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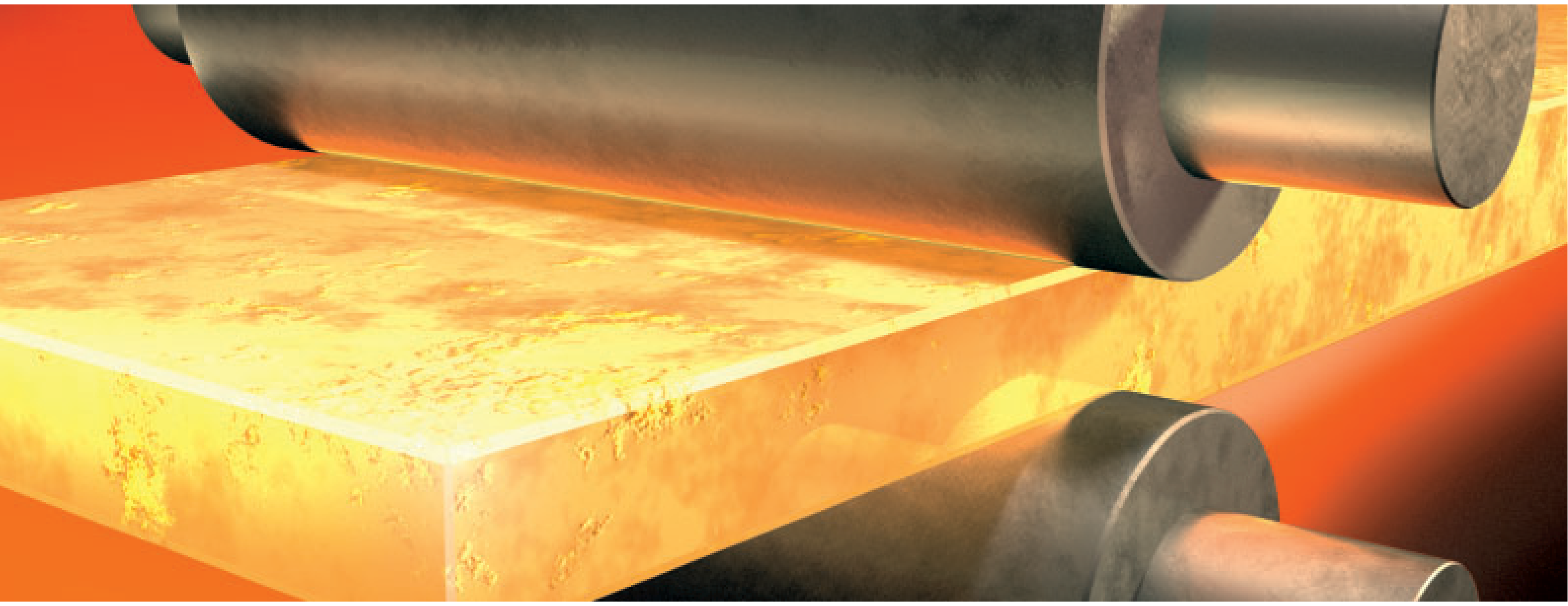


FAG Rolling Bearings in Rolling Mills

SCHAEFFLER

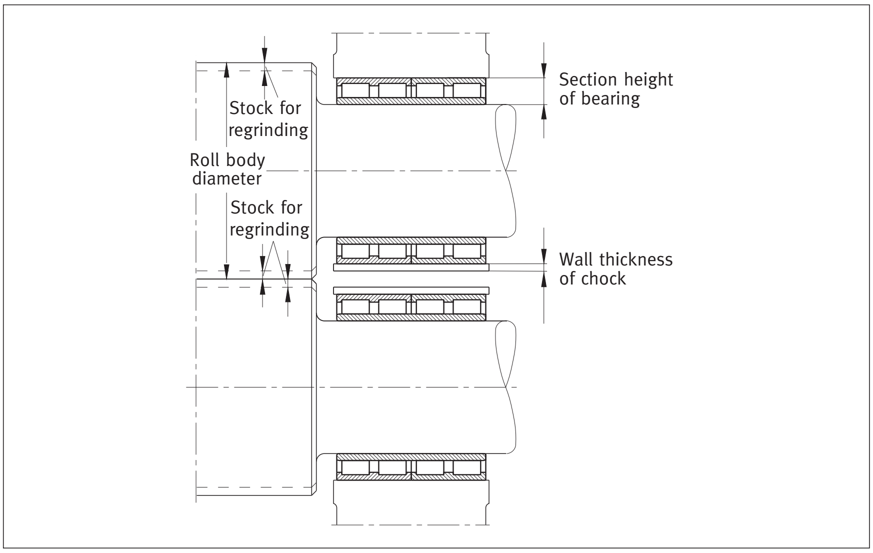
Foreword

Schaeffler has for many years worked on the design and production of bearings for rolling mills and gathered extensive experience in this field. This is presented in the present publication. A designer of rolling mills will find here the principles for selection and calculation of roll neck bearings. Their mounting and maintenance is also covered in detail. For any questions not covered under these principles, the Schaeffler engineering service can provide assistance. The dimensions and performance data of rolling bearings for rolling mills are given in Catalogue GL1. A selection of publications covering rolling mill bearing arrangements and fundamental subjects in bearing arrangement engineering, such as dimensioning, mounting and dismounting, lubrication and maintenance is given in the list on page 52 of this publication.

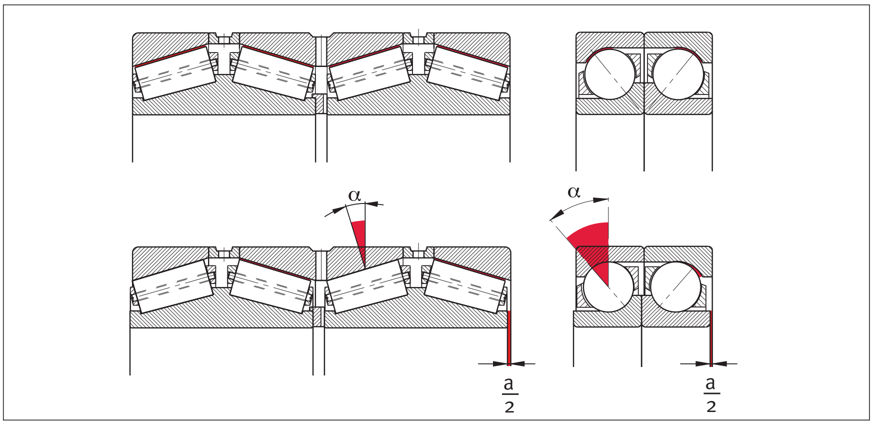
Contents

Design conditions

The rolling bearings in which the rolls in roll stands are supported, are subjected to high loads; the specific load is also high. In order that the bearings can reliably support the rolling forces, they must have high load carrying capacity.
On the other hand, the mounting space available for these bearings is restricted, especially in relation to the section height of the bearings, Figure 1. The diameter of the roll body, less a certain amount for regrinding and the wall thickness of the chock, determines the outside diameter of the bearing. Its bore corresponds to the diameter of the roll neck. If the load is very high, a compromise must be found between the diameter of the neck and its strength on the one hand and the section height of the bearing and its load carrying capacity on the other hand.
Since the roll neck bearings are very large in a radial direction, but are subjected to only slight load in an axial direction, the available mounting space is used as far as possible for supporting the radial forces.
Roller bearings have a higher load carrying capacity than ball bearings. As a result, the radial forces are supported by roller bearings only, namely cylindrical roller bearings, tapered roller bearings or spherical roller bearings. The material used for the bearing rings and rolling elements is through hardening rolling bearing steel or, in some cases, case hardening steel.
The frequent replacement of rolls influences the selection of bearing type. In general, the reworking of the roll bodies is accompanied by removal of the chocks. This is a demanding exercise in the case of non-separable bearings, such as spherical roller bearings, whose inner ring is fitted tightly on the roll neck. In the case of cylindrical roller bearings, the chock can be removed together with the outer ring and roller cage from the inner ring mounted on the neck.
Four-row tapered roller bearings or two spherical roller bearings arranged together on a cylindrical neck have a loose fit. As a result, the chocks can be easily removed; their field of application is restricted, however, by the loose fit.
If cylindrical roller bearings are fitted as radial bearings, the rolls must be axially supported in an additional axial bearing.
The separate support of the radial and axial forces is particularly advantageous where the axial guidance accuracy has an influence on the dimensional stability of the



1: Available mounting space

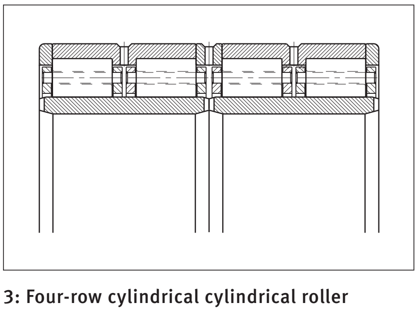


2: Axial internal clearance a as a function of radial internal clearance and contact angle α

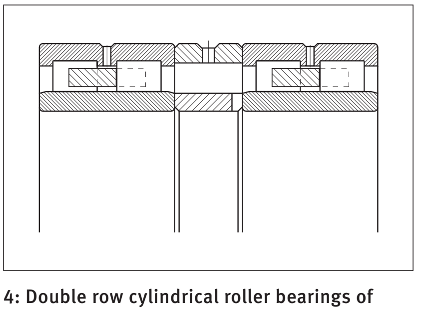
rolled stock, for example in stands for rolling of shape sections.
Axial bearings give a very high guidance accuracy, since they can be fitted with very small axial internal clearance or even without clearance. In contrast, radial bearings used to provide both axial and radial guidance always have a larger axial internal clearance.
Figure 2 (page 4) shows how the axial internal clearance a for a given radial internal clearance is dependent on the contact angle α. The ratio between the axial internal clearance and the radial internal clearance is at its largest in the case of spherical roller bearings. The values are smaller in four-row tapered roller bearings. The ratio is even smaller in the case of angular contact ball bearings.

Cylindrical roller bearings

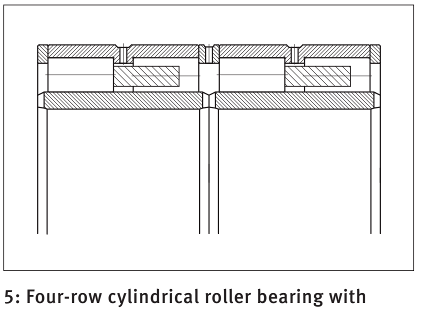
For a given mounting space, the highest load carrying capacity can be achieved with a cylindrical roller bearing. The cylindrical roller bearing is thus suitable for very high radial loads and, due to its low friction value, for very high speeds.
Cylindrical roller bearings of various designs are fitted in roll stands. The design considered in the individual case will depend on the type of application. Cylindrical roller bearings are mainly used with a tight inner ring fit. In some applications, cylindrical roller bearings with a loose inner ring fit have also proved effective.
In order to accommodate the maximum possible number of rollers in the bearing and achieve a high load carrying capacity, the bearings are designed with through-drilled rollers and pin-type cages, Figure 3. The pin-type cage comprises cage rings which laterally retain the pins passing through the centre of the rollers. This cage has very high strength. This is particularly important in the case of bearings in large stands that are subjected to strong acceleration and deceleration, for example in reversing type operation.
In order to achieve particularly high running accuracy, cylindrical roller bearings with a preground inner ring raceway are selected and finish ground together with the roll surface when the inner ring is mounted with a tight fit on the roll neck.
Figure 4 shows double row cylindrical roller bearings of dimension series 49. They are used in preference for the bearing arrangements of work rolls. In order to reduce the stresses occurring as a result of any moments, the bearing rings are separated from each other by inner and outer spacers. In these bearings, a high load carrying capacity is less important, while it is much more important that they are suitable for high speeds.
Cylindrical roller bearings in accordance with Figure 5 are used predominantly in fine-section and wire mills. They have machined brass or steel cages. In comparison with their suitability for high rolling speeds of up to 40 m/s, they have high load carrying capacity.
In the finishing sections of such lines with rolling speeds of up to 100 m/s and higher, rolling is carried out on a single strand. Normally, single row cylindrical roller bearings are used in these cases. The operating life achieved in practice with these bearings is sufficient for their purpose.



3: Four-row cylindrical cylindrical roller bearing with through-drilled rollers and pin-type cage



4: Double row cylindrical roller bearings of dimension series 49 with inner and outer spacers between bearing rings

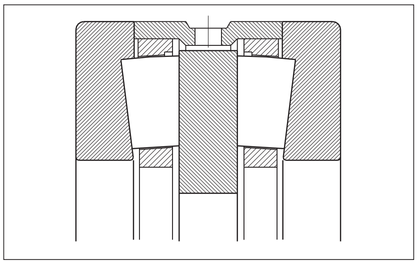


5: Four-row cylindrical roller bearing with machined cage for high rolling speed

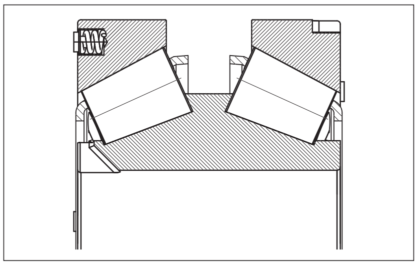
Roll neck bearings

Axial bearings

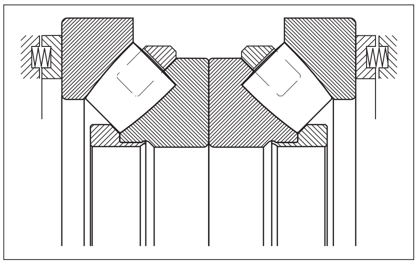
The chock is generally axially located in the roll stand at the operator's end. This chock transmits the axial forces to the roll stand. Bearings of various types are used as axial bearings. The bearings used for high axial forces and moderate speeds are axial tapered roller bearings (Figure 6), double row tapered roller bearings with a large contact angle (Figure 7) or axial spherical roller bearings (Figure 8). The axial tapered roller bearing has a ring fitted between the housing washers whose width is matched to the requisite axial internal clearance. Axial tapered roller bearings, double row tapered roller bearings and axial spherical roller bearings are used mainly in blooming mills, heavy plate mills and hot strip mills, where considerable axial forces occur in combination with low to moderate speeds. During operation, only one bearing row is subjected to pure axial load.
The other is not subjected to load. In order to prevent distorted rolling kinematics, the outer rings of the double row tapered roller bearings and the housing washers of the axial spherical roller bearings are preloaded on both sides to a minimum load by means of springs (Figures 7 and 8).
In strip mills, in fine-section and wire mills, the rolling speeds are so high in many cases that it is no longer possible to use axial tapered roller bearings and axial spherical roller bearings. The axial bearings used in these cases are angular contact ball bearings or deep groove ball bearings.
In back-up rolls for large four-high strip mills and foil mills, a deep groove ball bearing is often sufficient as an axial bearing, Figure 9. It generally has the same section height as the radial cylindrical roller bearing to which it is assigned.
In place of the deep groove ball bearing, a double row tapered roller bearing with a large contact angle can be used. The requisite basic load ratings can then be achieved with a significantly smaller bearing. In conjunction with considerably smaller adjacent parts, the double row tapered roller bearings allow more economical designs.
The work rolls in four-high strip mills and the rolls in two-high fine-section and wire mills are predominantly fitted with angular contact ball bearings as axial bearings (Figure 10).
The chock arranged on the drive end is not axially located in the roll stand, but is guided on the roll neck by the axial bearing. Since the guidance forces are not very high, a deep groove ball bearing is fitted here. This only increases the width of the bearing arrangement to a small degree.
It is advisable to use a deep groove ball bearing with the same section height as the radial bearing.
In some roll bearing arrangements, the same axial bearing is fitted on both the drive end and the operator's end. This gives simpler stockholding (page 48).
The deep groove ball bearings and angular contact ball bearings fitted in these bearings are only required to support axial forces. In order to prevent the outer rings from transmitting any radial forces, the chocks are relief-turned by a few millimetres at the seats for the bearing outer rings (see also Table 40, page 31).



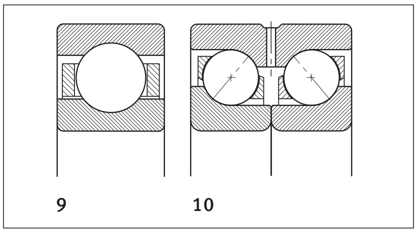
6: Double direction axial tapered roller bearing with intermediate ring



7: Double row tapered roller bearing with large contact angle and outer rings axially adjusted by springs



8: Pair of axial spherical roller bearings for supporting axial load in both directions

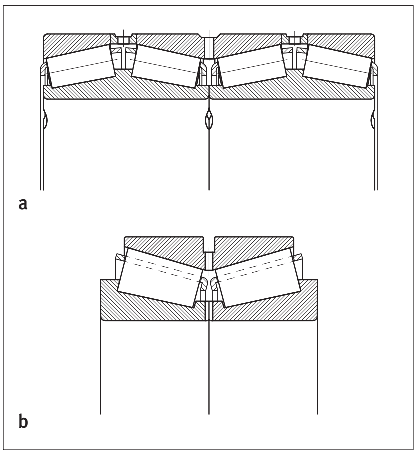


9: Deep groove ball bearing
10: Double row angular contact ball bearing

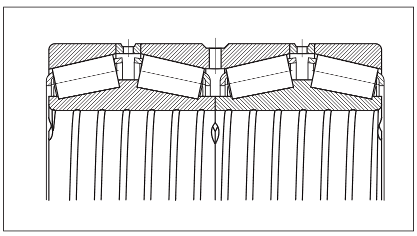
Roll neck bearings

Tapered roller bearings

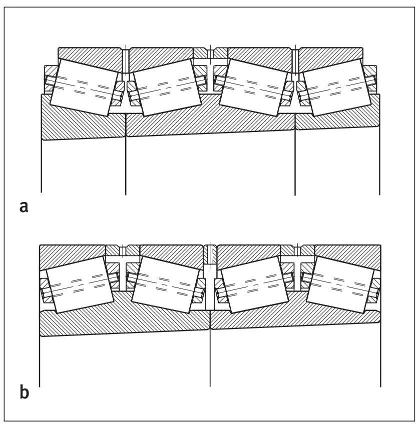
Due to the inclined position of the rollers, tapered roller bearings can support both radial and axial forces. Four-row and double row tapered roller bearings are used in rolling mills, Figure 11.
Tapered roller bearings are separable; unlike cylindrical roller bearings, however, it is not possible to fit the inner rings on the neck first, mount the outer rings in the chock and then slide the chock onto the roll neck.
The complete bearing must instead be mounted in the chock, after which the chock together with the bearing is slid onto the neck.
This means that the bearing inner ring must have a loose fit on the neck, although it should technically have a tight fit due to the circumferential load.
With a loose fit, the inner ring will inevitably creep on the neck. The neck will undergo heating and wear in this case. Wear can be restricted by good lubrication of the fit joint between the inner ring and roll neck, see also page 36.
In order to provide a grease reservoir and thus improve lubrication of the neck, a spiral groove is sometimes turned in the inner ring bore, Figure 12. The groove also provides for collection of abraded particles. For the same reason, radial grooves are provided in the lateral faces of the inner rings.
Where work rolls are supported in four-row tapered roller bearings, the low load means that little wear occurs. Furthermore, it is likely in most cases that the regrinding stock available on work rolls will have been used up and the rolls must be replaced before wear of the neck has a deleterious effect.
Large tapered roller bearings are, like cylindrical roller bearings, fitted with through-drilled rollers and pin-type cages. This design is necessary in reversing type stands due to the high inertia forces.
For the reasons given, the four-row tapered roller bearing with a cylindrical bore cannot be used in all roll neck bearing arrangements.
At high speeds and under high loads, the inner rings must have a tight fit. In these cases, bearings with a tapered bore are normally selected and mounted on a tapered roll neck, Figure 13.
In this way, the requisite tight fit can be easily achieved.
In the design shown in Figure 13a, the inner ring comprises a double ring and two single rings, while the outer ring comprises two double rings. Figure 13b shows a different design with an outer ring comprising four single rings separated by three spacer rings.
Schaeffler manufactures four-row tapered roller bearings in metric dimensions and metric tolerances as well as in inch size dimensions with inch size tolerances.



11: Tapered roller bearings
a: Four-row;
b: Double row



12: Four-row tapered roller bearing with spiral groove in inner ring bore



13: Four-row tapered roller bearing with tapered bore and pin-type cage.
a: Outer ring comprising two double rings
b: Outer ring comprising four single rings

Roll neck bearings

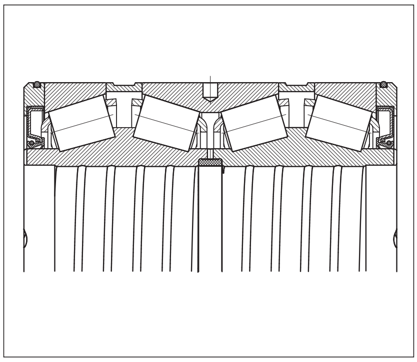
Tapered roller bearings

Sealed multi-row tapered roller bearings

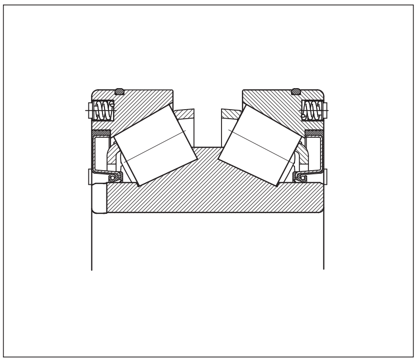
Work roll bearing arrangements in hot and cold rolling lines must be effectively sealed against large quantities of water or roll coolant that are mixed with contaminants. The work roll bearing arrangements are normally lubricated with grease. In order to reduce costs and protect the environment, plant operators try to reduce grease consumption. Better lubrication and cleanliness at the rolling contacts can help to increase bearing life.
In order to fulfil these objectives, Schaeffler has developed four-row tapered roller bearings with integrated seals, Figure 14.

The bearings have the same main dimensions as the unsealed bearings. A high quality rolling bearing grease is used that does not escape from the bearings and is only required in small quantities. The housing seals themselves are packed with simple, cheap sealing grease. Although the integrated seals reduce the design envelope available for the rollers, leading to a lower basic load rating, the sealed bearings normally have a longer life than the unsealed bearings due to the improved cleanliness in the lubrication gap.

Double row sealed tapered roller bearings are used as axial bearings for work rolls, Figure 15.



14: Sealed four-row tapered roller bearing of D1 design



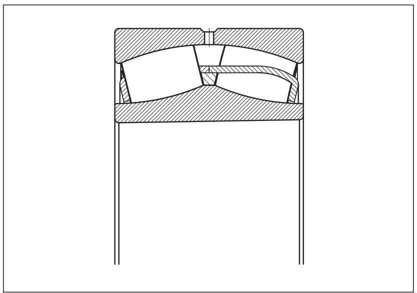
15: Sealed double row tapered roller bearing

Roll neck bearings

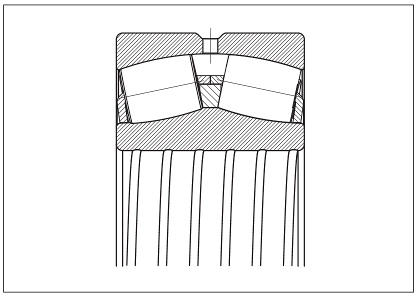
Spherical roller bearings · Axial tapered roller bearings for screw-down mechanisms

Spherical roller bearings

Spherical roller bearings are mainly used as roll neck bearings where the demands on axial guidance accuracy are not particularly high and the speed is low.
Since the available height is restricted, spherical roller bearings of the dimension series 240 and 241 are normally used. These bearings have a small section height, Figure 16.
Spherical roller bearings are self-aligning; they can support radial and axial forces. Since the axial internal clearance is between four and six times the radial internal clearance, their axial guidance accuracy is low. Spherical roller bearings can be used at low and moderate speeds. The rolling speed should be no more than approx. 12 m/s. Due to the self-alignment facility of the bearings, the chock can be supported easily in the roll stand: inaccuracy of the roll stand and deflection of the roll neck are compensated within the bearing. Spherical roller bearings are also used in prestressed stands where the chocks are fixed by means of tie bars and therefore cannot freely align themselves.
Where it must be possible to remove the spherical roller bearing quickly and easily from the roll neck and the rolling speed is low, the inner rings can have a loose fit. As in the case of tapered roller bearings (see Figure 12, page 7), spherical roller bearings can also have a spiral groove turned in the bearing bore to give better lubrication of the fit surfaces, Figure 17. Where the inner rings of spherical roller bearings have a tight fit on the roll neck, mounting and dismounting is easiest if bearings with a tapered bore are used. Mounting can be aided by means of the hydraulic method.
Spherical roller bearings are also used for rolls with cantilever mounting, since they can align themselves to the considerable deflections of the rolls occurring in these cases. Due to the relatively large axial internal clearance, an additional axial bearing must be incorporated in stands used for rolling of shape sections.



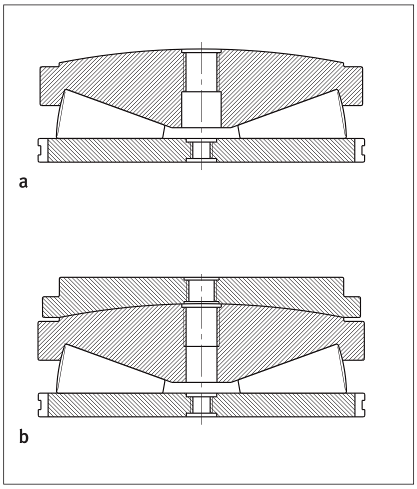
16: Spherical roller bearing



17: Spherical roller bearing with spiral groove in inner ring bore

Axial tapered roller bearings for screw-down mechanisms

Single direction axial tapered roller bearings are often mounted between the pressure spindle and the upper chock, Figure 18. Due to their low friction, these bearings reduce the screw-down forces.
This is particularly advantageous in large stands and in stands with frequent changes of rolled stock thickness.

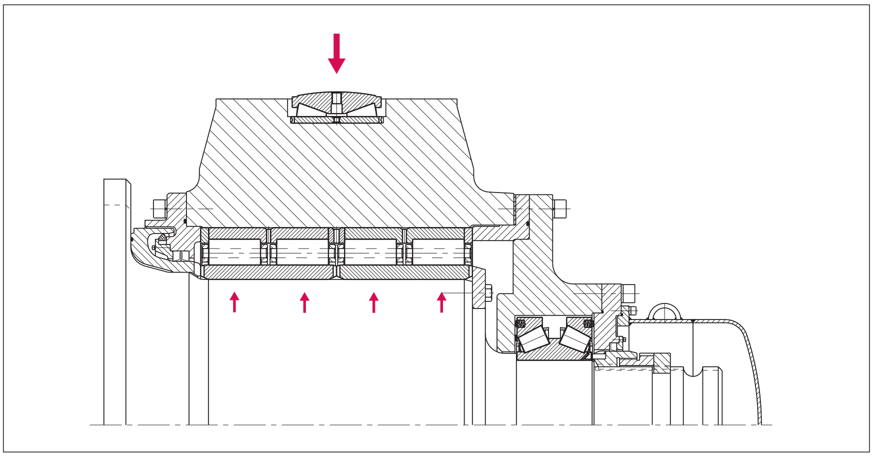


18: Axial tapered roller bearings for screw-down mechanisms
a: Design without pressure washer
b: Design with pressure washer

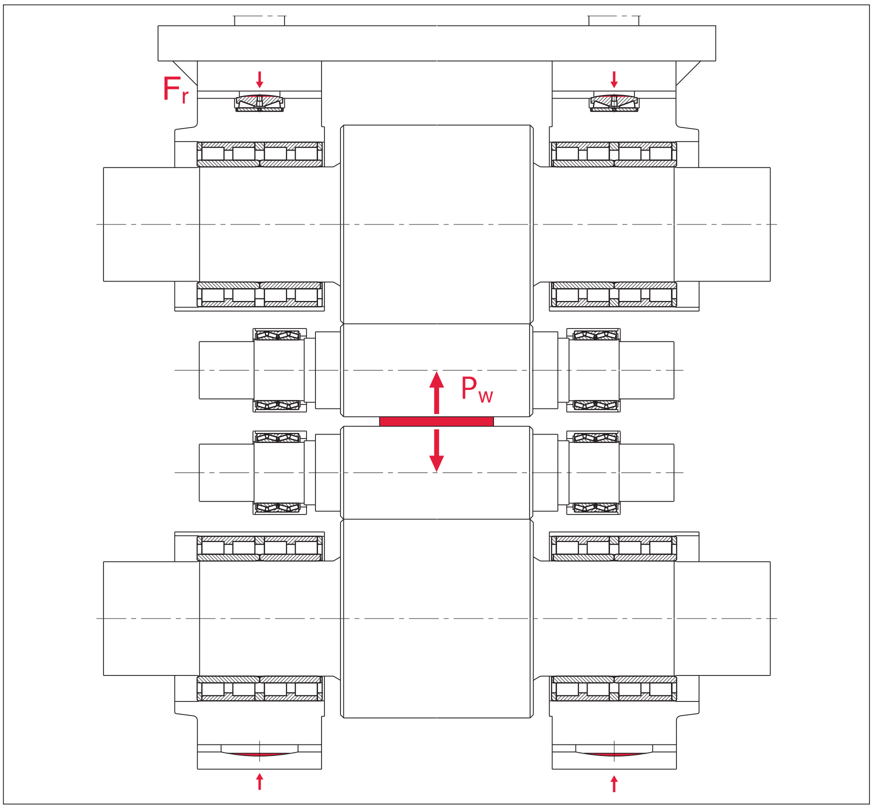
Calculation of rolling force is generally carried out with the aid of computer programs. A decisive influence is exerted by the type of rolled stock, the type of rolling (strip or groove rolling) and the proposed rolling schedule.
The rolling forces actually occurring sometimes differ considerably from the calculated results if the rolling schedule does not correspond to the projected schedule. Furthermore, the shocks occurring when the rolled stock enters the rolls is only considered in approximate terms in the calculation.
The rolling force in the initial pass may be more than twice the subsequent rolling force. The magnitude of these initial pass peaks is dependent on the configuration of the rolled stock and the temperature of the rolled stock ends. The initial pass peak in rolling force only occurs for a short time. It is not generally taken into consideration in life calculation.
It must not be overlooked, however, that the fatigue life of the rolling bearings is often reduced to a considerable extent by such stresses.
The distribution of the rolling force over the two bearing positions is dependent on the design of the roll stand and the type of rolled stock.

Self-aligning chocks

The chocks are supported independently in the roll stands.
The rolling forces are transmitted to the stands via pressure bearings (axial tapered roller bearings) with crowned contact surfaces. As a result, the chocks can adapt to the specific position of the roll neck as affected by roll deflection, non-uniform screw-down conditions etc. This ensures that all the rows of rollers in the multi-row bearings are subjected to uniform load, Figure 19. The rolled stock passes symmetrically between the bearing positions (Figure 20), each roll neck bearing being subjected to a load of $\frac{1}{2} \cdot P_w$.
$F_r = \frac{1}{2} \cdot P_w$



19: Self-aligning chock



20: Self-aligning chocks for strip rolling

Calculation of bearing load

Self-aligning chocks

Groove rolling

A distinction must be made between rolls with different grooves (e. g. in blooming mills) and rolls with identical grooves (e. g. in wire mills).
In rolls with different grooves, a pass schedule should be calculated indicating the time percentages and the rolling forces in the individual grooves. This can be used to determine the load acting on the two roll necks. Life calculation is based on the mean load acting on the most heavily loaded neck.

In rolls with identical grooves, the individual neck loads can be calculated from the pass schedule.

Alternatively, the following guide values can be used for the most heavily loaded neck:

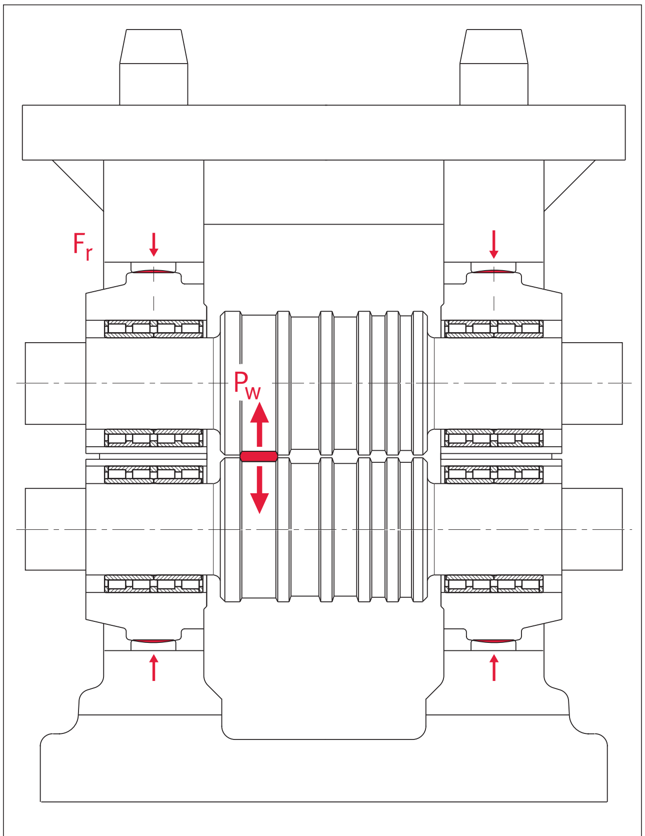
Single strand rolling:
max. neck load $F_r = 0{,}67 \cdot P_w$

Two-strand rolling:
max. neck load $F_r = 1{,}1 \cdot P_w$

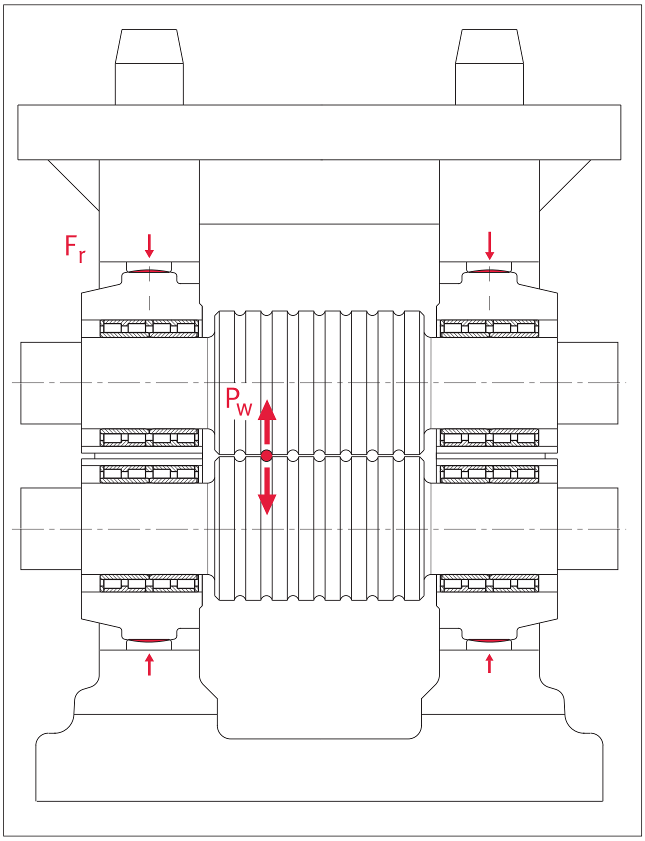
Four-strand rolling:
max. neck load $F_r = 2{,}0 \cdot P_w$

P_w = rolling force, relative to one strand.

The calculation of bearing load at variable speed and under variable load is described on page 17.



21: Self-aligning chocks:
Rolls with different grooves

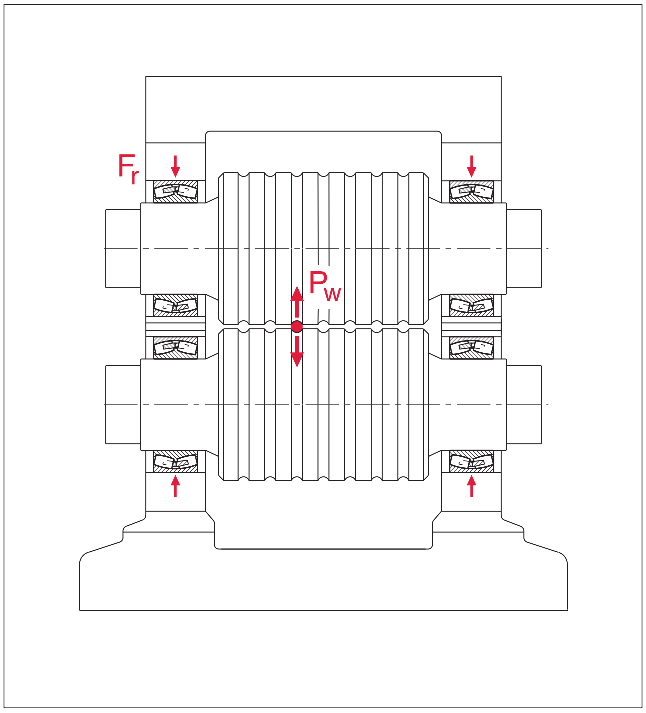


22: Self-aligning chocks:
Rolls with identical grooves

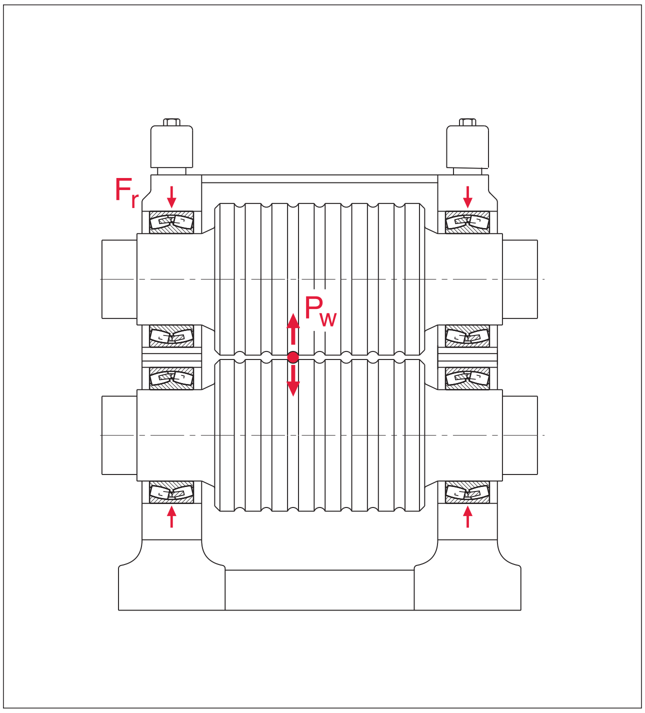
Rigid chocks

Both bearings are mounted in housings that are rigidly connected to each other. Roll deflections, neck offsets or misalignments cause mutual tilting of the two bearing rings. This has no influence on the bearings and their calculation if the necks are supported in spherical roller bearings.
In the case of double row or multi-row cylindrical roller bearings, it must be anticipated that there will be an uneven distribution of load over the rows of rollers. The method developed by Schaeffler for calculating roll deflection can be used to determine the load on the individual rows of rollers. It must then be checked whether the row of rollers subjected to higher load has an adequate fatigue life.
Rigid chocks are selected predominantly for shape section rolls.
The distribution of rolling force over the two roll necks can be calculated as shown on page 11.

The upper and lower chock are pressed together by the preload force, which means that they cannot adjust to misalignment. This can lead not only to roll deflection but also to offset of the two chocks relative to the roll axis. These stands are predominantly fitted with spherical roller bearings. If no axial bearing is provided, the axial force must be accommodated in the locating bearing.



23: Rigid chocks



24: Prestressed stands

Calculation of bearing load

Calculation of roll deflection and load conditions in the rolling bearings

Calculation of roll deflection and load conditions in the rolling bearings

The software Bearinx® can be used to calculate the deflection behaviour of elastic rolls under different load conditions that are supported by elastic means. The support reactions, internal stresses in the rolling bearings, the comparative stresses in the shafts and the most important calculation results are outputted in numerical and graphical form.

The following influences can be analysed:

- elasticity of plain and stepped solid and hollow rolls made from different materials, deformation due to shear forces.
- shaft loads due to rolling forces and bending moments or other external forces acting on the bearings.
- shaft support in the form of rolling bearings with non-linear elasticity, taking account of bearing geometry, bearing clearance, rolling element and raceway profiles as well as special conditions in support of loads.
- creation and calculation of any number of load cases (combinations of load and speed).

The following calculation results are outputted:

The deflection and inclination of the roll axis at any point, the shear forces and bending moments, the stresses, the bearing reaction forces, the bearing elasticity, the load conditions within the rolling bearings and pressure distribution at the rolling contacts of individual rolling elements. Based on the stress calculated for the individual rolling contacts, Bearinx® determines the bearing life to high precision.

Calculation example for roll deflection and load conditions in the rolling bearings

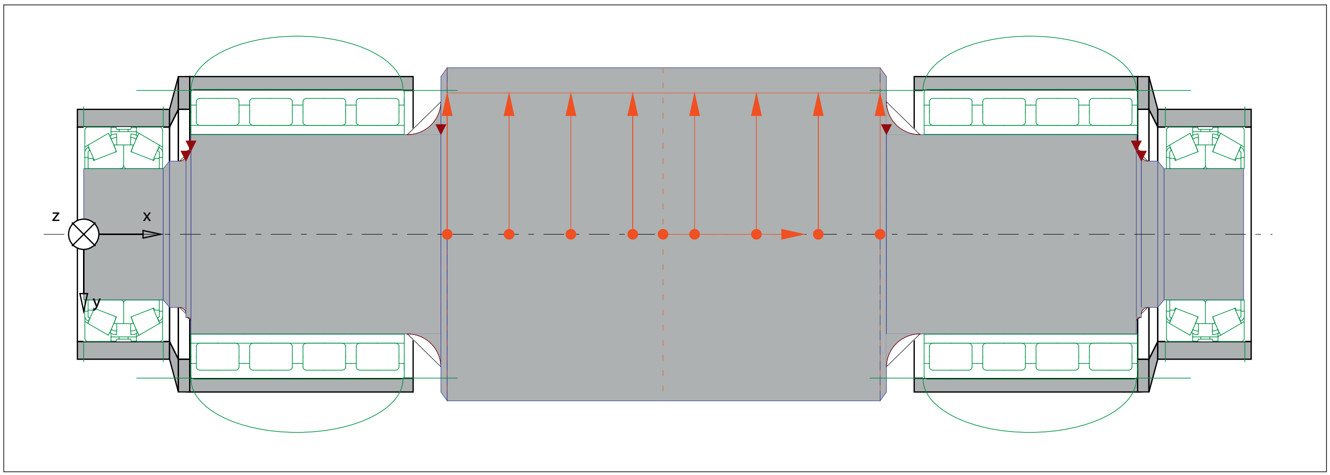
The subject of calculation is the work roll and back-up roll of a four-high cold rolling mill.

Load:
Rolling force $P_w = 8\,000$ kN

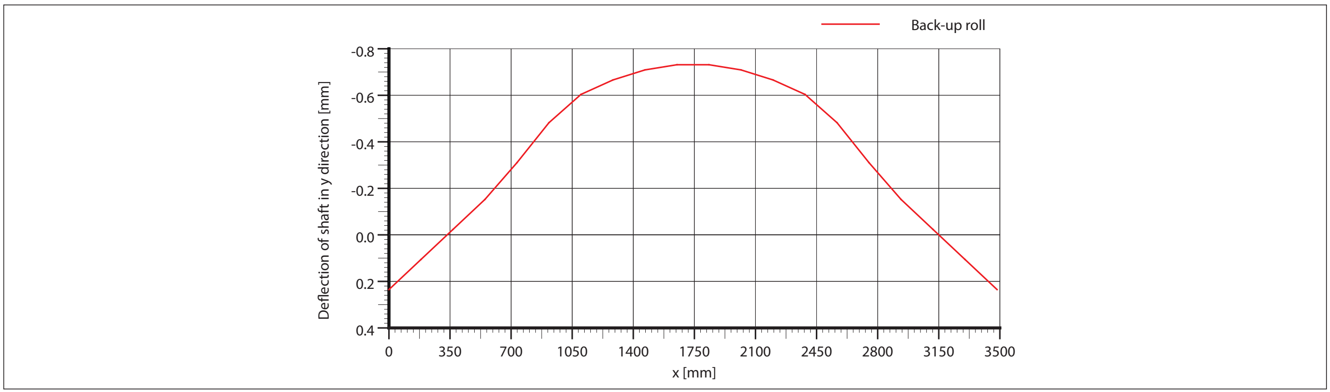
During the input operation, the external form of the roll is described. The rolling force can be inputted either as a line load or split into individual loads that act at different points on the roll body distributed over the width of the rolled stock. The chocks are considered as systems into which forces and/or moments are introduced. The self-alignment facility of the chocks is taken into consideration. Cylindrical roller bearings and tapered roller bearings are used as roll neck bearings. They have non-linear spring characteristics.

Calculation of bearing load

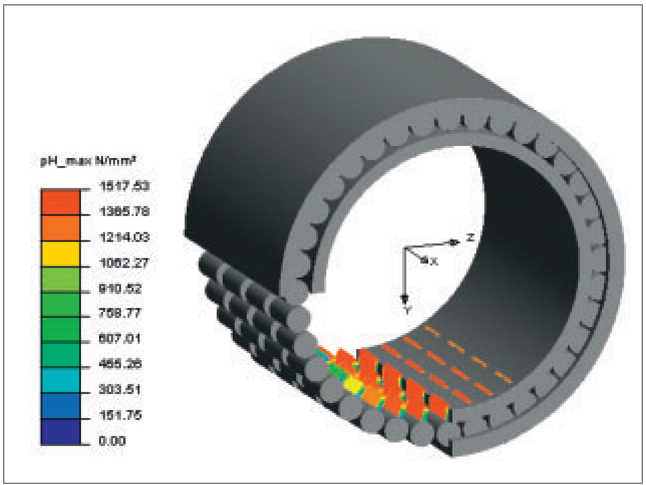
Back-up roll: Calculation of load conditions and pressures (pressure distribution)



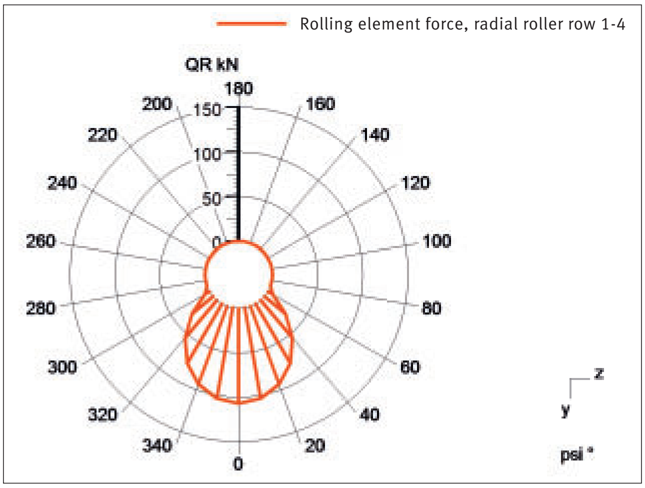
25a: Back-up roll bearing arrangement



25b: Resultant deflection of the back-up roll in the Y direction



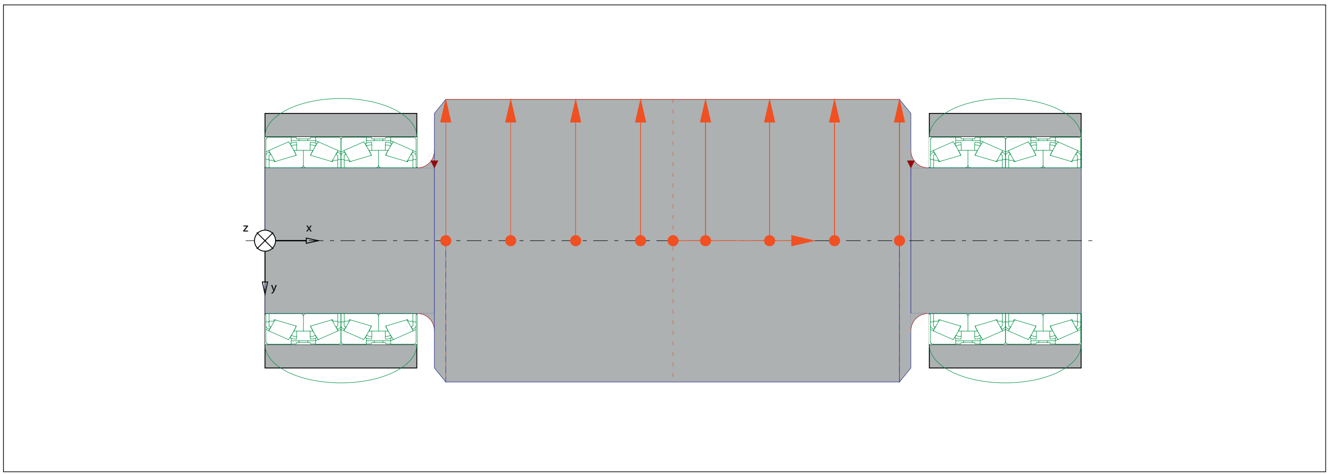
25c: Visualisation of the pressures acting on the four-row cylindrical roller bearing on the back-up roll



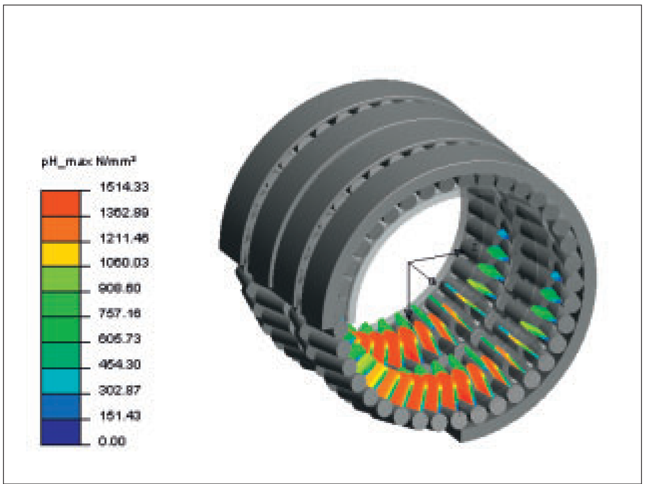
25d: Load distribution within the four-row cylindrical roller bearing on the back-up roll

Calculation of bearing load

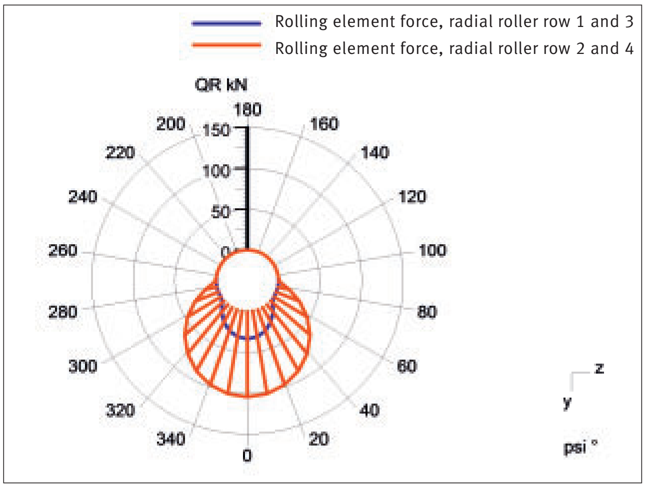
Work roll: Calculation of load conditions and pressures (pressure distribution)



26a: Work roll bearing arrangement



26b: Visualisation of the pressures in the four-row tapered roller bearing on the work roll



26c: Load distribution in the four-row tapered roller bearing on the back-up roll

Load carrying capacity and life

Bearings under static loading · Bearings under dynamic loading

In the calculation of dimensions, the loading in a bearing is compared with its load carrying capacity. A distinction is made here between dynamic loading and static loading. In the case of static loading, the bearing remains stationary (without relative motion between the rings) or rotates slowly. In these cases, the security against excessive plastic deformation of the raceways and rolling elements is checked.
Most bearings are subjected to dynamic loading. In these cases, the bearing rings rotate relative to each other. The dimension calculation checks the security against premature material fatigue of the raceways and rolling elements.

Bearings under static loading

In the case of static loading, calculation is carried out to demonstrate that a bearing with adequate load carrying capacity has been selected, using the static load safety factor S_0.

$$S_0 = \frac{C_0}{P_0}$$

where

S_0 Static load safety factor
C_0 Basic static load rating
P_0 Equivalent static load.

The static load safety factor S_0 is the safety factor against excessive plastic deformation at the contact points of the rolling elements.
Roll neck bearings are not normally checked for static load safety. However, bearings for screw-down mechanisms are an exception here. The recommendation here is: $S_0 = 1{,}8...2$
The basic static load rating C_0 is indicated for each bearing in the dimension tables in our catalogues.

Bearings under dynamic loading

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

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

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

L_{10} 10^6 revolutions
The basic rating life in millions of revolutions is the life reached or exceeded by 90 % of a sufficiently large group of apparently identical bearings before the first evidence of material fatigue develops

L_{10h} h
The basic rating life in operating hours according to the definition for L_{10}

C kN
Basic dynamic load rating

P kN
Equivalent dynamic bearing load for radial and axial bearings

p –
Life exponent;
for roller bearings: $p = 10/3$
for ball bearings: $p = 3$

n min^{-1}
Operating speed.

Equivalent dynamic load

The equivalent dynamic load P is a calculated value. This value is constant in magnitude and direction; it is a radial load for radial bearings and an axial load for axial bearings.
A load corresponding to P will give the same rating life as the combined load occurring in practice.

$$P = X \cdot F_r + Y \cdot F_a$$

P kN
Equivalent dynamic bearing load

F_r kN
Radial dynamic bearing load

F_a kN
Axial dynamic bearing load

X –
Radial factor from the dimension tables or product description

Y –
Axial factor from the dimension tables or product description.

The values for X and Y as well as guidelines on calculating the equivalent dynamic load for the various rolling bearings are indicated in Catalogue GL1.
While the radial loads acting on bearings in roll bearing arrangements can be determined with sufficient accuracy, little is generally known about the magnitude of the axial forces, which means that these must be estimated. Practice has shown that the following assumptions give adequate security:

for plain rolls (in two-high and four-high strip mills)
axial force = 0,5...2 % of rolling force
for grooved rolls
axial force = 5...10 % of rolling force

For radial bearings that support radial forces only:
$P = F_r$.

For axial tapered roller bearings that, due to their design, can support axial forces only:
$P = F_a$.

For four-row tapered roller bearings, only one row of rollers is normally considered.
For purely radial load or for $F_a/F_r \leq e$ is
$P = F_r$ (for one row).
For $F_a/F_r > e$ is
$P = 0,4 \cdot F_r + Y \cdot F_a$ (for one row).
e is an auxiliary calculation value, see dimension tables in GL1.
The calculation of the adjusted rating life and expanded adjusted rating life can be found in Catalogue GL1.

Equivalent operating values

The rating life equations assume that the bearing load P and bearing speed n are constant. If the load and speed are not constant, equivalent operating values can be determined that induce the same fatigue as the actual conditions.

The equivalent operating values calculated here already take account of the life adjustment factors a_3 or a_{ISO}. They must not be applied again when calculating the adjusted rating life.

Variable load and speed

If the load and speed vary over a time period T, the speed n and equivalent bearing load P are calculated as follows:

$$n = \frac{1}{T}\int_0^T n(t) \cdot dt$$

$$P = \sqrt[p]{\frac{\int_0^T \frac{1}{a(t)} \cdot n(t) \cdot F^p(t) \cdot dt}{\int_0^T n(t) \cdot dt}}$$

Variation in steps

If the load and speed vary in steps over a time period T, n and P are calculated as follows:

$$n = \frac{q_1 \cdot n_1 + q_2 \cdot n_2 + \ldots + q_z \cdot n_z}{100}$$

$$P = \sqrt[p]{\frac{\frac{1}{a_i} \cdot q_i \cdot n_i \cdot F_i{}^p + \ldots + \frac{1}{a_z} \cdot q_z \cdot n_z \cdot F_z{}^p}{q_i \cdot n_i + \ldots + q_z \cdot n_z}}$$

Variable load at constant speed

If the function F describes the change in load over a time period T and the speed is constant, P is calculated as follows:

$$P = \sqrt[p]{\frac{1}{T}\int_0^T \frac{1}{a(t)} \cdot F^p(t) \cdot dt}$$

Load varying in steps at constant speed

If the load varies in steps over a time period T and the speed is constant, P is calculated as follows:

$$P = \sqrt[p]{\frac{\frac{1}{a_i} \cdot q_i \cdot F_i{}^p + \ldots + \frac{1}{a_z} \cdot q_z \cdot F_z{}^p}{100}}$$

Constant load at variable speed

If the speed varies but the load remains constant, the following applies:

$$n = \frac{1}{T}\int_0^T \frac{1}{a(t)} \cdot n(t) \cdot dt$$

Constant load at speed varying in steps

If the speed varies in steps, the following applies:

$$n = \frac{\frac{1}{a_i} \cdot q_i \cdot n_i + \ldots + \frac{1}{a_z} \cdot q_z \cdot n_z}{100}$$

Roll neck bearings can in principle be lubricated with grease or oil. The lubricant should, as in the case of other rolling bearings, form a lubricant film capable of transmitting load that prevents the bearing parts from coming into contact with each other, leading to surface damage. The thickness and load carrying capacity of the lubricant film is dependent on the viscosity of the oil, the speed of the bearing, the bearing size and the lubrication characteristics. The lubricant also has the function of protecting the bearing parts against corrosion. Where sealing is present, it should lubricate the lips of the sealing rings (collar seals etc.) and act as a barrier agent.

Grease lubrication

Due to the simple seal arrangement and the ease of relubrication, roll neck bearings are lubricated with grease where permitted by the operating conditions. A large number of special rolling bearing greases are available in the market. However, these greases vary significantly in their key data and characteristics. The decision as to which grease should be used in a specific case must take account of the suitability and performance capability of the grease in the rolling bearing. The appropriate test results are not available in all cases and are not always published in a systematic manner in datasheets. Grease selection exclusively on the basis of product datasheets is therefore not recommended for rolling bearing applications with challenging requirements. Since the lubricant in the seal has functions that differ from those in the bearing arrangement, it would be advisable to lubricate the bearing and seal separately and select a lubricant for each lubrication system that is most suitable in the specific case. While this approach is correct in principle, there is often a tendency in practice, however, to restrict the variety of lubricants used. This reduces the risk of mixing up lubricants at relubrication and decreases the work involved in managing several lubricants. Schaeffler offers greases that have been particularly tested and found to be suitable, the FAG rolling bearing greases known as Arcanol. Table 27 gives an overview of the most important rolling bearing greases and their characteristics. It is recommended that detailed advice should be sought in every case. The suitability of FAG rolling bearing greases for the different bearing designs is known. Arcanol greases offer excellent protection against corrosion and are particularly resistant to the influence of water. In the case of unfamiliar greases, the supplier must demon-

27: The most important rolling bearing greases and their characteristics

Grease type Thickener	Consistency NLGI grade	Special remarks and application examples	FAG Arcanol	Operating temperature °C	Continuous limit temperature °C	Base oil viscosity at 40 °C mm²/s	Speed suitability	Load suitability
Li soap with EP additive	3	Universal rolling bearing grease, long lubrication interval	**MULTI3**	−20...+120	75	110	Moderate	Moderate
Li/Ca soap with EP additive	2	More difficult operating conditions, e. g. in back-up and work rolls, particularly sealed tapered roller bearings	**LOAD220**	−20...+140	80	245	High	High
Li soap with EP additive	2	More difficult operating conditions, especially at high speeds, sealed tapered roller bearings	**MULTITOP**	−50[1]...+140	85	82	Very high	High
Li/Ca soap with EP additive	2	Very difficult operating conditions, in particular high shock loads	**LOAD400**	−30[1]...+130	80	400	Moderate	Very high
Li/Ca soap with EP additive	1	Very difficult operating conditions, in particular high shock loads	**LOAD460**	−30[1]...+130	80	400	Moderate	Very high
Li/Ca soap with EP additive	2	Extremely difficult operating conditions, very high shock loads	**LOAD1000**	−20...+130	80	1000	Low	Very high

[1] Measurement values according to Schaeffler FE8 low temperature starting test

strate their suitability. If necessary, Schaeffler can carry out tests on suitability.

Selection of grease according to loading

Greases are produced in various consistencies. These are defined as NLGI grades, which are determined by means of worked penetration in accordance with ISO 2137. The higher the NLGI grade, the harder the grease.
For rolling bearings, greases of NLGI grades 1, 2 and 3 are used in preference. For the loading profiles typical of rolling mill bearings, greases of grades 2 and 3 should be used in preference. The lubricant must always contain an additive formulation that offers effective protection against wear.

Influence of bearing type

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

Bearings with point contact

In 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. Typically, greases with a base oil viscosity of ISO VG 100 or greater are used.

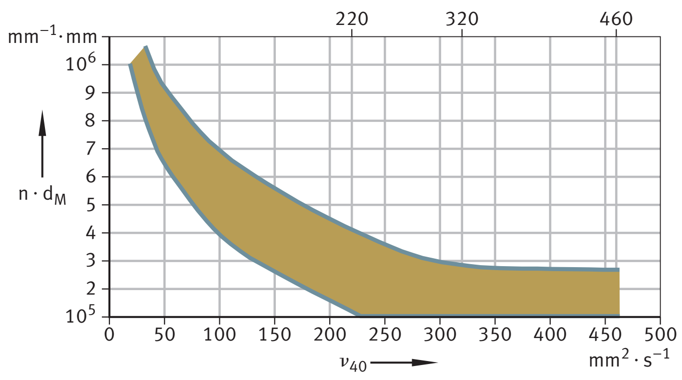
Bearings with line contact

Roller bearings with line contact place higher requirements on the grease. Not only is a larger grease quantity at the contact subjected to stress, but increased proportions of sliding motion in rolling contact and additional rib friction must also be expected. This hinders the formation of a lubricant film and can lead to mixed friction. As a countermeasure, greases for bearings with line contact exhibit higher base oil viscosity (ISO VG 150 to 460 and, in special cases, even higher). The consistency is normally NLGI 2.

Influence of speed

Greases have a maximum permissible speed parameter $n \cdot d_m$ that is specific to the design of rolling bearing. This speed parameter of the grease should be significantly higher than the requisite speed in the application. If the speed suitability of the grease is too low, this will lead to an increased bearing temperature and a reduced grease operating life. The speed parameter is dependent on the type and proportion of the thickener, the base oil type and the base oil viscosity. These data can be found in some cases in the technical datasheets for the greases.
Typically, greases for high speeds have a low base oil viscosity. Greases for low speeds have a higher base oil viscosity and are frequently also used as heavy duty greases. An overview of this relationship is shown in Figure 28.
For selection of a suitable grease, an initial guide can be given as follows:

- For rolling bearings rotating at high speeds or with a low requisite starting torque, a grease with a high speed parameter should be selected.
- For bearings rotating at low speeds, greases with a low speed parameter are recommended.



28: Speed suitability of various base oil viscosities

Base oil viscosity

In addition to the speed, the base oil viscosity has a direct influence on formation of the lubricant film. In normal cases, the base oil viscosity of the grease should therefore be selected such that good lubrication conditions are present under the operating mode. The base oil viscosity required in this case can be estimated with the aid of the kappa value, where the relationship is as follows: $\kappa = \nu/\nu_1$.
For effective lubricant film formation, the objective should be to achieve kappa values of 2.
With kappa values of less than 1, mixed friction must be expected, which leads to wear and premature bearing failure if insufficient protection against wear is provided. An effective additive formulation is even more important in this case.

Viscosity ratio

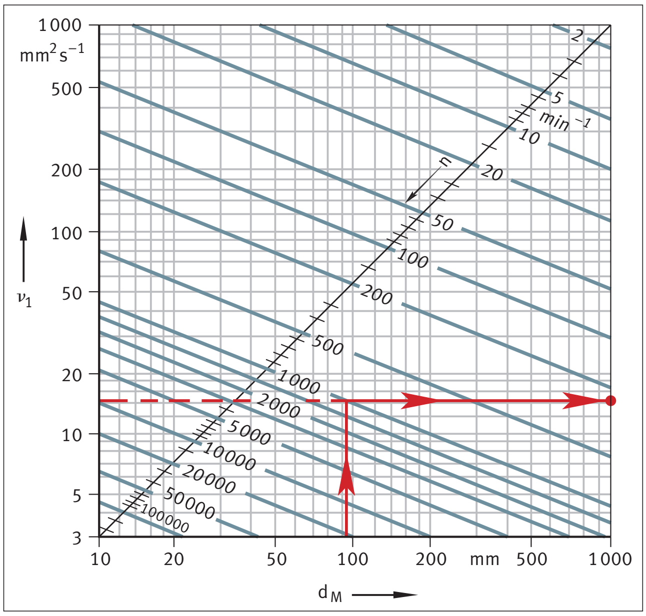
The viscosity ratio κ is an indication of the quality of lubricant film formation:

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

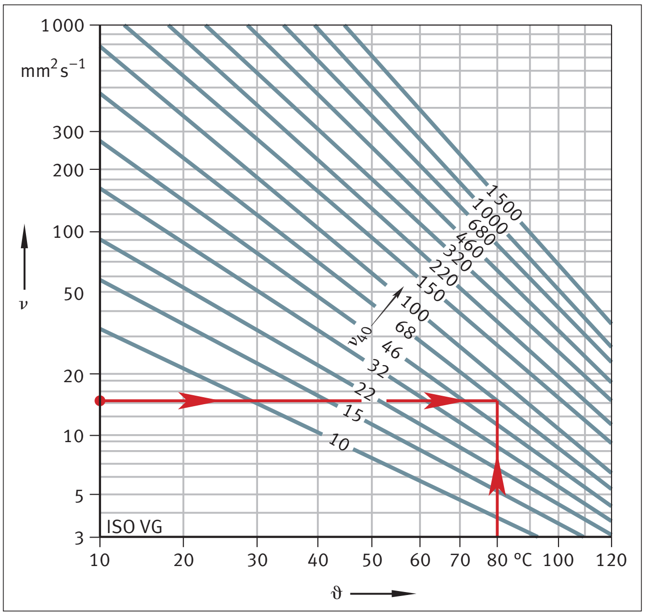
ν mm^2s^{-1}
Kinematic viscosity of lubricant at operating temperature

ν_1 mm^2s^{-1}
Reference viscosity of lubricant at operating temperature.

The reference viscosity ν_1 is determined from the mean bearing diameter $d_M = (D + d)/2$ and the operating speed n, Figure 29.
The nominal viscosity of the oil at +40 °C is determined from the requisite operating viscosity ν and the operating temperature ϑ, Figure 30. In the case of greases, ν is the operating viscosity of the base oil.
In the case of heavily loaded bearings with a high proportion of sliding contact, the temperature in the contact area of the rolling elements may be up to 20 K higher than the temperature measured on the stationary ring (without the influence of any external heat sources).



29: Reference viscosity ν_1
ν_1 = reference viscosity
d_M = mean bearing diameter
n = speed

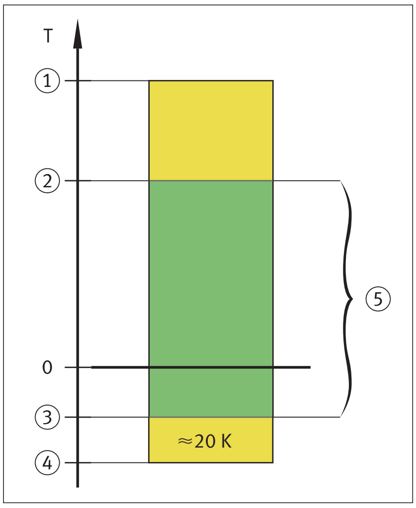


30: V/T diagram for mineral oils
ν = operating viscosity
ϑ = operating temperature
ν_{40} = viscosity at +40 °C

Influence of temperature

The operating temperature range of the grease should correspond to the range of possible operating temperatures in the rolling bearing. Figure 31 shows the temperatures that are important for greases.
The operating temperature range between the points ① and ④ 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.
It is generally recommended that greases should be used in accordance with the bearing temperature normally occurring in the standard operating range (between the temperatures ② and ③), in order to achieve reliable lubrication and an acceptable grease operating life.

The continuous limit temperature of the grease (②) is of particular importance here: The grease can only be used continuously up to this temperature without a reduction in its performance capability.
An appropriate estimate is that each increase of 15 degrees above the continuous limit temperature leads to a reduction by half in the grease operating life. This must always be taken into consideration in the selection of grease. The use of a grease with lower temperature suitability (continuous limit temperature < continuous operating temperature) is thus definitely possible but must be balanced by means of shorter relubrication intervals and in some cases by larger relubrication quantities.



31: Temperature parameters of a grease
① = maximum operating temperature,
② = (upper) continuous limit temperature,
③ = lower continuous limit temperature,
④ = minimum operating temperature,
⑤ = standard operating range

Other operating conditions

In selection of the grease, the position of the roll axis must also be taken into consideration.
If rolls are arranged vertically or at an incline, there is a risk that the grease will escape from the bearing and chock under gravitational force. It is recommended that baffle plates should be arranged under the bearing and that the grease selected should have particularly high adhesion and working resistance with a consistency grade 3, in certain circumstances consistency grade 2.
Relubrication is a further criterion for grease selection. Large relubrication quantities for bearings or seals and long lubrication ducts (e.g. when using central lubrication) require greases with good pumping behaviour. In the case of roll bearing arrangements that operate in a damp atmosphere and are often idle, there is a risk of corrosion because condensation can form in cooling. The greases used must therefore exhibit particularly good protection against corrosion. Bearing positions that may be exposed to spray water must be protected against the ingress of water by appropriate seal systems. The seal and bearing should be relubricated at short intervals.
Table 32 gives an overview of the aspects covered above and can be used, on the basis of the characteristics required, to select a suitable grease.

32: Criteria for grease selection

Selection criterion		Characteristics of the grease
Operating conditions	Vertical bearing axis	Grease to consistency grade 3, frequent relubrication when using softer greases
	Frequent relubrication	Grease with good pumping behaviour in central lubrication systems
	Continuous lubrication	Grease resistant to working, with known operating life and lubrication characteristics
Environmental conditions	Extreme temperatures	Grease whose operating temperature range corresponds to the operating temperature; with continuous relubrication, also greases that withstand the operating temperature at least for a short period and do not tend to solidify
	Contamination by foreign bodies	Stiff grease supports sealing, possibly a special sealing grease to NLGI grade 3
	Corrosion due to condensation	Emulsifying grease (e. g. lithium or lithium/calcium grease)
	Corrosion due to spray water	Water-repellent grease (e. g. calcium complex or lithium/calcium grease)

Oil lubrication

Viscosity requirements

In order that a lubricant film capable of supporting loads is formed in the bearing and the bearing achieves its calculated rating life, the oil must have a certain viscosity at operating temperature as a function of the speed and bearing size. This reference viscosity ν_1 is determined in accordance with Figure 29, page 20. If there are normal expectations for operating life, the operating viscosity ν of the oil for bearings with a small proportion of sliding motion should be at least as high as the reference viscosity ν_1.
Rolling bearing types with unfavourable kinematics (axially loaded roller bearings, large size bearings running at low speeds and under heavy loads) always require effective additives for protection against wear.
Where lubricant film formation is inadequate, these will then form boundary layers at the contact areas raceway/rolling element, rolling element/cage and rolling element/guidance rib that prevent wear and premature failure.

Other demands on the oil

Most rolling bearing oils are mineral oils that contain additives to improve their characteristics. These give, for example, better oxidation stability, improved protection against corrosion or reduced foaming. Dispersion agents hold finely distributed insoluble contaminants in suspension.
EP (extreme pressure) additives are recommended in all cases for roller bearings. For bearing arrangements subject to high thermal stress, oils with particularly good temperature and ageing resistance are available.
Synthetic oils are characterised by good viscosity/temperature behaviour (illustrated in V/T diagrams), which means that their viscosity varies less with temperature than the viscosity of mineral oils. This aspect is principally of importance for bearing arrangements that are subjected to varying temperatures. For extremely high temperatures, the synthetic oils such as polyalphaolefins and polyglycols that have significantly greater resistance to ageing are used in preference. The suitability of oils for the specific application must either be already known from practical experience or must be determined by means of tests.

Methods of oil lubrication

Recirculating oil lubrication is the lubrication method for the normal speed range of roll bearing arrangements and enables not only reliable lubrication but also cooling of the bearing and also carries harmful contaminants and water out of the bearing position. In the case of roll bearing arrangements, it is used as a cooling lubrication system

- where there are energy losses in the bearing itself, in other words under heavy loads and at high speeds,
- or where there is heating of the roll necks by external sources
- or where the heat dissipation conditions are unfavourable.

Oil injection lubrication, in which the lubricant is injected directly into the bearing under pressure via laterally arranged nozzles, is necessary in those cases where recirculating oil lubrication does not give sufficient cooling.
Oil injection lubrication permits extremely high speeds. Recirculating oil lubrication and oil injection lubrication require some outlay on inlet and outlet pipes, pumps, oil containers, filters and if necessary oil cooling systems.
In the case of **oil sump lubrication**, the small lateral spaces in the chocks can provide the bearings with only small quantities of oil. The oil is subjected to heavy stress and therefore undergoes rapid ageing. Oil changes must therefore be made frequently or synthetic oils with high resistance to ageing may need to be used.
In the case of **pneumatic oil lubrication** (minimal quantity lubrication), the oil is fed on a cycle into the lubrication pipe of the bearing by a metering unit and then delivered to the bearing by an air stream. The oil is not atomised. As a result, it is possible to use transmission oils of high viscosity with EP additives.
The small quantity of oil supplied is in addition to the oil sump that is absolutely essential for bearing lubrication. This ensures that lubrication occurs during start-up of the bearing and if there are short disruptions to the oil feed.
The position of oil outlet holes in the chock is defined for a horizontal shaft such that

- for cylindrical roller bearings the lowest rolling element is immersed to 2/3 of its diameter in oil,

- for tapered roller bearings the rolling element/guidance rib contact is still immersed in oil.

The sealing effect is aided by the continuous overpressure that builds up in the air stream in the housing as well as the air flowing outwards at the seals. The escaping oil still contains a small proportion of atomised oil which, on its escape, represents a certain environmental burden.

Design of lubrication system

Fill quantity with grease lubrication

The bearing arrangements should be greased as follows:

- Pack the bearings completely with grease in order to ensure that all functional surfaces are reliably supplied with grease.
- Fill any housing cavity adjacent to the bearing with grease only to the point where there is still sufficient space for the grease in the bearing. This prevents an excessive quantity of grease circulating through the bearing. The free spaces adjacent to the bearing in the chocks are normally large enough to accommodate the grease escaping from the bearing, so this grease filling is not necessary if the bearings run at high speeds.
- In the case of bearings running at very low speeds ($n \cdot d_m < 50\,000\ \text{min}^{-1} \cdot \text{mm}$), pack the bearing and housing completely with grease. The working friction occurring is insignificant.

Relubrication intervals with grease lubrication

The interval after which the grease in a bearing must be supplemented or renewed is dependent on the stress applied to the grease by bearing friction and on the operating speed. The bearing friction is determined by the influences arising from load and from the different conditions of motion in the individual bearing types. However it is also necessary, particularly for roll neck bearings, to take account of the environmental conditions and the effectiveness of the sealing: If the sealing action is inadequate, a drastic reduction in the relubrication interval may be necessary due to the damp atmosphere, spray water and rolling scale.
Guidelines for relubrication can be obtained by checking the condition of the grease and seals after certain running times, ideally when replacing the roll, to determine in particular whether contaminants have been able to enter the bearing.

Lubricant feed

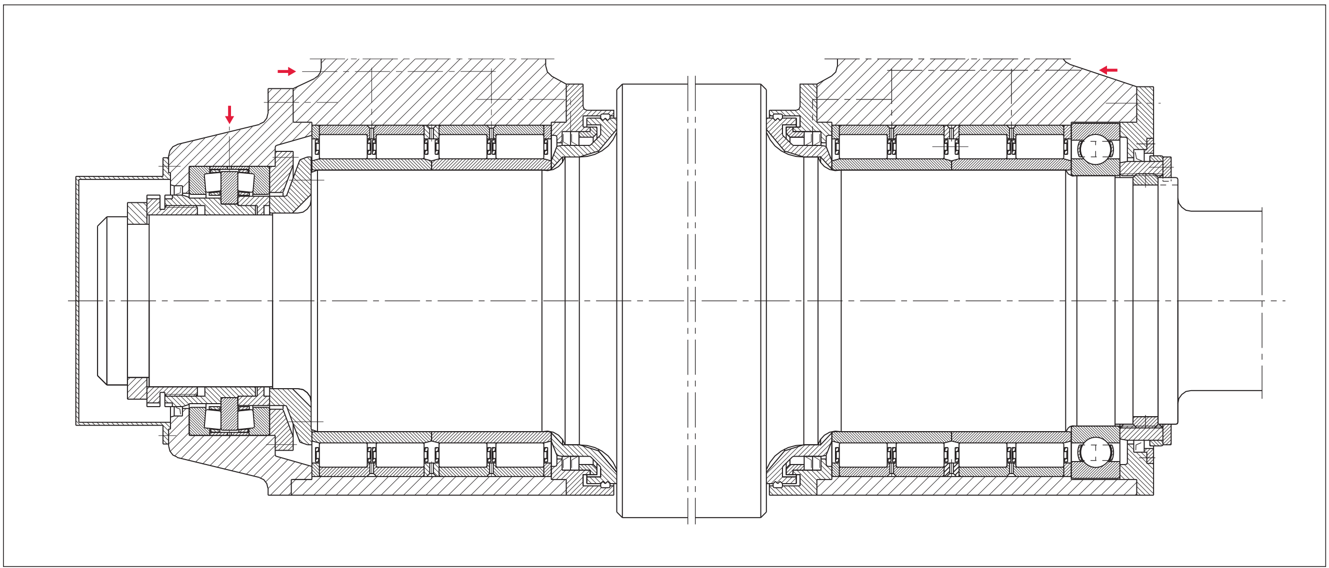
For effective lubrication, it is very important that the grease or oil is fed in a targeted manner.
The lubricant must be reliably supplied to the rolling and sliding surfaces. When using grease lubrication, it must be ensured that excess grease can escape.
Overlubrication can cause increased working action, leading to increased heat generation. This may become so great that the grease is damaged. Lubricant must also be specifically fed to the seals.

Grease lubrication

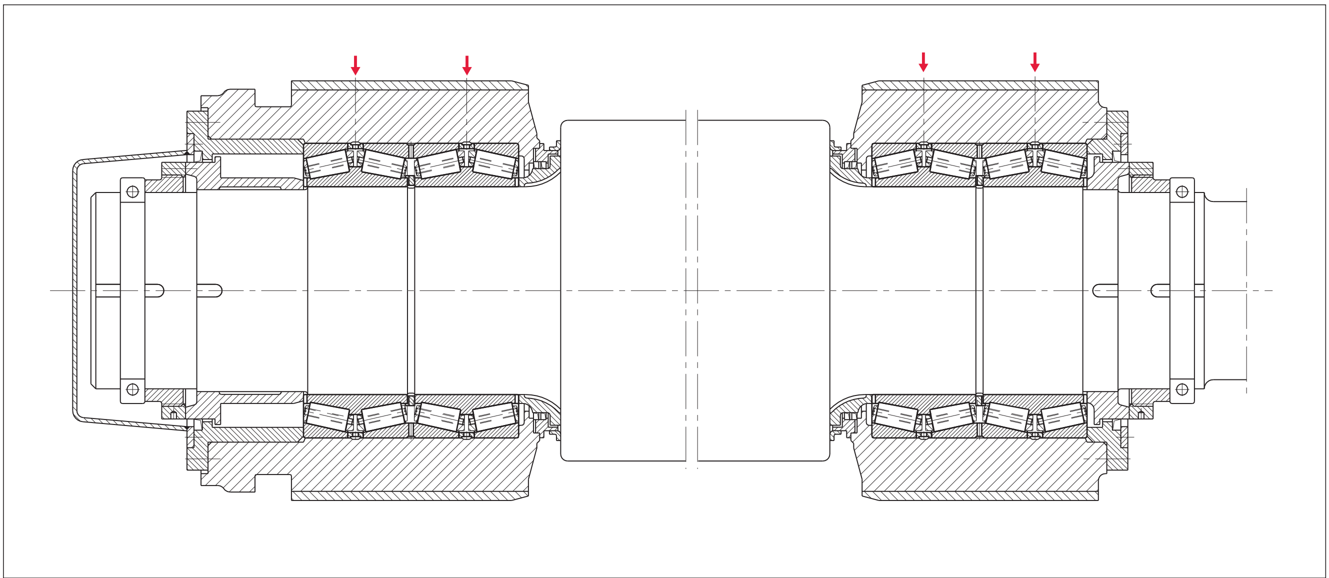
In four-row roller bearings supporting horizontally arranged shafts, the lubricant should be fed at two points, Figure 33. Ball bearings fitted as axial bearings (Figure 33 a, right) can be included in the lubrication of the radial bearings or lubricated separately. In contrast, axial tapered roller bearings (Figure 33 a, left) must always have a separate lubricant feed due to their higher demands in terms of lubrication. Double row angular contact ball bearings should also be lubricated separately where possible.
If the chocks are not removed (loose fit of inner rings) during stock regrinding of the roll body, relubrication must be provided through the roll neck. During assembly, sealed multi-row tapered roller bearings are packed with the optimum grease for the application. If the correct quantity of grease is used and distributed in the bearing, very long life values can be achieved.
We recommend that exit holes should be provided on both sides of the bearing, in order that the bearing seals should be exposed as little as possible to fluids.

Lubrication

Design of lubrication system



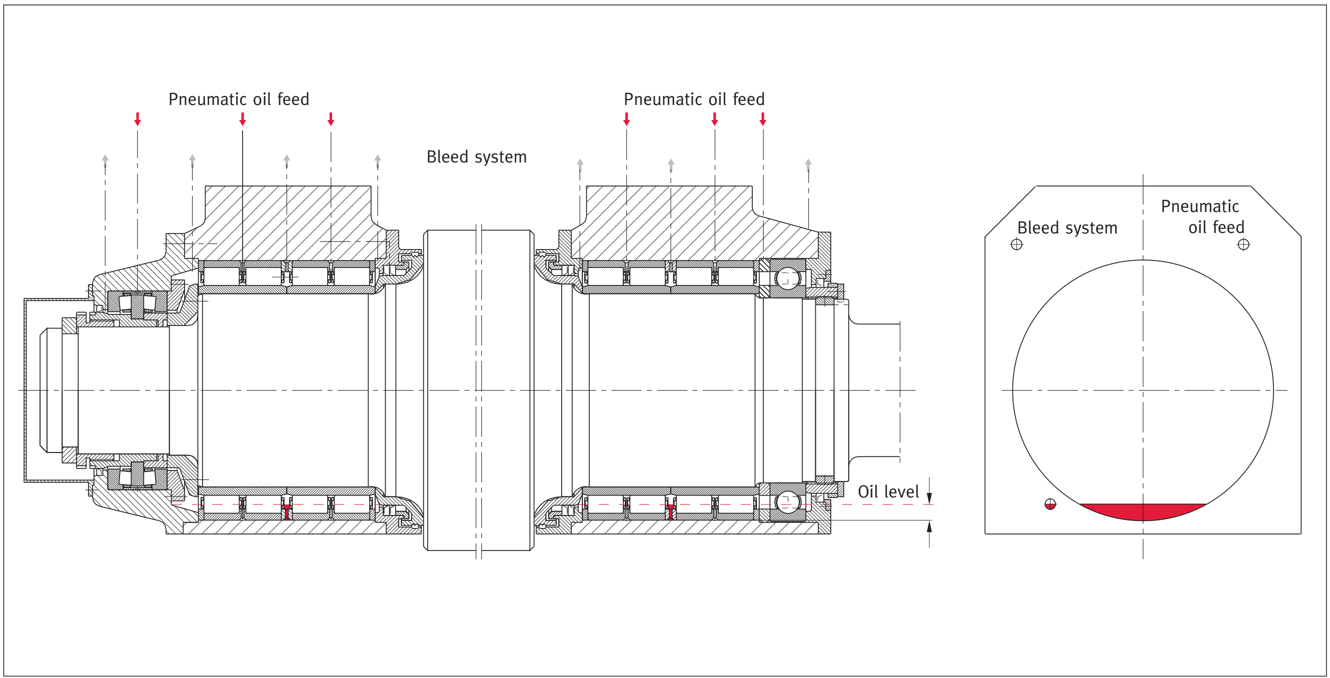
a) Roll bearing arrangement with four-row cylindrical roller bearings and axial bearings



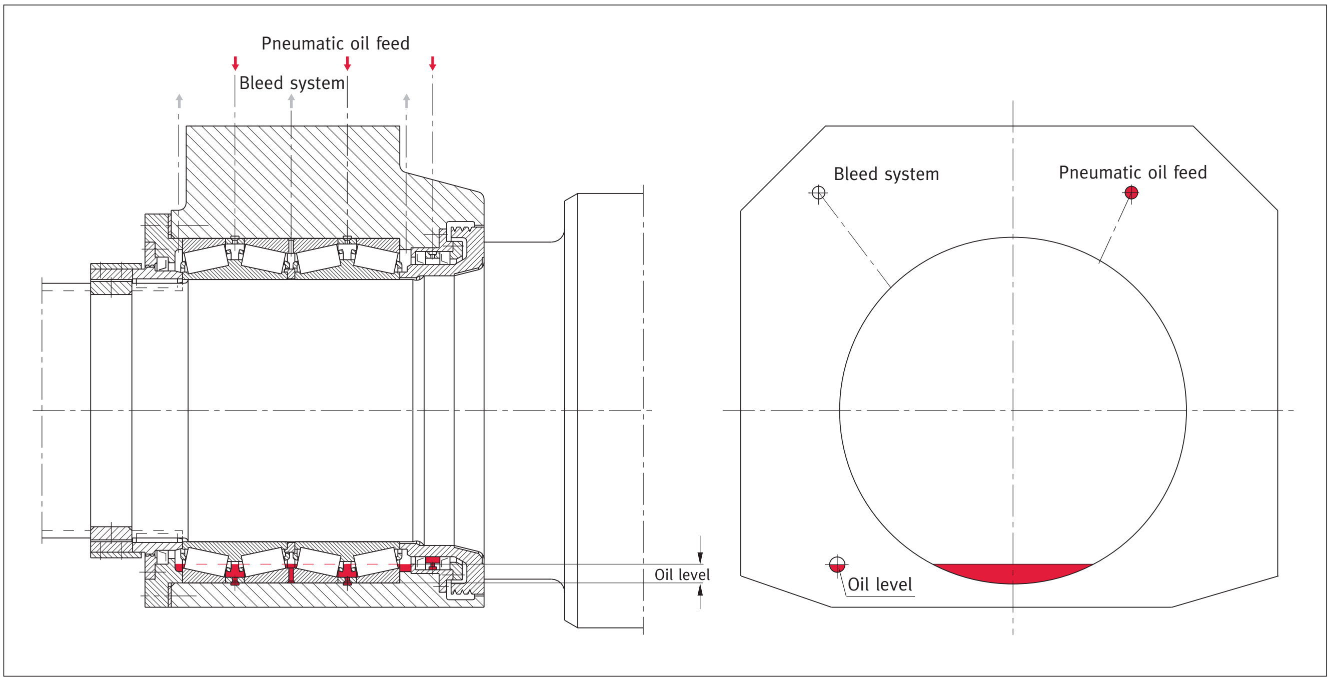
b) Roll bearing arrangement with four-row tapered roller bearings
33: Lubricant feed for four-row roller bearings (grease lubrication)

Design of lubrication system

Pneumatic oil lubrication



34: Feed of pneumatic oil supply for chocks with four-row cylindrical roller bearings and axial bearings

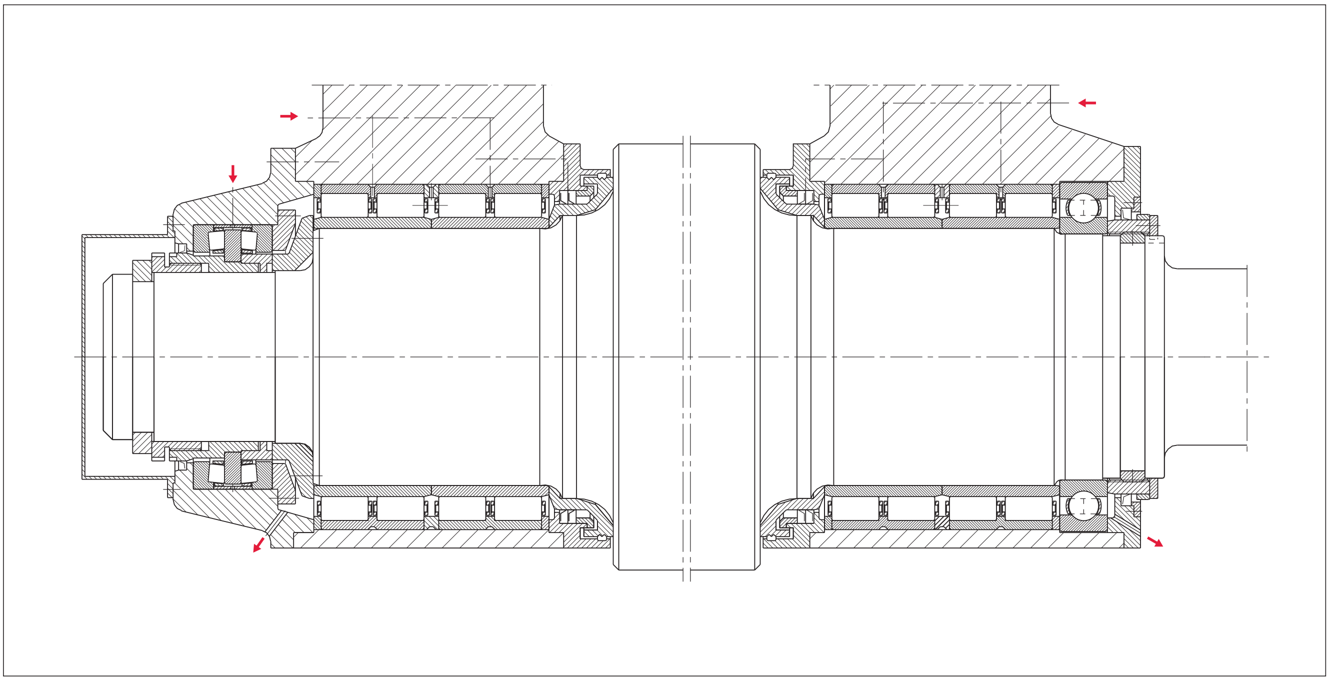


35: Feed of pneumatic oil supply for one chock with a four-row tapered roller bearing

Lubrication

Design of lubrication system

Recirculating oil lubrication



36: Oil feed and oil outlet in recirculating lubrication

Tolerances of roll neck bearings

Tolerances of roll neck bearings

37: Tolerances of radial and axial bearings with metric dimensions (normal tolerance)

Nominal dimension mm	Tolerance values µm Inner ring Δ_{dmp}		Outer ring Δ_{Dmp} Radial bearing		Axial bearing		Inner ring and outer ring $\Delta_{Bs} = \Delta_{Cs}$	
over 50 incl. 80	0	−15	0	−13	0	−19	0	−150
over 80 incl. 120	0	−20	0	−15	0	−22	0	−200
over 120 incl. 150	0	−25	0	−18	0	−25	0	−250
over 150 incl. 180	0	−25	0	−25	0	−25	0	−250
over 180 incl. 250	0	−30	0	−30	0	−30	0	−300
over 250 incl. 315	0	−35	0	−35	0	−35	0	−350
over 315 incl. 400	0	−40	0	−40	0	−40	0	−400
over 400 incl. 500	0	−45	0	−45	0	−45	0	−450
over 500 incl. 630	0	−50	0	−50	0	−50	0	−500
over 630 incl. 800	0	−75	0	−75	0	−75	0	−750
over 800 incl. 1 000	0	−100	0	−100	0	−100	0	−1 000
over 1 000 incl. 1 250	0	−125	0	−125	0	−125	0	−1 250
over 1 250 incl. 1 600	0	−160	0	−160	0	−160	0	−1 600
over 1 600 incl. 2 000	0	−200	0	−200	0	−200	0	−2 000

38: Tolerances of four-row tapered roller bearings with inch dimensions (normal tolerance)

Nominal dimension mm	Tolerance values µm Inner ring Δ_{dmp}		Outer ring Δ_{Dmp}		Inner ring and outer ring $\Delta_{Bs} = \Delta_{Cs}$
over 76,2 incl. 304,8	0	+25	0	+25	±1 524
over 304,8 incl. 609,6	0	+51	0	+51	±1 524
over 609,6 incl. 914,4	0	+76	0	+76	±1 524
over 914,4 incl. 1 219,2	0	+102	0	+102	±1 524
over 1 219,2	0	+127	0	+127	±1 524

Tolerances of roll neck bearings · Adjacent parts

Guidelines for fits

Bore diameter

Bore diameter
d Nominal bore diameter
d_s Single bore diameter

$$d_{mp} = \frac{d_{psmax} + d_{psmin}}{2}$$

Mean bore diameter in a radial plane
d_{psmax} Largest bore diameter in a radial plane
d_{psmin} Smallest bore diameter in a radial plane
$\Delta_{dmp} = d_{mp} - d$
Deviation of mean bore diameter from nominal dimension

Outside diameter

D Nominal outside diameter
D_s Single outside diameter

$$D_{mp} = \frac{D_{psmax} + D_{psmin}}{2}$$

Mean outside diameter in a radial plane
D_{psmax} Largest outside diameter in a radial plane
D_{psmin} Smallest outside diameter in a radial plane
$\Delta_{Dmp} = D_{mp} - D$
Deviation of mean outside diameter from nominal dimension

Width

B_s, C_s Width of inner ring and outer ring measured at one point
$\Delta_{Bs} = B_s - B$, $\Delta_{Cs} = C_s - C$
Deviation of single inner ring width and outer ring width from nominal dimension

Tolerance symbols
DIN ISO 1132, DIN 620

Guidelines for fits

Radial bearings

The inner rings of the radial bearings are subjected to circumferential load during operation. The inner rings should, where possible, have a tight fit on the roll neck.
In the case of four-row tapered roller bearings with a cylindrical bore, this requirement cannot be fulfilled for reasons of mounting, so a loose fit must be provided.
The inner rings of spherical roller bearings and cylindrical roller bearings also have a loose fit if the rolling speed is low and the bearing must be removed easily and quickly from the neck.
The outer rings of radial bearings have a loose fit in the chock, since they are subjected to point load. In an axial direction, the outer rings are located on the end face by the housing cover.

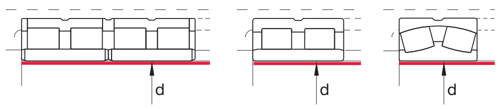
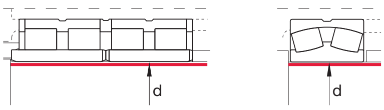
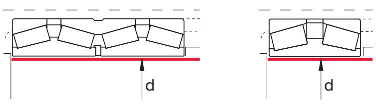
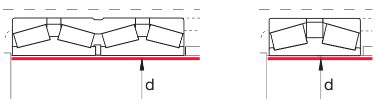
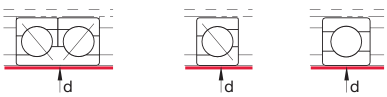
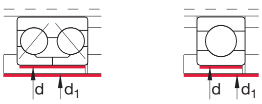
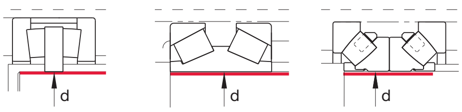
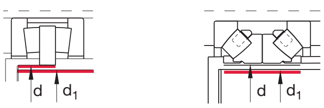
Axial bearings

The bearings used for axial guidance of the roll and guidance of the chocks are only subjected to axial load, so the inner rings can have a loose fit on the roll necks.
In some roll neck bearing arrangements, the bearings are placed on a sleeve for easier mounting.
A tight fit is advisable in this case.
The housing washers of axial tapered roller bearings are fitted loosely in the chocks. The outer rings of all other bearings used to provide axial guidance must be capable of radial displacement.
As a result, the housing bore must be significantly larger than the outside diameter of the outer rings.

Adjacent parts

Guidelines for fits

39: Tolerance zones for roll necks and sleeves (for bearing tolerances see page 28)

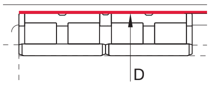
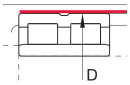
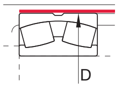
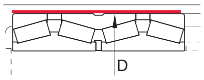
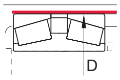
		Nominal dimension mm	Tolerance[1)] mm
	Cylindrical roller bearings and spherical roller bearings with tight fit	d < 170	p6
		d = 170...210	r6
		d > 210...225	+0,100...+0,130
		> 225...250	+0,110...+0,140
		> 250...280	+0,125...+0,160
		> 280...315	+0,140...+0,170
		> 315...355	+0,155...+0,190
		> 355...400	+0,170...+0,210
		> 400...450	+0,195...+0,230
		> 450	s6
	Cylindrical roller bearings and spherical roller bearings with loose fit	d	e7
	Tapered roller bearings, metric tolerances, with loose fit	d < 315	−0,180...−0,230
		d = 315...630	−0,240...−0,300
		> 630...800	−0,325...−0,410
		> 800	−0,350...−0,450
	Tapered roller bearings, inch tolerances, with loose fit	d = 101,6...127,0	−0,100...−0,125
		> 127,0...152,4	−0,130...−0,155
		> 152,4...203,2	−0,150...−0,175
		> 203,2...304,8	−0,180...−0,205
		> 304,8...609,6	−0,200...−0,249
		> 609,6...914,4	−0,250...−0,334
		> 914,4	−0,300...−0,400
	Angular contact ball bearings and deep groove ball bearings mounted on roll neck	d	e7
	Angular contact ball bearings and deep groove ball bearings mounted on sleeve	d	k6
		d_1	e9/H7
	Axial tapered roller bearings, double row tapered roller bearings (axial bearings), axial spherical roller bearings mounted on roll neck	d	e7
	Axial tapered roller bearings, axial spherical roller bearings mounted on sleeve	d	k6
		d_1	e9/H7

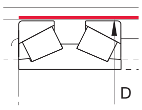
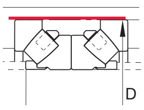
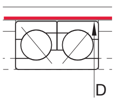
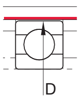
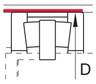
1) In the case of high speeds and bearings with a tapered bore, please contact us to discuss the tolerances for the adjacent parts.

Adjacent parts

Guidelines for fits

40: Tolerance zones for chocks

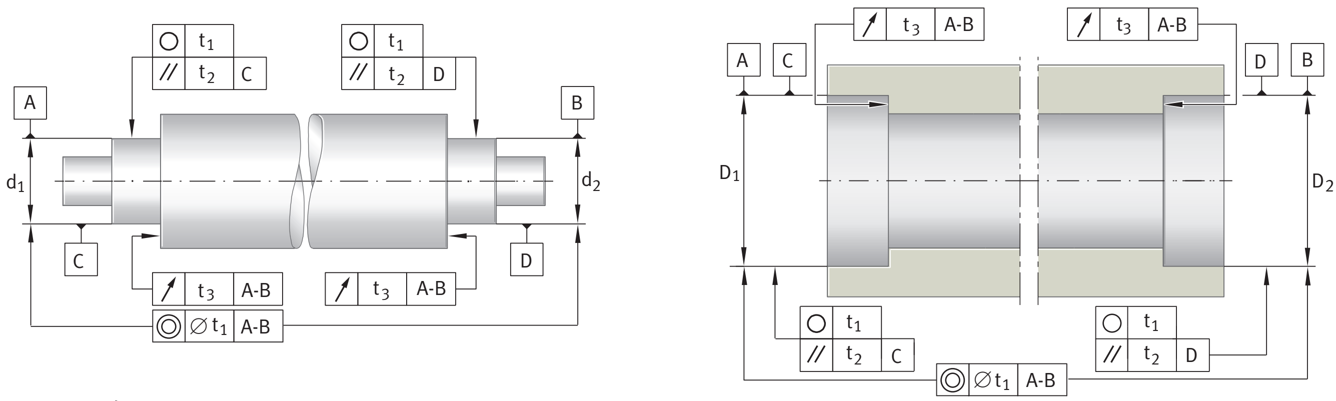
Radial bearings		Nominal dimension mm	Tolerance[1)] mm
  	Cylindrical roller bearings, spherical roller bearings and tapered roller bearings with metric tolerances	D ≤ 800	G6
		D > 800	G7
 	Tapered roller bearings with inch tolerance	D ≤ 304,8	+0,055...+0,080
		> 304,8...609,6	+0,101...+0,150
		> 609,6...914,4	+0,156...+0,230
		> 914,4...1219,2	+0,202...+0,300
		> 1219,2	+0,257...+0,380

Axial bearings		Nominal dimension mm	Tolerance[2)] mm
   	Tapered roller bearings, double row (axial bearings), axial spherical roller bearings, angular contact ball bearings and deep groove ball bearings	D ≤ 500	+0,6...+0,8
		> 500...800	+0,8...+1,1
		> 800	+1,2...+1,5
	Axial tapered roller bearings	D ≤ 800	G6
		D > 800	G7

1) For bearings with a tapered bore, please contact us to discuss the tolerances for the adjacent parts.
2) For high axial forces, please contact us to discuss the tolerances for the adjacent parts.

Machining tolerances for cylindrical bearing seats (DIN ISO 1101 and ISO 286)

41: Guide values for machining of shafts and housing bores, adjacent parts (sleeves, covers etc.)



t_1 = roundness
t_2 = parallelism
t_3 = axial runout of abutment shoulders

Bearing tolerance class	Bearing seating surface	Diameter tolerance	Roundness tolerance t_1	Parallelism tolerance t_2	Axial runout tolerance of abutment shoulder t_3
PN P6X	Shaft	IT6 (IT5)	Circumferential load IT4/2	IT4	IT4
			Point load IT5/2	IT5	
	Housing	IT7 (IT6)	Circumferential load IT5/2	IT5	IT5
			Point load IT6/2	IT6	
P5	Shaft	IT5	Circumferential load IT2/2	IT2	IT2
			Point load IT3/2	IT3	
	Housing	IT6	Circumferential load IT3/2	IT3	IT3
			Point load IT4/2	IT4	

IT grades in accordance with DIN ISO 286-1: 1988

Adjacent parts

Roughness of bearing seats

42: Guide values for surface roughness of rolling bearing seats (Roughness values apply to ground surfaces only)

Bearing tolerance class	Roughness		Nominal shaft diameter mm					Nominal housing bore diameter mm				
		over		50	120	250	500		50	120	250	500
		incl.	50	120	250	500		50	120	250	500	
			Roughness values µm									
Normal[1]	Roughness class		N5	N6	N7	N7	N7	N6	N7	N7	N8	N8
	Mean roughness value R_a CLA, AA[2]		0,4	0,8	1,6	1,6	1,6	0,8	1,6	1,6	3,2	3,2
	Roughness depth $R_t \approx R_z$		2,5	4	6,3	6,3	6,3	4; 6,3[*]	6,3; 8[*]	6,3; 10[*]	10; 16[*]	10; 16[*]
P6	Roughness class		N4	N5	N5	N6	N6	N5	N5	N6	N7	N7
	Mean roughness value R_a CLA, AA[2]		0,2	0,4	0,4	0,8	0,8	0,4	0,4	0,8	1,6	1,6
	Roughness depth $R_t \approx R_z$		1,6	2,5	2,5	6,3	6,3	2,5	2,5	6,3	6,3	6,3

[*] Roughness depths for flake graphite cast iron housings with turned fit surfaces.
[1] For more stringent requirements on running accuracy, the next highest surface quality must be selected.
[2] GBR: CLA (Centre Line Average Value); USA: (Arithmetic Average)

43: Roughness classes in accordance with DIN ISO 1302

Roughness class		N1	N2	N3	N4	N5	N6	N7	N8	N9	N10	N11	N12
Mean roughness value R_a	in µm	0,025	0,05	0,1	0,2	0,4	0,8	1,6	3,2	6,3	12,5	25	50
	in µinch	1	2	4	8	16	32	63	125	250	500	1000	2000

44: Permissible deviation of taper angle

Bearing width B	Dimensions mm > 16...25		> 25...40		> 40...63		> 63...100		> 100...160		> 160...250		> 250...400		> 400...630	
	Tolerances µm															
Taper angle tolerance AT_D to AT7 (DIN 7178) $(2 \cdot t_6)$	+8 0	+12,5 0	+10 0	+16 0	+12,5 0	+20 0	+16 0	+25 0	+20 0	+32 0	+25 0	+40 0	+32 0	+50 0	+40 0	+63 0

The taper angle tolerance AT_D applies vertical to the axis and is defined as the differential diameter.

Adjacent parts

Tolerances for roll necks and chocks

45: Tolerances for roll necks and chocks

	Nominal shaft diameter mm												
over	50	65	80	100	120	140	160	180	200	225	250	280	315
incl.	65	80	100	120	140	160	180	200	225	250	280	315	355
	Tolerances for roll necks µm												
e7	−60	−60	−72	−72	−85	−85	−85	−100	−100	−100	−110	−110	−125
	−90	−90	−107	−107	−125	−125	−125	−146	−146	−146	−162	−162	−182
e9	−60	−60	−72	−72	−85	−85	−85	−100	−100	−100	−110	−110	−125
	−134	−134	−159	−159	−185	−185	−185	−215	−215	−215	−240	−240	−265
f6	−30	−30	−36	−36	−43	−43	−43	−50	−50	−50	−56	−56	−62
	−49	−49	−58	−58	−68	−68	−68	−79	−79	−79	−88	−88	−98
g6	−10	−10	−12	−12	−14	−14	−14	−15	−15	−15	−17	−17	−18
	−29	−29	−34	−34	−39	−39	−39	−44	−44	−44	−49	−49	−54
k6	+21	+21	+25	+25	+25	+28	+28	+33	+33	+33	+36	+36	+40
	+2	+2	+3	+3	+3	+3	+3	+4	+4	+4	+4	+4	+4
n6	+39	+39	+45	+45	+52	+52	+52	+60	+60	+60	+66	+66	+73
	+20	+20	+23	+23	+27	+27	+27	+31	+31	+31	+34	+34	+37
p6	+51	+51	+59	+59	+68	+68	+68	+79	+79	+79	+88	+88	+98
	+32	+32	+37	+37	+43	+43	+43	+50	+50	+50	+56	+56	+62
r6	+60	+62	+73	+76	+88	+90	+93	+106	+109	+113	+126	+130	+144
	+41	+43	+51	+54	+63	+65	+68	+77	+80	+84	+94	+98	+108
s6	+72	+78	+93	+101	+117	+125	+133	+151	+159	+169	+190	+202	+226
	+53	+59	+71	+79	+92	+100	+108	+122	+130	+140	+158	+170	+190

	Nominal chock bore diameter mm												
over	80	100	120	140	160	180	200	225	250	280	315	355	400
incl.	100	120	140	160	180	200	225	250	280	315	355	400	450
	Chock bore tolerances µm												
G6	+34	+34	+39	+39	+39	+44	+44	+44	+49	+49	+54	+54	+60
	+12	+12	+14	+14	+14	+15	+15	+15	+17	+17	+18	+18	+20
G7	+47	+47	+54	+54	+54	+61	+61	+61	+69	+69	+75	+75	+83
	+12	+12	+14	+14	+14	+15	+15	+15	+17	+17	+18	+18	+20
H6	0	0	0	0	0	0	0	0	0	0	0	0	0
	+22	+22	+25	+25	+25	+29	+29	+29	+32	+32	+36	+36	+40
H7	0	0	0	0	0	0	0	0	0	0	0	0	0
	+35	+35	+40	+40	+40	+46	+46	+46	+52	+52	+57	+57	+63

Adjacent parts

Tolerances for roll necks and chocks

45 cont. Tolerances for roll necks and chocks

Nominal shaft diameter
mm

over	355	400	450	500	560	630	710	800	900	1000	1120	1250	1400
incl.	400	450	500	560	630	710	800	900	1000	1120	1250	1400	1600
Tolerances for roll necks µm													
e7	−125	−135	−135	−145	−145	−160	−160	−170	−170	−195	−195	−220	−220
	−182	−198	−198	−215	−215	−240	−240	−260	−260	−300	−300	−345	−345
e9	−125	−135	−135	−145	−145	−160	−160	−170	−170	−195	−195	−220	−220
	−265	−290	−290	−320	−320	−360	−360	−400	−400	−455	−455	−530	−530
f6	−62	−68	−68	−76	−76	−80	−80	−86	−86	−98	−98	−110	−110
	−98	−108	−108	−120	−120	−130	−130	−142	−142	−164	−164	−188	−188
g6	−18	−20	−20	−22	−22	−24	−24	−26	−26	−28	−28	−30	−30
	−54	−60	−60	−66	−66	−74	−74	−82	−82	−94	−94	−108	−108
k6	+40	+45	+45	+44	+44	+50	+50	+56	+56	+66	+66	+78	+78
	+4	+5	+5	0	0	0	0	0	0	0	0	0	0
n6	+73	+80	+80	+88	+88	+100	+100	+112	+112	+132	+132	+156	+156
	+37	+40	+40	+44	+44	+50	+50	+56	+56	+66	+66	+78	+78
p6	+98	+108	+108	+122	+122	+138	+138	+156	+156	+186	+186	+218	+218
	+62	+68	+68	+78	+78	+88	+88	+100	+100	+120	+120	+140	+140
r6	+150	+166	+172	+184	+199	+225	+235	+266	+276	+316	+326	+378	+378
	+114	+126	+132	+150	+155	+175	+185	+210	+220	+250	+260	+300	+300
s6	+244	+272	+292	+324	+354	+390	+430	+486	+526	+586	+646	+718	+798
	+208	+232	+252	+280	+310	+340	+380	+430	+470	+520	+580	+640	+720

Nominal chock bore diameter
mm

over	450	500	560	630	710	800	900	1000	1120	1250	1400	1600	1800
incl.	500	560	630	710	800	900	1000	1120	1250	1400	1600	1800	2000
Chock bore tolerances µm													
G6	+60	+66	+66	+74	+74	+82	+82	+94	+94	+108	+108	+124	+124
	+20	+22	+22	+24	+24	+26	+26	+28	+28	+30	+30	+32	+32
G7	+83	+92	+92	+104	+104	+116	+116	+133	+133	+155	+155	+182	+182
	+20	+22	+22	+24	+24	+26	+26	+28	+28	+30	+30	+32	+32
H6	0	0	0	0	0	0	0	0	0	0	0	0	0
	+40	+44	+44	+50	+50	+56	+56	+66	+66	+78	+78	+92	+92
H7	0	0	0	0	0	0	0	0	0	0	0	0	0
	+63	+70	+70	+80	+80	+90	+90	+105	+105	+125	+125	+150	+150

Conditions for inner rings with loose fits

A loose fit of the inner rings requires a minimum roll neck hardness in order to restrict wear of the roll neck. Wear of the roll neck is also considerably affected by the lubrication between the inner ring bore and the roll neck surface. If adequate lubrication of the roll neck is ensured over the whole operating period, a roll neck hardness of 35 to 40 Shore C is sufficient. For example, if the chocks are not removed as usual for grinding of the rolls, the fit gap between the inner rings and the roll neck is not always repeatedly supplied with fresh grease. In such cases, this can be remedied by special roll neck lubrication, Figure 46. Schaeffler urgently recommends this roll neck lubrication in the case of sealed four-row tapered roller bearings if the bearing arrangements will remain on the roll neck for a long period.

In order to minimise wear of the adjacent parts, these should be designed with a minimum hardness of 60 Shore C. In order to give better lubricant supply to the lateral faces, lubrication grooves should be provided on the end faces of the adjacent parts or the inner rings. The lateral faces are lubricated via these grooves and the fit joint between the inner ring and roll neck is supplied with lubricant.

Chocks

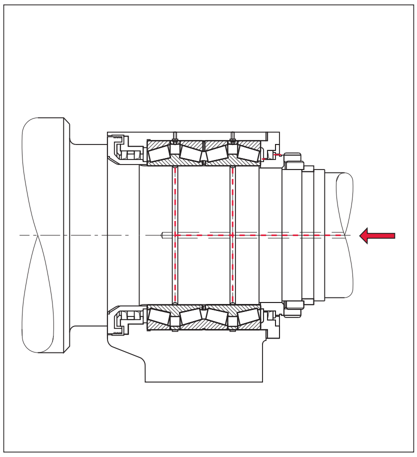
The rings of roll neck bearings are thin-walled in almost all cases. They must therefore be well supported; otherwise they cannot support the high forces occurring in operation. Good support of the bearing outer rings requires sufficiently rigid design of the chocks. Where chocks are made from cast steel with a minimum tensile strength of 450 N/mm^2, adequate rigidity is generally achieved if the design is carried out in accordance with the following equations:

$$h_A = (1{,}5 \ldots 2{,}0)\,\frac{D-d}{2}$$

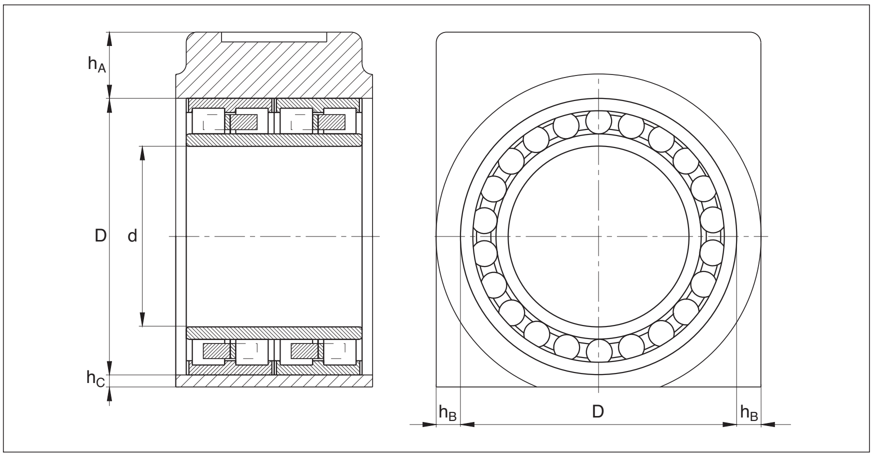
$$h_B = (0{,}7 \ldots 1{,}2)\,\frac{D-d}{2}$$

$$h_C = (0{,}15 \ldots 0{,}25)\,\frac{D-d}{2}$$

In this case, h_A is the upper, h_B the lateral and h_C the lower wall thickness of the chock in mm, d the bearing bore in mm and D the bearing outside diameter in mm (Figure 47). If the chocks correspond to these empirical equations, the influence of chock deformation on the stress exerted on the bearings, assuming the load is not too high, will generally remain within acceptable limits. Where extreme loads are present or new designs are in progress, however, it is recommended that the deformation of the chock and the effect on the bearing should be checked by means of calculation, see Figure 48 on the following page.



46: Bearing arrangement with lubrication holes in the roll neck



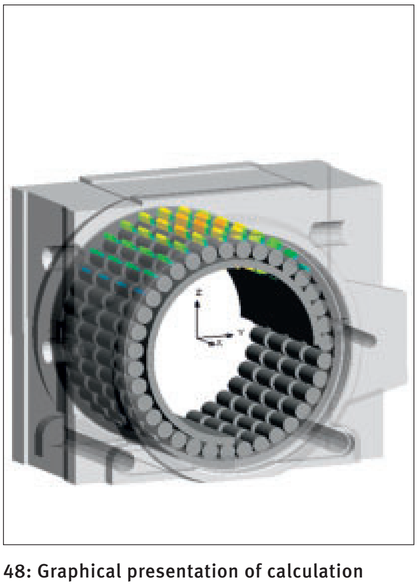
47: Wall thicknesses of a chock

Contact surfaces
for stand windows and chocks

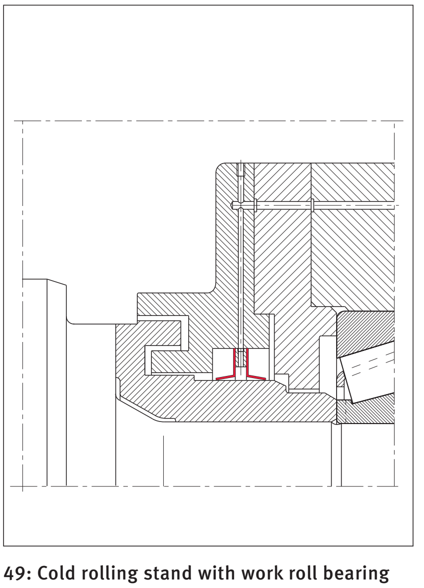
The surfaces on which the chocks are supported in the stand and on the screw-down mechanism must be crowned; this allows the chock bore to align itself parallel to the roll neck and the bearing can then support roll deflections over its entire width even if not precisely adjusted. The support surfaces should be hardened so that they do not undergo flattening at high loads. If multi-row bearings are used, the rolling mill designer must ensure that the screw-down mechanisms are positioned over the centre of the radial bearings, otherwise the rows of rollers will be subjected to uneven loads.

Design of seals

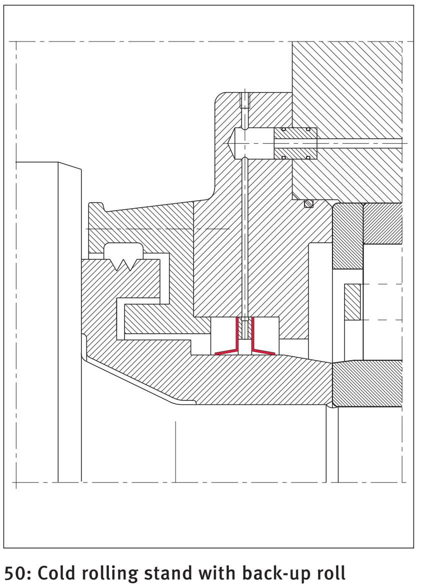
The seals should prevent the ingress of water, coolant liquids, rolling scale and other contaminants and should also retain the lubricant in the bearing. The type of seal to be considered in the individual case will depend on the rolling speed, the sealing action required, the type of lubrication and the operating temperature. Figures 49 and 50 show an example of a face seal for back-up rolls and work rolls in a cold rolling mill. For rolling on cold rolling mills, contamination of the rolled stock surface by the lubricant in the bearing arrangement must be prevented. The inner rotary shaft seal is therefore fitted so that the lip is directed towards the bearing. In all bearing arrangements sealed by means of rotary shaft seals, the surfaces on which the lip will slide must be machined to very high precision. The sliding surfaces must be bevelled so that the seal is not damaged during mounting. The seal lip must be lubricated regularly. Whenever a roll is replaced, the condition of the seal must be checked and the seals replaced if necessary.



48: Graphical presentation of calculation results



49: Cold rolling stand with work roll bearing arrangement



50: Cold rolling stand with back-up roll bearing arrangement

General guidelines for mounting and dismounting can be found in the Mounting Handbook MH1.
In addition, some important work processes in the operation of rolling mills are covered in further detail on the following pages.

Preparations for mounting

Before mounting of the bearings is started, the mating parts and adjacent parts, in other words roll necks, chocks, sleeves, covers etc. must be checked for dimensional accuracy and geometrical accuracy with reference to the design drawing.
The specified surface quality of the roll seats, chocks and lateral contact parts must be inspected.
All burrs and sharp edges resulting from machining must be broken or rounded.

Inspection of cylindrical roll necks

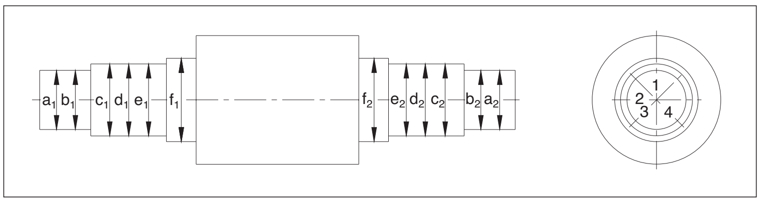
For acceptable dimensional and geometrical inspection, the roll necks must be measured at the seats for the radial bearings in three cross-sections (c-d-e) and at the seats for the axial bearings in two cross-sections (a-b).
The values for each of four diameters (1-2-3-4) should be determined, Figure 51. The measured values are documented in a measurement record.

Inspection of chocks

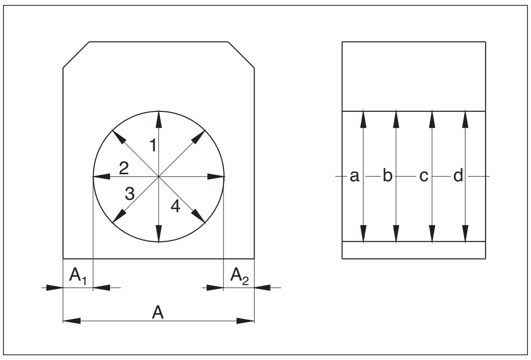
The chock bore should be checked in four cross-sections (a-b-c-d) at each of four diameters (1-2-3-4), Figure 52. Furthermore, the position of the chock bore relative to the chock face (A_1 and A_2) must be checked, if necessary with the locking ledges screwed into place (positional and geometrical tolerances, see Table 41, page 32).
The deviations from the nominal value should, as in roll neck inspection, be documented in a measurement record.
The adjacent parts must also be inspected; the important dimensions here are all those that result in the axial preload. It must also be checked whether the adjacent parts are free from impacts.
The lubrication holes must be cleaned. Air must then be blown through the holes for inspection.

Surface roughness

In order that a sufficient proportion of load-bearing surface is achieved at high bearing loads, the surface roughness of the rolling bearing seats must not exceed the guide values stated, see Table 42, page 33.



51: Measurement points for inspection of roll necks



52: Measurement points for inspection of chocks

Treatment of bearing seats

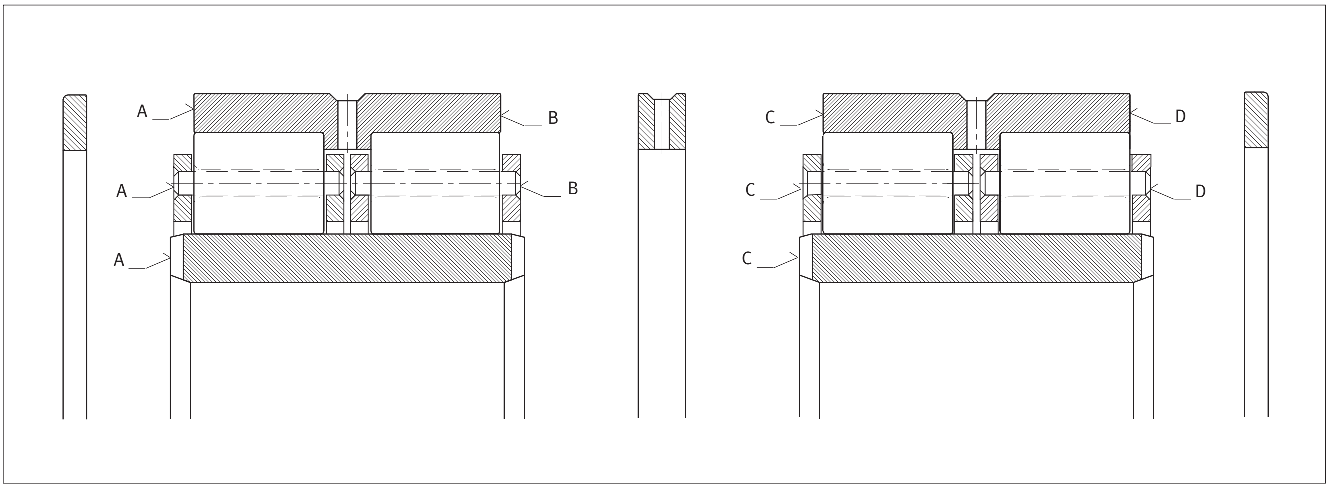
On all seats where rolling bearings have a sliding fit (chock) or tight fit (roll neck), the formation of fretting corrosion can be reduced if they are coated with a lubricant paste containing an anti-corrosion additive, for example with the FAG mounting paste Arcanol MOUNTING.PASTE. The seat areas must be thoroughly cleaned before the paste is applied. The paste should be applied in a thin layer such that the normally bright surface becomes matt.

Preparation of bearings for mounting

The bearings must not be removed from their original packaging until all preparations have been made to the chocks and rolls and the accessories are available. It is not normally necessary to remove the anti-corrosion oil.
This does not react with commonly used rolling bearing oils and greases.
The functional capability, load carrying capacity and operating life of a bearing is dependent not only on its quality but also on mounting. Mounting should therefore be entrusted to experienced fitters only. FAG fitters are available to carry out initial mounting, to instruct the fitters at the customer's plant and for all further eventualities. The following sections explain how mounting and dismounting is carried out on common roll bearing arrangements with four-row cylindrical roller bearings, four-row tapered roller bearings and spherical roller bearings.

Mounting of four-row cylindrical roller bearings

The four-row cylindrical roller bearings can be ordered either as a complete bearing or separately as an RZL part (outer ring + roller and cage assembly, e.g. Z-524678.RZL) and LZL part (inner ring, e.g. Z-524678.LZL). Each inner ring and outer ring is marked with the bearing designation (e. g. Z-524678.ZL) and the consecutive serial number (e. g. 11-585C) (Figure 57, page 41). At each mounting position, only parts of a bearing with the same serial number may be fitted. Furthermore, the sequence of the individual parts is identified in accordance with the schematic drawing (Figure 53) (e.g. A-B-C-D). The inner rings with one production number can, however, be allocated to outer rings with roller and cage assemblies having a different production number.



53: Marking and assembly of parts of four-row cylindrical roller bearings

First, the labyrinth ring or backing ring is, depending on the size of the interference, heated and shrink fitted on the roll neck. The ring must be axially clamped during cooling so that it abuts the roll body without a gap.

Mounting of inner rings

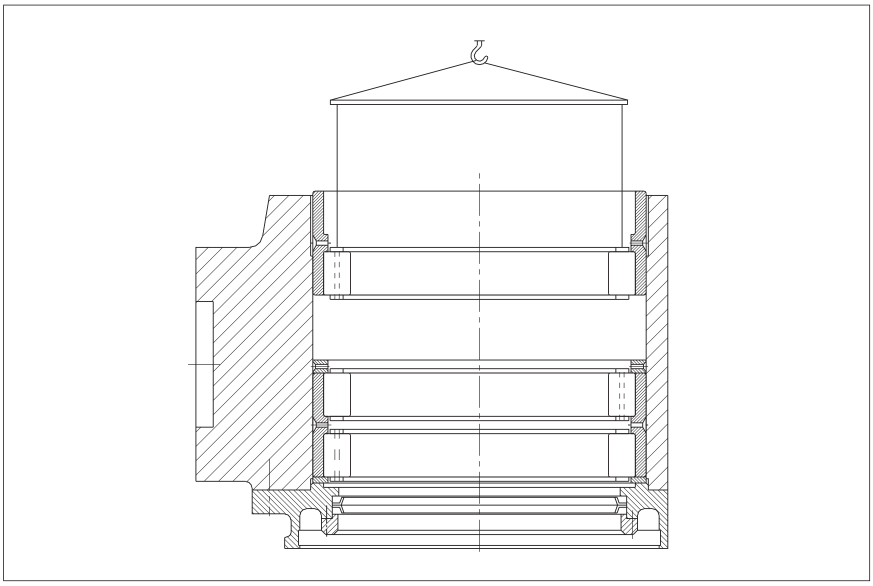
The inner rings of cylindrical roller bearings that are mounted with a tight fit on the roll neck must be heated to max. 120 °C before mounting. This is normally carried out by means of induction heating devices (Figure 54) or in an oil bath. This ensures uniform heating. In induction heating devices, a time or temperature control system prevents heating in excess of the heating temperature.
If heating is carried out in an oil bath, the heating temperature can be regulated using a thermostat. Inner rings of large dimensions and masses are often heated using a medium frequency induction heating device and a flexible inductor. After heating, smaller inner rings can be moved onto the roll neck by hand or by means of Bearing-Mate (Figure 55). In the mounting of larger bearing rings, lifting gear must be used.
After cooling, the rolling bearing rings should be fully abutted against the labyrinth ring. There must not be any gap between two bearing rings positioned adjacent to each other. For this reason, the rings must be axially clamped during cooling.

54: Example of an induction heating device (Heater600)

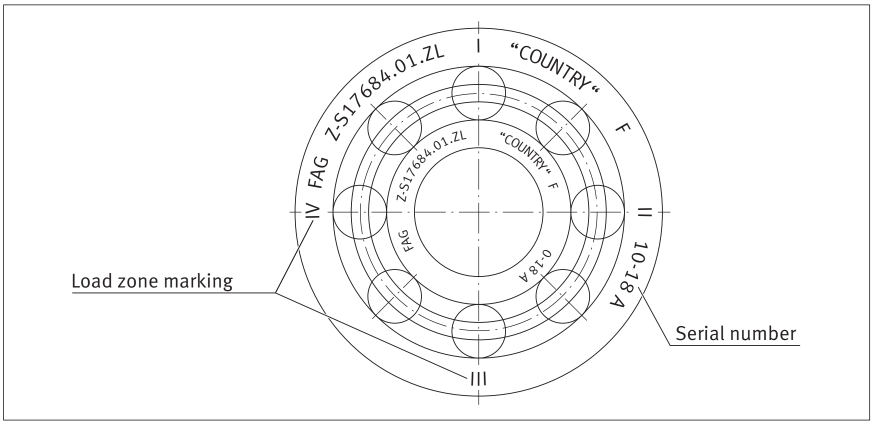

55: Mounting of the inner ring of a small cylindrical roller bearing using Bearing-Mate

Mounting of outer rings

The outer rings of cylindrical roller bearings have a sliding fit in the chocks. Smaller outer rings can be moved into the chock by hand. The outer rings or cages of larger bearings can be lifted into the chock, for example with the aid of a crane, Figure 56.
On the end faces of the outer rings, four zones with the numbers I, II, III and IV are marked, Figure 57. At first mounting, the outer rings are positioned such that the load acts on the load zone I. The load zones should be positioned in the same direction on all outer rings.
We recommend that the bearings should be thoroughly checked after a running time of 1 000 to 1 200 hours and the load zones of the outer rings changed. At the first load zone change, the outer rings in the chock must be rotated by 180° in load zone III and in all further changes in load zone II and IV respectively.



56: Mounting of a cylindrical roller outer ring using a crane



57: Marking of a four-row cylindrical roller bearing

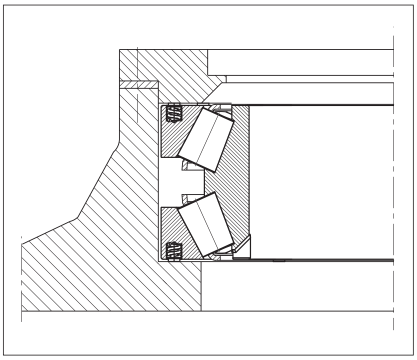
Mounting and maintenance

Mounting of axial bearings

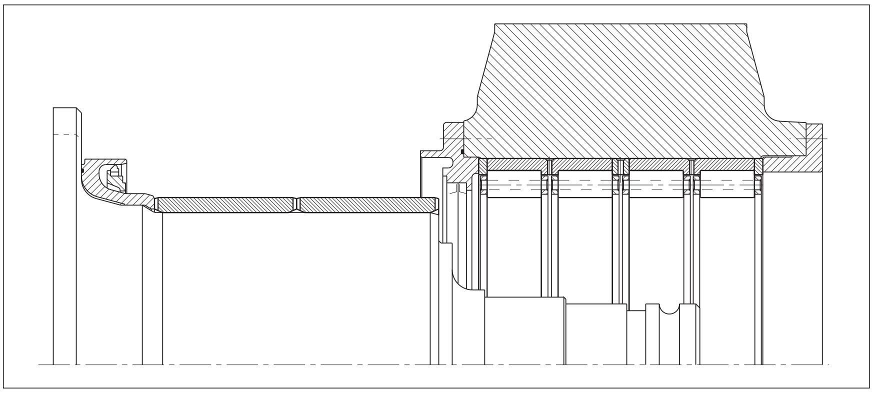
Axial bearings such as tapered roller bearings, angular contact ball bearings, deep groove ball bearings and spherical roller bearings must not be subjected to radial load. The bore of the chock is therefore 0,6 … 1,5 mm larger than the bearing outside diameter.
Double row tapered roller bearings with a large contact angle can be axially preloaded using springs, Figure 58. This prevents slippage of the row without load when an axial load acts in one direction.
In order to facilitate adjustment of the outer rings, there should be an axial clearance of 0,2 … 0,4 mm between the outer ring and the housing.
In the mounting of angular contact and deep groove ball bearings, the outer ring may sag due to the larger bore of the chock. As a result, there is a risk that only the upper balls will be subjected to axial load. In order to prevent this occurring, the cover screws are only lightly tightened after mounting of the axial bearing on the roll neck, so that the outer ring can still align itself.
The fitter moves the roll into its working position. The outer rings are then clamped in this position.

Mounting of preassembled chocks on roll necks

Once the labyrinth ring and the inner rings are shrink fitted on the roll neck, mounting of the preassembled chock can be started, Figure 59. The chocks with the outer rings must be carefully aligned so that they can be slid into place without constraint.
The further operations must be carried out with great care in order that the inner ring raceway is not damaged. It is advisable here to rotate the roll in order to make the sliding operation easier.
In the case of cylindrical roller bearings with a loose fit on the inner ring, the bore must be greased or oiled before sliding into place.



58: The outer rings of double row tapered roller bearings are adjusted by means of springs



59: The preassembled chock is slid onto the roll neck. The outer rib washer of the cylindrical roller bearing is secured by means of angle pieces (tabs)

Mounting and maintenance

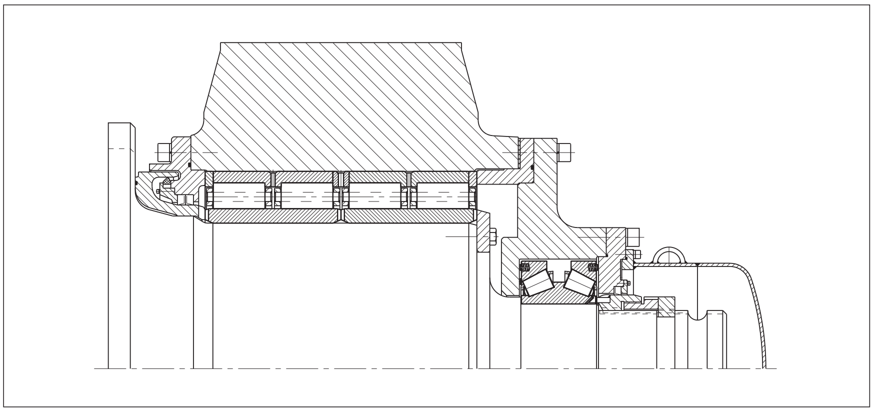
Mounting of four-row cylindrical roller bearings

Dismounting of bearing arrangement

After dismounting of the axial bearing retention on the roll, the chocks can be removed from the roll neck as a complete unit. For inspection of the roll neck bearings, the individual bearing parts are dismounted on the same basis as mounting but in the reverse sequence.
Withdrawal of the tightly fitted inner rings from the roll neck requires special equipment. The FAG medium frequency induction heating device has proved effective in this case, Figure 61.
In some cases, the inner rings are removed by hydraulic means. However difficulties can occur, mainly in the case of large bearings, if the fit surfaces are damaged by cold welding or frictional corrosion.

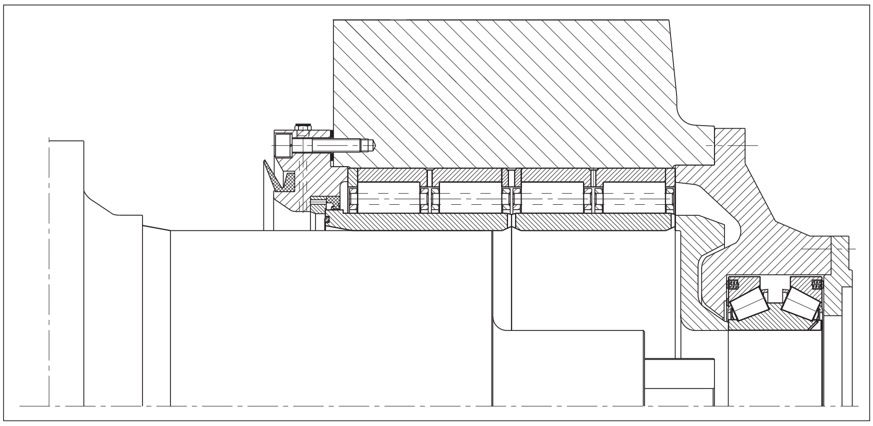
Loose fit of inner rings

In profile or fine-section mills, inner rings are sometimes positioned on the roll neck with a loose fit. When removing the bearing rings, the corresponding design of the labyrinth cover means that the labyrinth ring is removed as well and the inner rings are axially guided. The entire bearing arrangement remains together as a unit, Figure 62.



60: Fully assembled chock

61: Medium frequency technology with flexible inductor for thermal mounting and dismounting of cylindrical roller bearing inner rings



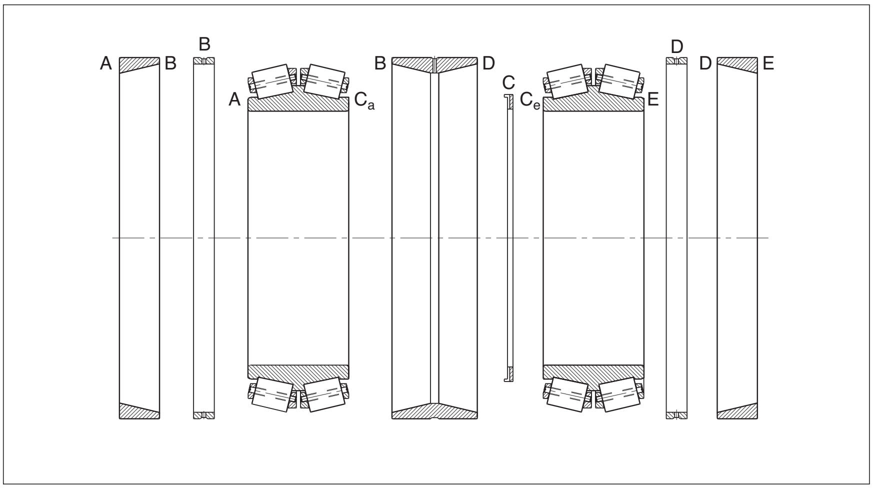
62: When the chock is replaced, the complete bearing arrangement is removed or slid into place respectively

Mounting of four-row tapered roller bearings

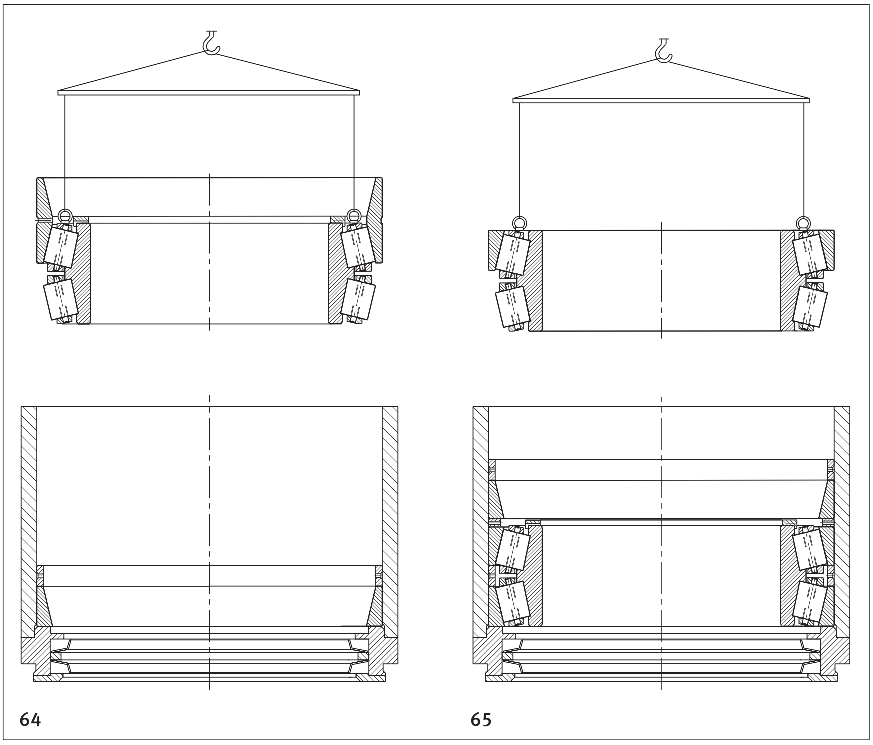
For information on mounting, see mounting manual WL 80 154.
In the case of four-row tapered roller bearings, the following markings are found: the bearing designation, the FAG company logo, the consecutive serial number and letters to indicate the assembly sequence. The marking of the bearing rings can be seen in Figure 63.
The intermediate rings B and D as well as C are matched when the bearing is supplied such that the correct axial internal clearance is achieved. The ring width and axial internal clearance are marked on the intermediate rings.
As in the case of the four-row cylindrical roller bearings, the circumference of the outer rings is divided into four load zones that are marked I, II, III and IV, see Figure 57, page 41.

Mounting

Four-row tapered roller bearings are mounted with a vertical axis, Figures 64 and 65. First, the narrow outer ring marked AB is fitted in the chock such that the load zone I lies in the direction of load.
The other bearing parts are then fitted in the sequence indicated in the schematic drawing (Figure 63).
Load zone I must also lie in the direction of load for the other outer rings.



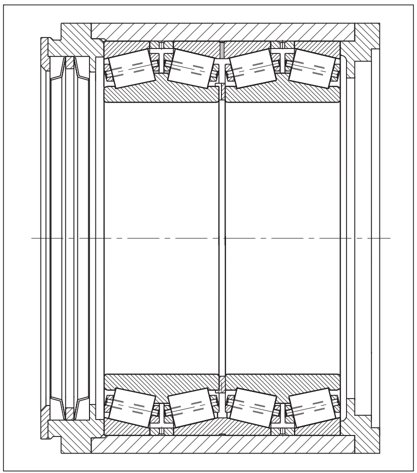
63: Sequence of assembly of bearing parts



64 and 65: The bearing parts are inserted in the chock.

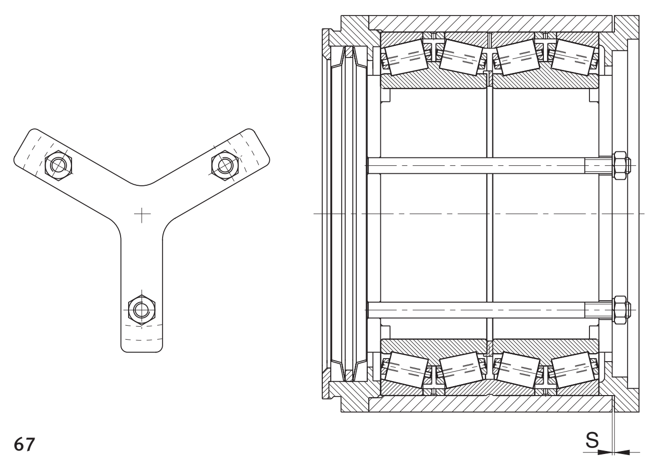
Mounting and maintenance

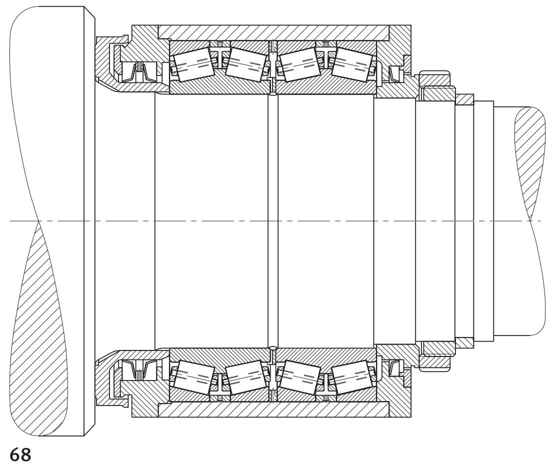
Mounting of four-row tapered roller bearings



66: The cover screws of the chock are first tightened lightly; the chock is then tilted

We recommend that the bearings should be thoroughly checked after a running time of 1 000 to 1 200 hours and the load zones of the outer rings changed. At the first load zone change, the outer rings in the chock must be rotated by 180° in load zone III and in all further changes in load zone II and IV respectively.
Once all parts of the bearing have been inserted, the cover screws that are initially without seals are lightly tightened, Figure 66.
The chock is turned over so that the bearing axis is horizontal.
Centring pieces are then attached to the outer end faces of the inner rings and are clamped by means of tie rods, Figure 67.
While the inner rings are continuously rotated, the nuts on the tie rods and the cover screws are tightened uniformly. A feeler gauge is used to check whether the inner rings and intermediate ring are fully abutted against each other.
The gap S between the chock and cover is then measured and a seal of width S-x is inserted.
The quantity x required for secure preload is dependent on the type of seal and is determined by the manufacturer of the rolling mill.
Once the outer rings are clamped by means of the cover, the centring pieces with the tie rods are removed. Experienced fitters dispense with the use of centring pieces and tie rods and rotate the inner rings during vertical insertion until the tapered rollers are fully abutted against the guidance ribs. Finally, the rotary shaft seals are positioned in the cover. The inner ring bores are greased or oiled. As soon as the labyrinth ring is shrink fitted, the chock is positioned on the roll neck.
The bearing is axially clamped by means of the shaft nut so that it is fully abutted on the labyrinth ring. While the nuts are being tightened, the chock should be rotated several times to left and right. The nut is then loosened again until there is a clearance of 0,2 to 0,6 mm between the inner ring and nut.
If the thread pitch is 3 mm, for example, the nut is loosened by 1/10 of a revolution.





67: The outer rings are clamped together while the inner rings are rotated
68: The mounted chock

Mounting and maintenance

Mounting of four-row tapered roller bearings

It is advisable not to grease the bearing arrangement until after mounting, otherwise the bearing may become contaminated. Greasing should ideally be carried out using a grease gun. If a grease gun is not available, the roller sets must be greased by hand before insertion in the chock.
In the case of rolls that are intended for very high rolling speeds, the chocks must not be packed completely with grease. Please consult us for information on the quantity required in the specific case.

Dismounting

If the chock is to be positioned on another roll when replacing rolls, it is simply necessary to unscrew the nut and the complete chock can be removed from the roll neck and slid onto the new roll. If the bearings must be dismounted in maintenance and inspection, this is carried out on the same basis as mounting but in the reverse sequence. Double row tapered roller bearings can be dismounted in the same way.

Maintenance

After a long running time, the axial internal clearance of four-row tapered roller bearings increases due to wear of the raceways.
It is therefore necessary to check the axial internal clearance from time to time.
If the axial internal clearance is too large, the outer and inner intermediate rings must be reground. The corrected axial internal clearance should be greater than the original axial internal clearance.
Further information can be found in WL 80 154.

Mounting of spherical roller bearings

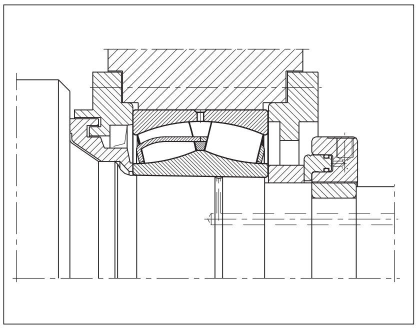
Spherical roller bearings with a loose fit and also with tight fit of the inner rings are fitted in rolling mills. Mounting is made easier if the inner rings have a loose fit. The bearings are first fitted in the chocks. The lateral covers are then screwed into place. Before the bearing arrangement is slid onto the roll, the bore of the inner rings should be greased. Sliding into place is made easier by means of a mounting sleeve.
Since the inner rings rotate on the roll, some clearance must be present between the lateral contact parts after mounting. Ideally, the clamping nut is first tightened and loosened again as in the case of tapered roller bearings. In this position, the nut is secured.
If a tight fit of the inner rings is necessary in the case of spherical roller bearings, bearings with a tapered bore are generally used. When changing rolls, the bearing can be transferred to the other roll if the tapered shaft necks and the width of the labyrinth rings are to a sufficiently tight tolerance.

Mounting of spherical roller bearings with tapered bore

The greased spherical roller bearing is positioned and clamped in the chock. The chock together with the spherical roller bearing is then slid onto the roll neck until it is fully seated on the neck.
Further pressing into place can be carried out using the hydraulic method. For this purpose, the roll necks must have oil grooves and oil feed ducts. It is advisable to use a hydraulic nut for pressing into place. Details on hydraulic pressing and the hydraulic nut can be found in the publications IS1 and TPI 196.
The spherical roller bearing is pressed on until it abuts the labyrinth ring, Figure 69. In order to maintain the specified drive-up distance, it must be ensured that the width of the labyrinth ring is matched to the actual diameter of the tapered roll seat.
The hydraulic nut is then removed. The roll nut is positioned, screwed into place and secured on the roll neck. Mounting is thus complete.

Dismounting of spherical roller bearings with tapered bore

The shaft nut is loosened by a few turns, at least by the drive-up distance of the bearing. If oil is then pressed between the fit surfaces, the bearing slips abruptly from the shaft seat as soon as a continuous film has been formed. Once the shaft nut has been unscrewed, the chock together with the spherical roller bearing can be lifted off and then mounted on another roll.



69: The spherical roller bearing is slid into place by a hydraulic nut using the hydraulic method

Mounting and maintenance

Stockholding · Statistical recording · Industrial Service

Stockholding

In order to avoid costly downtime, it is recommended that three and a half complete bearing sets should always be kept available for each stand. Of these, one bearing set is located on the roll fitted in the stand. A second set is among the rolls that are being reground.
A further set of rolls including the associated bearings should be ready for use at any time.
In order that these three sets can be completed in cases of bearing failure, a half bearing set should also be held in reserve. In the case of cylindrical roller bearings with a tight fit of the inner rings, it may be advisable to source additional inner rings, which are mounted on the roll necks. If the rolls must be replaced often, for example in profile rolling mills, this will save on frequent dismounting of the inner rings.

Statistical recording

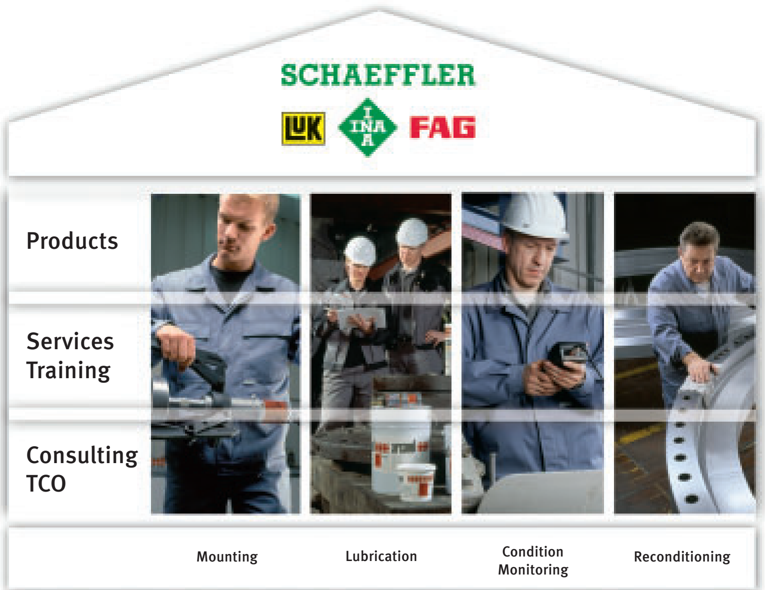
When the bearings are delivered, it is advisable to create an index card for each bearing and enter all important data. The entries should be expanded to include further operating data such as temperatures measured and rolling pressures. In this way, an important document is achieved which can be used for better assessment of the operating conditions and lifetime of the bearing than is possible in calculation on the basis of load assumptions.

Industrial Service

Within its Industrial Service concept, Schaeffler offers high quality products, services and training, Figure 70.
The Catalogue IS1 for „Mounting and Maintenance“ gives an overview of the portfolio:

- Mounting
- Lubrication
- Condition Monitoring
- Reconditioning

The employees of Schaeffler worldwide will be pleased to help you select the ideal products, services and training courses.



70: Portfolio

Storage of rolling bearings

The performance capability of modern rolling bearings lies at the boundaries of what is technically achievable. The materials, dimensional and geometrical tolerances, surface quality and lubrication have been optimised for maximum levels of function, which means that even slight deviations in functional areas, such as those caused by corrosion, can impair the performance capacity. In order to realise the full performance capability of rolling bearings, it is essential to match the anti-corrosion protection, packaging, storage and handling to each other. Corrosion protection and packaging constitute part of the bearing and are optimised such that they preserve all characteristics of the product at the same time as far as possible. In addition to protecting the surface against corrosion, this includes emergency running lubrication, friction, lubricant compatibility, noise behaviour, resistance to ageing and compatibility with rolling bearing components (cage and seal material).
Bearings that have been dismounted and will not be required for some time must be washed, preserved immediately and packed. Washing out should be carried out using kerosene.
For preservation, smaller bearings are dipped in anti-corrosion oil, while larger bearings should be carefully sprayed. Instead of then packing the bearings, they can be stored in oil.

If chocks with bearings fitted are not required again immediately, it must be checked whether there has been ingress of water. If this is the case, the grease packing should of course be renewed or, for example in the case of pneumatic oil lubrication, the bearing arrangement should be cleaned and preserved. For storage, the sides of the chocks are closed off using sealing shields.

Storage conditions for rolling bearings

As a basic prerequisite, parts must be stored in a closed storage area which cannot be affected by any aggressive media, such as exhaust gases from vehicles or gases, mist or aerosols of acids, lyes or salts. Direct sunlight should be avoided since, apart from the harmful effects of UV radiation, it can lead to wide temperature fluctuations in the packaging. The temperature should be constant and air humidity should be as low as possible. Jumps in temperature and increased humidity lead to condensation.
The following conditions must be fulfilled:

- frost-free storage, i.e. a temperature of +5 °C (this prevents formation of white frost, a maximum of +2 °C is permissible for up to 12 hours per day)
- maximum temperature +40 °C (to prevent excessive drainage of anti-corrosion oils)
- relative humidity 65 % (with changes in temperature, up to 70 % is permissible for a maximum of up to 12 hours per day).

Temperature and humidity must be continuously monitored. This can be carried out using a datalogger. The measurements must not be taken at intervals of no more than 2 hours.
At least 2 measurement points must be selected: The highest point and the lowest point in the vicinity of an external wall at which the goods can be stored.
Larger bearings with rings of relatively small thickness should not be stored standing but flat and supported over their whole circumference.

Notes

Notes

Selection of further FAG publications

The following list gives a selection of the available FAG publications.
Further information material is available by agreement.

Applications:

GL1	Large Size Bearings
PLS	The Bearing Solution for Strand Guide Rollers
TPI 129	Back-up Rolls for Multi-roll Cold Rolling Mills
TPI 148	Rolling Bearing Arrangements for Converters
TPI 157	Split Cylindrical Roller Bearings for the Bearing Arrangements of Rolling Mill Drive Shafts
TPI 176	Lubrication of Rolling Bearings
TPI 218	Sealed Spherical Roller Bearings
WL 41 140	Rolling Bearings for Rolling Mills
WL 43 165	Split Spherical Roller Bearings
WL 83 102	Rolling Bearing Damage

Maintenance:

IS1	Mounting and Maintenance of Rolling Bearings
TPI 170	FAG DTECT X1s
TPI 207	Reconditioning and Repair of Rolling Bearings
TPI 214	FAG SmartCheck
TPI WL 80-64	FAG Detector III
TPI WL 80-69	FAG ProCheck
WL 80 366	FAG Wear Debris Monitor
WL 80 368	Competence in Maintenance
WL 80 372	FAG DTECT X1s Flyer
WL 80 374	Light the Dark

Mounting:

MH1	Mounting Handbook
MON 90	Mounting Manual for Grease Lubricated Split FAG Spherical Roller Bearings and Housings
PDB 27	Induction FAG Heating Devices HEATER
TPI 138	Rolling Bearing Tolerances
TPI 168	Arcanol Rolling Bearing Greases
TPI 180	FAG Tools for Thermal Dismounting
TPI 182	FAG Alignment Tools
TPI 195	FAG Pressure Generators
TPI 196	FAG Hydraulic Nuts
TPI 200	FAG Heating Devices for Mounting of Rolling Bearings
WL 80-56	FAG Tools for Mechanical Mounting and Dismounting of Rolling Bearings
WL 80 112	Mounting of Rolling Bearings
WL 80 154	Mounting Manual: Four-row Tapered Roller Bearings
WL 80 369	New Possibilities for the Heating of Bearings
WL 80 376	FAG Medium Frequency Heating Device
WL 80 382	FAG CONCEPT8 – Compact Small Lubrication Device for Grease and Oil

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