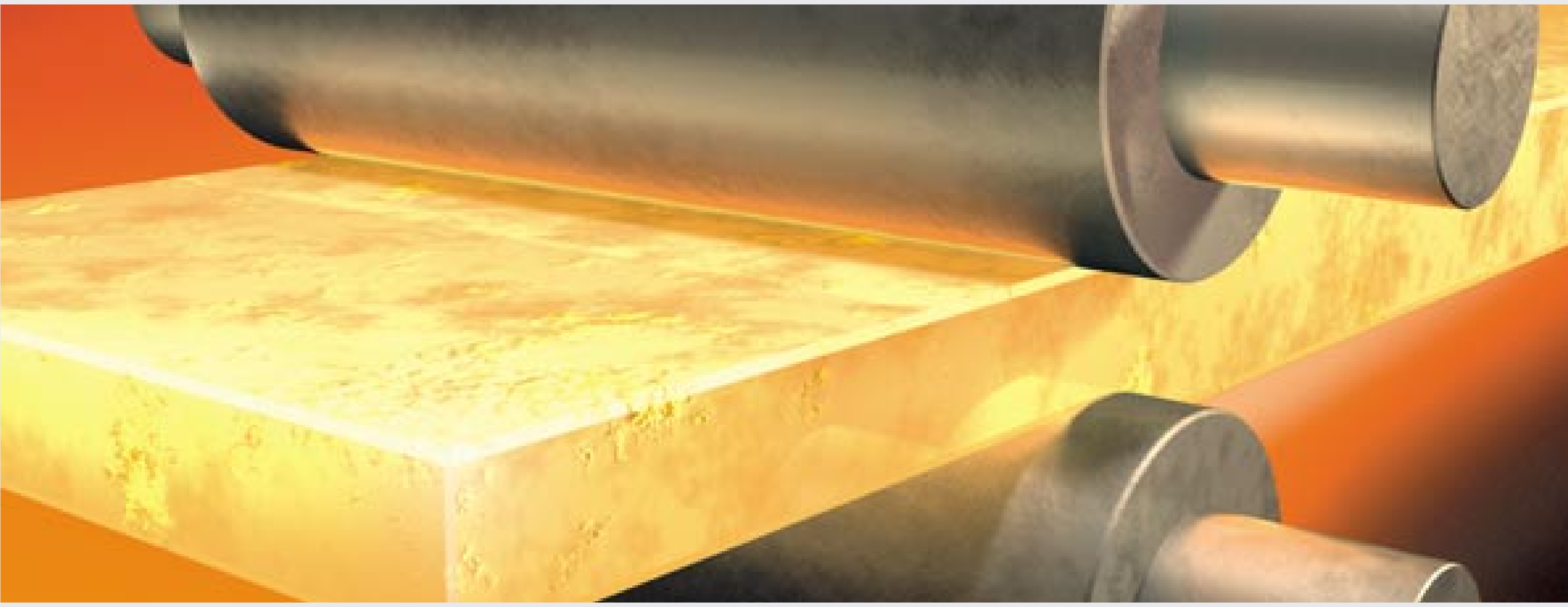


FAG Rolling Bearings for Rolling Mill Applications

A Member of the
Schaeffler Group

Preface

FAG Kugelfischer, the founder of the ball and ball bearing industry, has been producing ball and roller bearings of all types for over 100 years. FAG started designing and producing roll neck bearings very early and is well experienced in this field.
This publication includes the principles of roll neck bearing selection and calculation. The mounting and maintenance of these bearings are also explained in detail. Other specific questions not covered in this publication should be referred to FAG experts for advice.
The dimensions and technical data of the rolling bearings for rolling mills are given in FAG Publ. No. WL 41 140. A selection of publications about roll neck bearing arrangements and general topics concerning rolling bearing engineering (e.g. dimensioning, mounting and dismounting, lubrication and maintenance) is listed on page 68 of this publication.

Contents

Roll neck bearings

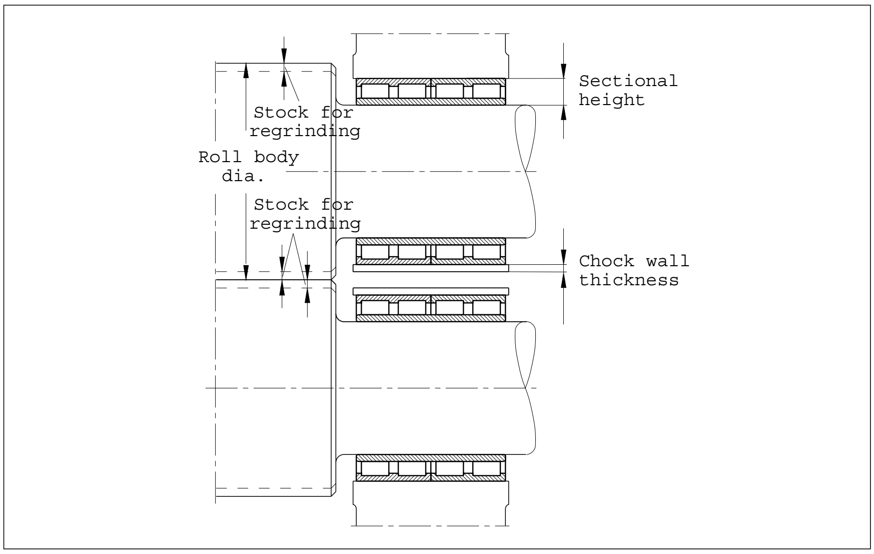
Conditions governing design

Conditions governing design

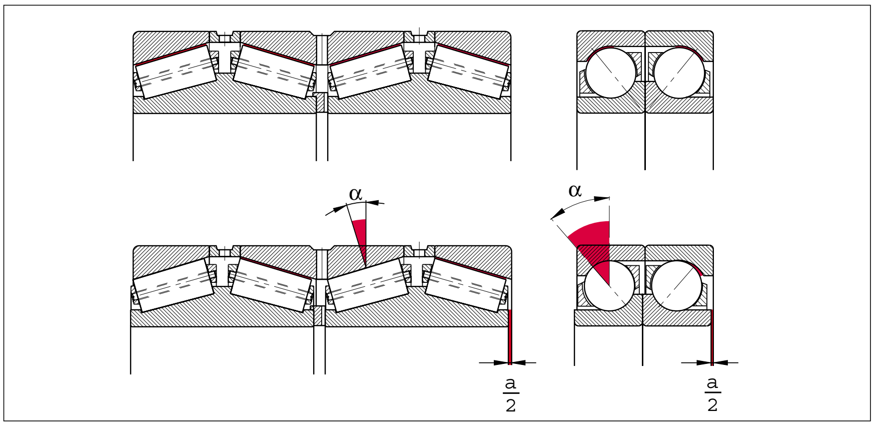
Roll neck bearings are very heavily loaded and subjected to high unit or specific pressure. Also, the mounting space particularly in radial direction is severely restricted as can be seen in fig. 1.
In consequence, a bearing of low sectional height yet with a very good load carrying capacity has to be provided.
The radial mounting space of a roll neck bearing is limited in that the outside diameter of the bearing is dictated by the roll body diameter minus a certain amount of stock allowed for roll regrinding and minus the wall thickness of the chock. Also, the roll neck diameter which controls the bearing bore diameter has, under high loading, to have adequate bending strength. A compromise will therefore have to be found between the diameter of the roll neck and its bending strength on the one hand and the bearing section height and its load carrying capacity on the other.
The available mounting space should be primarily used to accommodate the radial bearings since, compared to the radial loads, the axial loads are relatively small.
Roller bearings have a higher load carrying capacity than ball bearings. Therefore, roller bearings, i.e. cylindrical roller bearings, tapered roller bearings or spherical roller bearings, become the automatic choice for carrying the radial loads. These bearings are made of through-hardened rolling bearing steel or in some cases of casehardening steel.
The selection of the bearings for a specific application is influenced by the frequency of roll change.

Usually the chocks have to be removed when grinding the roll barrel. When using non-separable bearings such as spherical roller bearings, whose inner rings are tightly fitted on the roll neck, this task is made more difficult than is the case with cylindrical roller bearings where the chock together with the outer ring and the roller-and-cage assembly can be withdrawn, leaving the shrink-fitted inner ring on the roll neck.

Four-row tapered roller bearings or a pair of spherical roller bearings generally are loose-fitted on a cylindrical roll neck. Thus the chocks can be easily removed; however, the field of application of these bearing types is limited due to the unsuitability of loose fits for high-speed rolling.
If radial cylindrical roller bearings are used, the thrust loads have to be accommodated by a separate bearing. The provision of separate bearings



1: Available mounting space



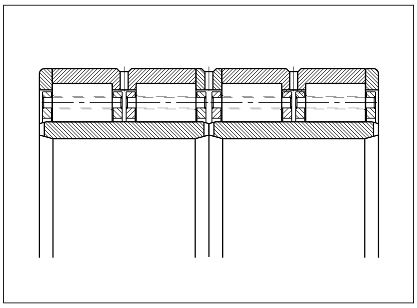
2: Axial clearance "a" as a function of radial clearance and contact angle α

Roll neck bearings

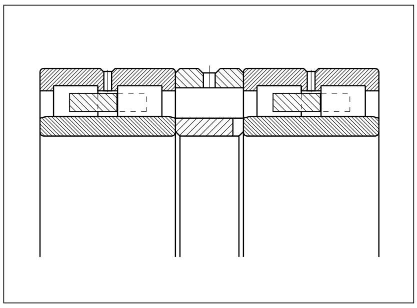
for the individual accommodation of radial and axial loads is particularly desirable in mills where close axial control is necessary for holding specified stock tolerances as is the case in shape-section rolling stands. Thrust bearings provide an extremely good axial guiding accuracy due to the very small or even zero axial clearance to which these bearings can be fitted. Radial bearings, on the other hand, having to perform the dual function of radial and axial guidance, will always have a larger axial clearance. Fig. 2 (page 4) shows how for a specified radial clearance the axial clearance "a" depends on the contact angle α. Spherical roller bearings feature the largest axial/radial clearance ratio. These values are smaller in the case of four-row tapered roller bearings. For angular contact ball bearings the ratio is smaller still.

Cylindrical roller bearings

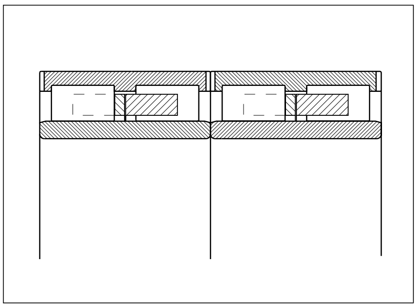
Within a given mounting space cylindrical roller bearings offer the greatest load carrying capacity. Consequently, this bearing type is suitable for the highest radial loads and – owing to its low friction coefficient – for the highest speeds.
Different types of cylindrical roller bearings are used for roll neck support. Which type will be used in specific cases depends on the rolling stand design.
To accommodate the maximum number of rollers, especially in larger bearings, and hence provide an optimum load carrying capacity, the bearings are equipped with through-bored rollers and fitted with pin-type cages (fig. 3). Such a cage consists of two rings which retain the rollers laterally and are connected by the pins which pass through the centre of the rollers. This cage offers a high strength which is particularly important for roll neck bearings which are subjected to great acceleration and deceleration – e.g. in reversing mills.
To achieve a particularly good running accuracy, cylindrical roller bearings with rough-ground inner ring raceways are used. The raceways are finish-ground together with the roll when the ring is mounted on the roll neck.
Fig. 4 shows double-row cylindrical roller bearings of series 49. They are used mainly for work rolls. To reduce stress resulting from possible tilting moments, the bearing rings are separated by inner and outer spacers. The load carrying capacity of these bearings is of secondary importance. Primarily they must be suitable for high speeds.
Cylindrical roller bearings as shown in fig. 5 are used generally in fine-section and wire mills. They feature machined brass or steel cages. They are not only suitable for high speeds (up to 40 m/s), but they can also accommodate high loads.
The finishing sections of such mills operating at rolling speeds of up to 100 m/s and more handle one single strand. Single-row cylindrical roller bearings are usually used for this application. Their service life is perfectly sufficient.



3: Four-row cylindrical roller bearing with through-bored rollers and so-called pin-type cages



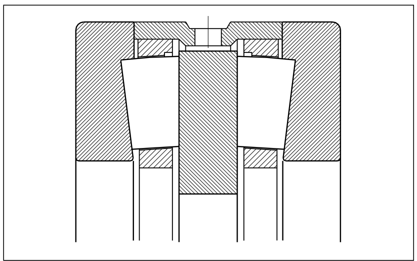
4: Double-row cylindrical roller bearings of dimensional series 49 with inner and outer spacers



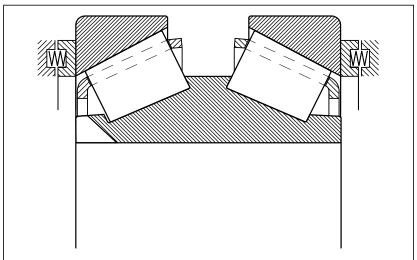
5: Four-row cylindrical roller bearing with machined cage for high rolling speed

Thrust bearings

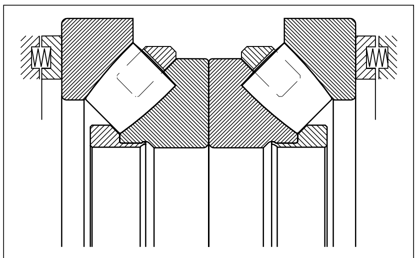
Normally the chock at the operator's end is axially located in the roll stand to which it transmits the axial forces. The thrust bearings can be of different types.
For high axial loads and medium speeds we recommend to use tapered roller thrust bearings (fig. 6), double-row tapered roller bearings with a large contact angle (fig. 7) or spherical roller thrust bearings (fig. 8).



6: Double-direction tapered roller thrust bearing with spacer ring

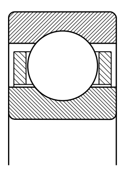


7: Double-row tapered roller bearing with large contact angle and outer rings axially preloaded with helical springs



8: Spherical roller thrust bearing pair for thrust accommodation in both directions

The tapered roller thrust bearing (fig. 6) features a spacer ring between the housing washers whose length is machined according to the axial clearance required.
Tapered roller thrust bearings, double-row tapered roller bearings and spherical roller thrust bearings are used principally in blooming mills, heavy-plate mills and hot strip mills, i.e. applications involving considerable axial forces combined with low to medium speeds. During operation only one roller row is purely axially loaded. The other row is not loaded. To make sure that the bearing kinematics is not impaired, the cups of the double-row tapered roller bearings and the housing washers of the spherical roller thrust bearings are on both sides preloaded with a minimum load by means of springs (figs 7 and 8).
In strip mills, in fine-section and wire mills the rolling speeds are often so high that tapered roller thrust bearings and spherical roller thrust bearings cannot be used.
In these cases the axial load is accommodated by angular contact ball bearings or deep groove ball bearings.
For the back-up rolls of large four-high strip and foil mills a deep groove ball bearing (fig. 9) will often be sufficient for accommodating the axial loads. Generally it has the same sectional height as the adjacent radial cylindrical roller bearing.
A double-row tapered roller bearing with a large contact angle can be used instead of the deep groove ball bearing. The required load ratings can thus be obtained with a considerably smaller bearing. These smaller double-row tapered roller bearings make it possible to use smaller surrounding components so that the cost for the overall construction can be reduced.
Work rolls of four-high strip mills and rolls of two-high fine-section and wire mills are generally fitted with angular contact ball bearings (fig. 10) to carry the axial loads.
The chock at the drive end is not axially located in the stand; it is held on the roll neck by a deep groove ball bearing, which is sufficient considering that the guiding forces are not very high. This does not considerably increase the overall width of the whole bearing arrangement. It is advisable to use a deep groove ball bearing of the same sectional height as the radial bearing.
Some roll neck bearing arrangements feature identical thrust bearings both at the drive end and at the operator's end. This simplifies stock-keeping.
The deep groove ball bearings and the angular contact ball bearings in these applications only serve to transmit axial loads. To positively prevent the outer rings from transmitting any radial loads, the chocks should be radially relief-turned a few millimeters at the seats provided for the outer rings (see also table 50 on page 39).



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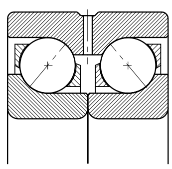

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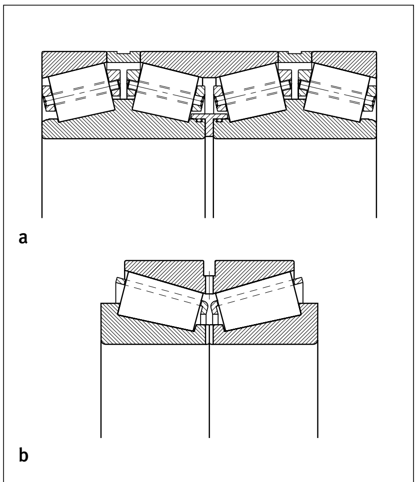
9: Deep groove ball bearing
10: Double-row angular contact ball bearing

Roll neck bearings

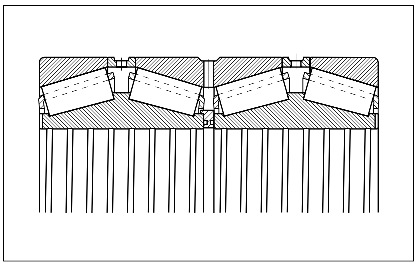
Tapered roller bearings

Tapered roller bearings

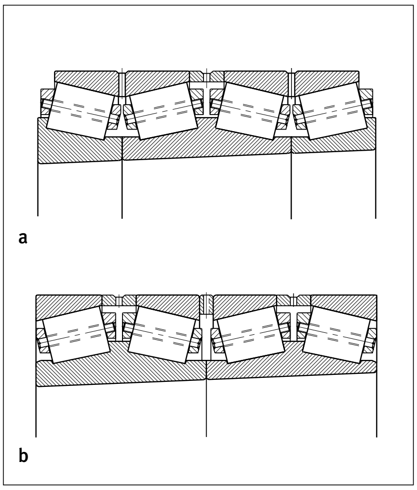
Due to the inclination of their rollers, tapered roller bearings accommodate radial and axial loads at the same time. Four-row and double-row tapered roller bearings (figs. 11a and b) are used in rolling mills.
Tapered roller bearings are separable. Despite this fact, however, it is not possible – as is the case with cylindrical roller bearings – to first fit the inner rings onto the roll neck, fit the outer rings into the chock and finally slip the chock onto the roll neck. The complete bearing has to be mounted into the chock, and then the chock together with the bearing pushed onto the roll neck. This means that the bearing inner ring must have a sliding fit on roll neck although technically (because of the circumferential load) it should be a tight fit.
The loose fit induces a condition of creep between the bearing bore and the roll neck, resulting in heat-up and wear. However, wear can be minimized by lubricating the surfaces of the inner ring and the roll neck well, see also page 44.
To provide space for extra grease, and thus improve the lubrication of the roll neck, spiral grooves are occasionally provided in the inner ring bore (fig. 12). This groove also serves to collect abraded particles. For the same reason radial grooves are also provided in the abutting inner ring faces.
In the case of work rolls supported by four-row tapered roller bearings, wear is moderate due to the low loads. Moreover, the allowable regrinding stock on the work rolls will most probably have been used up – necessitating new rolls – before roll neck wear has reached a point critical to continued bearing performance.
Large tapered roller bearings – like large cylindrical roller bearings – are provided with through-bored rollers and pin-type cages. This cage design is necessary for reversible stands because of the great acceleration and deceleration forces.
For the aforementioned reasons, four-row tapered roller bearings with a cylindrical bore cannot be used for all roll neck applications. Especially high speeds and loads call for a tight-fitted inner ring.
In these cases preference is usually given to tapered bore bearings fitted onto a tapered roll neck (fig. 13), whereby the required tight fit is simply achieved.
The inner ring of the design shown in fig. 13a consists of one double cone and two single cones, the outer ring of two double cups. Fig. 13b shows another design with four single cups separated by three spacer rings.
FAG manufactures four-row tapered roller bearings both with metric and inch dimensions and tolerances.



11: Tapered roller bearings
a: four-row
b: double-row



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



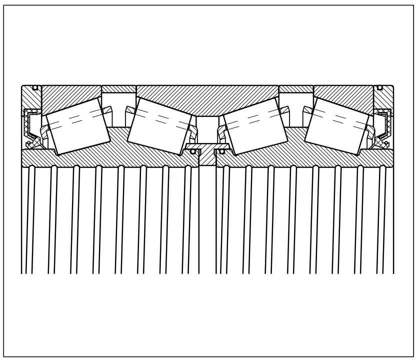
13: Four-row tapered roller bearing
with tapered bore and pin-type cage
a: outer ring consisting of two double cups
b: outer ring consisting of four single cups

Roll neck bearings

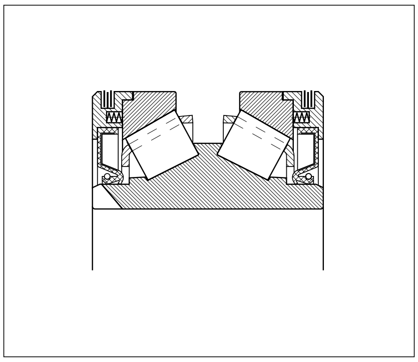
Sealed multi-row tapered roller bearings

Work roll bearing arrangements in hot and cold strip mills must be particularly efficiently sealed against large quantities of water or roll coolant mixed with dirt. Generally, work roll bearing arrangements are lubricated with grease. To cut grease cost and protect the environment, users try to reduce grease consumption. Longer bearing lives can be achieved by improving the lubrication and the cleanliness in the rolling contact areas.
In order to achieve these objectives, FAG has developed four-row tapered roller bearings with integrated seals (fig. 14) which have the same main dimensions as the unsealed bearings. High-grade rolling bearing grease is used which does not escape from the bearings, thus cutting grease consumption. The housing seals are packed with simple and cheap sealing grease. Sealed tapered roller bearings, due to the increased cleanliness in their lubricating gaps, generally have a longer life than unsealed ones although the integrated seals reduce the mounting space available for the rollers, resulting in a load rating reduction.

Sealed double-row tapered roller bearings are used as thrust bearings for work rolls (fig. 15).



14: Sealed four-row tapered roller bearing



15: Sealed double-row tapered roller bearing

Roll neck bearings

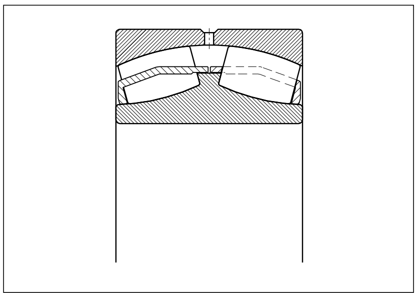
Spherical roller bearings · Tapered roller thrust bearings for screw-down mechanims

Spherical roller bearings

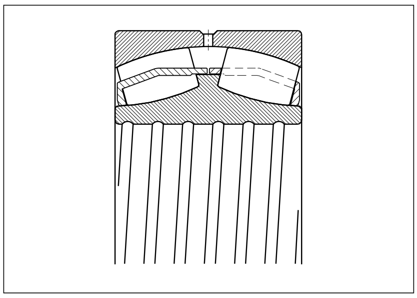
In rolling mills, spherical roller bearings are mainly used for low-speed roll neck applications without special demands on axial guiding accuracy. As the mounting space is limited in radial direction, preference is usually given to spherical roller bearings of series 240 and 241, both of which have a low sectional height (fig. 16).
Spherical roller bearings are self-aligning; they can accommodate radial and axial loads. Since their axial clearance is four to six times their radial clearance, their axial guiding accuracy is moderate.
Spherical roller bearings can be used for low and medium speeds. The rolling speed should not exceed 12 m/s. Owing to their self-aligning properties, the chock can be secured in the roll stand quite easily: Misalignments of the roll stand and roll neck bending are compensated for within the bearing.
This self-aligning property is also advantageous in mills with pre-stressed stands where the chocks are fixed with tie bars and consequently cannot align freely.
For applications where an easy, quick removal of the spherical roller bearings from the roll neck is required and where the rolling speed is low, the inner rings are loosely fitted on the roll neck.
Just like tapered roller bearings (fig. 12), spherical roller bearings can be provided with spiral grooves in the inner ring bore to improve lubrication of the surfaces in contact (fig. 17).
If the inner rings of spherical roller bearings are tightly fitted on the roll neck, mounting and dismounting will be easiest if bearings with a tapered bore are used. The hydraulic method simplifies mounting.
Spherical roller bearings are also preferable for cantilever roll mounting since they can compensate for the considerable roll deflections which have to be expected there.
In view of the relatively great axial clearance, profile rolls must be equipped with a supplementary thrust bearing.

Tapered roller thrust bearings for screw-down mechanisms

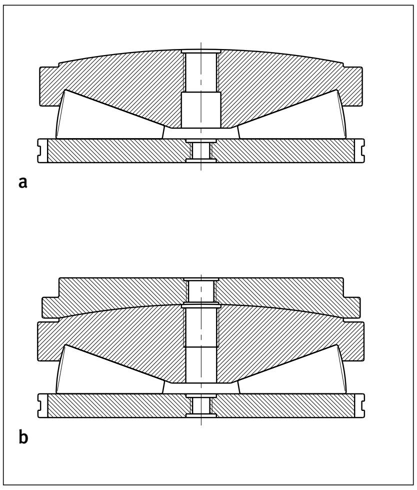
Single-direction tapered roller thrust bearings are frequently mounted between the pressure spindle and the upper chock (fig. 18). Due to their low friction, these bearings reduce the screw-down forces. This is particularly advantageous in large stands and in stands where the thickness of the rolled material changes frequently.



16: Spherical roller bearing



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



18: Tapered roller thrust bearings for screw-down mechanisms
a: design without pressure washer
b: design with pressure washer

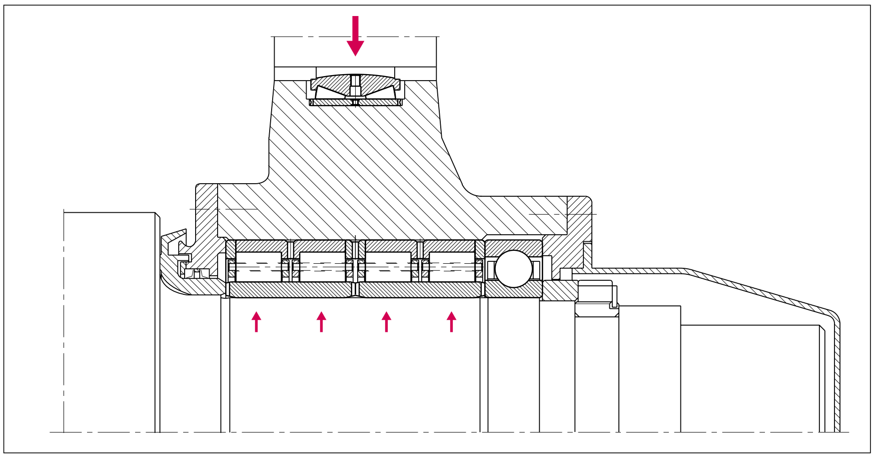
The magnitude of the rolling load today is generally determined with computer calculation programs. Major influences are the rolled stock, the rolling type (strip or groove rolling) and the proposed rolling schedule. The actual rolling loads sometimes differ considerably from the calculated values if the rolling schedule is not identical with the proposed one. Moreover, the shock loads at the entry of the material beetween the rolls escape calculation. The rolling load at the initial pass may be more than twice the constant load. The magnitude of the initial pass peak load depends on the configuration of the product entering the rolls and the temperature of the leading edge of the product. The initial pass peak load is of short duration and in most cases therefore does not enter into life calculation. However, it should not be overlooked that such stresses may occasionally drastically affect the service life of rolling bearings.
The distribution of the rolling load across the two bearing locations depends on the rolling stand design and the rolled stock.

Self-aligning chocks

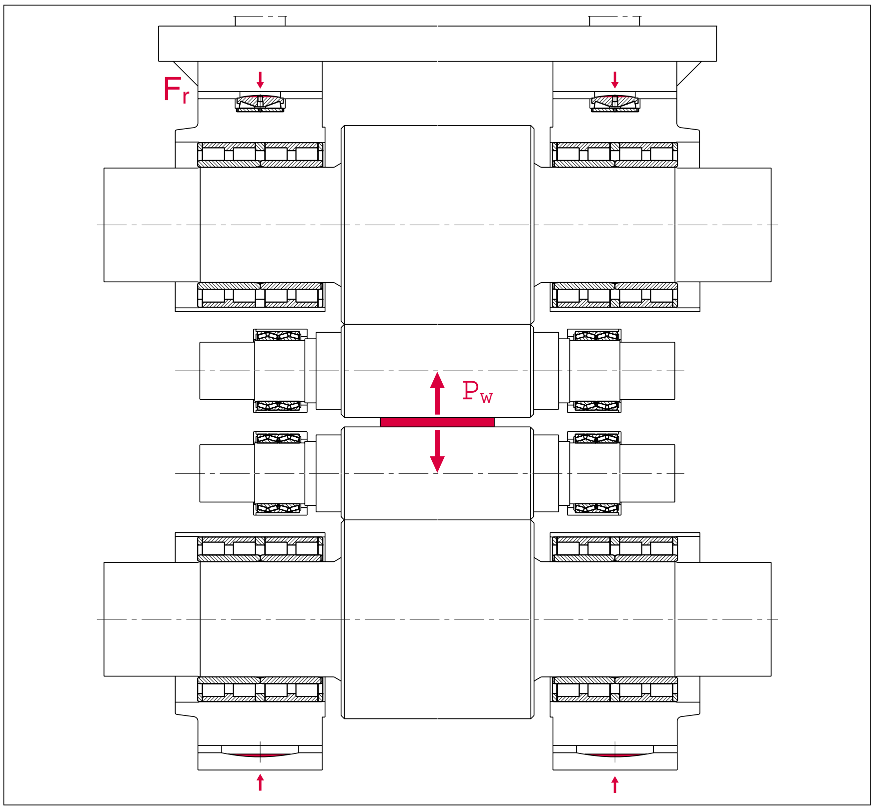
The chocks are supported separately in the stands. The rolling loads are transmitted to the stands through pressure bearings (tapered roller thrust bearings) with crowned support surfaces.
This enables the chocks to adapt to the position of the roll neck in case of roll deflection, poor screw-down conditions etc. This ensures that all roller rows of the multi-row bearings are evenly loaded (fig. 19).

The material passes symmetrically between the two bearing locations, each roll neck being loaded by ½ × P_w.

$F_r = ½ \times P_w$



19: Self-aligning chock



20: Self-aligning chocks for strip rolling

Calculation of bearing load

Self-aligning chocks

Groove rolling

It is necessary to distinguish between rolls with different grooves (e.g. in blooming mills) and rolls with identical grooves (e.g. in wire mills).
When rolling with different grooves (fig. 21), a rolling schedule should be established indicating the time percentages and rolling loads of the individual grooves in order to determine the load acting on the two necks. The fatigue life is based on the average load acting on the highest loaded neck.

For rolls with identical grooves (fig. 22), the different neck loads can be calculated from the rolling schedule.

Alternatively, the following indicative values can be assumed for the maximum neck load:

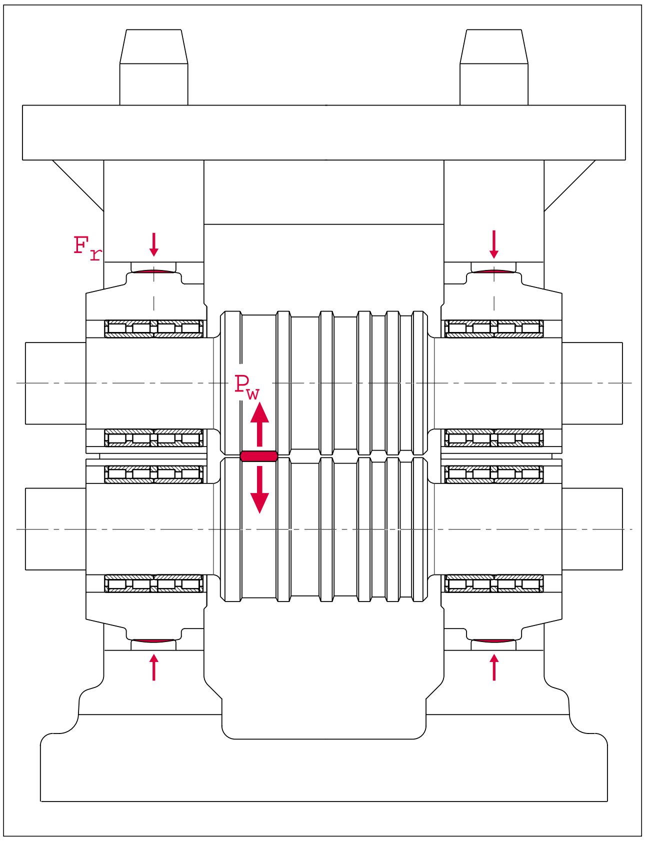
Single-strand rolling:
maximum neck load $F_r = 0{,}67 \times P_w$

Two-strand rolling:
maximum neck load $F_r = 1{,}1 \times P_w$

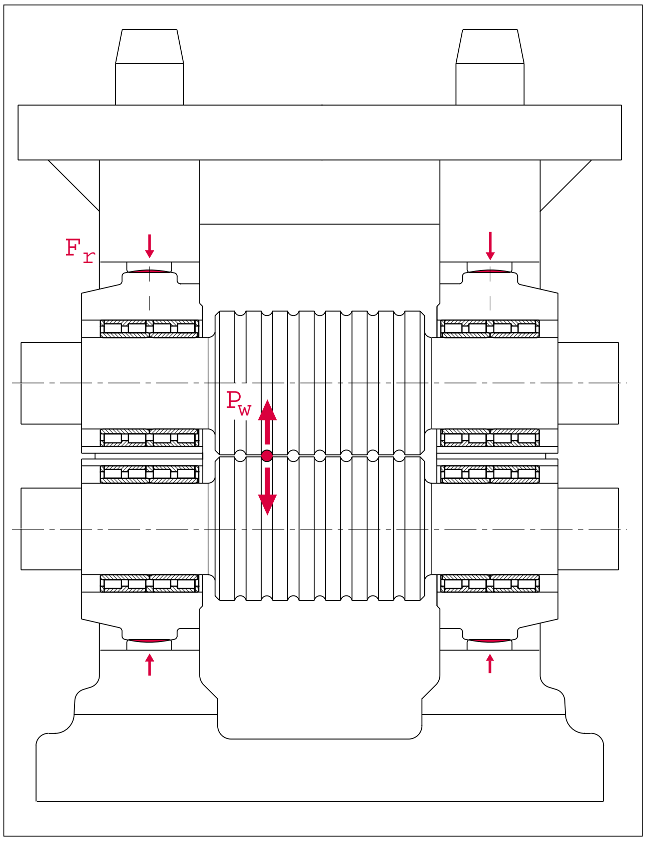
Four-strand rolling:
maximum neck load $F_r = 2{,}0 \times P_w$

P_w = Rolling load, relative to one strand

The calculation of the bearing load for variable speeds and roll loads is described on page 22.



21: Self-aligning chocks: Rolls with different grooves

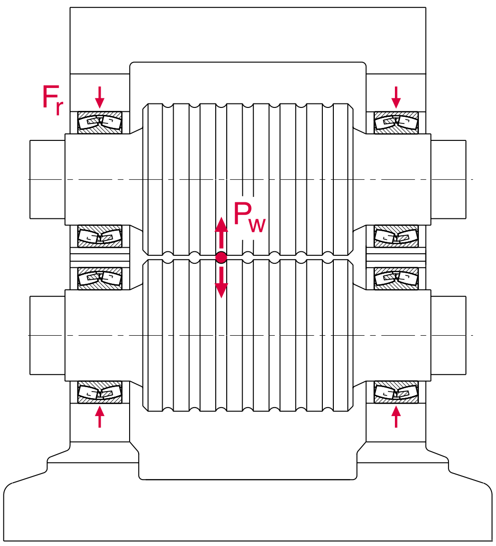


22: Self-aligning chocks: Rolls with identical grooves

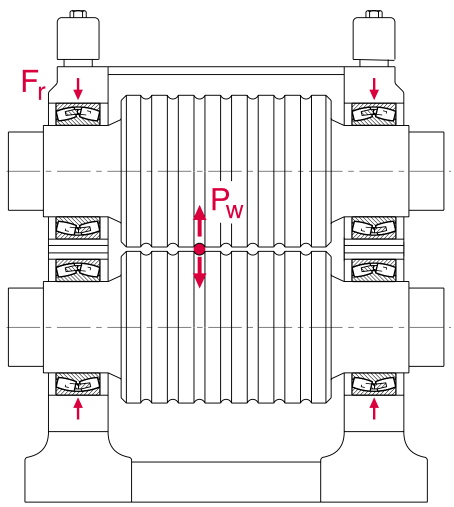
Rigid chocks

Both bearings are mounted into 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 bearing operation and calculation as long as the rolls are supported by spherical roller bearings.
When using double-row or multi-row cylindrical roller bearings, an uneven loading of the roller rows must be expected. FAG has developed a computerised method of calculating the roll deflection in order to determine the loads acting on the individual roller rows. Then it has to be checked if the higher loaded roller row offers a satisfactory fatigue life.
Rigid chock guidance is preferably chosen for groove rolling. The distribution of the rolling load on the two roll necks can be calculated as shown on page 11.

The upper and lower chocks are preloaded against each other so that they cannot adjust to any misalignment. This can cause both roll deflection and an offset of the two chocks relative to the roll axis. The majority of these stands are equipped with spherical roller bearings (figs. 23 and 24). If no separate thrust bearing is provided, the locating bearing must accommodate the axial load as well.



23: Rigid chock guidance



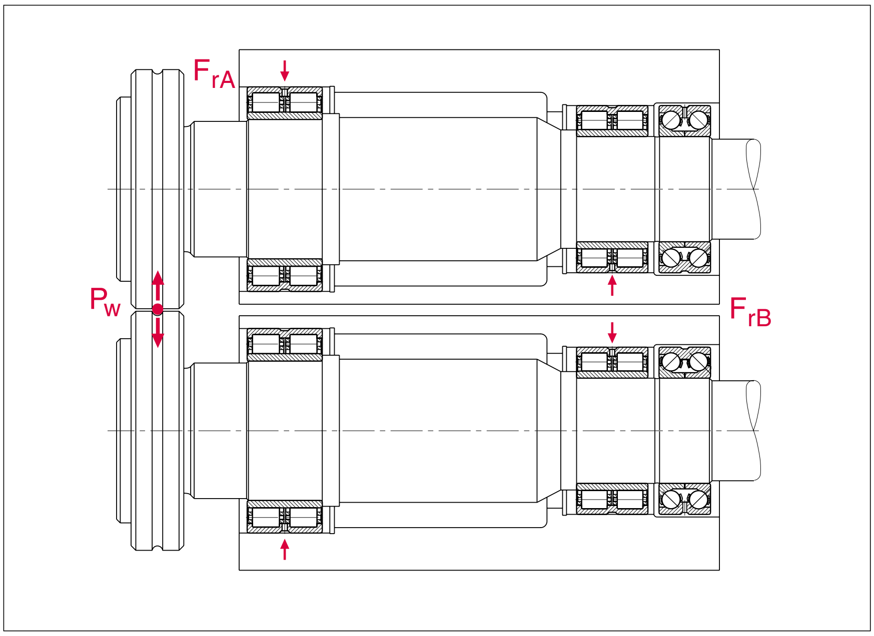
24: Pre-stressed stands

Cantilevered stands

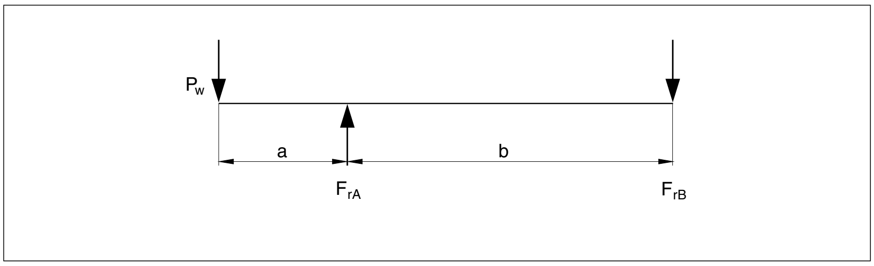
Rolling stands for small-section profile and wire rolling are provided with rolls with as a small a diameter as possible. In some cases, cantilevered rolls are used (fig. 25). When using multi-row bearings, the loads acting on them should be calculated from the roll deflection curve. In this way the fatigue life for the highest loaded roller row can be assessed.
The rolling load is distributed on the two supports as follows:

$$F_{rA} = P_w \times \frac{a + b}{b}$$

$$F_{rB} = F_{rA} - P_w$$



25: Cantilevered rolls



26: Load scheme to fig. 25

Calculation of bearing load

Calculation of roll deflection and of the load conditions within the rolling bearings

Calculation of roll deflection and of the load conditions within the rolling bearings

The calculation program Bearinx® can be used to calculate the bending behaviour of differently loaded, elastically supported elastic rolls. The support reactions, the internal bearing stresses, the equivalent stresses in the shafts and other important data are printed out (values) and drawn by a plotter.

The following influences can be analysed:

- Elasticity of plain and stepped solid or hollow shafts made of different materials, deformation due to shear loads.
- Shaft loads from rolling loads, from bending moments or from external forces acting on the bearings.
- Shaft support in the form of bearings with non-linear elasticity; the bearing geometry, the bearing clearance, the rolling element and raceway profiles as well as special load-transmitting conditions are taken into account.
- Any number of load cases (load/speed combinations) can be created and calculated.

The following calculation results are printed out:

The deflection and inclination of the roll axis at any point, the shearing forces and bending moments, the stresses, the bearing reaction forces, the bearing elasticity, the load conditions within the rolling bearings and the pressure distribution in the individual rolling elements' rolling contact areas. Based on the calculated stressing of the individual rolling contact areas, Bearinx® determines the life of the bearings with greater precision.

Calculating the roll deflection and load conditions within the rolling bearings (example)

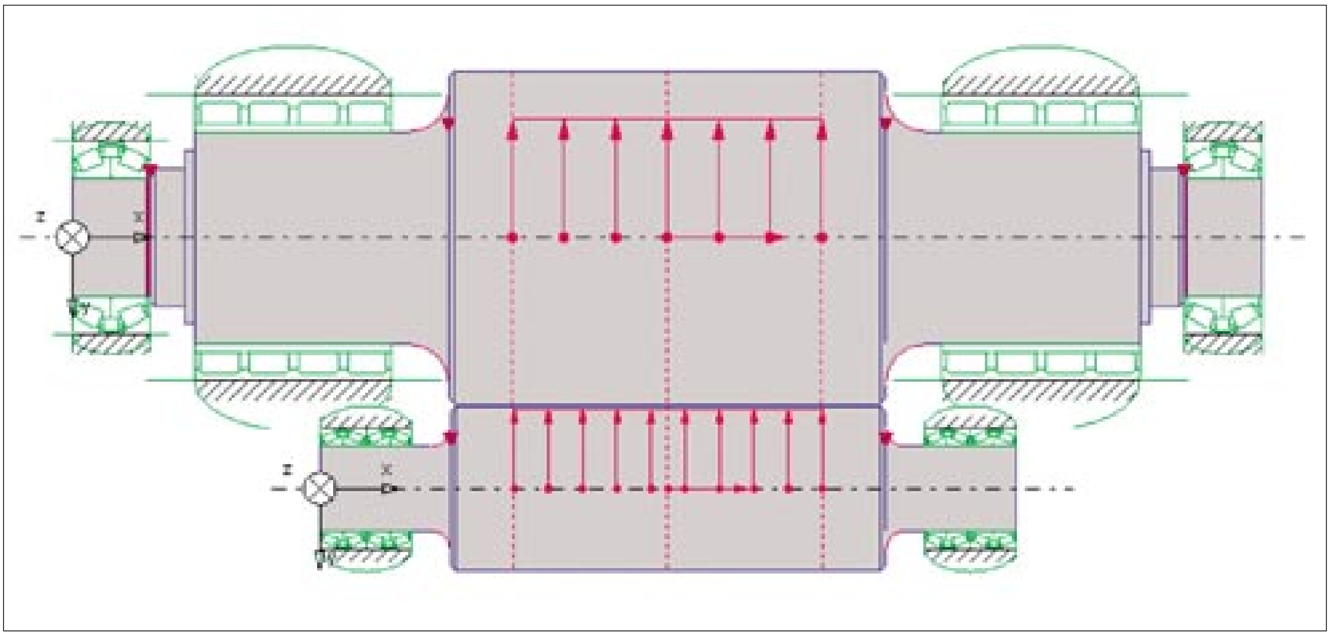
Subjects of calculation are the work roll and the back-up roll of a four-high cold rolling mill.

Load:
Rolling load $P_w = 8000$ kN

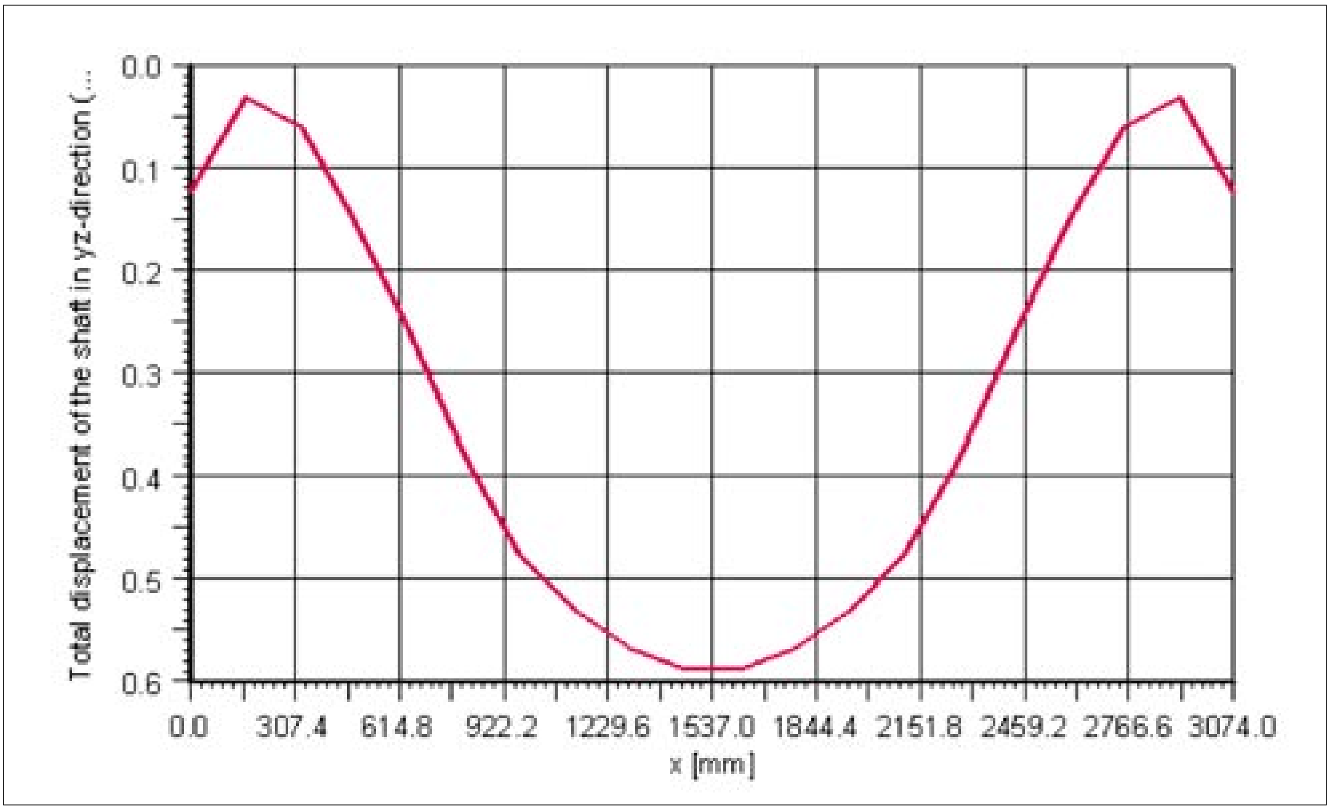
The input data describes the outline of the roll. The rolling load can either be input as a line load or split up into separate component loads which, arbitrarily distributed over the whole width of the rolled material, act on the roll barrel. The chocks are being considered as systems that are exposed to loads and/or moments. The self-aligning properties of the chocks will be taken into account. Roll bearings will be FAG cylindrical roller bearings and tapered roller bearings. Their spring characteristics are non-linear.

Calculation of bearing load

Calculation of roll deflection



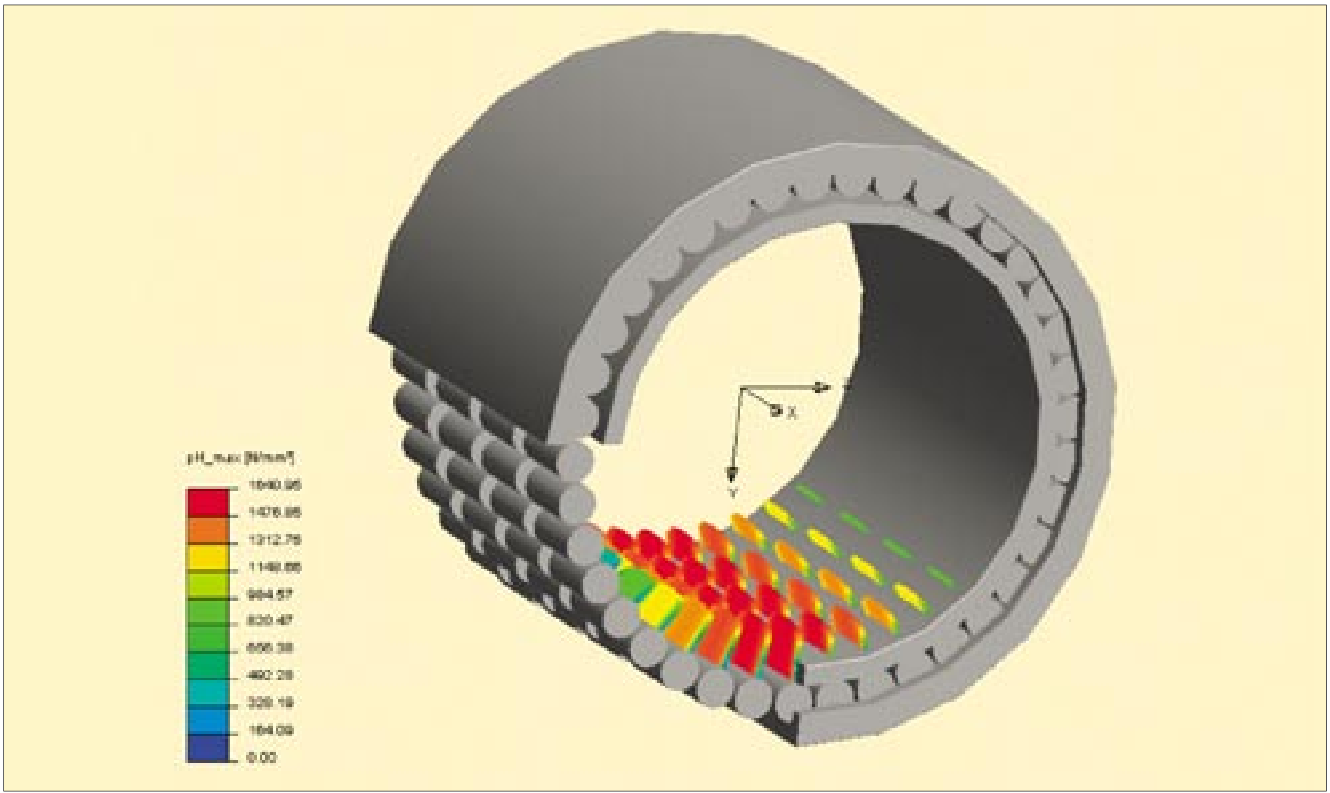
27a: Work roll and back-up roll bearing arrangement



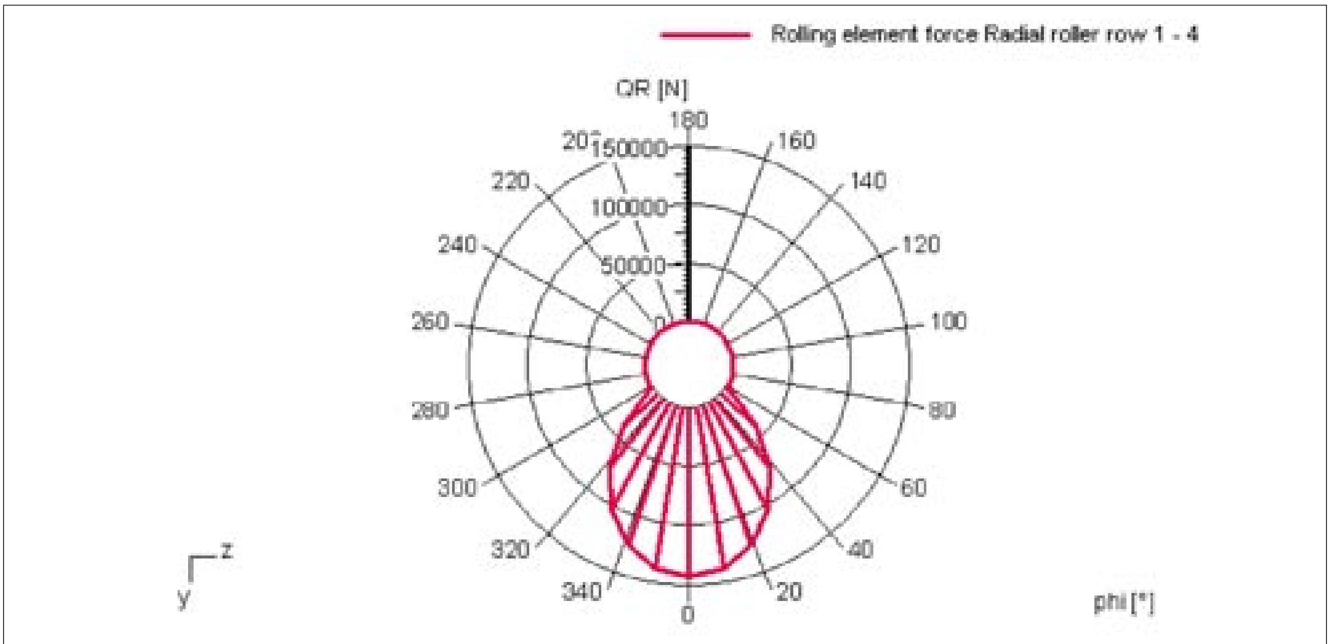
27b: Resultant back-up roll deflection in YZ direction

Calculation of bearing load

Calculating the load conditions and pressures (pressure distribution)



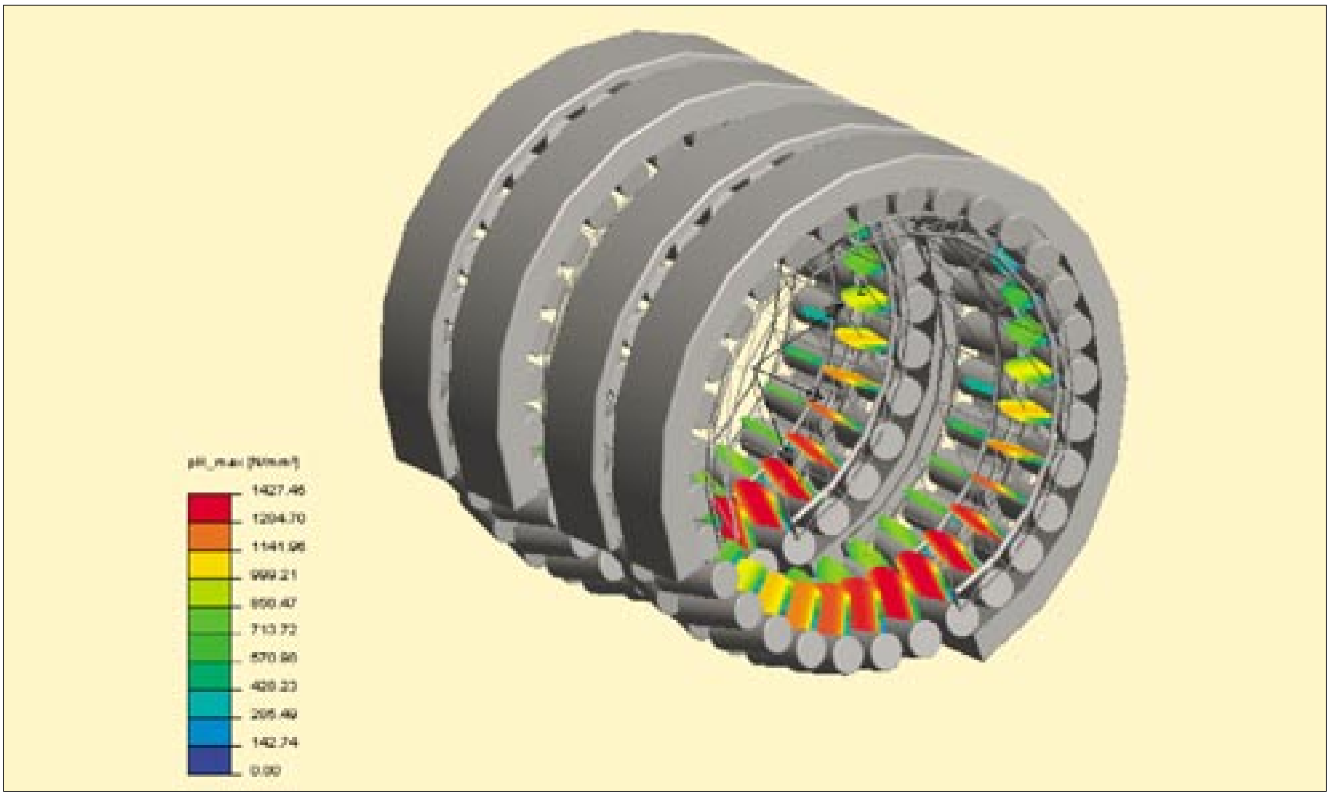
28a: Visualisation of the pressures acting on the four-row cylindrical roller bearing on the back-up roll



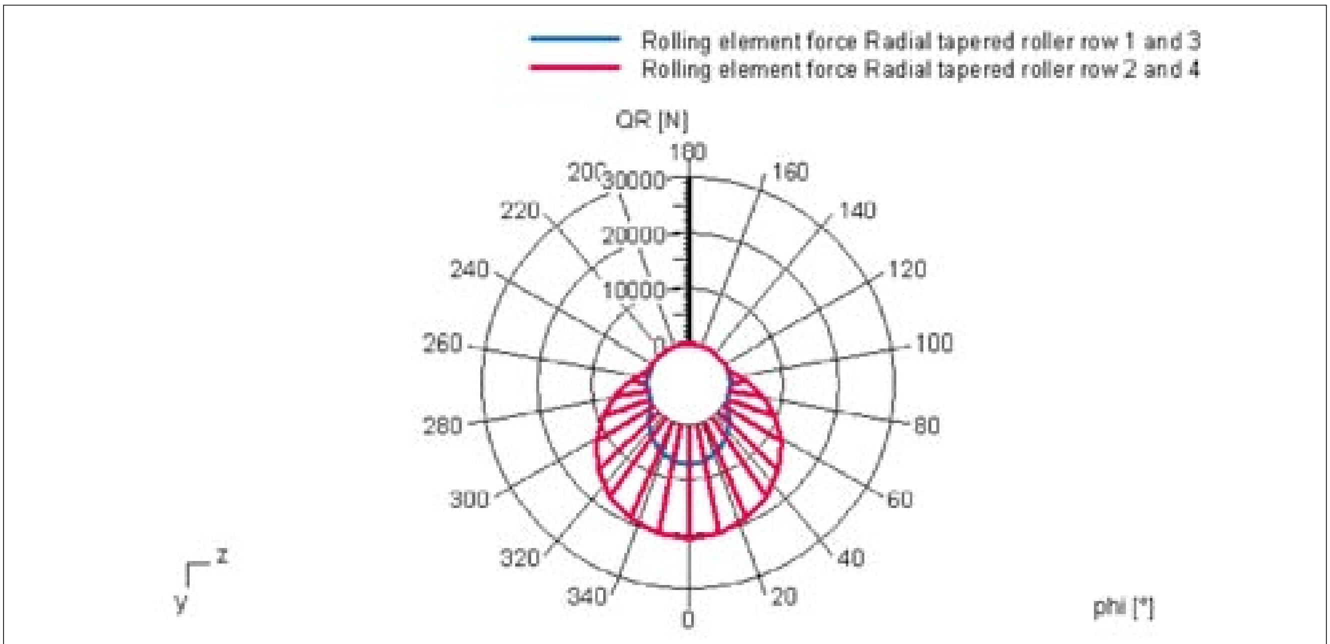
28b: Load distribution within the four-row cylindrical roller bearing on the back-up roll

Calculation of bearing load

Calculating the load conditions and pressures (pressure distribution)



29a: Visualisation of the pressures acting on the four-row tapered roller bearing on the work roll



29b: Load distribution within the four-row tapered roller bearing on the work roll

Dimensioning

Statically stressed bearings · Dynamically stressed bearings

Dimensioning calculation involves the comparison of a bearing's load with its load carrying capacity.
A differentiation is made between dynamic and static stress. Static stress implies that the loaded bearing is stationary (no relative movement between the rings) or is turning slowly. For these conditions the safety against excessive plastic deformations of the raceways and rolling elements is checked.
Most bearings are dynamically stressed. The rings turn relative to each other. The dimensioning calculation checks the safety against premature material fatigue of the raceways and rolling elements.

Statically stressed bearings

The calculation of the index of static stressing f_s serves to ascertain that a bearing with adequate load rating has been selected.

$$f_s = \frac{C_o}{P_o}$$

where
f_s Index of static stressing
C_o Static load rating [kN]
P_o Equivalent static load [kN]

The index of static stressing f_s is a safety factor against permanent deformations of the rolling elements. Roll neck bearings are usually not checked for static safety. The index f_s is determined, however, for the bearings of the screw-down mechanisms. A value of
$f_s = 1{,}8...2$
is recommended for this application.
The static load rating C_o [kN] is indicated for every bearing in the tables of FAG catalogues. This load (a radial one for radial bearings, an axial and centric one for thrust bearings) causes a theoretical contact pressure at the centre of the most heavily loaded contact area between rolling element and raceway of
p_o = 4200 N/mm² for ball bearings except self-aligning ball bearings
p_o = 4000 N/mm² for all roller bearings
Under the C_o load (equivalent to $f_s = 1$) a plastic total deformation of rolling element and raceway of about 1/10 000 of the rolling element diameter develops at the most heavily loaded contact area.
The equivalent static load P_o [kN] is a theoretical value. It is a radial load for radial bearings and an axial and centric load for thrust bearings. P_o causes the same stress at the centre of the most heavily loaded contact area between rolling element and raceway as the actual load combination. For the calculation of P_o, see FAG catalogues.

Dynamically stressed bearings

The standardized calculation method (DIN ISO 281) for dynamically stressed rolling bearings is based on material fatigue (formation of pitting) as the cause of failure. The life formula is:

$$L_{10} = L = \left(\frac{C}{P}\right)^p \qquad [10^6 \text{ revolutions}]$$

where
$L_{10} = L$ Basic rating life [10^6 revolutions]
C Dynamic load rating [kN]
P Equivalent dynamic load [kN]
p Life exponent

L_{10} is the basic rating life in millions of revolutions which is reached or exceeded by at least 90 % of a large group of identical bearings.
The dynamic load rating C [kN] is indicated for every bearing in the tables of the FAG catalogues. With this load an L_{10} rating life of 10^6 revolutions is reached.
The equivalent dynamic load P [kN] is a theoretical value. It is a radial load for radial bearings and an axial load for axial bearings which is constant in size and direction. P yields the same life as the actual load combination.

$$P = X \times F_r + Y \times F_a \qquad [\text{kN}]$$

where
P Equivalent dynamic load [kN]
F_r Radial load [kN]
F_a Axial load [kN]
X Radial factor
Y Thrust factor
The values for X and Y as well as information on the calculation of the equivalent dynamic load for the various bearing types can be found in the FAG catalogues and in Publ. No. WL 41 140 “FAG Rolling Bearings for Rolling Mills”.
While the radial loads acting on roll neck bearings can be determined with sufficient accuracy, little is known about the axial loads, which have to be estimated. In practice, it has been found satisfactory to assume the following values which are sufficiently safe:
for plain rolls (in two-high and four-high strip mills)
Axial load = 1 to 2 % of the rolling load
for grooved rolls
Axial load = 5 to 10 % of the rolling load

Dimensioning

Dynamically stressed bearings

$P = F_r$ for radial bearings which accommodate only radial loads.

For tapered roller thrust bearings which for design reasons can accommodate only axial loads, $P = F_a$.

$P = F_r$ (for one row) applies to purely radial loads or to $F_a/F_r \leq e$
If $F_a/F_r > e$,
$P = 0{,}4 \times F_r + Y \times F_a$ (for one row).
e is an auxiliary value, cp. FAG catalogues.

Different life exponents p are used for ball bearings and roller bearings.
p = 3 for ball bearings

$$p = \frac{10}{3}$$ for roller bearings

If the bearing speed is constant, the life can be expressed in hours

$$L_{h10} = L_h = \frac{L \times 10^6}{n \times 60} \text{ [h]}$$

where
$L_{h10} = L_h$ Basic rating life [h]
L Basic rating life [10^6 revolutions]
n Speed (revolutions per minute) [min^{-1}].

On converting the equation we obtain

$$L_h = \frac{L \times 500 \times 33\tfrac{1}{3} \times 60}{n \times 60}$$

$$\frac{L_h}{500} = \left(\frac{C}{P}\right)^p \times \frac{33\tfrac{1}{3}}{n}$$

or

$$\sqrt[p]{\frac{L_h}{500}} = \sqrt[p]{\frac{33\tfrac{1}{3}}{n}} \times \frac{C}{P}$$

where

$$f_L = \sqrt[p]{\frac{L_h}{500}}$$ Index of dynamic stressing,

i.e. $f_L = 1$ for a life of 500 hours,

$$f_n = \sqrt[p]{\frac{33\tfrac{1}{3}}{n}}$$ Speed factor,

i.e. $f_n = 1$ for a speed of $33\tfrac{1}{3}$ min^{-1}.
The table shown in fig. 32 lists the f_n values for ball bearings, the table in fig. 34 lists the fn values for roller bearings.

The life equation is therefore given the simplified form

$$f_L = \frac{C}{P} \times f_n$$

where
f_L Index of dynamic stressing
C Dynamic rating load [kN]
P Equivalent dynamic load [kN]
f_n Speed factor

Index of dynamic stressing f_L

The f_L value that is to be obtained for a correctly dimensioned bearing mounting is an empirical value obtained from field-proven identical or similar bearing mountings.
For comparisons with a field-proven bearing mounting the calculation of stressing must, of course, be based on the same former method. The usual data for the calculation and the f_L values are listed in the table in fig. 30.

For the conversion of f_L into the basic rating life L_h, see table 31 for ball bearings and table 33 for roller bearings.

30: Recommended f_L values and general stress conditions

Application	Recommended index of dynamic stressing f_L	Stress conditions
Roll stands	1...3	mean rolling load; rolling speed (f_L value determined by roll stand and rolling programme)
Rolling mill gears	3...4	nominal moment; nominal speed
Roller tables	2,5...3,5	weight of material, shocks; rolling speed

Dimensioning

Index of dynamic stressing f_L and speed factor f_n for ball bearings

31: f_L values for ball bearings

L_h h	f_L	L_h h	f_L	L_h h	f_L	L_h h	f_L	L_h h	f_L
100	0,585	420	0,944	1700	1,5	6500	2,35	28000	3,83
110	0,604	440	0,958	1800	1,53	7000	2,41	30000	3,91
120	0,621	460	0,973	1900	1,56	7500	2,47	32000	4
130	0,638	480	0,986	2000	1,59	8000	2,52	34000	4,08
140	0,654	500	1	2200	1,64	8500	2,57	36000	4,16
150	0,669	550	1,03	2400	1,69	9000	2,62	38000	4,24
160	0,684	600	1,06	2600	1,73	9500	2,67	40000	4,31
170	0,698	650	1,09	2800	1,78	10000	2,71	42000	4,38
180	0,711	700	1,12	3000	1,82	11000	2,8	44000	4,45
190	0,724	750	1,14	3200	1,86	12000	2,88	46000	4,51
200	0,737	800	1,17	3400	1,89	13000	2,96	48000	4,58
220	0,761	850	1,19	3600	1,93	14000	3,04	50000	4,64
240	0,783	900	1,22	3800	1,97	15000	3,11	55000	4,79
260	0,804	950	1,24	4000	2	16000	3,17	60000	4,93
280	0,824	1000	1,26	4200	2,03	17000	3,24	65000	5,07
300	0,843	1100	1,3	4400	2,06	18000	3,3	70000	5,19
320	0,862	1200	1,34	4600	2,1	19000	3,36	75000	5,31
340	0,879	1300	1,38	4800	2,13	20000	3,42	80000	5,43
360	0,896	1400	1,41	5000	2,15	22000	3,53	85000	5,54
380	0,913	1500	1,44	5500	2,22	24000	3,63	90000	5,65
400	0,928	1600	1,47	6000	2,29	26000	3,73	100000	5,85

32: f_n values for ball bearings

n min^{-1}	f_n	n min^{-1}	f_n	n min^{-1}	f_n	n min^{-1}	f_n	n min^{-1}	f_n
10	1,49	55	0,846	340	0,461	1800	0,265	9500	0,152
11	1,45	60	0,822	360	0,452	1900	0,26	10000	0,149
12	1,41	65	0,8	380	0,444	2000	0,255	11000	0,145
13	1,37	70	0,781	400	0,437	2200	0,247	12000	0,141
14	1,34	75	0,763	420	0,43	2400	0,24	13000	0,137
15	1,3	80	0,747	440	0,423	2600	0,234	14000	0,134
16	1,28	85	0,732	460	0,417	2800	0,228	15000	0,131
17	1,25	90	0,718	480	0,411	3000	0,223	16000	0,128
18	1,23	95	0,705	500	0,405	3200	0,218	17000	0,125
19	1,21	100	0,693	550	0,393	3400	0,214	18000	0,123
20	1,19	110	0,672	600	0,382	3600	0,21	19000	0,121
22	1,15	120	0,652	650	0,372	3800	0,206	20000	0,119
24	1,12	130	0,635	700	0,362	4000	0,203	22000	0,115
26	1,09	140	0,62	750	0,354	4200	0,199	24000	0,112
28	1,06	150	0,606	800	0,347	4400	0,196	26000	0,109
30	1,04	160	0,593	850	0,34	4600	0,194	28000	0,106
32	1,01	170	0,581	900	0,333	4800	0,191	30000	0,104
34	0,993	180	0,57	950	0,327	5000	0,188	32000	0,101
36	0,975	190	0,56	1000	0,322	5500	0,182	34000	0,0993
38	0,957	200	0,55	1100	0,312	6000	0,177	36000	0,0975
40	0,941	220	0,533	1200	0,303	6500	0,172	38000	0,0957
42	0,926	240	0,518	1300	0,295	7000	0,168	40000	0,0941
44	0,912	260	0,504	1400	0,288	7500	0,164	42000	0,0926
46	0,898	280	0,492	1500	0,281	8000	0,161	44000	0,0912
48	0,886	300	0,481	1600	0,275	8500	0,158	46000	0,0898
50	0,874	320	0,471	1700	0,27	9000	0,155	50000	0,0874

Dimensioning

Index of dynamic stressing f_L and speed factor f_n for roller bearings

33: f_L values for roller bearings

L_h	f_L	L_h	f_L	L_h	f_L	L_h	f_L	L_h	f_L
h		h		h		h		h	
100	0,617	420	0,949	1700	1,44	6500	2,16	28000	3,35
110	0,635	440	0,962	1800	1,47	7000	2,21	30000	3,42
120	0,652	460	0,975	1900	1,49	7500	2,25	32000	3,48
130	0,668	480	0,988	2000	1,52	8000	2,3	34000	3,55
140	0,683	500	1	2200	1,56	8500	2,34	36000	3,61
150	0,697	550	1,03	2400	1,6	9000	2,38	38000	3,67
160	0,71	600	1,06	2600	1,64	9500	2,42	40000	3,72
170	0,724	650	1,08	2800	1,68	10000	2,46	42000	3,78
180	0,736	700	1,11	3000	1,71	11000	2,53	44000	3,83
190	0,748	750	1,13	3200	1,75	12000	2,59	46000	3,88
200	0,76	800	1,15	3400	1,78	13000	2,66	48000	3,93
220	0,782	850	1,17	3600	1,81	14000	2,72	50000	3,98
240	0,802	900	1,19	3800	1,84	15000	2,77	55000	4,1
260	0,822	950	1,21	4000	1,87	16000	2,83	60000	4,2
280	0,84	1000	1,23	4200	1,89	17000	2,88	65000	4,31
300	0,858	1100	1,27	4400	1,92	18000	2,93	70000	4,4
320	0,875	1200	1,3	4600	1,95	19000	2,98	80000	4,58
340	0,891	1300	1,33	4800	1,97	20000	3,02	90000	4,75
360	0,906	1400	1,36	5000	2	22000	3,11	100000	4,9
380	0,921	1500	1,39	5500	2,05	24000	3,19	150000	5,54
400	0,935	1600	1,42	6000	2,11	26000	3,27	200000	6,03

34: f_n values for roller bearings

n	f_n	n	f_n	n	f_n	n	f_n	n	f_n
min^{-1}		min^{-1}		min^{-1}		min^{-1}		min^{-1}	
10	1,44	55	0,861	340	0,498	1800	0,302	9500	0,183
11	1,39	60	0,838	360	0,49	1900	0,297	10000	0,181
12	1,36	65	0,818	380	0,482	2000	0,293	11000	0,176
13	1,33	70	0,8	400	0,475	2200	0,285	12000	0,171
14	1,3	75	0,784	420	0,468	2400	0,277	13000	0,167
15	1,27	80	0,769	440	0,461	2600	0,270	14000	0,163
16	1,25	85	0,755	460	0,455	2800	0,265	15000	0,16
17	1,22	90	0,742	480	0,449	3000	0,259	16000	0,157
18	1,2	95	0,73	500	0,444	3200	0,254	17000	0,154
19	1,18	100	0,719	550	0,431	3400	0,25	18000	0,151
20	1,17	110	0,699	600	0,42	3600	0,245	19000	0,149
22	1,13	120	0,681	650	0,41	3800	0,242	20000	0,147
24	1,1	130	0,665	700	0,401	4000	0,238	22000	0,143
26	1,08	140	0,65	750	0,393	4200	0,234	24000	0,139
28	1,05	150	0,637	800	0,385	4400	0,231	26000	0,136
30	1,03	160	0,625	850	0,378	4600	0,228	28000	0,133
32	1,01	170	0,613	900	0,372	4800	0,225	30000	0,13
34	0,994	180	0,603	950	0,366	5000	0,222	32000	0,127
36	0,977	190	0,593	1000	0,36	5500	0,216	34000	0,125
38	0,961	200	0,584	1100	0,35	6000	0,211	36000	0,123
40	0,947	220	0,568	1200	0,341	6500	0,206	38000	0,121
42	0,933	240	0,553	1300	0,333	7000	0,201	40000	0,119
44	0,92	260	0,54	1400	0,326	7500	0,197	42000	0,117
46	0,908	280	0,528	1500	0,319	8000	0,193	44000	0,116
48	0,896	300	0,517	1600	0,313	8500	0,19	46000	0,114
50	0,885	320	0,507	1700	0,307	9000	0,186	50000	0,111

Dimensioning

Adjusted rating life calculation

Variable load and speed

If the load and speed for dynamically stressed bearings change over time, this fact must be taken into account when calculating the equivalent load. The curve is approximated by a series of individual loads and speeds of a certain duration q [%]. Then the equivalent dynamic load P is obtained from:

$$P = \sqrt[3]{P_1^{\,3} \times \frac{n_1}{n_m} \times \frac{q_1}{100} + P_2^{\,3} \times \frac{n_2}{n_m} \times \frac{q_2}{100} \ldots} \text{[kN]}$$

and the mean rotational speed n_m is obtained from:

$$n_m = n_1 \times \frac{q_1}{100} + n_2 \times \frac{q_2}{100} + \ldots [\,\text{min}^{-1}]$$

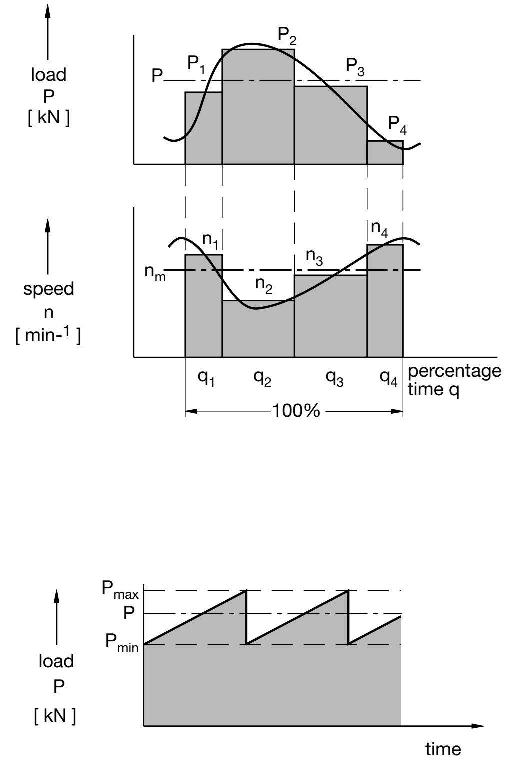
For the sake of simplicity, exponent 3 is indicated in the formulae for ball bearings and roller bearings. If the load is variable, but the speed constant:

$$P = \sqrt[3]{P_1^{\,3} \times \frac{q_1}{100} + P_2^{\,3} \times \frac{q_2}{100} + \ldots} \;[\,\text{kN}]$$

If the load grows linearly from a minimum value P_{min} to a maximum value P_{max} at a constant speed:

$$P = \frac{P_{min} + 2P_{max}}{3} \;[\,\text{kN}\,]$$

The mean value of the equivalent dynamic load may not be used for the calculation of the modified rating life (see page 23). The periods during which the same load type acts on a bearing must be summed up and the individual subsums entered in the L_{hnm} calculation. The modified rating life can then be calculated using the formula on page 28.



Adjusted rating life calculation

The basic rating life L or L_h more or less deviates from the actually attainable life of the rolling bearings. The equation $L = (C/P)^p$ takes into account only one of the operating conditions: the load. But the attainable life actually also depends on a number of other influences, e.g. the lubricating film thickness, the cleanliness in the lubricating gap, the lubricant doping and the bearing type. Therefore the standard DIN ISO 281:1993-01 has introduced, in addition to the basic rating life, the **“adjusted rating life”**, but so far there are no values for the factor which takes into account the operating conditions.

Dimensioning

Adjusted rating life calculation

Adjusted rating life

The adjusted rating life L_{na} is calculated using the following formula in accordance with DIN ISO 281:

$$L_{na} = a_1 \times a_2 \times a_3 \times L \quad [10^6 \text{ revolutions}]$$

or when expressed in hours:

$$L_{hna} = a_1 \times a_2 \times a_3 \times L_h \text{ [h]}$$

where

L_{na} Adjusted rating life [10^6 revolutions]

L_{hna} Adjusted rating life [h]

a_1 Life adjustment factor for reliability

a_2 Life adjustment factor for material characteristics

a_3 Life adjustment factor for operating conditions, in particular lubrication

L Basic rating life [10^6 revolutions]

L_h Basic rating life [h]

Life adjustment factor a_1 for requisite reliability

Rolling bearing failures due to fatigue are subject to statistical laws which is why the requisite reliability must be taken into account when calculating the fatigue life. Generally a 90% requisite reliability is assumed (corresponding to a failure probability of 10%). The L_{10} life is the basic rating life. The factor a_1 is used to express requisite reliabilities between 90 % and 99 %, see table 35.

Life adjustment factor a_2 for special material characteristics

Factor a_2 takes into consideration the characteristics of the material and its heat treatment. The standard permits factors of $a_2 > 1$ for bearings made of steel of particularly good cleanliness.

Life adjustment factor a_3 for special operating conditions

Factor a_3 takes into consideration the operating conditions, especially the lubrication condition under operating speed and operating temperature. DIN ISO 281:1993-01 does not yet include figures for this factor.

Modified rating life

Diverse and systematic laboratory investigations and the feedback from practical experience enable us today to quantify the effect of various operating conditions on the attainable life of rolling bearings. The life adjustment factors a_2 and a_3 that take into account the influences of special material characteristics and lubricating conditions and were introduced into DIN ISO 281 in 1977 were not quantified. For this reason several rolling bearing manufacturers have developed their own methods for calculating the adjusted rating life. These methods take into account the fact that the influences of material characteristics and lubrication are interdependent. FAG has published a calculation method for the factor a_{23}, which is needed to determine the attainable life, already a few years ago. This calculation method also shows that, under certain conditions, rolling bearings can be failsafe. In order to achieve a harmonisation and a better comparability with the life calculation methods of other leading manufacturers, FAG is introducing a new calculation method, which is based on DIN ISO 281 Bbl 1, by means of which the **modified rating life** L_{nm} is determined.

35: Factor a_1

Requisite reliability %	90	95	96	97	98	99
Factor a_1	1	0,62	0,53	0,44	0,33	0,21

Calculation of the modified rating life

The calculation method described in DIN ISO 281 Bbl 1:2003-4 for determining the modified rating life was derived from the methods developed by several rolling bearing manufacturers.
The modified rating life is obtained from

$L_{nm} = a_1 \times a_{DIN} \times L$ [10^6 revolutions]

and

$L_{hnm} = a_1 \times a_{DIN} \times L_h$ [h]

where

a_1	life adjustment factor for reliability (see page 40)
a_{DIN}	Life adjustment factor for operating conditions
L	Basic rating life [10^6 revolutions]
L_h	Basic rating life [h]

If influences vary during operation, the value of L_{hnm} has to be determined for each individual period of operation under constant conditions. Then the total adjusted rating life has to be determined on the basis of these values using the formula on page 28.

Life adjustment factor a_{DIN}

The standardised method for calculating a_{DIN} takes into account the following influences:
- the bearing load
- the lubrication condition (lubricant type and viscosity, additives, speed, bearing size)
- the fatigue limit of the material
- the bearing type
- the ambient conditions (contamination of the lubricant)

$a_{DIN} = f\,(e_C \times C_u/P,\ \kappa)$

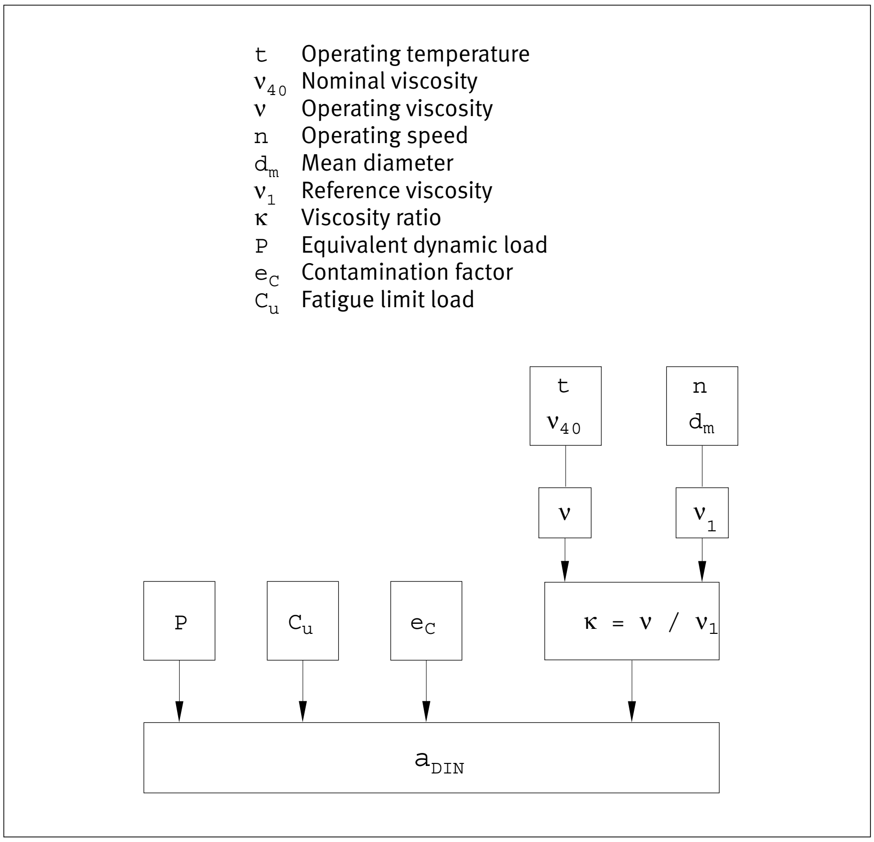
The fatigue limit load C_u takes into account the fatigue limit of the raceway material; the contamination factor e_C describes the increased stresses caused by contaminants in the bearing. P is the equivalent dynamic load.

$P = X \times F_r + Y \times F_a$ [kN]

where

F_r	Radial load [kN]
F_a	Axial load [kN]
X	Radial factor
Y	Thrust factor

The viscosity ratio κ is a measure of the lubricating film formation, see page 26.



36: Graph for determining a_{DIN}

Fatigue limit load C_u

According to DIN ISO 281/A2, the life modification factor a_{xyz} is determined by the ratio of the fatigue limit of the raceway material (σ_u) to the stress σ, which influences fatigue decisively.
The stress in the raceway, which influences fatigue decisively, primarily depends on the internal load distribution within the bearing and on the distribution of stress in the most heavily stressed rolling contact area. With ideal conditions in the rolling contact area, the fatigue limit of commonly used rolling bearing steels is reached at a Hertzian pressure of ca. 2200 N/mm².
The fatigue limit load C_u has been introduced for the practical application of the calculation method. C_u is determined in accordance with DIN ISO 281 Bbl. 1, based on the assumption of a contact pressure of 1500 N/mm².
Analogously to the static load rating C_0 according to DIN ISO 76, C_u is defined as the load at which the fatigue limit σ_u of the bearing material is just reached in the most heavily stressed contact area.
So the σ_u/σ ratio can be very approximately determined as a function of C_u/P.
The following factors have to be taken into account when determining C_u:

- type, size and internal geometry of the bearing
- profiles of rolling elements and raceways
- production quality
- fatigue limit of the material

The fatigue limit load values are indicated in FAG Publ. No. WL 41 140.

Contamination factor e_C

If the lubricant is contaminated with particles, cycling of these particles can generate permanent indentations in the raceways. Local increased stresses are generated at these indentations that reduce the life of the rolling bearing. This fact is taken into account by using the contamination factor e_C.
Guide values for the factor e_C are indicated in table 37.
The reduction of the rating life caused by solid particles in the lubrication gap is dependent on

- the type, size, hardness and number of particles
- the lubricant film thickness (viscosity ratio κ)
- the bearing size

The values indicated apply for contamination by solid particles. Other types of contamination such as contamination by water or other fluids cannot be taken into consideration here. If heavy contamination occurs ($e_C \to 0$) the bearings fail due to wear. The operating life is then considerably shorter than the calculated rating life.

37: Contamination factor e_C

Degree of contamination	Factor e_C $D_{pw} < 100$ mm	$D_{pw} \geq 100$ mm
Extreme cleanliness Particle size within lubricant film thickness Laboratory conditions	1	1
High cleanliness Oil filtered through extremely fine filter Sealed, greased bearings	0,8 to 0,6	0,9 to 0,8
Normal cleanliness Oil filtered through fine filter Greased, shielded bearings	0,6 to 0,5	0,8 to 0,6
Slight contamination Slight contamination of the lubricating oil	0,5 to 0,3	0,6 to 0,4
Typical contamination Bearing is contaminated with wear debris from other machine elements	0,3 to 0,1	0,4 to 0,2
Heavy contamination Bearing environment is heavily contaminated Bearing arrangement is insufficiently sealed	0,1 to 0	0,1 to 0
Severe contamination	0	0

D_{pw} pitch circle diameter; instead of D_{pw} the approximate mean bearing diameter $d_m = (D + d)/2$ can be used.

Viscosity ratio κ

The viscosity ratio κ is used as a measure of the quality of the lubrication film. κ is the ratio of the kinematic viscosity ν of the lubricant at operating temperature to the reference viscosity ν_1.
$\kappa = \nu/\nu_1$
The reference viscosity ν_1 is determined from diagram 38 as a function of the mean bearing diameter $d_m = (D + d)/2$ and the operating speed n.
The operating viscosity ν of a lubricating oil is obtained from the V-T diagram (fig. 39) as a function of the operating temperature t and the (nominal) viscosity of the oil at 40 °C.
In the case of lubricating greases, ν is the operating viscosity of the base oil.
Recommendations on oil viscosity and oil selection are given on page 31.

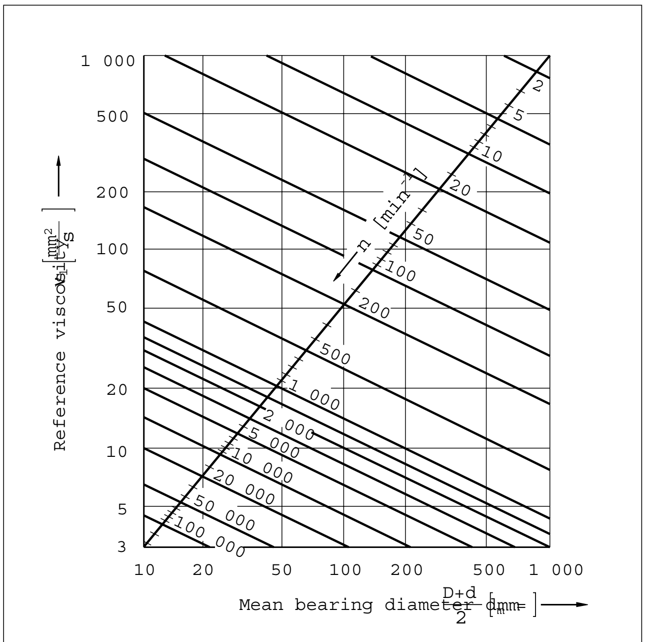
In heavily loaded bearings with a high percentage of sliding the temperature in the contact area of the rolling elements is up to 20 K higher than the temperature measurable at the stationary ring (without the influence of external heating).

Taking into account the effect of EP additives

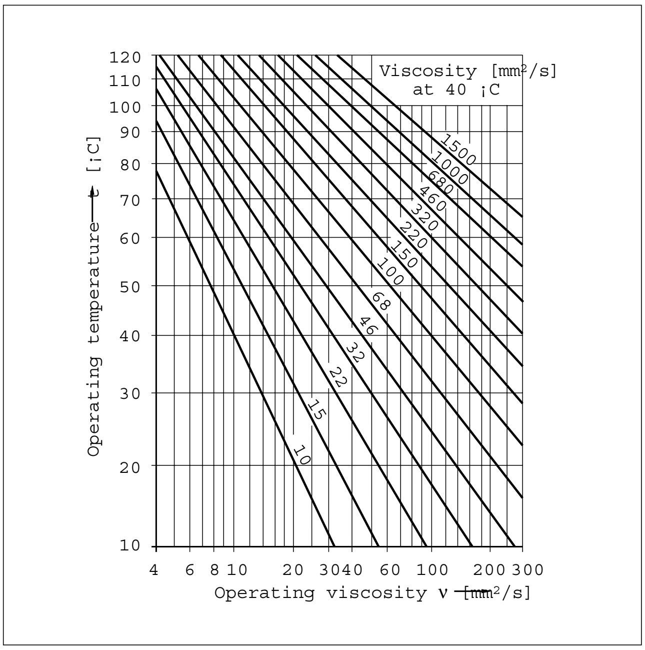
With a viscosity ratio of κ < 1 and a contamination factor $e_C \geq 0{,}2$, if lubricants with EP additives of proven effectiveness are used, the value κ = 1 can be used for the calculation. With a heavily contaminated lubricant (contamination factor $e_C < 0{,}2$) the effectiveness of the additives under the given contamination conditions must be proved. Proof of the effectiveness of the EP additives can be provided in field operation or in a rolling bearing test rig (FE 8) in accordance with DIN 51819-1.
If EP additives of proven effectiveness are used, i.e. if κ = 1, the life adjustment factor has to be limited to $a_{DIN} \leq 3$. If the value of a_{DIN} (κ) calculated for the actual κ is higher than 3, this value can be used for the calculation.

Diagrams for determining the life adjustment factor a_{DIN}

The life adjustment factor a_{DIN} can be taken from diagrams 40a to d on page 27 for
radial ball bearings (top left),
radial roller bearings (top right),
thrust ball bearings (bottom left),
roller thrust bearings (bottom right).
For κ > 4, the curve for κ = 4 shall be used.
If κ < 0,1 this calculation method is **not applicable**.



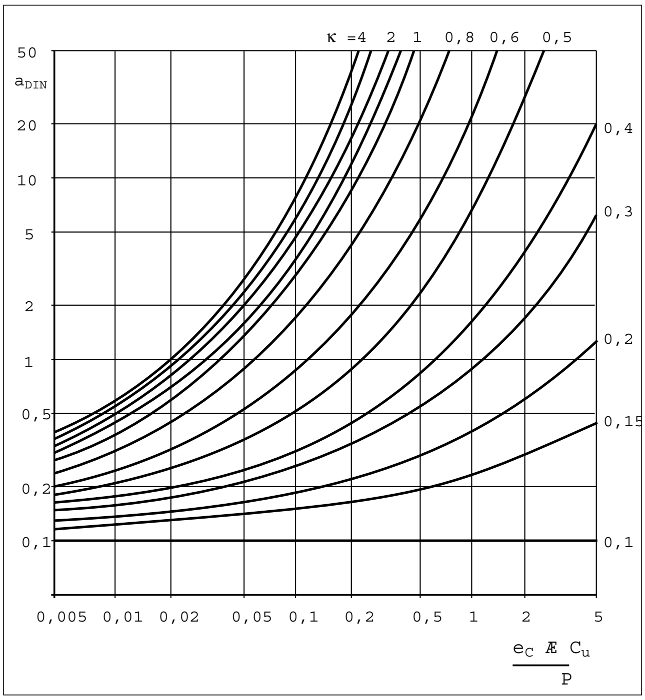
38: Reference viscosity ν_1



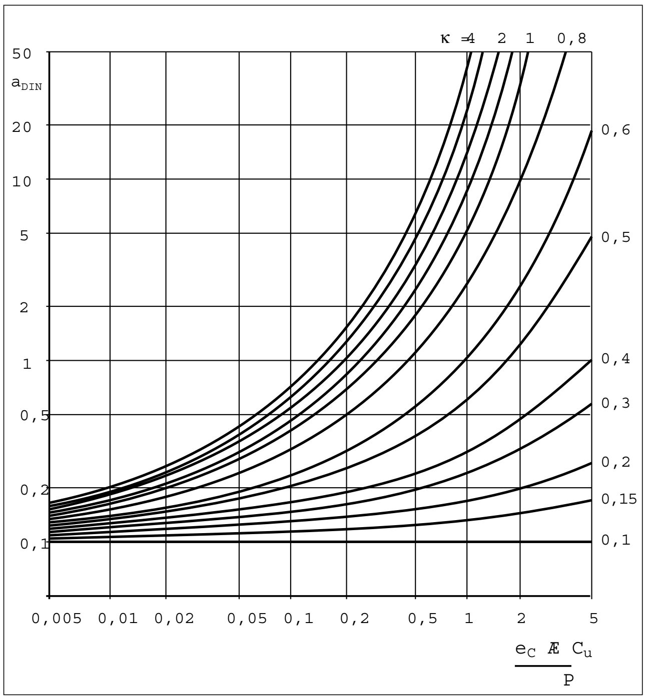
39: V-T diagram for mineral oils

Dimensioning

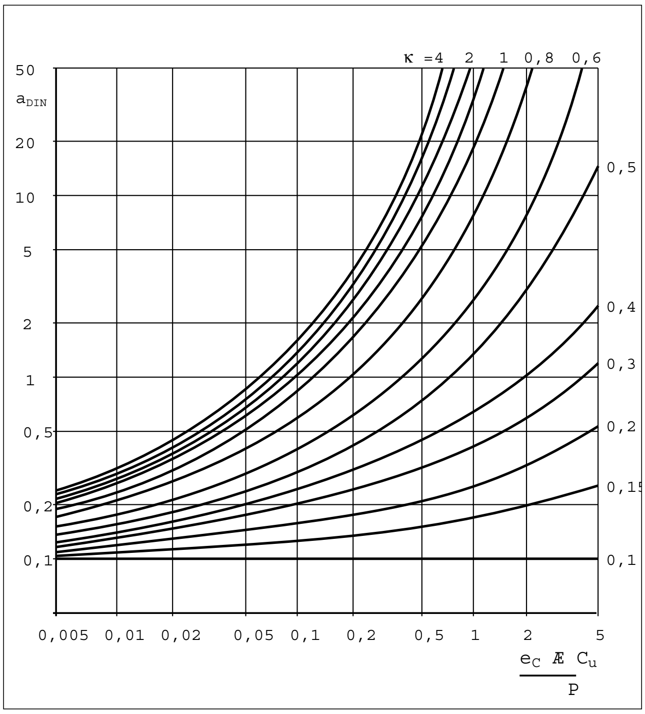
Adjusted rating life calculation



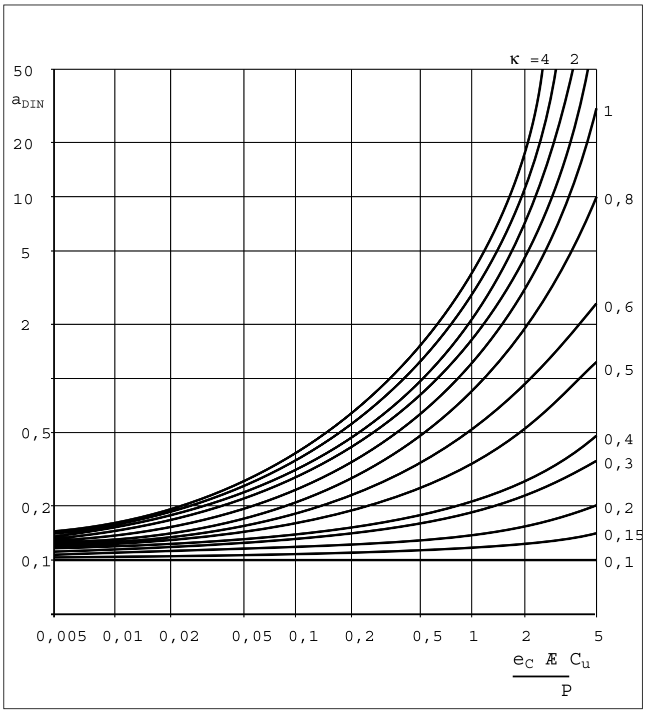
a: a_{DIN} for radial ball bearings



b: a_{DIN} for radial roller bearings



c: a_{DIN} for thrust ball bearings



d: a_{DIN} for roller thrust bearings

40: Life adjustment factor a_{DIN}

Dimensioning

Adjusted rating life calculation

Modified rating life under variable operating conditions

For applications where the load and the other life influencing parameters vary, the modified rating life (L_{hnm1}, L_{hnm2},...) must be calculated separately for each individual period of operation q [%] when the bearing operates under constant conditions. The modified rating life is calculated for the total operating time by means of the formula

$$L_{hnm} = \frac{100}{\frac{q_1}{L_{hnm1}} + \frac{q_2}{L_{hnm2}} + \frac{q_3}{L_{hnm3}} + \ldots}$$

Limits of life calculation

Like the former life calculation method, the adjusted rating life calculation takes only material fatigue into account as a cause of failure. The modified rating life obtained can only correspond to the actual service life of the bearing if the lubricant life, the service life of other components (e.g. the seal) or the life limited by wear are at least as long as the fatigue life of the bearing.

Bearing calculation on a PC

The BEARINX® calculation program combines extensive new calculation options with a user-friendly WINDOWS interface. It is particularly useful due to fast parametric analysis and automatic data import between individual calculation modules as well as an extensive bearing database. BEARINX® makes use of the higher precision of the calculation options available today that are described, for example, in addendum 4 to DIN ISO 281.
The program takes into account the influences of misalignment, operating clearance and loads acting on a bearing.

In roll neck bearings the lubricant – as in other rolling bearings – must form a load carrying film which prevents metal-to-metal contact between the bearing parts which would damage their surfaces. The thickness and load carrying capacity of the lubricant film depend on the viscosity of the oil, on the bearing speed and size, and on the lubricant properties. Also, the lubricant has to protect the bearing parts from corrosion, lubricate the lips of the seals (collar seals etc.) and fill the gaps of labyrinth seals.
Since the function of the lubricant in the sealing is different from its function in the bearing, it is advisable to lubricate bearings and seals separately and choose the best lubricant for each purpose. In many cases, however, this solution cannot be put into practice because of the hazard of mistaking one grease for the other (harmful mixture), complicated stockkeeping etc.

Grease Lubrication

Where operating conditions permit, grease is the lubricant of choice for roll neck bearings due to the simplicity of sealing and grease replenishment. Mineral oil companies produce a large number of special rolling bearing greases; however, these greases differ considerably in their composition and properties, making the selection of an appropriate grease for specific applications difficult. FAG offers a variety of particularly suitable bearing greases under the trade name Arcanol.

Table 41 lists a selection of FAG rolling bearing greases and their properties. For specific applications, we recommend customers to ask their grease supplier for exact data. The suitability of the FAG rolling bearing greases Arcanol for the various bearing types is known. But the supplier has to prove the suitability of unfamiliar greases. Where necessary, FAG can carry out performance tests for a customer.

41: A selection of rolling bearing greases and their properties

Grease type, thickener	Consistency (NLGI class)	Remarks, typical applications	FAG Arcanol	Temperature range °C	Base oil viscosity at 40 °C mm²/s	Speed	Load	Water resistance	Anti-corrosive properties
Lithium soap	3	universal rolling bearing grease, long lubrication intervals, e.g. in electric motors; good sealing grease	**MULTI3** (L71V)	−30...+140	80	medium	medium	stable up to 90°C	very good
Lithium/calcium soap; EP additives	2	rather difficult operating conditions, e.g. in back-up rolls and work rolls, esp. in sealed tapered roller bearings	**LOAD220** (L215V)	−20...+140	ISO VG 220	high	high	stable up to 90 °C	very good
Lithium soap; EP additives	2	rather difficult operating conditions, especially high speeds, esp. in sealed tapered roller bearings	**MULTITOP** (L135V)	−40...+150	85	very high	high	stable up to 90 °C	very good
Lithium/calcium soap; EP additives	2	very difficult operating conditions, esp. high impact loads	**LOAD400** (L186V)	−20...+140	400	medium	very high	stable up to 90 °C	very good
Lithium-calcium soap; EP additives	2	extremely difficult operating conditions, very high impact loads, e.g. in lifting tables	**LOAD1000** (L223V)	−20...+140	ISO VG 1000	low	very high	stable up to 90 °C	very good

Lubrication

Grease lubrication

Load and speed

Load and speed are of major importance when selecting an appropriate grease for a specific application. The stress due to speed can be estimated from the speed index $k_a \times n \times d_m$

where

- k_a Factor for the bearing type (see diagram 42)
- n Operating speed [min^{-1}]
- d_m Mean bearing diameter; $d_m = (D+d)/2$
- d Bearing bore [mm]
- D Bearing O.D. [mm]

The load ratio P/C is a measure of the specific loading

- P Equivalent dynamic load [kN] (see catalogues)
- C Dynamic load rating [kN] (see catalogues)

The table (fig. 42) can be used to determine which type of grease is suitable for specific operating conditions. For applications involving high speeds as well as high loads, the temperatures may increase, requiring a particularly temperature-resistant grease or special cooling measures. The upper speed and load limits of the lubricating greases must be observed; this information can be obtained from the mineral oil company or from FAG.

Other operating conditions

The **position of the roll axis** must also be taken into account in the grease selection. Vertical or inclined rolls may cause the grease to escape from the bearing and chock (effect of gravity). Therefore we recommend to provide baffle plates beneath the bearing and to choose a grease which is particularly adhesive and resistant to working (consistency class 3 or, where advisable, class 2).

Relubrication is another criterion for grease selection. If large quantities of grease are needed for bearings or seals, or if there are long lubricating ducts (e.g. central lubricating system), greases which can be pumped without problems, i.e. greases of consistency class 1 or 2, must be selected.
In damp environments and with longer idle times, roll neck bearings are exposed to a risk of **corrosion** due to condensation. Therefore, the greases used for these applications must have special anti-corrosive properties.
Bearing locations exposed to splash water must be protected by a seal. Seals and bearings should be relubricated at short intervals.
The table in fig. 43 shows an overview of the above criteria and permits users to select a suitable lubricating grease based on the required grease properties.

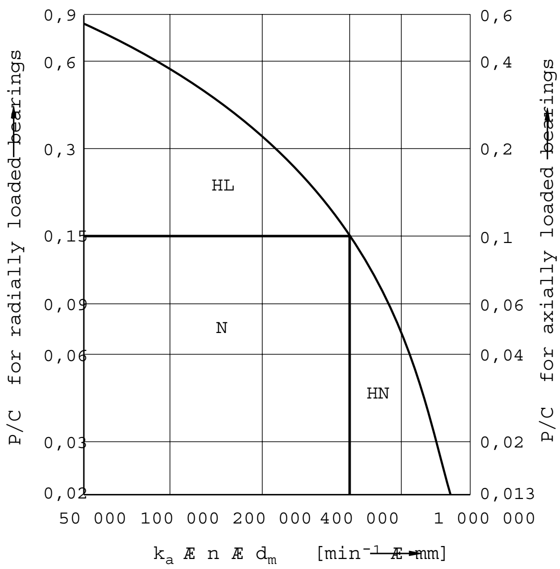
Range N
Normal operating conditions.
Rolling bearing greases K according to DIN 51825

Range HL
Heavy loads.
Rolling bearing greases KP according to DIN 51825 or other suitable greases

Range HN
High speed range
Greases for high-speed bearings.
For bearing types with $k_a > 1$, greases KP according to DIN 51825 or other suitable greases.

k_a values

- $k_a = 1$ deep groove ball bearings, angular contact ball bearings, four-point bearings, radially loaded cylindrical roller bearings
- $k_a = 2$ spherical roller bearings, tapered roller bearings
- $k_a = 3$ axially loaded cylindrical roller bearings, tapered roller thrust bearings



42: Grease selection from the load ratio P/C and the relevant bearing speed index $k_a \times n \times d_m$

Lubrication

Grease lubrication · Oil lubrication

43: Criteria for grease selection

Criteria for grease selection		Grease characteristics
Bearing stressing	Speed index Load ratio	Grease selection according to diagram 42 and table 41
Operating conditions	Position of roll axis	Grease of consistency class 3; softer greases require frequent relubrication
	Frequent relubrication	Good pumping capability, consistency class 1 or 2 for central lubricating systems
	For-life lubrication	Grease resistant to working whose service life and lubricating properties are known.
Environmental conditions	Extreme temperatures	Grease whose service life is suitable for the operating temperature; with constant relubrication, greases that resist the operating temperatures at least for a short period may be chosen too, but they must not solidify.
	Contamination by foreign matter	Stiff grease, consistency class 3
	Corrosion by condensed water	Emulsifying grease (e.g. lithium grease, lithium/calcium grease)
	Corrosion by splash water	Water-repellent grease (e.g. calcium complex grease, lithium/calcium grease)

Oil Lubrication

Viscosity requirements

Depending on the bearing speed and size, the oil must have a certain viscosity at operating temperature in order to form a load-carrying lubricant film and reach its rated life. This rated viscosity ν_1 is determined with the help of the diagram in fig. 38.
For a normal service life, bearings with a small amount of sliding motion should be lubricated with an oil whose operating viscosity ν at least equals the rated viscosity ν_1.
Rolling bearing types with unfavourable kinematics (axially loaded roller bearings, low-speed and highly loaded large size bearings) always require effective additives. If the lubricant film formed in the bearing is not adequate, these additives create a separating boundary film in the raceway/rolling element, rolling element/cage and rolling element/guiding lip contact areas. This boundary film prevents wear and premature fatigue.

Other demands on the oil

Most rolling bearing lubricating oils are mineral oils which contain additives that improve their properties.
These additives provide, for instance, a better oxidation stability, improve the oils' anticorrosive properties or reduce foaming. Dispersion agents keep finely distributed insoluble contaminants in suspension.
EP additives are important where $P/C > 0{,}15$ and the operating viscosity ν is lower than the rated viscosity ν_1.
High-temperature oils with superior non-deterioration properties are available for applications where the bearings are subjected to great thermal stressing. Some oils feature a favourable V-T behaviour, i.e. their viscosity varies less with temperature than the viscosity of normal oils; they are primarily used for applications where temperatures vary to some degree. For extremely high temperatures, synthetic oils oils (e.g. polyglycols or polyalphaolefins) are preferred to mineral oils because they are much more resistant to aging.
The suitability of oils for specific applications either has to be known from experience or determined in tests.

Methods of oil lubrication

Circulating oil lubrication is the lubricating method which enables – for the usual speed range of the roll neck bearings – not only safe lubrication but also provides cooling of the bearings and carries away contaminants and water from the bearing locations. In roll neck mountings it is provided as a cooling lubrication system for applications with

- energy losses within the bearing itself, i.e. with high loads and high speeds
- heating of the roll neck journals through external heat sources
- insufficient heat dissipation.

Oil injection lubrication, where the lubricant is injected directly into the bearing through lateral nozzles, is required where a circulating oil lubrication system is not sufficient to cool the bearing and the roll neck. Oil injection lubrication permits extremely high speeds.
Circulating oil lubrication and oil injection lubrication require some expenditure for inlets, outlets, pumps, oil reservoirs and, possibly, oil recoolers.
With **oil sump lubrication**, the small lateral spaces in the chocks can accommodate only a small amount of oil which, due to heavy stressing, deteriorates rapidly. Therefore the oil has to be changed frequently, or possibly non-aging synthetic oils have to be used.
Throwaway lubrication is frequently used for rolling mill bearings.
With **oil mist lubrication**, compressed air carries the atomized oil to the nozzle at the bearing where the oil particles form larger drops which are injected into the bearing. This small amount of oil adds to the oil sump which contributes to the bearing lubrication. It also ensures that during start-up of the bearing and during short malfunctions of the lubricating system all functional areas are adequately supplied with oil. With a horizontal shaft, the position of the oil drain holes in the chock is chosen such that the bottom rolling element is half immersed in the oil sump.
The constant overpressure created by the air current in the chock and the air escaping at the seals increase the effectiveness of the sealing. The escaping air usually contains some of the atomized oil which is harmful to the environment.
With **oil-air lubrication** the oil is intermittently fed to the lubricating pipe of the bearing via a metering unit and carried into the bearing by the air current. This oil is not atomized. Therefore high-viscosity transmission oils with EP additives can be used. As with oil mist lubrication, the air current increases the effectiveness of the sealing.
The volume of air can be selected within wide limits. For oil-air lubrication, an oil sump is also required.

Design of the lubricating system

Amount of grease

The bearings and housings should be greased as follows:

- The bearings should be packed to capacity with grease to make sure that all functional areas are supplied with grease.
- The housing space beside the bearings should be filled with grease to such an extent that the grease escaping from the bearings can be easily accommodated. In this way no excessive amount of grease will circulate through the bearing. Usually the free spaces in the chocks beside the bearings are just large enough to accommodate the grease escaping from the bearings; therefore these spaces do not have to be filled with grease if the bearings run at high speeds.
- Low-speed bearings ($n \times d_m < 50\,000$ min^{-1} × mm) and their housings should be packed with grease to capacity. The lubricant friction due to working is negligible.

Relubrication intervals (grease)

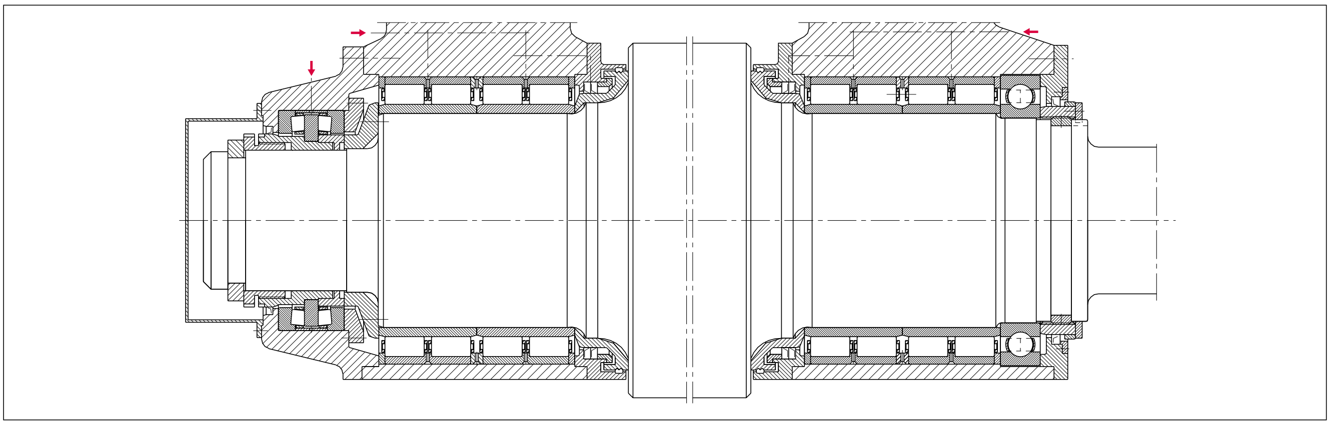
The exact relubrication or grease renewal intervals depend on the extent to which the grease has been stressed by bearing friction and speed. Bearing friction is a function of load and the result of the different kinematic conditions encountered in the various bearing types. Moreover – especially in roll neck applications – special thought should be given to the environmental conditions and the effectiveness of the sealing used. If the sealing is inadequate, moisture, splash water and mill scale would demand drastically reduced relubrication intervals.
Recommendations for relubrication intervals can be obtained by periodically examining the condition of grease and seals – preferably on the occasion of roll changes – to find out if contaminants have penetrated into the bearing.

Lubricant flow

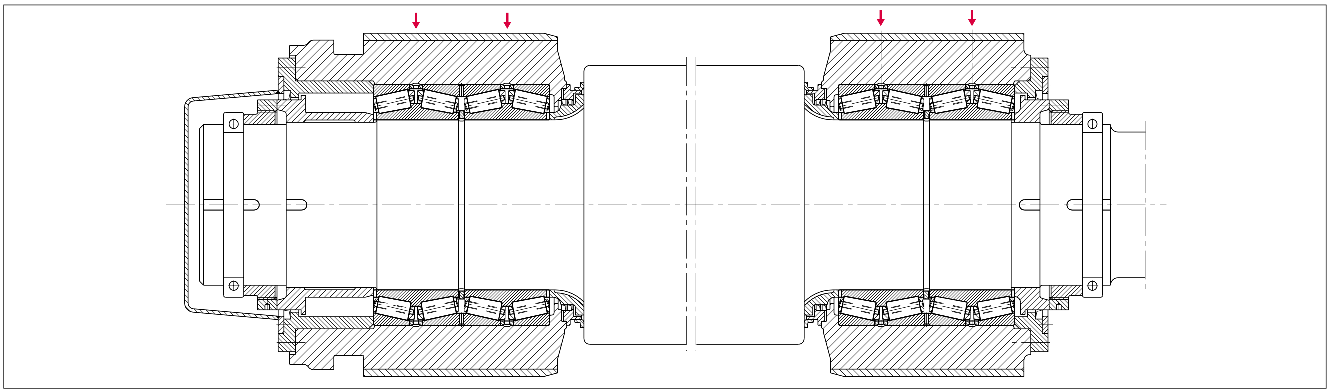
For effective lubrication, the grease or oil must safely reach the rolling and sliding surfaces. If grease is used, the excessive amount of grease must be able to escape. Overlubrication would cause an undesirable increase in churning action, creating a temperature rise with a resulting loss in grease lubricity. The seals must be effectively lubricated as well.

Grease Lubrication

Four-row roller bearings that support horizontal rolls should be supplied with lubricant at two points (fig. 44). Ball bearings that accommodate the thrust loads (fig. 44a, right) can be integrated into the general lubricating system but may also be lubricated separately. Tapered roller thrust bearings (fig. 44a, left) must be lubricated separately as their lubrication requirements are more demanding. Double-row angular contact ball bearings should also be lubricated separately. If the chocks are not removed for regrinding of the roll body (loose fit of the inner rings), the bearings must be regreased through the journal.
Sealed four-row tapered roller bearings are packed at the manufacturing plant to capacity with the grease best suited for a specific application. The right amount and distribution of grease within the bearing promise very long service lives. We recommend to provide drain holes on both sides of the bearing so that the bearing seals are exposed to moisture as little as possible.

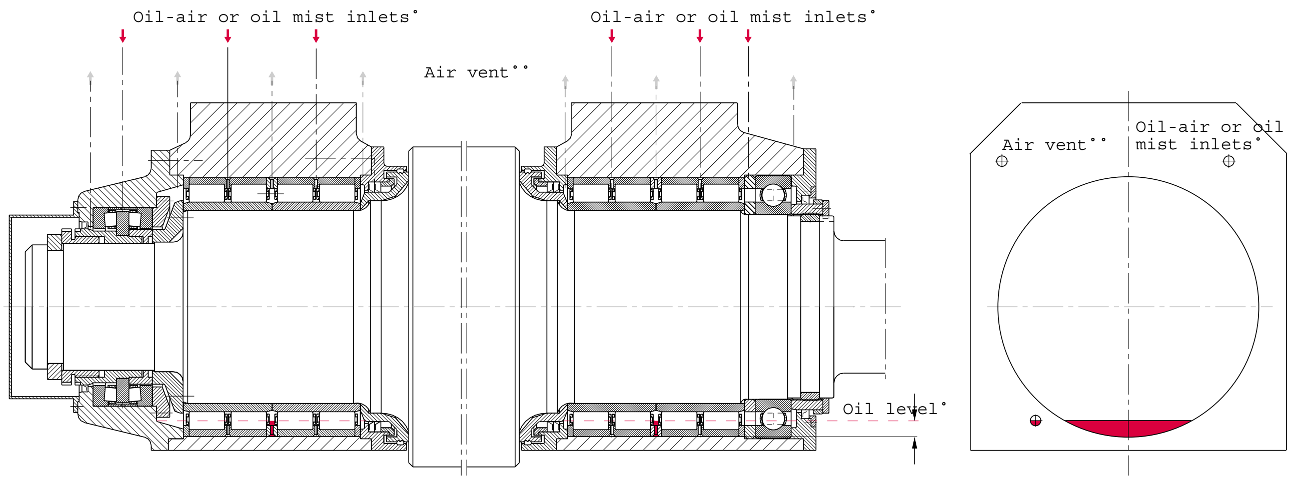


a) Roll neck bearing arrangement with four-row cylindrical roller bearings and thrust bearings

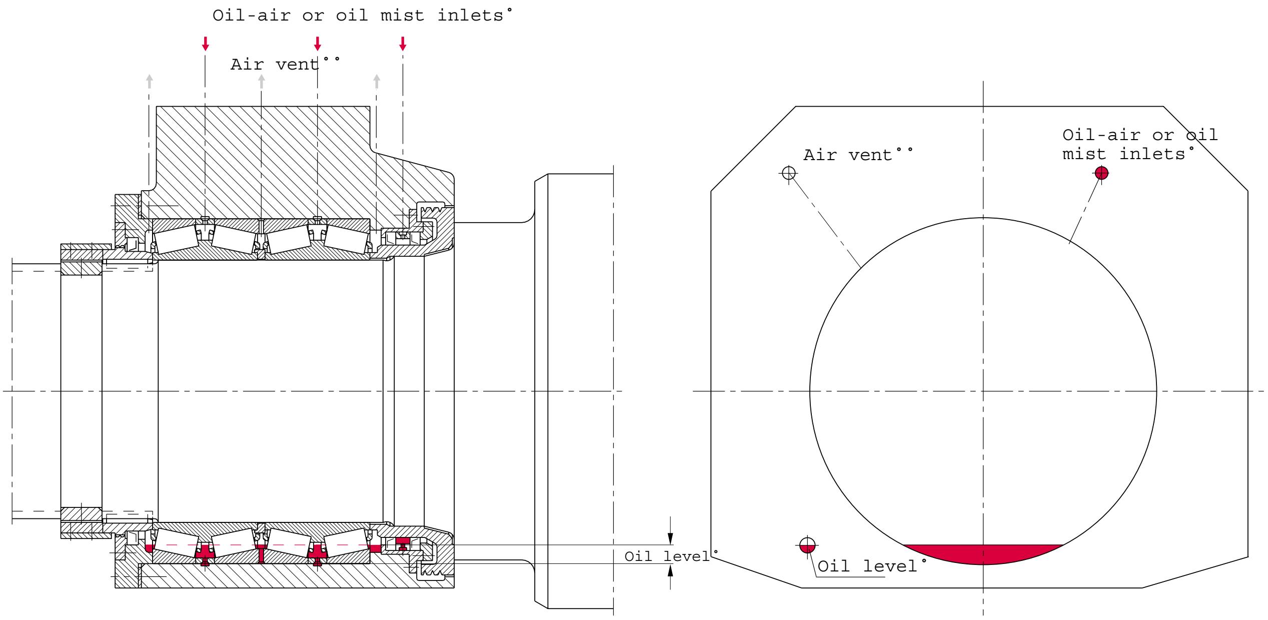


b) Roll neck bearing arrangement with four-row tapered roller bearings
44: Layout of lubrication system for four-row roller bearings (grease lubrication)

Oil mist lubrication, oil-air lubrication



45: Oil mist or oil-air lubrication system for chocks with four-row cylindrical roller bearings and thrust bearings

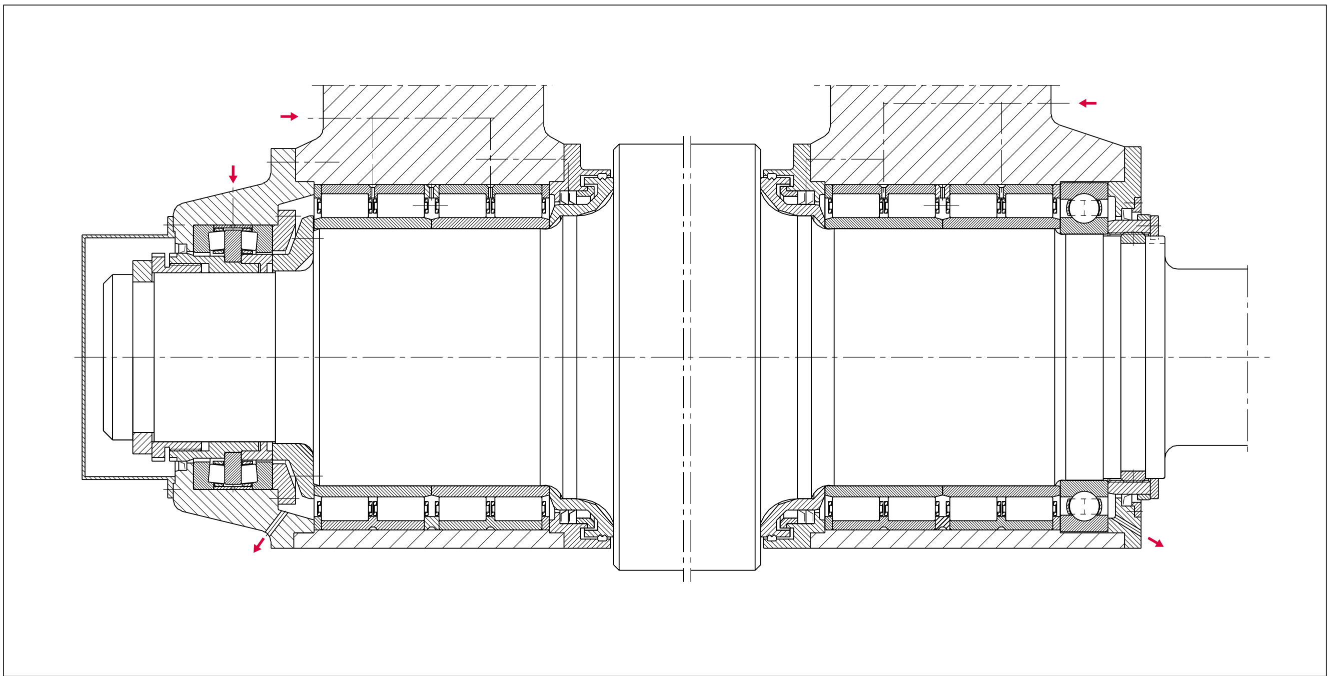


46: Oil mist or oil-air lubrication for a chock with a four-row tapered roller bearing

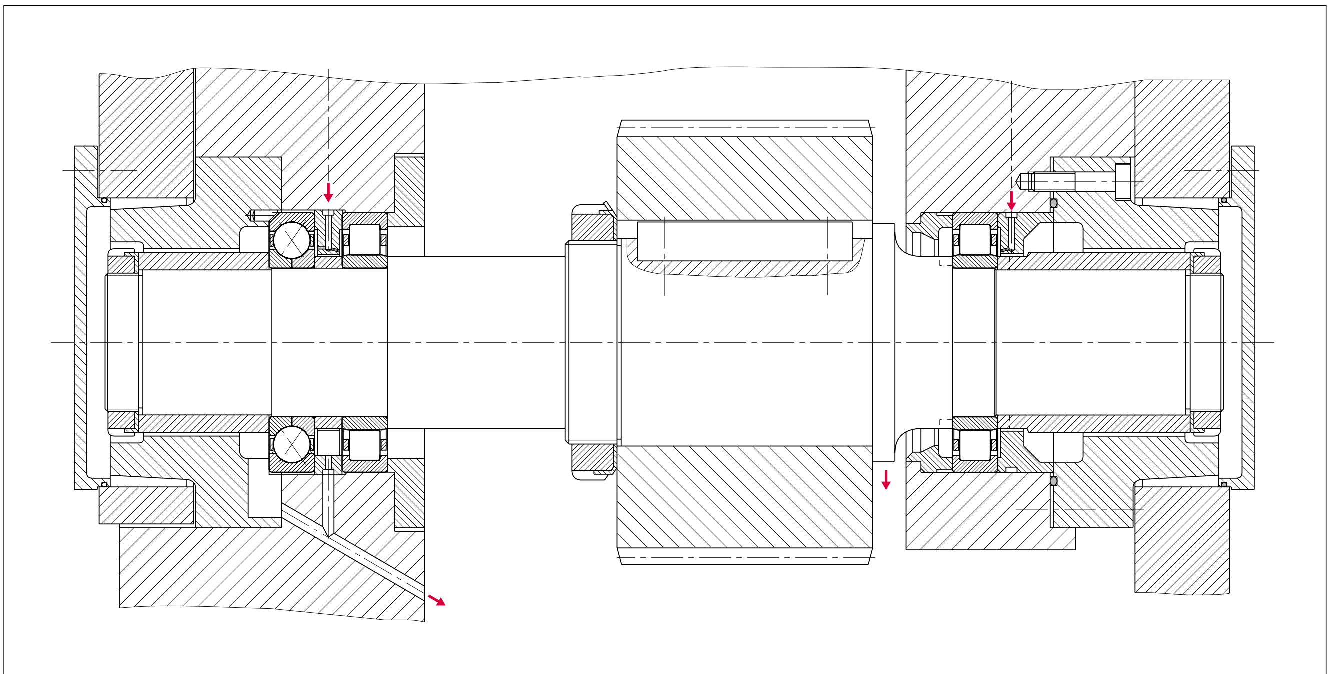
Lubrication

Design of the lubricating system

Circulating oil lubrication, oil injection lubrication



47: Oil inlet and oil outlet for circulating lubrication



48: Oil inlet and oil outlet for injection lubrication

Tolerances of roll neck bearings

Tolerances of roll neck bearings

Tolerances of metric radial and thrust bearings (normal tolerance)

Nominal dimensions mm				Tolerances µm Inner ring Δ_{dmp}		Outer ring Δ_{Dmp} Radial bearings		Thrust bearings		Inner and outer ring $\Delta_{Bs} = \Delta_{Cs}$	
over	50	to	80	0	−15	0	−13	0	−19	0	−150
over	80	to	120	0	−20	0	−15	0	−22	0	−200
over	120	to	150	0	−25	0	−18	0	−25	0	−250
over	150	to	180	0	−25	0	−25	0	−25	0	−250
over	180	to	250	0	−30	0	−30	0	−30	0	−300
over	250	to	315	0	−35	0	−35	0	−35	0	−350
over	315	to	400	0	−40	0	−40	0	−40	0	−400
over	400	to	500	0	−45	0	−45	0	−45	0	−450
over	500	to	630	0	−50	0	−50	0	−50	0	−500
over	630	to	800	0	−75	0	−75	0	−75	0	−750
over	800	to	1000	0	−100	0	−100	0	−100	0	−1000
over	1000	to	1250	0	−125	0	−125	0	−125	0	−1250
over	1250	to	1600	0	−160	0	−160	0	−160	0	−1600
over	1600	to	2000	0	−200	0	−200	0	−200	0	−2000

Tolerances of inch four-row tapered roller bearings (normal tolerance)

Nominal dimensions mm				Tolerances µm Cone Δ_{dmp}		Cup Δ_{Dmp}		Cup and cone $\Delta_{Bs} = \Delta_{Cs}$
over	76,2	to	304,8	0	+25	0	+25	±1524
over	304,8	to	609,6	0	+51	0	+51	±1524
over	609,6	to	914,4	0	+76	0	+76	±1524
over	914,4	to	1 219,2	0	+102	0	+102	±1524
over	1219,2			0	+127	0	+127	±1524

Tolerances of roll neck bearings · Surrounding parts

Tolerance symbols
DIN ISO 1132, DIN 620

Bore diameter

d Nominal bore diameter
d_s Single bore diameter

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

Single plane mean bore diameter
d_{psmax} Single plane maximum bore diameter
d_{psmin} Single plane minimum bore diameter
$\Delta_{dmp} = d_{mp} - d$
Single plane mean bore diameter deviation

Outside diameter

D Nominal outside diameter
D_s Single outside diameter

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

Single plane mean outside diameter
D_{psmax} Single plane maximum outside diameter
D_{psmin} Single plane minimum outside diameter
$\Delta_{Dmp} = D_{mp} - D$
Single plane mean outside diameter deviation

Width

B_s, C_s Single ring width (inner and outer ring)
$\Delta_{Bs} = B_s - B$, $\Delta_{Cs} = C_s - C$
Deviation of one single ring width (inner ring and outer ring) from the nominal value

Fits

Radial bearings

The inner rings of radial roll neck bearings are subjected to circumferential load during operation. They should therefore be tightly fitted on the roll neck. Due to mounting reasons, this is not possible with four-row tapered roller bearings with a cylindrical bore. Therefore a slide fit must be provided. The inner rings of spherical roller bearings and cylindrical roller bearings are slide-fitted if the rolling speed is low, and quick and easy extraction from the roll neck is desired.
The outer rings of radial bearings are subjected to point load. In this case no tight fit is required so that the rings may be slide-fitted in the chock. Axial location of the outer rings is provided by the end covers of the chocks.

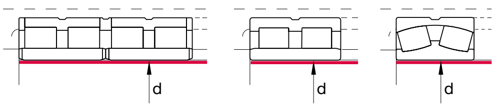
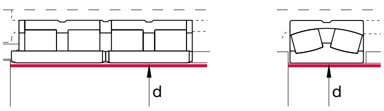
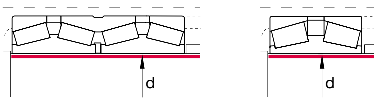
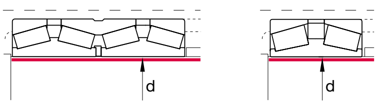
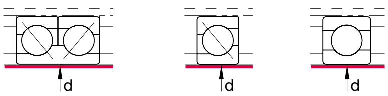
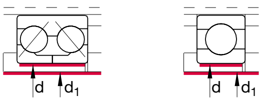
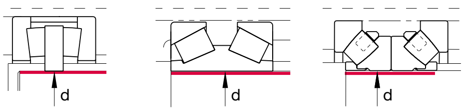
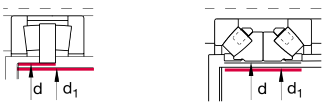
Thrust bearings

Bearings intended for axial location of the roll and location of the chocks are subjected to axial loads only so that the inner rings can be slide-fitted on the roll necks.
In some roll neck applications the thrust bearings are mounted on a sleeve for easier assembly. In this case a slightly tighter fit is advisable.
The housing washers of tapered roller thrust bearings are loosely fitted in the chocks. The outer rings of all other bearings serving to provide axial location must be radially relieved. Therefore, the housing bore must be significantly larger than the outside diameter of the outer rings.

Surrounding parts

Fits

49: Tolerance fields for roll necks and sleeves (bearing tolerances see page 36)

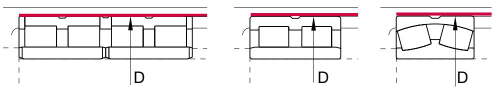
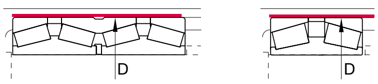
		Nominal diameter mm	Tolerance[1] mm
	Cylindrical roller bearings and spherical roller bearings (tight fit)	d < 200 d = 200...400 d > 400...630 > 630...800 > 800...1250 > 1250...1400 > 1400...1600	n6 p6/r6 +0,200...+0,260 +0,250...+0,330 +0,320...+0,420 +0,400...+0,550 +0,520...+0,650
	Cylindrical roller bearings and spherical roller bearings (slide fit)	d	e7
	Metric tapered roller bearings (slide fit)	d < 315 d = 315...630 > 630...800 > 800	−0,180...−0,230 −0,240...−0,300 −0,325...−0,410 −0,350...−0,450
	Inch tapered roller bearings (slide fit)	d = 101,6...127,0 > 127,0...152,4 > 152,4...203,2 > 203,2...304,8 > 304,8...609,6 > 609,6...914,4 > 914,4	−0,100...−0,125 −0,130...−0,155 −0,150...−0,175 −0,180...−0,205 −0,200...−0,249 −0,250...−0,334 −0,300...−0,400
	Angular contact ball bearings and deep groove ball bearings mounted on the roll neck	d	e7
	Angular contact ball bearings and deep groove ball bearings mounted on a sleeve	d d_1	k6 e9/H7
	Tapered roller thrust bearings, double-row tapered roller bearings (thrust bearings), and spherical roller thrust bearings mounted on the roll neck	d	e7
	Tapered roller thrust bearings, spherical roller thrust bearings mounted on a sleeve	d d_1	k6 e9/H7

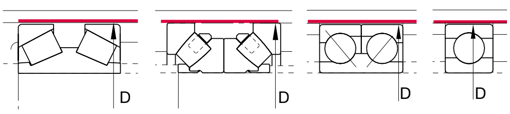
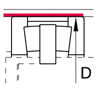
[1] For high speeds and bearings with a tapered bore, please contact FAG to determine the necessary tolerances of the surrounding parts.

Surrounding parts

Fits

50: Tolerance fields for chocks

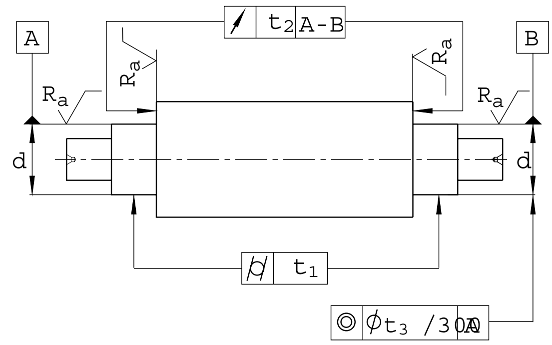
Radial bearings		Nominal diameter mm	Tolerance[1] mm
	Metric cylindrical roller bearings, spherical roller bearings and tapered roller bearings	D ≤ 800	H6
		D > 800	H7
	Inch tapered roller bearings	D ≤ 304,8	+0,055...+0,080
		> 304,8...609,6	+0,101...+0,150
		> 609,6...914,4	+0,156...+0,230
		> 914,4...1219,2	+0,202...+0,300
		> 1219,2	+0,257...+0,380

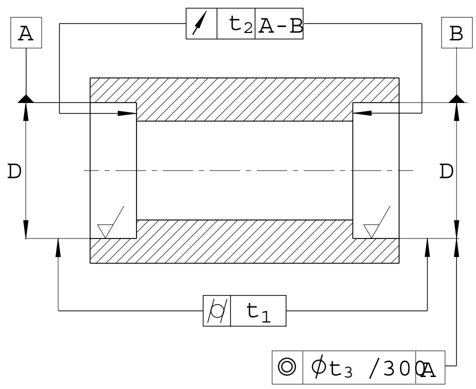
Thrust bearings		Nominal diameter mm	Tolerance[2] mm
	Tapered roller bearings, double-row (thrust bearings), spherical roller thrust bearings, angular contact ball bearings and deep groove ball bearings	D ≤ 500	+0,6...+0,8
		> 500...800	+0,8...+1,1
		> 800	+1,2...+1,5
	Tapered roller thrust bearings	D ≤ 800	H6
		D > 800	H7

[1] For bearings with a tapered bore, please contact FAG to determine the necessary tolerances of the surrounding parts.
[2] For high axial loads, please contact FAG to determine the necessary tolerances of the surrounding parts.

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

Recommendations for machining shafts, housing bores and surrounding parts (sleeves, covers etc.)





t_1 ⌭ Tolerance of cylindricity

t_2 ↗ Tolerance of runout

t_3 ◎ Coaxiality

Tolerance class of bearings	Bearing seat	Machining tolerance	Tolerance of cylindricity: Circumferential load t_1	Tolerance of cylindricity: Point load t_1	Tolerance of runout t_2	Coaxiality t_3
Normal, P6 X	Shaft	IT 6 (IT 5)	$\frac{IT\,4}{2}\left(\frac{IT\,3}{2}\right)$	$\frac{IT\,5}{2}\left(\frac{IT\,4}{2}\right)$	IT 4 (IT 3)	Half the dimensional tolerance
	Housing dia. Ø ≤ 150 mm	IT 6 (IT 7)	$\frac{IT\,4}{2}\left(\frac{IT\,3}{2}\right)$	$\frac{IT\,5}{2}\left(\frac{IT\,4}{2}\right)$	IT 4 (IT 3)	
	Housing dia. Ø > 150 mm	IT 7 (IT 6)	$\frac{IT\,5}{2}\left(\frac{IT\,4}{2}\right)$	$\frac{IT\,6}{2}\left(\frac{IT\,5}{2}\right)$	IT 5 (IT 4)	
P6	Shaft	IT 5	$\frac{IT\,3}{2}\left(\frac{IT\,2}{2}\right)$	$\frac{IT\,4}{2}\left(\frac{IT\,3}{2}\right)$	IT 3 (IT 2)	
	Housing	IT 6	$\frac{IT\,4}{2}\left(\frac{IT\,3}{2}\right)$	$\frac{IT\,5}{2}\left(\frac{IT\,4}{2}\right)$	IT 4 (IT 3)	

No values are indicated for ISO qualities IT4 and better where the nominal diameter is > 500 mm. In these cases half the machining tolerance is used.

Surrounding parts

Roughness of the bearing seats

Recommendations for the surface roughness of rolling bearing seats (the roughness values apply only to ground surfaces)

Tolerance class of bearings	Roughness	Nominal shaft diameter mm					Nominal housing bore mm				
	over		50	120	250	500		50	120	250	500
	to	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
	Average 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
	Average 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 grey-cast iron housings with turned fitting surfaces.
1) For more stringent requirements on running accuracy, the next higher surface quality must be selected.
2) GB: CLA (Centre Line Average Value); USA: AA (Arithmetic Average)

Roughness classes according to DIN ISO 1302

Roughness class		N1	N2	N3	N4	N5	N6	N7	N8	N9	N10	N11	N12
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 µinches	1	2	4	8	16	32	63	125	250	500	1000	2000

Surrounding parts

Tolerances of necks and chocks

Tolerances of necks and chocks

	Nominal shaft diameter mm												
over	50	65	80	100	120	140	160	180	200	225	250	280	315
to	65	80	100	120	140	160	180	200	225	250	280	315	355
	Roll neck tolerances 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

	Nominal chock diameter mm												
over	80	100	120	140	160	180	200	225	250	280	315	355	400
to	100	120	140	160	180	200	225	250	280	315	355	400	450
	Chock bore tolerances µm												
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

Surrounding parts

Tolerances of necks and chocks

Tolerances of necks and chocks

Nominal shaft diameter
mm

Roll neck tolerances
µm

over	355	400	450	500	560	630	710	800	900	1000	1120	1250	1400
to	400	450	500	560	630	710	800	900	1000	1120	1250	1400	1600
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

Nominal chock diameter
mm

Chock bore tolerances
µm

over	450	500	560	630	710	800	900	1000	1120	1250	1400	1600	1800
to	500	560	630	710	800	900	1000	1120	1250	1400	1600	1800	2000
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

Surrounding parts

Conditions required for loose-fitted inner rings

A loose fit of the inner rings requires a certain minimum roll neck hardness to limit wear of the roll necks. Roll neck wear is also considerably influenced by the lubricant film between inner ring bore and roll neck surface. If adequate lubrication of the roll neck is ensured over the entire period of operation, a roll neck hardness of 35 to 40 Shore C is sufficient.
If, for instance, the chocks are not, as usual, removed for the grinding of the rolls, the gap between the inner rings and the roll neck will not be repeatedly supplied with fresh grease. In such cases a separate roll neck lubricating system is provided, fig 51. FAG also urgently recommends such a roll neck lubricating system for sealed four-row tapered roller bearings if the bearings remain on the roll necks for a long time.

Grooves have to be provided in the abutment surfaces of parts adjacent to the inner ring. Through these grooves, grease will be supplied to this area and into the gap between inner ring and roll neck. Some bearing series are produced with such lubricating grooves in the inner ring faces so that no lubricating grooves have to be provided in the surrounding parts.

Chocks

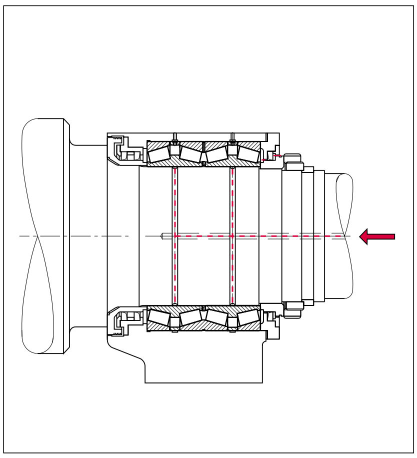
The rings of roll neck bearings are nearly always thin-walled. They must, therefore, be well supported; otherwise they would be unable to accommodate the high operating loads.
For efficient support of the bearing outer rings the chocks must be sufficiently rigid. Chocks made of cast steel with a minimum tensile strength of 450 N/mm² generally offer sufficient rigidity if their design is based on the following formulae:

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

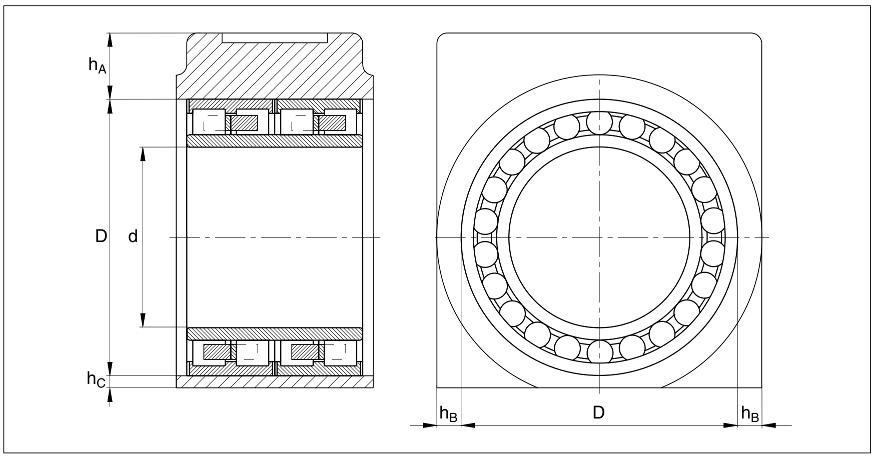
$$h_B = (0{,}7 \ldots 1{,}2)\ \frac{D - d}{2}$$

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

where h_A represents the upper, h_B the lateral and h_C the lower wall thickness of the chock in mm.
d represents the bearing bore in mm and D the bearing O.D. in mm (fig. 52). If the chock design is based on these formulae, and if the operating loads are not too high, the influence of chock deformation on the bearing stressing will usually remain within acceptable limits. For extreme loads and new designs it would be advisable to check the deformation of the chock and its effect on the bearing by calculation. This calculation can be effected quickly by means of a computer program developed by FAG.



51: Bearing arrangement with lubricating holes in the roll neck



52: Wall thicknesses of a chock

Surrounding Parts

Chocks

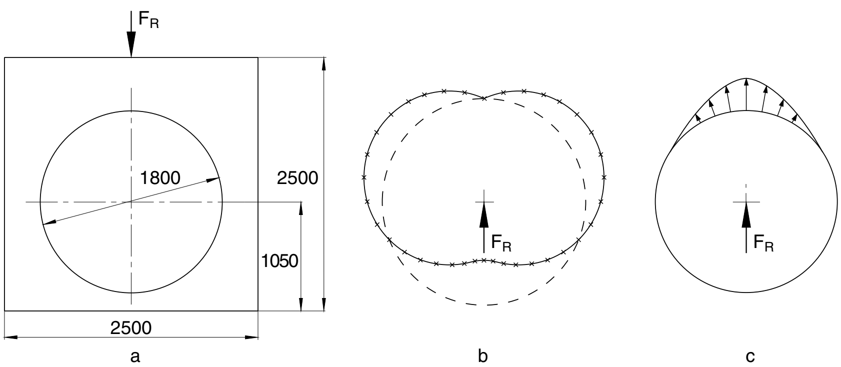
The deformation of the chock is determined by means of the strain-energy method, with the chock being regarded as a heavily curved closed beam with a variable cross section. The result of such a calculation is shown in fig. 53.
To ensure accurate running of the rolls and close gauge control of the rolled product, the chock clearance in the stand window should be small. On the other hand, there should be sufficient clearance to avoid jamming of the chocks at operating temperature. This applies not only to the chock located in the stand but also to the floating one. When specifying the clearance, the temperature gradient between chock and roll stand should be taken into account. At low rolling speeds, it is approx. 30 K, and at high rolling speeds approx. 50 K.

Stand windows and chock seats

A reasonable clearance for the chocks is obtained by adding up the tolerance field H9 and the heat expansion differential between chock and roll stand. Sometimes, however, mounting requirements necessitate a larger clearance.
The chocks seats in the stand windows and on the screw-down mechanism must be crowned; in this way the bore of the chock can align itself parallel to the roll neck, and the bearing can carry loads over its entire width even if it is not exactly adjusted and if roll deflections occur. The supporting areas should be hardened to resist flattening under these high loads. With multi-row bearings, the mill designer should make sure that the position of the screw-down mechanisms is dead in line with the centre of the radial bearing, as otherwise the roller rows would be unevenly loaded.

Coupling design, too, has an influence on the running behaviour of the rolls. Smooth running is, for instance, achieved by using couplings which are shrunk onto the drive-end neck of the roll.

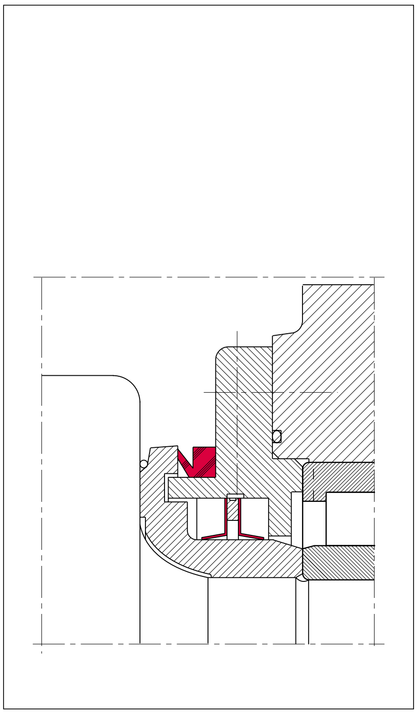
a) idealized form of the chock
b) chock deformation (exaggerated)
c) load distribution within the bearing



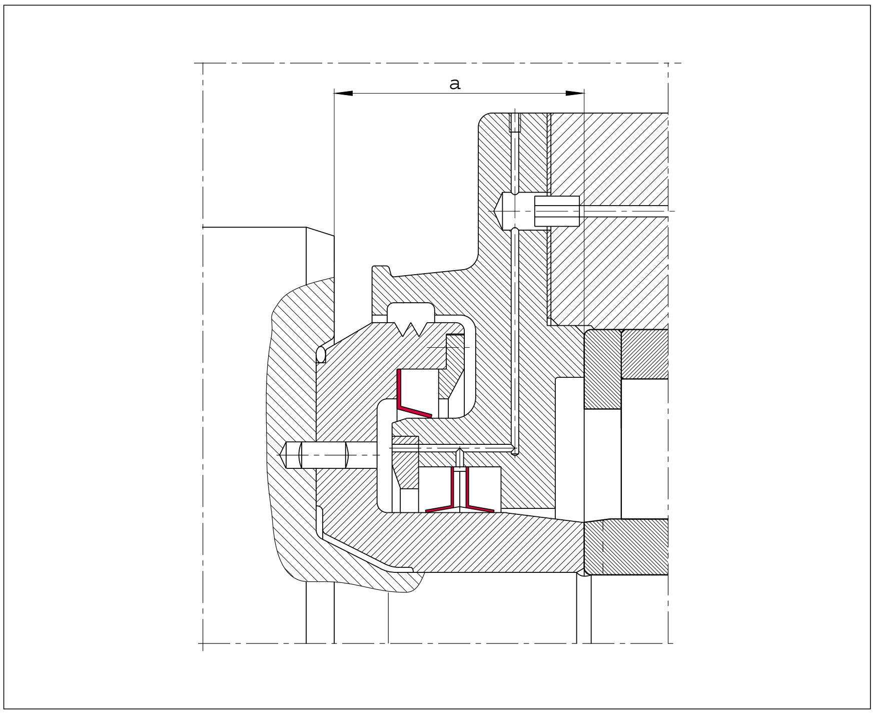
53: Graphic representation of the calculation results

Seals

The seals serve to prevent the penetration of cooling liquid, mill scale and other contaminants into the bearings; also, they prevent lubricant from escaping from the bearings.
The type of seal chosen for specific applications depends on the rolling speed, the required sealing efficiency, the lubricant and the operating temperature. The following illustrations show several types of seals used in rolling mill construction.
In hot-rolling stands, the bearings must be sealed against the ingress of water and mill scale. Fig. 54 shows an effective sealing for small roll stands. An axial sealing ring flings off any water coming near the bearing. A grease-filled labyrinth and two collar seals complete the sealing. During operation, grease is periodically replenished between the collar seals so that water and mill scale cannot penetrate into the bearing.
Additional collars further increase the efficiency of this sealing system. Additional protection against water is provided by grooves on the outside of the rotating labyrinth ring. The water collects in the groove provided in the stationary housing cover and is discharged through a hole in the bottom (fig. 55). The distance “a” between bearing and roll body is usually very short to keep the bending stresses in the roll neck as small as possible. The lateral space available for the sealing is then very limited so that the individual seals of combined sealing units must be positioned one above the other.
Often, collar seals cannot be used for the bearings in rolling stands of fine-section and wire mills due to the high speeds. In such cases seals built of piston rings (fig. 56) or face seals (fig. 57) are used. At high speeds, the lip of the face seals lifts off its contacting surface, thus preventing seal wear and frictional heat.



54: Sealing for small hot-rolling stands

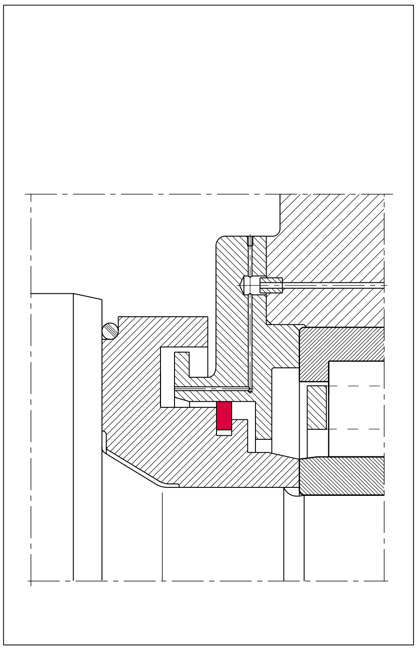


55: Sealing for large hot-rolling stands exposed to heavy contamination

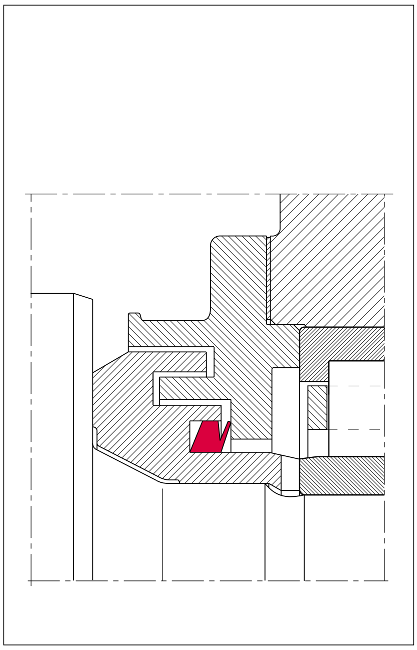
Surrounding parts

Seals

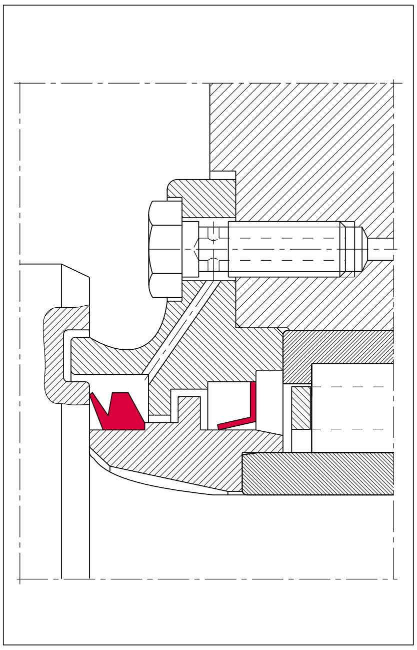
As already mentioned, in certain applications the bearing inner ring, and possibly the inner labyrinth ring as well, are slide-fitted on the roll neck (fig. 58). In this case not only the entrance to the bearing itself must be sealed – in fig. 58 by labyrinth and collar seal – but also the gap between the labyrinth ring and the roll neck. This gap is sealed by a face seal, fig. 58.
In hot-rolling mills with horizontal rolls, the seals at the drive end and at the operator's end are always identical. This is not possible with vertical rolls: since the openings of the labyrinth rings must be directed downwards so that the water can drain off, the labyrinth rings cannot be identical.
In hot-rolling stands, lubricant leaking from the bearings has no negative effect on the rolled material. In cold-rolling stands, however, any leakage from the bearings must be avoided since the lubricant might contaminate the material surface as well as the rolling fluid.
Therefore the inner collar seal is fitted with the lip directed towards the bearing (figs. 59 and 60 on page 48).
The surfaces on which the collar seal lips slide must be super finished. To avoid damage to the sealing lips during mounting, the sliding surfaces must be chamfered. The sealing lip must be lubricated regularly.
At the end not facing the roll, sealing is usually less difficult especially if a closed cover is provided. As a rule, one collar seal with the lip directed toward the bearing is sufficient. Where no end cover is provided, a second collar seal with its lip directed outward is often fitted.
The dimensions of the collar seals, face seals and piston ring type seals as well as the attainable sliding speed and the permissible ambient temperatures are indicated in the manufacturers' catalogues.



56: Sealing in a small rolling stand consisting of a piston ring and a labyrinth



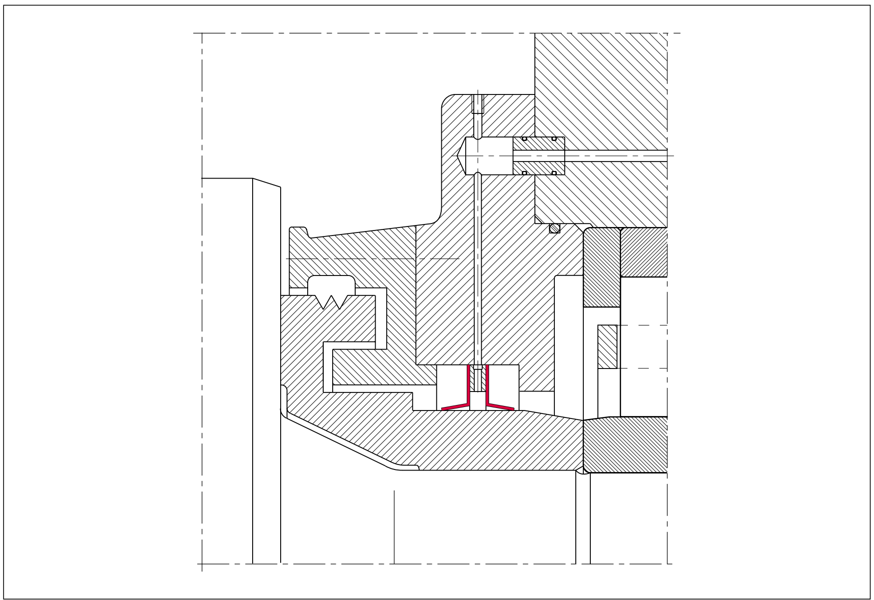
57: Sealing system for a high-speed roll neck bearing consisting of a face seal and a labyrinth



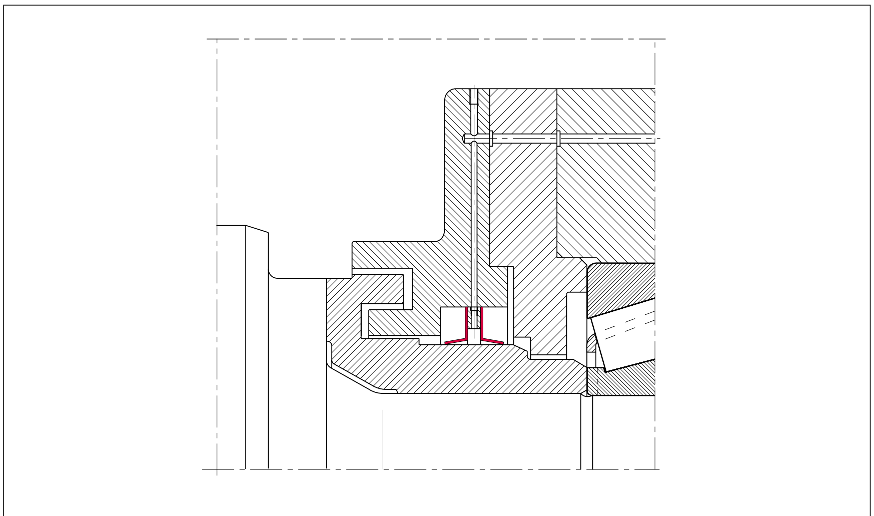
58: A face seal seals the gap between inner labyrinth ring and roll neck

Surrounding parts · Mounting and maintenance

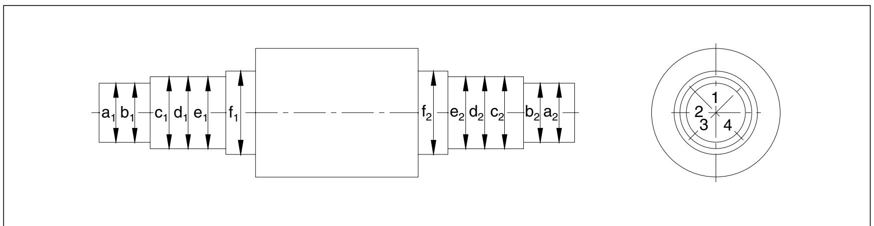
Preparations for mounting



59: In cold-rolling stands the sealing lip of the inner collar seal must be directed towards the bearing (back-up roll)



60: Symmetrically arranged collar seals for the work rolls of a cold-rolling stand



61: Measuring points for checking roll necks

Detailed mounting and dismounting instructions are contained in our Publication WL 80 100 "Mounting and Dismounting of Ball and Roller Bearings". In addition to the information given there, we will in the following explain a number of processes of importance for rolling mill operation.

Preparations for mounting

Prior to mounting the bearings, the surrounding parts – roll necks, chocks, sleeves, covers etc. – have to be checked for dimensional and geometrical accuracy against the design drawings.

Likewise, the specified surface finish of the roll necks, chocks and lateral abutting surfaces have to be checked. All sharp edges and burrs from machining must be removed.

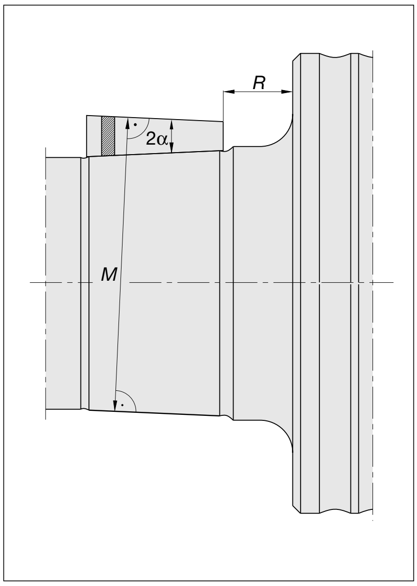
Checking of cylindrical roll necks

An exact check of the dimensional and geometrical accuracy of cylindrical roll necks necessitates four diameter readings to be taken (1-2-3-4) at each of the three positions (c-d-e) for the radial bearing seats and at each of the two positions (a-b) for the thrust bearing seats (fig. 61). The values measured are recorded in a measuring report.

Checking of tapered roll necks

For checking tapered roll necks (taper 1:12 or 1:30), we recommend to use the FAG taper measuring instrument MGK9205.
Tapered roll necks with a large diameter can be measured with a straight edge whose upper edge and lower edge form an angle, the taper angle of the journal = 2 α. If the upper edge of the straight edge is parallel to the generatrix of the roll neck, which is situated diametrically opposite to the straight edge, i.e. if M has the same magnitude at two measuring points, the taper angle of the roll neck is o.k. Moreover, the taper must have a certain ratio to a reference face, e.g. to the side face of the roll body.
FAG taper measuring instruments MGK9205 (fig. 63) are available in several sizes and designs, see TI No. WL 80-70.

A tolerance according to IT6 is permitted for the taper diameter. The permissible deviation of the taper angle from nominal is shown in table 65.



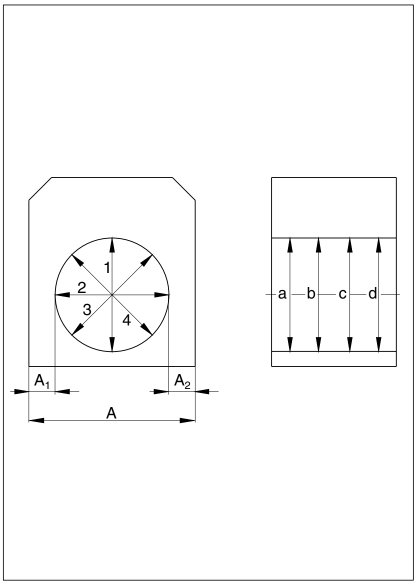
63a: Straight edge of the FAG Taper Measuring Instrument MGK9205

Checking of the chocks

The chocks should be measured across four diameters (1-2-3-4) in four lateral positions (a-b-c-d), fig. 64. Also, the position of the chock bore relative to the chock face (A_1 and A_2) has to be checked, if necessary with the locking ledges screwed on (for tolerances of form and position, see page 40).
The deviations from nominal should be recorded in a measuring report in the same way as those measured for the roll neck.
The surrounding parts also have to be checked, especially all dimensions leading to an axial preload. The surfaces of the surrounding parts must be square to the roll axis (for position and form tolerances, see page 40).
The lubricating holes have to be cleaned and subsequently checked by blowing compressed air through them.

62: Checking the roll neck of a small roll with an FAG dial indicator snap gauge.

63b: Measuring a tapered roll neck with an FAG Taper Measuring Instrument MGK9205; the stop touches the front datum surface.



64: Measuring points for checking the chock

Mounting and maintenance

Preparations for mounting

65: Permissible deviation of the 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	+8 +12,5	+10 +16	+12,5 +20	+16 +25	+20 +32	+25 +40	+32 +50	+40 +63
to AT7 (DIN 7178) ($2 \cdot t_6$)	0 0	0 0	0 0	0 0	0 0	0 0	0 0	0 0

The taper angle tolerance AT_D is measured perpendicular to the axis and is defined as the difference between the two diameters. When using FAG taper measuring instruments, the AT_D values must be halved (tolerance of angle inclination).
For bearings with nominal width values between two values indicated in the table, the taper angle tolerance AT_D must be determined by interpolation.

Example: Bearing with B = 90 mm

$$AT_D = \frac{\Delta \times AT_D}{\Delta B} \times B = \frac{25-16}{100-63} \times 90 = \frac{9}{37} \times 90 \mathrel{\hat{=}} 22\ \mu m\ (AT_D / 2 = t_6 \text{ amounts to } 0...+11\ \mu m)$$

Surface finish
The bearing seats must not be too rough so that a positive contact between mating parts and a satisfactory transmission of the high loads are ensured.

The bearing seat roughness of roll bearings should not exceed the values specified on page 41.

66: Measuring the bore of a large chock with an internal micrometer

67: Checking the surface roughness

Mounting and maintenance

Preparations for mounting · Mounting of four-row cylindrical roller bearings

Protection of the bearing seats

The formation of fretting corrosion in all seating areas where rolling bearings are fitted with a sliding fit (chock) or with a tight fit (roll neck) can be reduced by coating them with a lubricating paste with an anti-corrosion additive, e.g. with the FAG mounting paste Arcanol MOUNTING.PASTE. Prior to applying the coating, the seats have to be cleaned thoroughly. Such a thin layer of the paste should be applied that the normally bright surface is just dulled.

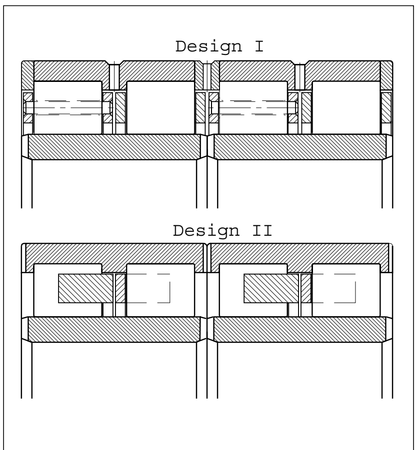
Preparation of bearings for mounting

The bearings should not be removed from their original packing until after the chocks and rolls and all accessories are ready for the mounting process. Normally the anti-corrosion oil does not have to be washed out as it does not react with any of the commonly used rolling bearing oils and greases. The performance, load carrying capacity and service life of a bearing depend not only on its quality but also on correct mounting. Therefore, the mounting should be done only by experienced fitters. FAG fitters are at the disposal of customers for initial mounting, for briefing the fitters at a customer's plant and for any other eventuality. In the following paragraphs we will explain how four-row cylindrical roller bearings, four-row tapered roller bearings and spherical roller bearings are to be mounted and dismounted in roll neck applications.

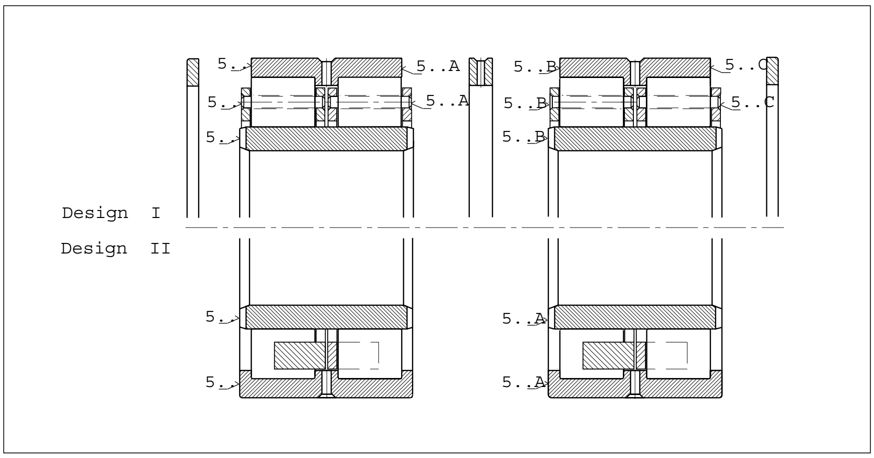
Mounting of four-row cylindrical roller bearings

Four-row cylindrical roller bearings differ by their design (fig. 68).

Design I:
Outer ring + roller set:
two outer rings, three loose lips, rollers and cages
Inner ring: two inner rings
Design II:
Outer ring + roller set: two outer rings with integral lips, rollers and cages
Inner ring: two inner rings
Customers can order either a complete bearing (e.g. FAG 524678) or the single components, e.g. outer ring with roller set (R524678) and inner ring (L524678). Each inner ring and outer ring is marked with the bearing code (e.g. 524678) and the serial number (e.g. 5), fig. 69. The inner rings at each bearing location must have the same serial number; the same applies to the outer rings (e.g. 5.. and 5..A). Inner rings of one serial number can, however, be matched with outer rings and roller sets with a different serial number.



68: Different designs I and II of four-row cylindrical roller bearings



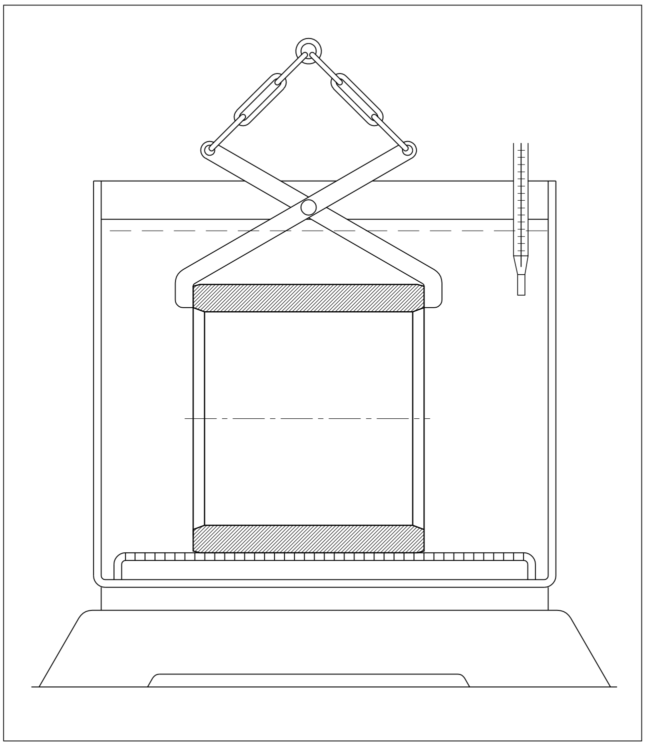
69: Marking and assembly of the components of four-row cylindrical roller bearings

Mounting and maintenance

First, the labyrinth ring or backing ring – depending on the interference – is heated to 150 to 170 °C and shrunk onto the roll neck. While the ring cools down, it must be axially clamped so that it abuts the roll body without a gap.

Mounting of the inner rings

The inner rings of cylindrical roller bearings with a cylindrical bore, which are tightly fitted on the roll neck, have to be heated to 80 to 100 °C prior to mounting. This is usually done in an oil bath (fig. 70) which ensures uniform heating. An excessively high heating temperature can be avoided by using a thermostat. After the rings are taken out of the oil bath, the oil in the bores and on the faces of the rings must be wiped off.
Frequently, the inner rings are dismounted by means of an induction heating device (see page 59). If such a device is available, it can also be used to heat the inner rings for mounting. After the heating process, smaller bearing rings are manually placed on the roll neck (fig. 71). For larger bearings, we recommend to use mounting tools, e.g. a pair of mounting grippers (fig. 70). The grippers always carry the ring with its axis in a horizontal position. If a rope is used to suspend the inner ring, it must, as a rule, be carefully adjusted in a horizontal position. After cooling down, the rolling bearing rings should abut the labyrinth ring without a gap. Neither should there be a gap between two adjacent labyrinth rings. Therefore the rings have to be axially clamped while they cool down. Smaller inner rings can also be mounted without a gap between them by pushing a so-called mounting sleeve against the ring face while the rings cool down.

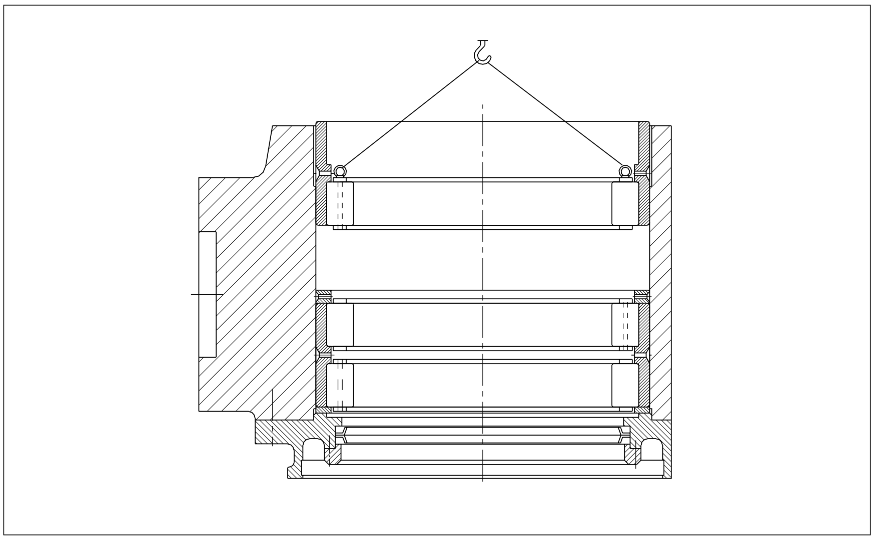


70: The inner ring of a cylindrical roller bearing is heated in an oil bath and then lifted out by means of a pair of grippers.

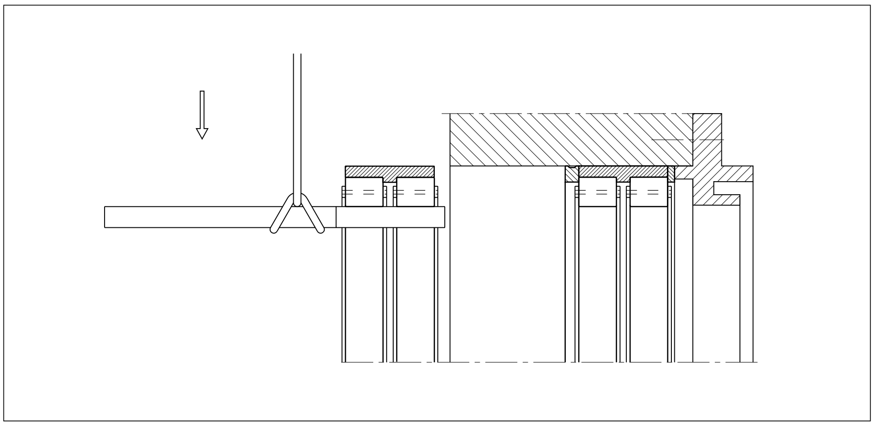
71: Fitting a small cylindrical roller bearing inner ring manually

Mounting of outer rings

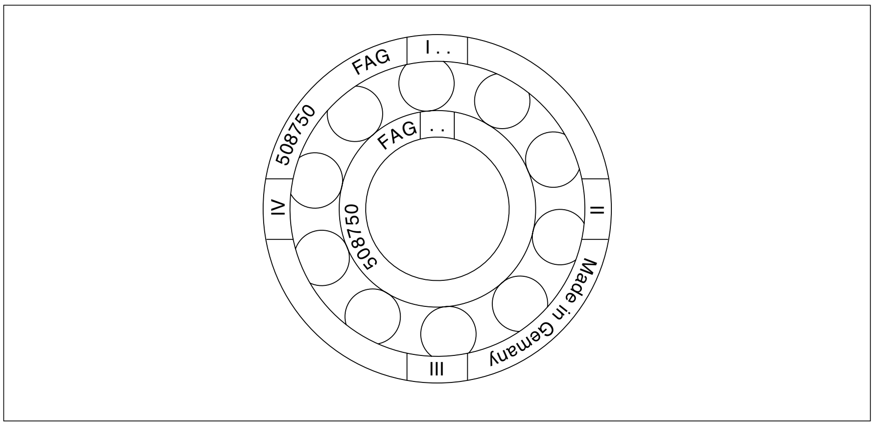
The outer rings of cylindrical roller bearings are slide-fitted in the chocks. Smaller outer rings can be inserted in the chock manually.
The outer rings or cages of larger bearings are usually provided with threaded holes for eye bolts so they can be inserted into the chock more easily (fig. 72).
If very large bearings have to be mounted with their axis in a horizontal position, the rings can, for instance, be inserted into the chock on a beam suspended in ropes (fig. 73).
The outer ring faces are divided into four zones which are marked I, II, III and IV (fig. 74). On initial mounting the outer rings are positioned such that the load is acting on zone I. The load zones of all outer rings should be positioned in the same direction.
We recommend to check the bearings thoroughly after 1000 to 1200 hours in operation. Then the outer rings should be rotated inside the chock by 180° to load zone III and, the next times, to load zone II and then IV.



72: Mounting of a cylindrical roller bearing outer ring by means of a crane



73: Mounting of a cylindrical roller bearing outer ring by means of a beam



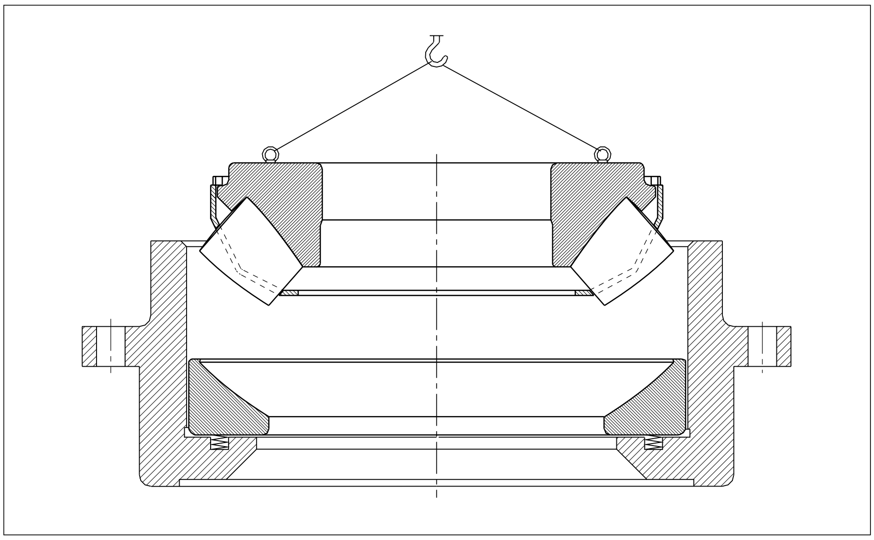
74: The load zones of a four-row cylindrical roller bearing are marked on the outer ring.

Mounting of thrust bearings

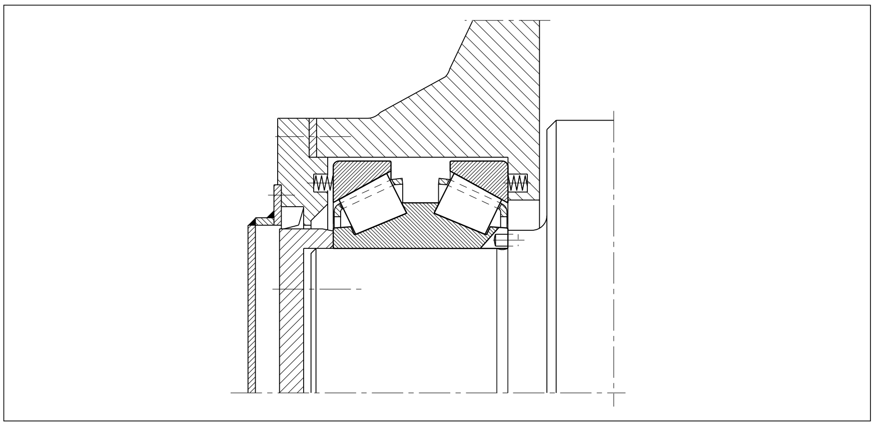
Angular contact ball bearings and deep groove ball bearings used as thrust bearings must not be loaded radially. Therefore the chock bore is usually 0,6...1,5 mm larger than the bearing outside diameter.
This may produce a radial displacement of the outer ring so that only the upper balls carry the thrust load. To prevent this, the fitter should bring the rolls, complete with chocks and radial bearings, into working position.
In this position, the center lines of the radial bearing inner rings are offset radially in relation to the outer ring centre line by half the radial clearance. The thrust bearings are then installed in this position.
To enable the outer ring to radially align under axial load, the cover bolts are tightened only slightly and then locked in this position.
Spherical roller thrust bearings are axially preloaded by means of springs; they are used primarily in work rolls. There must be sufficient radial and axial clearance between outer ring and housing (fig. 75).
The cups of double-row tapered roller bearings with a large contact angle are also axially adjusted by means of springs (fig. 76).

Mounting of pre-assembled chocks on the roll necks

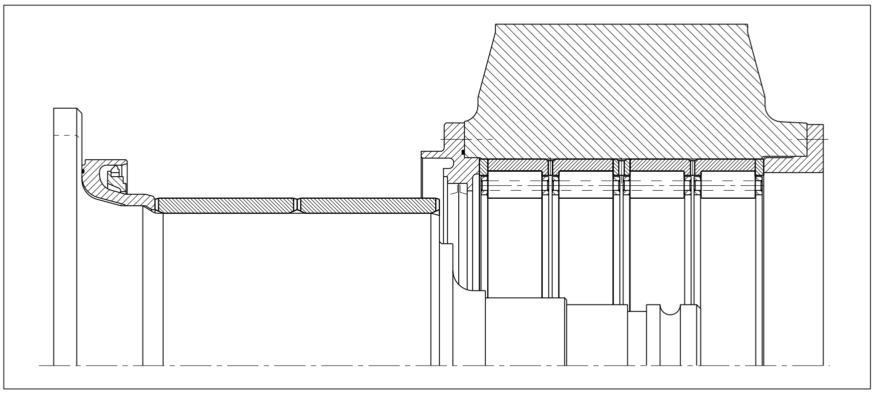
Once the labyrinth ring and the inner rings have been shrink-fitted onto the roll neck, the pre-assembled chock can be pushed onto the roll neck (fig. 77).



75: Mounting of thrust bearing components in the housing

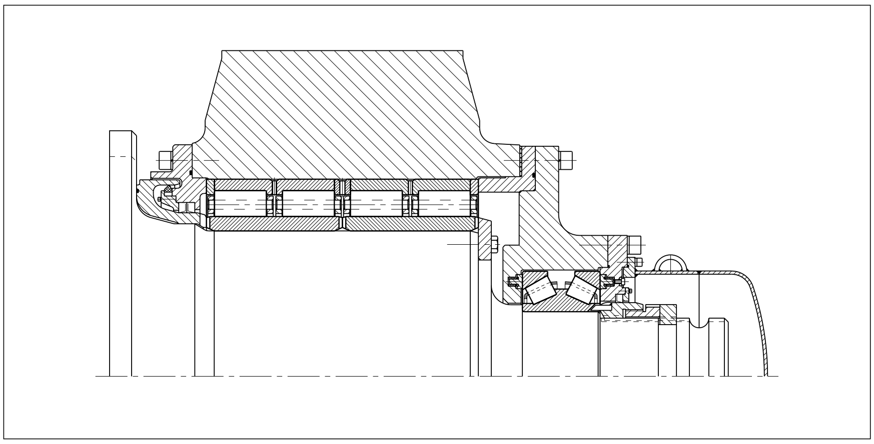


76: The cups of double-row tapered roller bearings are adjusted by means of springs.

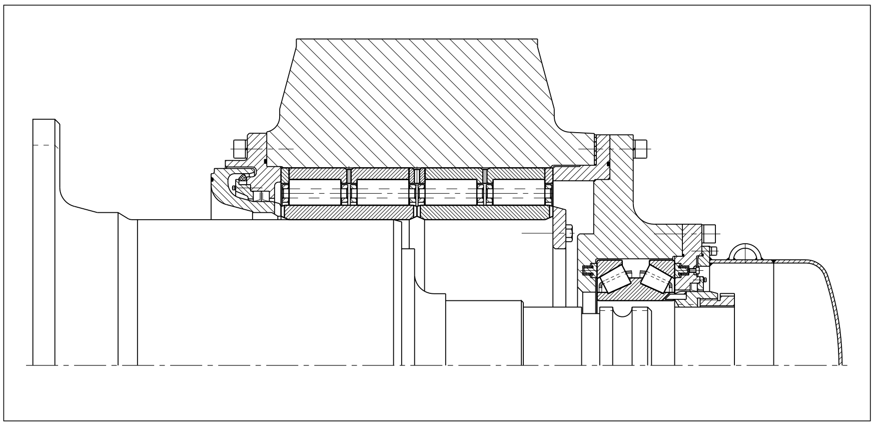


77: The pre-assembled chock is pushed onto the roll neck.
The loose lip of the cylindrical roller bearing outer ring is temporarily retained by angle pieces.

If the inner rings are to be fitted loosely on the roll neck, their bores have to be greased or oiled before mounting.
The chocks, complete with outer rings and thrust bearings, are usually carried to the roll neck by means of a crane and aligned as closely as possible with the roll neck so that the chocks can be pushed easily onto the neck. In the process, no score marks must be produced on the rollers and inner rings.
The inner rings or the sleeves on which the thrust bearings are mounted must be clamped by means of a locknut so that they cannot creep during operation, causing wear. The locknut must be secured.



78: Completely assembled chock



79: When changing the chock, the complete unit is replaced.

Dismounting

After removing the parts holding the thrust bearings on the roll, the chocks can be withdrawn from the roll neck as complete units.
When the rolls are replaced, the chock can be mounted onto the new roll after the inner rings have been mounted.
Dismounting of the different bearing components for inspection is effected in the reverse order of mounting, using the same equipment.
Withdrawal of the tightly fitted inner rings from the roll neck requires special equipment.
FAG induction heating devices (see page 59) have proved suitable for this purpose. In some cases the inner rings are extracted hydraulically. But there may be problems, especially with large bearings, if the fitting surfaces are damaged by cold welding or fretting corrosion. In exceptional cases the inner rings can also be heated with a gas burner (page 60).

Loose fit of the inner rings

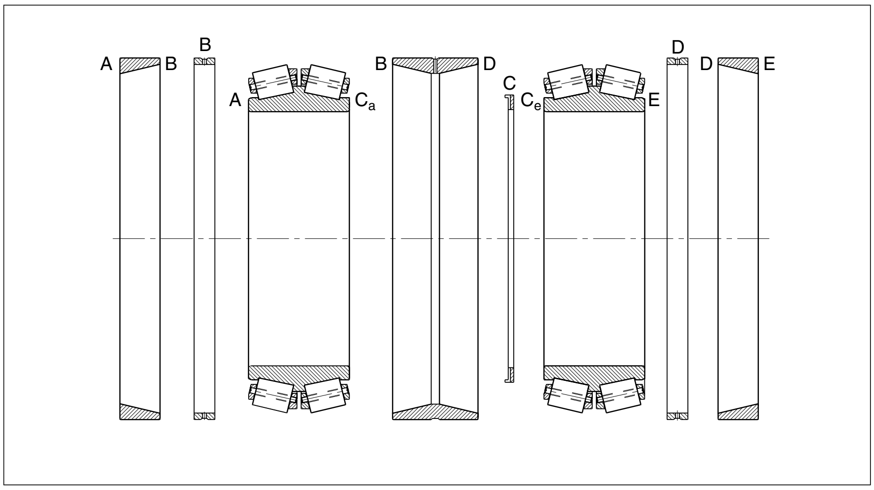
In section or light-section mills with a frequently changing rolling programme, the inner rings are sometimes slide-fitted onto the roll necks. The labyrinth ring facing the roll body is also fitted loosely. The labyrinth cover is designed such that, on removal of the bearing rings, the labyrinth ring is removed as well, serving as axial guidance for the inner rings. Thus the components of the bearing arrangement are held together as a unit (fig. 79).

Mounting of four-row tapered roller bearings

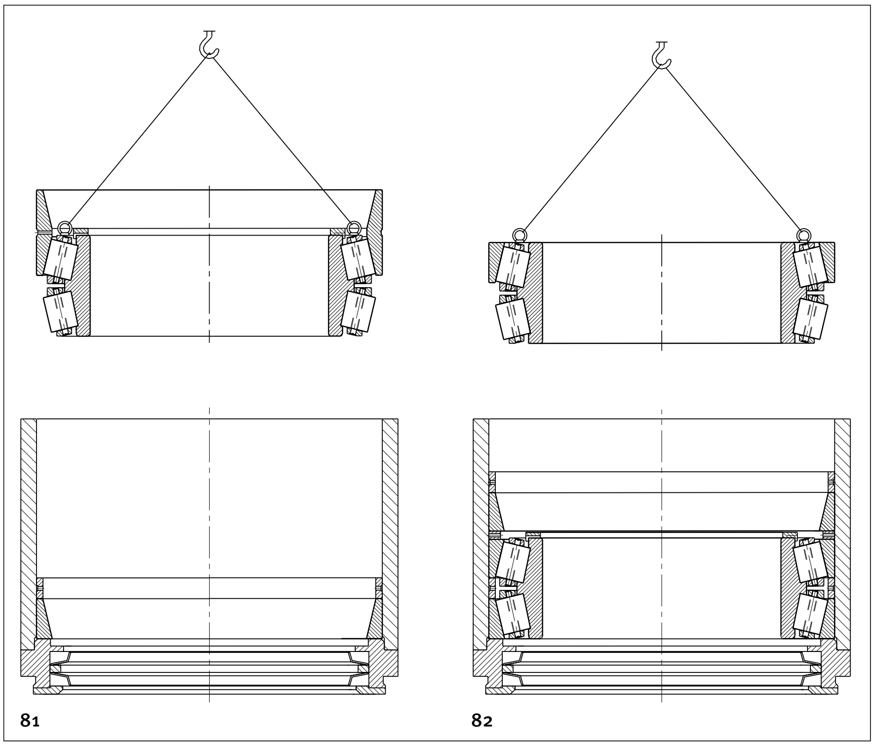
The components of four-row tapered roller bearings are marked as follows: bearing code, FAG logo, serial number and letters to ensure the correct mounting order. Fig. 80 shows how the sides of the bearing rings are marked.
The widths of the spacer rings B, D and C have been machined such that the correct axial clearance is obtained. The ring width and axial clearance are indicated on the spacers.
Like the outer rings of four-row cylindrical roller bearings, the cups are divided into four zones marked I, II, III and IV (fig. 74).

Mounting

Four-row tapered roller bearings are mounted vertically. First, the narrow cup marked AB is inserted into the chock with its load zone I in the direction of the load. Then the other components are inserted in the order shown in fig. 80. The load zones I of all cups must be turned in the direction of the load.
The cages of large four-row tapered roller bearings feature threaded holes for eye bolts (only designed for transport purposes). They facilitate the handling and mounting of the bearing components (figs. 81 and 82).
After all bearing components have been mounted, the cover bolts are tightened slightly, the seal not yet being in place (fig. 83).



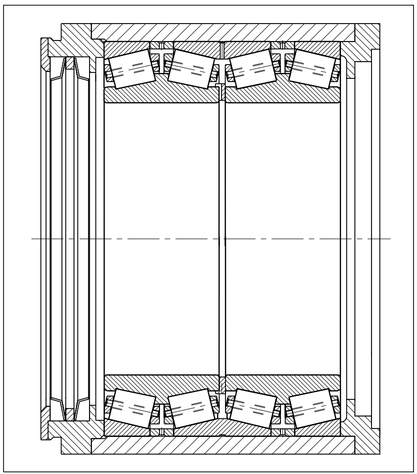
80: Sequence of assembly of the bearing components



81 and 82: The bearing components are inserted into the chock

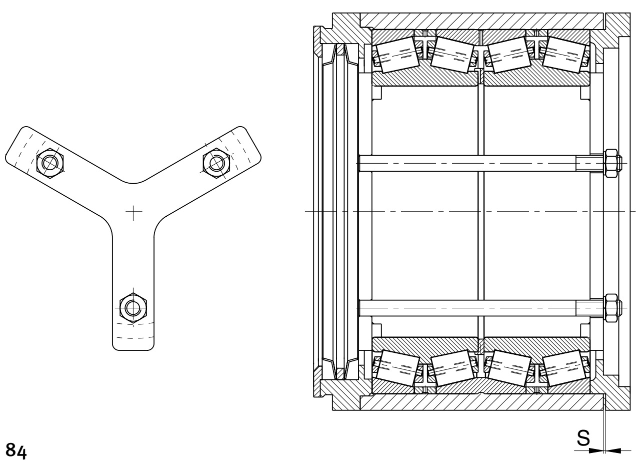
Mounting and maintenance

Mounting of four-row tapered roller bearings



83: The cover bolts of the chock are tightened slightly, and the chock is turned into a horizontal position.

Then the chock is turned over so that the bearing axis is horizontal. Centering pieces are attached to the cone's outer faces and clamped with tie rods (fig. 84).
While constantly turning the inner rings, the nuts at the tie rods and the cover bolts are tightened evenly.
A feeler gauge is used to check if cones and spacers abut each other without a gap. Then the gap S between chock and cover is measured, and a seal of the required width (S + x) inserted.
The magnitude of x required for a safe preload depends on the type of seal used and is determined by the rolling mill manufacturer. Once the cups and the cover have been clamped tightly, the centering pieces and tie rods can be removed.
Experienced fitters can do without centering pieces and tie rods.
While inserting the cones into the vertical chock, they constantly rotate the cones until the tapered rollers positively contact the guiding lips. Then they insert the radial seals into the cover.
The cone bores are greased or oiled. After the labyrinth ring has been shrunk on, the chock is mounted onto the roll neck.
The bearing is axially clamped with the locknut until it abuts the labyrinth ring without a gap. While tightening the nut, the chock should be turned several times to the left and right. Then the nut is loosened again until there is a gap of 0,2 to 0,4 mm between cone and nut. With a thread pitch of 3 mm, this distance corresponds to one tenth of a nut turn.
It is advisable to grease the bearing only after mounting to avoid contamination of the grease. This is best done by means of a grease gun. If no grease gun is available, the roller-and-cage assemblies have to be greased manually prior to mounting.
Where rolling speeds are very high, the chocks must not be completely filled with grease. Please ask FAG for information on the appropriate amount of grease for specific applications.



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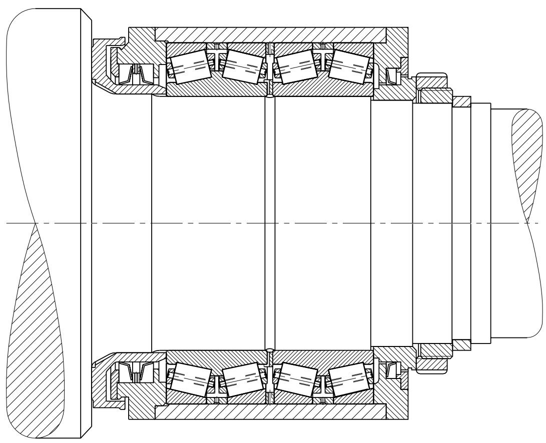


85

84: The cups are clamped together while the cones are constantly rotated.
85: The mounted chock

Dismounting

If the chock is to be mounted onto a different roll on the occasion of roll replacement, all the fitter needs to do is remove the nut, withdraw the complete chock from the neck and mount it onto the new roll. If the bearings have to be dismounted for maintenance and inspection, dismounting is effected in the reverse order of mounting. Double-row tapered roller bearings can be dismounted in the same way.

Maintenance

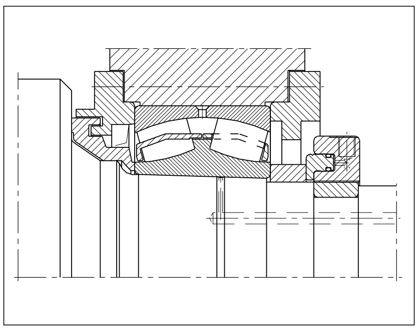
After some time in operation, the axial clearance of the four-row tapered roller bearings increases due to wear of the running surfaces. It is therefore necessary to check the axial clearance from time to time.
If the axial clearance is too large the spacers must be reground. The corrected axial clearance should be somewhat larger than the original axial clearance. More details are given in our "Mounting and maintenance instructions for four-row tapered roller bearings".

Mounting of spherical roller bearings

Spherical roller bearing inner rings are installed in rolling stands with a tight or loose fit.
If the inner ring may have a loose fit, mounting is easy. First, the bearings are inserted into the chock and the lateral covers are screwed on. Prior to mounting the chocks with the bearings onto the roll necks, the inner ring bores should be greased. The mounting process is made easier by using a mounting sleeve.
Since the inner rings revolve on the roll neck, a small amount of clearance must be provided between the lateral abutment surfaces. This axial clearance is best obtained by first tightening the nut and then loosening it again as with tapered roller bearings. In this position the nut is secured.
If a tight fit of the spherical roller bearing inner rings is required, bearings with a tapered bore are usually used.
On the occasion of roll replacement, the bearing can be mounted onto the new roll provided that the tapered roll necks and the width of the labyrinth rings have very close tolerances.

Mounting of tapered-bore spherical roller bearings

The greased tapered roller bearing is inserted into the chock and clamped. The chock with the spherical roller bearing is pushed onto the roll neck until it abuts the roll neck. Then the chock is pushed even further onto the roll neck using the hydraulic method. For this purpose the roll necks must have oil grooves and oil ducts. The chock is best pressed up onto the roll neck by means of a hydraulic nut. Details on hydraulic mounting and hydraulic nuts are given in the FAG publications WL 80 102 and TI No. 80-57.
The spherical roller bearing is pushed onto the roll neck until it abuts the labyrinth ring (fig. 86). To ensure that the specified drive-up distance is observed, the width of the labyrinth ring must be exactly adapted to the actual diameter of the tapered roll neck.
Then the hydraulic nut is removed and the locknut placed on the roll neck, tightened and secured.



86: Mounting of a spherical roller bearing by means of a hydraulic nut.

Dismounting of tapered-bore spherical roller bearings

The locknut is loosened by a few turns, the distance corresponding at least to the drive-up distance.
If now oil is pressed between the fitting surfaces, the bearing will suddenly come off as soon as an uninterrupted oil film has formed.
After removing the locknut, the chock with the spherical roller bearing can be removed from the roll neck and mounted onto another roll neck.

Methods for the mounting and dismounting of cylindrical roller bearing inner rings (tight fit)

Induction heating*)

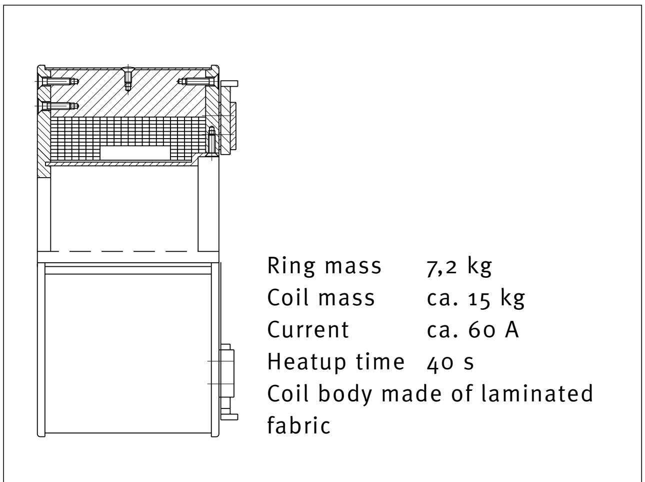
To release the tight fit of cylindrical roller bearing inner rings, they must be quickly heated up to 60 to 80 K above ambient temperature, that is to 80 to 100 °C for a roll temperature of 20 °C. During this process the roll neck should heat up as little as possible to obtain sufficient clearance between the inner ring and the roll neck to remove the inner ring.
Induction heating equipment operating at 50 Hz (A.C.) has proved suitable for mounting and dismounting medium-sized and large cylindrical roller bearing inner rings. With the usual wall thickness of the inner rings, the roll neck temperature increases – depending on the roll neck volume – by only 5 to 10 K while the inner rings reach temperatures of 80 to 100 °C. This temperature must be reached – and is reached by FAG equipment – within 0,5 to 1,5 minutes for small and medium-sized bearings and within 2,5 to 5 minutes for larger bearings.

*) see FAG Publ. No. WL 80 107 "Induction Heating Equipment"

FAG induction coils for 400 V and low-voltage induction coils

Mounting equipment should not be too complex and expensive since it is used only rarely. For the occasional mounting and dismounting of small and medium-sized cylindrical roller bearing inner rings with diameters of up to ca. 200 mm, a 400 V device is the best solution (fig. 87).

For large bearing rings, an induction coil which is connected directly to the 400 V mains is too impractical; it weighs several times as much as the parts to be mounted. For regular mounting of large and medium-sized rings it is advisable to operate induction heating equipment on low voltage (fig. 88). A transformer is connected between two phases of the 400 V mains and the induction coil; the secondary circuit of the transformer must be adjustable between 20 and 40 V.



87: Diagrammatic view of a pedal switch induction coil (400 V) for cylindrical roller bearing inner rings with a bore diameter of 130 mm

88: FAG low-voltage induction coil with transformer

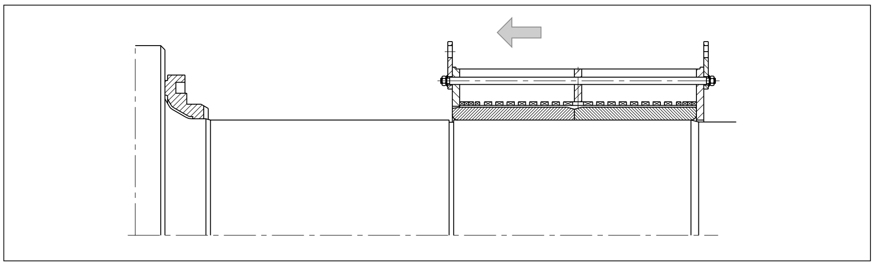
Mounting and maintenance

Methods for the mounting and dismounting of cylindrical roller bearing inner rings

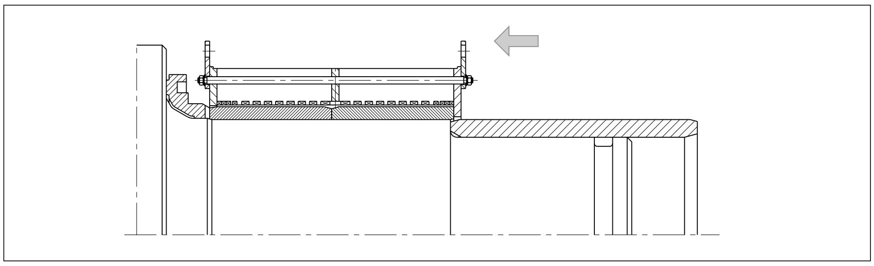
Using low-voltage equipment offers technical and economic advantages. The electric coil can be cooled with water so that it does not heat up. Thus the copper winding can be higher loaded. The device is lighter and, due to the improved electric coupling of the single or double layer coil, it offers an increased efficiency.
The devices can be used to heat inner rings for mounting and dismounting.
When shrink-fitting the inner rings onto the roll neck, it is advisable to first put them onto the neck end, heat them up to mounting temperature and then push them onto their seat together with the induction coil (fig. 89). If the roll neck end is not shaped to guide the rings, we recommend to use a mounting sleeve (fig. 90).
The bearing rings and roll necks are magnetized by the induction heating process; they must be demagnetized after mounting. This can also be done by means of the induction coil. The induction coil is pulled over the mounted part with the current switched on and then slowly removed to a distance of 1 to 2 meters from the parts. As the distance increases, the effect of the magnetic field decreases until its influence is so feeble that the parts are sufficiently demagnetized.

Heating with gas burners

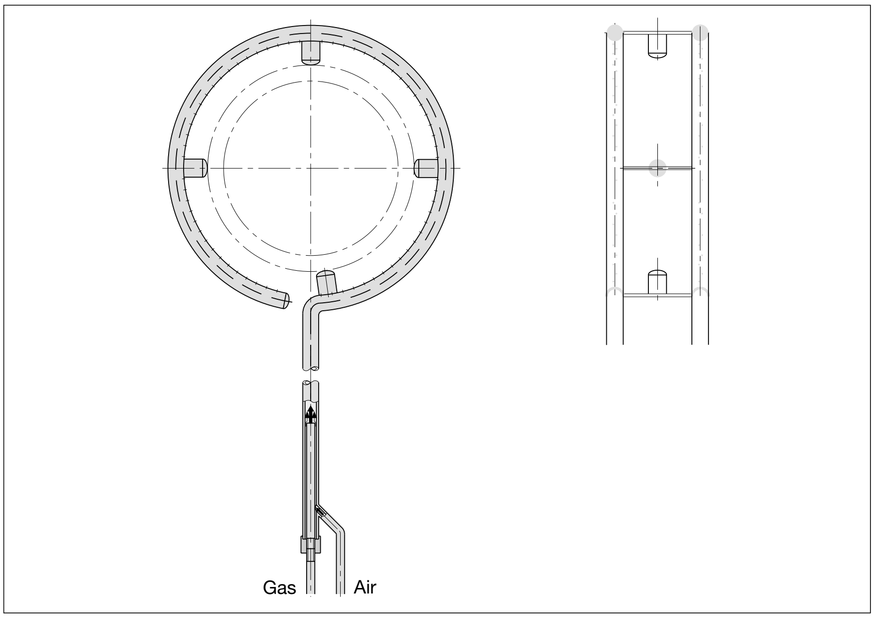
In cases where the rings cannot be dismounted by means of induction coils or the hydraulic method, only one possibility is left: heating by flame. This method should only be employed as an emergency measure. Ring burners (fig. 91) have proved to be an acceptable solution.
The burner should be positioned ca. 50 mm from the ring surface. For the usual gas pressure the diameter of the burner jets is 2 mm. The jets are spaced over the circumference 25 mm apart. The temperature and length of the flames are adjusted by the addition of air. Guiding pieces ensure a



89: Mounting of two inner rings by means of an induction coil. The device is also used for dismounting.



90: The rings are guided on a mounting sleeve.



91: Gas ring burner

concentric position of the ring burner and an even heating of the bearing rings. The ring burner must be moved back and forth across the ring in axial direction to ensure even heating of the rings.
In rare cases a welding torch is used, which renders the bearing rings unservicable as temperatures of more than 300 °C are reached. Then fluorinated materials, e.g. Viton® seals, can release gases and vapours which are detrimental to health. Please observe the relevant safety data sheet.

Mounting facilities for roll couplings and labyrinth rings

Induction heating of roll couplings

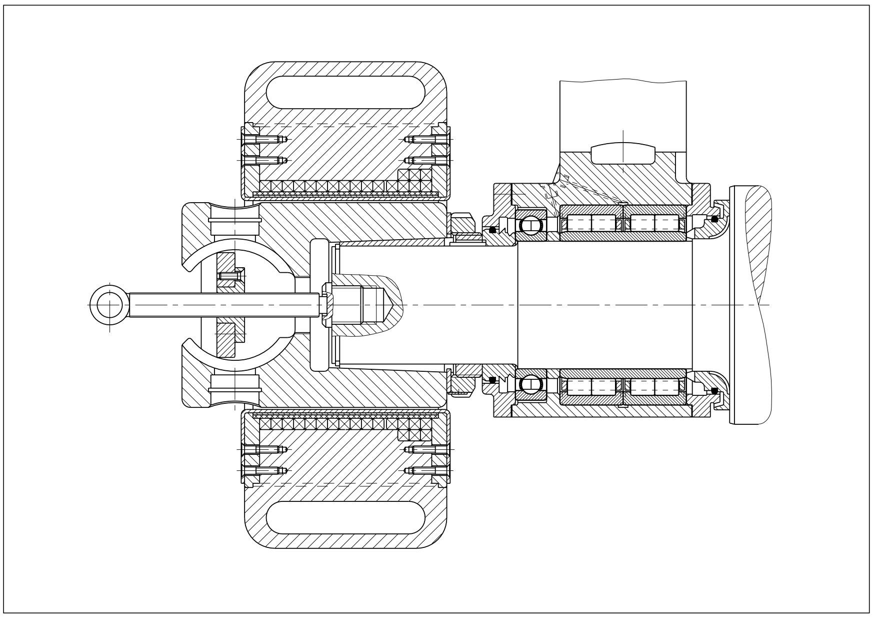
Apart from the inner rings, in some cases the couplings are also tightly fitted on the roll necks of high-speed wire and light-section trains.
During each roll change the roll couplings must be removed and mounted onto other rolls. With the formerly used hydraulic method this procedure took much time and, particularly after repeated mounting and dismounting, it became difficult. Therefore FAG has developed induction heating equipment similar to that used for inner rings. In this way the mounting times have been reduced considerably.
As a rule, the couplings are mounted onto the roll necks with an interference of 1,5 to 1,8 ‰.
A mounting temperature of 170 to 200 °C is required to release the interference fit. This temperature is reached within 70 to 360 seconds, depending on the coupling size.

To date, induction coils for couplings weighing up to 485 kg are available. For several years these devices have proved their worth in a number of rolling mills.
The design of such an induction coil is shown in fig. 92. The replaceable tapered sleeve between roll neck and coupling which was required for the hydraulic mounting method was split and retained as a wearing part. Couplings with a cylindrical seat and without sleeve are also induction-heated.
For dismounting a coupling shrunk onto a cylindrical seat, it is advisable to suspend the roll vertically such that the coupling comes off by its deadweight once the required temperature above ambient is reached. Couplings seated on a tapered sleeve are pulled from the roll neck by means of an extractor with the roll in a horizontal position.

The induction coils for couplings use the same transformers as the induction coils for cylindrical roller bearing inner rings. Of course the acquisition cost of an induction coil is higher than that of hydraulic tools. But since many roll necks must be provided with oil grooves and oil holes for the hydraulic mounting method, and since the expense for drilling long co-axial holes in large rolls is high, induction coils pay after a short time.
Moreover, mounting with an induction coils is cleaner and faster.

Induction coils for labyrinth rings

FAG also offers induction coils (operated with mains voltage) and low-voltage coils for heating labyrinth rings.



92: Design of the induction coil for couplings. The extractor makes removal of the coupling easier.

Mounting and maintenance

Mounting facilities for roll couplings and labyrinth rings · Statistical stockkeeping

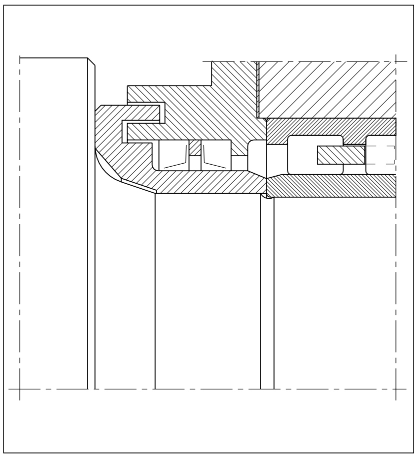
Labyrinth rings (fig. 93) are usually shrunk onto the roll neck with a large interference to prevent them from working loose if they are heated in operation by the rubbing seal.
It is often difficult to remove tightly fitted labyrinth rings. By means of an induction coil they can be heated to 150 to 200 °C within a few minutes, and the shrink fit is released. The devices are inexpensive if a factory already has the switching unit or the transformer for an induction coil for inner rings.

Spares

To prevent costly downtimes it is advisable to always have three and a half complete sets of bearings for each stand. One of these sets is mounted on the rolls actually in operation. A second set is mounted on the rolls being reground. A third set of rolls with bearings should always be readily available. An additional half set of bearings should be kept in stock in order to complete the bearing sets in the event of a bearing failure.
For cylindrical roller bearings with tightly fitted inner rings it is advisable to procure additional inner rings and mount them onto the roll necks. If the rolls have to be changed frequently, e.g. in section rolling mills, frequent mounting and dismounting of the inner rings is then not necessary.

Statistical stockkeeping

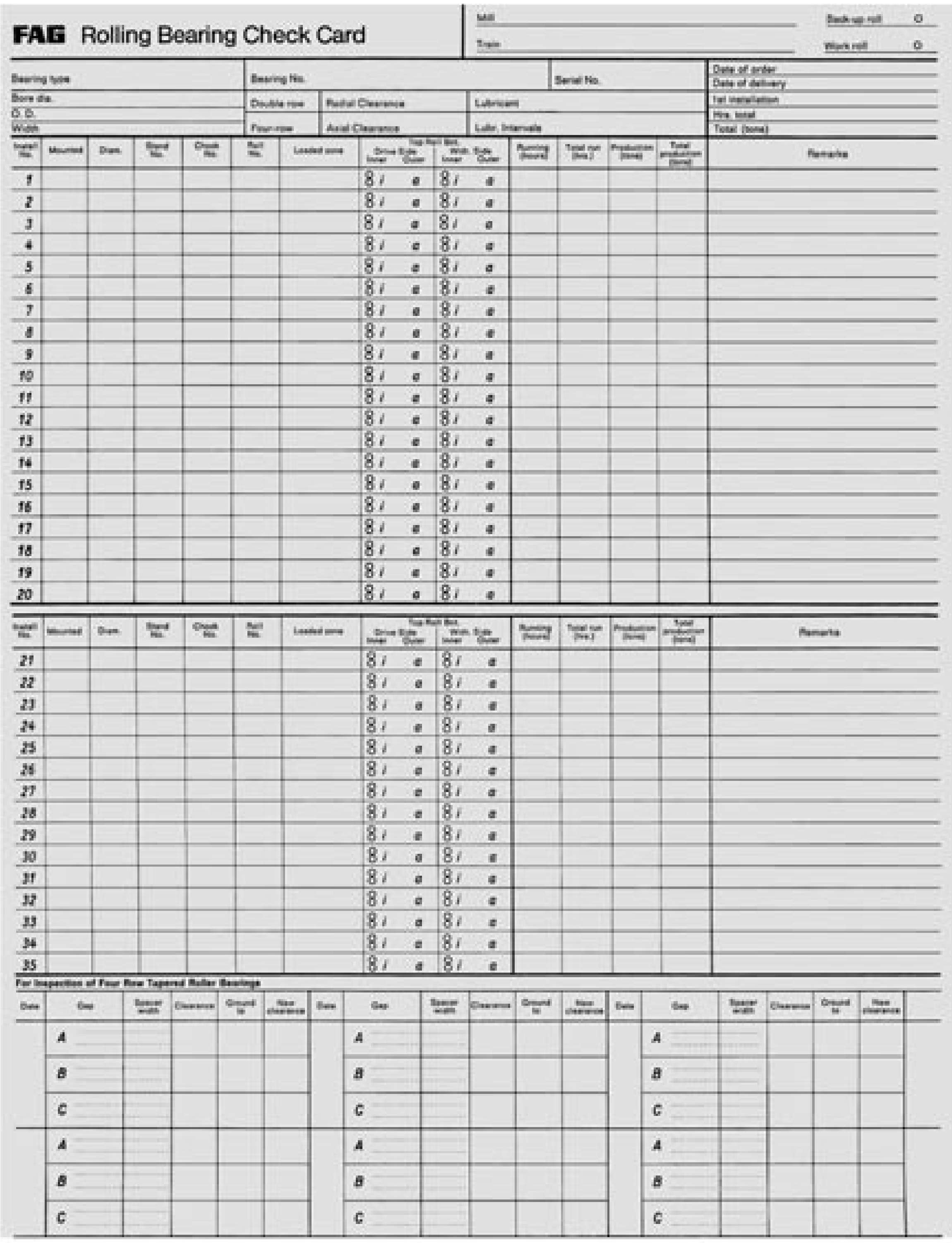
Upon arrival at the mill, a record card should be prepared for each roll neck bearing (page 63) on which all important data are recorded. It should be completed by operating data, e.g. temperatures and rolling loads measured during operation. Such constantly updated records ensure a more realistic evaluation of operating conditions and bearing performance than would be possible by basing the calculation on assumed loads.



93: Labyrinth ring of a work roll of a 4,2 m heavy-plate stand

Mounting and maintenance

Statistical stockkeeping



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Storage

The bearings should be stored in their original packaging. They should be unpacked only when they are actually needed. In this way contamination and corrosion can best be avoided. Large bearings with relatively thin-walled rings should not be stored standing upright but laid flat, supported over their entire circumference.
During transport, care must be taken not to damage the direct wrapping which, for large bearings, usually consists of plastic foil.
FAG rolling bearings are immersed into anti-corrosion oil and thus protected from environmental influences as long as they are kept inside their wrapping. This protection is effective for a long time but only if the bearings are stored in a dry and frost-proof room. Of course, no corrosive chemicals such as acids, ammonia or chloride of lime must be stored in the same room.
Bearings dismounted for temporary storage must be washed and immediately afterwards preserved and wrapped. It is advisable to wash them with kerosene.
Smaller bearings are preserved by dipping them into anti-corrosion oil. Larger bearings are sprayed carefully with anti-corrosion oil.
Instead of wrapping the bearings, they can be stored immersed in an oil tank.
If chocks, complete with bearings, are not to be used for some time, they must be checked to make sure that no water has penetrated into the chocks. If there is any water, the bearings must of course be filled with fresh grease or – in the case of oil mist lubrication – cleaned and preserved. For storage, the chocks are closed with covers on both sides.

Worked example

Roll neck mountings of a wire mill

Builder:
SMS Schloemann-Siemag AG, Düsseldorf and Hilchenbach

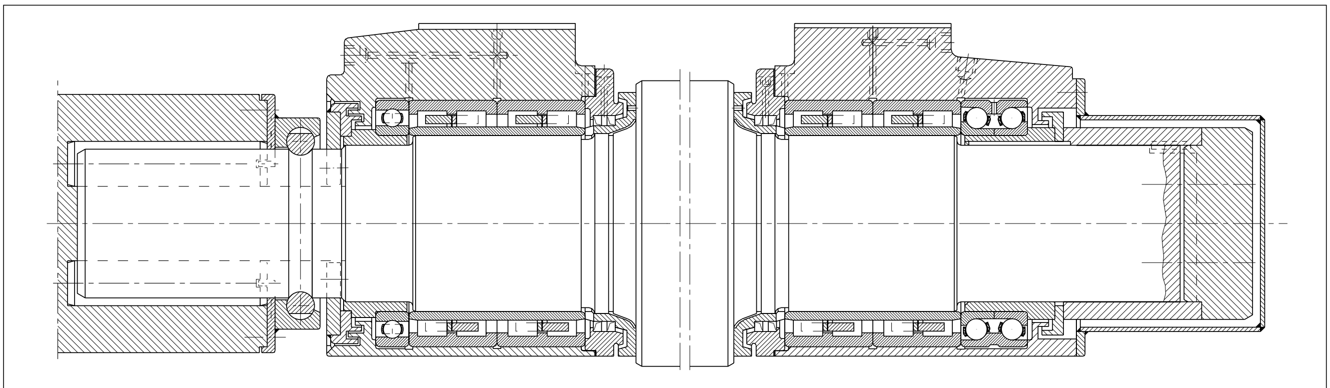
The wire mill is designed for a final speed of 50 m/s. A close final tolerance is specified for the rolled wire.
The line is made up of three sections:
- Roughing section (six stands)
- Intermediate section (eight stands)
- Finishing section

80×80 mm billets are reduced continuously down to wire with diameters of 12 to 5,5 mm.
The roughing and intermediate sections handle two strands. Leaving the intermediate section, the two strands are fed to the separate single-strand finishing sections.
The first stand of the roughing section comprises rolls with a body diameter of 450 mm, the second stand has rolls with a body diameter of 420 mm. Both roll sizes are supported by identical bearings.
The body diameter of the remaining stands of the roughing section is 380 mm.

In the intermediate section, the first two stands are also equipped with rolls having a body diameter of 380 mm. On the remaining stands of the intermediate section, the roll body has a diameter of 320 mm.
The roll necks of the roughing and intermediate sections are radially supported in four-row cylindrical roller bearings. Double-row angular contact ball bearings are provided for transmitting the axial forces.
Location of the drive end chocks is provided by deep groove ball bearings.
The dimensions and load ratings of the different bearings are listed in the table below.

95: Bearing dimensions and load ratings

Stand	Roll body dia-meter mm	Radial bearings	Dimensions mm	dyn. load rating kN	Thrust bearings Locating bearing end	Dimensions mm	dyn. load rating kN	Floating bearing end	Dimensions mm
1	450	Cylindrical roller bearing, four-row FAG 507336	260×370×220	2200	Angular contact ball bearing, double-row FAG 508731A	260×369,5×92	390	Deep groove ball bearing FAG 507338A	260×369,5×46
2	420	Cylindrical roller bearing, four-row FAG 507336	260×370×220	2200	Angular contact ball bearing, double-row FAG 508731A	260×369,5×92	390	Deep groove ball bearing FAG 507338A	260×369,5×46
3-8	380	Cylindrical roller bearing, four-row FAG 508727	230×330×206	2080	Angular contact ball bearing, double-row FAG 508732A	230×329,5×80	320	Deep groove ball bearing FAG 508729	230×329,5×40
9-14	320	Cylindrical roller bearing, four-row FAG 508657	190×270×200	1660	Angular contact ball bearing, double-row FAG 508658A	190×269,5×66	224	Deep groove ball bearing FAG 502288	190×269,5×33



Roll neck bearing arrangement of the intermediate section

Worked example

96: Calculation of fatigue life

Stand or pass no.	Radial bearings Radial load kN	 Speed n min⁻¹	 Speed coefficient f_n	 Dynamic load rating C kN	 Index of dynamic stressing f_L	 Fatigue life L_h h	Thrust bearings Axial load kN	 Equivalent bearing load kN	 Speed coefficient f_n	 Dynamic load rating C kN	 Index of dynamic stressing f_L	 Fatigue life L_h h
1	1080	9,08	1,477	2200	3,01	19700	99	92	1,543	390	6,54	>60000
2	530	13,47	1,312	2200	5,45	>60000	48	44	1,353	390	12	>60000
3	680	19,77	1,170	2080	3,58	35100	61	57	1,19	320	6,68	>60000
4	340	26,45	1,072	2080	6,56	>60000	31	29	1,08	320	11,9	>60000
5	530	36,75	0,971	2080	3,81	43200	49	45	0,968	320	6,88	>60000
6	360	51,9	0,876	2080	5,06	>60000	33	31	0,863	320	8,91	>60000
7	330	71,5	0,795	2080	5,01	>60000	30	28	0,775	320	8,86	>60000
8	210	99,2	0,721	2080	7,14	>60000	19	17	0,695	320	13,1	>60000
9	200	156,1	0,629	1660	5,22	>60000	18	17	0,598	224	7,88	>60000
10	140	207,3	0,578	1660	6,85	>60000	13	12	0,544	224	10,2	>60000
11	180	264,2	0,537	1660	4,95	>60000	16	15	0,502	224	7,5	>60000
12	120	364,8	0,488	1660	6,75	>60000	11	10	0,45	224	10,1	>60000
13	250	411,2	0,471	1660	3,13	22400	23	21	0,433	224	4,62	>60000
14	100	485,8	0,448	1660	7,44	>60000	9,3	8,7	0,409	224	10,5	>60000

Fatigue life

When determining the bearing loads the fact has to be taken into account that the rolling stands of the roughing and intermediate sections handle two strands. The roll neck loads are calculated according to the indicative values given on page 11. 5 % of the individual maximum rolling load is assumed as axial load (see also page 18).
The calculation of the fatigue life for the cylindrical roller bearings and angular contact ball bearings is summarized in the table above.
The theoretical fatigue life of nearly all bearings exceeds 60000 hours. However, these values are not likely to be reached in practice. The service life will be shorter due to wear.

Machining tolerances, bearing clearance

All cylindrical roller bearings in this line are fitted tightly. The roll necks of the roughing and intermediate sections are machined to r6.
The outer ring seats in the roughing and finishing sections are machined to J7. To simplify mounting and dismounting, the inner rings of angular contact ball bearings serving as thrust bearings are mounted on sleeves.
Since it must be possible to displace the rolls laterally relative to each other for groove aligning purposes, the double-row angular contact ball bearing of one roll in each stand is mounted on a threaded bushing. The roll can be axially adjusted in the chock by means of this bushing. The thrust bearings have a very small axial clearance.

Lubrication

The bearings of the roughing and intermediate section are lubricated with grease.

Notes

Selection of FAG publications

The following list gives a selection of FAG publications. For further information please contact FAG.

Publ. No. WL 41 140	FAG Rolling Bearings for Rolling Mills – Bearing Tables
CD-medias4.0	INA-FAG Rolling Bearing Catalog
CD-WLS	Rolling Bearing Learning System
Publ. No. WL 17 109	FAG Rolling Bearings in Rolling Mills
Publ. No. WL 17 114	Sealed FAG Spherical Roller Bearings
Publ. No. WL 80 100	Mounting and Dismounting of Rolling Bearings
Publ. No. WL 80 102	How to Mount and Dismount Rolling Bearings Hydraulically
Publ. No. WL 80 107	FAG Induction Heating Equipment
Publ. No. WL 80 110	Radial Clearance Reduction of FAG Spherical Roller Bearings with Tapered Bore
Publ. No. WL 80 123	All About the Rolling Bearing – FAG Training Courses on Rolling Bearings Theory and Practice
Publ. No. WL 80 134	FAG Video: Mounting and Dismounting Rolling Bearings
Publ. No. WL 80 135	FAG Video: Hydraulic Methods for the Mounting and Dismounting of Rolling Bearings
Publ. No. WL 80 151	FAG Repair Service for Large Rolling Bearings
Publ. No. WL 80 250	FAG Mounting and Maintenance Equipment and Services for Rolling Bearings
Publ. No. WL 81 115	Rolling Bearing Lubrication
Publ. No. WL 81 116	Arcanol · Rolling-Bearing Tested Grease
Publ. No. WL 81 122	Automatic Lubricators “COMPACT” and “CHAMPION” Motion Guard
Publ. No. WL 82 102	Rolling Bearing Damage
TI No. WL 00-11	FAG Videos on Rolling Bearings
TI No. WL 17-7	Split Cylindrical Roller Bearings for Rolling Mill Drive Shafts
TI No. WL 40-48	FAG Life Calculation – Introduction of the Modified Rating Life
TI No. WL 80-14	Mounting and Dismounting of Spherical Roller Bearings with Tapered Bore
TI No. WL 80-47	FAG Induction Heating Devices
TI No. WL 80-50	FAG Pressure Generators
TI No. WL 80-53	Rolling Bearing Mounting Cabinet and Mounting Sets - A basic course for vocational training
TI No. WL 80-57	FAG Hydraulic Nuts
TI No. WL 80-62	FAG Detector II – the “Mobile Phone” Among the Data Collectors
TI No. WL 80-63	Rolling Bearing Diagnosis with the FAG Bearing Analyser III
TI No. WL 80-70	Measuring Tapered Journals with an FAG Taper Measuring Instrument of Series MGK9205; Dimensioning of Tapered Journals

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