SCHAEFFLER

Rolling bearings in traction motors

Deep groove ball bearings, four-point contact bearings, cylindrical roller bearings

Technical Product Information

1 Foreword

Schaeffler is a leading manufacturer of rolling bearings and plain bearings. In close partnership with manufacturers and operators, carefully matched solutions are developed for any application in rail vehicles. The product portfolio includes wheelset bearings including housings as well as bearings and components for traction motors and gearboxes, for wagon joints and tilting mechanisms, for doors and sliding panels.

Bearing arrangements for rail vehicles are subjected to extreme loads and are required to fulfil high safety standards. Schaeffler has over 140 Jahre of experience in the railway sector and therefore can offer a comprehensive range of technical expertise, top quality and products that are perfectly tailored to any application. The reliability of the bearings is tested under extreme conditions on test rigs we have developed in-house. The railway test shop at the FAG Schweinfurt site is recognised and certified as a test facility for rail vehicle bearing arrangements by the Federal German Railway Authority.

It is additionally authorised by the DAP (Deutsches Akkreditierungssystem Prüfwesen GmbH) to carry out tests on the performance capabilities of wheelset bearings for railway applications in accordance with DIN EN ISO/IEC 17025:2000.

Our services include expert application advisory work, rolling bearing calculations, testing and assembly. We have a close-meshed network of external sales engineers, service and sales technicians working worldwide for you to ensure short travel distances and rapid response times. As a special service, Schaeffler offers professional and cost-efficient reconditioning of railway bearings.

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2 Factors influencing the design of bearing arrangements

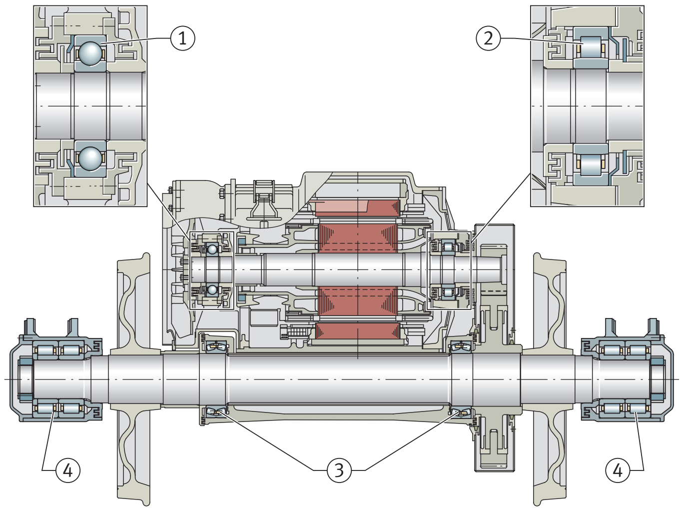
2.1 Influencing factors

Modern electric drives for the railway industry are subject to certain fundamental requirements such as cost-efficiency, reliability, operational reliability and safety, adequate service life and low maintenance outlay. The fulfilment of these fundamental requirements is heavily influenced by the acting forces, additional loads and environmental influences. Because rolling bearings are wear parts of the drive, all industry-specific requirements must be considered and fulfilled. The bearing arrangement on the electric drive rotor shaft must be carefully adapted to the specific operating and environmental conditions. Different load spectra must be taken into account for the different drive concepts to yield efficient, operationally reliable and safe, and economical bearing arrangements. The lubrication and sealing must be configured such that the bearings are neither undersupplied nor oversupplied with lubricant in any operating status. ➤7|1.

For the bearing arrangement for an electric drive, the design engineer must consider the following influencing factors:

- load
- speed
- current flow, continuity
- installation space
- shaft arrangement
- vibration behaviour
- temperature influence
- rating life requirement
- structural safety
- environmental conditions
- friction
- lubrication
- maintenance
- mounting and dismounting

1 Traction motor bearing arrangement in an electric multiple unit



1	Bearing, non-drive side	2	Bearing, drive side
3	Nose-suspended bearing, here: spherical roller bearing	4	Wheelset bearing

2.2 Design of traction motor bearing arrangements

The computer-aided design of electric motor bearing arrangements is generally carried out using the material fatigue theory. For bearings in standard motors and series motors, the operating life is usually identical to the grease (operating) life because, depending on the axle height, many such motors may have sealed bearings with for-life lubrication. Here, the grease life determines the rating life of the bearing arrangement.

For traction motors, the bearing sizing is determined in the same way as for standard motors with the help of the usual fatigue life calculation. In the industry, fatigue life is normally given not in hours, but in kilometres travelled. The rating life of the bearings in a traction motor is set by the fatigue life of the bearings, as they are usually relubricated.

In the design of a traction motor bearing arrangement, the usually specified target maintenance interval must be considered alongside the calculated rating life. Because the calculated rating life should be as long as possible, but the bearing size should be as small as possible to allow for higher speeds and longer lubrication intervals, the design usually involves a compromise.

In the design of the traction motor bearing arrangement, special attention must be paid to the following:

- speed capability of bearings and grease
- minimum load capacity of bearings
- axial load capacity for use of cylindrical roller bearings as locating bearings
- thermal stability and vibration resistance of grease
- selection of a suitable cage guidance type and cage design

The correct fit and selection of the correct internal clearance are also extremely important for long-lasting, reliable operation.

3 Bearing types

The following bearings are used in the traction motors for supporting the electric drives:

- Deep groove ball bearings
- Four-point bearings as a special solution or project-specific requirement
- Cylindrical roller bearings

All bearings with basic designs have normal tolerances (PN). In addition, bearings with tighter tolerances of P6 and P5 can be supplied. The bearing tolerance is indicated in the bearing designation by the corresponding suffix.

The internal clearance of the bearing is selected according to the specifications and in most cases lies in the range C4 (Group 4) or C5 (Group 5). If a special radial internal clearance is implemented for a specific project, it is indicated separately after the bearing designation.

The axial internal clearance for designs NUP, NJ and HJ is also regulated by the corresponding standards and provides for a greater axial bearing clearance.

The cage design depends on the size and application. Single-piece brass or bronze solid cages (rolling element-guided) are preferred for cylindrical roller bearings. Two-piece riveted brass cages that are usually guided by the rolling elements are used for ball bearings.

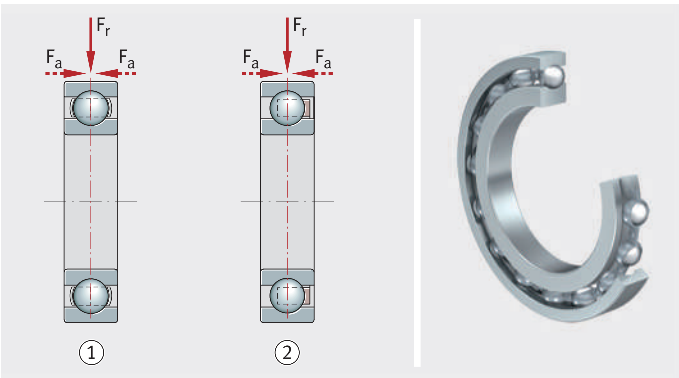
Schaeffler offers suitable electrical insulation solutions to avoid damage due to electric currents.

The suffix F1 used with a cylindrical roller bearing is the standard identification for a traction motor bearing.

3.1 Deep groove ball bearings

Single-row deep groove ball bearings support radial and axial forces and are suitable for high speeds ➤8|2. Deep groove ball bearings cannot be disassembled. Because of its versatility and its favourable price/performance ratio, the deep groove ball bearing is the most commonly used bearing type.

2 Deep groove ball bearings



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1	Bearing with sheet metal cage	2	Bearing with plastic cage

F_a	N	Axial load
F_r	N	Radial load

Deep groove ball bearings in traction motors usually have normal tolerances and Group 4 or Group 5 radial internal clearances. The nominal contact angle α_0 is 0°. For axial loading and an increased internal clearance, the contact angle can increase to 20°. The support of the elliptical contact surface must therefore always be checked.

The speed capability is high to very high.

3.1.1 Cage

The rolling element-guided two-piece brass cage is usually used for electric drives. Due to their low maximum service temperatures, which for TVP2 standard cages is +120 °C, plastic cages are rarely used.

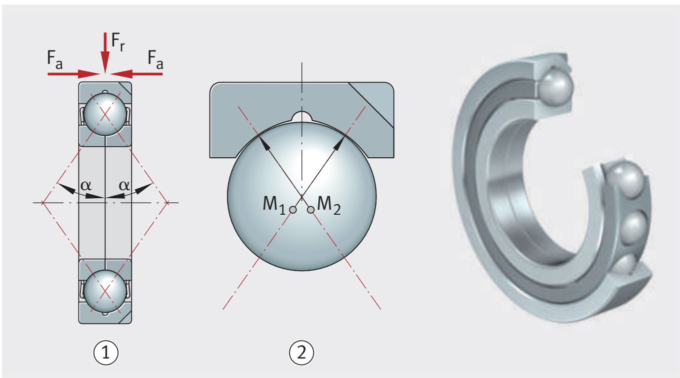
3.1.2 Heat treatment

Deep groove ball bearings for traction motors are made to be dimensionally stable up to +150 °C through a heat treatment; this is not additionally indicated by a suffix. Further dimensional stabilisation measures are possible, but they must be explicitly ordered. Dimensional stabilisation measures are indicated by suffixes.

3.2 Four-point contact bearings

Four-point contact bearings are single-row angular contact ball bearings that support axial forces in both directions and low radial forces ➤9|3. The four-point contact bearing has a split inner ring to accommodate a large number of balls. The high load-carrying capacity in the axial direction is achieved through the large number of balls, the high raceway shoulders and the contact angle of 35°.

3 Four-point contact bearings



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1	Four-point contact bearing with retaining slot and split inner ring	2	Raceway geometry

F_a	N	Axial load
F_r	N	Radial load
M_1, M_2	–	Centres of curvature of outer ring raceway
α	°	Contact angle

Since the centres of curvature of the arc-shaped raceways on the inner and outer rings are offset relative to each other, however, the balls are in contact with the bearing rings at four points under radial load. Therefore, four-point contact bearings should only be used with predominantly axial loads.

In order to avoid four-point contact during operation, four-point contact bearings are used to support the axial loads and are thus unsupported on the outer ring.

For reliable operation, a constantly acting axial force is required.

3.2.1 Cage

Four-point contact bearings in traction motors are usually oil-lubricated and have an outer rib-guided single-piece brass cage. Grease lubrication is usually unsuitable for this cage type.

3.2.2 Heat treatment

The four-point contact bearings for traction motors are heat-treated such that they are dimensionally stable up to +150 °C.

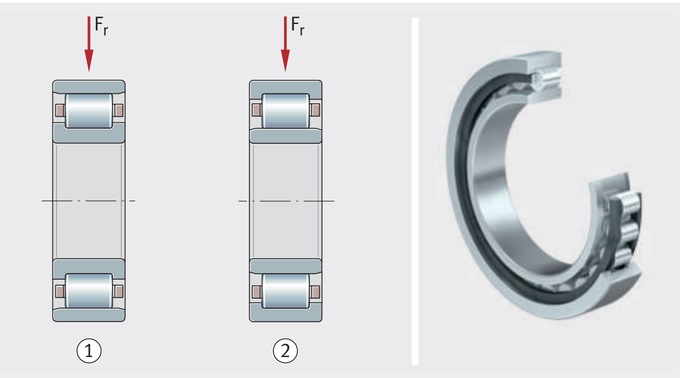
3.3 Cylindrical roller bearings

Cylindrical roller bearings have a high radial load-carrying capacity. In traction motors, they are mainly used on the drive side.

3.3.1 Non-locating bearings

The different single-row cylindrical roller bearing designs differ in terms of the arrangement of the ribs ➤10 | 4. The NU design has two ribs on the outer ring and a ribless inner ring. The N design has an inner ring with two ribs and a ribless outer ring. Cylindrical roller bearings with the NU and N designs are used as non-locating bearings; they allow length compensation within the bearings.

4 Non-locating bearings



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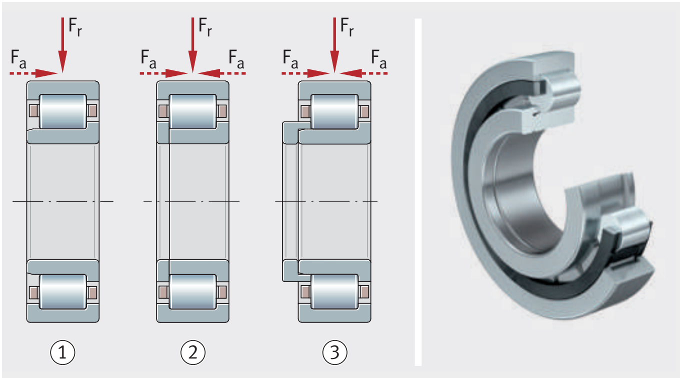
1	Cylindrical roller bearing N	2	Cylindrical roller bearing NU
F_r	N	Radial load	

3.3.2 Semi-locating bearings and locating bearings

NJ cylindrical roller bearings each have two ribs on the outer ring and one rib on the inner ring and are semi-locating bearings ➤11 | 5. They can support not only high radial forces but also axial forces in one direction and can therefore guide shafts axially in one direction. In the opposite direction, they act as non-locating bearings.

NUP cylindrical roller bearings are installed as locating bearings to support alternating axial forces. This bearing has two ribs on the outer ring, a rigid rib on the inner ring and a loose rib washer. A cylindrical roller bearing NJ with an L-section ring HJ also forms a locating bearing. Ribs subjected to load must be supported across their entire height.

5 Semi-locating bearings and locating bearings



1	Cylindrical roller bearing NJ (semi-locating bearing)	2	Cylindrical roller bearing NUP with rib washer (locating bearing)
3	Cylindrical roller bearing NJ with L-section ring HJ (locating bearing)		

F_a	N	Axial load
F_r	N	Radial load

For traction motors of types NUP or NJ and HJ (NH), a special axial internal clearance is used.

3.3.3 Tilting

The modified line contact between rollers and raceways prevents edge stresses and allows for a certain amount of angle adjustment in the single-row cylindrical roller bearings.

The following values should not be exceeded:

- 4' for 10, 19, 2, 3 and 4 series bearings
- 3' for 22 and 23 series bearings

3.3.4 Cage

Solid brass or bronze cages are recommended for traction motors. The guidance method depends on the bearing type and the type of lubrication.

3.3.5 Heat treatment

The cylindrical roller bearings for traction motors are standardly dimensionally stable up to +150 °C.

3

3.3.6 Axial load-carrying capacity

The axial load-carrying capacity depends on the following factors:

- the size of the sliding surfaces between the ribs and the end faces of the rolling elements
- the sliding velocity at the ribs
- lubrication of the contact surfaces
- bearing tilt

! Ribs subjected to load must be supported across their entire height.

The permissible axial load $F_{a\ per}$ must not be exceeded. This will prevent heating to impermissibly high temperatures.

The axial load limit $F_{a\ max}$ must not be exceeded. This will prevent impermissible pressures on the contact surfaces.

The ratio F_a/F_r should not exceed 0,4. Continuous axial loading without simultaneous radial loading is not permissible.

ƒ1 Permissible axial load

$$F_{a\ per} = k_S \cdot k_B \cdot d_M^{1,5} \cdot n^{-0,6} \leq F_{a\ max}$$

D	mm	Outside diameter
d_M	mm	Mean bearing diameter (d+D)/2
$F_{a\ per}$	N	Permissible axial load
$F_{a\ max}$	N	Maximum axial load
k_B	–	Factor dependent on the bearing series
k_S	–	Factor dependent on the lubrication method
n	min^{-1}	Operating speed (nominal speed)

ƒ2 Maximum axial load

$$F_{a\ max} = 0,075 \cdot k_B \cdot d_M^{2,1}$$

d_M	mm	Mean bearing diameter (d+D)/2
$F_{a\ max}$	N	Maximum axial load
k_B	–	Factor dependent on the bearing series

1 Factor k_S for the lubrication method

Lubrication methods [1]	**Factor k_S**
Minimum heat dissipation, drip feed oil lubrication, oil mist lubrication, low operating viscosity ($\nu < 0,5 \cdot \nu_1$)	7,5 ... 10
Poor heat dissipation, oil sump lubrication, oil spray lubrication, low oil flow	10 ... 15
Good heat dissipation, recirculating oil lubrication (pressurised oil lubrication)	12 ... 18
Very good heat dissipation, recirculating oil lubrication with oil cooling, high operating viscosity ($\nu > 2 \cdot \nu_1$)	16 ... 24

[1] Doped oils should be used: e.g. CLP (DIN 51517:2018) and HLP (DIN 51524) of ISO-VG classes 32 to 460 and ATF oils (DIN 51502:1990) and gearbox oils (DIN 5112:2016) of SAE viscosity classes 75 W to 140 W.

2 Bearing factor k_B for series

Series	Bearing factor k_B
NJ2..-E, NJ22..-E, NUP2..-E, NUP22..-E	15
NJ3..-E, NJ23..-E, NUP3..-E, NUP23..-E	20
NJ4	22

Misalignment of the bearing, for example due to shaft deflection, can lead to alternating loads on the inner ring ribs. In this, case the axial load must be restricted, for a bearing tilt of up to 2 arcmin, to F_{as}.

3 Permissible axial load with misalignment F_{as}

$$F_{as} = 20 \cdot d_M^{\;1,42}$$

d	mm	Inside diameter
D	mm	Outside diameter
d_M	mm	Mean bearing diameter (d+D)/2
F_{as}	N	Permissible axial load with misalignment of max. 2 arcmin

4

4 Cage

The rolling bearings in the unsuspended or only slightly suspended traction drives are subjected to highly dynamic additional loads. These additional loads especially act on the cage. They are very complex and difficult to define, so the rolling bearings in highly loaded traction drives are usually equipped with solid brass or bronze cages.

The most important functions of the cage are as follows:

- to separate the rolling elements from each other in order to keep friction and heat generation as low as possible
- to maintain the rolling elements at the same distance from each other in order to ensure uniform load distribution
- in bearings that can be swung out and disassembled, to prevent the rolling elements from falling out
- to guide the rolling elements in the unloaded zone of the bearing

4.1 Loads

The cage is driven by the circulating rolling element set. Sliding motions occur on the guidance surfaces, which are above all the cage pockets. The guidance forces result from the mass of the cage, possible displacements of the centre of gravity, and accelerations and decelerations between the rolling elements and cage. They are normally low. The cage is not involved in the transfer of external forces from one bearing ring to the other via the rolling elements.

Due to the sliding friction that occurs, the lubrication regime at the guidance surfaces of the cage is particularly important in relation to wear and operating life. In most cases, and especially when grease lubrication is used, it is almost impossible to prevent a mixed friction regime. The wear occurring as a result is, however, negligibly small in terms of its practical effect on the operating life under the forces occurring in normal cases.

4.2 Cage types

Rolling bearing cages are subdivided into sheet metal and solid section cages. The design of the solid cage is particularly important in heavily loaded cylindrical roller bearings.

6 Tractor motor bearing with the current two-piece M1 brass cage

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In a cage with crosspiece rivets, rivet pins are created on the cage crosspieces. The crosspiece riveting makes it possible to achieve a smaller spacing of the rollers. This means that more rollers can be accommodated in the bearing than in a cage with normal riveting.

Due to the increasing dynamic loads on bearings in traction motors, Schaeffler recommends the use of single-piece solid brass cages for cylindrical roller bearings in the future. These window cages have the highest stiffness (shape retention) and are therefore preferred for use in highly loaded bearings. A newly developed series cage with the suffix MP is standardly available for many cylindrical roller bearings.

7 Traction motor bearing with single-piece MP cage

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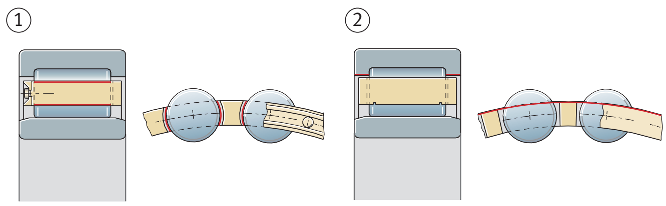
4.2.1 Guidance types and characteristics

In cage guidance, a distinction is made between rolling element guidance and rib guidance ➤15|8.

8 Guidance of cages



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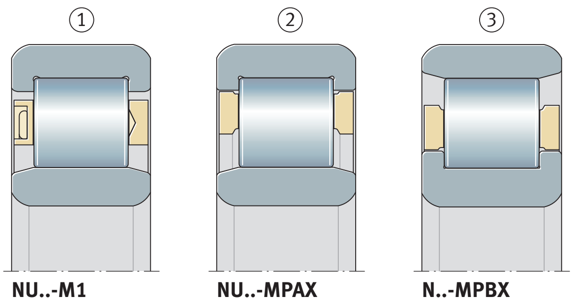
1	Rolling element guidance	2	Rib guidance

In traction motors, the cage is usually directly guided by rolling elements and the bearings are usually greased ➤16|9. As experience has shown, rib guidance with grease lubrication entails a risk of lubricant starvation on the guidance surfaces. There are exceptions in other areas of application, but these applications require more relubrication. Rib-guided cages are used when the bearings are subjected to strong vibrations or extremely high speeds. Experience has shown that this type of guidance requires oil lubrication of the rolling bearing.

9 Cage guidance types



1	NU220-M1 (example)	2	NU220-MPAX (example)
3	N220-MPBX (example)		

4.3 Cage damage

Cage damage and unsatisfactorily short bearing running times due to cage wear are often explained by a cage design that is unsuitable for the respective operating conditions. The superimposition of primary and secondary effects makes it difficult in many cases to identify or at least narrow down the sources of the problems. The assessment and evaluation of cage damage or premature cage wear is also hindered by the fact that the effects of some problems may be diminished and thereby delayed in time through the use of solid cages. In this case, only the secondary effect is then combated and the actual root cause remains unidentified.

The most common causes of problems are as follows:

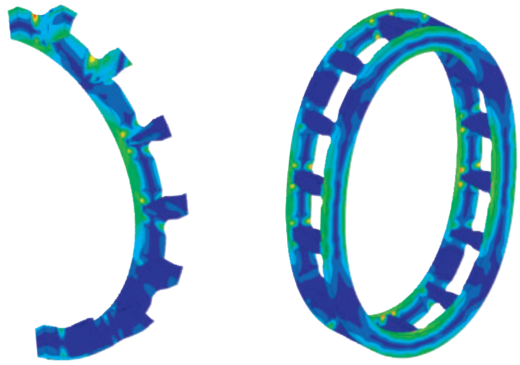
- disrupted running conditions caused by misalignment of bearing rings relative to each other
- non-uniform running of the cage, resulting in continuous acceleration and deceleration forces due to insufficient contact between the rolling partners due to insufficient loading
- impeded rolling element and cage running due, for example, to the effects of foreign bodies, an unsuitable or impermissible (hardened) lubricant or possibly overlubrication
- unexpected vibrations and shocks
- disrupted running due to high axial forces or radial stress resulting from insufficient operating clearance
- any type of lubrication problem
- other effects arising from the drive or from the drive control system

4.4 Cages for high shock and oscillation loads

In freight locomotives, the cages in cylindrical roller bearings are predominantly subjected to very high shock and vibration loads. These additional loads are very difficult to identify, but lead to suboptimal design if insufficient knowledge of them has been obtained. Possible consequences are cage breakage and premature drive failure.

The additional forces transmitted directly from the rail via the wheel to the rotor shaft subject the cage through deformation to additional bending alternating load and through shock to additional forces that are then transmitted directly via the rolling elements to the cage crosspiece. The additional load due to shock and vibration loading is shown as an example in ➤17 | 10.

10 Loads on the cage



To counteract these special additional loads, special cages with an optimised and reinforced design are used. Using special materials to further enhance the operational reliability and safety is an absolute priority for such cages. Differentiation is made between alloys produced by centrifugal casting and by hot rolling. Purity and pore size in particular are important quality criteria.

Characteristics of a special cage:

- optimised cage pocket clearance
- optimised contact surface (pocket geometry)
- suitable cage guidance type
- reinforced design (cage design)
- increased stiffness
- reduction of notch influences

5

5 Operating parameters

5.1 Minimum load on radial bearings

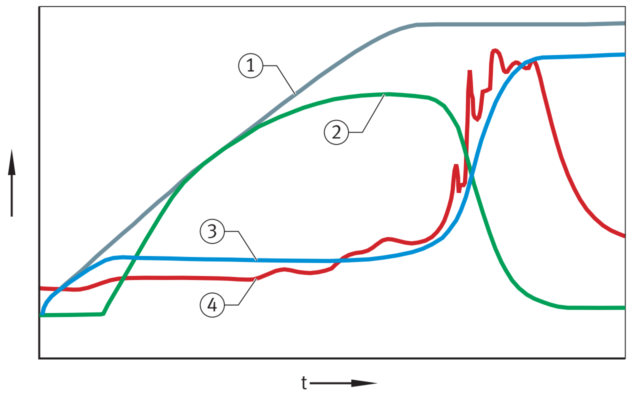
In certain speed and load ranges, slip occurs between the rolling elements and raceways, i.e. the rolling contact partners move at different speeds relative to each other. The primary cause is an insufficient bearing load. Due to the lack of friction locking between the inner ring raceway and the rolling elements, pure rolling motion is not present. The rolling element and cage speeds are lower than in the case of kinematically ideal rolling. In the industry, slip is frequently referred to as cage squealing. It presents itself acoustically as a high-pitched clinking sound and can be diagnosed relatively easily. Relubrication usually makes the slipping noise go away, but only for a short time.

The slip itself is not that critical, but slippage breakdown is. Scoring of the raceway surfaces and high cage loads are consequences of slippage breakdown. This is understood to mean an abrupt change in the rotational speed of the cage from the slip phase to kinematically ideal rolling conditions. The transition from hydrodynamic lubrication to mixed friction with metal-to-metal contact in the contact area between the rolling elements and the raceway is responsible for this short-term change in speed of the cage and the rolling elements.

Slip investigations on the test rig

An unloaded cylindrical roller bearing is brought up to the test speed. With kinematically ideal rolling, the cage and the shaft run at synchronised speeds. If they do not, then slip occurs. When slip breakdown occurs, the noise level rises sharply. The slip noise is not a result of scoring of the raceway surfaces, but is rather the result of friction-induced vibration.

11 Slip investigations on the test rig



1	Shaft speed	2	Cage slip
3	Cage speed	4	Structure-borne noise development

The risk of such slip is particularly high for bearings under low loads. In these cases, an increase in the load is recommended. If the load cannot be increased, smaller bearings with lower dynamic load ratings or capacities C_{dyn} should be installed if possible.

3 Minimum load on radial bearings

Bearing type	Minimum load
Cylindrical roller bearings	$P > C_{0r} / 60$
Deep groove ball bearings	$P > C_{0r} / 100$
Four-point contact bearings	$F_a \geq 1{,}2 \cdot F_r$

5.2 Speed parameters of standard bearings

The permissible speed is dependent on the design of the entire bearing arrangement, in other words the rolling bearing itself (type, size, accuracy and cage design), as well as on the operating conditions ➤20 | 12.

The operating conditions are as follows:

- magnitude and direction of the load
- lubrication method, type and amount of lubricant
- environment
- design and tolerances of adjacent parts
- heat dissipation through adjacent parts

The highest permissible speed for a rolling bearing in operation can be restricted by various criteria. The decisive factor is usually the operating temperature, which rises with increasing speed.

The speed limits stated in the catalogues are guide values indicating the speeds that can still be reliably achieved with bearings of normal design and tolerances given good installation conditions and a normal load ratio. The speed achievable in practice depends greatly on the factors described above.

In connection with this, the characteristic speed value is often used as a design parameter for rolling bearings. The characteristic speed value is the product of the speed n and the mean bearing diameter d_M.

4 Characteristic speed

$$n \cdot d_M$$

d_M	mm	Mean bearing diameter (d+D)/2
n	min^{-1}	Operating speed or equivalent speed

5 Mean bearing diameter

$$d_M = \frac{D + d}{2}$$

d	mm	Inside diameter
D	mm	Outside diameter
d_M	mm	Mean bearing diameter (d+D)/2

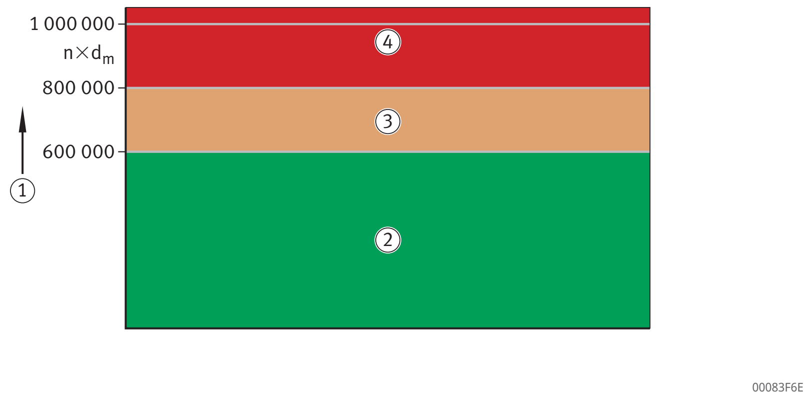
With grease lubrication, maximum speeds with an order of magnitude of $1 \cdot 10^6$ $min^{-1} \cdot mm$ for radial ball bearings and radial cylindrical roller bearings can be achieved. Oil lubrication is required for higher speeds.

Standard deep groove ball bearings or cage-guided cylindrical roller bearings can be operated in the green zone without any reservations. For higher characteristic speeds (yellow or red zone), the following parameters must be investigated or adjusted:

- radial internal clearance
- accuracy class
- rolling elements (size, material)

- machining tolerances of adjacent parts
- cage (material, type and guidance)
- lubrication (grease or oil)
- lubrication method for oil lubrication (circulating oil, oil mist or oil injection)

12 Schematic of reference speed ranges and limits for deep groove ball bearings and cylindrical roller bearings



1	Characteristic speed	2	Standard range
3	Extended range	4	HighSpeed range

The actual limiting speeds for the individual bearings are indicated in the product tables in the catalogue HR 1, Rolling Bearings.

5.3 Thermal stabilisation and retained austenite

The dimensions of the rolling bearings at a specific operating temperature must remain unchanged even after passing through the entire operating temperature range. The dimensional instability of hardened but improperly post-treated rolling bearing steel would cause significant dimensional changes in a short time at a high bearing operating temperature.

Negative consequences of the dimensional changes would be as follows:

- changes in bearing clearance
- loosening of shrink-fit connections
- premature rolling bearing failure

Causes of dimensional changes are related to changes in the microstructural phases retained austenite and martensite. In hardened rolling bearing steel, not only the influence of temperature and time but also the influence of the load lead, on one hand, to an increase in volume due to the transformation of retained austenite and, on the other hand, to a decrease in volume due to the carbon precipitation in the martensite. The overall change in dimensions results from the superimposition of the two subprocesses.

For this reason, dimensionally stable rolling bearings require a differentiated tempering treatment following hardening in the production process. Tempering forestalls the decomposition of residual austenite and the precipitation of carbon in the martensite.

The designations for the dimensional stability levels and the operating temperatures are regulated in DIN 623-1:2020-06. The associated treatments are left to the rolling bearing manufacturer.

Rolling bearings for traction motors are standardly heat-treated such that they can be used up to an operating temperature of +150 °C and for some types up to +200 °C. At operating temperatures of +120 °C and higher, rolling bearings require a special heat treatment. The different suffixes and the associated maximum operating temperatures are shown in ➤21 | ▦4.

▦4 Operating temperature and suffixes for dimensionally stabilised bearings

Maximum operating temperature °C	**Suffix for dimensionally stabilised bearings**
+120	SN
+150	S0
+200	S1
+250	S2
+300	S3

ƒ6

$$\Delta d = D \cdot \Delta RA \cdot 0{,}08 \frac{\mu m}{mm \cdot \%RA}$$

Δd	–	Dimensional change
D	mm	Outside diameter
ΔRA	%RA	Retained austenite content

5

6

6 Calculation

Certain points must be considered in the selection of the bearing type, in the bearing design and with regard to the surrounding structure.

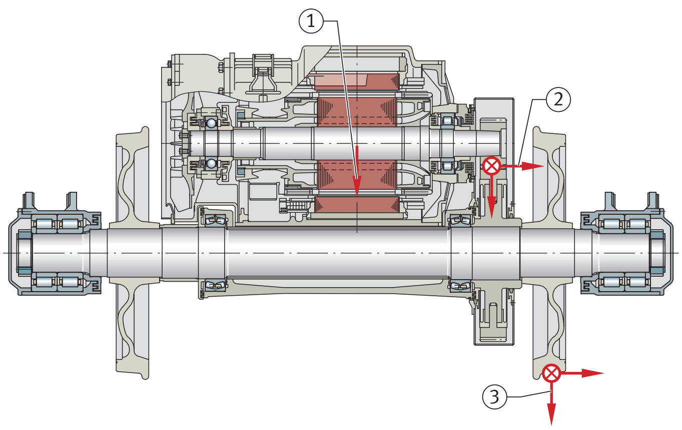
Tested standard bearings are primarily used in traction motors, especially for use in railway applications. If special or additional loads are to be expected, special solutions are preferred.

The most frequently used bearing types have already been discussed in detail in the "Bearing types" section. The bearing arrangement concept used most frequently is the arrangement comprising a deep groove ball bearing as a locating bearing and a cylindrical roller bearing as a non-locating bearing ➤22 | 13. Unlike in stationary standard motors, the two bearings on the inner and outer rings are mounted with a tight fit on the shaft and in the housing due to the increased vibration load. The displacement function is ensured with the help of the cylindrical roller bearing. Particular attention must be paid to the determination of the radial internal clearance. The installed bearing at the steady-state operating temperature should have a minimal radial internal clearance. Examples of typical loads to be considered during the design of the traction motor bearing arrangement are shown in ➤22 | 13.

The loads include the following:

- loads at the rotor's centre of gravity
- loads from the coupling or gearing or from the cardan shaft drive
- additional loads from shocks and vibrations

13 Classic bearing arrangement on a traction motor with loads

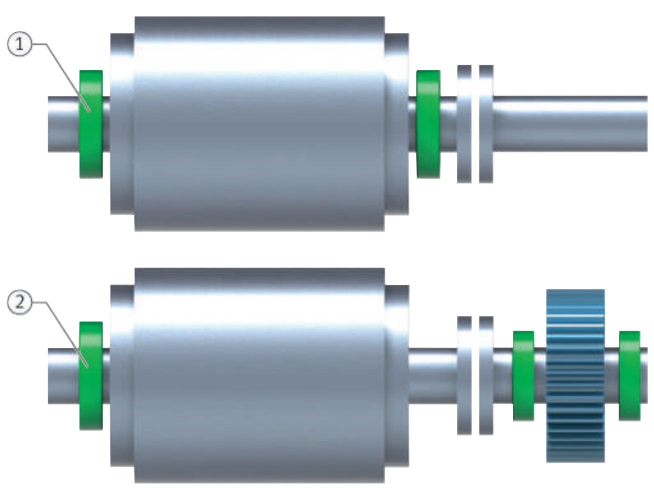


1	Rotor forces	2	Forces from gearing
3	Additional loads from shocks and vibrations		

6.1 Bearing arrangement and drive concepts

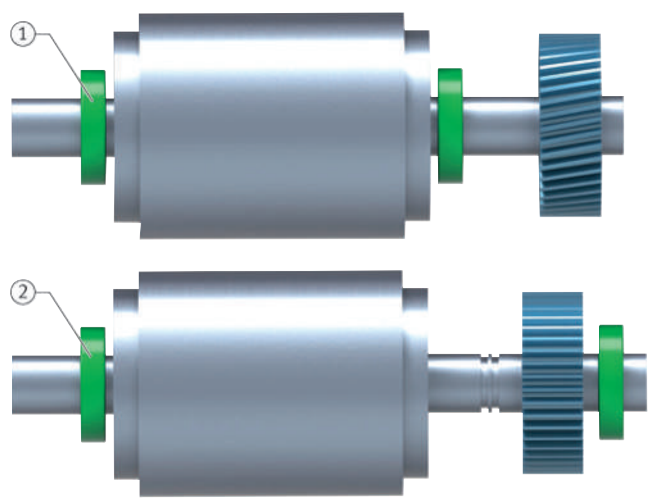
Representations of the bearing arrangement and drive concepts generally used in traction motors are provided in ➤23 | 14 ➤23 | 15.

14 Coupling drive



1	Bearing, both sides	2	Bearing, one side

15 Gearbox drive, direct drive with a pinion



1	Bearing, both sides	2	Bearing, one side

6.2 Basic rating life

The standardised method for calculating the rating life in accordance with ISO 281 for dynamically loaded rolling bearings is based on material fatigue (pitting) as the cause of failure.

7 Basic rating life in millions of revolutions

$$L_{10} = \left(\frac{C}{P}\right)^{p}$$

C	N	Basic dynamic load rating
L_{10}	10^6	Basic rating life in millions of revolutions
p	–	Life exponent • Roller bearings: p = 10/3 • Ball bearings p = 3
P	N	Equivalent dynamic bearing load

6

ƒ8 Basic rating life in operating hours

$$L_{10h} = \frac{16666}{n} \cdot \left(\frac{C}{P}\right)^p$$

C	N	Basic dynamic load rating
L_{10h}	h	Basic rating life in operating hours
n_m	min^{-1}	Mean speed
p	–	Life exponent • Roller bearings: p = 10/3 • Ball bearings p = 3
P	N	Equivalent dynamic bearing load

ƒ9 Basic rating life in kilometres

$$L_{10\ km} = L_{10} \cdot \frac{\pi \cdot D_R}{i} \cdot 10^3$$

D_R	m	Wheel diameter
L_{10}	10^6	Basic rating life in millions of revolutions
i	–	Transmission ratio

6.3 General equations for calculations and aids

ƒ10 Mean speed

$$n_m = n_1 \cdot \frac{q_1}{100} + n_2 \cdot \frac{q_2}{100} + \ldots$$

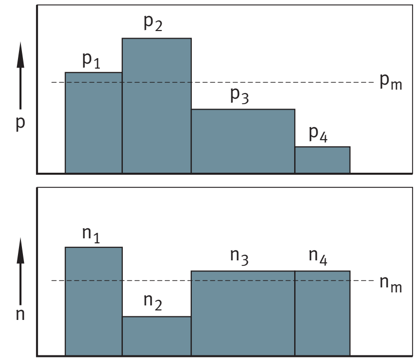
n	min^{-1}	Speed
n_m	min^{-1}	Mean speed
q	%	Proportion

ƒ11 Variable load and speed

$$p = \sqrt[3]{p_1^3 \cdot \frac{n_1}{n_m} \cdot \frac{q_1}{100} + p_2^3 \cdot \frac{n_2}{n_m} \cdot \frac{q_2}{100} + \ldots}$$

n	min^{-1}	Speed
n_m	min^{-1}	Mean speed
p	N	Variable load
q	%	Proportion

16 Load and speed



p	Load	n	Speed

12 Variable load

$$p = \sqrt[3]{p_1^3 \cdot \frac{q_1}{100} + p_2^3 \cdot \frac{q_2}{100} + \ldots}$$

p	N	Variable load
q	%	Proportion

13 Power

$$P = \frac{M \cdot n}{9550}$$

n	min^{-1}	Speed
P	kW	Power

14 Train speed

$$v = \frac{n \cdot D_R}{5{,}3 \cdot i}$$

D_R	m	Wheel diameter
i	–	Transmission ratio
n	min^{-1}	Speed
v	km/h	Train speed

15 Transmission ratio

$$i = \frac{n_1}{n_2} = \frac{z_2}{z_1} = \frac{r_2}{r_1}$$

i	–	Transmission ratio
n	min^{-1}	Speed
r	mm	Pitch circle radius
z	–	Number of teeth on gear wheel

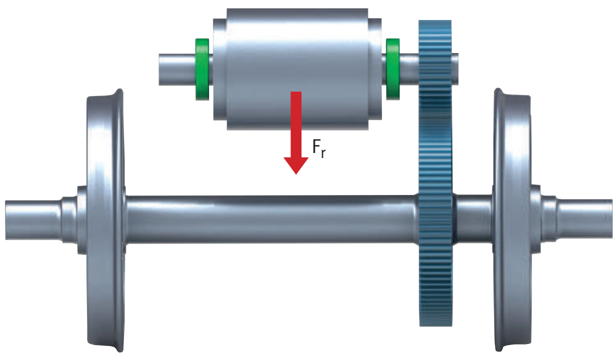
Index 1 indicates the driving wheel and index 2 indicates the driven wheel.

6

6.4 Loads at the rotor's centre of gravity

The bearing forces in electric machines are calculated from the rotor mass ➤26 | 17. In this case, the safety factor f_z includes any imbalance and the magnetic pull. For electric traction drives, this factor was also used as a shock safety factor and was used in accordance with the suspension type. In theory and in the calculation, a safety factor f_z = 1,5 to 2,5 can be applied. Shock loads are especially considered.

17 Forces from the rotor mass



16 Radial force of the rotor with F_B

$$F_r = F_g + F_B$$

F_B	N	Force due to magnetic pull and imbalance
F_g	N	Gravitational force
F_r	N	Radial force

17 Radial force of rotor with f_z

$$F_r = F_g + f_z$$

F_g	N	Gravitational force
F_r	N	Radial force
f_z	N	Total safety factor f_z= 1,5 ... 2,5

18 Gravitational force of rotor

$$F_g = m_R \cdot g$$

F_g	N	Gravitational force
g	m/s^2	Acceleration due to gravity g = 9,81 m/s^2
m_R	kg	Rotor mass

6.5 Loads from the drive concepts

There are three established drive concepts for traction motors.

These include the classic coupling connections and the spur and helical gearing concepts as well as the cardan drive that is flange-mounted directly on the traction motor. In all three drive concepts, additional forces that affect the bearing are generated. These forces must be taken into account in the design.

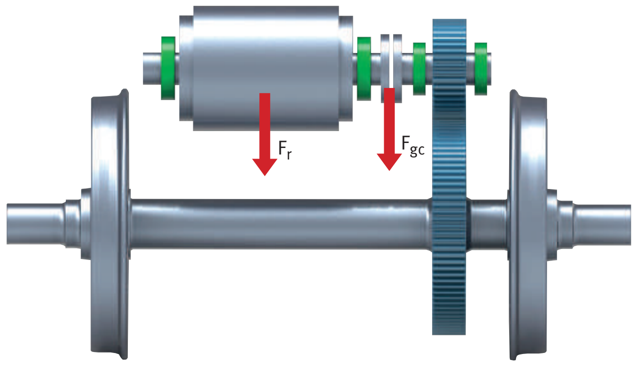
If you have any questions about the cardan drive, please contact the Application Technology department at Schaeffler.

6.5.1 Loads due to the coupling

Elastic or adjustable couplings are the most commonly used connecting elements between the traction motor and the gearbox. The coupling output generally represents a clearly defined load composed of an additional radial as well as an axial force component. A bending load must often also be taken into account.

Half the weight of the coupling is usually applied as the coupling force. The possible axial force component must be requested from the coupling or traction motor manufacturer ➤27 | 18.

18 Additional forces due to the coupling



19 Coupling radial force

$$F_{rC} = \frac{1}{2} \cdot F_{gC}$$

F_{gC}	N	Gravitational force of coupling
F_{rC}	N	Coupling radial force

20 Gravitational force of coupling

$$F_{gC} = m_C \cdot g$$

F_{gC}	N	Gravitational force of coupling
g	m/s^2	Acceleration due to gravity g = 9,81 m/s^2
m_C	kg	Coupling mass

21 Coupling axial force

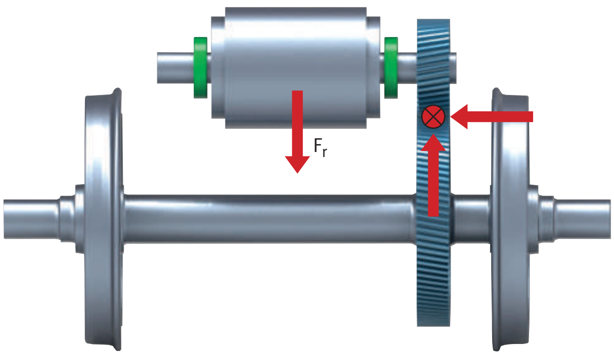
$$F_{aC} = \text{as stated}$$

F_{aC}	N	Coupling axial force

6.5.2 Loads due to the gearing

Additional gearing type-dependent gearing forces act on the bearing and are decisive for the selection and sizing of the bearing arrangement ➤28 | 19 ➤28 | 21.

19 Additional forces due to the gearing



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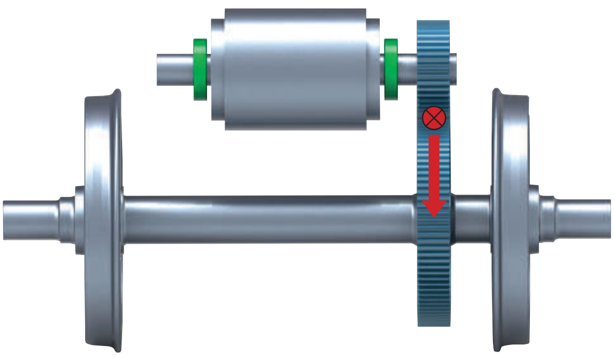
6.5.2.1 Spur gearing

In this section, the tangential, radial and axial forces arising due to the spur gearing are considered ➤28 | 20 ➤28 | 21.

20 Spur gearing



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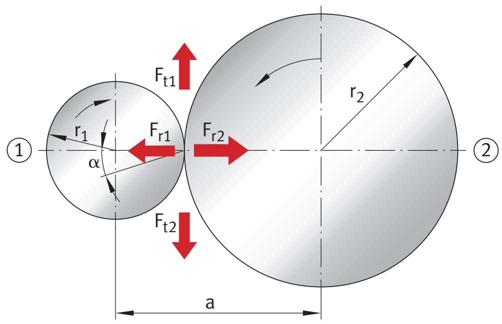
21 Tangential and radial forces arising due to the spur gearing



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1	Gear 1, driving	2	Gear 2, driven

22 Tangential force

$$F_{t1} = F_{t2} = \frac{M_{d1}}{r_1} = \frac{M_{d2}}{r_2}$$

F_t	N	Tangential force
M	Nm	Torque
r	m	Gear radius

23 Radial force

$$F_{r1} = F_{r2} = F_{t1} \cdot \tan \alpha$$

F_t	N	Tangential force
F_r	N	Radial force
α	°	Mesh angle

24 Axial force

$$F_{a1} = F_{a2} = 0$$

F_a	N	Axial force

25 Gear radius r_1

$$r_1 = \frac{a}{i+1}$$

a	m	Shaft spacing
i	–	Transmission ratio
r	m	Gear radius

26 Gear radius r_2

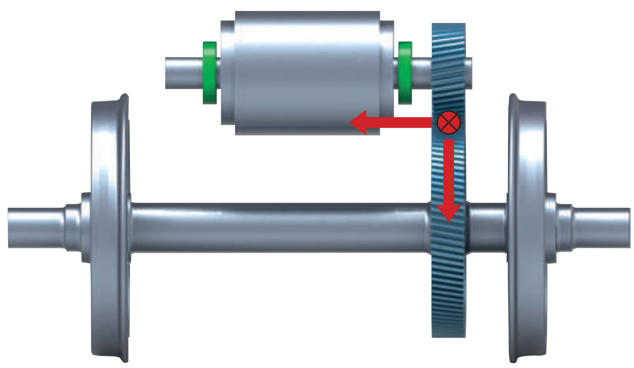
$$r_2 = a - r_1$$

a	m	Shaft spacing
r	m	Gear radius

6.5.2.2 Helical gearing

In this section, the tangential, radial and axial forces arising due to the helical gearing are considered ➤29 | 22 ➤30 | 23.

22 Helical gearing



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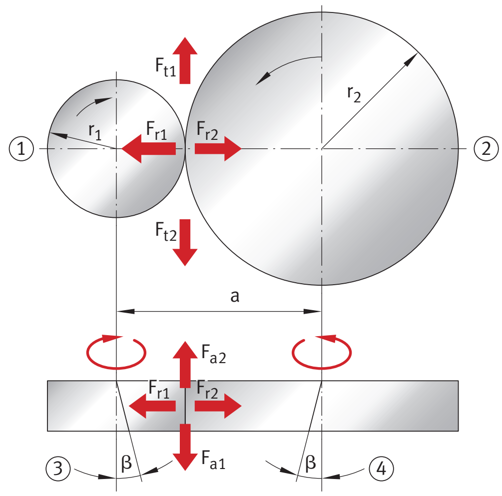
23 Tangential, radial and axial forces arising due to the helical gearing



1	Gear 1, driving	2	Gear 2, driven
3	Helix, left-handed	4	Helix, right-handed

27 Tangential force

$$F_{t1} = F_{t2} = \frac{M_{d1}}{r_1} = \frac{M_{d2}}{r_2}$$

F_t	N	Tangential force
M	Nm	Torque
r	m	Gear radius

28 Radial force

$$F_{r1} = F_{r2} = \frac{F_{t1} \cdot \tan \alpha}{\cos \beta}$$

F_r	N	Radial force
F_t	N	Tangential force
α	°	Mesh angle
β	°	Helix angle

29 Axial force

$$F_{a1} = F_{a2} = F_{t1} \cdot \tan \beta$$

F_a	N	Axial force
F_t	N	Tangential force
β	°	Helix angle

30 Gear radius r_1

$$r_1 = \frac{a}{i+1}$$

a	m	Shaft spacing
i	–	Transmission ratio
r	m	Gear radius

ƒ31 Gear radius r_2

$$r_2 = a - r_1$$

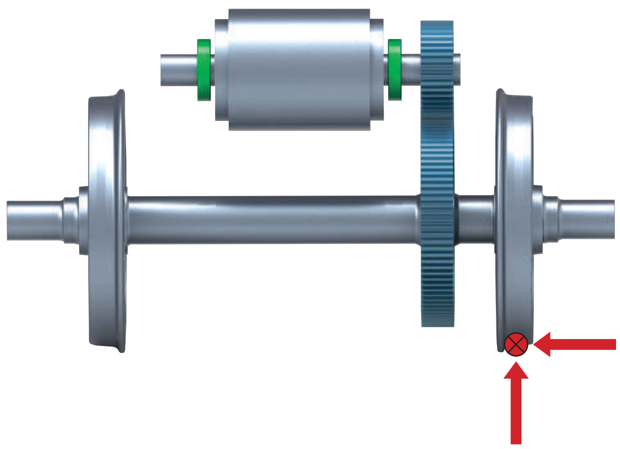
a	m	Shaft spacing
r	m	Gear radius

6.6 Additional loads from shocks and vibrations

6

Other additional loads that must be taken into account in the bearing design are loads from vibrations and shocks ➤31 | 24. The magnitude of the load is relatively difficult to determine and must be measured. Empirical values from other similar projects or comparable values from international standards are often used in the determination of the vibration load.

24 Additional loads from shocks and vibrations



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ƒ32 Acceleration force

$$F_{acc} = m \cdot a$$

a	m/s^2	Direction-dependent acceleration
F_{acc}	N	Acceleration force
m	kg	Mass

In the determination of the vibration load, additional loads due to the mass and the corresponding acceleration in accordance with the applicable equation above are calculated and taken into consideration as an additional load in the calculation. Thus, the accelerations (loads) acting on the system from the three directions x, y and z are included in the calculation. The length of time that these additional loads act is defined in cooperation with the customer.

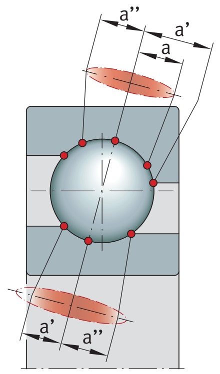
If no explicit load data for vibrations are submitted or the actually occurring vibration accelerations are not known, initial design of the bearing arrangement can be carried out using the effective values from DIN EN 61373-04 (VDE 0115-106) or EN 61373.

In the bearing design and in the consideration of the vibration accelerations, a distinction is made between the dynamic and the static case.

6

The dynamic case captures the additional load and takes it into account in the rating life calculation. The static case only allows for a plastic deformation safety test and tests the elliptical contact area with axial loading of a deep groove ball bearing ➤32 | 25. If cylindrical roller bearings of types NJ and HJ or NUP are used as locating bearings, the maximum axial load-carrying capacity of the ribs must be checked.

25 Elliptical contact area on a deep groove ball bearing



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Excerpt from DIN EN 61373-04 for dynamic loading and static loading ➤32 | 5.

5 Effective values for dynamic loading

Category		Orientation	Acceleration m/s²
1	Class A Mounted on the vehicle body	Vertical	0,75
		Transverse direction	0,37
		Longitudinal direction	0,5
	Class B Mounted on the vehicle body	Vertical	1,01
		Transverse direction	0,45
		Longitudinal direction	0,7
2	Mounted on the bogie	Vertical	5,4
		Transverse direction	4,7
		Longitudinal direction	2,50
3	Mounted on the wheelset	Vertical	38
		Transverse direction	34
		Longitudinal direction	17

6 Effective values for static loading

Category		Orientation	Peak acceleration	Nominal duration
			A	D
			m/s^2	ms
1	Class A and Class B Installed on the vehicle body	Vertical	30	30
		Transverse direction	30	30
		Longitudinal direction	50	30
2	Mounted on the bogie	All	300	18
3	Mounted on the wheelset	All	1000	6

k_S > 5/6; better: 1,0.

The contact ellipse index k_S indicates the extent to which the elliptical contact surface is actually supported by the raceway. Both elliptical contact areas are included in the calculation of the contact ellipse index k_S. If the raceway is sufficiently wide, an index > 1 is yielded. In this case, k_S can be interpreted as a safeguard against truncation of the contact ellipse. The minimum distance between the centre of the contact ellipse and the two shoulders is used for the calculation of k_S.

7 Lubrication

7.1 The functions of lubrication

The main function of the lubrication of rolling bearings is to prevent or reduce contact between rolling and sliding surfaces. As a result, friction and wear are kept to a low level.

The lubricant fulfils the following tasks:

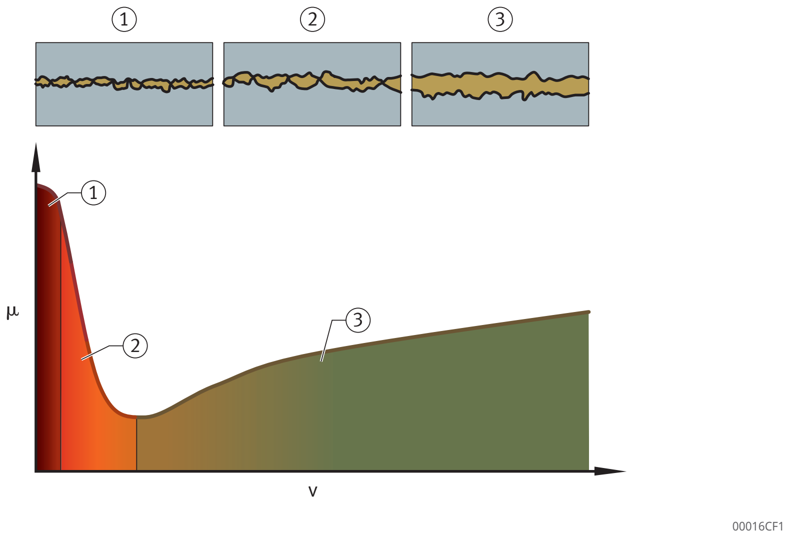
- provides protection against corrosion
- dissipates heat from the bearing (in the case of oil lubrication)
- flushes out wear particles and contaminants (recirculating oil lubrication with filtration of the oil)
- supports the sealing effect of bearing seals (grease collar, pneumatic oil lubrication)

7.2 Lubrication and friction regimes

The friction and wear behaviour and the achievable fatigue life for lubricated contact depend strongly on the separating effect of the lubricating film.

The Stribeck curve shows the possible friction regimes of boundary friction, mixed friction and hydrodynamic friction as a function of the hydrodynamically effective speed. All three regimes occur with oil as well as grease lubrication. In addition to speed, viscosity plays an important role. For grease lubrication, the viscosity of the base oil is decisive, but the grease thickener can also have a lubricating film-forming effect.

26 Stribeck curve



1	Boundary lubrication	2	Mixed lubrication
3	Full lubrication		

μ	–	Friction
v	–	Velocity

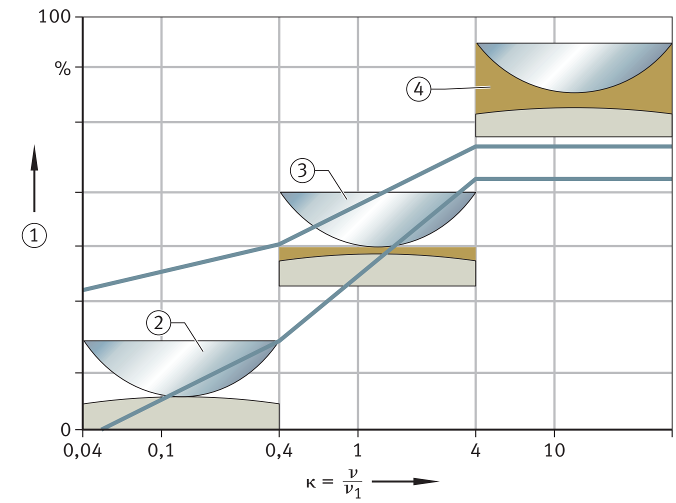
The fatigue life of rolling bearings is affected by the lubricating film.

The lubricating film thickness is influenced by the following:

- lubricant characteristics
- macrogeometry and microgeometry of the contact surfaces
- speed of the contact surfaces relative to each other

Separation of the contact surfaces by the lubricating film is desired.

27 Lubrication regimes during operation



1	Separation of contact surfaces	2	Metal-to-metal contact
3	Partially elastohydrodynamic (EHD), mixed friction	4	EHD conditions

κ	–	Viscosity ratio

7 Viscosity ratio

κ –	**Viscosity ratio**	**Effect**
0,4 ... 1	Moderate mixed friction	A reduction in the basic rating life is to be expected.
1 ... 2	Mixed friction	With a good level of cleanliness, the basic rating life can be achieved.
2 ... 4	Mixed friction, full-fluid lubricating film	With a good level of cleanliness, the basic rating life is achieved.
> 4	Full lubrication	With maximum cleanliness and a moderate load, rolling bearings can be durable under these conditions.

The quality of the lubrication state in lubricated rolling contacts can be described by the viscosity ratio κ. The lubricating film thickness is based on the surface roughness. If the lubricating film thickness is greater than the surface roughness, separating lubrication of the rolling partners occurs and there is no damage-relevant contact between the surfaces.

A so-called reference viscosity ν_1 that is just sufficient to ensure separation of the surfaces at the operating temperature was defined. This makes the procedure easy to use. The viscosity ratio κ was then defined as the ratio of the viscosity of the lubricant at the operating temperature to the reference viscosity.

7

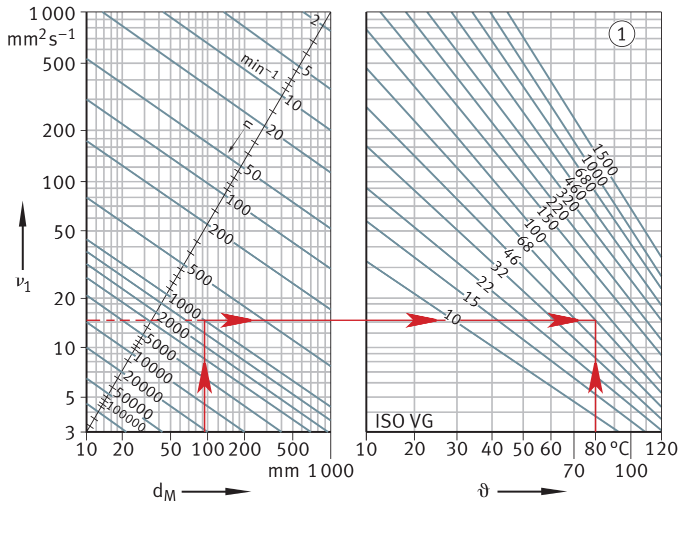
33 Viscosity ratio

$$\kappa = \frac{\nu}{\nu_1}$$

κ	–	Viscosity ratio
ν	mm^2/s	Kinematic viscosity of the lubricant at operating temperature
ν_1	mm^2/s	Reference viscosity of the lubricant at operating temperature

The reference viscosity ν_1 can be determined with the help of the mean bearing diameter d_M= (d + D)/2 and the operating speed n.

28 Reference viscosity and V/T diagram for mineral oils



d_M	mm	Mean bearing diameter (d+D)/2
n	min^{-1}	Operating speed (nominal speed)
ν_1	mm^2/s	Reference viscosity of the lubricant at operating temperature
ϑ	°C	Temperature

7.3 The supply of lubricant to bearings

Depending on the loading conditions and the design, a rolling bearing can be lubricated both with lubricating oil and with grease. In the case of greases, lubrication is performed mainly through the base oil that is released in small amounts over time by the thickener.

The lubricant amount actually required by a rolling bearing is extraordinarily small. Due to the operational reliability of the bearing arrangement, however, it is normally estimated at a higher value in practice. However, too much lubricant in the bearing can lead to damage. If excess lubricant cannot be forced out, the splashing or churning work leads to temperatures at which the lubricant may be affected or even broken down. Over-greasing can thus lead to premature bearing failure.

Adequate supply is ensured by the following:

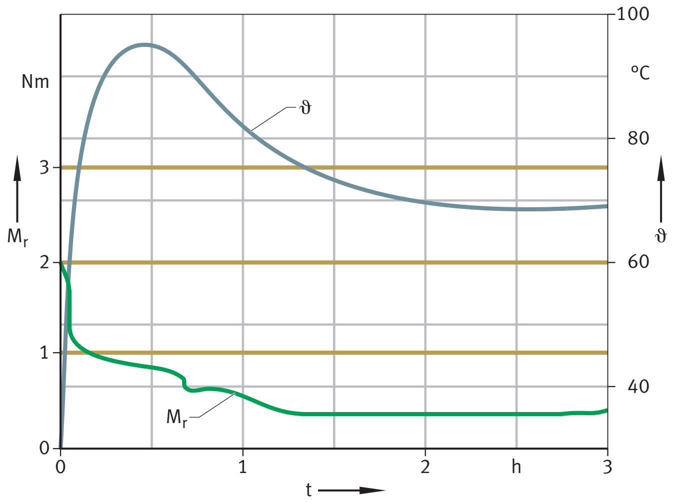
- selection of the correct lubricant amount and distribution in the bearing
- consideration of the operating life of the lubricant
- appropriate addition of lubricant or lubricant replacement
- targeted design of the bearing position
- suitable sealing

7.4 Initial greasing and regreasing

Instructions for correctly greasing bearings:

- Fill the bearings such that all functional surfaces definitely receive grease.
- Fill the existing housing cavity adjacent to the bearing with grease only to the point where there is still sufficient space for the grease forced out of the bearing. This is intended to avoid co-rotation of the grease. If the bearing is adjoined by a larger, unfilled housing cavity, cover washers or sealing washers as well as baffle plates should ensure that an appropriate amount of grease remains in the vicinity of the bearing.
- The sealing action of a gap seal is improved by the formation of a stable grease collar. This effect is supported by continuous relubrication.
- If the correct degree of filling is used, favourable friction behaviour and low grease loss will be achieved.
- If there is a pressure differential between the two sides of the bearing, the flow of air may drive the grease and the released base oil out of the bearing and may also carry contamination into the bearing. In such cases, pressure balancing is required by means of openings and holes in the adjacent parts.
- If a high temperature is expected in the bearing, an appropriate grease as well as a grease reservoir should be provided if the surrounding construction allows for this. The reservoir should be provided with an oil-releasing surface of maximum possible size facing the bearing and with an open space between the surface and the bearing. A favourable amount for the reservoir is two or three times the normal fill level. The reservoir should preferably be provided in equal volumes on both sides of the bearing, or, as a temporary solution, on one side.
- At relatively high characteristic speeds, the bearing temperature may be elevated in the starting phase, often for several hours, if the grease amount has not been adjusted correctly. The greater the degree of filling of the bearings and the cavities adjacent to the bearings with grease is and the more difficult it is for grease to escape freely, the higher the temperature is and the longer the phase in which the temperature is elevated is. A remedy is a so-called interval running-in process with appropriately determined standstill periods for cooling. If suitable greases and grease amounts are used, equilibrium is achieved after a very short time.

29 Frictional torque and temperature



M_R	Nm	Frictional torque
t	h	Time
ϑ	°C	Temperature

7.5 Selection of a suitable lubricant

Selection of a suitable lubricant is decisive for the reliable function of the bearing. The optimum operating life can be achieved, for example, if suitable greases are selected.

Factors influencing the selection of a suitable lubricant:

- bearing type
- speed
- temperature
- load

7.5.1 Influence of bearing type

A distinction is made between rolling bearings with point contact (ball bearings) and rolling bearings with line contact (cylindrical roller bearings).

Rolling bearings with point contact

For rolling bearings with point contact, at each point where the rolling element contacts the raceway, only a relatively small volume of grease is loaded. In addition, the rolling kinematics of ball bearings exhibit only relatively small proportions of sliding motion. The specific mechanical stress placed on greases in bearings with point contact is therefore significantly less than in bearings with line contact.

Rolling bearings with line contact

Rolling bearings with line contact place greater demands on the grease. Not only is a greater amount of grease loaded during contact, but also sliding and rib friction can always be expected. This prevents the formation of a lubricating film and results in wear.

7.5.2 Influence of speed

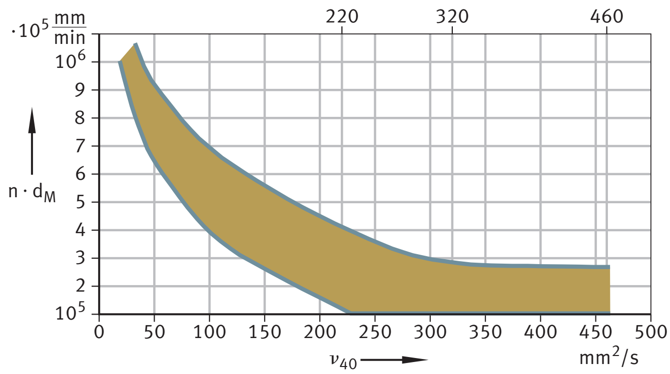
Like rolling bearings, greases have maximum permissible characteristic speeds. The characteristic speed of a grease should always match the characteristic speed of the bearing or be suitable for the operating conditions.

For greases, the characteristic speed depends on the respective thickener, the base oil type and the proportional composition.

Typically, greases for high speeds have a low base oil viscosity. They are also suitable for low temperatures.

Greases for low speeds have a higher base oil viscosity and are frequently used as heavy-duty greases. Because the characteristic speed also depends on other parameters such as the bearing design, it is not a pure material characteristic. As a rule, several design-specific characteristics are given in the technical data sheets for the greases.

30 Characteristic speed for greases



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$n \cdot d_M$	mm/min	Characteristic speed
ν_{40}	mm²/s	Base oil viscosity of grease at +40 °C

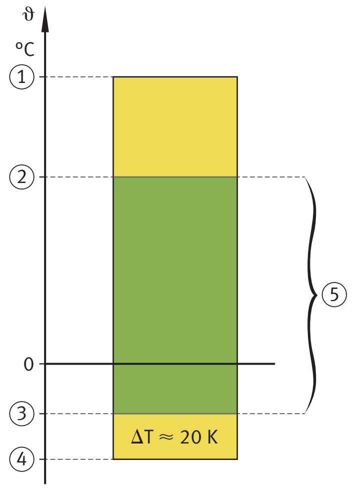
7.5.3 Influence of temperature

The temperature range of a grease must correspond to the range of possible operating temperatures in the rolling bearing and is referred to as the operating temperature range. The operating temperature range depends on the thickener, the base oil, the additives and the production process.

At low temperatures, greases release very little base oil. This can result in lubricant starvation. Schaeffler therefore recommends that greases should not be used below the lower continuous limit temperature $\vartheta_{lowerlimit}$. This is approx. 20 K above the lower operating temperature of the grease as stated by the grease manufacturer.

The grease life is based on the standard operating range, which is limited at the high end by the reference temperature $\vartheta_{reference}$. This reference temperature must not be exceeded if a temperature-related decrease in the grease life is to be avoided.

31 Operating temperature range



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1	Upper operating temperature according to grease manufacturer	2	$\vartheta_{reference}$
3	$\vartheta_{lowerlimit}$	4	Lower operating temperature according to grease manufacturer
5	Standard operating range		

ϑ	°C	Temperature
ΔT	K	Temperature difference

7.5.3.1 Operating temperature range

The operating temperature range of the grease should be suitable for the respective operating temperatures of the rolling bearing.

The grease manufacturers specify operating temperature ranges for their greases in accordance with DIN 51825 (Lubricants – Lubricating greases K). The operating temperature range is dependent on the thickener type, the proportion of thickener, the base oil type, the proportion of base oil, the production quality and the production process.

The upper value is determined in accordance with DIN 51825 by means of testing using the rolling bearing grease test rig FE9. At the upper operating temperature, a 50% failure probability (F_{50}) of at least 100 h must be achieved in this test.

The lower value is determined in accordance with DIN 51825 by means of the flow pressure. The flow pressure of a grease is the pressure required to press a stream of grease through a defined nozzle. For greases of type K, the flow pressure at the lower operating temperature must be less than 1400 mbar.

The use of flow pressure in determining the lower operating temperature only indicates, however, whether the grease can be moved at this temperature. It cannot be used to give an indication of the suitability of the grease for use in rolling bearings at low temperatures.

In addition to the lower operating temperature of a grease, therefore, the low-temperature frictional torque is also determined in accordance with ASTM D 1478 or IP 186/93. At the lower operating temperature, the starting torque must not exceed 1 Nm and the running torque must not exceed 0,1 Nm.

Schaeffler recommends that greases should be used in accordance with the bearing temperature normally occurring in the standard operating range in order to achieve a reliable lubricating action and an acceptable grease operating life.

The operating temperature range for a grease can be taken from the corresponding data sheet.

7.5.4 Influence of load

For a load ratio C/P < 10, greases that have higher base oil viscosities and in particular contain anti-wear additives (EP) are recommended. These additives form a reaction layer on the metal surface that gives protection against wear. Such greases are identified in accordance with DIN 51825 by KP. Their use is also recommended for bearings with an increased proportion of sliding motion (including slow running) or line contact as well as under combined loads (radial, axial).

Greases with solid lubricants such as PTFE or molybdenum disulphide should preferably be used for applications in the boundary or mixed friction range. The solid lubricant particle size should not exceed 5 μm. Because of their low load carrying capacities, which cannot be compensated for through the use of corresponding additives, silicone lubricants must only be used at very low loads $P \leq 0{,}03 \cdot C$.

7.6 Grease operating life

7.6.1 Basic grease operating life

The basic grease operating life t_f depends on the bearing-related characteristic speed.

ƒ34 Bearing-specific characteristic speed

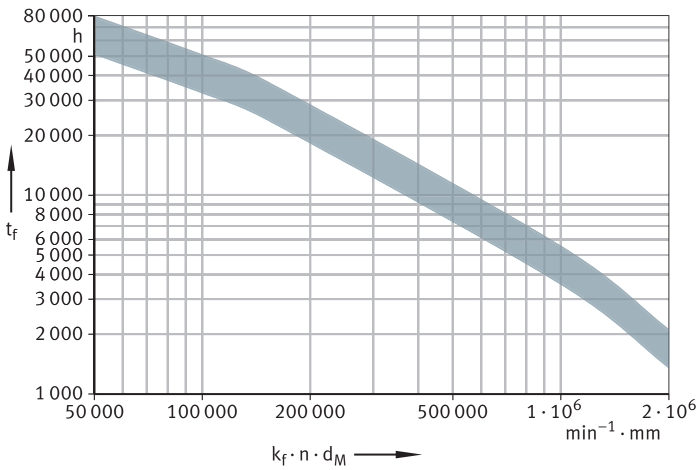
$$k_f \cdot n \cdot d_M$$

d_M	mm	Mean bearing diameter (d+D)/2
k_f	–	Bearing factor, as a function of bearing type
n	min^{-1}	Operating speed or equivalent speed

The method of determining the basic grease operating life can be used under the following conditions:

- greases with proven performance capabilities for use in bearings
- bearing arrangements with bearing temperatures that are lower than the reference temperature $\vartheta_{reference}$ of the grease
- load ratio of $C_0/P \geq 20$
- constant speed and load
- load acting in the main direction of bearing loading
  - radial bearings: radial
  - axial bearings: axial
- radial bearings with horizontal axis of rotation
- rotating inner ring
- bearing arrangement with no disturbing environmental influences

32 Determination of the basic grease operating life for steel/steel



t_f	h	Basic grease operating life
$k_f \cdot n \cdot d_M$	mm/min	Bearing-specific characteristic speed

8 Factor k_f, as a function of bearing type

Bearing type	Factor k_f
Deep groove ball bearings, single-row, Generation C	0,8
Deep groove ball bearings, single-row	1
Angular contact ball bearings, single-row	1,6
Angular contact ball bearings, single-row, X-life	1,3
Four-point contact bearings	1,6
Four-point contact bearings, X-life	1,3
Cylindrical roller bearings, single-row	2

7.6.2 Grease operating life

The grease operating life t_{fG} applies where this is below the calculated bearing life and the bearings are not relubricated.

35 Guide value for grease operating life

$$t_{fG} = t_f \cdot K_T \cdot K_P \cdot K_U$$

t_f	h	Basic grease operating life
t_{fG}	h	Guide value for grease operating life in operating hours
K_T	–	Correction factor for increased temperature
K_P	–	Correction factor for load
K_U	–	Correction factor for environment

7.6.2.1 Temperature factor K_T for increased temperature

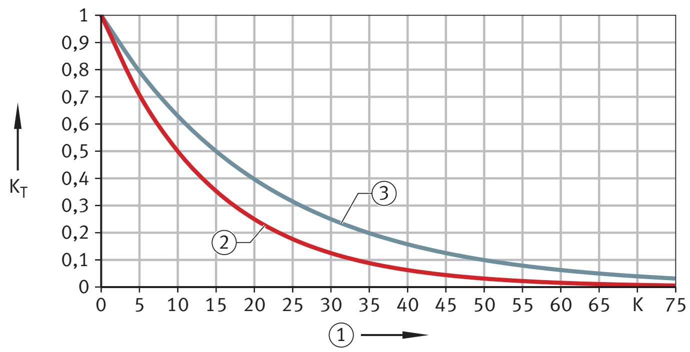
An increase in temperature leads to acceleration of the reaction rate and thus of the oxidation and ageing rates.

If the bearing temperature is above the reference temperature $\vartheta_{reference}$, K_T should be determined by means of the diagram.

The temperature factor K_T must not be used if the bearing temperature is higher than the upper operating temperature of the grease used. In this case, a different grease should possibly be selected or Schaeffler should be consulted.

Depending on the grease quality, temperature factors $K_T > 1$ may also be possible below the reference temperature.

33 Temperature factor K_T



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1	K above $\vartheta_{reference}$	
K_T	–	Correction factor for increased temperature

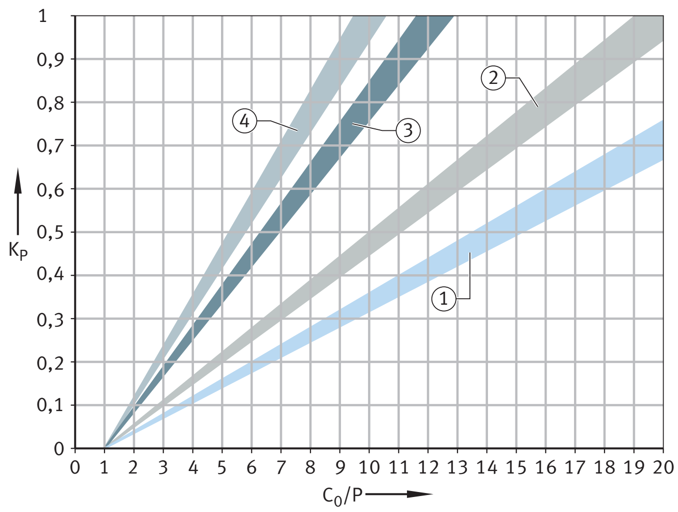
9 Temperature factor K_T

Curve	Lubricating grease group	Rolling bearing grease Arcanol
2	GA08	MULTITOP
	GA15	MULTI3
	GA16	LOAD150
	GA17	LOAD1000
	GA47	–
3	GA11	MULTI2
	GA13	LOAD220
	GA14	LOAD400
	GA22	LOAD460

7.6.2.2 Correction factor K_P for increased load

Under higher bearing load, greases are subjected to greater strain. This effect can be taken into consideration through the factor K_P, which depends on the load ratio C_0/P and the bearing type.

34 Load correction factor K_P



K_P	–	Correction factor for load
C_0/P	–	Load ratio

10 Load correction factor K_P

Curve	Bearing type
1	Axial angular contact ball bearings, double-row
2	Cylindrical roller bearings, double-row (not valid for NN30)
3	Four-point contact bearings
	Cylindrical roller bearings LSL, ZSL
	Cylindrical roller bearings, full-complement
	Cylindrical roller bearings, single-row (constant, alternating, without axial load)
4	Deep groove ball bearings, single- and double-row
	Angular contact ball bearings, single- and double-row

7.6.2.3 Environment factor K_U

The environment factor K_U takes into account the effects of humidity, shaking forces, slight vibrations and shocks. Vibrations with low amplitudes can be the cause of tribocorrosion.

! The environment factor does not take into account any extreme environmental influences such as water, aggressive media, contamination, nuclear radiation or extreme vibrations such as those occurring in vibratory machines. In relation to contamination, the influence of contamination on rating life calculation must also be noted.

11 Environment factor K_U

Environmental influence	Environment factor K_U
Low, e.g. on test rig	1
Moderate (standard)	0,8
High, e.g. in open-air application	0,5

7.6.3 Relubrication intervals

If rolling bearings are relubricated, it is important to pay attention to the lubrication interval in order to ensure the reliable functioning of the bearings.

! For reasons of operational reliability, relubrication intervals of > 1 a (year) are not recommended.

36 Guide value for relubrication interval

$$t_{fR} = 0{,}5 \cdot t_{fG}$$

t_{fR}	h	Guide value for relubrication interval in operating hours
t_{fG}	h	Guide value for grease operating life in operating hours

The grease used for relubrication should be the same as that used for initial greasing.

If grease mixtures are unavoidable, the miscibility and compatibility of the greases should be tested ➤46 | 7.6.6.

If supply lines are filled pneumatically, the filling volume of the supply lines must be included in the calculation of the relubrication amount.

7.6.4 Relubrication amounts

The initial lubrication amount and the relubrication amount are usually calculated based on the free or undisturbed space. The relubrication amount can be estimated.

37 Estimation of relubrication amount

$$m = D \cdot B \cdot X$$

B	mm	Width
D	mm	Outside diameter
X	-	Factor (see following table)

12 Factor X for relubrication intervals

Relubrication interval	Factor X
Weekly	0,002
Monthly	0,003
Yearly	0,004

Where lubrication intervals are long, the aim should be to achieve grease replacement. Substantial replacement of used grease by fresh grease is achieved with the aid of a larger relubrication amount. A large relubrication amount is necessary above all if the used grease has already been damaged due to higher temperature.

Schaeffler recommends use of Bearinx for more precise determination of the relubrication amount.

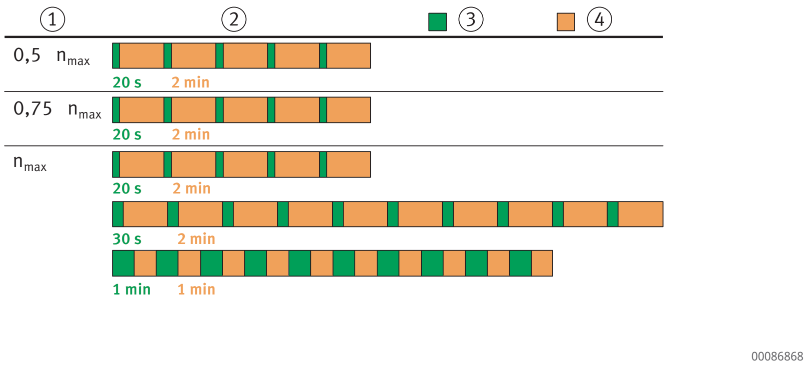
Bearinx | Calculation modules | https://www.schaeffler.de/std/1FEB

7.6.5 Grease distribution

For rapidly rotating bearings with a characteristic speed > 500000 $min^{-1} \cdot mm$, a grease distribution run is needed.

The running-in process consists of several start-stop cycles with different speeds and running times and, very importantly, with stationary periods following the running periods. The number of cycles required varies according to the bearing size, the number of bearings, the maximum speed and the bearing environment.

35 Suggested grease distribution run



1	Speed	2	Running times and stationary times
3	Running time	4	Stationary time

Further cycles should be carried out with extended running times and shortened stationary times until the steady-state temperature is reached.

7.6.6 Miscibility of lubricants

! Mixtures of greases should be avoided.

If mixtures of greases are unavoidable, the following conditions must be met:

- All greases must have the same base oil basis.
- The thickener types must match.
- The base oil viscosities must be similar (they must not differ by more than one ISO VG class).
- The consistencies must be the same (NLGI grade).

The miscibility can be roughly estimated, but without a guarantee of operational reliability.

13 Miscibility of lubricants (base oils)

	Mineral oil	**PAO**	**Ester oil**	**Polyglycol oil**	**Silicone oil**	**Alkoxyfluoro oil**
Mineral oil	+	+	+	–	o	–
PAO	+	+	+	–	o	–
Ester oil	+	+	+	o	–	–
Polyglycol oil	–	–	o	+	–	–
Silicone oil	o	o	–	–	+	–
Alkoxyfluoro oil	–	–	–	–	–	+

+	mixing generally non-critical
o	miscible in individual cases, verification required
–	mixing not permissible

14 Compatibility of different thickener types

	Lithium soap	**Lithium complex**	**Sodium complex**	**Calcium complex**	**Aluminium complex**	**Barium complex**	**Bentonite**	**Polycarbamide**	**PTFE**
Lithium soap	+	+	–	+	–	+	–	–	+
Lithium complex	+	+	o	+	o	o	–	o	+
Sodium complex	–	o	+	o	o	o	–	o	+
Calcium complex	+	+	o	+	o	o	o	o	+
Aluminium complex	–	o	o	o	+	o	–	–	+
Barium complex	+	o	o	o	o	+	+	o	+

	Lithium soap	Lithium complex	Sodium complex	Calcium complex	Aluminium complex	Barium complex	Benton-ite	Polycarb-amide	PTFE
Bentonite	–	–	–	o	–	+	+	–	+
Polycarbamide	–	o	o	o	–	o	–	+	+
PTFE	+	+	+	+	+	+	+	+	+

+ mixing generally non-critical
o miscible in individual cases, verification required
– mixing not permissible

Before mixing is done, the lubricant manufacturer must always be consulted. Even if the preconditions are fulfilled, the performance capability of the grease mixture may be impaired. Relubrication should only be carried out using greases of comparable performance capability. Before a switch to a different grease grade is made, the previous grease must first be flushed out as far as this is permitted by the design. Further relubrication should be carried out after a shortened period. If incompatible greases are mixed, this can lead to considerable structural changes. Substantial softening of the grease mixture may also occur.

8 Electrical insulation

8.1 Rolling bearing currents – causes and remedial measures

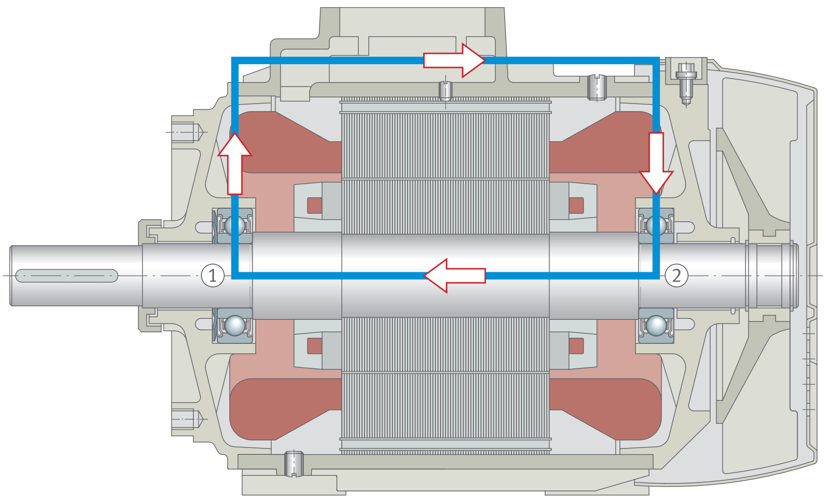
Depending on the motor, frequency converter and the operating conditions involved, there are mainly three different types of unwanted electrical currents that occur in electric motors. The resulting remedial measures are also selected on the basis of the cause or current type. Conductive elements, improved grounding and insulation for rolling bearings are among the measures that have proved successful.

Circular currents

In the case of very large motors or generators with a small number of poles, magnetic asymmetries can cause a low-frequency shaft voltage. For motors with a frame size of 100 mm and above that are operated by frequency converters, the high-frequency currents that flow through the slot insulation of the stator lead to a high-frequency shaft voltage.

Without insulated bearings, low-frequency shaft voltage and high-frequency shaft voltage lead to the phenomenon of circular currents. An effective and practical remedy to this issue is available in the form of rolling bearings with a ceramic oxide coating (Insutect A) and hybrid rolling bearings from Schaeffler. A rolling bearing coated with Insutect A is a particularly frequent choice of insulation for the bearing on the fan side.

36 Induced voltage along a shaft



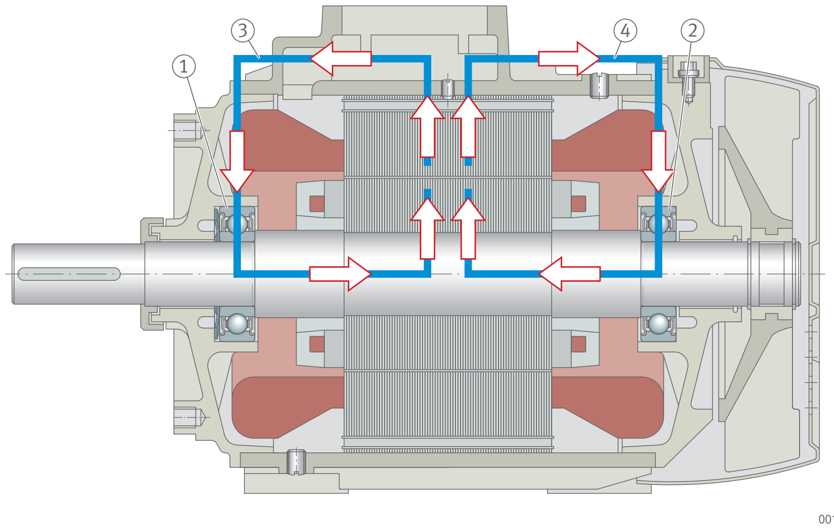
1	Bearing, drive end side	2	Bearing, fan side

Discharge currents

Common mode voltage is an unwanted occurrence in electric motors that are operated with frequency converters. This voltage, which is present between the shaft and housing, can lead to discharge currents, particularly in small electric motors with frame sizes of up to 315 mm, that can flow separately through each of the two bearings.

Two hybrid bearings with ceramic rolling elements and/or current conductive solutions have proven to be effective remedies against discharge currents. Alternatively, a solution is also available in the form of an Insutect A coating, which must be selected in a suitable layer thickness. The motor and adjacent parts determine which is the better solution for the respective application.

37 Discharge currents



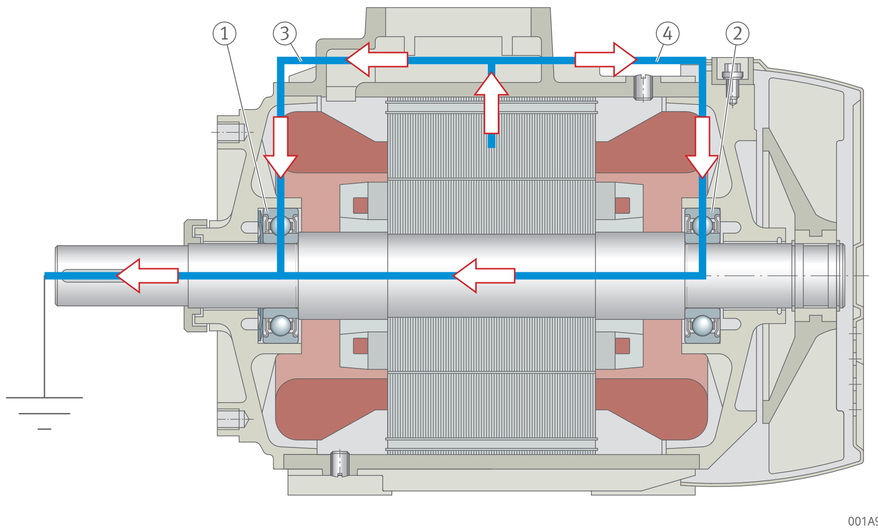
1	Bearing, drive end side	2	Bearing, fan side
3	Possible current path 1	4	Possible current path 2

Rotor ground currents

Particularly in larger stationary electrical machines, poor grounding conditions can result in a current that flows from the housing, via the bearings, to the shaft and through the driven or driving unit.

Suitable remedies for such currents include grounding that is suitable for high-frequency currents, an insulated coupling or insulation of the rolling bearings on the drive and fan sides.

38 Rotor ground current



1	Bearing, drive end side	2	Bearing, fan side
3	Possible current path 1	4	Possible current path 2

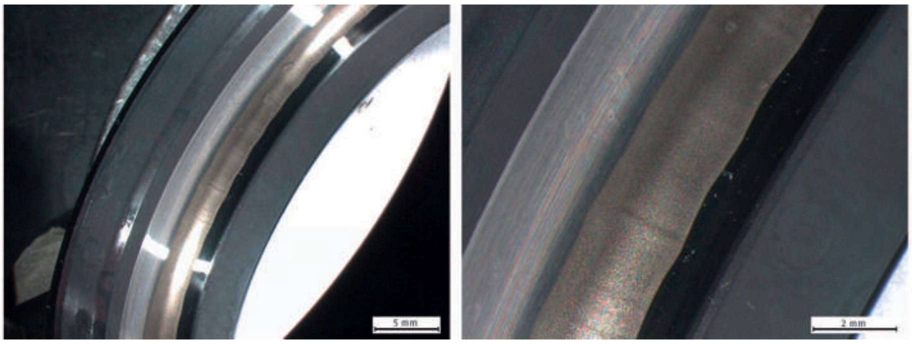
8.2 Typical bearing damage in current passage

The same surface changes always occur regardless of whether a bearing is subjected to direct current or alternating current up to frequencies in the MHz range.

8.2.1 Marks on raceways and rolling elements

In many cases, uniformly matt grey marks are created by current passage on the raceways and rolling element surfaces. These marks can also be caused by other influences, for example by abrasive substances in contaminated lubricant.

39 Marks on raceways



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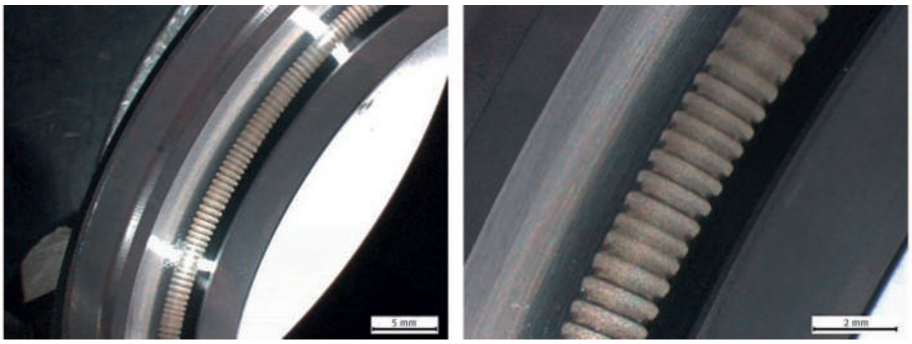
8.2.2 Fluting

Fluting consists of periodic patterns running in the direction of rolling on areas of the surface of different depth. Current passage is often the cause of fluting.

40 Fluting



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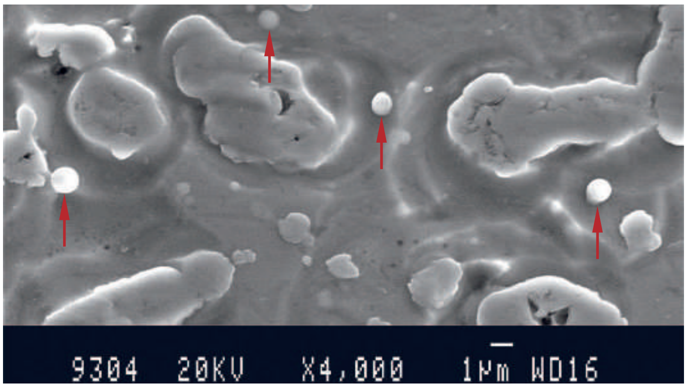
The use of a scanning electron microscope (SEM) reveals that both structures are composed of melt craters and welding beads in the μm range. The damage structures overlap the overrolled raceway in a tightly packed form. This demonstrates current passage through the bearing.

41 Damage structures under the scanning electron microscope (SEM)



8.2.3 Development of bearing damage

The melt craters and welding beads develop during electrical discharge between the micropeaks that are present on raceways and on the rolling element surfaces. Where a lubricant film is fully formed, this is punctured by the spark at the thinnest point and the base points of the spark are melted for a short period.

With metal-to-metal contact in the mixed friction range, the surfaces involved become fused together. This fusion is then immediately broken apart again by the rotation of the bearing, during which material becomes detached from the surfaces and immediately solidifies to form welding beads. Some of the welding beads are mixed with the lubricant and are deposited on the metal surfaces. As overrolling continues, the craters and welding beads can be flattened and smoothed. Under a continuing flow of current, the surface layers involved repeatedly melt in this way over time.

Fluting is responsible for most bearing failures ➤50 | 40.

The commonly applied theory behind its formation is as follows:
When a rolling element rolls over any sufficiently large melt crater it undergoes radial motion. The parameters for this radial motion are dependent on the internal geometry, speed and load of the bearing. When the rolling element swings back, the lubricant film thickness is reduced. New sparkovers increasingly occur in this area and a self-sustaining process is initiated, which produces the periodic structures on the raceway. After a time, fluting can cover the raceway of the ring in the load zone or the entire circumference of the raceway. This fluting leads to further increases in bearing vibration and ultimately to failure of the bearing.

8.2.4 Influence on the lubricant

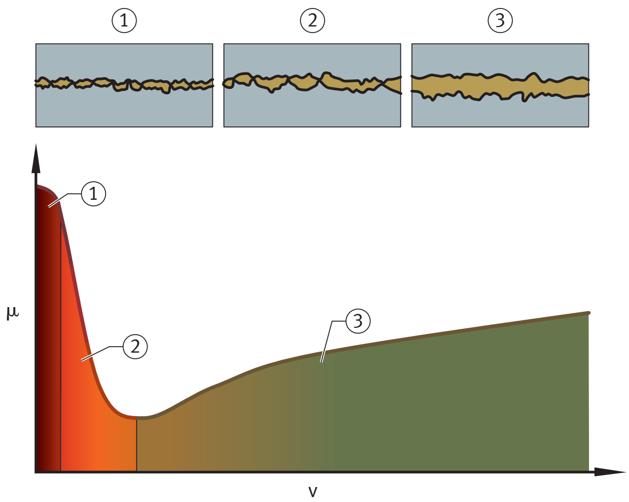
Current passage also has a negative effect on the lubricant. The base oil and additives are damaged. Premature ageing and an increased concentration of iron particles cause a marked deterioration in the lubrication characteristics and noise behaviour of the bearing.

8

8.3 Electrical behaviour of rolling bearings

8.3.1 Electrical behaviour of an uncoated rolling bearing

42 Electrical behaviour of an uncoated rolling bearing



1	Boundary lubrication	2	Mixed lubrication
3	Full lubrication		

μ	–	Friction
v	–	Velocity

The electrical behaviour of a rolling bearing is dependent on the lubrication condition. The lubrication condition in which solid body contact is predominantly present with only partial fluid friction is described as boundary lubrication (area 1). In this condition, the bearing can be viewed as having ohmic resistance.

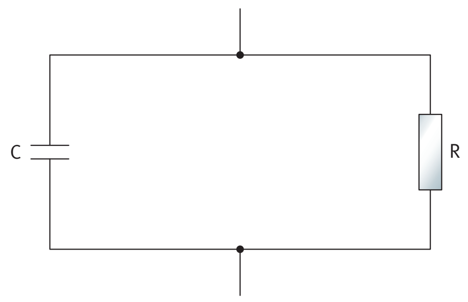
If the rolling contact surfaces moving relative to each are separated almost completely by a lubricant film, the term full lubrication is used (area 3). With full lubrication, the rolling bearing exhibits capacitance behaviour.

In the case of mixed friction or partial lubrication (area 2), a thin lubricant film is present, but the rolling contact surfaces continue to come into contact. In such cases, the rolling bearing exhibits both ohmic and capacitive resistance.

8.3.2 Electrical behaviour of a current-insulated rolling bearing

A current-insulated bearing can be viewed as a parallel connection of ohmic resistance and capacitance.

43 Parallel circuit, resistance and capacitance



C	F	Electrical capacitance
R	Ω	Ohmic resistance

For good insulation, the ohmic resistance should be as high as possible and the capacitance as low as possible.

The decisive factor in the selection of current insulation is the type of voltages present. In the case of direct current voltage and alternating current voltage at 50 Hz or 60 Hz , the ohmic resistance is decisive. In the case of higher-frequency alternating current, the capacitive resistance of the bearing is decisive. These high-frequency alternating currents are often found in electric motors that are operated using frequency converters, where frequencies ranging from several 100 kHz to several MHz are typically encountered.

2 types of electrical resistance are important:

- Direct current voltage resistance
  The direct current voltage resistance of bearings with an Insutect A coating is at least 50 MΩ at room temperature, based on the coating J20AB. Thus, according to Ohm’s Law I = V/R, only currents significantly below 20 µA are possible with voltages up to 1000 V. Currents below 20 µA are not critical to bearings.
- Alternating current voltage resistance
  Frequency converters that generate unwanted and high-frequency currents in the range of several 100 kHz to several MHz are being used in an increasing number of applications. In this frequency range, ohmic resistance plays a minor role. The decisive factor here is the capacitive impedance of the bearing insulation, which should be as high as possible, and is largely determined by the frequency of the bearing current and the capacitance of the bearing.

38 Impedance dependent on frequency and capacitance

$$Z = \frac{1}{2 \cdot \pi \cdot f \cdot C}$$

C	F	Electrical capacitance
f	Hz	Frequency
Z	Ω	Impedance

The capacitance of a rolling bearing with an Insutect A coating can be calculated using the following formula:

39 Capacitance from area and layer thickness

$$C = \varepsilon_0 \cdot \varepsilon_r \cdot \left(\frac{A}{s}\right)$$

ε_0	As/Vm	Electric field constant
ε_r	–	Relative permittivity, dependent on material
A	mm²	Area, coated
C	F	Electrical capacitance
s	mm	Layer thickness

8

A large coating thickness and a small coated surface will nevertheless lead to a low capacitance and thus to a high impedance.

In practice, a reliable criterion for assessing the level of hazard presented by current passage has been found to be the calculated current density J_S, in other words the effective amperage divided by the total contact area between the rolling elements and the inner or outer ring of the bearing. The calculated current density is dependent on the bearing type and the operating conditions. At current densities with effective amperages above around 0,1 A/mm², there is the risk of current damage. In addition, white etching cracks (WECs) can also occur at lower current densities. White etching cracks are the result of interactions between certain lubrications and an additional load, e.g. current passage.

40 Current density

$$J_S = \frac{I}{A_{eff}} \left[A/mm^2\right]$$

A_{eff}	mm²	Effective contact area
I	A	Amperage
J_S	A/mm²	Effective current density

8.4 Ceramic-coated bearings

Ceramic-coated bearings are standard bearings in which either the inner ring or outer ring has the ceramic coating Insutect A.

8.4.1 Types of coatings

44 Deep groove ball bearing with ceramic coating on the outer ring

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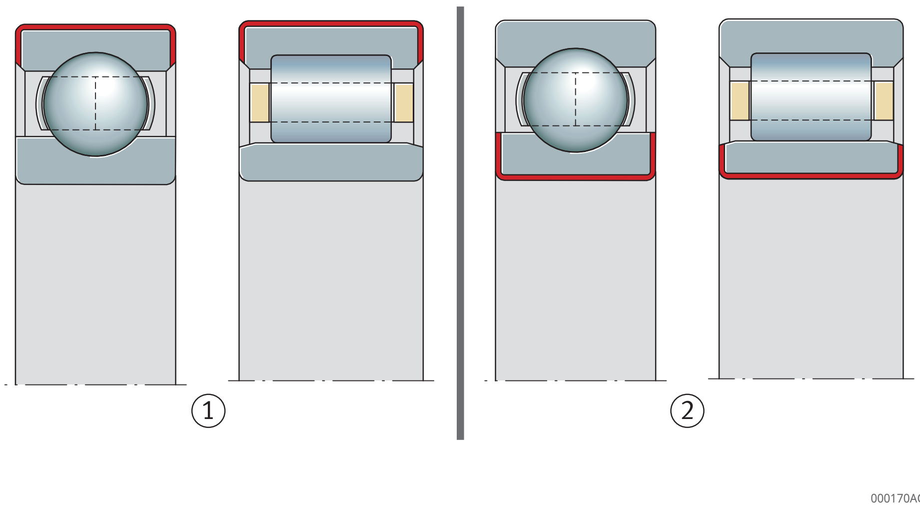
45 Cylindrical roller bearing with ceramic coating on the outer ring

Bearings coated with Insutect A have a high degree of electrical insulation protection. Bearings with an oxide ceramic coating are identified by the suffix J20 and an additional letter combination of GA, GB or GI. Previously, the suffixes AB, AA or C were used. The oxide ceramic layer is very hard, resistant to wear and has good thermal conductivity.

The external dimensions of the current-insulated rolling bearings correspond to the dimensions in accordance with DIN 616 (ISO 15). Current-insulated bearings are therefore interchangeable with standard bearings.

The various type of coatings for bearings are shown in cross-section.

46 Types of coatings



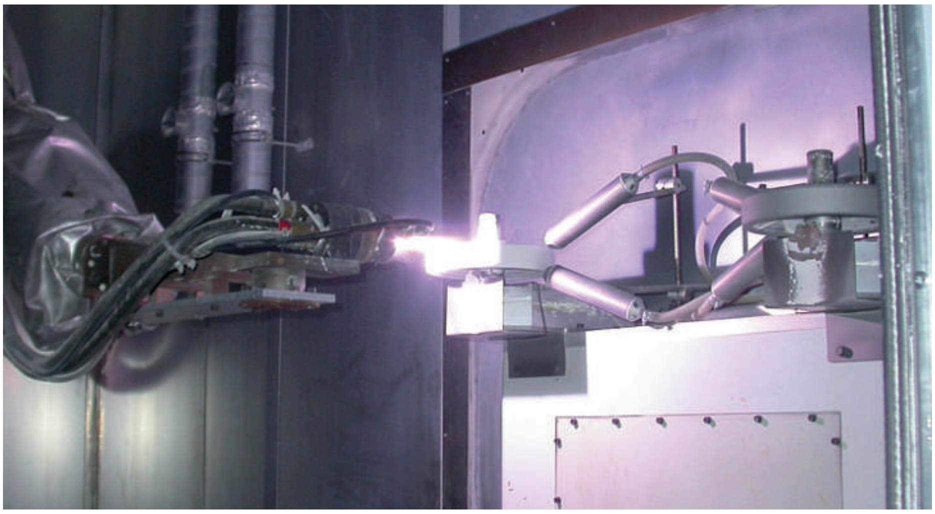
1	Outer ring coating: J20GA, J20GB	2	Inner ring coating: J20GI

8.4.2 Coating method

The bearings are coated using the plasma spray method. In the plasma spray method, an electric arc is generated between two electrodes and the inert gas introduced is thus ionised. The resulting plasma jet is then used as a carrier jet for the aluminium oxide powder fed into the device. The aluminium oxide powder is melted and sprayed at high velocity onto the outer or inner ring. The base material is roughened prior to application of the oxide layer. The oxide layer is then sealed.

8

47 Plasma spray method



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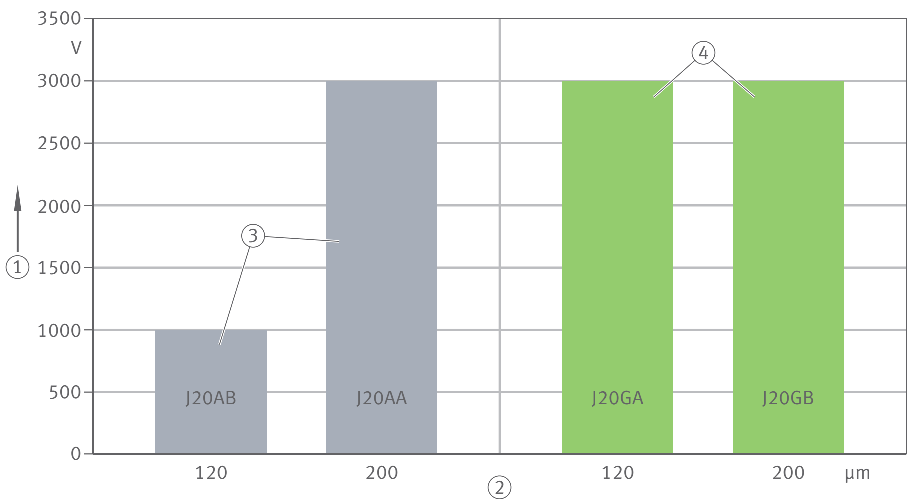
8.4.3 Increased insulation performance with the new J20G coating

Systematic further developments have led to improvements in the insulation properties of rolling bearings with the Insutect A coating. In addition to improving electrical properties in dry environments, a significant increase in performance has also been achieved in damp operation conditions. A comparison with the values for the previous coating is provided in the following figures.

48 Dielectric strength in a dry and damp environment



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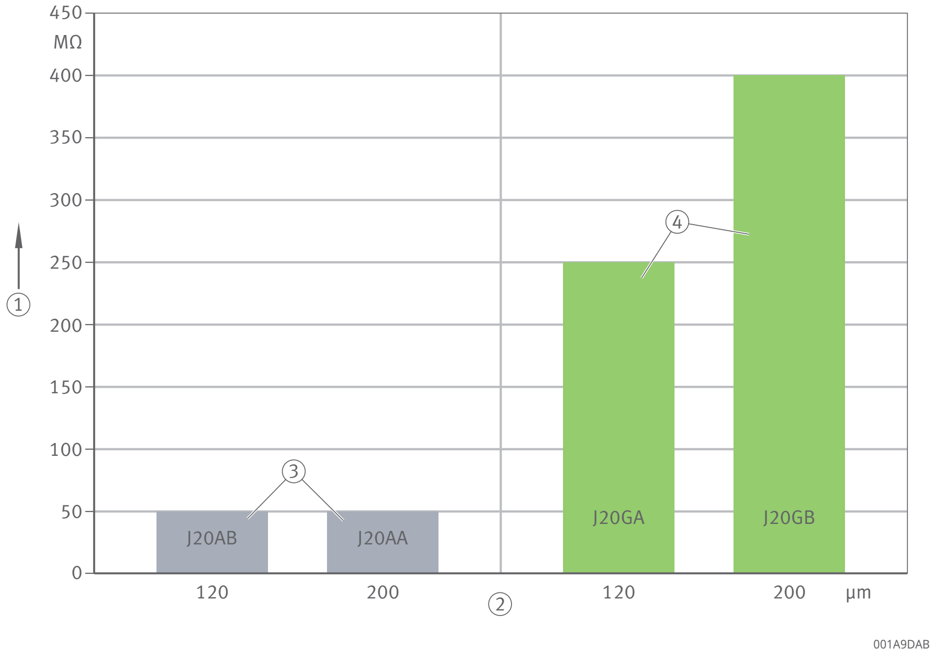
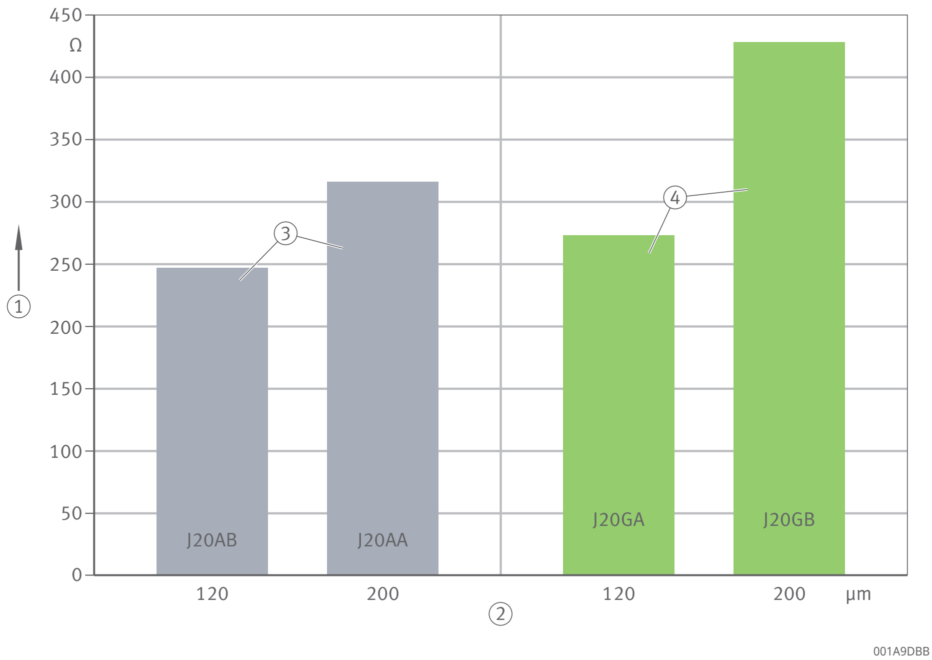
1	Puncture voltage	2	Layer thickness
3	Current coating	4	New coating

49 Ohmic resistance in a dry environment



1	Ohmic resistance	2	Layer thickness
3	Current coating	4	New coating

50 Impedance in a dry and damp environment, bearing type 6314, f = 100 kHz

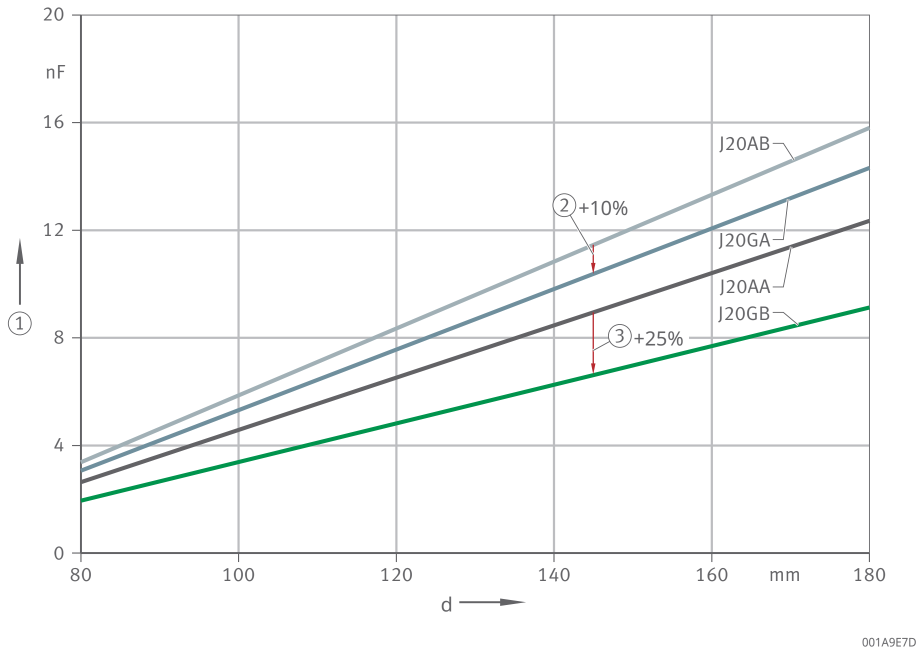


1	Impedance	2	Layer thickness
3	Current coating	4	New coating

8.4.4 Coating parameters

Electrical simulations and calculations are increasingly being used to find the right insulation solution – processes in which the electrical properties of rolling bearings play a key role. The electrical insulating effect of the lubricant film can only be determined if the exact operating parameters are known. Schaeffler offers expert guidance to assist you in this respect. The impedance or capacitance is crucial for the electrical properties of the insulation layer on Insutect A bearings. Guide values for the capacitance of deep groove ball bearings of bearing series 60, 62 and 63 can be found in the following figures. The lowest possible capacitance is necessary in order to reduce the current passage through the bearing to optimum effect. The values shown can also be used as an initial approximation for other designs with the same external dimensions, e.g. for cylindrical roller bearings in the same dimension range. The values for capacitance are also valid for use in damp environments, e.g. 90 % relative humidity.

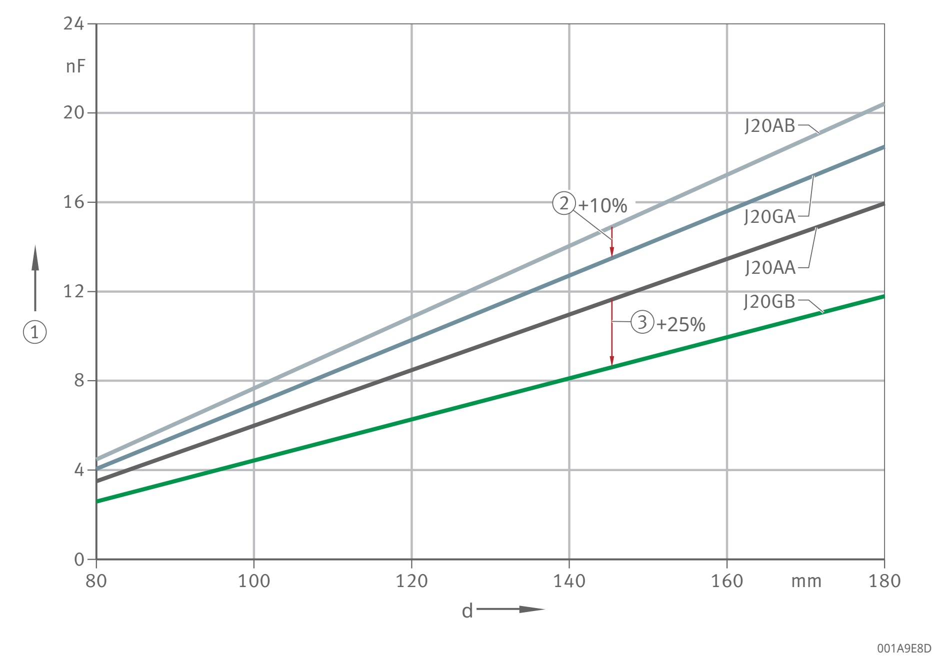
51 Guide values for the rolling bearing capacitance of bearing series 60



1	Electrical capacitance	2	Improvement in capacitance of J20GA over J20AB of 10%
3	Improvement in capacitance of J20GB over J20AA of 25%		

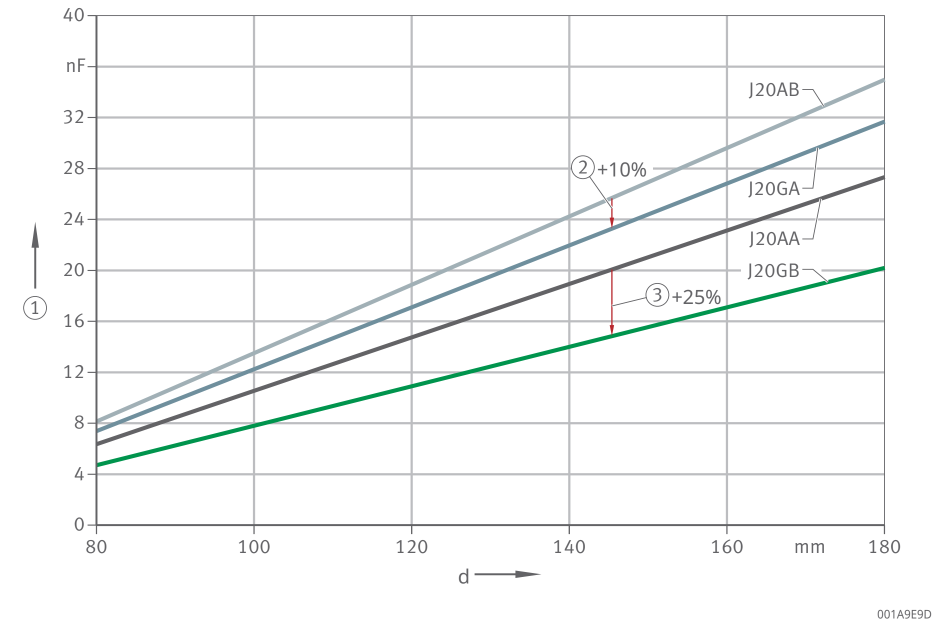
d	mm	Bore diameter

52 Guide values for the rolling bearing capacitance of bearing series 62



1	Electrical capacitance		2	Improvement in capacitance of J20GA over J20AB of 10%
3	Improvement in capacitance of J20GB over J20AA of 25%			
d	mm	Bore diameter		

53 Guide values for the rolling bearing capacitance of bearing series 63



1	Electrical capacitance		2	Improvement in capacitance of J20GA over J20AB of 10%
3	Improvement in capacitance of J20GB over J20AA of 25%			
d	mm	Bore diameter		

15 Parameters for coating types in accordance with Insutect A J20G

Parameter		J20GA	J20GB	J20GI
Bearing coating	–	Outer ring	Outer ring	Inner ring
Layer thickness	μm	120	200	120
Operating environment	–	Dry, damp	Dry, damp	Dry, damp
Electric strength	DCV	3000	3000	3000
Ohmic resistance	MΩ	250	400	250
Impedance, 6314, f = 100 kHZ	Ω	273	428	583
Possible inside diameters	mm	–	–	≥ 70
Possible outside diameters	mm	70 ... 800	70 ... 800	... 800

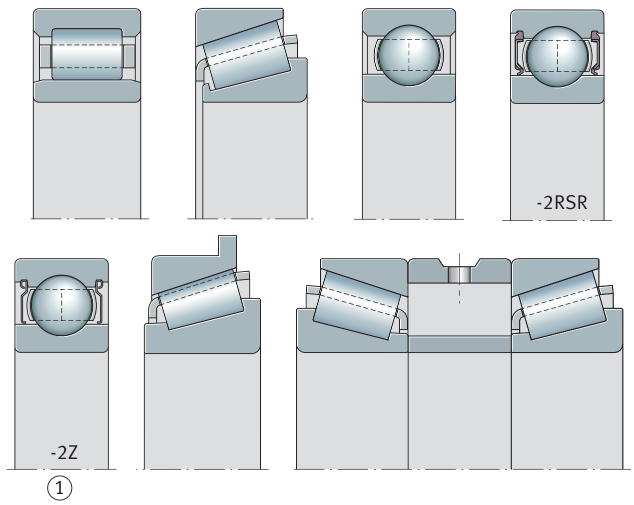
The bearing surface of coated rings is cylindrical. If grooves or lubrication holes are present, we recommend that you contact the responsible Application Engineering department at Schaeffler.

The coated rings are subjected to a 100 % insulation inspection.

8.4.5 Bearing designs with ceramic coatings

The available bearing designs with ceramic coatings are shown in cross-section.

54 Examples of bearing designs



1	Only with J20GI coating

8.4.6 Ordering examples

55 Deep groove ball bearing with coated outer ring

62 14 – 2RSR – J20GA – C3

Bearing series
62 Deep groove ball bearing

Bore code
14 14 · 5 = 70 mm

Sealing
2RSR Sealed on both sides, contact rubber seal

Coating
J20GA Current insulation coating on the outer ring, layer thickness 120 µm

Radial internal clearance
C3 Radial internal clearance

001B190B

56 Cylindrical roller bearing with coated outer ring

NU10 18 – XL – MP – F1 – J20GB – C4

Bearing series	
NU10	Cylindrical roller bearing
Bore code	
18	18 · 5 = 90 mm
Premium brand	
XL	X-life
Cage	
MP	Brass cage, single-piece
Variant	
F1	Variant for traction motors
Coating	
J20GB	Current insulation coating on the outer ring, layer thickness 200 µm
Radial internal clearance	
C4	Radial internal clearance

001B191B

57 Deep groove ball bearing with coated inner ring

63 30 – M – J20GI – C4

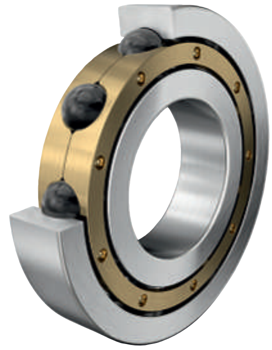
Bearing series	
63	Deep groove ball bearing
Bore code	
30	30 · 5 = 150 mm
Cage	
M	Brass cage
Coating	
J20GI	Current insulation coating on the inner ring, layer thickness 120 µm
Radial internal clearance	
C4	Radial internal clearance

001B192B

8.5 Hybrid bearing

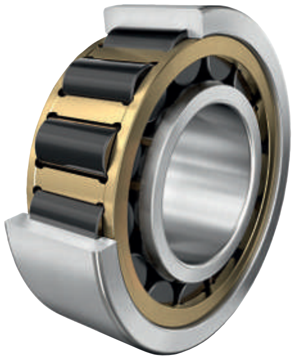
An alternative to Insutect A bearings is FAG hybrid bearings. The rings of the hybrid bearings are made from rolling bearing steel and the rolling elements are made from ceramic. The hybrid bearings are identified by the prefix HC. The rolling elements are highly resistant to wear and perform the function of current insulation. Hybrid bearings are available as ball bearings and as cylindrical roller bearings.

58 Hybrid ball bearing



0019D336

59 Hybrid cylindrical roller bearing



001A9EE3

Advantages of hybrid bearings

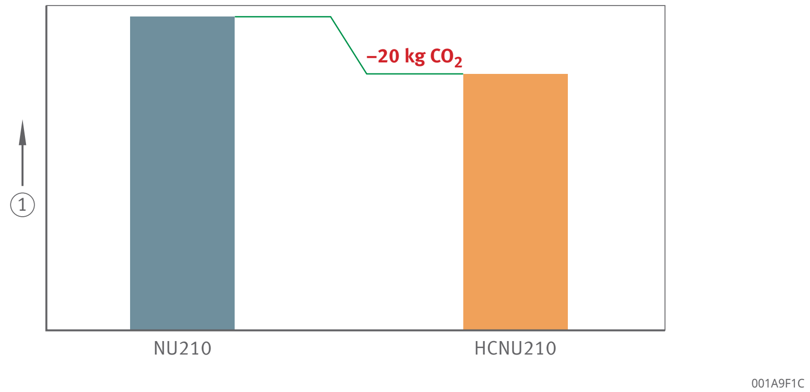
Hybrid bearings have advantages over ceramic-coated bearings:

Hybrid bearings offer very high resistance to current passage. Their direct current resistance, even at high temperatures, is in the GHz range. A typical value for capacitance is 40 pF and is thus lower by a factor of 100 than for bearings with a ceramic coating.

Hybrid bearings allow higher speeds at lower friction and thus lower temperatures in operation. The low weight of the rolling elements leads to lower friction. Less friction reduces CO_2 emissions in the application. A comparison between the CO_2 emissions of a standard cylindrical roller bearing and a hybrid cylindrical roller bearing operating in the drive of a high-speed train over a period of one year and an approximate distance of 600,000 km shows a CO_2 reduction of 20 kg.

Hybrid bearings have better emergency running characteristics than standard bearings.

60 Comparison of CO_2 emissions



1	CO_2 emissions

Other characteristics

In comparison with standard bearings, hybrid bearings have the following characteristics:

- comparable basic dynamic load ratings C_r in accordance with ISO 20056-1
- comparable basic static load ratings C_{0r} in accordance with ISO 20056-2
- 20 % higher limiting speeds n_G

Hybrid bearings have the same dimensions and are therefore suitable for retrofitting. In addition, hybrid bearings offer twice the grease operating life of standard bearings. Hybrid bearings also offer advantages over standard bearings in terms of lifetime costs. The use of hybrid bearings can optimise product-specific life cycle costs by up to 20 %.

Our Sales Engineers will be pleased to advise you in the selection of the best economic and technical solution.

9 Service products and special solutions

9.1 Housing unit with relubrication facility for traction motors

For rotor bearing arrangements in electric drives, Schaeffler works in consultation with customers to develop and manufacture complete housing units with relubrication facilities that are individually designed in accordance with the application and can also be equipped with measurement technology ➤65 | 61.

61 FKB housing unit with relubrication facility for traction motors

00019990

The FKB housing units from Schaeffler offer numerous advantages in the design, manufacture, mounting and maintenance of rotor bearing arrangements. Due to the significantly simplified design of the end shields, the work involved in the development of new drive concepts is also reduced. Thanks to their highly developed design, these bearing solutions are extremely easy to maintain.

The rotor bearing arrangement must be matched to the particular operating and ambient conditions of an electric drive concept. Depending on the drive concept, widely differing bearing loads must be taken into consideration in order to achieve electric drives that are efficient, operationally reliable and cost-effective. The lubrication and sealing must be configured such that the bearings are neither undersupplied nor oversupplied with lubricant in any operating status.

Maintenance costs should be kept low. These requirements are fulfilled by the housing units for traction motors.

Depending on the size, a housing unit can be made of any of the following materials:

- spheroidal graphite cast iron EN-GJS-400-15
- steel with a minimum tensile strength R_m of 400 N/mm^2

Due to the customer-specific concept, the housing units can be matched to the specific geometries of the adjacent constructions without significant additional work. They can be produced individually for different standard bearing types and series. As a result, there are no problems in using either standard bearings or electrically insulated bearings in order to prevent damage caused by the passage of current ➤66|62 ➤66|64.

62 Cylindrical roller bearings

00019991

63 Electrically insulated bearings

00017269

64 Hybrid bearings (hybrid cylindrical roller bearings)

0001726B

For special bearings, special solutions that allow continued use of existing products are also available. The housing units developed for grease lubrication have appropriate relubrication facilities, thus reducing the costs and work associated with maintenance.

Special designs with containers for collecting used grease are also possible and can be adapted to the respective customer requirements.

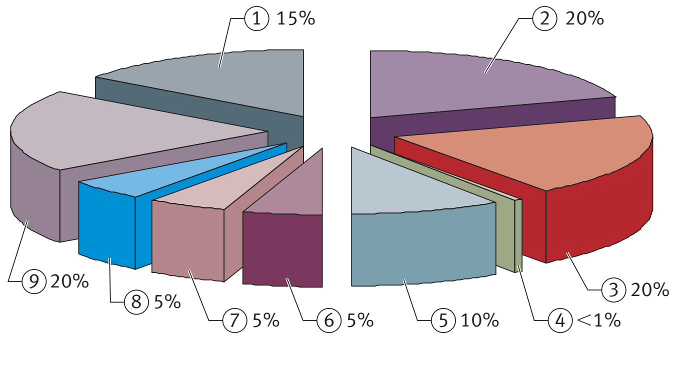
Further customer-specific modifications, such as sensors or measurement technology, can also be realised without major outlay.

9.2 Automated lubricators for traction motors

Almost half of all bearing failures can be traced back to inadequate or incorrect lubrication ➤67 | 65. Bearing failures can be very costly. Bearing damage along the powertrain leads to unplanned and expensive downtime.

In many cases, this results in production downtime and repair costs that not infrequently run to tens of thousands of euros.

65 Causes of failure for rolling bearings, source: Drive technology, 93



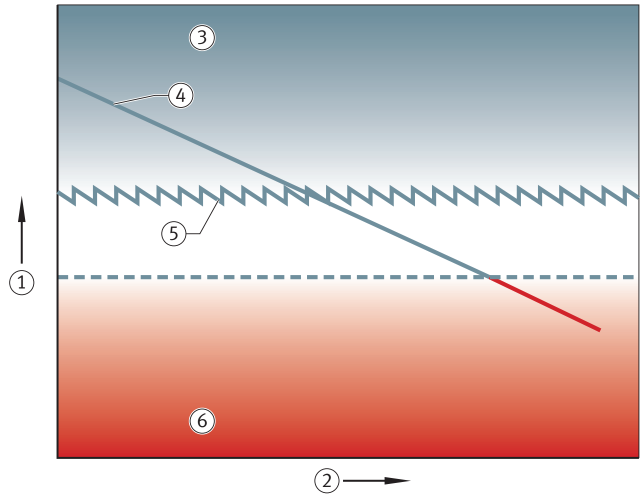
1	Lubricant starvation	2	Unsuitable lubricant
3	Aged lubricant	4	Material defects and manufacturing defects
5	Unsuitable bearing selection	6	Secondary damage
7	Mounting errors	8	Liquid contaminants
9	Solid contaminants		

If automatic lubricators are used, such causes of failure and the associated damage can be largely avoided ➤68 | 66.

Advantages of the lubricators are as follows:

- avoidance of lubricant starvation and thus individual and precise supply of each bearing point with the most suitable lubricant
- fully automatic, maintenance-free operation due to continuous relubrication
- extended life and maintenance intervals
- higher asset availability
- considerable cost savings

66 Automatic relubrication



1	Lubricant amount	2	Operating life
3	Sufficient amount of lubricant	4	for-life lubrication
5	Continuous lubrication	6	Insufficient lubrication

The automatic single-point lubrication systems and multiple-point lubrication systems in the CONCEPT family can supply up to eight lubrication points constantly, precisely and within a wide range of temperatures ➤69 | 67.

Advantages of the lubrication systems:

- universally applicable for grease or oil
- lubricant supply individually adapted to the bearing point
- simple, user-friendly handling
- no need for manual relubrication
- control via integrated timer or external PLC
- output of error messages via display, status LED and multifunction interface

67 Optimal lubrication concept

001B32E2

Further information

TPI 252 | Lubricators |
https://www.schaeffler.de/std/1D4E

9.3 Arcanol rolling bearing greases

Special rolling bearing greases such as Arcanol offer the best conditions for achieving reliable, durable and cost-effective bearing arrangements. As a result, bearings that fail prematurely because they were lubricated with the wrong grease are increasingly a thing of the past.

Schaeffler has been working with renowned lubricant manufacturers for many years to develop lubricants that are particularly suitable for rolling bearings. Before a grease can be included in the Arcanol range, it is subjected to a series of tests in the Schaeffler lubricant laboratory, where its properties are examined.

On the lubricant test benches FE8 (testing in accordance with DIN 51819) and FE9 (testing in accordance with DIN 51821), life, friction and wear tests are carried out on the greases in rolling bearings. Only greases with the best characteristics are then selected to undergo the subsequent tests under simulated field conditions in far more complex rolling bearing test rigs. If the results meet the requirements of the stringent Schaeffler specifications, the grease then receives the Arcanol seal of approval. In addition, we test every single batch to ensure the uniform quality of the product. It is only after this final test that approval can be given for designation of the grease as Arcanol.

The range is graduated such that almost all areas of application can be optimally covered using these greases. Overview of the chemical and physical data, application areas and suitability of these greases ➤70 | 17.

Advantages of the Arcanol greases:

- 100% testing to guarantee constant quality and thus a longer rolling bearing life
- development and field tests carried out by application and tribology experts
- close cooperation at all times with well-known lubricant manufacturers
- optimal design for rolling bearing applications
- reduced costs through
  - longer maintenance intervals
  - lower friction
  - less wear and bearing damage
  - considerably longer bearing operating life
  - increased operational reliability and safety

16 Classification and compositions of rolling bearing greases Arcanol

Arcanol	Classification	Thickener	Base oil
MULTI2	Ball bearing grease, low-noise for D ≤ 62 mm	Lithium soap	Mineral oil
MULTI3	Standard ball bearing/radial insert ball bearing grease for D > 62 mm	Lithium soap	Mineral oil
MULTITOP	Universal high-performance grease	Lithium soap	Partially synthetic oil
TEMP90	Rolling bearing grease, low-noise, to +160 °C	Polycarbamide	Mineral oil
TEMP110	Universal grease for higher temperatures	Lithium complex soap	Partially synthetic oil
TEMP120	Grease for high temperatures and high loads	Polycarbamide	Synthetic oil
TEMP200	Rolling bearing grease for T > +150 °C to +260 °C	PTFE	Alkoxyfluoro oil
LOAD150	Multi-purpose grease for automotive applications, high-performance grease for line contact	Lithium-calcium soap	Mineral oil
LOAD220	Heavy-duty grease, wide speed range	Lithium-calcium soap	Mineral oil
LOAD400	Grease for high loads, shocks	Lithium-calcium soap	Mineral oil
LOAD460	Grease for high loads, vibrations, low temperatures	Lithium-calcium soap	Mineral oil
LOAD1000	Grease for high loads, shocks, large bearings	Lithium-calcium soap	Mineral oil
SPEED2,6	Standard spindle bearing grease	Lithium soap	Synthetic oil
FOOD2	Grease with foodstuffs approval	Aluminium complex soap	Synthetic oil
VIB3	Grease for oscillating motion	Lithium complex soap	Mineral oil
CLEAN-M	Clean room grease, grease resistant to radiation	Polycarbamide	Ether oil
MOTION2	High-performance grease paste for oscillating applications and plain bearing arrangements	Lithium soap	Synthetic oil
MOUNTING-PASTE2	Anti-seize paste with additional protection against tribo-corrosion for bearings, screws and bolts	Lithium soap	PAO oil

17 Properties of Arcanol rolling bearing greases as a function of the base oil

Arcanol	Operating temperature range	Upper continuous limit temperature	NLGI grade	Characteristic speed	Kinematic viscosity	
		$T_{upperlimit}$		$n \cdot d_M$	At +40 °C	At +100 °C
	°C	°C		$min^{-1} \cdot mm$	mm^2/s	mm^2/s
MULTI2	–30 ... +120	+75	2	500000	110	11
MULTI3	–30 ... +120	+75	3	500000	80	10
MULTITOP	–50 ... +140	+80	2	800000	82	12,5
TEMP90	–40 ... +160	+90	3	700000	148	15,5
TEMP110	–35 ... +160	+110	2	500000	130	14
TEMP120	–30 ... +180	+120	2	300000	400	40
TEMP200	–30 ... +260	+200	2	300000	550	49
LOAD150	–20 ... +140	+95	2	500000	160	15,5
LOAD220	–20 ... +140	+80	2	500000	245	20
LOAD400	–40 ... +130	+80	2	400000	400	27
LOAD460	–40 ... +130	+80	1	400000	400	25,8
LOAD1000	–20 ... +130	+80	2	300000	1000	38
SPEED2,6	–40 ... +120	+80	2 ... 3	2000000	25	6
FOOD2	–30 ... +120	+70	2	400000	150	18

Arcanol	Operating temperature range	Upper continuous limit temperature	NLGI grade	Characteristic speed	Kinematic viscosity	
		$T_{upperlimit}$		$n \cdot d_M$	At +40 °C	At +100 °C
	°C	°C		$min^{-1} \cdot mm$	mm^2/s	mm^2/s
VIB3	–30 ... +150	+90	3	350000	170	14
CLEAN-M	–30 ... +180	+90	2	850000	103	12,8
MOTION2	–40 ... +130	+75	2	500000	50	8
MOUNTING-PASTE2	–30 ... +150	+120	2	–	100	13,5

9

10 Checklist

Scope of performance

FAG housing unit ______________________

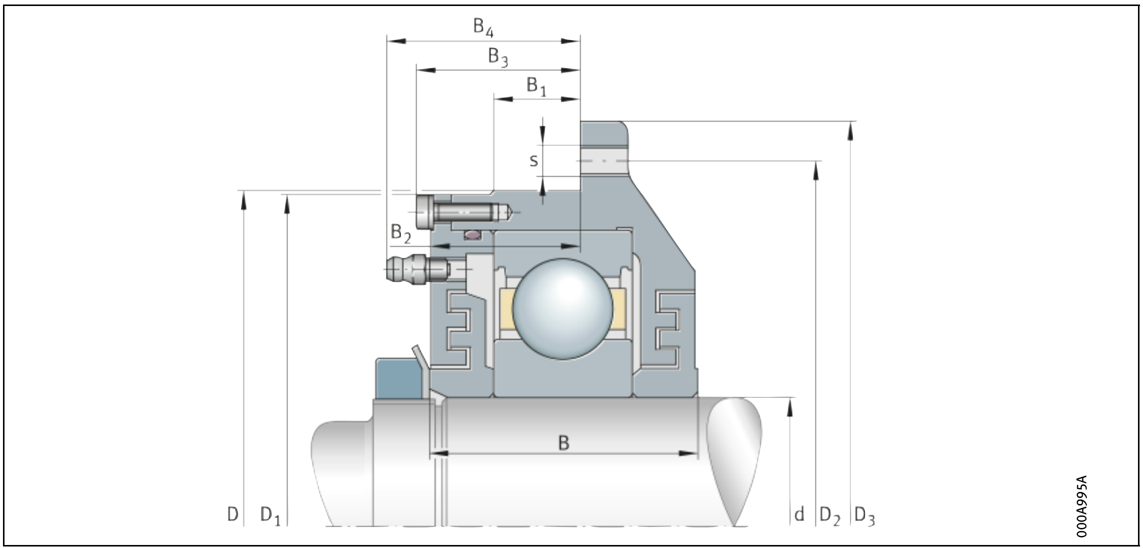
Rolling bearing ______________________

Tab washer ______________________

Locknut ______________________

Dimensions

B1	______	mm	∅d	______	mm
B2	______	mm	∅D	______	mm
B3	______	mm	∅D1	______	mm
B4	______	mm	∅D2	______	mm
			∅D3	______	mm



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