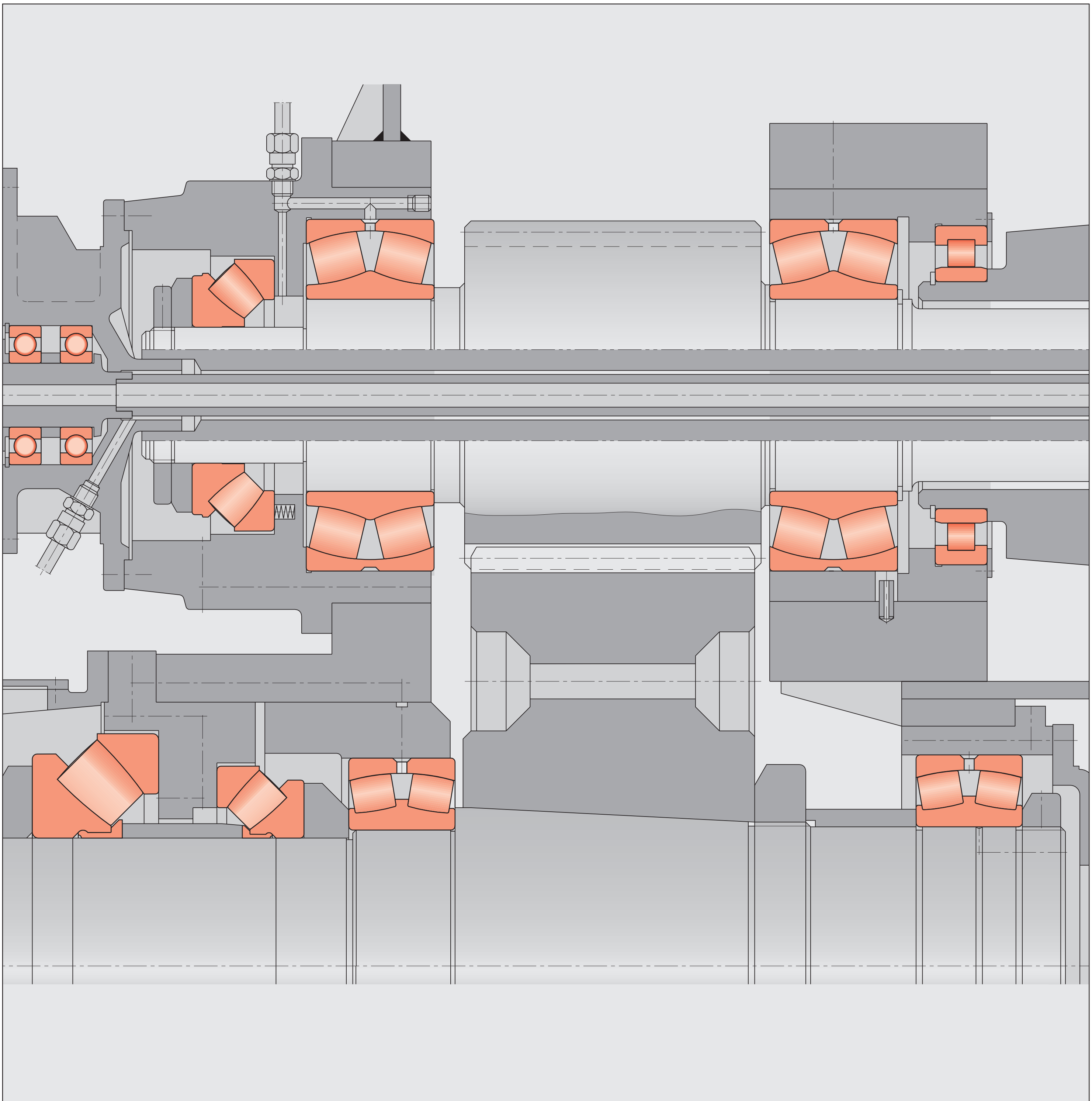
The Design of Rolling Bearing Mountings

PDF 5/8:
Paper machines
Lifting and conveying equipment

FAG Rolling Bearings

FAG OEM und Handel AG

Publ. No. WL 00 200/5 EA



The Design of Rolling Bearing Mountings

Design Examples covering
Machines, Vehicles and Equipment

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Preface

This publication presents design examples covering various machines, vehicles and equipment having one thing in common: rolling bearings.

For this reason the brief texts concentrate on the rolling bearing aspects of the applications. The operation of the machine allows conclusions to be drawn about the operating conditions which dictate the bearing type and design, the size and arrangement, fits, lubrication and sealing.

Important rolling bearing engineering terms are printed in italics. At the end of this publication they are summarized and explained in a glossary of terms, some supplemented by illustrations.

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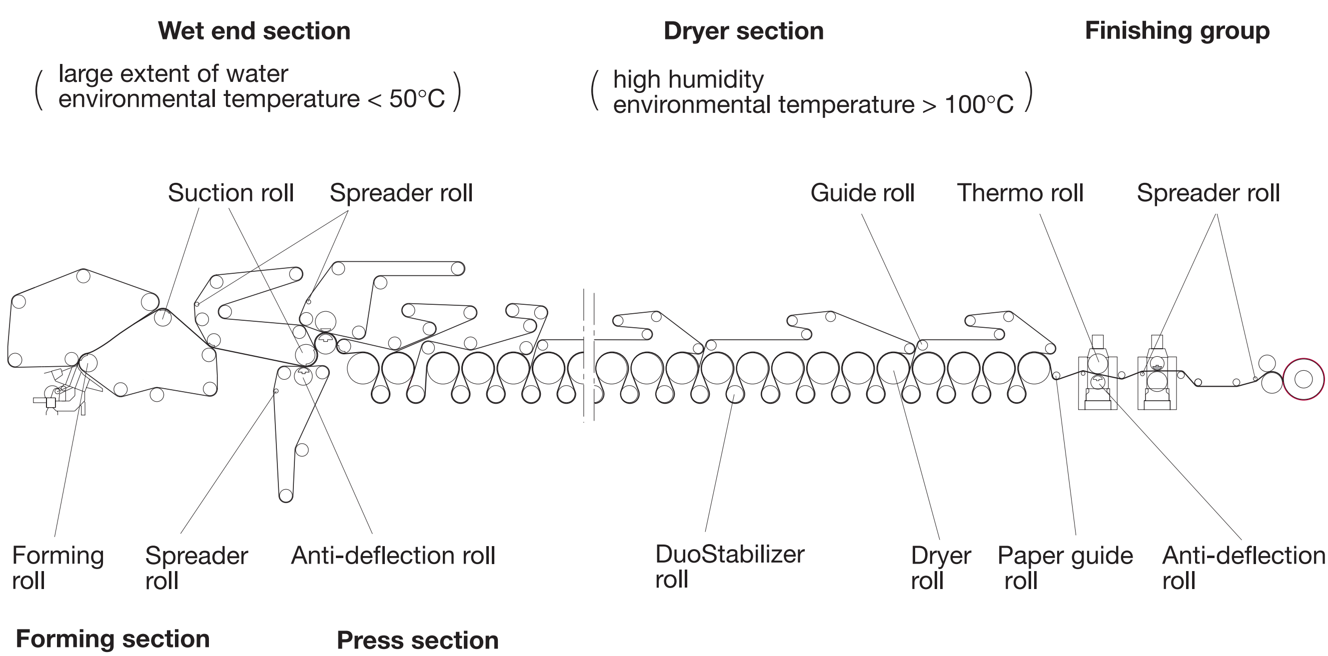
65–72 Paper Machines

Modern paper machines are extensive plants which frequently stretch well beyond 100 m in length and have numerous rolls. The demand for utmost operational reliability is priority number one when designing and dimensioning bearing locations: if trouble arises at just one roll the whole plant has to be shut down. For this reason the bearings are designed for a far longer *nominal life* (*index of dynamic stressing* f_L = 5...6) than in other industrial equipment. A high degree of cleanliness in the bearings is decisive for a long *service life*. This demands utmost *sealing* reliability, particularly against moisture, and design diversity based on the type of roll in question.

Lubrication also influences the *bearing life* greatly. All roll bearings in modern paper machines are connected to an *oil* circulation system for operational reliability and maintenance purposes. The bearings in the wet end section of older paper machines are still lubricated with *grease* (lower environmental temperatures).

In the dryer section, bearings for rope sheaves, spreader rolls and sometimes guide rolls are still lubricated with *grease*.
Due to high temperatures in the area of the dryer roll, bearing lubrication is particularly critical. Therefore *oils* of the *viscosity class* ISO VG 220 or 320 are used. Lightly doped *mineral oils* and *synthetic oils* are suitable (high ageing stability), which correspond to the requirements for dryer roll oils and have proven themselves in the field or successfully stood dynamic testing on the FAG test rig FE8.
Lubrication can be improved considerably (increasing the *operating viscosity*) by insulating the hollow journals of the dryer rolls and thus reducing the bearing temperature.
The following examples show the structure of some main bearing locations in the paper industry, for example refiners, suction rolls, press rolls, dryer rolls, guide rolls, calender thermo rolls, anti-deflection rolls and spreader rolls.



A modern paper machine

65 Refiners

Wood chips from the wood chopper which have been softened and steamed by water are broken down and crushed in the refiner by means of crushing wheels rotating in reverse motion with knife sections. Temperatures up to 160 °C result from this process (steamed wood chips, crushing) and can lead to increased operating temperatures in bearings depending on their construction.

Operating data

Axial load from crushing process 400 kN;
Radial load (rotor/shaft) 15 kN per bearing;
Speed 600 min^{-1};
Temperature in locating bearing 80 °C, in floating bearing 70 °C.

Bearing selection, dimensioning

With the high axial loads which have to be accommodated, an *attainable life* $L_{hna} \geq 80{,}000$ hours is required. A second thrust bearing is necessary since the axial load acts mainly in the direction of the *locating bearing* but can also be acting in the opposite direction. Thus the *locating bearing* arrangement is made up of two symmetrically arranged spherical roller thrust bearings FAG 29460E. For the rollers to remain undisturbed when the axial load is "reversed" both bearings must be preloaded with springs (minimum load) at the outer rings.
A spherical roller bearing FAG 23052K.MB is mounted as a *floating bearing* and can easily accommodate shaft deflection. Thermal length variations of the shaft are compensated for in between bearing outer ring and housing (sliding *fit*). The bearing is mounted directly on the tapered shaft seat and fastened with a locknut HM3052.
The *floating bearing* reaches a *nominal life* L_h of well over 200,000 hours. Excellent bearing lubrication is required due to slippage hazard when loads are low ($P/C \approx 0.02$).
A *nominal life* of $L_h = 50{,}600$ h is calculated for the left *locating bearing* 29460E. With *oil* circulation lubrication, good cleanliness and a bearing temperature of 70 °C, *factor* a_{23} is 3.2. An *attainable life* $L_{hna} =$ 162,000 h results from the *adjusted life calculation.*

The right *locating bearing* only has a slight axial load (spring preload). The *attainable life* L_{hna} is over 200,000 h for this bearing.

Machining tolerances

Floating bearing: The inner ring has *circumferential load* and is attached to the tapered bearing seat of the shaft.
Roundness tolerance IT5/2 (DIN ISO 1101);
Taper angle tolerance AT7 (DIN 7178).
Bearing seat of housing bore according to G7.

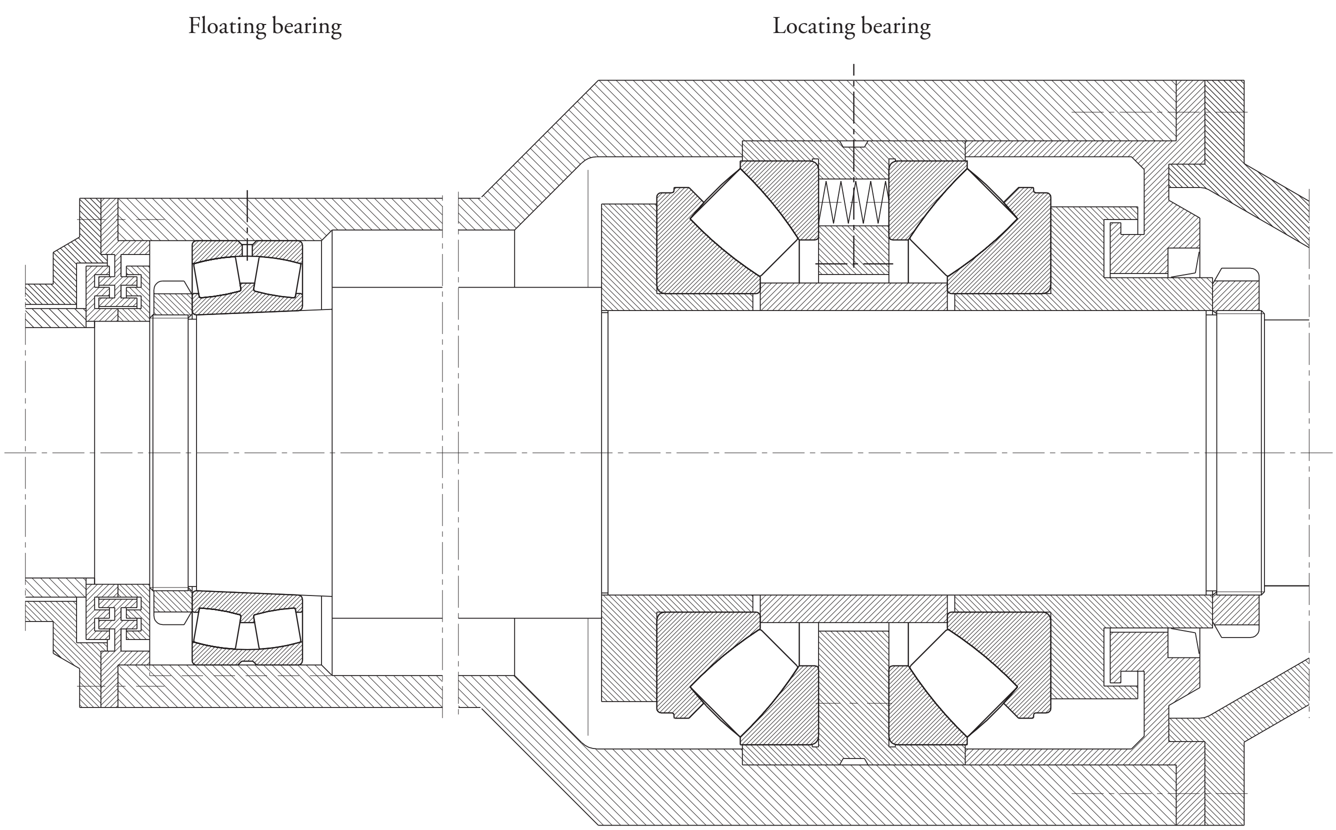
Locating bearing: For mounting reasons, both shaft and housing washer are in sleeves. The bearing seats are machined according to k6 and G7 for the shaft sleeves and housing sleeves respectively.

Lubrication

A *lubricating oil* ISO VG 150 with *EP additives* is used for *locating* and *floating bearings.*
The radial spherical roller bearing has oil circulation lubrication with 0.8 l/min. Oil jet lubrication is provided for the spherical roller thrust bearings. This ensures adequate *oil* constantly at the highly-stressed contact areas between roller face and lip. The *oil* is supplied through the side of the bearing via the spacer sleeve. The minimum *oil* flow rate for both bearings is 8 l/min (good heat dissipation from bearing). The oil is filtered in cirulation and cooled back to a temperature of 40 °C.

Sealing

There are two labyrinths on the side of the crushing wheel connected to one another and filled with grease which protect the bearings from water and contamination and prevent *oil* escaping from the bearings. On the outer side of the *locating bearing* a shaft sealing ring prevents *oil* escape.



65: Refiner bearings

66 Suction rolls

Suction rolls are found in the wire or press section of a paper machine. They are hollow cylinders up to 10 m in length which have several small holes all around their circumference. Some water is removed from the web due to the rotating roll shell and the vacuum inside the roll. The suction box, as interior axle, is stationary. The roll shell is driven by planet wheels in modern paper machines.

Operating data

Roll length 7,800 mm; roll diameter 1,600 mm; rotation 278 min^{-1} (speed 1,400 m/min); roll weight 270 kN; wire tension 5 kN/m.

Bearing selection, dimensioning

The diameter of the suction box is decisive for the size of the bearing. We recommend bearings with a *dynamic load rating* as low as possible; the higher specific bearing load reduces the danger of slippage. *Self-aligning bearings* are necessary as misalignment could arise. Roll weight, wire tension and rotational speed are the main criteria for dimensioning the bearings.
FAG spherical roller bearings FAG 239/850K.MB.C3 with tapered bore (K 1:12) and increased *radial clearance* are used. The bearings are mounted directly on the tapered shaft seats for running accuracy reasons. The hydraulic method is applied to facilitate mounting.
The *locating bearing* provides axial guidance for the rolls while the *floating bearing* compensates for any length variations caused by displacement of the outer ring in the housing bore.
The *nominal life* for both bearings is L_h > 100,000 h. The *attainable life* reaches over 200,000 h when the operating temperature is 60 °C and *oil* ISO VG 68 (*viscosity ratio* $\kappa > 2$; *factor* $a_{23} = 2.2$) is used.

Machining tolerances

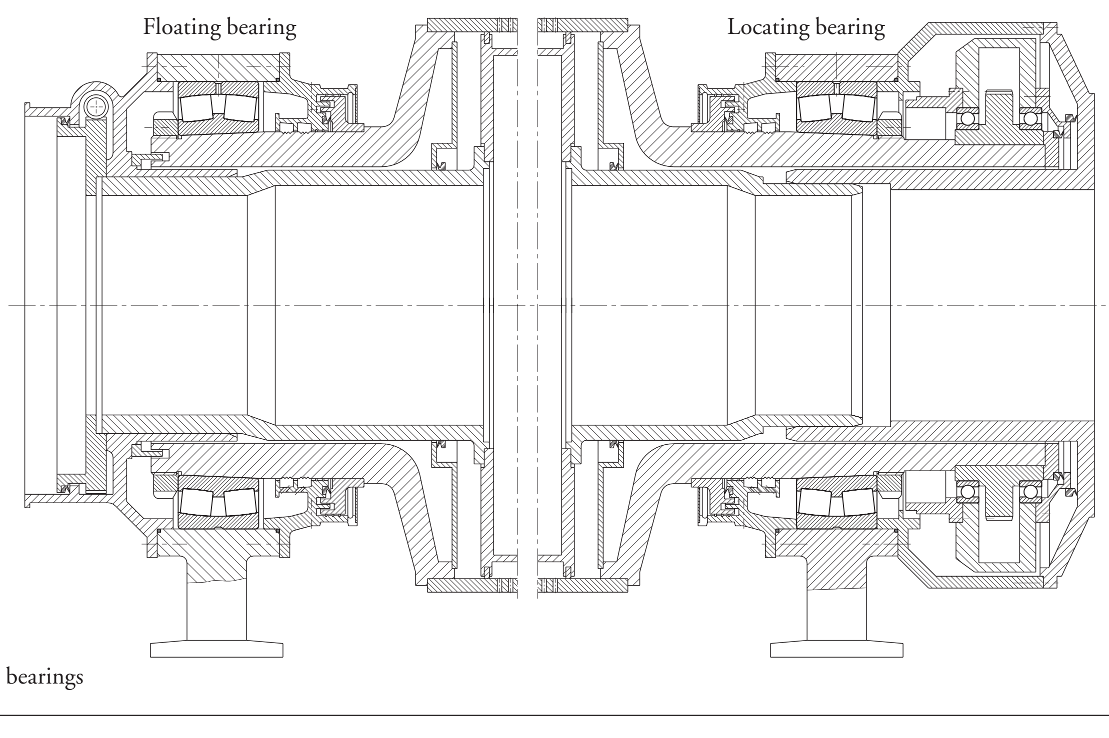
The inner ring has *circumferential load* and is attached to the tapered bearing seat of the shaft.
Roundness tolerance IT5/2 (DIN ISO 1101); taper angle tolerance AT7 (DIN 7178).
Housing bores according to G7 due to *point load* at the outer ring.

Lubrication

The spherical roller bearings are supplied by circulation lubrication with a *mineral oil* quantity of 8 l/min. A *mineral oil* with sufficient *viscosity* and *EP additives* is selected. *Additives* with good anti-corrosive properties and water separation ability are also required. An effective lubrication is achieved with an oil supply to the centre of the bearing.

Sealing

Any *oil* which escapes is thrown off via splash grooves into oil collecting chambers and directed back. At the roll side a baffle plate and multiple grease-filled labyrinth with integrated V ring prevent water penetrating from the outside.



66: Suction roll bearings

67 Central press rolls

The paper web runs through the press rolls on a felt cloth and a large amount of water is pressed out of it. Modern press sections consist of one central press roll against which one or more (suction) press rolls are pressed. The central press roll is solid, made of granite/steel or steel with a protective coating.

Operating data

Roll length 8,800 mm; roll diameter 1,500 mm; speed 1,450 m/min; roll weight 750 kN. Pressure by 3 rolls at 30°, 180° and 210°; bearing temperature about 60 °C. Direct drive.

Bearing selection, dimensioning

Self-aligning spherical roller bearings of the series 231 or 232 with a very high load carrying capacity are chosen due to the high radial load and the misalignment which is possible between the bearing locations. A low cross section height is also important for these bearings since the height of the housing is restricted by the roll diameter. The roll weight and the load components of the pressure rolls yield a resulting bearing load F_r = 300 kN.
A spherical roller bearing FAG 231/600K.MB.C3 is mounted at every bearing location. The bearings with tapered bore (taper 1:12) are pressed directly onto the tapered shaft seat by means of the hydraulic method. The *floating bearing* at the operator's end permits temperature-depending length variations of the roll by shifting the outer ring in the housing. The *locating bearing* is at the drive end.

The *nominal life* calculated is L_h > 100,000 h with a speed of 308 min^{-1}. With good lubrication (*viscosity ratio* $\kappa \approx 3$, *basic factor* $a_{23II} = 3$) and improved cleanliness (*contamination factor* V = 0.5) in the lubricating gap $L_{hna} \gg 100{,}000$ h according to the *adjusted rating life calculation.*

Machining tolerances

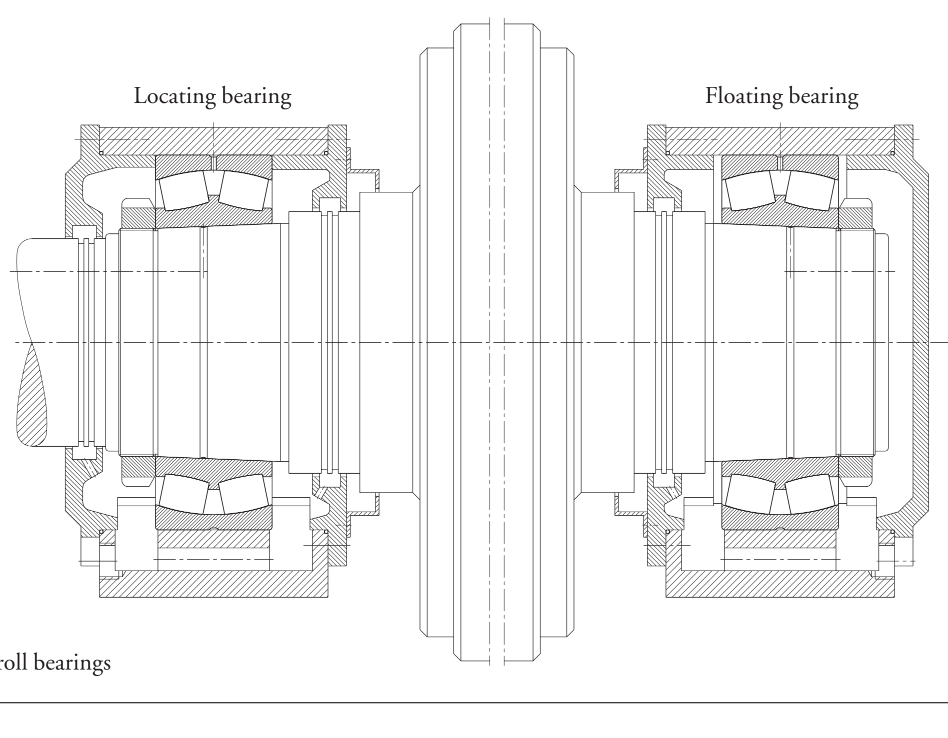
The inner ring has *circumferential load* and is attached to the tapered bearing seat of the shaft.
Roundness tolerance IT5/2 (DIN ISO 1101); taper angle tolerance AT7 (DIN 7178).
Housing bores according to G7 since there is *point load* at the outer ring.

Lubrication

The spherical roller bearings are supplied with a minimum *oil* quantity of 7 l/min by circulation lubrication. A *mineral oil* of sufficient *viscosity* (ISO VG 100) and *EP additives* is used. *Additives* with good anti-corrosive properties and water separation ability are also required. An effective lubrication is achieved with an oil supply to the centre of the bearing.
Oil returns to both sides of the bearing via oil collecting pockets and connecting holes.

Sealing

Oil splash grooves in the roll journal prevent *oil* escaping at the cover passage.
Non-rubbing and maintenance free gap-type *seals* protect the bearings from environmental influences.



67: Central press roll bearings

68 Dryer rolls

The remaining water in the dryer section is evaporated. The paper runs over numerous heated dryer rolls guided by endless dryer wires (formerly dryer felts). The dryer rolls are steam heated (temperature depends on the type of paper, its thickness and speed, and on the number of dryer rolls). The high temperatures of the heating steam transfer to the bearing seats stressing the rolling bearings accordingly. Today, the journals through which the steam flows are insulated in order to keep bearing temperatures low.

Operating data

Working width 5,700 mm; roll diameter 1,800 mm; paper speed 1,400 m/min (rotational speed 248 min^{-1}); heating temperature 165 °C (7 bar); roll weight 90 kN. Felt pull 4.5 kN/m; wrap angle 180°; environmental temperature under the dryer section hood approx. 95 °C; insulated journal bores.

Bearing selection

The bearing load is calculated from the roll weight, felt pull and temporary water fill. *The floating bearing* is loaded with 75 kN, the *locating bearing* with 83 kN taking into account the drive force. Heating the dryer roll leads to heat expansion which in turn leads to considerable changes in length with such long rolls. *Self-aligning* rolling bearings are necessary due to the misalignment arising between both bearing locations.
A double-row cylindrical roller bearing of the dimension series 31 is provided as *floating bearing* at the operator's end. It easily compensates for length variations in the bearing between the rolls and the inner ring raceway. With its spherical sliding surface a plain spherical bearing's seating ring accommodates any alignment inaccuracy of the journal. A double-row self-aligning cylindrical roller bearing FAG 566487K.C5 with the dimensions 200x340x112 mm is mounted. A spherical roller bearing FAG 23140BK.MB.C4 is mounted as *locating bearing* on the drive end.
Both bearings have about the same *operating clearance* in order to avoid any detrimental preload during the heating-up stage which may lead to a maximum temperature difference of 50 K. The spherical roller bearing has an increased *radial clearance* according to C4 (260...340 microns), the cylindrical roller bearing an increased *radial clearance* according to C5 (275...330 microns).

Both bearings have a tapered bore (K 1:12) and are mounted by the hydraulic method directly onto the tapered journals.

Since the cylindrical roller bearing and the spherical roller bearing have the same dimensions unsplit PMD plummer block housings (FAG PMD3140AF or BF) are applied both at the drive end and at the operator's end.

Due to increased operating temperature, both bearings are given special heat treatment (isotemp) and are thus dimensionally stable up to 200 °C.

Bearing dimensioning

An *attainable life* $L_{hna} \geq 250{,}000$ hours is required for dryer roll bearings. Lubrication decisively influences the *adjusted rating life.* Under an average operating temperature of 100°C the *operating viscosity* $\nu \approx 16\ mm^2/s$ for a *mineral oil* with a nominal *viscosity* of 220 mm^2/s (ISO VG 220).
The *rated viscosity* is determined from the speed and the mean bearing diameter $d_m = (200 + 340)/2 = 270$ mm to $\nu_1 = 25\ mm^2/s$.
The *viscosity ratio* is then:
$\kappa = \nu/\nu_1 = 16/25 = 0.64$.
With the *value* K = 1 a *basic factor* $a_{23II} = 1.1$ is obtained for the spherical roller bearing.
The values K = 0 and $a_{23II} = 1.4$ apply to the cylindrical roller bearing.
With normal cleanliness (*cleanliness factor* s = 1) the *factor* $a_{23} = a_{23II} \cdot s$
1.1 for the spherical roller bearing,
1.4 for the cylindrical roller bearing.
The *attainable life* $L_{hna} = a_1 \cdot a_{23} \cdot L_h$ is therefore well over 250,000 h for both bearings.

Machining tolerances

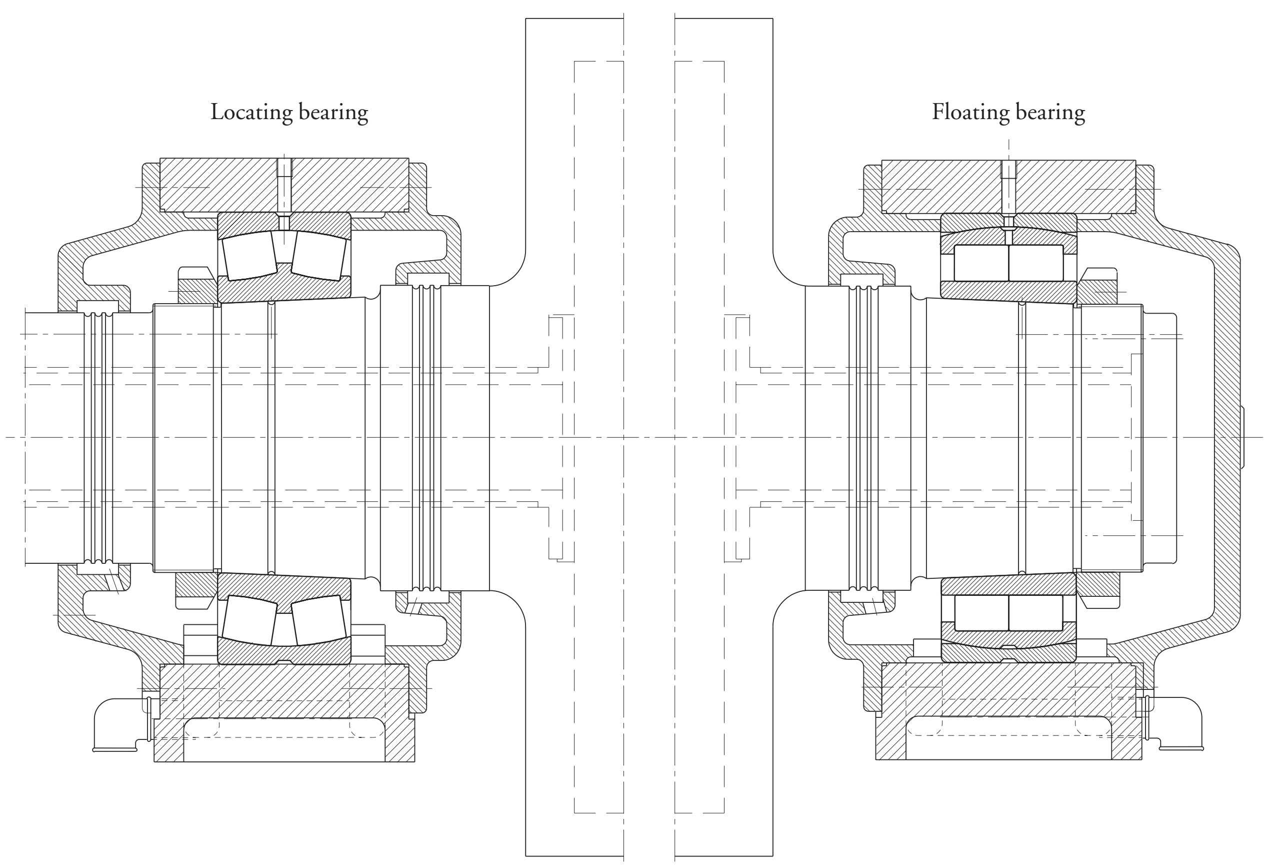
The inner rings have *circumferential load* and have a tight *fit* on the tapered roll journal. The journals have oil ducts so the bearings can be mounted and dismounted by means of the hydraulic method. Roundness tolerance IT5/2 (DIN ISO 1101), taper angle tolerance AT7 (DIN 7178). Bearing seats in the housing bore according to G7.

Lubrication

The bearing housings are connected to a central *oil* circulation lubrication system so that heat is constantly dissipated from the bearing. High-grade *mineral oils* ISO VG 220 or 320 are used which must have a high *operating viscosity,* thermal stability, good protection against *wear,* good water separation ability and a high degree of cleanliness. A minimum *oil* quantity of 1.6 l/min is guided directly to the centre of the bearing via a lubricating groove and lubricating holes in the outer ring.
The oil can be carried off at both sides of the bearing with the central oil system. The danger of oil retention and leakage is minimized considerably. Any contaminants or *wear* particles which might penetrate the bearing are immediately washed out of it with this method of lubrication.

Sealing

Gap-tape *seals,* which are non-rubbing and maintenance-free, are provided as *sealing* for the journal passages. The *oil* is thrown off via splash grooves and *oil* collecting chambers and flows back through return holes to the two *oil* cavities on the housing floor. Cover *seals* make the housing of the paper machine *oil* proof.



68: Dryer roll bearings

69 Guide rolls

Guide rolls guide, as the name indicates, and turn the wire and felt cloth in the wet end and dryer sections of a paper machine. The same bearings are used for the guide rolls in both areas. Lubrication and *sealing* differ, however, depending on the place of application.
In older machines the wet end section is usually lubricated with grease, and the dryer section with *oil.*
In modern machines both sections have *oil* circulation lubrication. Due to different operating conditions separate oil circuits are necessary for the wet end and dryer sections.
The larger the machine the more often it is found to be faster. For this reason the bearing inner rings are mounted with a tapered bore directly on the tapered roll journal.

Wet end section
Depending on the positions of the bearings in the machine they are subject to a small or large degree of moisture. Water must not penetrate the housing particularly when machines are being high-pressure cleaned.

Dryer section
Environmental temperatures of about 95 °C lead to great length variations and place high demands on lubrication. The operating temperature of the bearings can be 115 °C.

Operating data

Useful width 8,800 mm
Roll diameter 700 mm
Paper speed 1,650 m/min ($n = 750\ \text{min}^{-1}$)
Roll weight $F_G \approx 80$ kN
Paper pull 1 kN/m (tensile load $F_z \approx 9$ kN)
Wrap angle 180°
Bearing temperature approx. 105 °C

Bearing selection, dimensioning

The bearings must be able to accommodate loads and compensate for misalignment at the same time (misalignment, bending). An increased *radial clearance* according to C3 is necessary due to temperature differences. Spherical roller bearings FAG 22330EK.C3 are mounted.

Bearing load:

$P = (F_G + F_z)/2 = (80 + 9)/2 = 44.5$ kN

The diameter of the roll journal is determined by the roll rigidity required. As a result there is a high *index of dynamic stressing* f_L corresponding to a *nominal life* L_h of well over 200,000 hours. The *attainable life* is even higher with such good lubrication conditions.

The housings can be in standing or suspended position or can be laterally screwed on. They are designed for *oil* circulation lubrication.

Machining tolerances

The inner rings have *circumferential load* and are directly fitted to the tapered roll journal. The roll journal have oil ducts so the bearings can be mounted and dismounted with the hydraulic method.
Roundness tolerance IT5/2 (DIN ISO 1101); taper angle tolerance AT7 (DIN 7178).

Bearing seats in the housing bore according to G7.

Lubrication

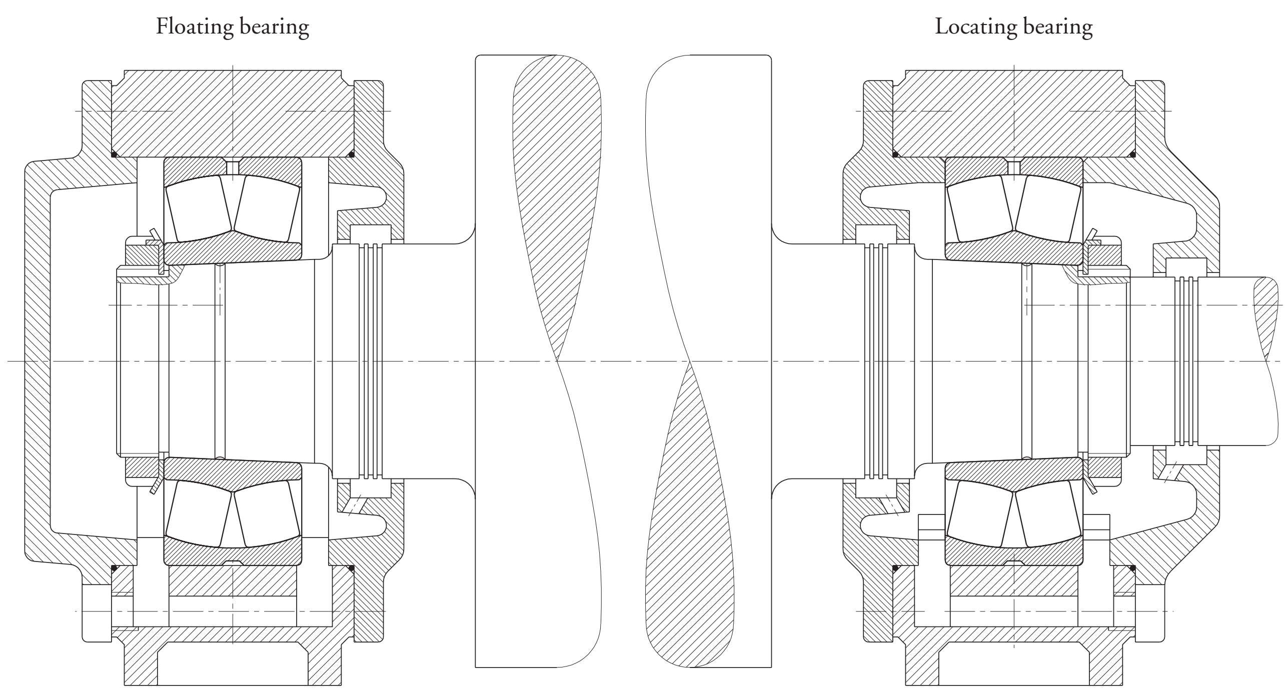
In the dryer section: see example 68 (Dryer rolls) since the bearings are connected to the *oil* circuit of the dryer rolls. Minimum flow rate 0.9 l/min.

In the wet end section: see example 66 (Suction rolls) and 67 (Central press rolls), since the bearings are connected to the *oil* circuit of the wet section rolls. Minimum flow rate 0.5 l/min.

Sealing

Gap-type *seals,* which are non-rubbing and maintenance-free, prevent *oil* from escaping through the cover passages in the dryer section.

The bearings in the wet end section must have relubricatable labyrinth *seals* to prevent water from penetrating. Remaining *oil* is thrown off by splash grooves into collecting chambers and directed back. Cover *seals* make the housing oilproof.



69: Guide roll bearings (dryer section)

70 Calender thermo rolls

The paper passes through the so-called calender stack after leaving the dryer section. Soft calenders smooth the surface of the paper thus improving its printability. The calender consists of two pairs of rolls. One calender roll (steel) lies above a counter roll, another below one. The counter roll is the so-called anti-deflection roll (elastic material). Soft calender rolls can be heated by water, steam, or oil. The gap or the "nip" pressure depends on the type of paper.

Operating data

Useful width approx. 7 m
Rotation 350 min^{-1} (speed 1,100 m/min)
Heated by oil at 200...250 °C
Insulated roll journal
Operating temperature at bearing inner ring 130 °C.

Bearing selection, dimensioning

The radial bearing load depends on the application of the calender roll as lower or upper roll, on the weight F_G and the variable pressure load with percentage of time.

$P_1 = F_G + F_{nip\,min}$ = 600 kN
$P_2 = F_G + F_{nip\,med}$ = 990 kN
$P_3 = F_G + F_{nip\,max}$ = 1,260 kN
$P_4 = F_G - F_{nip\,min}$ = 60 kN
$P_5 = F_G - F_{nip\,med}$ = 390 kN
$P_6 = F_G - F_{nip\,max}$ = 720 kN

Percentages of time: P_1, P_4 : 10 % each
P_2, P_3, P_5, P_6 : 20 % each

The sum of the roll weight and the nip load acts for the application as bottom roll whereas their difference acts for the application as top roll.

Taking the maximum load for designing the bearing would lead to overdimensioning (*equivalent dynamic load* $P < 0.02 \cdot$ *dynamic load rating* C) in the case of application in the top roll. Slippage may occur with such a low load which in turn can lead to bearing damage when lubrication is inadequate. In order to avoid this problem, smaller bearings with a smaller *dynamic load rating* C should be selected so that P/C > 0.02. The risk of breaking through the lubricating film drops with the smaller roller mass.

Requirements with respect to load carrying capacity and *self-alignment* are met by spherical roller bearings. The cross section height of the bearing is limited by the diameter of the roll journal and roll shell. The relatively wide spherical roller bearings FAG 231/560AK.MB.C4.T52BW are mounted.
The *nominal life* L_h = 83,000 h with given loads and percentages of time.

With a *lubricating oil* ISO VG 220 the viscosity ratio is κ = 0.71 under an operating temperature of 130 °C. An *attainable life* L_{hna} > 100,000 h is obtained with the *adjusted rating life calculation* (where $f_{s*} > 12$; a_{23II} = 1.2; V = 0.5; s = 1.6).

The increased *radial clearance* C4 is required due to the danger of detrimental radial preload in the bearing during the heating up phase when the temperature difference is great. With a *speed index* $n \cdot d_m$ = 224,000 $min^{-1} \cdot$ mm we recommend bearings with increased running accuracy according to specification T52BW.

Machining tolerances

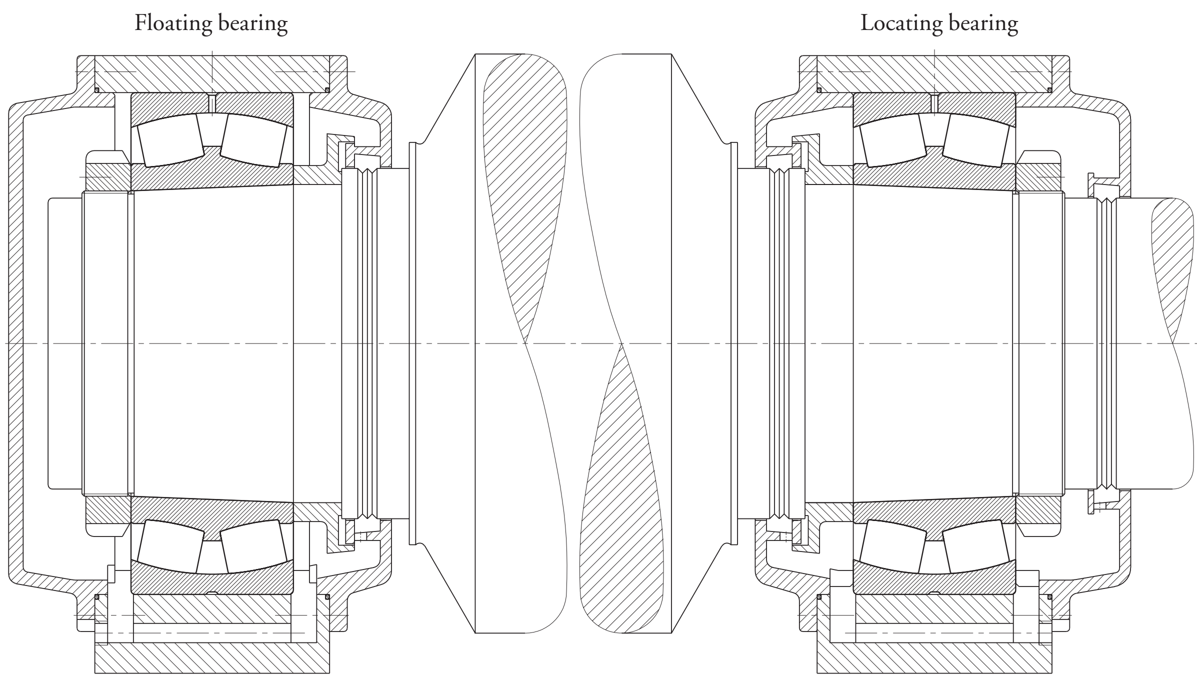
The inner rings have *circumferential load* and are directly fitted on the tapered roll journal. The roll journals have oil ducts so that the hydraulic method can be applied for mounting and dismounting the bearings. Roundness tolerance IT5/2 (DIN ISO 1101), taper angle tolerance AT7 (DIN 7178).
Bearing seats in the housing boring according to F7.

Lubrication

Oil circulation lubrication with a *synthetic oil* ISO VG 220, suitable in quality, which has stood dynamic testing on the FAG test rig FE8.
By supplying a large amount of *oil* to the centre of the bearing (minimum flow rate 12 l/min) heat dissipation is achieved as well as a low thermal stress of the oil. Any contaminants or wear particles are washed out of the bearing. Oil returns at both sides of the bearing via oil collecting pockets and connecting holes.

Sealing

Angle rings at the roll side prevent direct oil escape at the cover holes. Remaining oil is thrown off by splash grooves into collecting chambers and directed back. Cover *seals* make the housing oilproof.



70: Calender thermo roll bearings

71 Anti-deflection rolls

Anti-deflection rolls are found in both the press section and in calenders. They provide for an even paper thickness across the web and a consistently high paper quality. The drive is at the *locating bearing* end. Its power is transmitted via gearing and the hypoid teeth coupling to the roll shell.
The adjustment roll is pressed against the mating roll (calender roll) under very high pressure. As a result the mating roll is bent and the form of the roll shell changed. The shell of the adjustment roll must adjust to this form.
The anti-deflection roll consists of a stationary axle and a rotating roll shell. Control elements which can be pressure-balanced separately are provided on the axle. They support the roll shell hydrostatically and effect its adjustment. The roll shell is shaped like the bent mating roll by the changing pressure giving the paper an even thickness.

Operating data

Roll length 9,300 mm; roll diameter 1,025 mm; roll weight 610 kN; shell weight 210 kN; pressure 700 kN; circumferential velocity 1,500 m/min ($n = 470\ min^{-1}$); bearing temperature 55 °C.

Bearing selection, dimensioning

A *service life* of > 100,000 h is required. The bearing only has a guidance function when in operation (with pressure and closed gap).
Spherical roller bearings FAG 23096MB.T52BW (*dynamic load rating* C = 3,800 kN) are used.
Due to the danger of slippage bearings of the series 239 with a low *load rating* should be selected.
The bearings are produced with a reduced radial runout (specification T52BW), since running inaccuracy of the rotating roll shell influences the quality of the paper web.

Machine tolerances

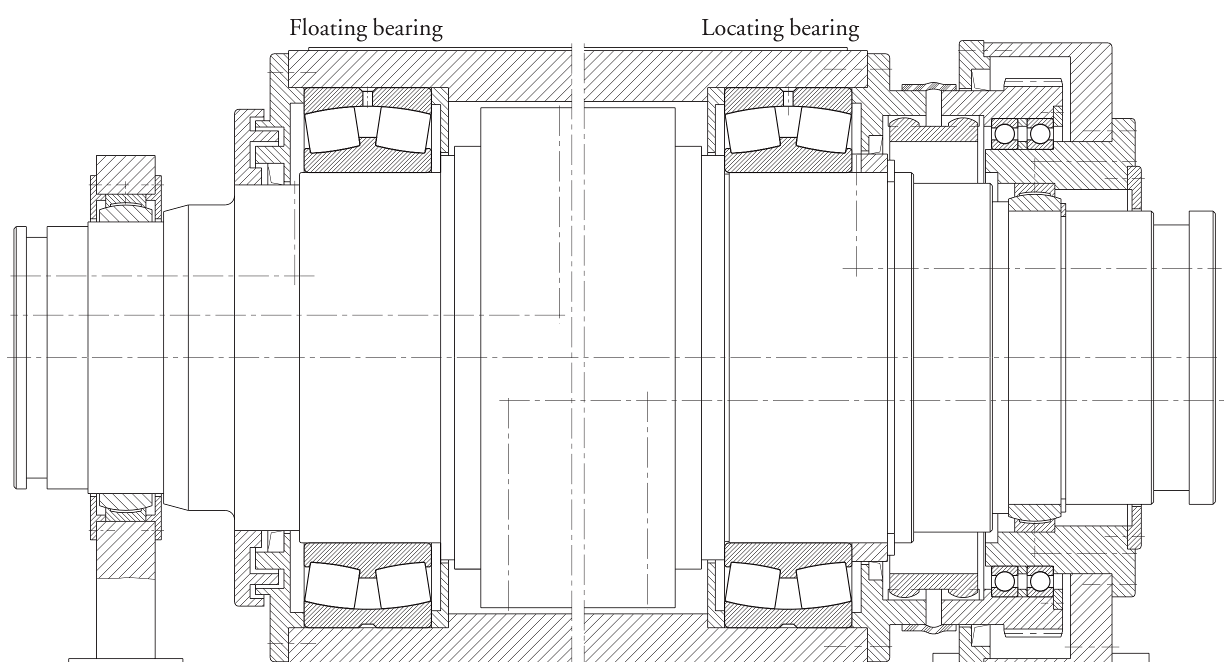
Bearing seats on the axle according to f6 due to *point load* for the inner rings.
The outer rings have *circumferential load* and a tight *fit;* the bearing seats in the housings are machined to P6.

Lubrication

When dynamic misalignment and/or slippage may occur, a very good lubrication system must always provide a load-carrying lubricating film. The bearings are supplied with the *lubricating oil* used for the hydraulic system (ISO VG 150 with *EP additives*). The oil is fed laterally to the bearings via holes. In new designs and particularly with heated rolls, the *lubricating oil* is fed via lubricating holes in the inner ring directly to the bearing contact areas.
The deep groove ball bearings of the transmission arranged at the *locating bearing* side are supplied with *oil* via a separate oil circuit.

Sealing

The bearings are *sealed* externally with a shaft seal. To the roll side a baffle plate provides for an *oil* reservoir in the bearing area.



71: Anti-deflection roll bearings

72 Spreader rolls

Paper webs transported in lengthwise direction tend to creasing. Spreader rolls stretch or expand in cross direction the webs running over them. They flatten creases and any middle or end parts of the web which are loose. Spreader rolls consist of a stationary axle which is bent symmetric to its longitudinal axis, and around which the roll shell rotates. Tube-shaped sections make up the roll shell and are arranged to rotate freely and have angular freedom. The sections adjust to one another in such a way that the bending form of the axle is reflected on the shell surface. Depending on the case of application – wet end section, dryer section, or subsequent processing – the sections are made of stainless steel or provided with a flexible coating (e.g. rubber).

Operating data

Roll length 8,300 mm, consisting of 22 sections; weight/section plus wire or paper web pull at 30° wrap angle 2 kN/m; a radial load of just 0.5 kN per bearing results therefrom.
Rotation of roll shell 1,160 min^{-1}.
Operating temperature in the wet end section 40 °C; in the dryer section and in subsequent processing with infrared drying temperatures can reach 120 °C.

Bearing selection, dimensioning

With rotating outer ring, extremely smooth running is required from the bearings since the sections in the wet end section and in the dryer section or subsequent processing are only driven by the wire tension and the paper web respectively.
High operational reliability is also necessary since the failure of one bearing alone means that the whole spreader roll has to be dismounted.

FAG 61936.C3 deep groove ball bearings are selected. The increased *radial clearance* C3 permits easy adjustment of the sections. With the low load, the bearings have a *nominal life* L_h of well over 100,000 hours.

Machining tolerances

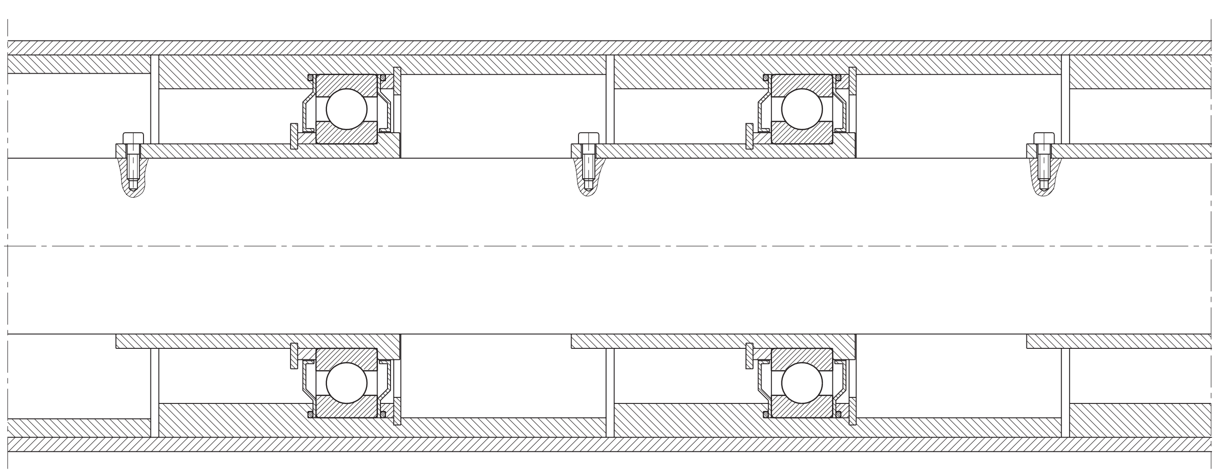
As the outer ring of the bearing rotates with the roll shell it is given a tight *fit* with M6 tolerance and is secured axially by a snap ring.
The inner ring has *point load* and is fitted to the shaft sleeve with h6. Due to the bent roll axle and for assembly reasons the sleeve is loosely fitted and axially attached with a screw.

Lubrication

The bearings are greased for life, i.e. no relubrication is provided for. The selection and filling quantity of *lubricating grease* is determined by the demand for smooth running as well as a *service life* of up to five years (8,000 operating hours per year). Low-friction *greases* (e.g. greases of class LG10 for the wet end section) are advantageous with high speeds and low loads.

Sealing

Non-rubbing dust shields are used for *sealing* due to the smooth running required. They are stuck to the bearing outer ring on both sides so the *base oil* centrifuged from the *lubricating grease* cannot escape. Round cord seals also provide for oil tightness.



72 Spreader roll bearings

73 Run wheel of a material ropeway

Operating data

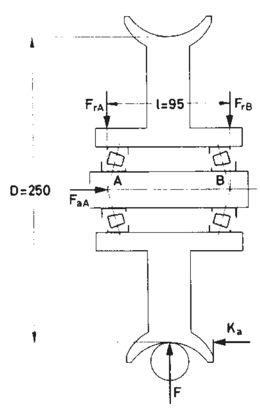
Speed $n = 270\ min^{-1}$; radial load $F_r = 8$ kN. Thrust loads as guidance loads only, considered by 20 % of the radial load: $K_a = 1.6$ kN.

Bearing selection

Each run wheel is supported by two tapered roller bearings FAG 30306A. The bearings are assembled in *O arrangement* which provides for a wider bearing *spread* than an *X arrangement.* The wider the spread, the lower the additional bearing load from thrust load K_a.

Bearing dimensioning

As thrust load K_a acts at the wheel circumference, it generates radial reaction forces at the bearing locations.



Bearing A:
$F_{rA} = F_r/2 + K_a \cdot (D/2)/l = 4 + 1.6 \cdot 125/95 = 6.1$ kN
The thrust load $K_a = 1.6$ kN acts toward bearing A.
Bearing B:
$F_{rB} = F_r/2 - K_a \cdot (D/2)/l = 4 - 1.6 \cdot 125/95 = 1.9$ kN

Radial loads acting on a shaft supported on two tapered roller bearings generate axial reaction loads which have to be considered in the calculation of the *equivalent dynamic load.* These internal loads together with the external thrust loads should, therefore, be taken into account for *life* calculation (see FAG catalogue WL 41 520, chapter "Tapered roller bearings").

Data for tapered roller bearings FAG 30306A (designation to DIN ISO 355: T2FB030):
dynamic load rating C = 60 kN,
Thrust factor $Y = Y_A = Y_B = 1.9$.

Thus,
$F_{rA}/Y = 6.1/1.9 = 3.2$; $F_{rB}/Y = 1.9/1.9 = 1$ and consequently $F_{rA}/Y > F_{rB}/Y$

The second condition proven is
$K_a > 0.5 \cdot (F_{rA}/Y - F_{rB}/Y) = 0.5\ (3.2 - 1) = 1.1$
For calculation of bearing A the following thrust load F_{aA} must, therefore, be taken into account:
$F_{aA} = K_a + 0.5 \cdot F_{rA}/Y = 1.6 + 0.5 \cdot 1.9/1.9 = 2.1$ kN

Consequently, the *equivalent dynamic load* P_A of bearing A is:

$P_A = 0.4 \cdot F_{rA} + Y\ F_a = 0.4 \cdot 6.1 + 1.9 \cdot 2.1 = 6.45$ kN
With this load, the indicated *dynamic load rating* and the *speed factor* $f_n = 0.534$ ($n = 270\ min^{-1}$) the i*ndex of dynamic stressing.*

$f_L = C/P_A \cdot f_n = 60/6.45 \cdot 0.534 = 4.97$

This value corresponds to a *nominal rating life* of more than 100,000 hours. Since this calculation is based on the most unfavourable load conditions, the thrust load acting constantly at its maximum and only in one direction, the bearing is adequately dimensioned with regard to *fatigue life.* The *service life* will probably be terminated by *wear,* especially under adverse operating conditions (high humidity, heavy contamination).
The load carrying capacity of bearing B does not need to be checked since its loading is much less than that of bearing A.

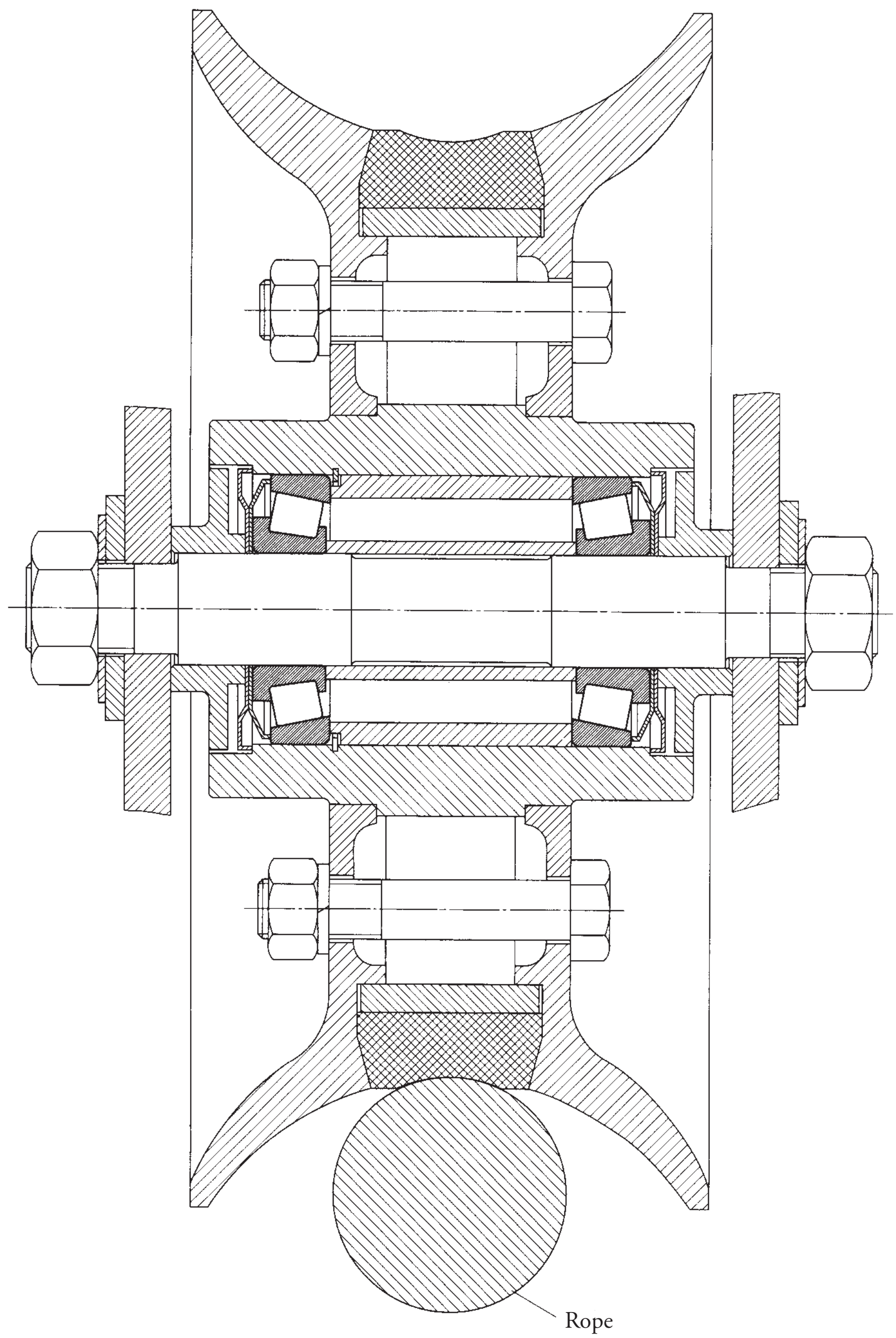
Machining tolerances

The run wheel mounting is a so-called hub mounting, i.e. the run wheel, with the two cups, rotates about a stationary shaft. The cups carry *circumferential load* and are thus tight-fitted. The shaft is machined to h6, the hub bore to M6.

Lubrication, sealing

The bearings and the free spaces have to be filled during mounting with *grease,* e. g. FAG rolling bearing *grease Arcanol* L186V. The grease filling will last for approximately one year.

In the example shown, the bearings are sealed by spring steel seals (Nilos rings).



73: Run wheel of a material ropeway

74 Rope return sheaves of a passenger ropeway

In this example of a passenger ropeway, eight sheaves are installed at the mountain station and another eight at the valley station including the sheaves in the valley station tensioning weight pit. The sheave diameters are 2.8 and 3.3 meters.

Bearing selection, dimensioning

The valley station sheaves and the tensioning weight sheaves are fitted with spherical roller bearings FAG 22234E. The sheaves at the mountain station are supported by spherical roller bearings FAG 22240B.MB.

The load on the bearings FAG 22234E installed in the tensioning weight sheaves is P = 65 kN each; with a *dynamic load rating* C = 1,100 kN and a *speed factor* f_n = 0.838, corresponding to a speed of 60 min^{-1}, the *index of dynamic stressing:*

$f_L = C/P \cdot f_n = 1{,}100/65 \cdot 0.838 = 14.2.$

This shows that the bearings are more than adequately dimensioned with regard to *fatigue life.*

The one-piece sleeve carrying the bearings allows convenient changing of the rope sheaves.

Machining tolerances

The outer rings carry *circumferential load* and require, therefore, a tight *fit.* To safeguard the spherical roller bearings against detrimental axial preloading, the design is of the *floating mounting* type. The outer rings are securely locked via the two covers by means of a spacer ring. The centre lip of sleeve H is slightly narrower than the spacer so that the sheave can float axially on the sleeve via the loosely fitted inner rings. The sleeve is locked to prevent it from rotating with the inner rings.

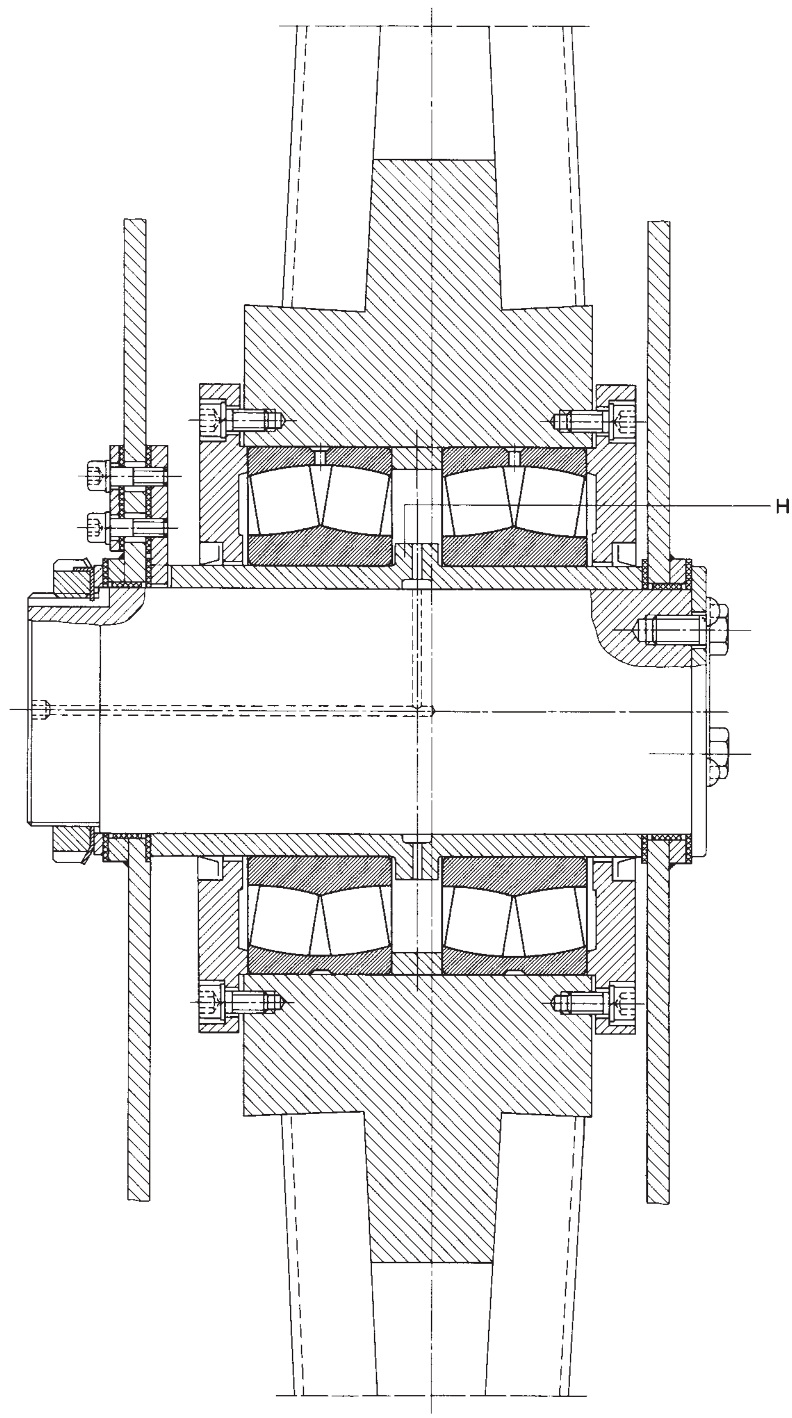
Sleeve to g6; hub bore to M6;

The sleeve has a sliding *fit* on the shaft.

Lubrication, sealing

Grease lubrication with FAG rolling bearing grease *Arcanol* L186V. Relubrication by means of lubricating holes in the shaft.

A shaft *seal* ring in the covers provides adequate protection against contamination.



74: Rope return sheaves of a passenger ropeway

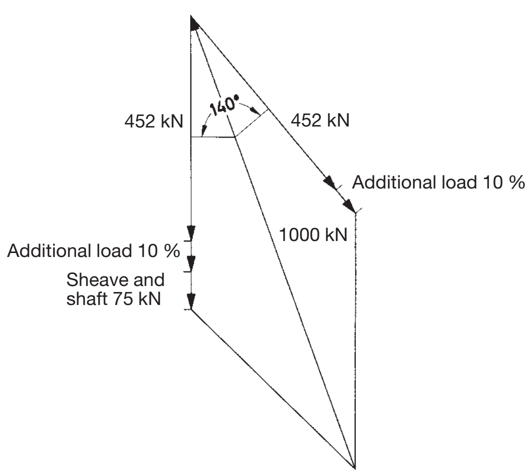
75 Rope sheave (underground mining)

These sheaves are arranged in the head frames of the pits. The rope fastened to the cage runs from the drive sheave or the drum of the hoist into the mine by passing over the rope sheaves.

Operating data

Static rope load 452 kN; weight of rope sheave and shaft 75 kN; rope sheave diameter d_S = 6.3 m; haulage speed v = 20 m/s; wrap angle 140°.

Acceleration forces are taken into account by assuming 10 % of the static rope load.



Bearing selection, dimensioning

From the parallelogram of forces the resultant load is approximately 1,000 kN. Since the two bearings are symmetrically arranged, the radial load per bearing is P = 500 kN.

Speed $n = v \cdot 60/(d_S \cdot \pi) = 20 \cdot 60/(6.3 \cdot 3.14) = 60\ \text{min}^{-1}$; this yields a *speed factor* f_n = 0.838.

The recommended *index of dynamic stressing* f_L is 4...4.5. With 4.5, the *nominal rating life* is about 75,000 hours. It should be borne in mind that only in rare cases the rope sheave bearings fail due to material fatigue; usually their *service life* is terminated by *wear.*

Thus, the required *dynamic load rating* C for the spherical roller bearing is calculated as follows:

$C = f_L/f_n \cdot P = 4.5/0.838 \cdot 500 = 2{,}680$ kN

Spherical roller bearings FAG 23252BK.MB with a *dynamic load rating* C = 2,900 kN were chosen.

The bearings feature a high load carrying capacity and compensate for potential housing misalignments, shaft deflections and deformations of the head frame.

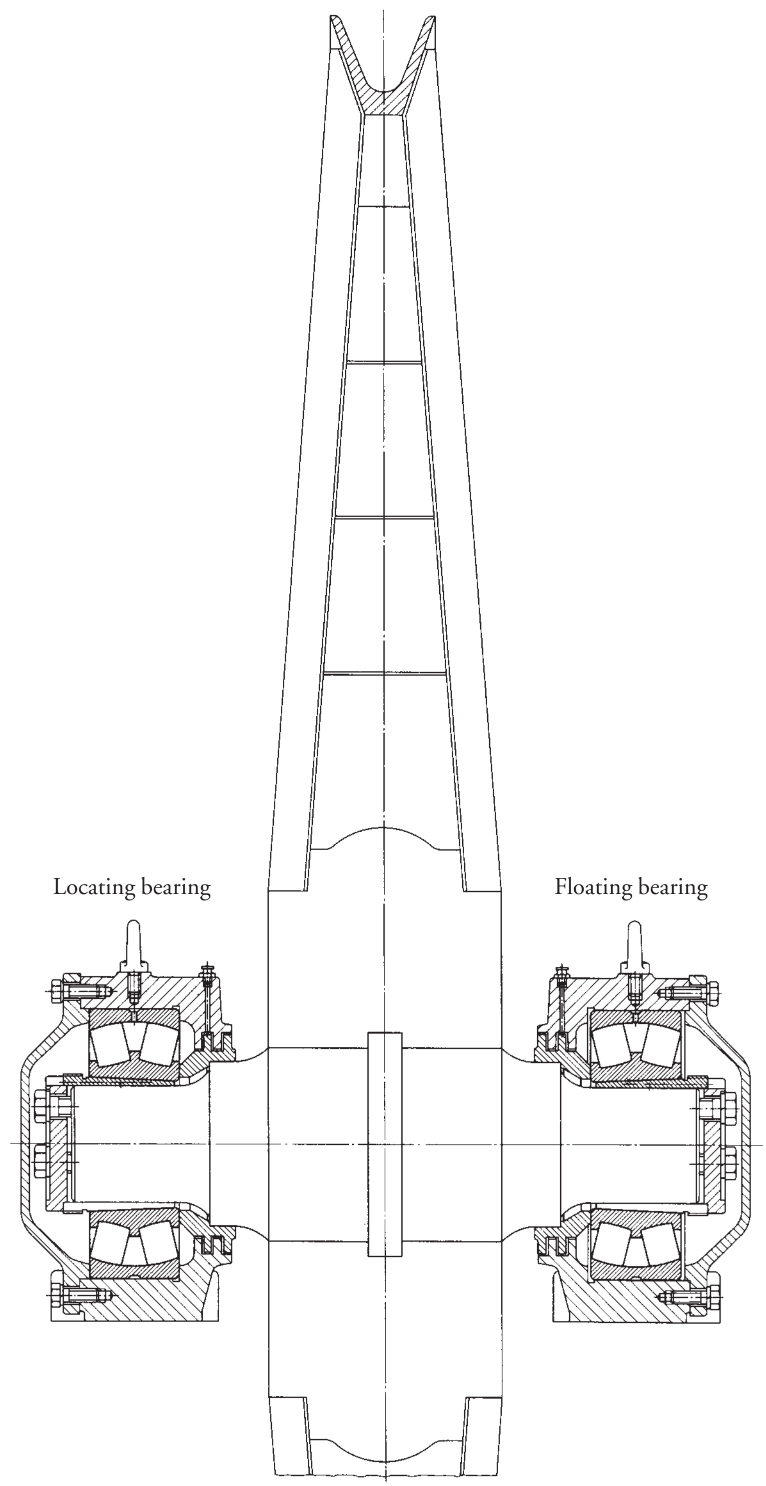
Machining tolerances

One bearing acts as the *locating bearing,* the other one as the *floating bearing.* Both bearings have a tapered bore (K 1:12). They are mounted on the shaft journal with withdrawal sleeves (FAG AH2352H). Mounting and dismounting is simplified by using the hydraulic method. For this purpose the withdrawal sleeves feature oil grooves and ducts. The spherical roller bearings are supported by FAG plummer block housings FS3252AHF and FS3252AHL.

Shaft journal to h6, cylindricity tolerance IT5/2 (DIN ISO 1101).
Housing to H7.

Lubrication, sealing

Grease lubrication with FAG rolling bearing grease *Arcanol* L186V.
A multiple labyrinth protects the bearings against contamination. Replenishment of labyrinth *grease* is effected about every 4...6 weeks.



75: Rope sheave (underground mining)

76 Rope sheave of a pulley block

In pulley blocks it is customary to arrange several sheaves on a common shaft. To achieve minimum overall pulley block width, the sheaves and their bearings should, therefore, be as compact as possible.

Bearing selection

For the rope sheaves of pulley blocks the wrap angle is 180°. Thus the radial load on the bearing is twice the rope pull. Thrust loads, resulting from a possible inclined rope pull, and the moments caused by them are low and can be neglected for *bearing life* calculation. Adequate bearing *spread* for load accommodation is achieved by mounting either two bearings or one double-row bearing. Deep groove ball bearings are satisfactory for accommodating the loads in this application.
The bearings are mounted on a sleeve, forming a ready-to-mount unit with the sheave which can be easily replaced.

Operating data and bearing dimensioning

Rope pull S	40 kN
Bearing load	
$F = 2 \cdot S$	80 kN
Speed n	30 min^{-1}
Speed factor f_n	1.04
Bearings mounted	2 deep groove ball bearings FAG 6220
Dynamic load rating	$C = 2 \times 122$ kN
Equivalent dynamic load	$P = F/2 = 40$ kN
Index of dynamic stressing	$f_L = C/P \cdot f_n$ $= 122/40 \cdot 1.04 = 3.17$
Nominal rating life	$L_h = 16{,}000$ h

Usually, an *index of dynamic stressing* $f_L = 2.5...3.5$ is used for rope sheaves. This corresponds to a *nominal rating life* of 8,000 to 20,000 hours.
Thus the bearings are adequately dimensioned compared with established field applications.

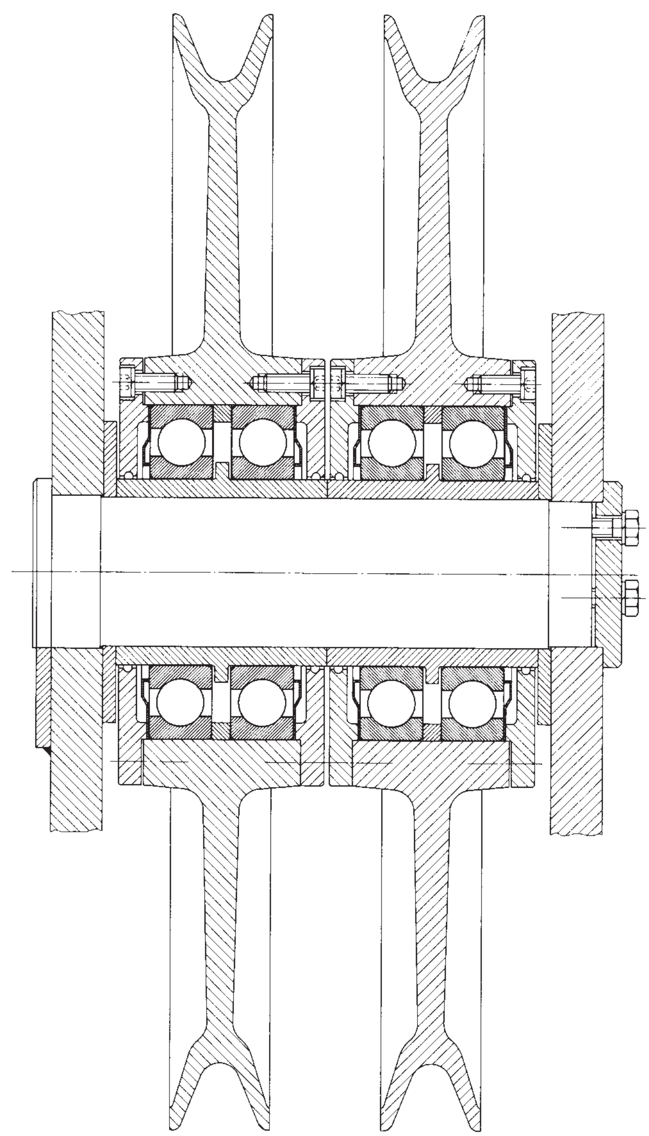
Machining tolerances

The mounting is a so-called hub mounting, i.e. the pulley, with the outer rings, rotates about a stationary shaft. The outer rings carry *circumferential load* and are press-fitted: hub to M7.
The inner rings carry *point load* allowing a loose *fit* or sliding fit: shaft sleeve to g6 or h6.

Lubrication, sealing

The sheave bearings are lubricated with lithium soap base *grease* of *penetration* class 3 (*Arcanol* L71V). High loads (load ratio P/C > 0.15) require a lithium soap base *grease* of *penetration* class 2 and *EP-additives* (*Arcanol* L186V). One *grease* filling normally lasts for several years.

The rope sheave in this example is sealed by spring steel *seals* (Nilos rings).



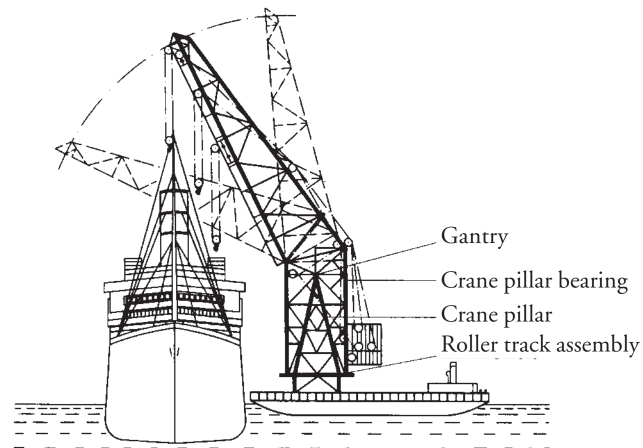
76: Rope pulleys with deep groove ball bearings

77–78 Gantry of a floating crane

Floating cranes are used in harbours for transportation of heavy and bulky goods, in shipyards for repair work and for ship outfitting. Due to their mobility they are an ideal complement to stationary cranes.

The pillar of the crane described is attached to the ship. The slewing gantry with the crane superstructure is supported on the crane pillar. The bearing mounting has to take up the weight of the superstructure and the payload. Since the common centre of gravity of payload and gantry is outside the pillar axis, a tilting moment is produced causing horizontal reaction forces in the bearings at the upper and lower pillar end.

At the upper pillar end the gantry runs on the so-called pillar bearing mounting. It consists either of one single spherical roller thrust bearing or one spherical roller bearing combined with one spherical roller thrust bearing, depending on the amount of radial loading.



At the pillar foot the gantry is supported on a roller-track assembly (see example no. 79).

77 Crane pillar mounting with a spherical roller thrust bearing

Operating data

Thrust load (crane superstructure and payload) F_a = 6,200 kN; radial load (reaction forces resulting from tilting moment and wind pressure) F_r = 2,800 kN; speed n = 1 min^{-1}.

Bearing selection, dimensioning

The thrust load, consisting of the weight of the slewing superstructure and the payload, is much higher than the radial load resulting from the tilting moment and wind pressure. Therefore, the crane pillar bearing must have a high thrust load carrying capacity. Moreover, the bearing must be *self-aligning* to compensate for misalignment and elastic deformation unavoidable on these crane structures. Due to the low speed of 1 min^{-1} the bearing is chosen with regard to its static load carrying capacity.
A spherical roller thrust bearing FAG 294/630E.MB with a *static load rating* of C_0 = 58,500 kN; factor X_0 = 2.7 is selected.

For spherical roller thrust bearings under *combined load* the ratio F_r/F_a must be small in order to ensure that most of the rollers transmit loads. Condition: $F_r/F_a \leq 0.55$.

In this example

$F_r/F_a = 2{,}800/6{,}200 = 0.45$

Thus the *equivalent static load*

$$P_0 = F_a + X_0 \cdot F_r = F_a + 2.7 \cdot F_r = 6{,}200 + 2.7 \cdot 2{,}800 = 13{,}800 \text{ kN}$$

The *index of static stressing*

$f_s = C_0/P_0 = 58{,}500/13{,}800 = 4.24$

Thus, the requirement $f_s \geq 4$ for spherical roller thrust bearings (FAG catalogue WL 41 520) whose housing and shaft washers – as in this example – are fully supported is met.

With f_s values $\geq 4 ... \leq 6$ the shaft washer and the housing washer must be fully supported axially, and good radial support of the housing washer must also be provided.

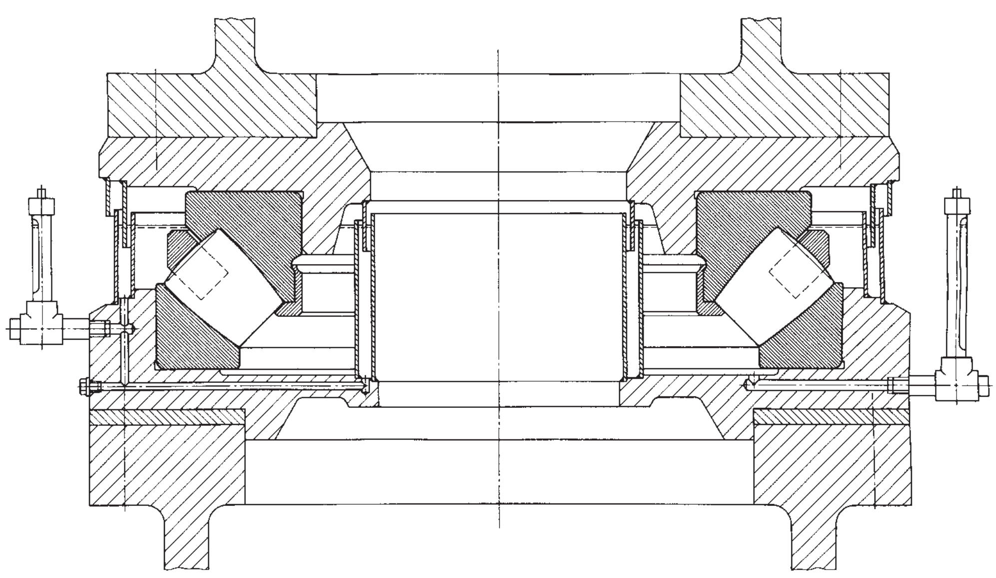
Machining tolerances

Shaft washer to j6; housing washer to K7

Lubrication, sealing

Oil bath lubrication, with the rollers fully immersed in *oil*. The *oil* level should be maintained to the upper edge of the shaft washer and is controlled by means of an *oil* level indicator.

Due to adverse ambient conditions existing for floating crane applications, high-efficiency *seals* must be provided (*oil*-filled labyrinths). The inner and the outer labyrinth are interconnectd by oil holes. The *oil* level in the labyrinths is also checked with an *oil* level indicator.



77: Crane pillar mounting with a spherical roller thrust bearing

78 Crane pillar mounting with a spherical roller thrust bearing and a spherical roller bearing

Operating data

Thrust load (crane superstructure and payload) F_a = 1,700 kN; radial load (reaction forces resulting from tilting moment and wind pressure) F_r = 1,070 kN; speed n = 1 min^{-1}.

Bearing selection, dimensioning

In this case $F_r/F_a > 0.55$. The radial load is relatively high. Therefore, it is accommodated by an additional radial bearing, a spherical roller bearing. The two bearings are mounted so that their pivoting centres coincide. Thus angular alignability is ensured. A thrust washer inserted between the two bearings prevents excessive radial loading on the *thrust bearing*. The spherical roller bearing size depends on that of the spherical roller thrust bearing. The outside diameter of the *radial bearing* must be larger than the housing washer of the *thrust bearing*. To ensure close guidance of the crane superstructure, the reduced *radial clearance* C2 is provided for the *radial bearing*.
Crane pillar mountings with one spherical roller bearing and one spherical roller thrust bearing provide compact designs. They require, however, a wider mounting space than mountings with one single spherical roller thrust bearing.
The mounting features a spherical roller thrust bearing FAG 29440E with the *static load rating* C_0 = 8,500 kN and a spherical roller bearing FAG 23056B.MB.C2 with the *static load rating* C_0 = 3,000 kN.
For calculating the *equivalent static load* for the spherical roller thrust bearing it is assumed that the friction at the thrust washer, acting as a radial load, is 150 kN. Thus $F_r/F_a < 0.55$ for the spherical roller thrust bearing.

Equivalent static load:

$P_0 = F_a + X_0 \cdot F_r = F_a + 2.7 \cdot F_r$ for $F_r \leq 0.55\ F_a$
$= 1{,}700 + 2.7 \cdot 150 = 2{,}100$ kN

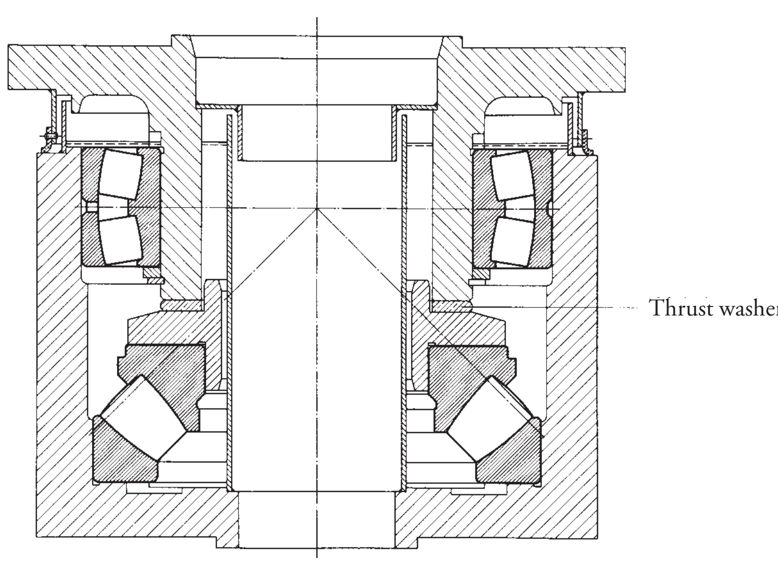
For the spherical roller bearing:
$P_0 = F_r = 1{,}070$ kN
Hence the *indices of static stressing* $f_s = C_0 / P_0$ are:
Spherical roller thrust bearing = 8,500 / 2,100 = 4.05
Spherical roller bearing = 3,000 / 1,070 = 2.8
These values show that the bearings are safely dimensioned.
The shaft washer and housing washer of spherical roller thrust bearings with f_s values of ≥ 4...≤ 6 must be fully supported axially; good radial support of the housing washer is also required.

Machining tolerances

Spherical roller thrust bearing:
Shaft washer to j6,
housing washer to K7
Spherical roller bearing:
shaft to j6; housing to J7

Lubrication, sealing

The bearing housing is filled with *oil* beyond the upper edge of the spherical roller bearing, i.e. the bearings run in an oil bath. Thus they are well protected against condensation water and corrosion.
Outer *sealing* is provided by labyrinths. In view of the adverse ambient conditions an additional, rubbing *seal* with elastic lip is provided. Inner sealing is effected by the tube communicating with the housing, and a labyrinth.



78: Crane pillar mounting with a spherical roller thrust bearing and a spherical roller bearing

79 Roller track assembly

The radial bearing mounting at the pillar foot consists normally of several rollers travelling on a circular track. Each of these rollers is supported by two bearings, the upper bearing being the *locating bearing,* the lower one the *floating bearing.*

Operating data

The maximum load on one roller is 2,200 kN. Thus, each bearing is loaded with P_0 = 1,100 kN.

Bearing selection, dimensioning

The rollers transmit only the horizontal loads resulting from the tilting moment. To cater for the misalignment conditions inherent in structural steelwork and for wheel axle deflection, *self-aligning bearings* have to be provided.

Spherical roller bearings FAG 23230ES.TVPB with *static load rating* C_0 = 1,630 kN are mounted.
With an *equivalent static load* P_0 = 1,100 kN an *index of static stressing*

$f_s = C_0/P_0$ = 1,630 / 1,100 = 1.48

is calculated.

This value meets the requirements for smooth running of the bearing.

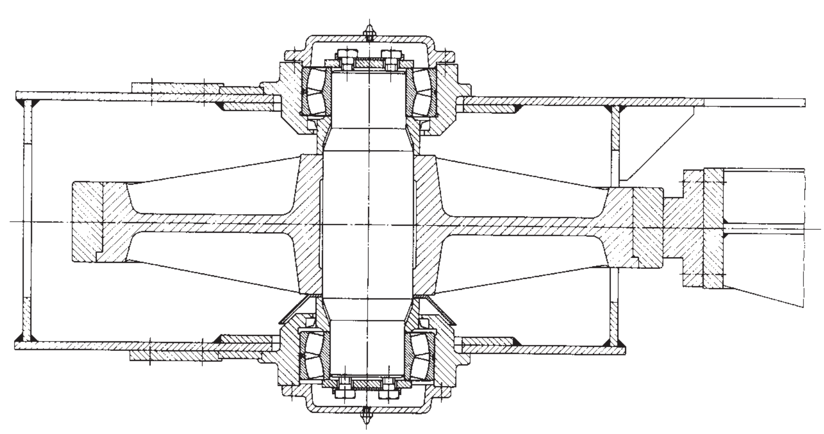
Machining tolerances

The inner rings carry *circumferential load* and are fitted tightly.
Shaft to k6; housing to H7.

Lubrication, sealing

The bearings and housing cavities are packed to capacity with a lithium soap base *grease* with *EP additives* (FAG rolling bearing grease *Arcanol* L186V). Relubrication is possible through lubricating nipples in the housing cover.

Outer *sealing* is provided by the housing cover, inner sealing by a shaft seal ring. A flinger ring between roller and lower bearing additionally protects the lower shaft seal ring against dirt and rubbed-off particles.



79: Roller-track assembly

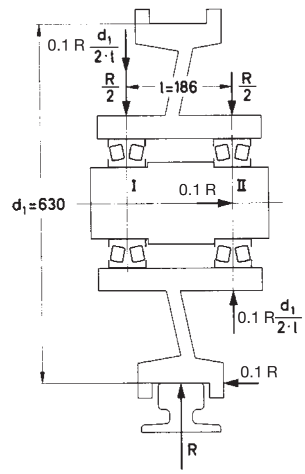
Crane run wheels

Bearings in crane run wheels have to accommodate the heavy loads resulting from the deadweight of the crane and the payload, and axial and radial reaction loads resulting from the axial guiding loads between wheel flange and rail.

80 Crane run wheel

Operating data

Wheel load R = 180 kN; operating speed n = 50 min^{-1}; wheel diameter d_1 = 630 mm; bearing centres l = 186 mm.



Bearing selection

The bearings fitted in run wheels are often designed as hub mountings. The run wheel rotates, together with the bearing outer rings, about a stationary shaft. Spherical roller bearings are used because of their very high load carrying capacity.

The bearings fitted are two spherical roller bearings FAG 22220E. The distance between the two bearings should not be too small in order to keep the bearing reaction loads resulting from the wheel-rail contact within reasonable limits.
This bearing arrangement is standardized by DIN 15 071. The two spherical roller bearings run on a sleeve to allow for rapid replacement of the complete run wheel unit. It is a *floating bearing arrangement,* the inner rings being displaceable on the sleeve. Depending on the thrust load direction, either the left-hand or the right-hand bearing abuts the sleeve collar. This arrangement allows optimum bearing loading, since the bearing which accommodates the additional thrust loads is relieved of radial load due to the tilting moment from the thrust load.

Bearing dimensioning

The weight of the crane and the maximum payload are known. The thrust acting between wheel and rail can, however, only be estimated. The *equivalent dynamic load* P acting on the bearings is calculated in accordance with DIN 15 071; this standard specifies the thrust resulting from friction between wheel and rail to be 10 % of the radial load. The bearing loads P_I (bearing I) and P_{II} (bearing II) are:

$P_I = X \cdot [R/2 + 0.1 \cdot R \cdot d_1 / (2 \cdot l)]$

$P_{II} = X \cdot [R/2 - 0.1 \cdot R \cdot d_1 / (2 \cdot l)] + Y \cdot 0.1 \cdot R$

With the radial factor X = 1 and e = 0.24 for $F_a/F_r \leq e$ the thrust factor Y = 2.84.

Thus $P_I = 90 + 18 \cdot 630/372 = 120.5$ kN $= P_{max}$

$P_{II} = 90 - 30.5 + 2.84 \cdot 18 = 110.6$ kN $= P_{min}$

Assuming that the bearing loads vary linearly between P_{min} and P_{max},

$P = (P_{min} + 2 \cdot P_{max})/3 = (110.6 + 241)/3 = 117.2$ kN

With the *dynamic load rating* C = 360 kN and the *speed factor* f_n = 0.885 (n = 50 min^{-1}) the *index of dynamic stressing*

$f_L = C/P \cdot f_n = 360/117.2 \cdot 0.885 = 2.72$

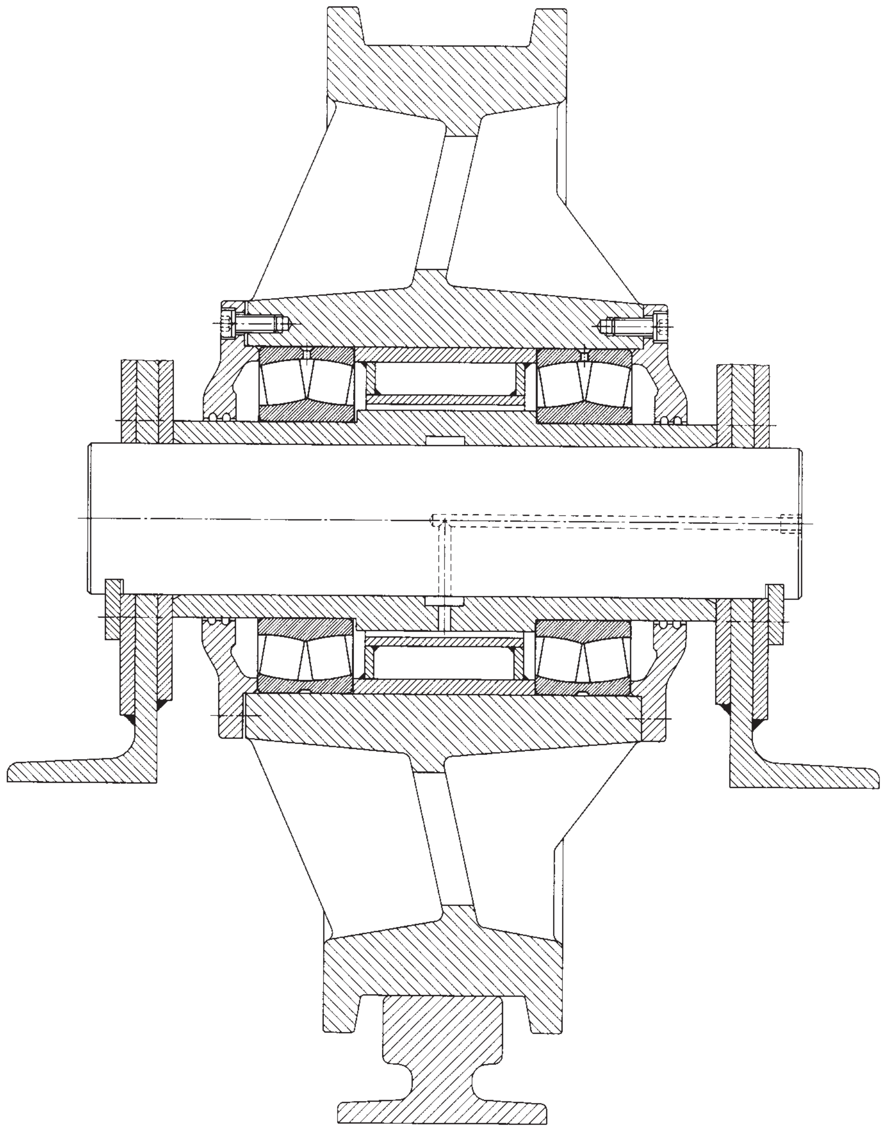
With the generally recommended value for crane run wheels f_L = 2.5...3.5, the bearing mounting is adequately dimensioned.

Machining tolerances

The bearing outer rings, which carry *circumferential load,* are tight *fits.* The hub is machined to M7, the sleeve to g6, thus providing for a slide *fit* for the inner rings. This prevents detrimental axial preloading and simplifies bearing mounting and dismounting.

Lubrication, sealing

The bearings are lubricated with a lithium soap base *grease* with *EP additives* (FAG rolling bearing grease *Arcanol* L186V). The *relubrication interval* is approximately one year.
Gap-type *seals* or simple rubbing *seals* are in most cases satisfactory.



80: Crane run wheel with spherical roller bearings

81 Crane hook

The load suspended from a crane hook often has to be swivelled before being lowered. Therefore, the hooks of heavy-duty cranes are designed for these swivelling motions.

Bearing selection, dimensioning

Since the weight of the payload acts vertically downward, the load is pure thrust. Therefore, loose radial guidance of the shaft in the crosshead is satisfactory.

The load carrying capacity of the bearing is based on its *static load rating*. A thrust ball bearing FAG 51152FP with a *static load rating* C_0 = 1,020 kN is mounted. Based on the maximum hook load of 1,000 kN plus a safety margin of 10 %, the *index of static stressing* $f_s = C_0/P_0 = 1{,}020 / 1{,}100 = 0.93$; i. e., permanent deformation occurs at maximum load. However, it is so small that it does not interfere with the swivelling of the load.

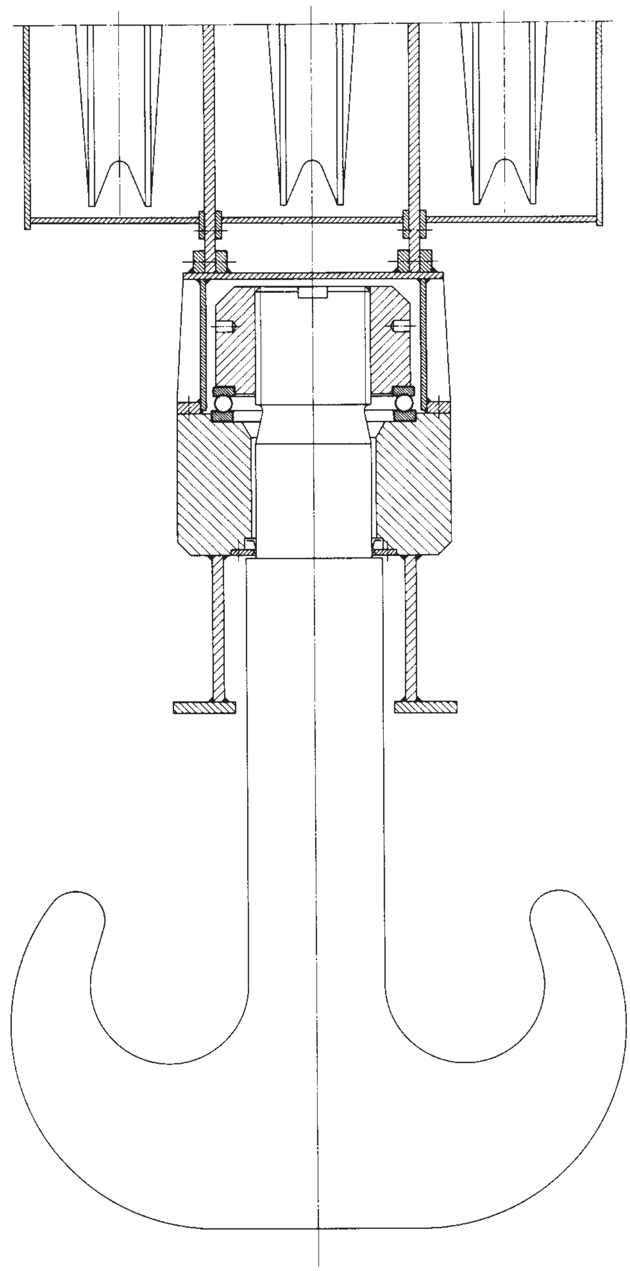
The bearing is *adjusted* against the collar at the hook shaft by means of a locknut. This prevents the shaft washer from separating when the crane hook is set on the ground.

Machining tolerances

The bearing seats are machined to j6 (washer) and to H7 (housing).

Lubrication, sealing

The bearing assembly is packed to capacity with lithium soap base *grease* with *EP additives* (FAG rolling bearing grease *Arcanol* L186V). Maintenance of the bearing is not required. Above the crane hook nut a sheet steel cap is provided which protects the bearing against contamination.



81: Crane hook mounting

82 Mast guidance bearings of a fork lift truck

The fork lift carriage must run smoothly in order to handle the live loads efficiently. This requirement is satisfied by mast guide rollers and chain return sheaves.
Mast guide rollers (HMFR) and chain sheaves (KR) of modern fork lift trucks are largely fitted with double-row angular contact ball bearings.

Bearing selection, bearing design

Mast guide rollers
FAG HMFR30x75x20.75 are preferably used for fork carrier and lifting frame. They can accommodate radial loads, thrust loads and the moments resulting from these. The mast guide rollers feature thick-walled outer rings and can, therefore, accommodate even high, shock-type loads.
The profile and dimensions of the outer ring are largely dictated by the standardized U-beam dimensions.

Chain sheaves
Chain sheaves FAG KR30x75x28/27 are attached to the hydraulically actuated upper section of the mast and serve to deflect the pull chain.

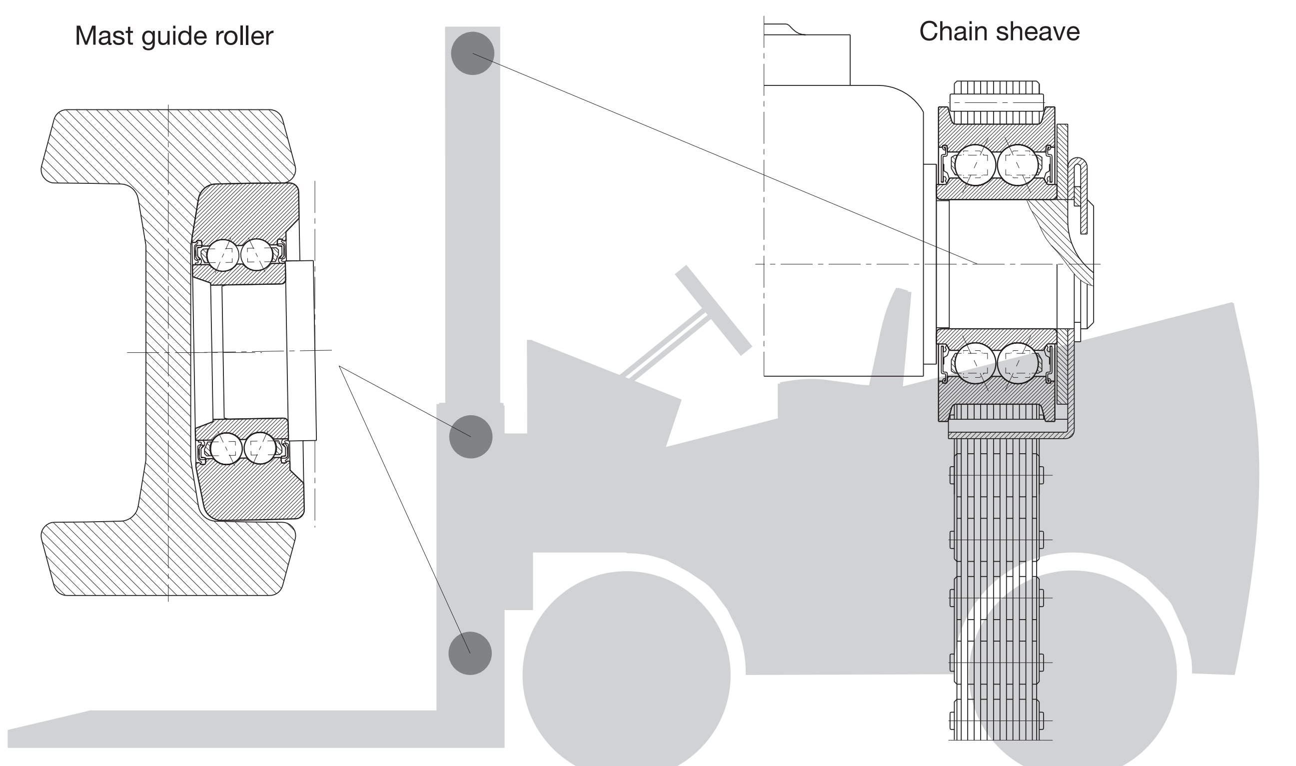
Due to their relatively thick-walled outer ring, the bearings can accommodate high radial loads made up of the deadweight of the fork lift carriage, including fork and live load. The outer ring profile is dictated by the pull chain used; lateral guidance is provided by the two lips. The distance between the two ball rows, together with the *contact angle,* provides for a wide *spread* so that the return sheaves can also accommodate tilting forces and axial guiding forces.
Roller mounting is simple; they are simply placed on the pin; axial preloading by a screw is not required. Chain return sheaves are axially locked.

Machining tolerances

The inner rings of der mast guide rollers and return sheaves carry *point load,* thus a loose *fit* is satisfactory. The pin is machined to j6.

Lubrication, sealing

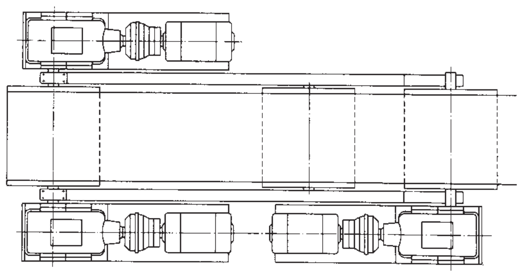
The bearings are lubricated for *life* with a lithium soap base *grease* (*EP additives*).
Sealing is provided by single- or double-lip RSR *seals.*



82: Mast guide roller and chain return sheave for a fork lift truck

83 Head pulley of a belt conveyor

One head pulley is not sufficient for very long belts, steeply inclined belts or heavily loaded belts. In such cases several head pulleys are mounted in tandem. In this application, two head pulleys are arranged at the drive station. Three identical driving motors are used: the first pulley is driven from both ends, the second one from one end only.



Operating data

Power consumption 3 x 430 kW; belt width 2,300 mm; belt speed 5.2 m/s; conveying capacity 7,500 m^3/h; pulley diameter 1,730 mm.

Bearing selection, dimensioning

The shaft of the head pulley is supported on plummer blocks. The shaft diameter is dictated by strength considerations, thus determining the bearing bore and housing size. Spherical roller bearings FAG 23264K.MB are mounted. The one-piece plummer block housings FAG BND3264K are made of cast steel GS-45. One of the plummer blocks acts as the *locating bearing,* the other one as the *floating bearing.* To simplify mounting and dismounting hydraulic sleeves are used.

With an *index of dynamic stressing* $f_L \approx 4$ the bearings are adequately dimensioned compared to field-proven bearing arrangements. Often the *bearing life* is limited by wear on *rolling elements* and raceways and is generally shorter than the *nominal rating life* (approx. 50,000 h), calculated with the *index of dynamic stressing* f_L. Improved cleanliness during mounting and operation, and a suitable lubricant, reduce *wear,* thus increasing the *bearing life.* These influences are taken into account in the *adjusted rating life calculation* by the *factor* a_{23}.

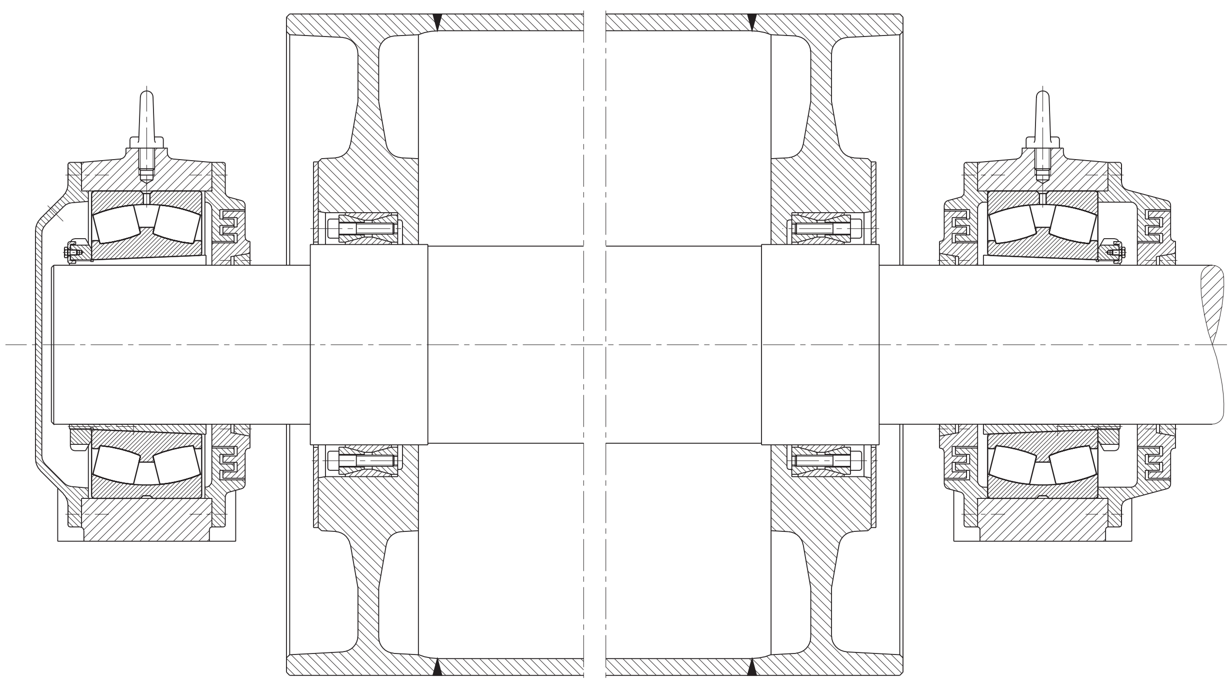
Machining tolerances

The bearing inner rings carry *circumferential load.* They are fitted on the shaft with adapter sleeves FAG H3264HG.
Shaft to h8 and cylindricity tolerance (DIN ISO 1101) IT5/2; housing bore to H7.

Lubrication, sealing

Grease lubrication with a lithium soap base grease of *penetration* class 2 with *EP additives* (FAG rolling bearing grease *Arcanol* L135V or L186V).

The housing covers and rings on the shaft form nonrubbing labyrinth *seals.* These multiple labyrinths are filled with the same *grease* as the bearings and prevent penetration of foreign matter. In very dusty environments relubrication at short intervals is required. *Grease* is injected into the bearing until some of the spent grease escapes from the labyrinths.



83: Head pulley bearing arrangement of a belt conveyor

84 Internal bearings for the tension/ take-up pulley of a belt conveyor

Non-driven pulleys in belt conveyors are frequently fitted with internal bearings. The bearings are integrated into the pulley so that the pulley body revolves about the stationary shaft.

Operating data

Belt width 3,000 mm; belt speed 6 m/s; pulley diameter 1,000 mm; pulley load 1,650 kN.

Bearing selection, dimensioning

These pulleys are supported either in two spherical roller bearings (fig. a) or in two cylindrical roller bearings (fig. b). The internal design of the cylindrical roller bearings allows the *rolling elements* to accommodate load-related shaft deflections without edge running.

In a spherical roller bearing arrangement, an FAG 23276BK.MB with an adapter sleeve FAG H3276HGJ is used as *locating bearing* and an FAG 23276B.MB is used as *floating bearing.*

In a cylindrical roller bearing arrangement, the *floating bearing* is an FAG 547400A and the *locating bearing* an FAG 544975A. Both cylindrical roller bearings have the main dimensions 360 x 680 x 240 mm and are interchangeable with spherical roller bearings FAG 23276BK.MB with an adapter sleeve FAG H3276HGJ.

The bearings must feature the required *dynamic load rating* C/the required bore diameter. With an *index of dynamic stressing* $f_L > 4$, the bearings are sufficiently dimensioned with regard to *fatigue life.*

Often, the actual *bearing life* is considerably shorter than the *nominal rating life* determined on the basis of the f_L value. The cause is *wear* in raceways and on *rolling elements* as a result of adverse ambient conditions. Improved cleanliness during mounting and in operation as well as the utilization of a suitable lubricant have a positive effect on the *bearing life.* These influences are taken into account in the a*djusted rating life calculation* and in the *modified life calculation* in accordance with DIN ISO 281. It is used for example to compare the effects of different lubricants. The *fatigue life* calculated for pulley bearings with this method in most cases is not equivalent to the *attainable life* as the *service life* is mainly limited by *wear.*

Machining tolerances

In view of the *circumferential load* and the relatively high amount of load the outer rings must be a very tight *fit* in the pulley bore. Tolerances, see table below.

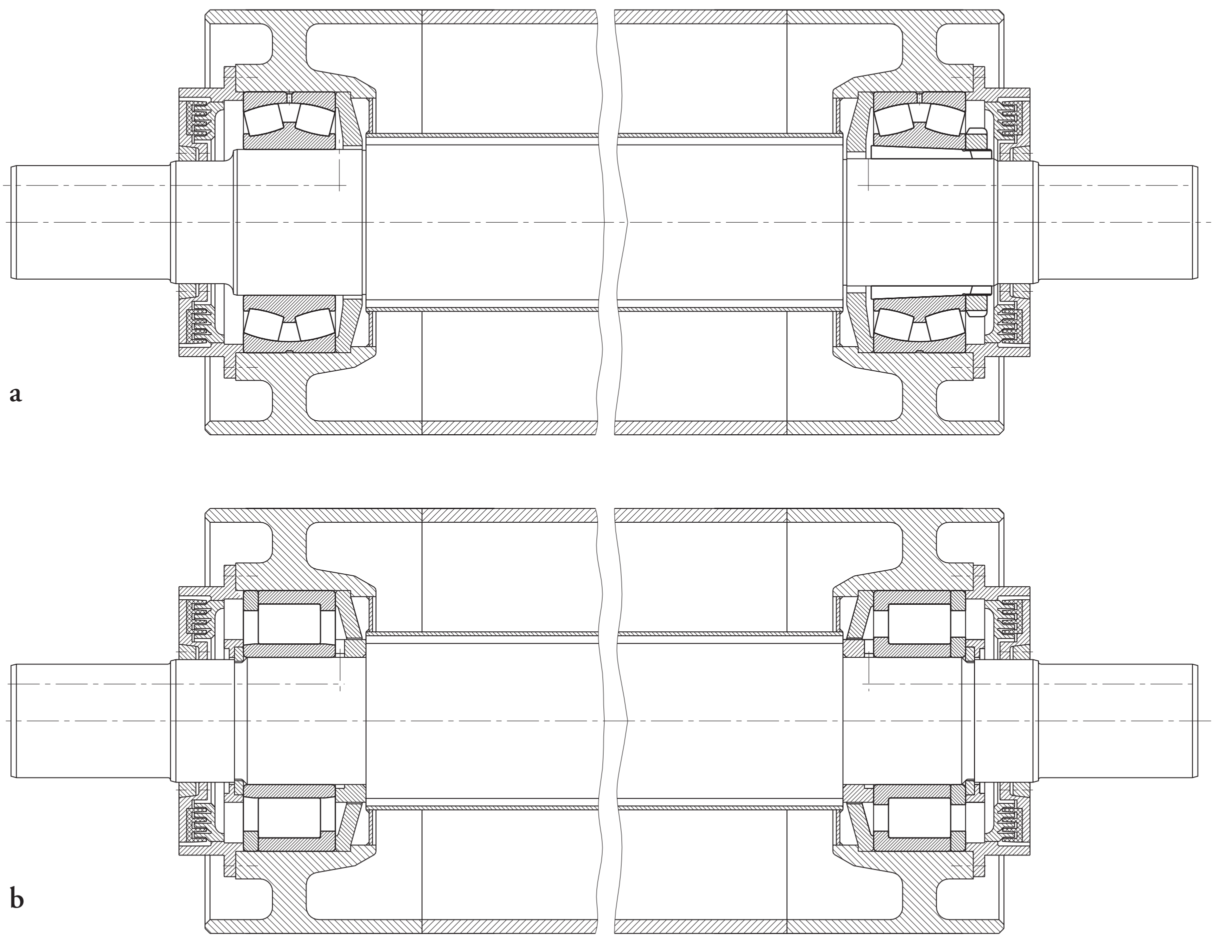
Lubrication, sealing

The bearings are lubricated with a lithium soap base *grease* of *penetration* 2 with *EP additives* (FAG rolling bearing grease *Arcanol* L186V).

External *sealing* of the bearings is provided by non-rubbing labyrinth *seals* or multi-collar rubbing seals. In both cases the labyrinths are filled with the same *grease* as the bearings. To supply the bearings with fresh grease and to increase the sealing effect, relubrication is effected at short intervals (depending on the amount of dirt) via the stationary shaft.

Machining tolerances

Bearing	Seat	Diameter tolerance	Cylindricity tolerance
Spherical roller bearing as *locating bearing*	Shaft Pulley bore	h8 M7	IT5/2 IT5/2
Spherical roller bearing as *floating bearing*	Shaft Pulley bore	g6 M7	IT5/2 IT5/2
Cylindrical roller bearing *locating bearing, floating bearing*	Shaft Pulley bore	g6 N7	IT5/2 IT5/2



a

b

84: Internal bearings for the tension / take-up pulley of a belt conveyor

Belt conveyor idlers

Many industries use belt conveyors for transporting bulk materials. The conveyors run on idlers and may extend over many miles; thus the number of idlers needed may be very large. Consequently, bearing mounting design is dictated by cost-saving considerations.

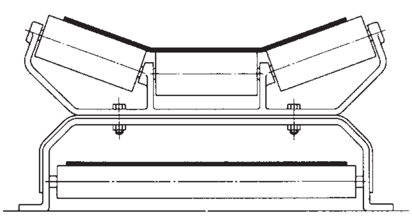
Idler arrangement

Small belt conveyor systems feature idlers rigidly linked to a frame. Large belt conveyor systems feature idler garlands linked to each other by flexible joints.

85 Rigid idlers

Operating data

Capacity I_m = 2,500 t/h; Design: troughed belt with three idlers per station; the two outer idlers are arranged at an angle of 30° to the horizontal; distance between two idler stations l_R = 1,200 mm; idler diameter d = 108 mm, belt weight G_G = 35 kg/m, deadweight per roller G_R = 6 kg; belt speed v = 3 m/s; acceleration due to gravity g = 9.81 m/s^2.



Bearing selection

Idler mountings are usually internal bearing arrangements (hub mountings), i.e. the idler rotates about a stationary shaft.
Since a belt conveying plant requires a large number of roller bearings, deep groove ball bearings, which are produced in large quantities at low cost, are preferably used. This allows a simple and economical idler design.

Bearing dimensioning

Idler speed $n = \dfrac{v \cdot 60 \cdot 1{,}000}{d \cdot \pi} = 530\ \text{min}^{-1}$

For ball bearings, the *speed factor* f_n = 0.4.

Load per idler station:

$$F = g \cdot l_R \cdot \left(\frac{I_m}{3.6 \cdot v} + G_G \right) =$$
$$= 9{,}81 \cdot 1{,}2 \cdot \left(\frac{2{,}500}{3.6 \cdot 3} + 35 \right) = 3{,}137\ \text{N}$$

For a trough angle of 30° the horizontal centre idler takes up approximately 65 % of this load. Thus the load on the centre idler is

$$F_R = 0.65 \cdot F + g \cdot G_R = 0.65 \cdot 3{,}137 + 9.81 \cdot 6 =$$
$$= 2{,}100\ \text{N} = 2.1\ \text{kN}$$

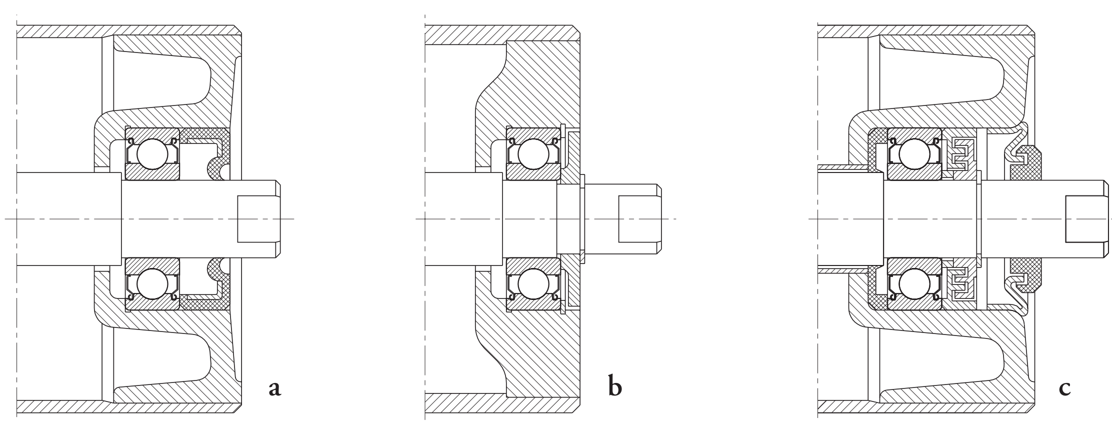
Equivalent dynamic bearing *load:*

$P = F_r = F_R/2 = 1.05$ kN

The usual *index of dynamic stressing* for idler bearings f_L = 2.5...3.5. With f_L = 3, the required *dynamic load rating* C of a bearing

$C = f_L \cdot P/f_n = 3 \cdot 1.05/0.4 = 7.88$ kN

Deep groove ball bearings FAG 6204.2ZR.C3 having a *dynamic load rating* C = 12.7 kN are mounted.



85a...c: Idler *sealing* variations

Generally, the *service life* of a bearing is not terminated by fatigue but by *wear* in raceways and on *rolling elements* as a result of contamination. Increased cleanliness during mounting and efficient *sealings* increase the *bearing life*. The *ajdusted rating life calculation* is used for comparing different *seal* designs.
New idler bearings feature utmost cleanliness (V = 0.3). However, in the course of operation the lubricant gets heavily contaminated by particles (V = 3).
As the bearings in belt conveyor systems fail as a result of *wear,* the values obtained by the *adjusted rating life calculation* (L_{hna}) usually are not equivalent to the actually attainable lives.

Machining tolerances

The two deep groove ball bearings are mounted onto the idler shaft in a *floating bearing arrangement.* As the inner rings are subjected to *point load* the shaft is machined to h6 or js6. The outer rings are subjected to *circumferential load* and are pressed, therefore, into the idler end with an M7 interference *fit.*

Lubrication, sealing and maintenance

The deep groove ball bearings FAG 6209.2ZR.C3 are packed, at the manufacturing plant, with a lithium soap base *grease* of *penetration* class 2 which is sufficient for the entire *bearing service life.* Such a *grease* is also used for the *sealing.*
With idler bearings, both the *attainable life* and the lubricant *service life* may be considerably reduced by *grease* contamination during operation so that the *sealing* selected is decisive. Figs. 85a...c show various types of *sealing* for belt conveyor idlers.
Simple *seals* (figs. 85a and b) are used for clean environments. Fig. 85c shows an idler *seal* for brown coal open pit mining.

In addition to the rigidly troughed belt conveyors the garland type belt conveyors are being increasingly used. The idlers of each station are linked to each other by flexible joints. These joints may consist of a wire rope, a chain link (flat chain, round chain), hinge or similar.
Idler garlands accommodate impacts elastically; in the event of problems with a roller the individual garland is lowered and can be replaced relatively easily if necessary.

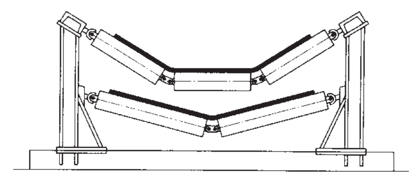


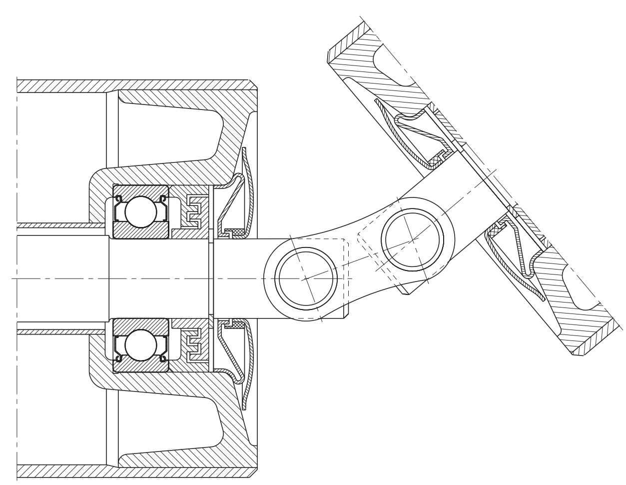
Fig. 86 shows idler garlands connected by chain links. These idlers are part of a conveying installation for rock phosphate. The bearings fitted are deep groove ball bearings FAG 6303.2ZR.C3.

Machining tolerances

Idler ends to M7, shaft to h6 or js6.

Lubrication, sealing, maintenance

The deep groove ball bearings (design .2ZR) are *sealed* by dust shields on both sides and filled with FAG rolling bearing *grease,* a lithium soap base grease of *penetration* class 2. The grease filling suffices for idler *service life.* A grease chamber with a non-rubbing labyrinth seal is provided at the outboard end. The second, adjacent chamber is closed by a shield pressed into the hub bore. A baffle plate protects the bearing against coarse particles.



86: Idlers connected by chain link

87 Bucket wheel shaft of a bucket wheel excavator

Bucket wheel excavators are mainly used for brown coal open pit mining. The bucket wheel shaft carries the bucket wheel, the bull gear and the transmission housing. It is supported in the boom ends.

Operating data

Input power 3 x 735 kW; theoretical conveying capacity 130,000 m^3 / day; bucket wheel speed 3 min^{-1}.

Bearing selection

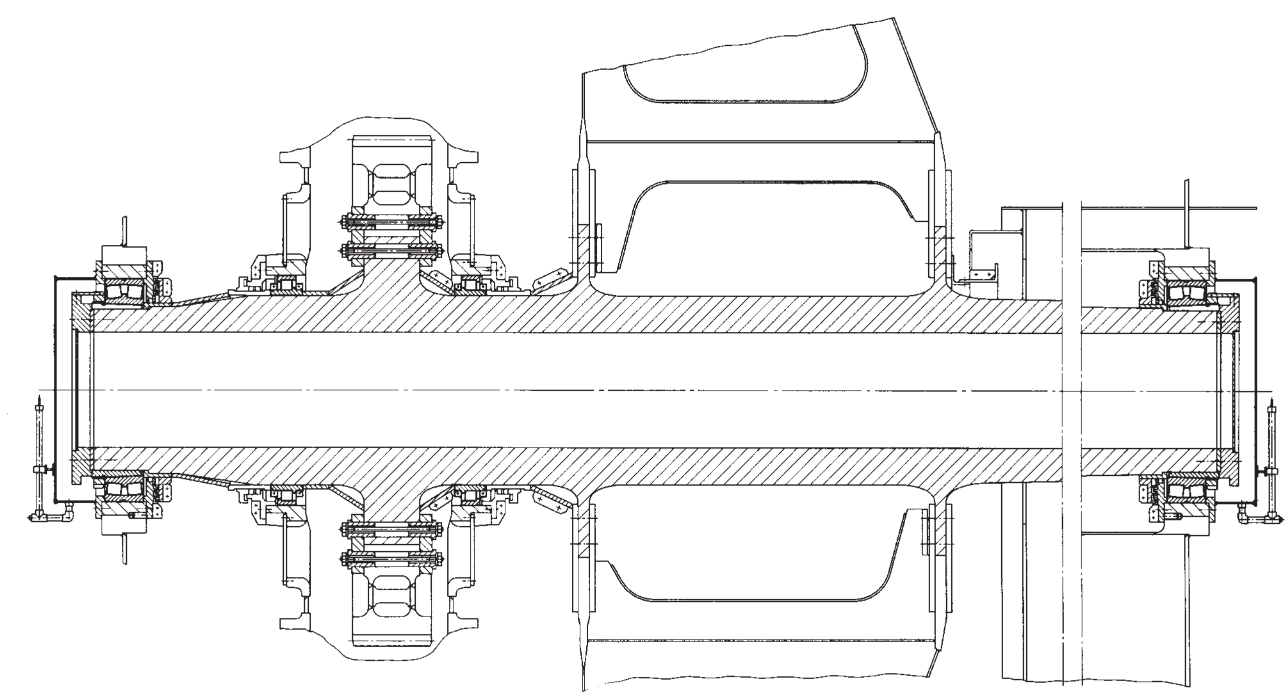
The bearings of the bucket wheel shaft are subjected to high shock-type loads. Moreover, shaft deflections and misalignments must be expected. For this reason, only *self-aligning* roller bearings are suitable for supporting the shaft. At both shaft ends, spherical roller bearings FAG 239/900K.MB with withdrawal sleeves FAG AH39/900H are mounted as *locating bearings.* Thermal length variations of the shaft are compensated for by the elastic surrounding structure. The *radial clearance* of the spherical roller bearings is eliminated during mounting by pressing in the withdrawal sleeves. Only a split bearing can be provided on the bucket wheel side of the transmission box due to the solid forged shaft flange to which the bull gear is attached. If an unsplit bearing were to be provided on the opposite side of the transmission box it could only be replaced after dismounting the spherical roller bearing first. For this purpose the entire bucket wheel shaft would have to be removed from the boom. This is avoided by using a split FAG cylindrical roller bearing of dimensions 1,000 x 1,220 x 170/100 mm on this side as well. The increased *axial clearance* of the two cylindrical roller bearings yields a *floating bearing arrangement.* Each bearing accommodates axial guiding loads in only one direction. The inner ring halves are attached to the shaft by means of separate locking rings. The calculated *nominal rating life* of all bearings is over 75,000 hours.

Machining tolerances

All inner rings are subjected to *circumferential load.* The spherical roller bearings FAG 239/900K.MB are hydraulically fastened to the shaft (machined to h8) by means of withdrawal sleeves FAG AH39/900H. The split cylindrical roller bearings sit directly on the shaft which is machined to m6 in this place. All outer ring seats are toleranced to H7.

Lubrication, sealing

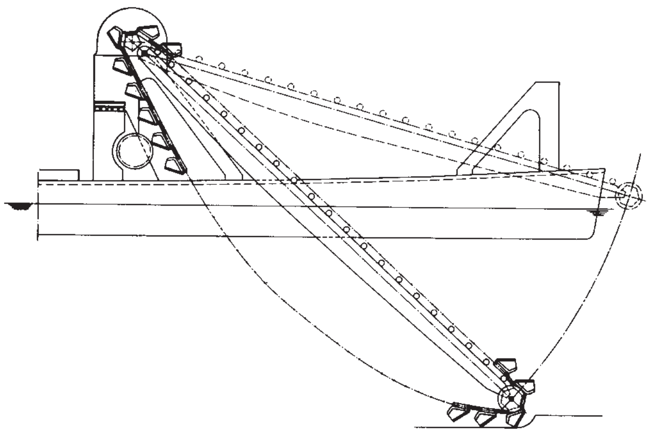
The spherical roller bearings are *oil*-bath lubricated. The split cylindrical roller bearings are supplied by the draining *oil* from gearwheel lubrication.
The *sealing* is a combination of labyrinth and rubbing *seal.* The labyrinths at the spherical roller bearings can be relubricated.



87: Bucket wheel mounting

88 Bottom sprocket of a bucket chain dredger

Bucket chain dredgers perform dredging work in waterways. The buckets are carried by a continuous chain from the bottom sprocket to the top sprocket over a large number of support rolls and back.



Operating data

Ladder length 32 m; number of buckets 44; maximum dredging depth approximately 14 m; radial load on bottom sprocket approximately 250 kN.

Bearing selection

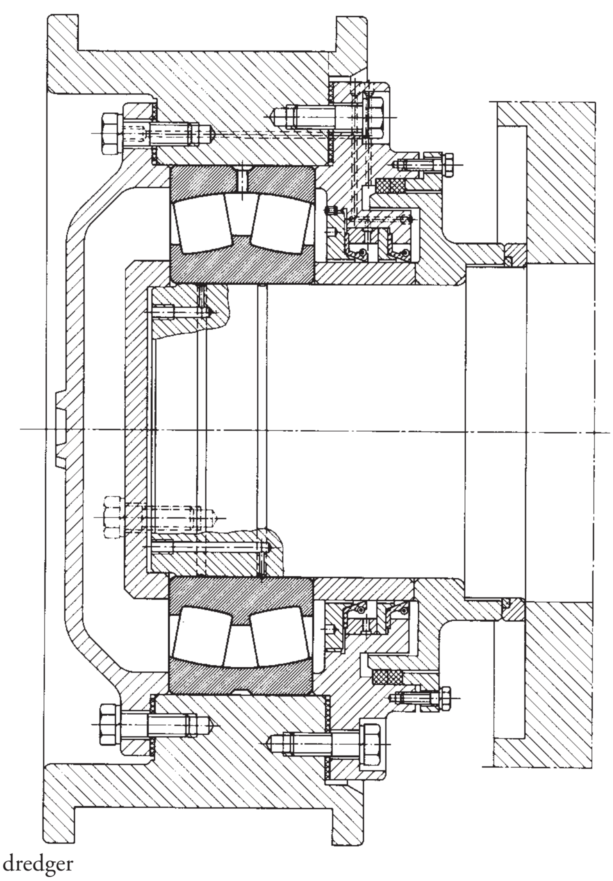
Rugged operation and unvoidable misalignment between the housings at both ends of the sprocket shaft call for *self-aligning bearings.* The bearings used are spherical roller bearings FAG 22240B.MB. Both bottom sprocket shaft bearings are designed as *locating bearings.* However, the bearings are not nipped axially, the housing being mounted with clearance in its ladder yoke seat. For easier bearing dismounting the shaft journal is provided with oilways and grooves for hydraulic dismounting.

Machining tolerances

Circumferential load on the inner ring.
Shaft journal to m6; housing to J7.

Lubrication, sealing

The *grease* in the bearing (FAG rolling bearing *grease Arcanol* L186V) is renewed at intervals of 1 1/2 to 2 years coinciding with the general overhaul period of the dredger.
The bottom sprocket is constantly immersed in water. This requires waterproof *sealing.* Each bearing location is, therefore, fitted with two rubbing *seals* (shaft seals with bronze garter spring) and, in addition, with two packing rings (stuffing box). The shaft seals run on a bush of seawater-resistant material. The stuffing box can be retightened by means of a cover. *Grease* is regularly pumped into the labyrinth between the shaft seals and packing rings.



88: Bottom sprocket of a bucket chain dredger

89 Drive unit of a finished goods elevator

Finished-goods elevators are used, for example, for charging salt granulating plants. The material is conveyed in buckets attached to a chain. The chain is driven by the tumbler situated at the upper end.

Operating data

Input power 22 kW; speed 13.2 min^{-1}; radial bearing load 90 kN.

Bearing selection

As shaft deflections and misalignments have to be expected the drive shaft is supported on *self-aligning bearings.* Selecting split spherical roller bearings FAG 222SM125T ensures that the heavy drive unit with the torque arm does not have to be dismounted in the event of repair.
As a result, the downtimes of the plant and the cost of production loss are considerably lower than they would be with one-piece bearings. To limit the variety of bearings used, a split spherical roller bearing was provided at the free shaft end as well.
Split spherical roller bearings have a cylindrical bore. Inner ring, outer ring and *cage* with roller set are split into halves.

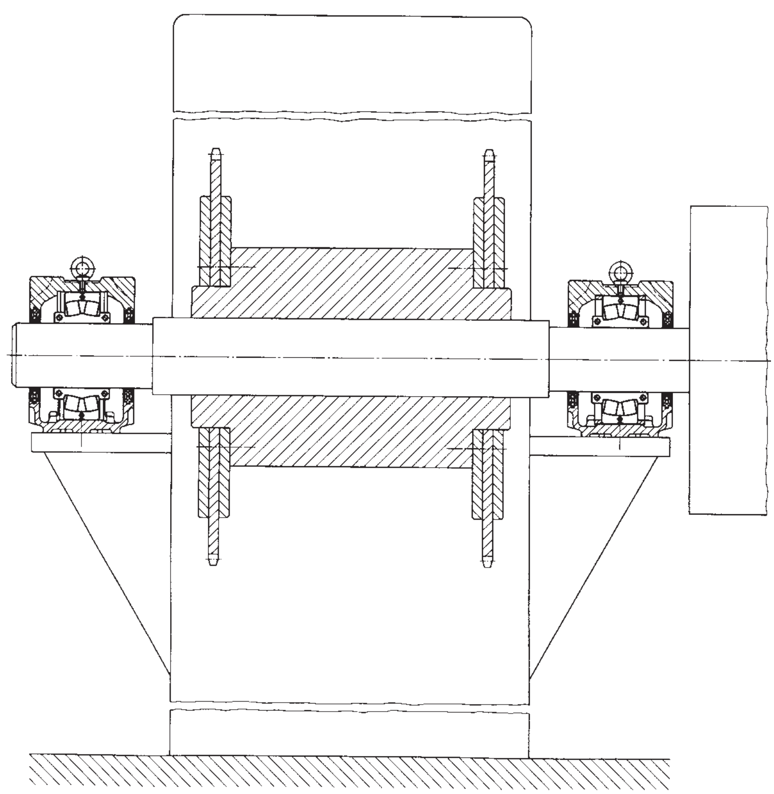
The split inner ring halves are braced together by means of four dowel screws and attached to the shaft. Both outer ring halves are fitted together without a gap by means of two dowel screws.
The drive-end bearing is mounted with two locating rings and acts as the *locating bearing;* the bearing at the opposite end is the *floating bearing.* Split spherical roller bearings FAG 222SM125T are designed in such a way that they can be mounted into split series housings FAG SNV250 instead of one-piece spherical roller bearings with an adapter sleeve. Outside diameter, outer ring width and shaft seat diameter are identical.
The theoretical *fatigue life* L_h of the bearings is over 100,000 hours.

Machining tolerances

Shaft to h6...h9;
housing to H7

Lubrication, sealing

The bearings are lubricated with *grease.* The housings are connected to a central lubricating system so that continuous relubrication is ensured.
The shaft openings on both sides of the housing are each sealed by a two-lip *seal.*



89: Drive unit of a finished goods elevator