

FAG



Rolling Bearings in Traction Motors

Technical Product Information

SCHAEFFLER

Foreword

Schaeffler is, through its brands INA and FAG, one of the world's leading manufacturers of rolling 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 axlebox bearings together with housings, 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 more than 140 years of experience in the rail sector. We can thus offer comprehensive technical expertise, the highest quality and precise matching of products to any area of 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 Schweinfurt site is recognised and certified as a test facility for rail vehicle bearing arrangements by the Federal German Railway Authority.

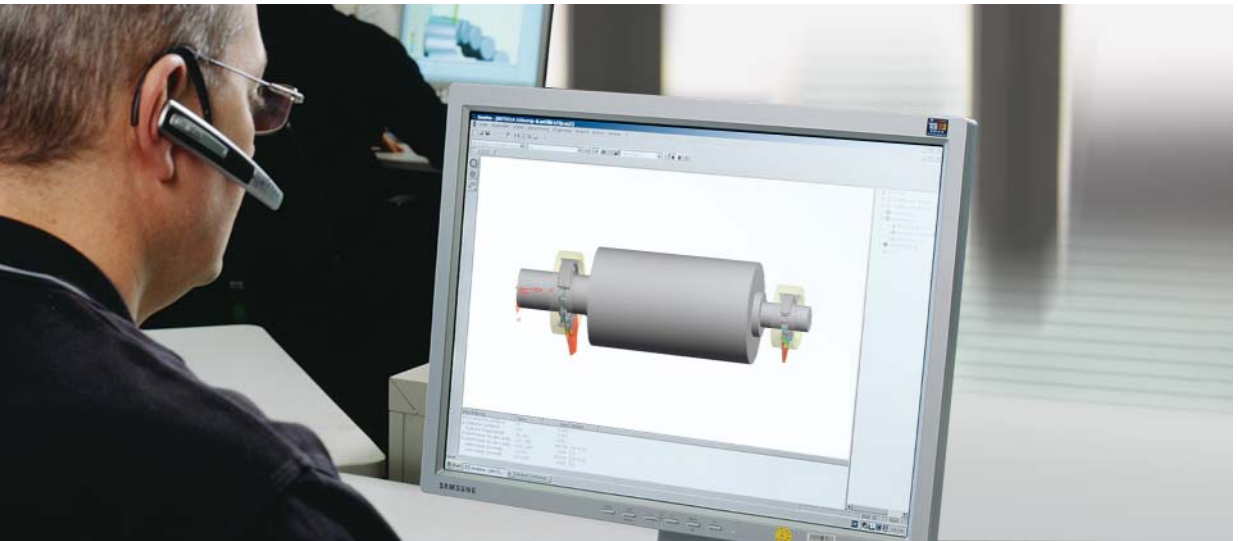
In addition, it is authorised by the DAP (German Accreditation System for Testing GmbH) to carry out testing in the area "Performance tests on axlebox bearings – Railway applications" in accordance with DIN EN ISO/IEC 17025:2000.

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 the skilled and economical reconditioning of railway bearings.

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Factors influencing the design of the bearing arrangement

Influencing factors
Design of traction motor bearing arrangements

Factors influencing the design of the bearing arrangement

Influencing factors

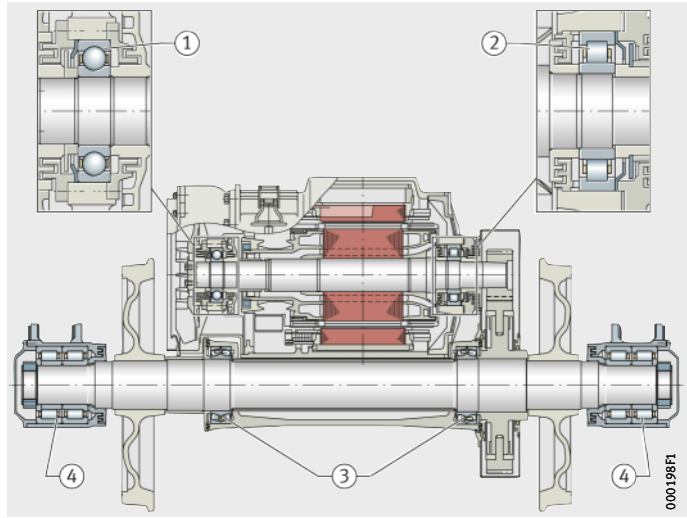
A modern electric drive for the rail industry is subject to certain fundamental requirements such as cost-effectiveness, reliability, operational security, adequate operating life and low maintenance outlay. The fulfilment of these basic requirements is heavily influenced by the forces present, additional loads and ambient influences. Since the rolling bearings belong to the wear parts of the drive, all the particular sector-specific requirements must be taken into consideration and fulfilled. The bearing arrangement of the rotor shaft in the electric drive must be carefully matched to the particular operating and ambient conditions. Depending on the drive concept, differing load cycles must be taken into consideration in order to achieve a bearing arrangement that is efficient, operationally secure and cost-effective. The lubrication and sealing must be configured such that the bearings are neither undersupplied nor oversupplied with lubricant in any operating mode, *Figure 1*, page 7.

For the bearing arrangement of an electric drive, the designer must take account of the following influencing factors:

- load
- speed
- current flow, current passage
- design envelope
- shaft arrangement
- oscillation behaviour
- influence of temperature
- rating life requirement
- static load safety
- ambient conditions
- friction
- lubrication
- maintenance
- mounting and dismounting.

- ① Bearing on non-drive side
- ② Bearing on drive side
- ③ Nose bearing, in this case spherical roller bearing
- ④ Axlebox bearing

Figure 1
Traction motor bearing arrangement
in an electric powered unit



Factors influencing the design of the bearing arrangement

Design of traction motor bearing arrangements

The computer-aided design of electric motor bearing arrangements is generally carried out in accordance with the material fatigue theory. The operating life of the bearings in standard motors or series motors is in most cases identical with the grease operating life since many such motors, depending on the axis height, are fitted with sealed bearings with “for life” lubrication. In this case, the rating life of the bearing arrangement is determined by the grease operating life.

In traction motors, the bearing size is determined exactly as in the case of the standard motor with the aid of the normal fatigue life calculation method. As is normal in the sector, this is indicated not in hours but in kilometres travelled. The rating life of the bearings in a traction motor is specified by means of the fatigue life of the bearings, since these are generally relubricated.

When defining a traction motor bearing arrangement, account must be taken of not only its calculated rating life but also the normally specified or targeted maintenance interval. Since the largest possible bearing is required on the basis of the calculated rating life but a small bearing is required on the basis of the speed suitability and lubrication interval, it is often necessary to find an acceptable compromise in the design process.

In the design of the traction motor bearing arrangement, particular attention must therefore be paid to:

- the speed suitability of the bearings and the grease
- the minimum load on the bearings
- the axial load carrying capacity when using a cylindrical roller bearing as a locating bearing
- the temperature and oscillation stability of the grease
- the selection of the suitable cage guidance type and the cage design.

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



Bearing types

Deep groove ball bearings
Four point contact bearings
Cylindrical roller bearings

Bearing types

In traction motors, the following bearings are used for the support of electric drives:

- deep groove ball bearings
- four point contact bearings, as a special solution or project-specific requirement
- cylindrical roller bearings.

All bearings of the basic design have normal tolerances PN. In addition, bearings with restricted tolerances P6 and P5 are available. This is indicated in the bearing designation by an appropriate suffix.

The internal clearance of the bearings is carefully selected and, in most cases, moves in the range C4 (Group 4) or C5 (Group 5). A special radial internal clearance may be realised for a specific project and this is explicitly stated at the end of the bearing designation.

The axial internal clearance of the NUP, NJ and HJ design is also regulated by appropriate standards and provides an increased axial internal clearance.

The cage design is based accordingly on the size and application. In preference, cylindrical roller bearings have single-piece solid section cages (guided by the rolling elements) made from brass or bronze. For ball bearings, brass cages of a two-piece riveted design are used that are normally guided by the rolling elements.

In order to prevent current damage, Schaeffler offers appropriate solutions for current insulation, see the pages starting page 69.

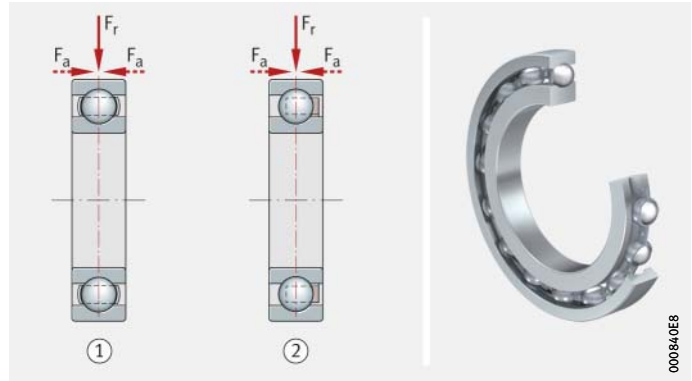
All cylindrical roller bearings with the suffix F1 have an additional standard identifier that represents the characteristic of a traction motor bearing.

Deep groove ball bearings

Single row deep groove ball bearings can support radial and axial forces and are suitable for high speeds, *Figure 1*. Deep groove ball bearings are not separable. Due to its versatile usability and its favourable price/performance ratio, the deep groove ball bearing is the most widely used bearing type.

- F_r = radial load
 F_a = axial load
- ① Bearing with sheet metal cage
 - ② Bearing with plastic cage

Figure 1
Deep groove ball bearings



In traction motors, deep groove ball bearings usually have normal tolerances and have a radial internal clearance in accordance with Group 4 or Group 5. The nominal contact angle α_0 is 0° . In the case of axial load and an increased internal clearance, the contact angle can increase to 20° . The support of the contact ellipse should therefore always be checked.

The speed suitability is high to very high.

Cage

For electric drive applications, a two-piece brass cage guided by the rolling elements is normally used. Since plastic cages have low temperature suitability, which in the case of TVP2 standard cages is $+120^\circ\text{C}$, they are seldom used.

Heat treatment

Due to heat treatment, the deep groove ball bearings for traction motors are dimensionally stable up to $+150^\circ\text{C}$, without any additional notation by means of a suffix. Further dimensional stabilisations are possible, but these require explicit ordering. Dimensional stabilisations are indicated by means of suffixes.

Bearing types

Four point contact bearings

Four point contact bearings are single row angular contact ball bearings that can support axial forces in both directions and low radial forces, *Figure 2*. The inner ring of four point contact bearings is split, which allows a large number of balls to be accommodated. The high load carrying capacity in an axial direction is achieved as result of the large number of balls, the high raceway shoulders and the contact angle of 35° .

- F_r = radial load
 - F_a = axial load
 - α = contact angle
 - M_1, M_2 = centres of curvature of outer ring raceway
- ① Four point contact bearing with retaining slot and split inner ring
 - ② Raceway geometry

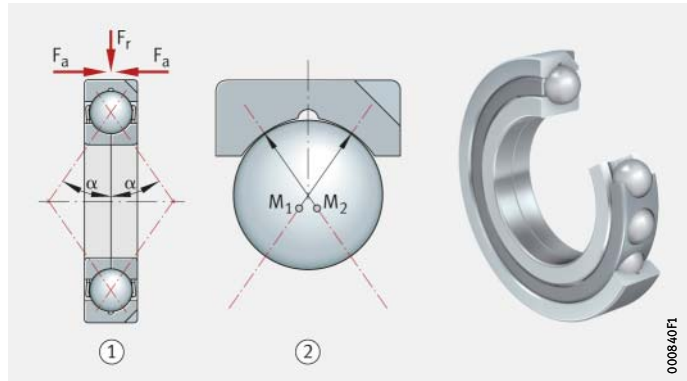


Figure 2
Four point contact bearings

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

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

For operationally secure application, a continuously acting axial force is a prerequisite.

Cage If four point contact bearings are used in traction motors, these are normally lubricated using oil and have a single-piece brass cage guided on the outer rib. This cage type is normally unsuitable for grease lubrication.

Heat treatment The four point contact bearings for traction motors are heat treated such that they are dimensionally stable up to $+150^\circ\text{C}$.

Cylindrical roller bearings

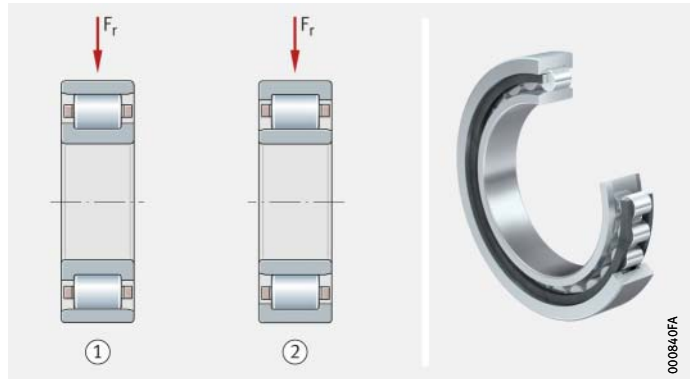
Cylindrical roller bearings have a high radial load carrying capacity. In traction motors, they are predominantly used on the drive side.

Non-locating bearings

The various designs of single row cylindrical roller bearings differ in the arrangement of the ribs, *Figure 3*. The design NU has two ribs on the outer ring and a ribless inner ring. In the case of design N, the inner ring has two ribs; the outer ring is ribless. Cylindrical roller bearings of the design NU and N are used as non-locating bearings; they facilitate length compensation within the bearing.

- F_r = radial load
- ① Cylindrical roller bearing N
 - ② Cylindrical roller bearing NU

Figure 3
Non-locating bearings



Bearing types

Semi-locating and locating bearings

Cylindrical roller bearings NJ have two ribs on the outer ring, one rib on the inner ring and are semi-locating bearings, *Figure 4*. 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.

As locating bearings for the support of alternating axial forces, cylindrical roller bearings NUP are fitted. They have two ribs on the outer ring; the inner ring has a rigid rib 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.

- F_r = radial load
 F_a = axial load
- 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)

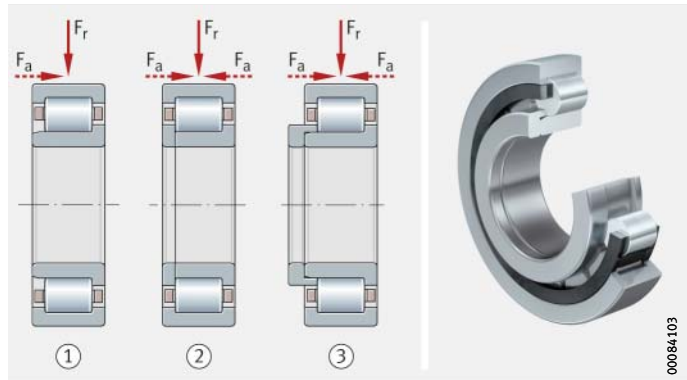


Figure 4
Semi-locating and locating bearings

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

Tilting

The modified line contact between the rollers and the raceways prevents edge stresses and allows a certain self-adjustment facility of the single row bearings.

The following values should not be exceeded:

- 4' in bearings of series 10, 19, 2, 3, 4
- 3' in bearings of series 22, 23.

Cage

For traction motors, solid brass or bronze cages are used. The type of guidance depends on the bearing type and type of lubrication.

Heat treatment

As standard, the cylindrical roller bearings for traction motors are dimensionally stable up to at least +150 °C.

Axial load carrying capacity

The axial load carrying capacity is dependent on:

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



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

The permissible axial load $F_{a \text{ per}}$ must not be exceeded, in order to avoid impermissibly high temperatures.

The axial limiting load $F_{a \text{ max}}$ must not be exceeded, in order to avoid impermissible pressure at the contact surfaces.

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

The permissible and the maximum load are calculated with the aid of the following equations:

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

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

$F_{a \text{ per}}$ N
Permissible axial load

k_S –
Factor dependent on the lubrication method, see table, page 16

k_B –
Factor dependent on the bearing series, see table, page 16

d_M mm
Mean bearing diameter $(d + D)/2$, see Catalogue HR 1, Rolling Bearings

n min^{-1}
Operating speed

$F_{a \text{ max}}$ N
Maximum axial load.

Bearing types

Factor k_S for the lubrication method

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

¹⁾ Doped lubricating oils should be used, such as CLP (DIN 51517) and HLP (DIN 51524) of ISO VG grade 32 to 460 and ATF oils (DIN 51502) and gearbox oils (DIN 5112) of SAE viscosity grades 75 W to 140 W.

Bearing factor k_B

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

Skewing of the bearing, for example due to shaft deflection, can lead to alternating stresses on the inner ring ribs. In this case, the axial load must be restricted up to bearing tilting of max. 2 angular minutes to F_{as} .

Permissible axial load:

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

F_{as} N

Permissible axial load up to bearing tilting of max. 2 angular minutes

d_M mm

Mean bearing diameter $(d + D)/2$, see Catalogue HR 1, Rolling Bearings.



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Cage

Loads

Cage types

Cage damage

Cages for high shock and oscillation loads

Cage

The rolling bearings in traction drives that are unsprung or only slightly sprung are subjected to high dynamic additional loads. These additional loads act on the cage in particular. They are very complex and difficult to define, with the result that the rolling bearings in heavily loaded traction drives are normally fitted with solid brass or bronze cages.

The most important functions of the cage are:

- to separate the rolling elements from each other, in order to minimise friction and heat generation
- to maintain the rolling elements at the same distance from each other, in order to ensure uniform load distribution
- to prevent the rolling elements from falling out in bearings that can be dismantled or swivelled out
- to guide the rolling elements in the unloaded zone of the bearing.

Loads

The cage is driven by the rotating rolling element set. Sliding motions occur on the guidance surfaces, which are principally the cage pockets. The guidance forces result from the inherent mass of the cage, possible displacements of the centre of gravity, accelerations and decelerations between the rolling elements and cage. These are normally small. 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 occurring, 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 using grease lubrication, it is almost impossible to prevent a mixed friction regime. The wear occurring as a result is, however, negligibly small in its practical effect on the operating life under the forces occurring in normal cases.

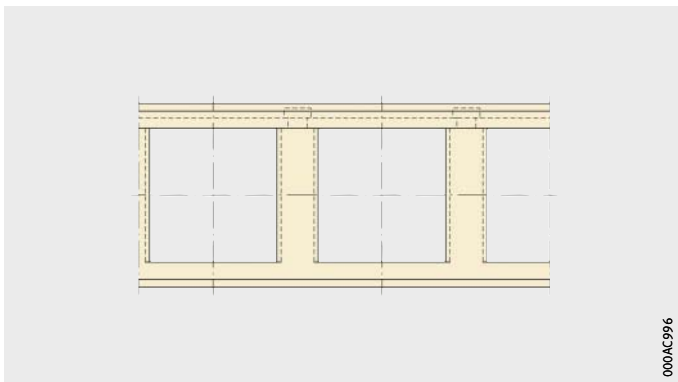
Cage types

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.

The current two-piece solid brass cage is shown in *Figure 1*.

With crosspiece rivets, M1 design

Figure 1
Two-piece solid brass cage



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.

A further design of solid cage is the single-piece window cage, which is produced by milling, *Figure 2*.

The so-called window cage has the highest form stability of all solid cages. It is therefore used in preference for heavily loaded cylindrical roller bearings.

Milled, MPAX design

Figure 2
Solid brass cage



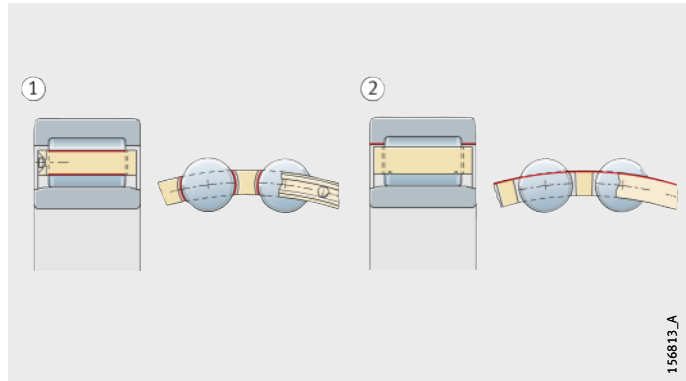
Cage

Guidance types and characteristics

In cage guidance, a distinction is made between rolling element guidance and rib guidance, *Figure 3*.

- ① Rolling element guidance
- ② Rib guidance

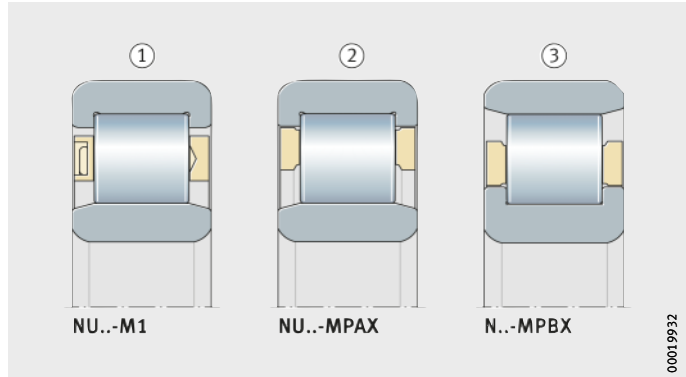
Figure 3
Guidance of cages



In traction motors, the cage is generally guided by the rolling elements and the bearings are normally lubricated using grease, *Figure 4*. As shown by practical experience, rib guidance when using grease lubrication involves the risk of lubricant starvation of the guidance surfaces. Exceptions are found in other areas of application, but require greater outlay for relubrication. Rib-guided cages are used where the bearings are subjected to strong oscillations or extreme speeds. Based on experience, this type of guidance requires oil lubrication of the rolling bearing.

- ① Example of NU220-M1
- ② Example of NU220-MPAX
- ③ Example of N220-MPBX

Figure 4
Cage guidance types



Cage damage

Cage damage and unsatisfactorily short running times of bearings as a result of cage wear are in many cases explained by a cage design that is unsuitable for the relevant operating conditions. The overlay of primary and secondary effects actually 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 effect of such problems may be diminished and, as a result, delayed in time through the use of solid cages. This means that only the secondary action can be countered, while the actual cause remains unknown.

The most frequent causes of problems are:

- disrupted running conditions, caused by bearing rings running while tilted relative to each other
- non-uniform running of the cage and the resulting acceleration and deceleration forces, caused by inadequate contact of the rolling partners due to insufficient load
- inhibited running between the rolling elements and cage under the influence of foreign bodies, a lubricant that is unsuitable or (impermissibly) hardened or possibly overlubrication
- unexpected oscillations and vibrations
- running defects due to high axial forces or radial distortion as a result of insufficient operating clearance
- any type of lubrication defect
- other effects arising from the drive or drive control system.

Cage

Cages for high shock and oscillation loads

In the field of freight locomotives, the cages in cylindrical roller bearings are predominantly subjected to very high shock and oscillation loads. These additional loads are very difficult to identify but, if they are insufficiently known, can lead to suboptimal design that ends in cage breakage and early failure of the drive.

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 oscillation load is shown as an example in *Figure 5*.

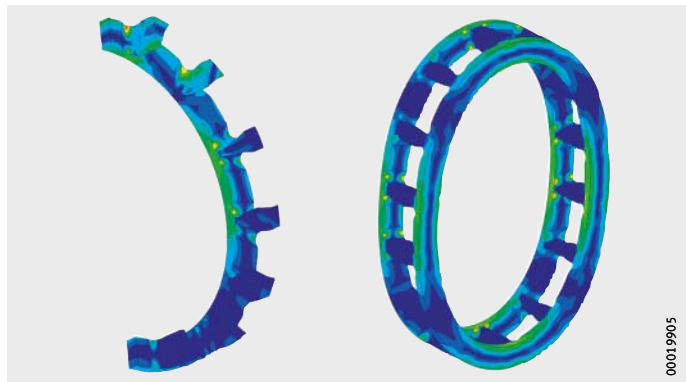
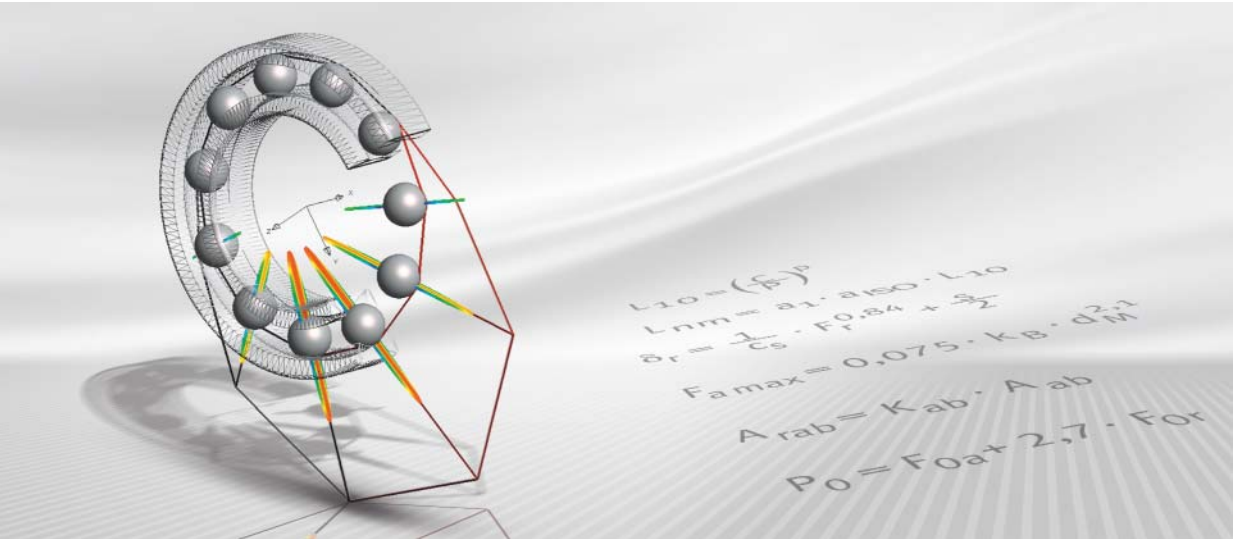


Figure 5
Loads on the cage

In order to counteract these particular additional loads, special cages of an optimised and enhanced capacity design are used. For such cages, special materials in order to additionally increase the operational security are of absolute priority. A distinction must be made for various alloys that are produced in the centrifugal casting or hot rolling process. In particular, purity and pore size are important quality criteria.

The characteristics of a special cage are as follows:

- optimised cage pocket clearance
- optimised contact surface (pocket geometry)
- suitable type of cage guidance
- higher capacity design (cage design)
- increased rigidity
- reduction in notch influences.



Operating parameters

- Minimum load on radial bearings
- Speed parameters of standard bearings
- Thermal stabilisation and residual austenite

Operating parameters

Minimum load on radial bearings

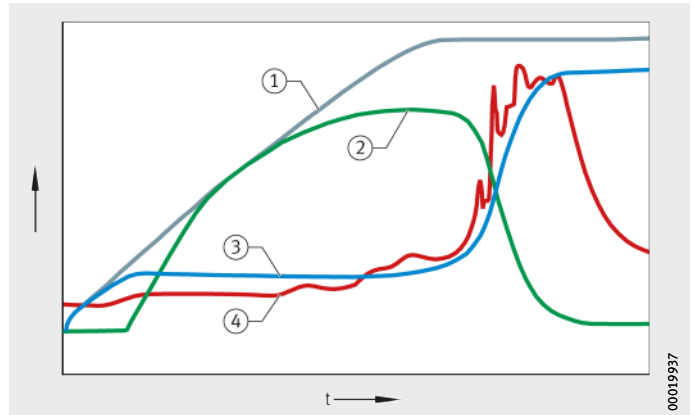
In certain speed and load ranges, slippage occurs between the rolling elements and raceways, i. e., the rolling contact partners have differing circumferential velocities. The primary cause is insufficient bearing load. Due to the lack of friction locking between the inner ring raceway and rolling elements, pure rolling motion is not present. The speed of the rolling elements and cage are lower than in the case of kinematically satisfactory rolling. In practice, slippage is frequently described as cage squealing. It becomes noticeable acoustically by a high clashing noise and can be diagnosed relatively easily. The slippage noise generally disappears during relubrication, but only for a short time.

The slippage itself is less critical, but slippage breakdown is important. Smearing on the raceways and high cage loads are consequences of slippage breakdown. This is defined as an abrupt change in speed from the slippage phase to kinematically acceptable rolling conditions. This rapid change in speed of the cage and rolling elements is due to the transition from hydrodynamic lubrication to mixed friction with metallic contact in the contact area of the rolling elements and raceway.

The results of slippage investigations on the test rig are shown in *Figure 1*, page 25. A cylindrical roller bearing without load is brought up to test speed. The cage speed should run synchronously with the shaft speed under kinematically acceptable rolling. If this is not the case, slippage is present. If slippage breakdown occurs, the noise level increases considerably. The slippage noise is not a result of the smearing on the raceways but an oscillation induced by friction.

- ① Shaft speed
- ② Cage slippage
- ③ Cage speed
- ④ Structure-borne sound development

Figure 1
Slippage investigations
on the test rig



The risk of such slippage is particularly high in the case of bearings under low loads. Based on experience, it is recommended in such cases that the load should be increased; see table. If the load cannot be changed, attempts should be made to fit smaller bearings with a low basic dynamic load rating C_{dyn} .

Minimum load on radial bearings

Bearing type	Minimum load
Ball bearings	$P/C > 0,01$
Cylindrical roller bearings	$P/C > 0,02$
Cylindrical roller bearings, X-life	$P/C_0 < 60$
Full complement roller bearings	$P/C > 0,04$

Operating parameters

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, *Figure 2, page 27*.

The operating conditions are:

- the magnitude and type of load
- the lubrication method, type and quantity of lubricant
- the environment
- the design and accuracy of the adjacent parts
- heat dissipation by the adjacent parts.

The highest permissible speed for a rolling bearing in operation can be restricted by various criteria: the decisive factor is most often the operating temperature increasing with speed.

The speed limits stated in the catalogues are guide values for which speeds can still be reliably achieved with bearings of normal design and accuracy under good mounting conditions and a normal load ratio. The speed achievable in practice is heavily dependent on the factors described above.

In rolling bearing technology, the speed parameter is used as a measurement value on many occasions in this connection.

The speed parameter is the product of the speed n and the mean bearing diameter d_m .

Calculating the speed parameter:

$$n \cdot d_m$$

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

n	min^{-1}
Operating speed or equivalent speed	
d_m	mm
Mean bearing diameter	
D	mm
Bearing outside diameter	
d	mm
Bearing inside diameter.	

With grease lubrication, the maximum speeds achievable are of the order of magnitude of $1 \cdot 10^6 \text{ min}^{-1} \cdot \text{mm}$ for radial ball bearings and radial cylindrical roller bearings.

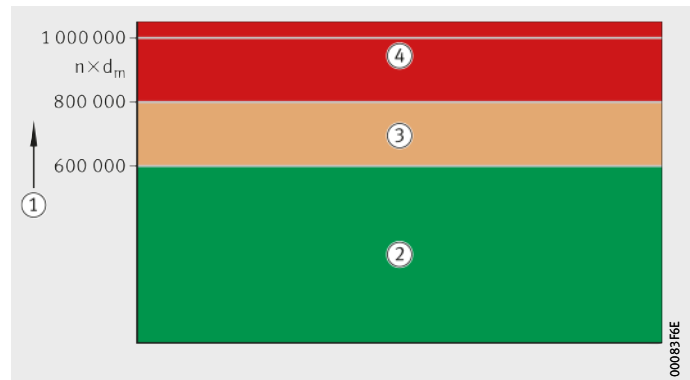
If higher speeds are required, a transition to oil lubrication is necessary.

The ranges of speed limits are shown schematically in *Figure 2*. Standard deep groove ball bearings or cage-guided cylindrical roller bearings can be operated in the green zone without any reservations. If higher speed parameters are required (yellow or red zone), the following parameters must be additionally investigated or adjusted respectively:

- radial internal clearance
- accuracy class
- rolling elements (size, material)
- machining tolerances of the adjacent parts
- cage (material, type and guidance)
- lubrication (grease or oil)
- type of lubrication using oil (recirculating oil, oil mist or oil injection).

- ① Speed parameter
- ② Standard range
- ③ Expanded range
- ④ HIGH SPEED range

Figure 2
Guide values for speed limits
of deep groove ball bearings and
cylindrical roller bearings



The actual limiting speeds of the individual bearings are given in the dimension table in Catalogue HR 1, Rolling Bearings.

Operating parameters

Thermal stabilisation and residual austenite

At a certain operating temperature, the dimensions of the rolling bearings must remain identical even after passing through the entire operating temperature range. The dimensional instability of hardened but incorrectly post-treated rolling bearing steel would, at higher bearing operating temperature, cause larger dimensional changes within a short space of time.

The negative consequences of dimensional changes would be:

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

The dimensional changes are caused by changes in the structural components of residual austenite and martensite. It is not only the influence of temperature and time but also the influence of load that leads hardened rolling bearing steel to undergo, on the one hand, a volume increase due to the decomposition of residual austenite and, on the other hand, to a volume decrease due to the precipitation of carbon in the martensite. The overall change is the result of overlay between the two subprocesses.

Dimensionally stable rolling bearings therefore require, during production, differentiated tempering treatment after hardening. Tempering forestalls the decomposition of residual austenite and the precipitation of carbon in the martensite.

The designation of the dimensional stability stages and the operating temperatures is regulated in DIN 623-1, however the treatments for this purpose are left to the discretion of the rolling bearing manufacturer.

As standard, the rolling bearings for traction motors are heat treated such that they can be used up to an operating temperature of +150 °C and, depending on the type, up to +200 °C. At or above an operating temperature of +120 °C, rolling bearings require special heat treatment. The different suffixes and the associated maximum operating temperatures are shown in the table, page 29.

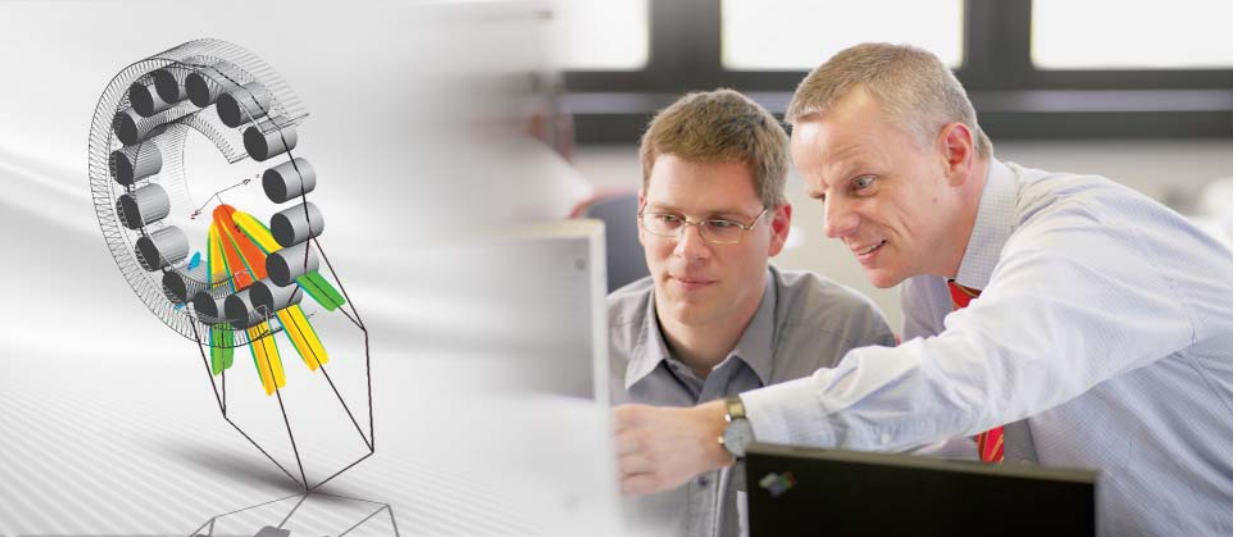
**Operating temperature and
suffixes for dimensionally
stabilised bearings**

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

If decomposition of residual austenite does occur as a result of unacceptably high operating temperatures in bearings that have not been dimensionally stabilised, the dimensional change occurring can be estimated in accordance with the following equation:

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

Δd Dimensional change
 d Bearing diameter
 ΔRA Residual austenite content.
 —
 mm
 %RA



Calculation

- Bearing arrangement and drive concepts
 - Basic rating life
- General calculation equations and aids
 - Loads at the rotor centre of gravity
 - Loads due to the drive concepts
- Additional loads due to shocks and oscillations

Calculation

When defining the bearing type and bearing design as well as in the design of the adjacent construction, particular attention must be paid to certain points.

In traction motors, especially for use in railway applications, tested standard bearings are predominantly used. If special or additional loads are expected, preference is given to special solutions.

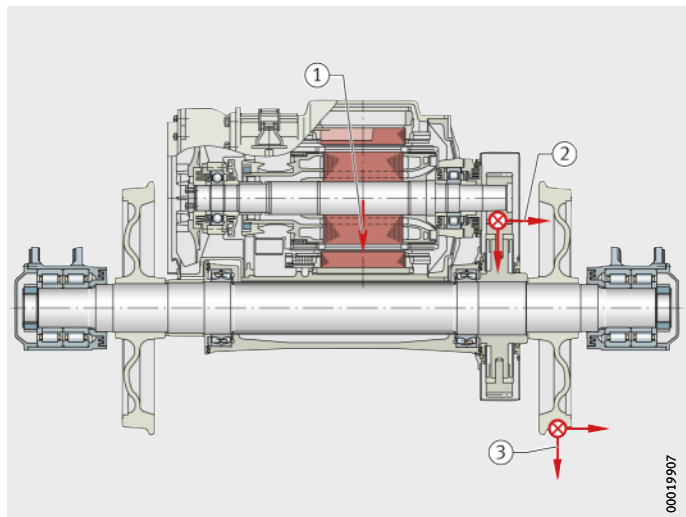
The bearings used most frequently have already been discussed in detail in the chapter Bearing types. The bearing arrangement concept used most frequently is the arrangement comprising a deep groove ball bearing as a locating bearing and a cylindrical roller bearing as a non-locating bearing, *Figure 1*. In contrast to stationary standard motors, both bearings are mounted by means of the inner ring and outer ring on the shaft and in the housing with a tight fit due to the increased oscillation load. The displacement function is ensured with the aid of the cylindrical roller bearing. Particular attention must be paid, however, to the definition of the radial internal clearance. The radial internal clearance of the mounted bearing warm from operation should be as small as possible. The typical loads that must be taken into consideration in design of the traction motor bearing arrangement are shown as examples in *Figure 1*.

The loads include:

- loads at the rotor centre of gravity
- loads due to the coupling or tooth set or propshaft drive
- additional loads due to shocks and oscillations.

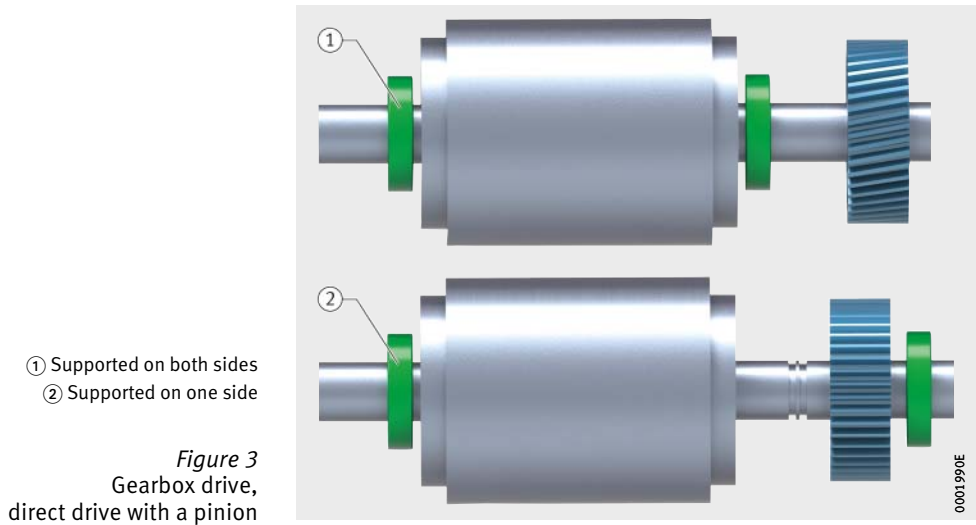
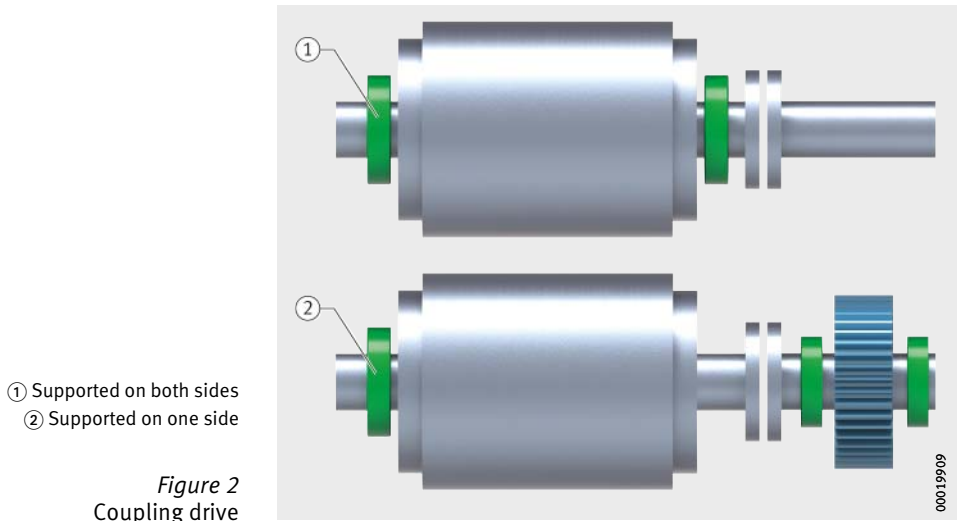
- ① Rotor forces
- ② Forces due to the tooth set
- ③ Additional loads due to shocks and oscillations

Figure 1
Classic bearing arrangement
of a traction motor
with the active loads



Bearing arrangement and drive concepts

Illustrations of the bearing arrangement and drive concepts generally used in traction motors are shown in *Figure 2* and *Figure 3*.



Calculation

Basic rating life

The standardised method for calculating the rating life in accordance with ISO 281 for rolling bearings subjected to dynamic loads is based on material fatigue (pitting formation) as a cause of failure.

The basic rating life L_{10} and L_{10h} is determined as follows:

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

$$L_{10h} = \frac{16\,666}{n} \cdot \left(\frac{C}{P} \right)^p$$

L_{10} 10^6 revolutions

Basic rating life in millions of revolutions that is achieved or exceeded by 90% of a sufficiently large group of apparently identical bearings before the first evidence of material fatigue develops

C N

Basic dynamic load rating

P N

Equivalent dynamic bearing load for radial and axial bearings

p -

Life exponent;

for roller bearings: $p = 10/3$

for ball bearings: $p = 3$

L_{10h} h

Basic rating life in operating hours according to the definition for L_{10}

n min^{-1}

Operating speed.

As is normal in the sector, the rating life is often expressed in kilometres travelled:

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

$L_{10 \text{ km}}$ km

Basic rating life in kilometres

L_{10} 10^6 revolutions

Basic rating life in millions of revolutions that is achieved or exceeded by 90% of a sufficiently large group of apparently identical bearings before the first evidence of material fatigue develops

D_R m

Wheel diameter in metres

i -

Transmission ratio between traction motor and wheel.

General calculation equations and aids

Variable load and speed

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

p	N
Variable load	
n_1, n_2	min^{-1}
Speed	
q_1, q_2	-
Proportion in %	
n_m	min^{-1}
Mean speed.	

Calculation of mean speed:

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

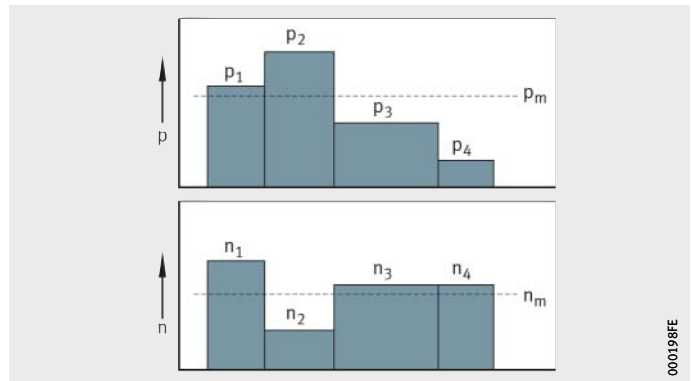
p = load
n = speed

Figure 4
Load and speed

Variable load

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

p	N
Variable load	
p_1, p_2	N
Single load	
q_1, q_2	-
Proportion in %.	



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Calculation

Power

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

P	kW
Power	
M	Nm
Torque	
n	min ⁻¹
Speed.	

Velocity of train

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

v	km/h
Train velocity	
n	min ⁻¹
Motor speed	
D _R	m
Wheel diameter	
i	–
Transmission ratio.	

Transmission ratio

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

i	–
Transmission ratio	
n ₁ , n ₂	min ⁻¹
Speed	
z ₁ , z ₂	–
Number of teeth on a gear	
r ₁ , r ₂	mm
Pitch radius.	

The index 1 indicates the driving wheel,
the index 2 indicates the driven wheel.

Loads at the rotor centre of gravity

The bearing forces in static electrical machinery are calculated from the rotor mass, *Figure 5*. In this case, the safety factor f_z takes account of any unbalance and of the magnetic attraction. In the case of electric traction drives, this factor has also been included as a shock factor and is used in accordance with the suspension type. In theory and calculation, the safety factor $f_z = 1,5 \dots 2,5$ can be used. Shock loads are analysed in particular.

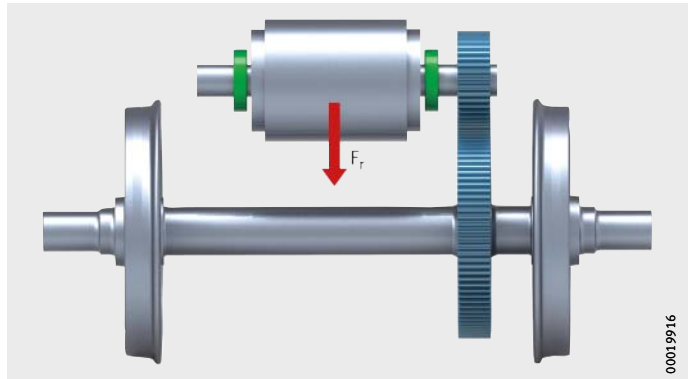


Figure 5
Forces due to the rotor mass

Calculating the radial force on the rotor:

$$F_r = F_g + F_B$$

$$F_r = F_g \cdot f_z$$

F_r	N
Radial force	
F_g	N
Weight force	
F_B	N
Force due to magnetic attraction and unbalance	
f_z	-
Total safety factor $f_z = 1,5 \dots 2,5$.	

Calculating the weight force of the rotor:

$$F_g = m_R \cdot g$$

F_g	N
Weight force	
m_R	kg
Mass of rotor	
g	m/s^2
Gravitational acceleration $g = 9,81 m/s^2$.	

Calculation

Loads due to the drive concepts

In principle, a distinction is made between three drive concepts in the field of traction motors.

These include the classic coupling connections and the spur and helical tooth set concepts as well as the still frequently encountered cardanic drive that is flange mounted directly to the traction motor. In all three drive concepts, further additional forces are generated that act on the bearing arrangement. These must be taken into consideration in the design.

If you have any questions, please contact the Application Engineering function of Schaeffler.

Loads due to the coupling

Elastic or adjustable couplings are the most frequently used connecting elements between the traction motor and gearbox. Coupling outputs generally represent a clearly defined load that comprises an additional radial force and also an axial force component. A bending load must often be taken into consideration.

The coupling force normally applied is half the weight force of the coupling, for which the possible axial force component must be requested from the coupling manufacturer or the traction motor manufacturer, *Figure 6*.

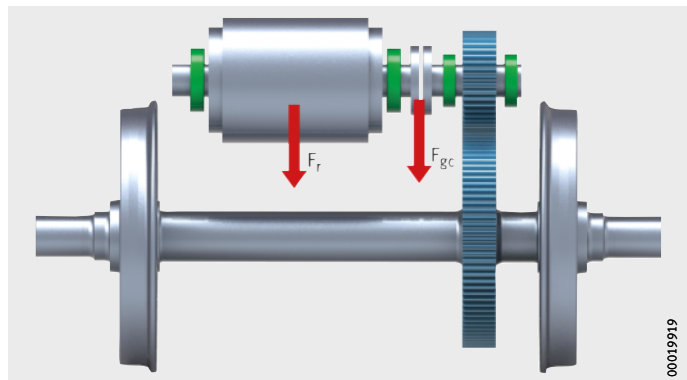


Figure 6
Additional forces due to the coupling

Calculating the radial force on the coupling:

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

F_{rC} N
Radial force on coupling

F_{gC} N
Weight force of coupling.

Calculating the weight force of the coupling:

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

F_{gC} N
Weight force of coupling

m_C kg
Mass of coupling

g -
Gravitational acceleration $g = 9,81 \text{ m/s}^2$.

Calculating the axial force on the coupling:

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

F_{aC} N
Axial force on coupling.

Calculation

Loads due to the tooth set

As a function of the tooth set type, additional tooth set forces act on a bearing that have a decisive influence on the selection and dimensioning of the bearing arrangement, *Figure 7 to Figure 9*.

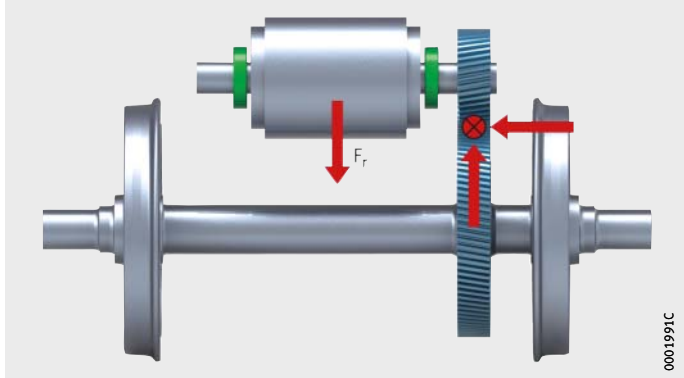


Figure 7
Additional forces due to the tooth set

Spur tooth set

This section examines the tangential, radial and axial forces due to the spur tooth set, *Figure 8 and Figure 9*.

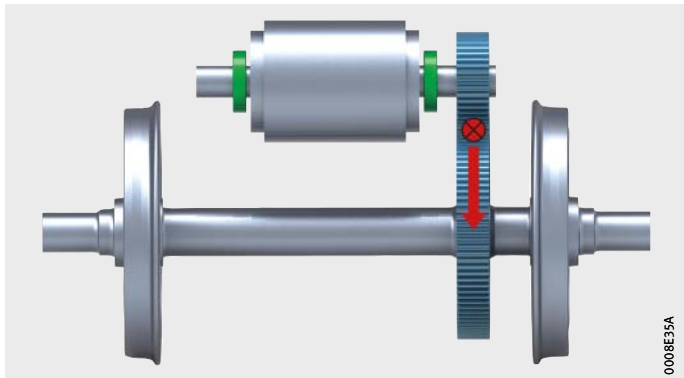
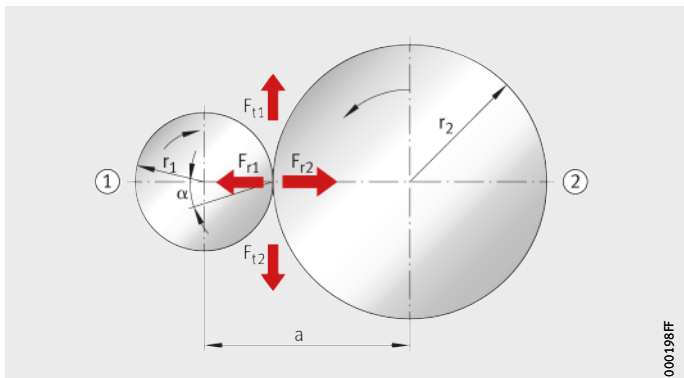


Figure 8
Spur tooth set

- ① Gear 1, driving
- ② Gear 2, driven

Figure 9
Tangential and radial forces due to the spur tooth set



Calculating the tangential force:

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

F_{t1}, F_{t2} N
Tangential force
 M_{d1}, M_{d2} Nm
Torque
 r_1, r_2 m
Gear radius.

Calculating the radial force:

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

F_{r1}, F_{r2} N
Radial force
 F_{t1} N
Tangential force
 α °
Mesh angle.

Calculating the axial force:

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

F_{a1}, F_{a2} N
Axial force.

Calculating the radius of gear 1:

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

r_1 m
Radius of gear 1, driving
 a m
Shaft spacing
 i -
Transmission ratio.

Calculating the radius of gear 2:

$$r_2 = a - r_1$$

r_2 m
Radius of gear 2, driven
 a m
Shaft spacing
 r_1 m
Radius of gear 1, driving.

Calculation

Helical tooth set This section examines the tangential, radial and axial forces due to the helical tooth set, *Figure 10* and *Figure 11*.

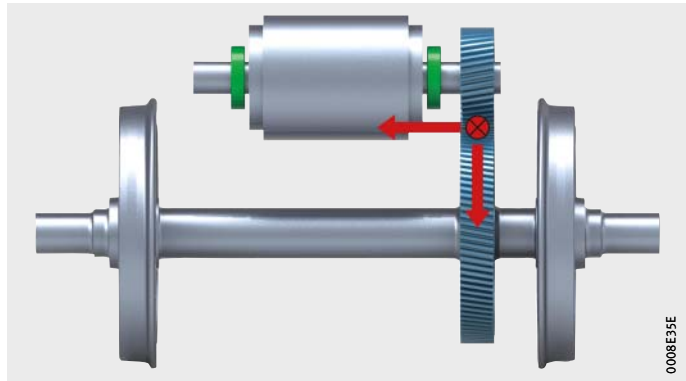
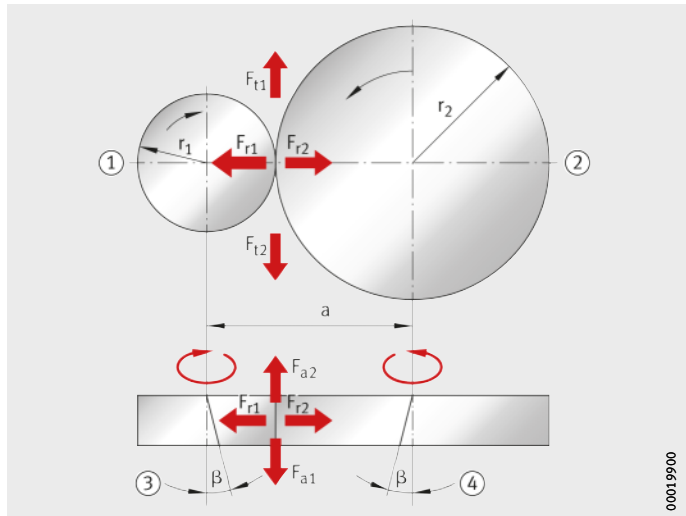


Figure 10
Helical tooth set



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

Figure 11
Tangential, radial and axial forces due to the helical tooth set

Calculating the tangential force:

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

F_{t1}, F_{t2} Tangential force	N
M_{d1}, M_{d2} Torque	Nm
r_1, r_2 Gear radius.	m

Calculating the radial force:

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

F_{r1}, F_{r2} Radial force	N
F_{t1} Tangential force	N
α Mesh angle	°
β Helix angle.	°

Calculating the axial force:

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

F_{a1}, F_{a2} Axial force	N
F_{t1} Tangential force	N
β Helix angle.	°

Calculating the radius of gear 1:

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

r_1 Radius of gear 1, driving	m
a Shaft spacing	m
i Transmission ratio.	-

Calculating the radius of gear 2:

$$r_2 = a - r_1$$

r_2 Radius of gear 2, driven	m
a Shaft spacing	m
r_1 Radius of gear 1, driving.	m

Calculation

Additional loads due to shocks and oscillations

Further additional loads that must be taken into consideration in the bearing arrangement design are loads due to oscillations and shocks, *Figure 12*. The magnitude of the load is relatively difficult to identify and requires measurement. The identification of the oscillation load is often carried out using empirical values from other similar projects or comparative values from the international standard.

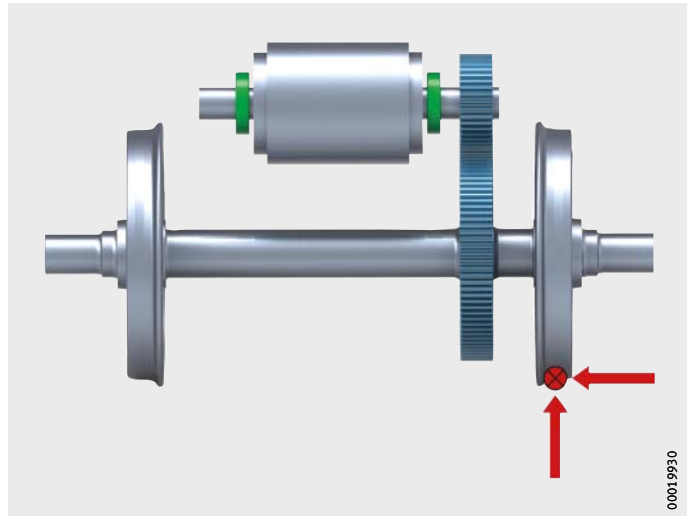


Figure 12
Additional loads due to shocks and oscillations

Calculating the acceleration force:

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

F_{acc} Acceleration force N

m Mass kg

a Acceleration as a function of direction. m/s^2

In the identification of the oscillation load, additional load data due to the mass and the corresponding acceleration in accordance with the valid equation above are calculated and taken into consideration as an additional load in the calculation. In this way, the accelerations (loads) are incorporated in the calculation that act from the three directions x , y and z on the system. The effective duration of these additional loads is defined in consultation with the customer.

If no explicit load data for oscillations are presented or the oscillation accelerations actually occurring are not known, initial design of the bearing arrangement can be carried out using the effective values from DIN EN 61373:2011-04 (VDE 0115-106) or EN 61373:2010.

In bearing design and when taking account of the oscillation accelerations, a distinction is made between the dynamic and the static case.

If the dynamic case applies for identification of the additional load and for taking account of the same in the rating life calculation, the static case is used only for checking security against plastic deformation and for checking the contact ellipse under axial load of a deep groove ball bearing, *Figure 13*. If cylindrical roller bearings of the types NJ and HJ or NUP are used as locating bearings, the maximum axial load carrying capacity of the ribs must be checked.

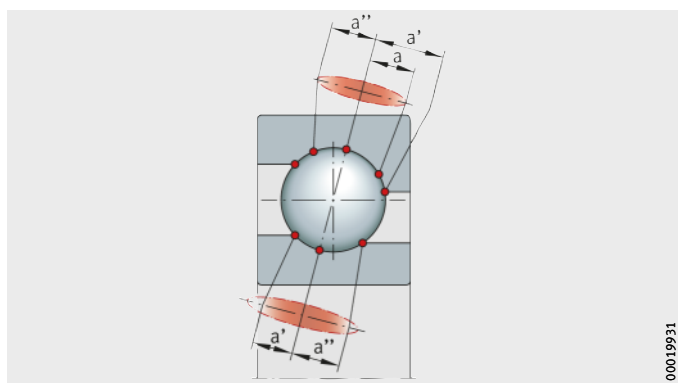


Figure 13
Contact ellipse
of a deep groove ball bearing

Calculation

For an excerpt from DIN EN 61373:2011-04 for dynamic loading and static loading, see tables.

Effective values for dynamic loading

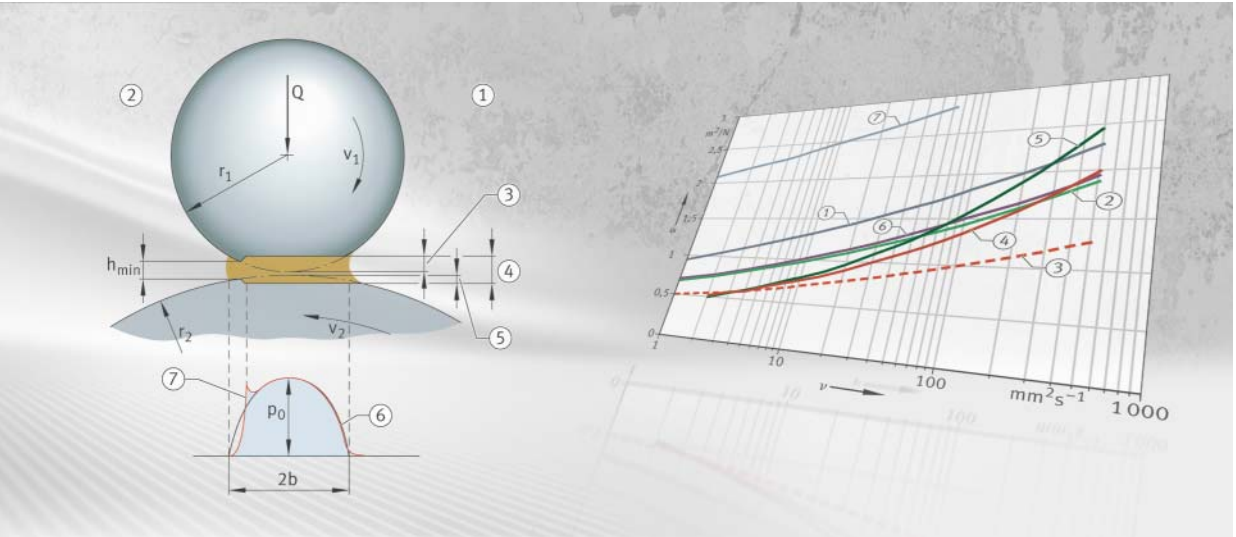
Category	Orientation	Acceleration m/s ²
1 Class A Mounted on the vehicle body	Vertical	0,75
	Transverse direction	0,37
	Longitudinal direction	0,5
Class B Mounted on the vehicle body	Vertical	1,01
	Transverse direction	0,45
	Longitudinal direction	0,7
2 Mounted on the bogie	Vertical	5,4
	Transverse direction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

Effective values for static loading

Category	Orientation	Peak acceleration A m/s ²	Nominal duration D ms
1 Class A and Class B Mounted 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	1 000	6

$k_s > 5/6$, preferably 1,0.

The contact ellipse parameter k_s indicates how far the contact ellipse is actually supported by the raceway. Both sides of the contact ellipse are taken into consideration in the calculation of the contact ellipse parameter k_s . If the raceway is sufficiently wide, this gives a parameter greater than 1. In this case, k_s can be interpreted as the security against truncation of the contact ellipse. The minimum distance between the centre of the contact ellipse and the two shoulders is used here for the calculation of k_s .



Lubrication

- Functions of the lubricant
- Lubrication and friction regimes
- The supply of lubricant to bearings
- Initial greasing and new greasing
- Selection of the suitable lubricant
- Grease operating life

Lubrication

Functions of the lubricant

The principal function of the lubrication of rolling bearings is to prevent wear and premature fatigue and thus to ensure a sufficiently long operating life. Furthermore, lubrication should contribute to achieving favourable running characteristics.

The lubricant is intended to perform the following functions,

Figure 1:

- form a lubricant film on the contact surfaces that is sufficiently capable of supporting loads and thus preventing wear and premature fatigue ①
- dissipate heat in the case of oil lubrication ②
- provide additional sealing for the bearing against external solid and fluid contaminants in the case of grease lubrication ③
- provide damping of running noise ④
- protect the bearing against corrosion ⑤.

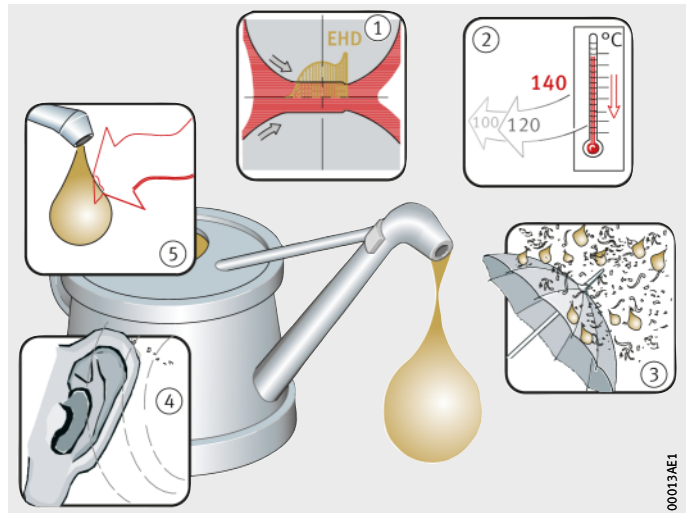
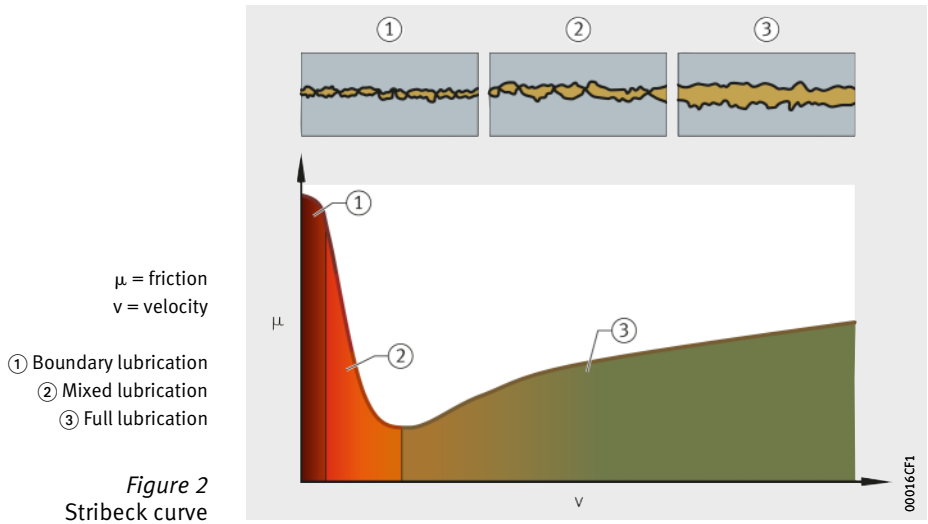


Figure 1
Functions of the lubricant

Lubrication and friction regimes

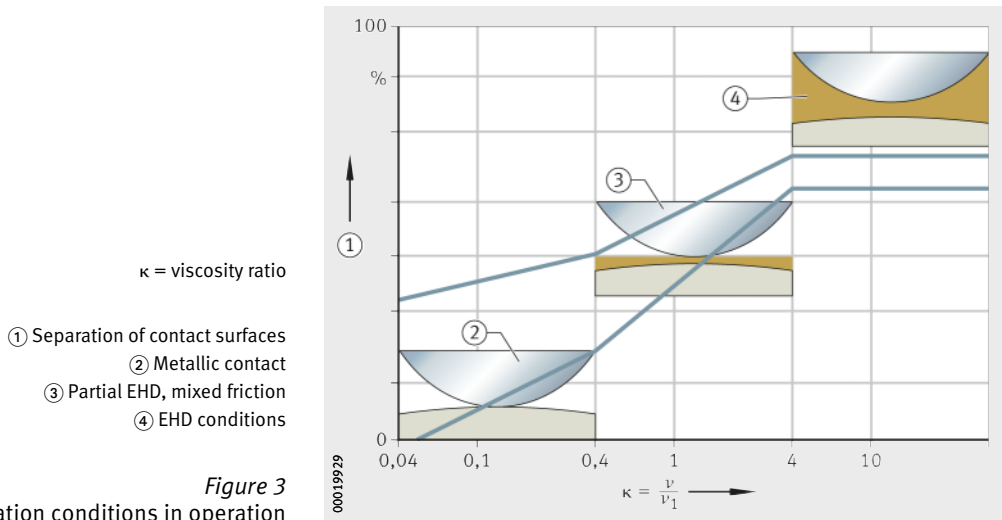
The friction and lubrication behaviour and the achievable life of the rolling bearing are dependent on the lubrication regime and the resulting friction regime. The possible lubrication regimes are delineated in the Stribeck curve, *Figure 2*.

All three regimes may occur in oil and grease lubrication. The lubrication regime in grease lubrication is determined primarily by the viscosity of the base oil. In addition, the thickener in the grease acts as a lubricant.



Lubrication

The rating life of rolling bearings is influenced by the lubricant film. The lubricant and its characteristics, matched to the operating conditions, as well as the macrogeometry and microgeometry of the contact surfaces determines the lubricant film thickness. The objective is to achieve separation of the contact surfaces, *Figure 3*.



The viscosity ratio κ is an indication of the quality of lubricant film formation:

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

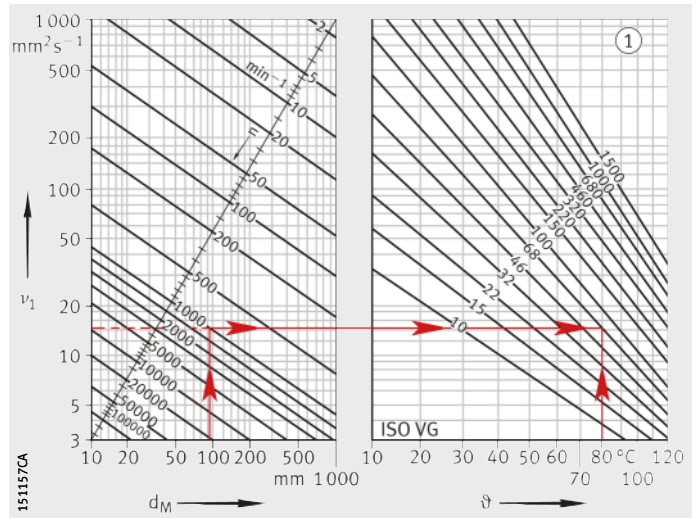
κ Viscosity ratio
 ν Kinematic viscosity of the lubricant at operating temperature
 ν_1 Reference viscosity of the lubricant at operating temperature.

The reference viscosity ν_1 is determined with the aid of the mean bearing diameter $d_M = (D + d)/2$ and the operating speed n , *Figure 4*, page 51.

n = operating speed
 v_1 = reference viscosity
 d_M = mean bearing diameter $(d + D)/2$
 ϑ = operating temperature

① Viscosity mm^2s^{-1} at $+40\text{ }^\circ\text{C}$

Figure 4
 Reference viscosity and
 V/T diagram for mineral oils



The supply of lubricant to bearings

The lubricant quantity 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 escape, the splashing or churning work will lead to temperatures at which the lubricant may be impaired or even destroyed. Overgreasing can lead to premature failure of the bearing.

In general, an adequate supply is ensured through the following:

- selection of the correct lubricant quantity and distribution in the bearing
- attention to the operating life of the lubricant
- appropriate addition of lubricant or lubricant replacement
- targeted design of the bearing position
- suitable sealing.

Lubrication

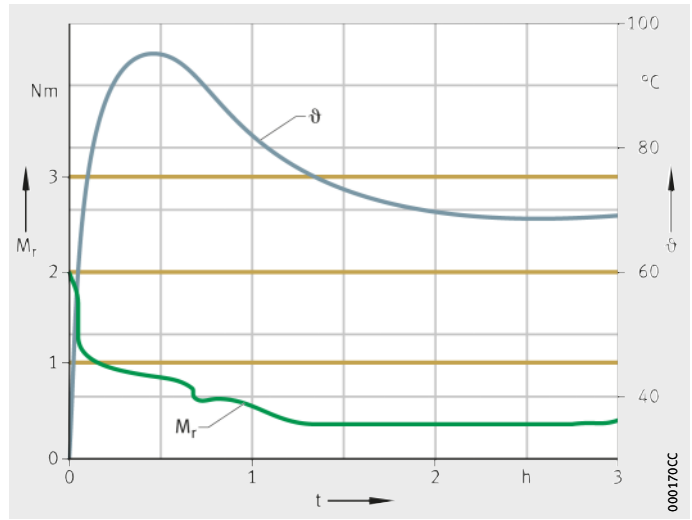
Initial greasing and new greasing

In the greasing of bearings, the following guidelines must be observed:

- Fill the bearings such that all functional surfaces definitely receive grease.
- Fill any housing cavity adjacent to the bearing with grease only to the point where there is still sufficient space for the grease displaced from the bearing. This is intended to avoid co-rotation of the grease. If a large, unfilled housing cavity is adjacent to the bearing, sealing shields or washers as well as baffle plates should be used to ensure that an appropriate grease quantity 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, the appropriate grease should be supplemented by a grease reservoir that has a surface as large as possible facing the bearing and that dispenses oil. The favourable quantity for the reservoir is two to three times the normal degree of filling. The reservoir must be provided either on one side of the bearing or preferably to an identical extent on both sides.
- In the case of higher speed parameters, the bearing temperature may settle at a higher value, in some cases over several hours, if the grease quantity during the starting phase has not been set correctly, *Figure 5*, page 53. The temperature will be higher and the increased temperature will remain for longer, the more the bearings and the cavities adjacent to the bearings are filled with grease and the more difficult it is for grease to escape freely. A remedy is a so-called interval running-in process with appropriately determined standstill periods for cooling. If suitable greases and grease quantities are used, equilibrium is achieved after a very short time.

M_r = frictional torque
 t = time
 ϑ = operating temperature

Figure 5
Frictional torque and temperature



Lubrication

Selection of the suitable lubricant

Selection of the correct lubricant is decisive for reliable function of the bearing. The optimum bearing operating life can be achieved if suitable lubricants are selected.

The following must be taken into consideration:

- bearing type
- speed
- temperature
- load.

Influence of bearing type

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

In bearings with point contact (ball bearings), each overrolling motion at the rolling contact exerts stress on only a relatively small volume of grease. 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 (roller bearings) place higher requirements on the grease. Not only is a larger grease quantity at the contact subjected to strain, but sliding friction and rib friction are also to be expected. This prevents the formation of a lubricant film and therefore leads to wear.

Influence of speed

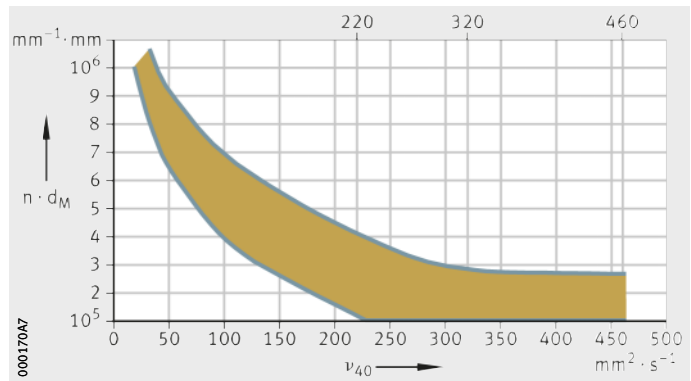
As in the case of rolling bearings, greases have a maximum speed parameter $n \cdot d_M$. The speed parameter of the bearing should always be a good match for the speed parameter of the grease, *Figure 6*.

In the case of a grease, this is dependent on the type and proportion of the thickener, the base oil type and the proportion of base oil. These data can be found in the technical data sheets for the greases.

Typically, greases for high speeds have a low base oil viscosity and are based on ester oil. 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. The speed parameter of a grease is not a material parameter but is dependent on the bearing type and the required minimum running time.

$n \cdot d_M$ = speed parameter
 ν_{40} = base oil viscosity at +40 °C

Figure 6
Speed parameter for greases



Lubrication

Influence of temperature

The temperature range of the grease must correspond to the range of possible operating temperatures in the rolling bearing. Grease manufacturers state this for rolling bearing greases K in accordance with DIN 51825. 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 stability at high temperature is dependent principally on the production quality and the production process. It is generally recommended that greases should be used in accordance with the bearing temperature normally occurring in the standard operating range, in order to achieve reliable lubrication and an acceptable grease operating life, *Figure 7*.

At low temperatures, greases release very little base oil. This can result in lubricant starvation. Schaeffler therefore recommends that greases are not used below the lower continuous limit temperature $T_{\text{lowerlimit}}$, *Figure 7*. This is approx. 20 K above the lower operating temperature of the grease as stated by the grease manufacturer.

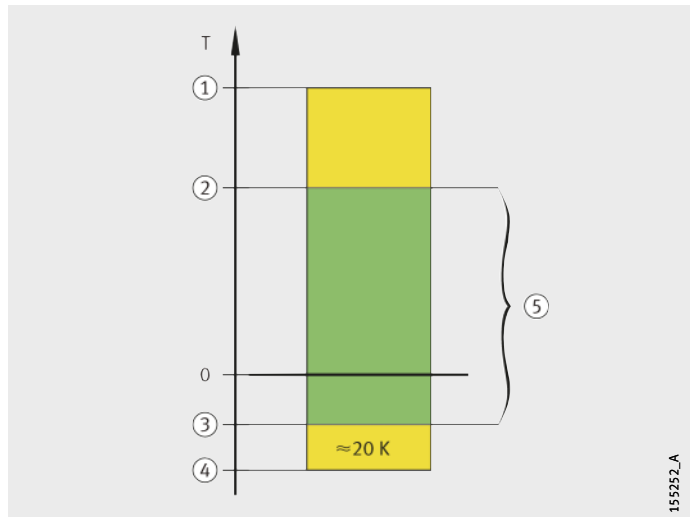
The upper continuous limit temperature $T_{\text{upperlimit}}$ must not be exceeded if a temperature-induced reduction in the grease operating life is to be avoided, see page 59.

T = operating temperature

- ① Upper operating temperature according to grease manufacturer
- ② $T_{\text{upperlimit}}$
- ③ $T_{\text{lowerlimit}}$
- ④ Lower operating temperature according to grease manufacturer
- ⑤ Standard operating range

Figure 7

Operating temperature range



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Operating temperature range

The operating temperature range of the grease must correspond to the range of possible operating temperatures in the rolling bearing.

Grease manufacturers state an operating temperature range for their rolling bearing greases K in accordance with DIN 51825.

The upper value is defined in accordance with DIN 51821 by means of testing on the FAG rolling bearing grease test rig FE 9. At the upper operating temperature, a 50% failure probability (F_{50}) of at least 100 hours must be achieved in this test.

The lower value is defined in accordance with DIN 51825 by means of flow pressure. The flow pressure of a grease is the pressure required to press a stream of the 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. This cannot be used to give an indication of its suitability 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 1000 Nmm and the running torque must not exceed 100 Nmm.

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 of a grease can be found in the corresponding datasheet.

Lubrication

Influence of load For a load ratio $C/P < 10$ or $P/C > 0,1$, greases are recommended that have higher base oil viscosity and in particular anti-wear additives (EP). 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 be used in preference for applications in the boundary or mixed friction range (chemical lubrication). The solid lubricant particle size must not exceed a value of $5 \mu\text{m}$. Silicone lubricants have a low load carrying capacity that cannot be compensated by an appropriate additive package and may therefore be used only under very low loads $P \leq 3\% C$.

Grease operating life

Basic grease operating life

The basic grease operating life t_f is dependent on the bearing-specific speed parameter $k_f \cdot n \cdot d_M$ and is calculated using *Figure 8*.

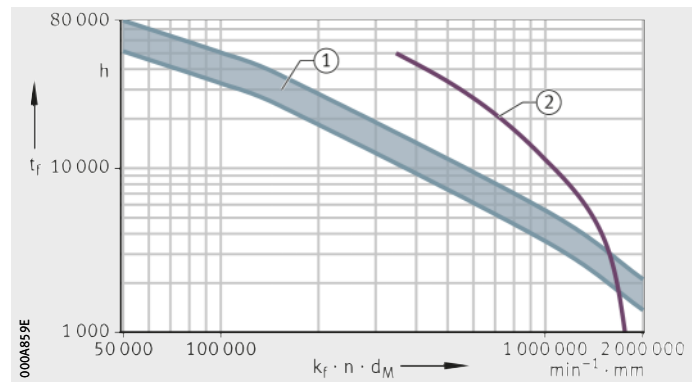
The basic grease operating life in accordance with *Figure 8* is valid in the following cases:

- greases with a proven performance capability for bearings
- bearing arrangements where the bearing temperature is lower than the upper continuous limit temperature of the grease $T_{upperlimit}$
- a load ratio of $C_0/P \geq 20$
- constant speed and load
- load in the main direction (radial in radial bearings, axial in axial bearings)
- radial bearings with a horizontal axis of rotation
- a rotating inner ring
- bearing arrangements without disruptive environmental influences.

t_f = basic grease operating life
 $k_f \cdot n \cdot d_M$ = bearing-specific speed parameter

- ① Steel/steel
- ② Steel/ceramic

Figure 8
 Calculation of basic grease operating life



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
Tapered roller bearings	4
Spherical roller bearings without central rib	8
Spherical roller bearings with central rib	8

Lubrication

Grease operating life

The grease operating life t_{fG} applies where this is below the calculated bearing life and the bearings are not relubricated. A guide value for the grease operating life t_{fG} can be determined in approximate terms using the following equation:

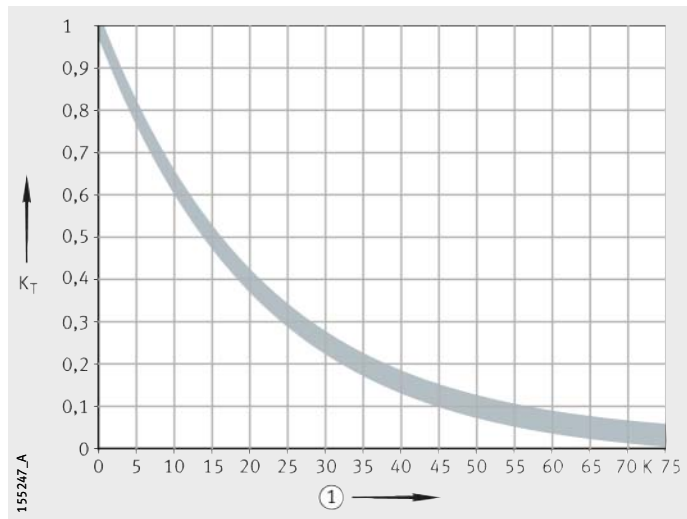
$$t_{fG} = t_f \cdot K_T \cdot K_P \cdot K_R \cdot K_U \cdot K_S$$

t_{fG}	h
Guide value for grease operating life	
t_f	h
Basic grease operating life	
K_T	-
Correction factor for increased temperature	
K_P	-
Correction factor for increased load	
K_R	-
Correction factor for oscillation	
K_U	-
Correction factor for environmental influences	
K_S	-
Correction factor for vertical shaft.	

If the operating temperature exceeds the value $T_{upperlimit}$, the grease operating life will decrease. In approximate terms, a reduction by half of the relubrication interval should be expected with each increase of 15 K in the temperature, *Figure 8*, page 59.

K_T = temperature factor
 ① ΔT above $T_{upperlimit}$

Figure 9
 Temperature factor



Correction factor
for increased temperature

An increase in temperature leads to an acceleration in the speed of reaction and thus of oxidation or ageing.

As a rule of thumb, the following applies: an increase in temperature of 15 K will reduce the grease operating life by half. In the case of high grade greases, however, this effect is only pronounced above the so-called upper continuous limit temperature $T_{upperlimit}$. If the bearing temperature is above $T_{upperlimit}$, the reduction in the grease operating life due to temperature must be determined, *Figure 10*.



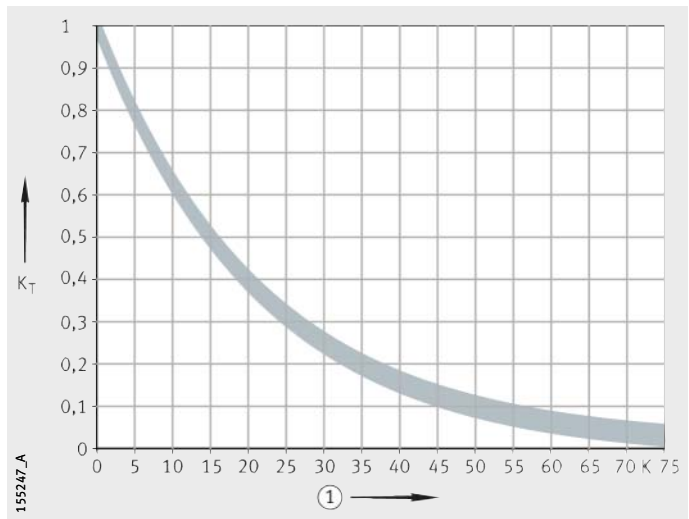
This diagram must not be used if the bearing temperature is higher than the upper operating temperature of the grease used. If necessary, another grease must be selected.

K_T = temperature factor
① K above $T_{upperlimit}$

Figure 10
Temperature factor

Correction factor
for increased load

Under higher bearing load, greases are subjected to greater strain. As a function of the load ratio C_0/P and the bearing type, this influence can be taken into consideration using the factor K_p , *Figure 11*, page 62.



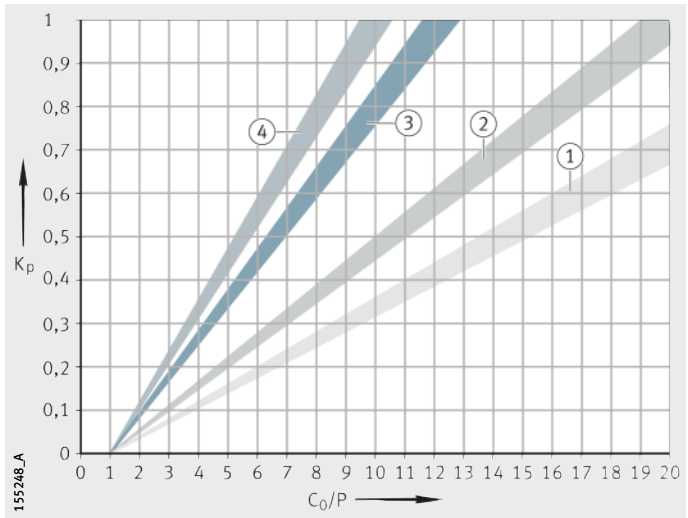
Lubrication

K_p = load factor
 C_0/P = ratio between basic static load rating and equivalent dynamic bearing load

①, ②, ③, ④ see table

Figure 11
 Load factor

Load factor K_p



Curve ¹⁾	Bearing type
①	Axial angular contact ball bearings, double row
	Axial deep groove ball bearings
	Axial needle roller bearings, axial cylindrical roller bearings
	Crossed roller bearings
②	Spherical roller bearings with central rib
	Needle roller and cage assemblies, needle roller bearings
	Drawn cup needle roller bearings, open ends and close end
	Cylindrical roller bearings, double row (not valid for NN30)
	Yoke type track rollers PWTR, stud type track rollers PWKR
	Yoke and stud type track rollers, with cage or full complement cylindrical roller set
③	Yoke and stud type track rollers with full complement needle roller set
	Cylindrical roller bearings LSL, ZSL
	Tapered roller bearings
	Spherical roller bearings without central rib (E1)
	Barrel roller bearings
	Cylindrical roller bearings, full complement
	Cylindrical roller bearings, single row (constant, alternating, without axial load)
Four point contact bearings	
④	Deep groove ball bearings (single row, double row)
	Angular contact ball bearings (single row, double row)
	Self-aligning ball bearings
	Ball bearing type track rollers (single row, double row)
	Radial insert ball bearings, housing units

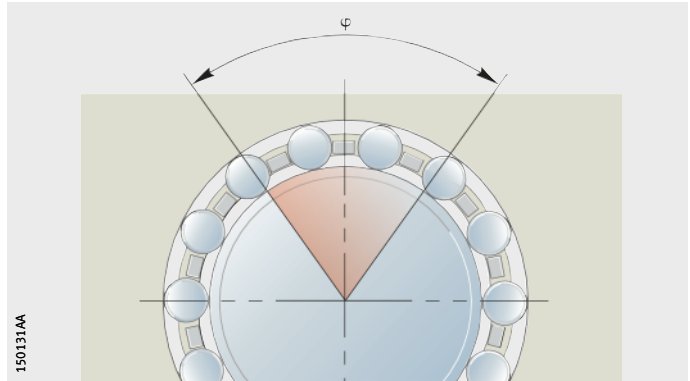
¹⁾ Curves, Figure 11.

Oscillation factor

Oscillating movements place a higher strain on the grease than continuously rotating bearings. The strain is placed continuously on the same grease volume, since no new grease can be drawn into the lubrication contact. As a result, the grease at the contact becomes depleted. In order to reduce fretting corrosion, the lubrication interval should be shortened. The reduction-inducing influence can be taken into consideration using the oscillation factor K_R , *Figure 13*. This is active starting from a swivel angle $\varphi < 180^\circ$, *Figure 12* and *Figure 13*.

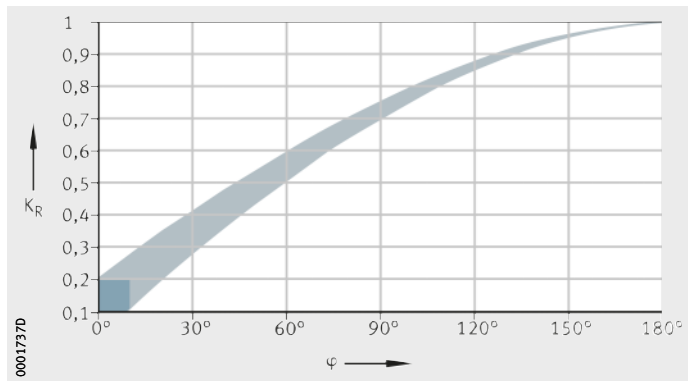
φ = swivel angle

Figure 12
Swivel angle



K_R = oscillation factor
 φ = angle of oscillation

Figure 13
Oscillation factor



Lubrication

Environment factor The factor K_U takes account of the influences of moisture, shaking forces, slight vibrations and shocks; see table.



It does not take account of extreme environmental influences such as water, aggressive media, contamination, nuclear radiation and 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.

Environment factor

Environmental influence	Environment factor K_U
Slight (for example, test rig)	1
Moderate (standard)	0,8
Heavy (for example, outdoor application)	0,5

Factor for vertical shaft

If increased escape of grease is expected, for example in the case of radial bearings with a vertical axis of rotation, this influence must be taken into consideration using the factor K_S , see table.

Factor

Vertical shaft	Factor K_S
Vertical shaft (depending on sealing)	0,5 to 0,7
Otherwise	1

Relubrication intervals

If rolling bearings are relubricated, attention must be paid to the lubrication interval in order to ensure reliable function of the bearings.



For reasons of operational security, relubrication intervals > 1 year are not recommended.

Experience shows that a guide value for most applications is:

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

t_{fR} h
Guide value for relubrication interval

t_{fG} h
Guide value for grease operating life.

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

If other greases are used, the miscibility and compatibility of the greases must be checked.

If feed lines filled with air are present, the filling volume of the feed lines should be included in calculation of the relubrication quantity.

Relubrication quantities

The initial greasing and relubrication quantity is normally calculated together with the calculation of the free or undisturbed internal space.

In order to give an initial estimate, the following equation can be helpful.

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

m g
Relubrication quantity
 D mm
Bearing outside diameter
 B mm
Bearing width
 X –
Factor, see table.

Relubrication intervals

Relubrication interval	Factor X
weekly	0,002
monthly	0,003
annually	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 grease quantity. A large relubrication quantity is necessary in principle if the used grease has already been damaged due to higher temperature.

Lubrication

Grease distribution

In the case of bearings rotating at very high speeds (speed parameter $> 500\,000 \text{ min}^{-1} \cdot \text{mm}$), a grease distribution cycle is necessary.

The running-in process comprises several cycles of start/stop operation at different speeds and for different running times, while the stationary times after each cycle are very important. The necessary number of cycles required may vary according to the bearing size, the number of bearings, the maximum speed and the bearing environment, *Figure 14*.

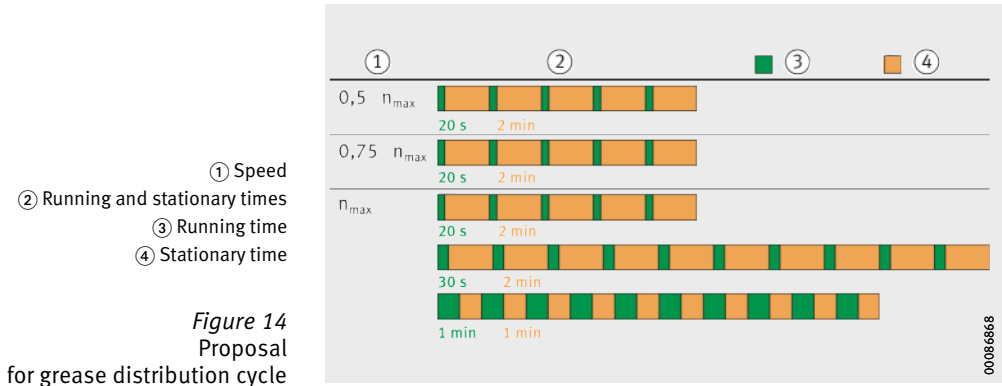


Figure 14
Proposal

for grease distribution cycle

Further cycles should be carried out for an increased running time and with a shortened stationary time until the equilibrium temperature is achieved.

Miscibility of lubricants

Mixtures of greases should be avoided if at all possible.

Preconditions

If they are unavoidable, the following preconditions must be fulfilled:

- the base oil must be the same
- the thickener types must match
- the base oil viscosities must be similar (they must not differ by more than one ISO VG grade).
- the consistency must be identical (NLGI grade).

Miscibility can be estimated in approximate terms using the following tables. However, this gives no guarantee of operational security:

Miscibility of lubricants (base oils)

Base oil	Mineral oil	PAO	Ester oil	Polyglycol oil	Silicone oil	Alkoxyfluoro oil
Mineral oil	●	●	●	–	○	–
PAO	●	●	●	–	○	–
Ester oil	●	●	●	○	–	–
Polyglycol oil	–	–	○	●	–	–
Silicone oil	○	○	–	–	●	–
Alkoxyfluoro oil	–	–	–	–	–	●

- Mixing generally non-critical
- Miscible in individual cases, but checking should be carried out
- Mixing not permissible

Lubrication

Compatibility of different thickener types

	Lithium soap	Lithium complex	Sodium complex	Calcium complex	Aluminium complex	Barium complex	Bentonite	Polycarbamide	PTFE
Lithium soap	●	●	–	●	–	●	–	–	●
Lithium complex	●	●	○	●	○	○	–	○	●
Sodium complex	–	○	●	○	○	○	–	○	●
Calcium complex	●	●	○	●	○	○	○	○	●
Aluminium complex	–	○	○	○	●	○	–	–	●
Barium complex	●	○	○	○	○	●	●	○	●
Bentonite	–	–	–	○	–	●	●	–	●
Polycarbamide	–	○	○	○	–	○	–	●	●
PTFE	●	●	●	●	●	●	●	●	●

- Mixing generally non-critical
- Miscible in individual cases, but checking should be carried out
- Mixing not permissible



Before mixing, the lubricant manufacturer must always be consulted. Even if the preconditions are fulfilled, the performance capability of the mixed grease may be impaired. Relubrication should only be carried out using greases of comparable performance capability. If a different grease grade is to be used, 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.



Current insulation

- Current-insulating bearings as a preventive measure
- Typical bearing damage in current passage
- Ceramic-coated bearings
- Hybrid bearings

Current insulation

Current-insulating bearings as a preventive measure

In general, it is difficult to eliminate the causes of electrical currents in the rolling bearing. Nevertheless, bearing damage can be prevented if it is possible to stop or significantly reduce the flow of current. Current-insulating rolling bearings of numerous designs are now available for this purpose. The components that should be insulated are dependent on the type of electrical current present.

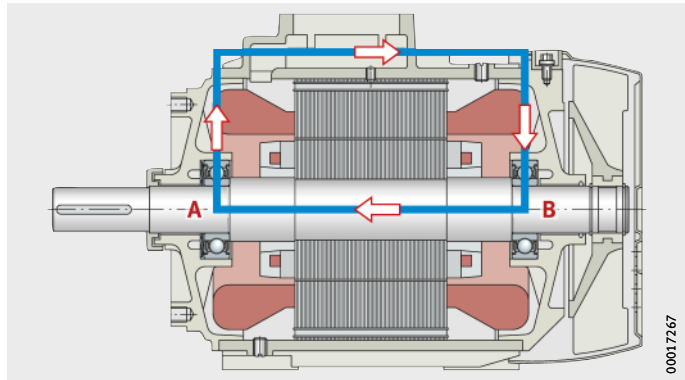
Induced voltage along a shaft

An induced voltage along a shaft leads to a circuit that is completed via bearing 1, the housing and bearing 2, *Figure 1*.

One cause of such shaft voltages is frequently the asymmetrical distribution of the magnetic flow in the motor, which is observed in particular in motors with a small number of pairs of poles. In this case, it is sufficient to interrupt the flow of current by insulating one of the two bearings. In general, the bearing on the non-drive side is insulated.

A = drive side
B = non-drive side

Figure 1
Current flow due to induced voltage along the shaft



Current flow between the shaft and housing

If a voltage occurs between the shaft and housing, the currents flow in the same direction through each of the two bearings. The main cause to be considered is the common mode voltage in converters. It is recommended here that both bearings should be insulated, *Figure 2*.

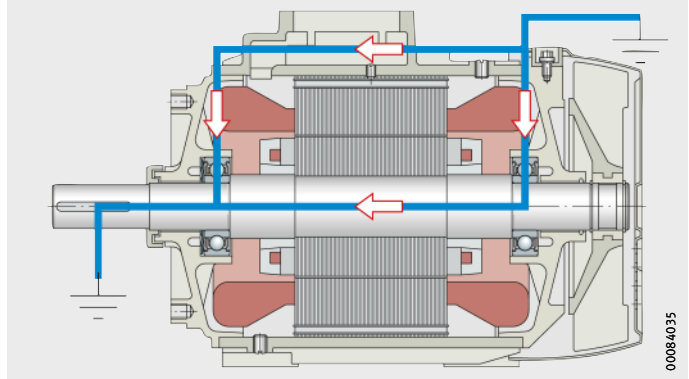


Figure 2
Current flow between the shaft and housing

Current insulation

Typical bearing damage in current passage

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

Marks on raceways and rolling elements

In many cases, uniformly matt grey marks are seen on the raceways and rolling element surfaces. This visual characteristic is non-specific and can also be caused by other influences, for example oil containing abrasives, *Figure 3*.



Figure 3
Marks on raceways

Fluting

The so-called fluting consists of periodic patterns running in the direction of rolling on areas of the surface of different depth. This surface pattern can in most cases be attributed to current passage, *Figure 4*.

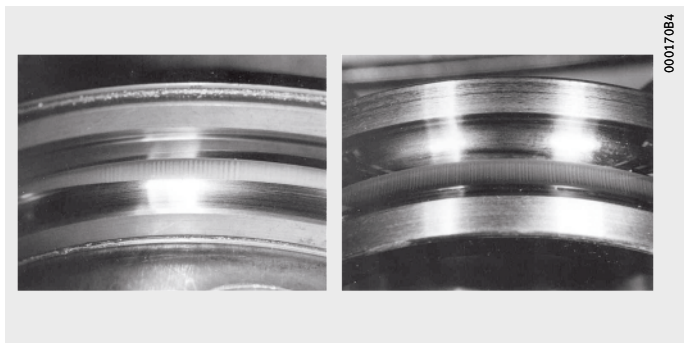


Figure 4
Fluting

Damage structures

It is only with the aid of scanning electron microscopy (SEM) that it becomes apparent that both damage structures are characterised by melt craters and welding beads of μm size that cover the overrolled surfaces in a tightly packed form, *Figure 3* and *Figure 4*, page 72. This demonstrates current passage, *Figure 5*.

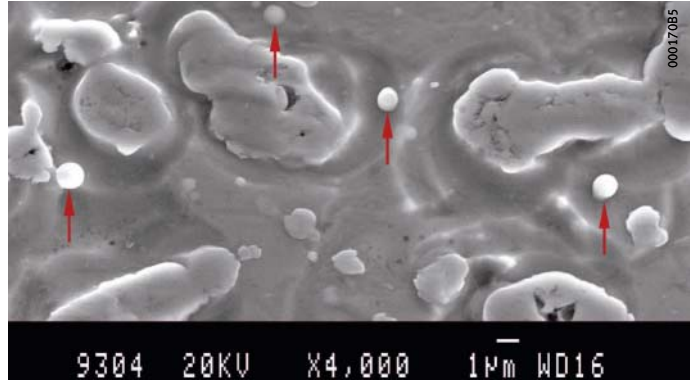


Figure 5
Damage structures
under the scanning electron
microscope (SEM)

Current insulation

Ceramic-coated bearings

Ceramic-coated bearings (Insutect A) are standard bearings in which the inner ring or outer ring has a current-insulating ceramic coating, *Figure 6* and *Figure 7*.

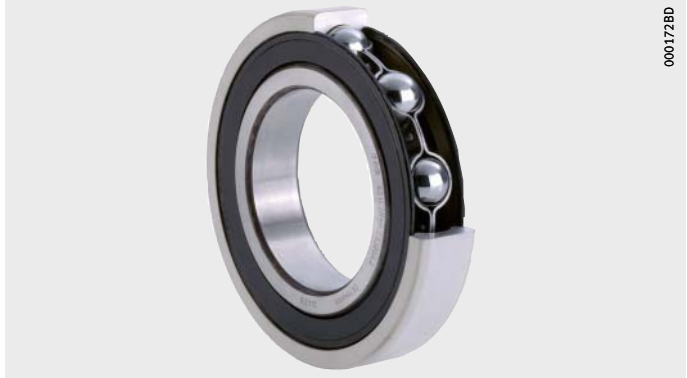


Figure 6
Ceramic-coated
deep groove ball bearing

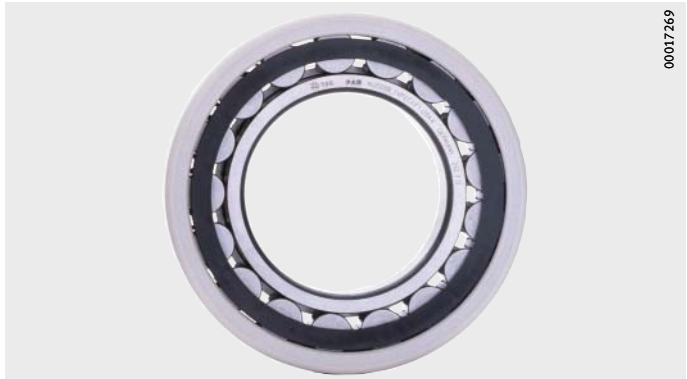


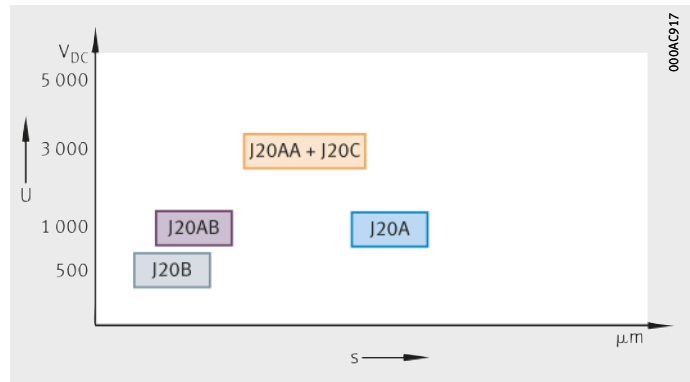
Figure 7
Ceramic-coated
cylindrical roller bearing

FAG bearings coated with Insutect A give high insulation protection. The bearings coated with oxide ceramic are indicated by the suffix J20 with an additional letter A, AA, AB, C or B, *Figure 8*. These layers are applied to the bearing surfaces using the plasma spray method. 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 thus interchangeable with standard bearings.

U = puncture voltage
s = coating thickness

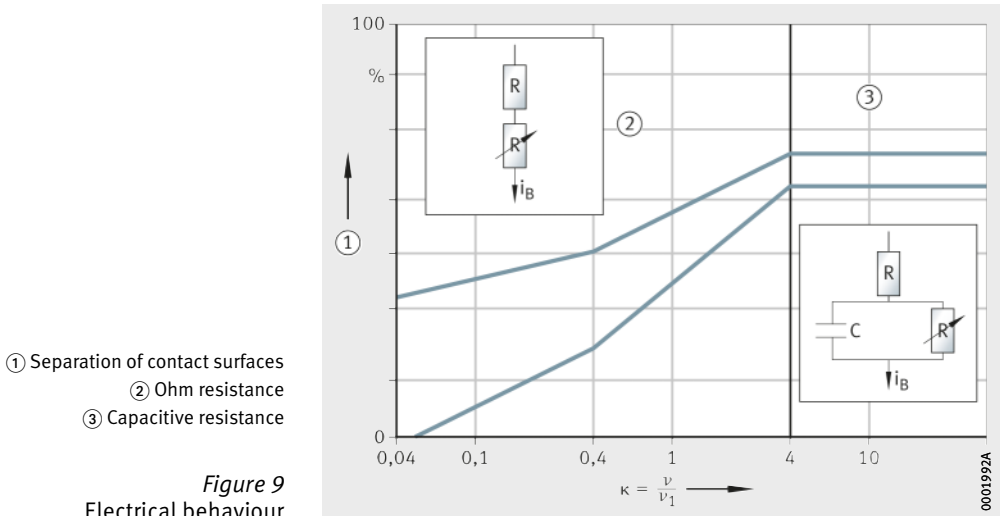
Figure 8
Overview of coatings



Current insulation

Electrical behaviour of an uncoated bearing

The electrical bearing behaviour under different lubrication regimes is shown as examples in *Figure 9*. If the elastohydrodynamic state is achieved in operation, i.e. the rolling partners are completely separated by the lubricant, the bearing acts in a capacitive manner. If the state of complete separation is left behind and enters a regime of mixed and boundary friction, the bearing changes its electrical behaviour and this can now be regarded as Ohm resistance.



Puncture strength

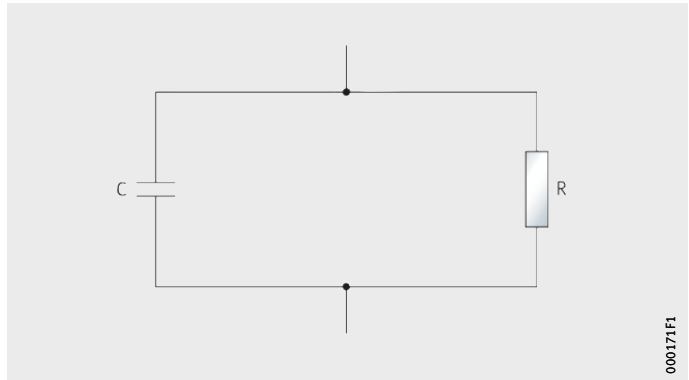
The coatings are subjected to 100% quality inspection and ensure a puncture strength as follows:

- as insulation up to DC 3 000 V (J20AA, J20C)
- as insulation up to DC 1 000 V (J20A, J20AB)
- as insulation up to DC 500 V (J20B).

In principle, a current-insulating bearing can be integrated as a parallel circuit between resistance and capacitance, *Figure 10*.

C = capacitance
R = resistance

Figure 10
Parallel circuit
Resistance and capacitance



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

The decisive factor in selecting current insulation is the time behaviour of the voltages present. This depends in the case of direct current voltage or slowly varying alternating current voltage on the Ohm resistance and in the case of higher frequency alternating current voltage (which is often found in converters) on the capacitive resistance of the bearing.

Current insulation

A distinction must be drawn between two mechanisms:

- Direct current voltage resistance:
 - At room temperature, it is at least 50 MΩ which, according to Ohm's law $I = U/R$ at voltages up to 1000 V means only currents significantly below 20 μA, which are non-critical for bearings.
- Alternating current voltage resistance:
 - The insulated unit represents a capacitance C, which can accumulate charges. Under the influence of an alternating current voltage, this leads to an alternating current due to direct contact between the rolling element and raceway. For the effective values of current and voltage under a harmonic time curve with a pulsance ω , the equation $I = U \cdot \omega \cdot C$ applies.
 - On a similar basis to Ohm's Law, $Z = 1/(\omega \cdot C)$ is designated as the capacitive resistance of the bearing. The capacitance of a bearing with oxide ceramic is typically 2 nF to 20 nF, depending on the bearing size. Its capacitive resistance at a frequency of 50 Hz therefore lies in the range 0,15 MΩ to 1,5 MΩ, in other words significantly lower than its direct current voltage resistance. At higher frequencies, this value is reduced even further. Nevertheless, it will in most cases be significantly higher than the resistance of the non-insulated bearing, which is only very low at voltages at or above approx. 1 V (1 Ω or less).

At the same voltage and same frequency, the flow of current is determined by the capacitance of the system. A large coating thickness s and a small coated contact area A relative to the bearing leads to a low capacitance and thus to a lower current density, see equation.

$$C = \epsilon_0 \cdot \epsilon_r \cdot \left(\frac{A}{s} \right)$$

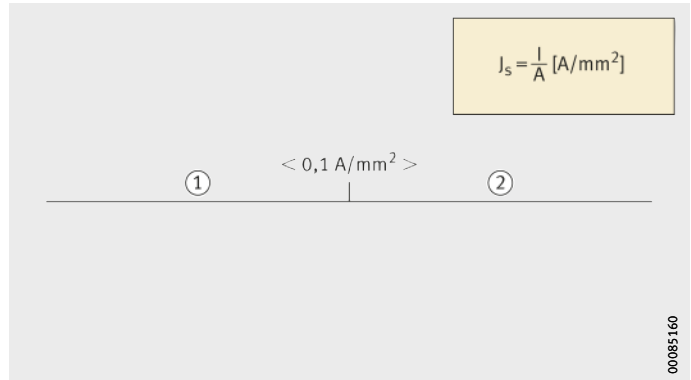
C	F
Capacitance of the bearing arrangement	
ϵ_0	Fm ⁻¹
Dielectric constant	
ϵ_r	–
Relative dielectric factor, dependent on material	
A	m ²
Coated surface	
s	m
Coating thickness.	

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, respectively, the inner or outer ring of the bearing. This is dependent on the bearing type and the operating conditions. At current densities with effective amperages below approx. $0,1 \text{ A/mm}^2$, there is no danger of damage due to current passage according to our present level of knowledge. At values above $0,1 \text{ A/mm}^2$, current damage must be expected, *Figure 11*.

J_s = effective current density
 I = amperage
 A = effective contact surface

- ① No damage
- ② Damage

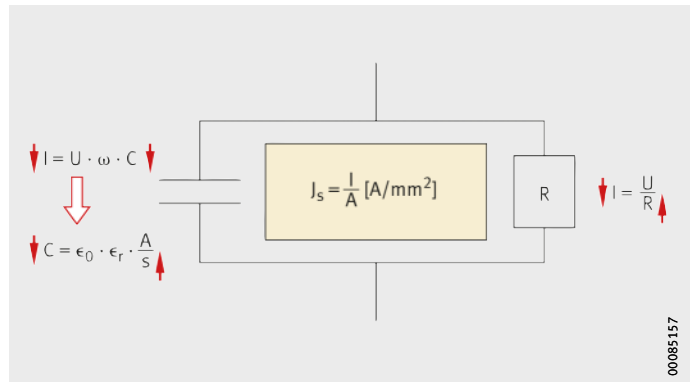
Figure 11
 Current density



For good insulation, the Ohm resistance should be as high as possible and the capacitance as low as possible. This is shown as an example in *Figure 12*.

J_s = effective current density
 I = current
 A = effective contact surface
 U = voltage
 ω = pulsatance
 C = capacitance
 ϵ_0 = dielectric constant
 ϵ_r = relative dielectric factor
 s = layer thickness
 R = Ohm resistance

Figure 12
 Insulation effect



Current insulation

The electrical capacitance as a function of the coated surface and the coating used is shown in the diagram, *Figure 13*.

C = capacitance
d = bore diameter

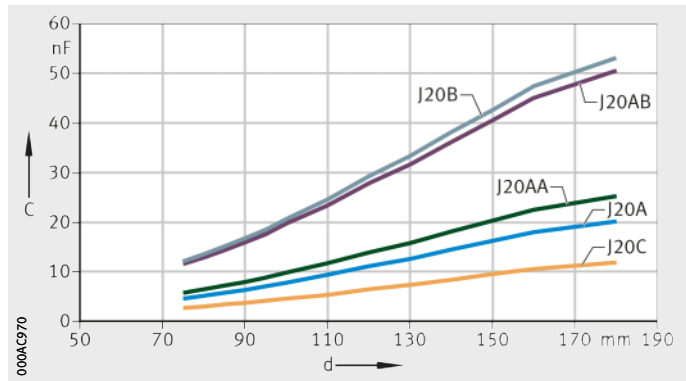


Figure 13

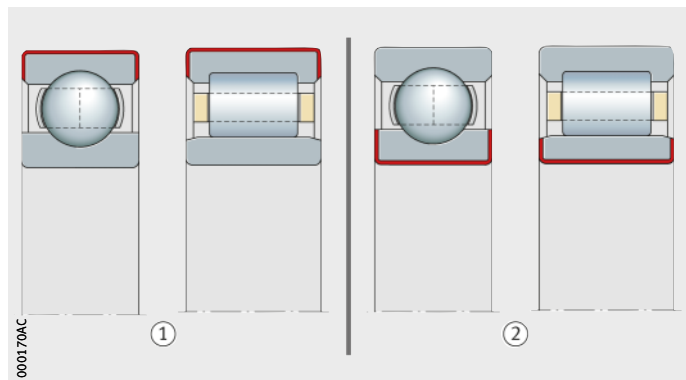
Relationship between electrical capacitance and coating used

Types of coatings and value ranges

The various type of coatings for bearings are shown in cross-section, *Figure 14*. The parameters and value ranges of the types of coatings are shown in a comparison, see table, page 81.

- ① External coating J20A, J20B, J20AB, J20AA
- ② Internal coating J20C

Figure 14
Types of coatings



**Parameters
Types of coatings**

Parameter	Unit	J20AB	J20AA	J20C
Bearing coating	–	Outer ring	Outer ring	Inner ring
Layer thickness	μm	100	200	200
Operating environment	–	Dry, damp	Dry, damp	Dry, damp
Dielectric strength	DC V	≤ 1 000	≤ 3 000	≤ 3 000
Dry	MΩ	50	50	50
Damp	MΩ	10	10	10
Possible dimensions				
Inside diameter	mm	–	–	70
Outside diameter	mm	70 – 800	70 – 800	800

**Parameters
Types of coatings
(continued)**

Parameter	Unit	J20B	J20A
Bearing coating	–	Outer ring	Outer ring
Layer thickness	μm	≤ 120	> 200
Operating environment	–	Dry	Dry
Dielectric strength	DC V	≤ 500	≤ 1 000
Dry	MΩ	50	50
Damp	MΩ	–	–
Possible dimensions			
Inside diameter	mm	–	–
Outside diameter	mm	70 – 1 400 ¹⁾	70 – 1 400 ¹⁾

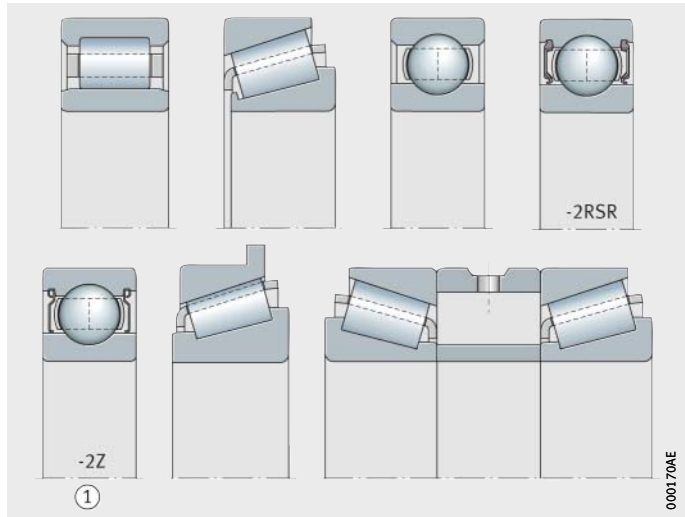
¹⁾ Preferred for use at 800 mm to 1 400 mm outside diameter.

The bearing surfaces of the rings to be coated must be cylindrical. If grooves or lubrication holes are present, please contact the relevant Application Engineering function of Schaeffler.

Current insulation

Bearing designs with ceramic coating

The available bearing designs with ceramic coatings are shown in cross-section, *Figure 15*.



Other bearing designs can be supplied coated by agreement.

Hybrid bearings

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 extremely resistant to wear and perform the function of current insulation.

FAG hybrid bearings offer not only a high resistance value but very low capacitance and thus the best protection against high frequency currents in converter operation.

Hybrid bearings are available as ball bearings and as cylindrical roller bearings, *Figure 16* and *Figure 17*.

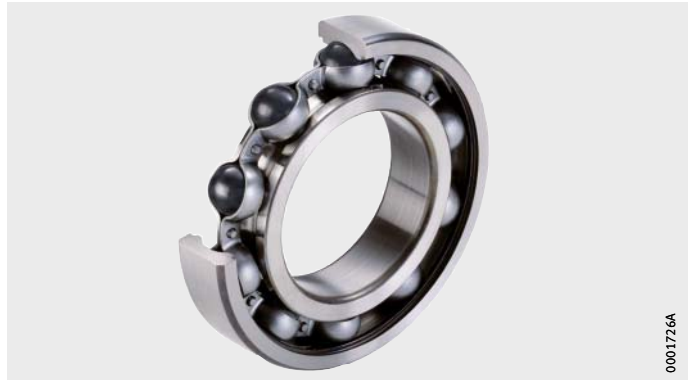


Figure 16
Hybrid ball bearing



Figure 17
Hybrid cylindrical roller bearing

Current insulation

Advantages of hybrid bearings

Hybrid bearings have advantages in comparison with ceramic-coated bearings:

- Hybrid bearings offer very high resistance to current passage. Their direct current voltage resistance, even at high temperatures, is in the $G\Omega$ range. A typical value for capacitance is 40 pF and is thus lower by a factor of 100 than bearings with ceramic coating.
- Hybrid bearings allow higher speeds at lower friction and thus lower temperatures in operation. The low mass of the rolling elements leads to lower friction. Lower friction reduces the CO_2 emissions in application. The comparison of CO_2 emissions from a standard and a hybrid cylindrical roller bearing in the drive system of a high speed train over an operating period of one year at approx. 600 000 km/year is shown in *Figure 18*.
- Hybrid bearings have better emergency running characteristics than standard bearings.
- No coating present that could be damaged during incorrect handling.

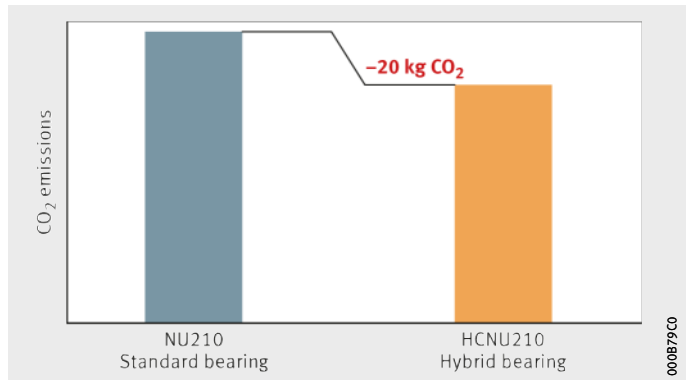


Figure 18

Comparison of CO_2 emissions with NU210 and HCNU210

Other characteristics

In comparison with standard bearings, hybrid bearings have:

- 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 ball bearings have the same dimensions and can be retrofitted.

In addition, hybrid bearings offer a longer grease operating life than standard bearings.

If the life cycle cost of a bearing is analysed, the hybrid bearing is superior to the standard bearing. The product-specific life cycle costs can be optimised through the use of hybrid bearings by up to 20%.

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



Service products and special solutions

- FAG housing unit with relubrication facility
for traction motors
- Automatic lubricators for traction motors
- Rolling bearing greases Arcanol

Service products and special solutions

FAG housing unit with relubrication facility for traction motors

For rotor bearing arrangements in electric drives, Schaeffler works in consultation with customers to manufacture complete housing units with relubrication facility that are individually designed in accordance with the application and can also be equipped with measurement technology, *Figure 1*.



Figure 1
FAG housing unit FKB
with relubrication facility
for traction motors

FAG housing units FKB offer numerous advantages in the design, production, 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 FAG housing units for traction motors.

Depending on the size, the housing units are manufactured as follows:

- spheroidal graphite cast iron EN-GJS-400-15 or
- steel with a minimum tensile strength R_m of 400 N/mm².

Due to the customer-specific concept, the housing units can be matched to the specific geometry of the adjacent construction 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 current-insulating bearings in order to prevent damage caused by current passage, *Figure 2 to Figure 4.*



Figure 2
Cylindrical roller bearing

00019991



Figure 3
Current-insulating bearing

00017269

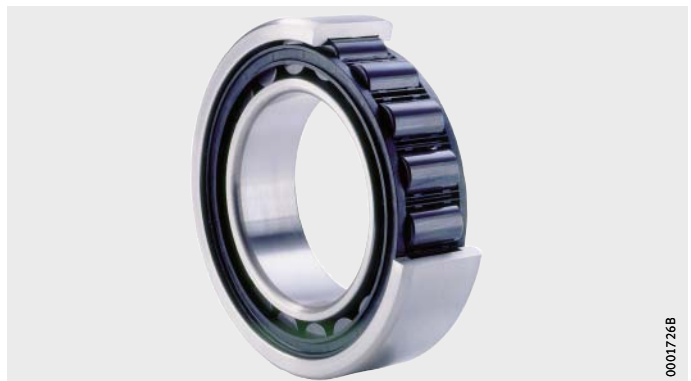


Figure 4
Hybrid bearing
(hybrid cylindrical roller bearing)

00017268

Service products and special solutions

For special requirements, special solutions are also available that allow continued use of existing products, see page 95. The housing units developed for grease lubrication have an appropriate relubrication facility, thus reducing the costs and work associated with maintenance.

Special designs with a container for collecting used grease are also possible and can be adapted to the appropriate customer requirements.

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

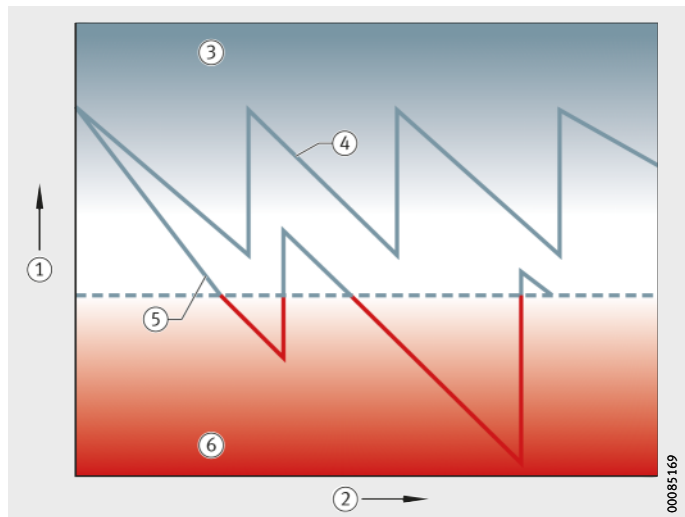
Automatic lubricators for traction motors

Almost half of all bearing failures can be traced back to inadequate or incorrect lubrication, *Figure 5*. Bearing failures can be very costly. Bearing failures in the drive train can lead to unplanned, costly breakdowns.

In many cases, this incurs costs for lost production and repairs that not infrequently run to tens of thousands of Euros.

- ① Lubricant quantity
- ② Operating life
- ③ Adequate lubricant quantity
- ④ Theory
- ⑤ Practice
- ⑥ Lubricant starvation

Figure 5
Manual relubrication



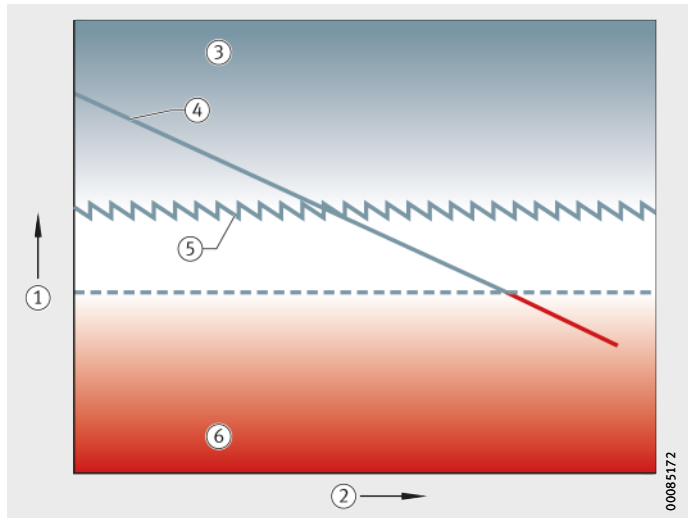
Such damage can be prevented by using automatic lubricators, *Figure 6*.

Lubricators have the following advantages:

- individual, precise supply to each bearing position using the most suitable lubricant
- fully automatic, maintenance-free operation due to continuous relubrication
- extended service life and maintenance intervals
- increased availability of plant
- considerable cost savings.

- ① Lubricant quantity
- ② Operating life
- ③ Adequate lubricant quantity
- ④ "For life" lubrication
- ⑤ Continuous lubrication
- ⑥ Lubricant starvation

Figure 6
Automatic relubrication



Service products and special solutions

The automatic single-point and multi-point lubrication systems of the CONCEPT family supply up to eight lubrication points with lubricant constantly, precisely and irrespective of temperature, *Figure 7*.

- Maintenance-free and cost-effective
- Versatile in application
- Individually matched to the bearing position
- Precise quantity dispensing even over long periods
- Simple, user-friendly handling
- High flexibility
- No manual relubrication necessary.



Figure 7
Optimum lubrication concept

Rolling bearing greases Arcanol

Special rolling bearing greases such as Arcanol offer the best conditions for achieving reliable, durable and cost-effective bearing arrangements. As a result, premature failure of bearings due to incorrect grease selection is increasingly a thing of the past.

In co-operation with renowned lubricant manufacturers, Schaeffler has for many years developed 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, in which its characteristics are examined.

On the lubricant test rigs FE8 (testing in accordance with DIN 51819) and FE9 (testing in accordance with DIN 51821), the greases are tested in rolling bearings for service life, friction and wear. Only the 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 to transfer the grease to containers under the name of Arcanol.

The range is graduated such that almost all areas of application can be optimally covered using these greases. For an overview of the chemical-physical data, areas of application and suitability of these greases, see table, page 92.

The advantages of Arcanol greases include:

- 100% tested, guaranteeing constant quality of lubricants for longer rolling bearing life
- developed and tested in the field by application and tribology experts
- close co-operation at all times with well known lubricant manufacturers
- optimally designed for rolling bearing applications
- reduced costs through
 - longer maintenance intervals
 - lower friction
 - less wear and bearing damage
 - considerably longer bearing operating life
 - increased operational security.

Service products and special solutions

Arcanol rolling bearing greases

Arcanol grease	Classification
MULTI2	Ball bearing grease, low-noise, for $D \leq 62$ mm
MULTI3	Ball bearing grease, radial insert ball bearing grease, standard for $D > 62$ mm
MULTITOP	Universal high performance grease
TEMP90	Rolling bearing grease, low-noise, up to $+160$ °C
TEMP110	Universal grease for higher temperatures
TEMP120	Grease for high temperatures and high loads
TEMP200	Rolling bearing grease for $T > +150$ °C to $+260$ °C
LOAD150	Multi-purpose grease for automotive applications, high performance grease for line contact
LOAD220	Heavy duty grease, wide speed range
LOAD400	Grease for high loads, shocks
LOAD460	Grease for high loads, vibrations, low temperatures
LOAD1000	Grease for high loads, shocks, large bearings
SPEED2,6	Standard spindle bearing grease
FOOD2	Grease with foodstuffs approval
VIB3	Grease for oscillating motion
CLEAN-M	Clean room grease, grease resistant to radiation
MOTION2	High performance grease paste for oscillating applications and plain bearing arrangements

Thickener Base oil	Operating temperature range °C	Upper continuous limit temperature $T_{upperlimit}$ °C	NLGI grade	Speed parameter $n \cdot d_M$ $\text{min}^{-1} \cdot \text{mm}$	Kinematic viscosity	
					at 40 °C mm^2/s	at 100 °C mm^2/s
Lithium soap Mineral oil	-30 to +120	+75	2	500 000	110	11
Lithium soap Mineral oil	-30 to +120	+75	3	500 000	80	10
Lithium soap Partially synthetic oil	-50 to +140	+80	2	800 000	82	12,5
Polycarbamide Mineral oil	-40 to +160	+90	3	700 000	148	15,5
Lithium complex soap Partially synthetic oil	-35 to +160	+110	2	500 000	130	14
Polycarbamide Synthetic oil	-30 to +180	+120	2	300 000	400	40
PTFE Alkoxyfluoro oil	-30 to +260	+200	2	300 000	550	49
Lithium complex soap Mineral oil	-20 to +140	+95	2	500 000	160	15,5
Lithium/calcium soap Mineral oil	-20 to +140	+80	2	500 000	245	20
Lithium/calcium soap Mineral oil	-40 to +130	+80	2	400 000	400	27
Lithium/calcium soap Mineral oil	-40 to +130	+80	1	400 000	400	25,8
Lithium/calcium soap Mineral oil	-20 to +130	+80	2	300 000	1 000	38
Lithium soap Synthetic oil	-40 to +120	+80	2 to 3	2 000 000	25	6
Aluminium complex soap Synthetic oil	-30 to +120	+70	2	400 000	150	18
Lithium complex soap Mineral oil	-30 to +150	+90	3	350 000	170	14
Polycarbamide Ether oil	-30 to +180	+90	2	850 000	103	12,8
Lithium soap Synthetic oil	-40 to +130	+75	2	500 000	50	8

Checklist

Scope of performance

FAG housing unit _____

Rolling bearing _____

Tab washer _____

Locknut _____

Dimensions

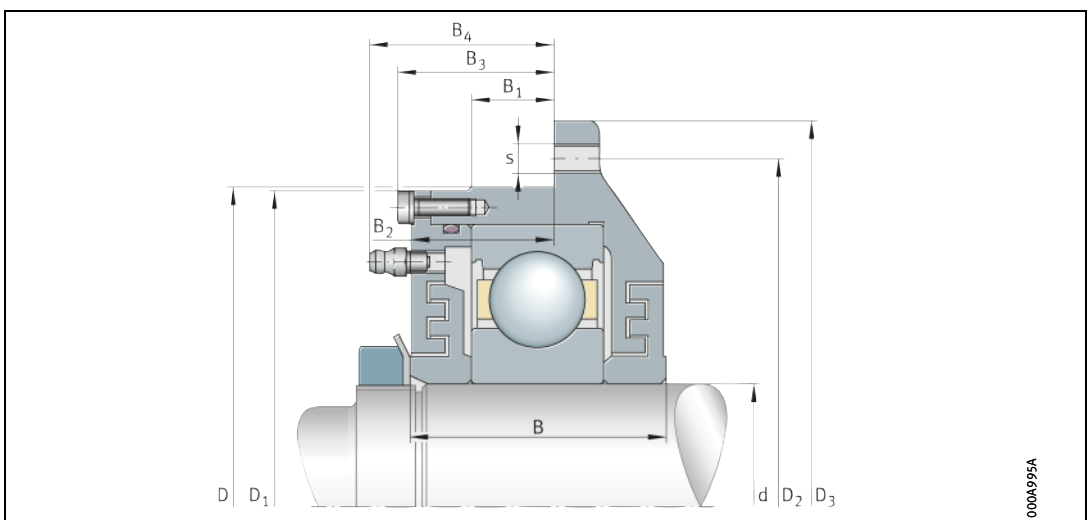
B1 _____ mm $\varnothing d$ _____ mm

B2 _____ mm $\varnothing D$ _____ mm

B3 _____ mm $\varnothing D1$ _____ mm

B4 _____ mm $\varnothing D2$ _____ mm

$\varnothing D3$ _____ mm



**Schaeffler Technologies
AG & Co. KG**

Georg-Schäfer-Straße 30
97421 Schweinfurt
Germany
Internet www.schaeffler.de/en
E-mail info.de@schaeffler.com

In Germany:

Phone 0180 5003872
Fax 0180 5003873

From other countries:

Phone +49 9721 91-0
Fax +49 9721 91-3435

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