease Institute).

Critical speed: When a shaft rotates (e.g. the screw of a ball screw assembly), bending vibrations occur. The "critical speed" is the rotary speed that is equivalent to the first bending frequency of the shaft. If the shaft is operated at the critical speed, resonance occurs, which can lead to destruction of the system.

Deformation/deflection: The change in shape of a solid body in response to an applied force. A distinction is made between elastic and plastic deformation. The term elastic deformation or elastic deflection is used to describe a case where the body returns to its original shape once the force ceases to be applied, i.e., there is no permanent deformation. Plastic deformation describes a case where there is lasting deformation once the force ceases to be applied.

Dry lubricant: A solid lubricant incorporated into a lacquer-like matrix of organic or inorganic binder. Dry lubricants are also called lubricating varnishes.

Dynamic load capacity C: The load at which a sufficiently large number of apparently identical bearings will achieve the specified nominal life. In the case of ball screw assemblies and rotary anti-friction bearings, the nominal life is 1 million revolutions. The dynamic load capacity of linear motion guides, such as profiled rail systems and linear bushings and shafts, is based on a nominal life of 100 km.

Dynamic load moments M_t and M_L : These are comparative dynamic moments which cause a load equivalent to the dynamic load capacity C. A distinction is made between the dynamic torsional load moment M_t and the dynamic longitudinal load moment M_L .

Friction coefficient: A dimensionless number that represents the relationship between the friction force and the applied load (normal force).

Gothic profile: One of the possible forms of running tracks in rolling bearing guides and ball screw assemblies. Unlike the circular-arc profile, the running track in the gothic version is composed of two running tracks per side. This produces an ogival form, resulting in four-point contact on the rolling element.

Hexapod: A spatial positioning machine with 6 legs of variable length and 6 independently controllable drive components. This enables mobility in all 6 degrees of freedom (3 translatory and 3 rotary). Typical applications for hexapods are spindle guides for machine tools or motion mechanisms for driving and flight simulators.

HRC: This abbreviation, followed by a numerical value, is an indication of hardness as measured using the Rockwell C method. Other methods for measuring the hardness of steel are Rockwell A, B and F, Brinell and Vickers. In all hardness testing methods, a defined body, e.g. a diamond cone in the Rockwell C method, is pressed into the sample with a defined test force. The permanent impression in the sample is then measured and serves as a measure of the sample's hardness.

Incremental encoder: Sensor equipment used to detect changes in the position of linear or rotary components. An incremental encoder can detect travel as well as direction. It contains a track with periodic markings and measures changes in position by scanning the track and counting the number of marks passed. Only relative changes are recorded, not the component's absolute position. When the encoder is switched on, or after a power failure, a homing cycle to a reference mark must therefore be performed in order to determine the component's absolute position. Some incremental measuring systems also have additional features, e.g. distance-coded reference marks, that eliminate the need for a homing cycle. Incremental encoders are used to measure travel, speeds or angles of rotation on machine tools, in handling and automation systems, and in measuring and testing equipment.

7 Appendix

7.2 Glossary

Interchangeability: Precision manufacturing makes it possible to combine components of the same size (e.g. runner blocks and guide rails) irrespective of their design, accuracy or pre-load classes.

Lead: Relating to screws or threaded shafts, the lead is the linear distance traveled per revolution of the screw or shaft. In the case of a single thread (single-start screws), this is the distance between two thread crests or two grooves (running tracks).

Limit switches: Switches used to monitor the end position of moving parts. They emit a signal when the component reaches a certain position, usually the beginning or end of a stroke. The signal can be electrical, pneumatic or mechanical. Typical forms of limit switches with electrical signals are roller lever switches or non-contacting switches such as photoelectric sensors and proximity switches.

Linear motor: Electric motor that produces a linear (translatory) motion instead of rotary motion. The operating principle of a linear motor corresponds to that of a three-phase motor. The excitation windings (stator), arranged in a circle in a three-phase motor, are arranged in a planar configuration in the case of a linear motor. Here, the rotor is pulled along travel path by the linearly moving magnetic field. Linear motors enable direct linear motion and forces to be produced, i.e. without gears. They are therefore also referred to as direct drives.

Pitching: A rotary movement around the transverse axis (Y-axis) and one of the three basic rotational movements of a body in space.

Positioning accuracy: The positioning accuracy is the maximum deviation between the actual position and the target position, as defined in VDI/DGQ 3441.

Reference switch: Switch used to detect the position of a moved component, e.g. the carriage of a linear motion system. The switch emits a signal when the component reaches a defined position (reference mark). Reference switches are required for incremental measuring systems or motors with incremental encoders during start-up and after any interruption to the power supply.

Repeatability: Repeatability indicates how precisely a linear motion system positions itself when approaching a position repeatedly from the same direction (unidirectional motion). It is stated as the deviation between the actual position and the target position.

Resonance: The forced vibration of a vibratory system when subjected to periodic external excitation. If the frequency of the forced vibration is close to the natural frequency of the vibratory system, the amplitude of the vibrations can rapidly become many times larger than that of the external stimulus. If the damping forces in the system are weak, the amplitude will rise uncontrollably, ultimately resulting in destruction of the system.

Reversing mode: In reversing mode, a component, e.g. the carriage of a linear motion system, is moved alternately forwards and backwards.

Rolling: A rotary movement around the longitudinal axis (X-axis) and one of the three basic rotational movements of a body in space.

Short stroke: Short-stroke applications are applications in which not all of the rolling elements recirculating within the bearing component arrive in the load-bearing zone during execution of the stroke. The precise definition differs from product to product. The consequences can be premature material fatigue, leading to failure of the guide units. Short-stroke applications must be taken into account when calculating the life expectancy.

Solid lubricant: A substance that alone or in combination with other substances forms an uninterrupted sliding and separating film on metal surfaces. These films are so thin that fits and tolerances do not have to be altered. Solid lubricants are mostly only required and used for lubrication tasks under extreme conditions (e.g. when operating in the mixed friction range). The most well-known are graphite, molybdenum disulfide, various plastics, (e. g. PTFE) and heavy metal sulfide.

Static load capacity C_0 : The static load which results in a permanent overall deformation of the rolling element and the raceway corresponding to approximately 0.0001 times the rolling element diameter. Deformations of this order have no noticeable effect on the smoothness of travel.

Stick-slip effect: Term used to describe the phenomenon of backsliding between solid bodies being moved against each other, which can occur at low sliding speeds. A rapid motion sequence takes place as a result of adhesion, jamming, separation and sliding. This leads to vibrations, which can create noises, such as the squealing of a railcar's wheels when traveling round a curve in the track. The problem can be remedied by increasing the sliding speed, lubrication or appropriate selection of materials.

Transmission/transmission ratio: The transmission and conversion of movements, linear and rotary speeds, forces and torques in a geared mechanism. The transmission ratio (also known as reduction ratio) is the ratio between the drive variable and the output variable, e.g. the ratio of input speed to output speed.

7 Appendix

7.2 Glossary

Viscosity: A measure of a liquid's resistance to flow when subjected to shear stresses. This resistance is due to the liquid's internal friction.

Worked penetration: Penetration is a method used to measure the consistency of lubricants. In grease lubricants, a cone of defined configuration is allowed to penetrate vertically into the test sample under prescribed conditions (ISO 2137), and the depth of penetration is then measured. A distinction is made between unworked penetration and worked penetration. Worked penetration is the penetration of the cone immediately after the sample has been worked in a container fitted with a plunger by stroking the plunger 60 times within one minute at a temperature of +25 °C.

Yawing: A rotary movement around the vertical axis (Z-axis) and one of the three basic rotational movements of a body in space.

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